Modelling of material properties for MEMS structures

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The paper presents a combined experimental and theoretical study to determine the actual dynamic Young's modulus ($E_{dynamic}$) for the mechanical structure of a silicon microaccelerometer. The resonance (natural) frequencies of the suspension arms of accelerometer were measured using MSA-500 Micro System Analyzer and, then, introduced in analytical expressions in order to find $E_{dynamic}$. The obtained results were used to validate the model of numerical simulation relative to the measured resonance frequencies of analyzed MEMS accelerometer.

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1. Introduction

The mechanical properties of the materials used in microelectromechanical system (MEMS) fabrication are of great importance for designing the microdevices. These mechanical properties are strongly dependent on size, manufacturing processes of MEMS structures and crystal orientation in materials, due to anisotropy – preferred directions in the lattice of single crystalline materials, such as silicon wafers – the structural material chosen for the application under study.

Among the mechanical properties of interest one can be mentioned:

-elastic properties (Young's modulus and Poisson's ratio), directly related to the device performance (mechanical behaviour);

-internal stress, causing the deformation of the microstructure;

-strength, determining how much force can be applied to a MEMS device (fracture strength, flexural strength, yield strength, tensile strength, compressive strength);

-fatigue/aging, when MEMS devices are exposed to cyclic or constant stress for a long time during operation, generally accelerated life tests to check the reliable work of the microdevices.

Several testing methods using various test samples were developed, e.g. [1-3]. They are based on the idea that it is essential to measure the mechanical properties of MEMS materials at the same size scale and to employ the same processes with those used in MEMS devices.

The established specimens for determination of Young's modulus (E) are structures such as cantilevers or fully clamped bridges, which can be fabricated using a variety of surface- or bulk- micromachining techniques from any material, with sufficient stiffness to permit a free-standing structure. Extraction of E by using these specimens types is possible from dynamic characteristics such as resonance frequencies.

The resonance frequencies can be computed analytically for simple geometries like straight beams. In other cases, for multilayer structures or more complex shapes, the resonance frequencies can be computed using finite element method (FEM) – a particularly interesting approach for predicting at the same time the sensitivity and detection limits of these versatile microdevices, e.g. [4].

In a previous work, [5], we carried out a comparison between the experimental values and those computed with FEM of the resonance frequencies for the mechanical structure of a bulk-micromachined silicon accelerometer. The structure of this sensor consists of an inertial mass, symmetrically situated, and supported by four microbeams, as can be seen in Fig. 1. The sensitive elements (piezoresistors) are diffused into the suspension arms, nearly the rim and the inertial mass, on perpendicular directions.

The difference enough high (about 18% for the first vibration mode) between the experimental and numerically computed values of the resonance frequencies were explained by various behaviours of the four microbeams because of the technological execution errors and the material anisotropy. Values of the elastic constants found in the scientific literature, neglecting the orthotropic behaviour of silicon, were used for the model of numerical simulation.

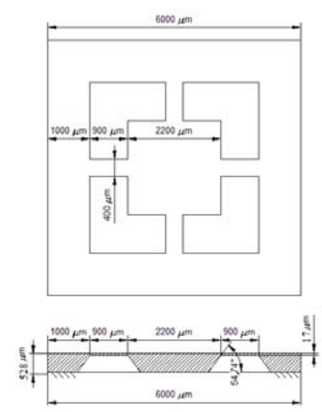


Fig. 1. Sizes of the ideal structure of microaccelerometer.

In order to validate this isotropic model of computation, in this paper we present a combined experimental and theoretical study for determining the actual dynamic Young's modulus ($E_{dynamic}$). The vibration tests were performed on microcantilevers detached from the analyzed accelerometer structure by using the MSA-500 Micro System Analyzer, a measurement instrument for analysis and visualization of structural vibrations and surface topography in microstructures such as MEMS devices. The computation of $E_{dynamic}$ has been made using analytical expressions according to beam and plate models. After resuming the FEM modal analysis with $E_{dynamic}$ on the isotropic model of mechanical structure of the accelerometer, a good agreement was found between the computed and experimental values of the resonance (natural) frequencies.

2. Theoretical considerations

The suspension arms of inertial mass of the microaccelerometer were modelled as cantilever beams and cantilever rectangular plates with expressions of resonance frequencies, given in (1) and (2), respectively [6]:

$$f_{1,2,3} = \frac{(n_{1,2,3})^2}{2\pi l^2}\frac{h\sqrt{3}}{6}\sqrt{\frac{E}{\rho}} \tag{1}$$

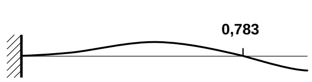
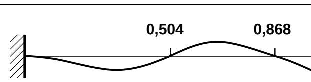
where: f – resonance (natural) frequency;

l, h – length and thickness of the beam, respectively;

E, ρ – Young's modulus and density of the beam material, respectively;

$n_{1,2,3}$: 1.875, 4.694, and 7.855 corresponding to the first, second and third vibration mode (bending), respectively (Table 1).

Table 1. Shape of vibration modes for homogenous and constant section cantilever beams.

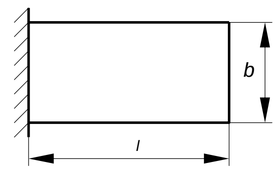
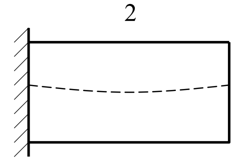
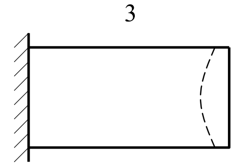
Supporting type	Vibration mode	Mode shape and nodes (indicated by fractions of length from the left end)
Clamping-free end	1	
	2	
	3	

$$f_{1,2,3} = \frac{\alpha_1}{2\pi}\sqrt{\frac{D}{\rho h l^4}} \tag{2}$$

$$D = \frac{Eh^3}{12(1-\nu^2)} \tag{3}$$

where: f – resonance (natural) frequency;
l, h – length and thickness of the plate, respectively;
E, ν, ρ – Young's modulus, Poisson's ratio and density of the plate material, respectively;
$\alpha_{1,2,3}$ – coefficients computed depending on length (l)/width (b) ratio (Table 2), for the first, second and third vibration mode (bending, torsion, and bending), respectively.

Table 2. Vibration modes and nodal lines for cantilever rectangular plates.

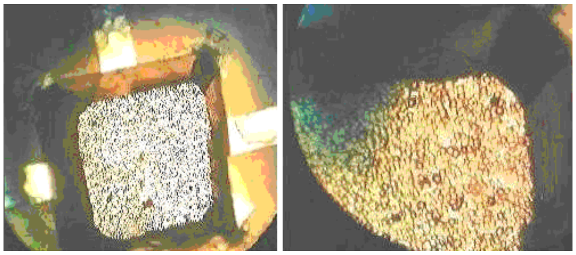
Vibration mode \ l/b	2	3
1 	3.472	3.450
2 	14.93	34.73
3 	21.61	21.52

3. Experimental procedure

The mechanical structure of accelerometer was released by a wet anisotropic etching process on both sides of the silicon substrate. We used (100) silicon wafers and did patterning so that the edges of the suspension arms were in the <110> directions. As etch and boron diffusion (for piezoresistors) mask, a thermally grown oxide was used. Its patterning was performed by a standard photolithographic process.

The anisotropic etchant (KOH) exposed the {111} planes.

The patterning mask was provided with compensation structures to avoid convex corner undercut of the inertial mass. In absence of an etch stop layer, etching time was estimated to create the required microbeam thickness and inertial mass height. The executed structure of microaccelerometer is shown in Fig. 2.



(a) *(b)*

Fig. 2. Photos of the microaccelerometer final structure (a) with a detailed corner of the inertial mass (b).

For the vibration tests, three cantilevers together with the rim were detached from the structure, Fig. 3.

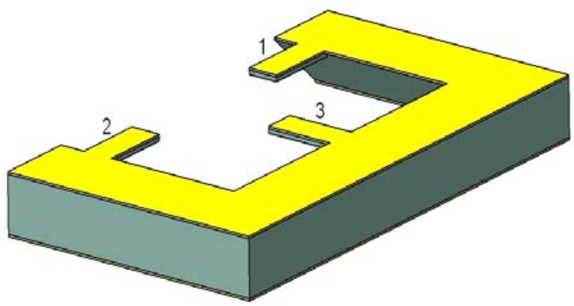


Fig. 3. Scheme of cantilevers used in out-of-plane vibration test.

In order to characterize the geometry and the dynamic behaviour of performed structures, the MSA-500 Micro System Analyzer (Polytech), Fig. 4, was used.

It has a unique combination of non-contact measurement techniques: Scanning Laser-Doppler Vibrometry for characterization of out-of-plane vibrations; Stroboscopic Video Microscopy for measurement of in-plane motion and vibration; White Light Interferometry (WLI) for determination of surface topography; Geometry Scan data acquisition for the vibration measurement.

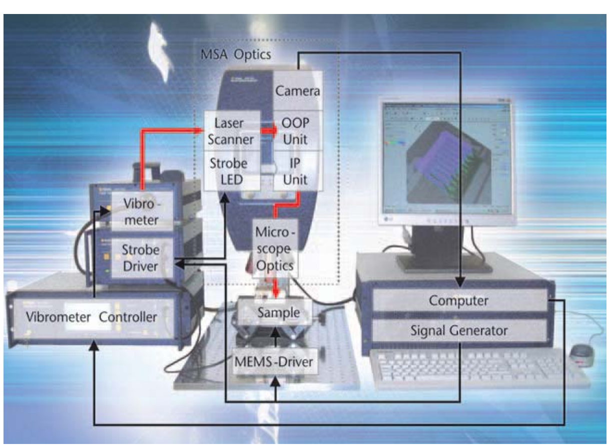


Fig. 4. Schematic representation of the MSA.

The measurements were made on individual chips bonded through the structure rim onto a metal rigid plate of 12 x 12 x 5 mm^3. The plate there is fixed through an elastic double adhesive layer on a piezoelectric excitator consisting of a multilayer ceramics piezoelectric actuator mounted in an elastic pre-tensioned housing.

The equipment has two laser beams whose signals are optically subtracted in the interferometer. The first (reference) beam can be positioned manually (using software) in the visual field of microscope, while the second (measuring) beam is automatically moved in the points of a scanning grid defined by the user, Fig. 5.

During measurement, the reference beam is fixed at a point on the structure rim, and the other one remains mobile. So, only the vibration of the structure points relative to the rim is measured.

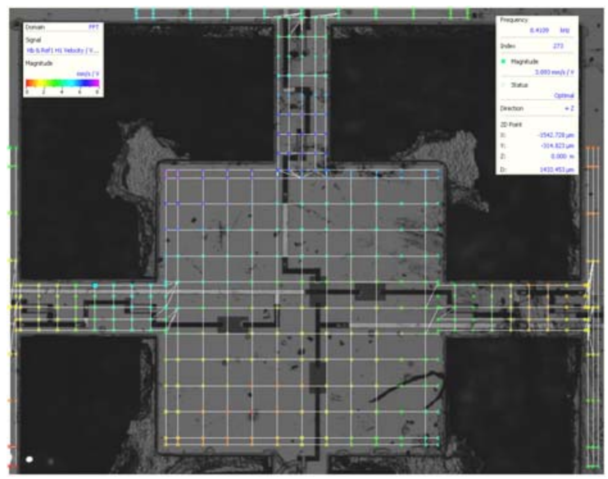


Fig. 5. Image of the accelerometer structure with a scanning grid.

This manner of differential measurement avoids the influence of excitator dynamics on the determinations for the analyzed structure.

Every point, test was performed by using a sinusoidal signal applied to the piezoelectric excitator. The signal was varied in a domain of frequency ranged from 0.5 up to 150 kHz, with 6400 measuring intervals. For the low-frequency modes (met at the determination of accelerometer vibration modes), the resolution in the detection of natural frequencies has been improved by limiting the frequency of testing at 10 kHz.

4. Results

The actual shape of vibration modes and the measured resonance frequencies for the tested cantilevers are given in Fig. 6 and Table 3.

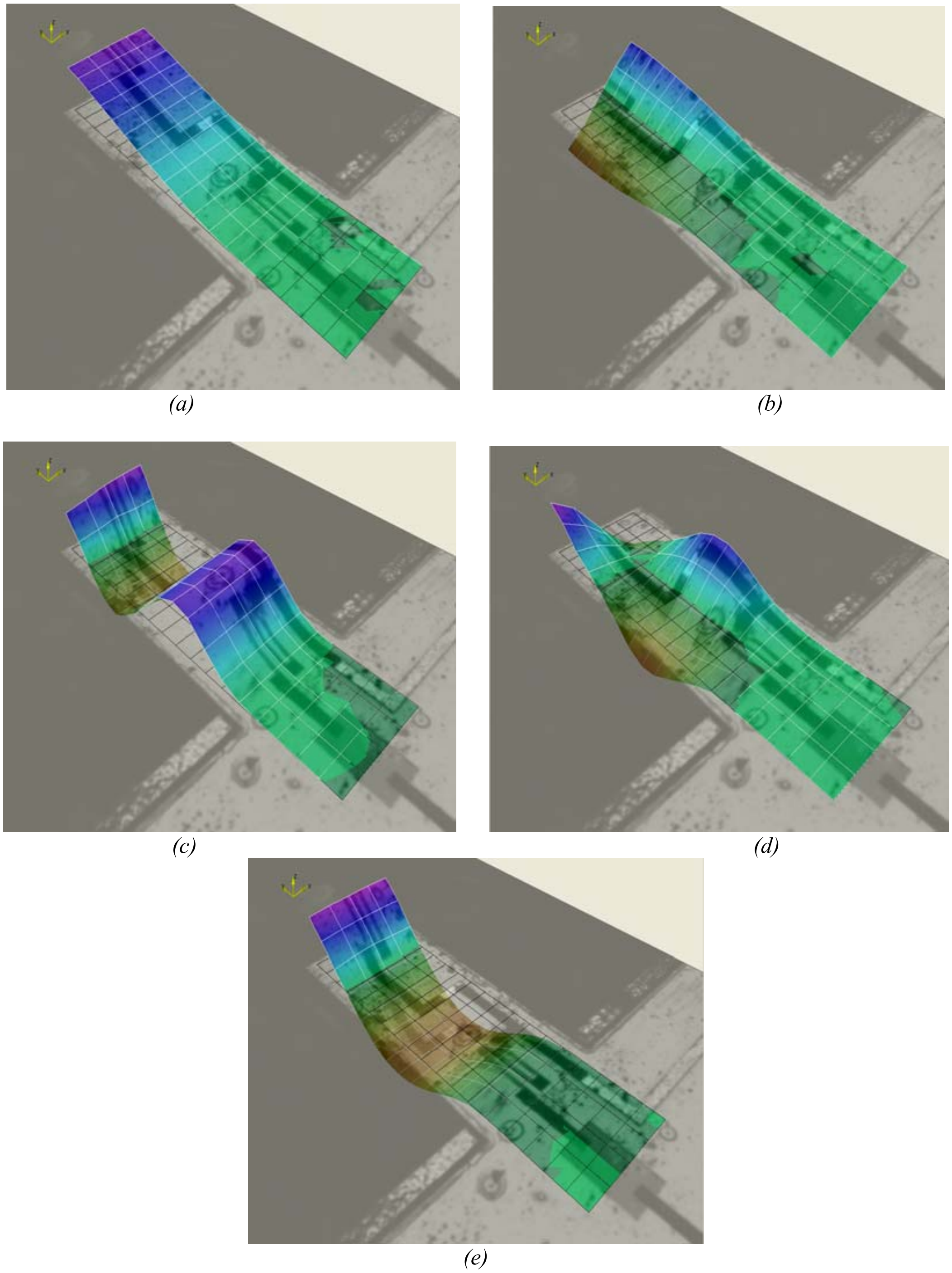


Fig. 6. Vibration modes of the tested cantilevers: the first mode (a), the second mode (b), the third mode (c), the fourth mode (d) and the fifth mode (e).

Table 3. The resonance frequencies measured on cantilever structures using MSA-500 Micro System Analyzer.

Vibration mode	Cantilever position	Resonance frequency (kHz) – average values
f_1 - bending, Fig. 6a	1	22.852
	2	20.885
	3	20.654
f_2 - torsion, Fig. 6b	1	96.289
	2	92.682
	3	93.291
f_3 - bending, Fig. 6c	1	148.378
	2	142.142
	3	143.330
f_4 - torsion, Fig. 6d	1	321.269
	2	313.014
	3	318.047
f_5 - bending, Fig. 6e	1	420.839
	2	409.648
	3	408.964

Observation: at high frequencies, signal to noise ratio is very low, and accuracy in identifying the natural frequencies is reduced.

The values of the measured resonance frequencies (Table 3) were introduced in the equations (1) and (2), for suspension arms of dimensions given in Table 4.

Table 4. Dimensions of tested cantilevers.

Cantilever position	l (μm)	b (μm)	h (μm)
1	877	409.5	17
2	944.5	408	
3	870	411	

The material data used in computation were:

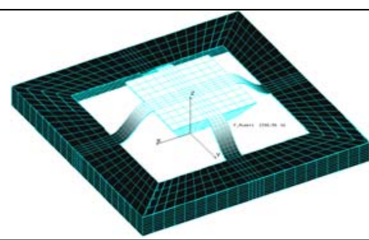
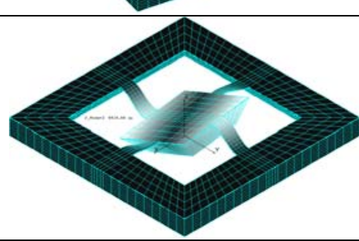
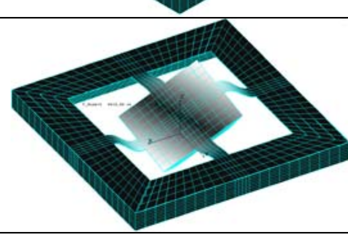
$\rho_{Si} = 2330\ kg/m^3$, $\nu_{Si} = 0.22$.

The obtained values for $E_{dynamic}$ in the two cases, beam model and plate model, are centralized in Table 5.

Table 5. Dynamic Young's modulus values computed with analytical models.

Mode	Position	$E_{dynamic}$ (10^{11} N/m^2) beam model	$E_{dynamic}$ (10^{11} N/m^2) plate model	Position	Mode
1	1	0.955	0.932	1	1
	2	1.073	1.048	2	
	3	0.755	0.737	3	
3	1	1.025	0.792	1	2
	2	1.265	0.859	2	
	3	0.926	0.735	3	
5	1	1.051	0.985	1	3
	2	1.340	1.175	2	
	3	0.961	0.895	3	

Table 6. The first three modes of the accelerometer, experimentally determined and computed with $E_{dynamic}$

Normal mode	Mode shape	f (Hz) FEM, with $E_{dynamic}$ beam model	f (Hz) FEM, with $E_{dynamic}$ plate model	f (Hz) measured with MSA-500 Analyzer
1		2568.4	2398.6	2384.4
2		4730.9	4419.9	4229.7; 4235.9
3		4730.9	4419.9	4837.5; 4835.9

The average values, $E_{dynamic}$ (beam model) = 1.039 $\cdot 10^{11}$ N/m^2 and $E_{dynamic}$ (plate model) = 0.906 $\cdot 10^{11}$ N/m^2 have resulted lower than that of bulk silicon, $E_{(100)}$ = 1.295 $\cdot 10^{11}$ N/m^2, e.g. [7], which was used in the model of numerical simulation from [5].

5. Validation by numerical simulation on the structure of mems accelerometer

For the ideal structure of accelerometer (Fig. 1), considering it entirely made from silicon, as an isotropic model, a numerical simulation was performed by means of COSMOS software package using 9,600 elements of 3D solid interconnected in 12,573 nodes.

The natural frequencies from the first to the third vibration mode obtained by FEM, for $E_{dynamic}$ in the two cases - beam model and plate model, compared with the resonance frequencies measured with MSA-500 Analyzer, are shown in Table 6.

6. Conclusions

A good agreement was found between the values of the resonance (natural) frequencies numerically computed and experimentally determined, with errors of about 7.7% (beam model) and 0.6% (plate model), for the first vibration mode of the structure of MEMS accelerometer.

Given the importance of accuracy of the calculation model relative to the physical model, future research will consider the orthotropic behaviour of silicon.

For structures made from material bulk by etching processes, Young's modulus depends not so the size, but especially of the crystalline orientation.

Because, at resonance, the suspension arms have great deformations, even though the acceleration at input is low, layers of cover for amortization should be studied.

The internal stresses from microbeams, which appear at high levels of boron doping (for piezoresistors), should be studied also.

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