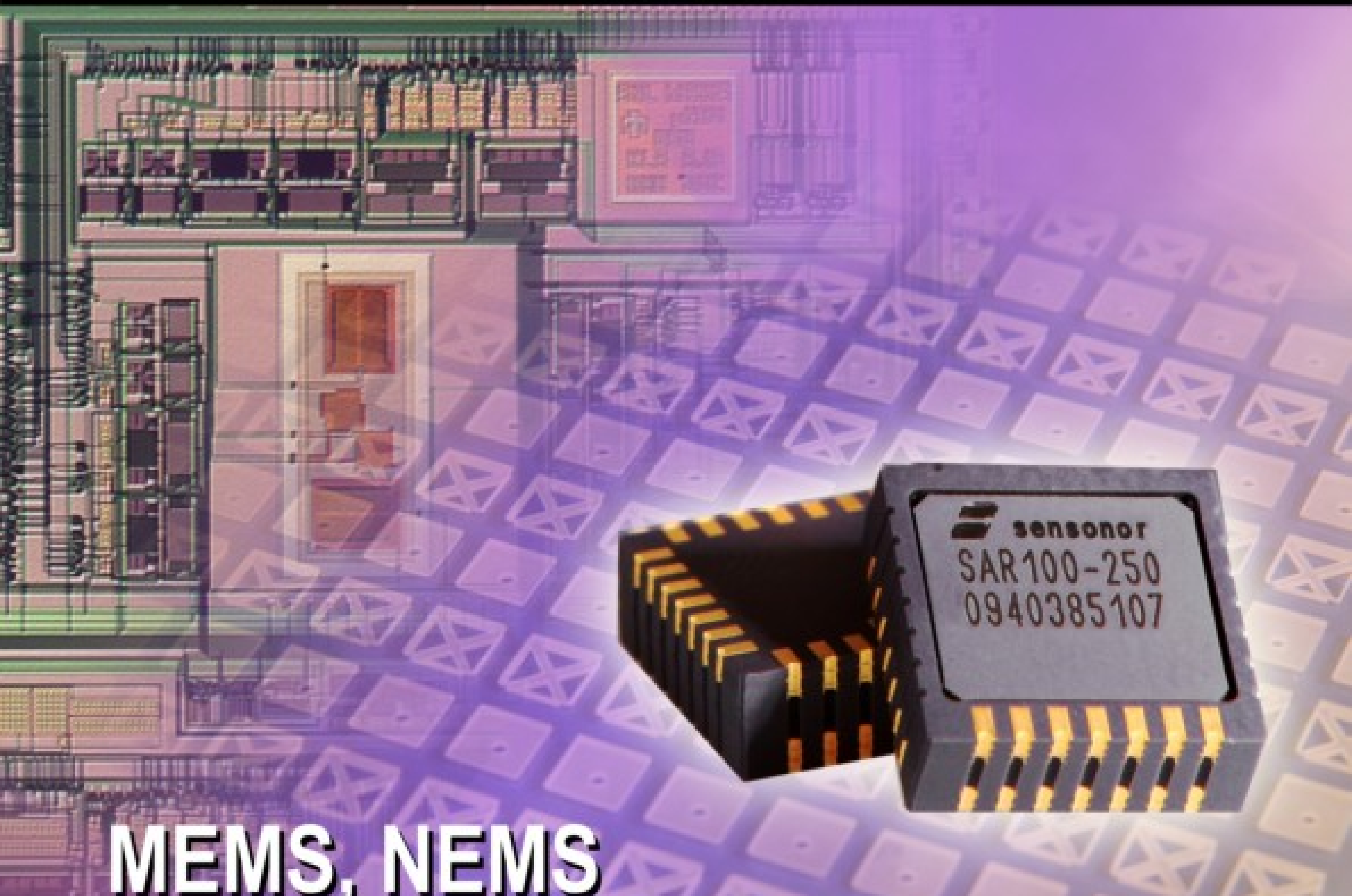
ISSN 1726-5479

SENSORS & TRANSDUCERS

vol. 115
4/10



MEMS, NEMS and Modern Technologies

International Frequency Sensor Association Publishing

IFSA

Sensors & Transducers

Volume 115, Issue 4,
April 2010

www.sensorsportal.com

ISSN 1726-5479

Contents

Volume 115
Issue 4
April 2010

www.sensorsportal.com

ISSN 1726-5479

Research Articles

Authors are encouraged to submit article in MS Word (doc) and Acrobat (pdf) formats by e-mail: editor@sensorsportal.com
Please visit journal's webpage with preparation instructions: http://www.sensorsportal.com/HTML/DIGEST/Submition.htm

International Frequency Sensor Association (IFSA).

Sensors & Transducers Journal, Vol. 115, Issue 4, April 2010, pp. 29-42

Sensors & Transducers

ISSN 1726-5479

http://www.sensorsportal.com

On the Modeling of a MEMS Based Capacitive Accelerometer for Measurement of Tractor Seat Vibration

[1]M. Alidoost, [2]G. Rezazadeh, [1]M. Hadad-Derafshi
[1] Agriculture Machinery Mechanic. Dep. Urmia University, Urmia, Iran
[2] Mech. Eng. Dept. Urmia University, Urmia, Iran
[2] E-mail: g.rezazadeh@mail.urmia.ac.ir

Received: 8 January 2010 /Accepted: 20 April 2010 /Published: 27 April 2010

Abstract: Drivers of heavy vehicles often face with higher amplitudes of frequencies range between 1-80 Hz. Hence, this range of frequency results in temporary or even sometimes permanent damages to the health of drivers. Examples for these problems are damages to the vertebral column and early tiredness, which both reduce the driver's performance significantly. One solution to this problem is to decrease the imposed vibration to the driver's seat by developing an active seat system. These systems require an online measuring unit to sense vibrations transferred to the seat. The measuring unit can include a capacitive micro-accelerometer on the basis of MEMS which measure online vibrations on the seat. In this study, the mechanical behavior of a capacitive micro-accelerometer for the vibration range applied to a tractor seat has been simulated. The accelerometer is capable to measure step, impact and harmonic external excitations applied to the system. The results of the study indicate that, with increasing the applied voltage, the system sensitivity also increases, but the measuring range of vibrations decreases and vice versa. The modeled accelerometer, at damping ratio of 0.67 is capable to measure accelerations within the frequency range of lower than 130 Hz. *Copyright © 2010 IFSA.*

Keywords: MEMS, Accelerometer, Capacitive, Tractor seat vibration

1. Introduction

Work tasks on a tractor necessitate a number of actions which the operator performs them, that puts different demands on the body. Examples of these demands are front steering of tractor, looking backward to observe and control the implement, force required to operate clutch and brake pedals, etc. The task and workplace specify the postures and make a sample of loading on the structures of the body of the operator. These inputs stimulate mechanisms of discomfort that need to be quantified in terms of mechanical requirements for seat design and its behavior [1]. Occupational heavy vehicle drivers are exposed to considerably large magnitudes of vibration in the relative low frequency range, which cause discomfort, trouble, and several health and safety risks in extreme cases. The driver in such an environment is exposed to vibrations that happen simultaneously along the three translational and rotational axes. The low frequency and high magnitude whole-body vibration (WBV) has been related with reduced comfort and health. In accordance with ISO 2631/1, vibrations from 1 to 80 Hz for tractor seat are important [2].

Transmission of vehicle vibration to the occupant, body pressure distributed under and supporting both the buttocks, thighs and back of an operator, control of posture either statically or dynamically through differing load paths, clothing and seat covering material, perceptions and interior ergonomic characteristics are sources of such discomfort. Chronic health effects of WBV include disorders of the musculoskeletal structure, such as spine and supporting structure. The studies of many epidemiologic have suggested that there is a strong relation between the exposure to WBV and feeling of pain in low back [3, 4]. The best procedure for reducing WBV is to solve the problem at its original. If the WBV is created by the engine, so it should be redesigned to eliminate or reduce the vibrations. For off-road vehicles, the driving areas where there are no roads are the most important cause of the WBV comes from. The WBV caused in this manner needs to be attenuated with a vibration cancellation system. Therefore, considerable efforts are being made to design effective suspension seats and continuously investigate the transmission of vibration to and through the human body to enhance the understanding of human response to vibration. Thus seat is one component affecting that can modify loads on body structures to reduce operator's discomfort. WBV excites the natural frequency of the body so it is harmful to humans. The dynamics of the human body has been researched to specify which frequencies are most harmful. Gniady and Bauman (1991) from Aura Systems determined the natural frequencies of the human body in the sitting position [5]. The human body usually has a natural frequency between 4 and 5 Hz. The major effects in work-places due to WBV have been reported to occur below 80 Hz [6]. Table 1 shows a number of vital body parts, and the corresponding natural frequencies. The agricultural and forestry vehicles have been most widely studied with regard to their ride vibration and safety [3, 4, 7].

Table 1. Natural frequency ranges of human body.

Body Part	Frequency range(Hz)
Trunk	3-6
Chest	4-6
Spine	3-5
Shoulders	2-6
Stomach	4-7
Eyeball	20-25
Thorax	3
Heart	4-5
Head	30
Colon	20-25

The suspensions and tires properties, apart from the vehicles weights and dimensions of vehicles, strong affect the nature of transmitted vibration. Such vehicles cause appreciable frequency-weighted magnitudes of vibration transmitted to the driver seat along the longitudinal and lateral axes, which are summarized in table 2 [4, 8, 9]. The vector sum of accelerations a_v is also presented in the Table 2.

Table 2. Magnitudes of frequency-weighted rms accelerations due to vibration Measured along the longitudinal and lateral and vertical axis, on the seats of agricultural tractors.

Operation	$a_{wx}(m/s^2)$	$a_{wy}(m/s^2)$	$a_{wz}(m/s^2)$	$a_v(m/s^2)$
Tractor on road	0.22-1.04	0.37-1.2	1.42-2.15	1.8-3.09
Tractor off road	0.24-1.79	0.51-1.70	1.25-2.99	1.58-4.57
Tractor Harrowing	0.2-0.69	0.2-0.8	0.38-0.96	0.85-1.67
Tractor ploughing	0.3-1.3	0.2-0.6	0.3-0.59	0.87-1.26

One solution to this problem is to develop an active seat capable of canceling the vibrations felt by the operator. In active suspension systems need an accelerometer for measuring online vibration transmitted to driver seat. Micro-Electro Mechanical Systems (MEMS) are a means to measure or manipulate physical systems [10]. MEMS can be almost any mechanical system that interacts physically or measures physical properties but was built using micro fabrication techniques with electrical components. The MEMS devices currently made today use equipment that is inexpensive and readily available to this new market. The market for MEMS devices has reached to 5.1 billion dollar in 2005. We expect that the MEMS markets will reach 9.7 billion dollar in 2010, representing a compound annual growth rate of almost 15 % [11].

MEMS accelerometers play an important role in the field of sensors [12]. Accelerometers can be arranged according to their form of transduction mechanism: capacitive, piezoelectric, piezoresistive and thermal. One of the most widely used in MEMS-based accelerometers is capacitive sensing plane [13]. This is due to their high sensitivity, good Dc response and noise performance, low drift, low temperature sensitivity, low power dissipation and simple structure [14].

Due to the importance of the measurement of acceleration on a tractor seat, along with the development of the MEMS technology for micro-instrumentation systems, in the last few decades, it seems that, the design of an accelerometer on the base of MEMS is a matter of importance. In this paper, a mathematical model was developed to study the mechanical behavior of a capacitive type micro-accelerometer for online measuring of the vibration on the tractor seat.

2. Model Description

The capacitive micro-accelerometer consists of a microbeam, fixed at one end (Fig. 1). The microbeam is separated from the substrate by a gap distance g_0 .A constant voltage is provided for both plates. The behavior of microbeam against applied external forces appeared to be similar to the theory of beam vibration. A concentrated mass, m_0, is fixed at one end of the beam which assures the operation of the instrument within the range of the tractor seat vibration. As the external force is applied to the beam, it vibrates and hence, the capacity of the condenser changes which indicates the mount of the force applied to the system. The micro-accelerometer shown in Fig. 1 has length (l), width (b), and thickness (h) in micro scale.

3. Mathematical Modeling

The theory of flexural vibration is used to study the behavior of a cantilever beam model. Beams are part of a 1-dimentional structure which, according to the type of loading applied to their surface, indicate a bending behavior and transfer this load to its supports as shearing force and bending moment.

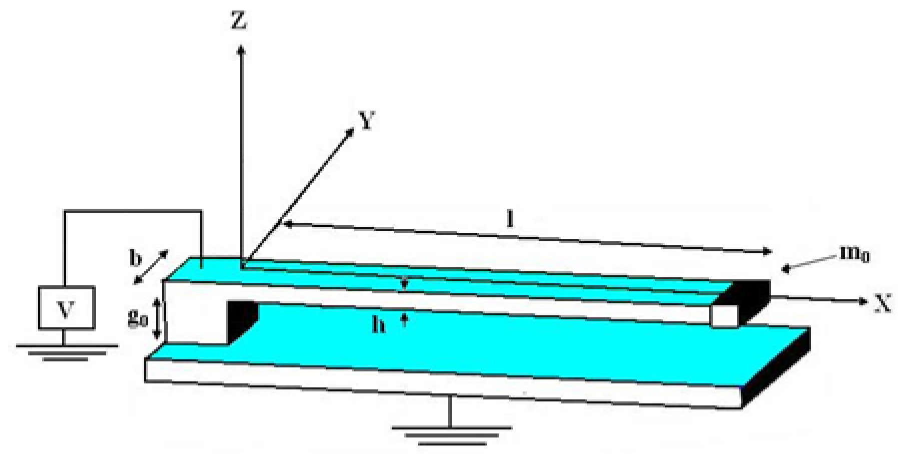


Fig. 1. Capacitive accelerometer.

3.1. Non-linear Static Equation on Cantilever Beam

The equations showing small deflections on the beams under transversal loads, on the basis of Euler-Bernolli beam theory, and ignoring shearing strains, are as follows:

$$EI\frac{d^4w_s(x)}{dx^4}=\frac{\varepsilon_0 bV^2}{2(g_0-w_s(x))^2} \tag{1}$$

In the above Eq. (1); w_s is the static deflection in the z direction, E the Module of Elasticity, I moment of inertia about to y axis, V is the voltage between the substrate and the micro beam, ε_0 is the permittivity of air and F_e, electro-static force applied per unit length [15]. The boundary conditions at the fixed end are:

$$w_s(0)=0, \qquad \frac{dw_s}{dx}\Big|_{x=0}=0, \tag{2}$$

And boundary conditions dominant on the free end of the beam are:

$$\frac{d^2w_s}{dx^2}\Big|_{x=l}=0, \qquad \frac{d^3w_s}{dx^3}\Big|_{x=l}=0. \tag{3}$$

3.2. Differential Equation Dominant in Dynamic behavior of Cantilever

As in static equation, ignoring shearing strain dynamic equation can be written as:

$$EI\frac{\partial^4 w_T(x,t)}{\partial x^4}+\left(\rho A+m_0\delta(x-l)\right)\left(\frac{\partial^2 w_T(x,t)}{\partial t^2}+a\right)+c\frac{\partial w_T(x,t)}{\partial t}=\frac{\varepsilon_0 bV^2}{2(g_0-w_T(x,t))^2} \tag{4}$$

In this Eq. (4); A is the area of cross section of the beam, ρ, density, a, acceleration that should be measured, c, damping coefficient per unit length of the beam, m_0, mass at the tip of the cantilever beam, on the beam, $\delta(x-l)$, is the Delta function, and w_T, is the beam displacement [16]. The total displacement can be obtained by summing the two static and dynamic ones:

$$w_T(x,t) = w_s(x) + \varepsilon(x,t) \tag{5}$$

Here, $\varepsilon(x,t)$ are the small vibrations about the static equilibrium position. Thus the Eq. (4) can be written as:

$$EI\frac{\partial^4 (w_s(x)+\varepsilon(x,t))}{\partial x^4} + \left(\rho A + m_0\delta(x-l)\right)\left(\frac{\partial^2 (w_s(x)+\varepsilon(x,t))}{\partial t^2} + a\right) + c\left(\frac{\partial (w_s(x)+\varepsilon(x,t))}{\partial t}\right) = \frac{\varepsilon_0 b V^2}{2(g_0 - (w_s(x)+\varepsilon(x,t))^2} \tag{6}$$

Considering Taylor Expansion, the electro-static force about static equilibrium position, w_s, for the second hand of the Eq. (6) would be:

$$\frac{\varepsilon_0 b V^2}{2(g_0 - (w_s(x)+\varepsilon(x,t))^2} = \frac{\varepsilon_0 b V^2}{2(g_0 - w_s(x))^2} + \frac{\varepsilon_0 b V^2}{(g_0 - w_s(x))^3}\varepsilon(x,t) + O(2) \tag{7}$$

Ignoring the terms of higher levels in the formula (7) and solving it in the Eq. (6) will result in:

$$EI\frac{d^4 w_s(x)}{\partial x^4} + EI\frac{\partial^4 \varepsilon(x,t)}{\partial x^4} + \left(\rho A + m_0\delta(x-l)\right)\left(\frac{\partial^2 \varepsilon(x,t)}{\partial t^2} + a\right) + c\left(\frac{\partial \varepsilon(x,t)}{\partial t}\right) = \frac{\varepsilon_0 b V^2}{2(g_0 - w_s(x))^2} + \frac{\varepsilon_0 b V^2}{(g_0 - w_s(x))^3}\varepsilon(x,t) \tag{8}$$

By omitting the static equation derived from Eq. (8), the relationship for the small vibrations about the static equilibrium position will be calculated as follow:

$$L(\varepsilon(x,t)) = EI\frac{\partial^4 \varepsilon(x,t)}{\partial x^4} + \left(\rho A + m_0\delta(x-l)\right)\left(\frac{\partial^2 \varepsilon(x,t)}{\partial t^2} + a\right) + c\left(\frac{\partial \varepsilon(x,t)}{\partial t}\right) - \frac{\varepsilon_0 b V^2}{(g_0 - w_s(x))^3}\varepsilon(x,t) = 0 \tag{9}$$

Here, $L(\varepsilon(x,t))$ is the operator of linear differential and ε satisfies the boundary condition, similar to the boundary condition in Eqs. (2) and (3).

3.3. Charge of the Capacitor

Considering the applied model and the form of displacements, the following formula is suggested for measuring the charge of the capacitor:

$$C(t) = \int_0^l \frac{\varepsilon_0 b}{(g_0 - w_T(x,t))}dx \tag{10}$$

4. Numerical Solution

4.1. Static Equation

Solving the non-linear equations is difficult and usually there is no analytical solution for them. Numerical solutions, by using finite difference method (FDM) or finite element method (FEM), result in non-linear algebraic equations which is also very time consuming process to solve them and most of the time these solutions are not stabilized. Therefore, a team of researchers have changed the non-linear equation governing on static deflections of beam about w_S, to linear equation. However, where w_S =0, especially when the applied voltage is high and the deflections are also high, occurrence of several errors are likely. Therefore, for solving the non-linear equations, step-by-step linearization

(SSLM) was used [17]. According to this method, if w^k, is considered accelerometer displacement for the applied voltage, V^k, then:

$$w^{k+1} = w^k + \psi(x) \tag{11}$$

$$V^{k+1} = V^k + \delta V \tag{12}$$

Then considering the Eq. (1), following equations can be written:

$$EI\frac{d^4(w^{k+1})}{dx^4} = \frac{\varepsilon_0 b(V^{k+1})^2}{2(g_0 - w^{k+1})^2} \tag{13}$$

$$EI\frac{d^4(w^k)}{dx^4} = \frac{\varepsilon_0 b(V^k)^2}{2(g_0 - w^k)^2} \tag{14}$$

By subtracting the Eq. (14) from the Eq. (13):

$$EI\frac{d^4(\psi)}{dx^4} = \frac{\varepsilon_0 b(V^{k+1})^2}{2(g_0 - (w^k + \psi))^2} - \frac{\varepsilon_0 b(V^k)^2}{2(g_0 - w^k)^2} \tag{15}$$

Developing the Taylor expansion of the force about w^k, for the first term of the right side of the Eq. (15), it can be written:

$$\frac{\varepsilon_0 b(V^{k+1})^2}{2(g_0 - (w^k + \psi))^2} = \frac{\varepsilon_0 b(V^{k+1})^2}{2(g_0 - w^k)^2} + \frac{\varepsilon_0 b(V^{k+1})^2}{(g_0 - w^k)^3}\psi \tag{16}$$

Finally, the linear equation below can be used to calculate ψ:

$$EI\frac{d^4(\psi)}{dx^4} - \frac{\varepsilon_0 b(V^{k+1})^2}{(g_0 - w^k)^3}\psi = \frac{\varepsilon_0 b((V^{k+1})^2 - (V^k)^2)}{2(g_0 - w^k)^2} \tag{17}$$

Using central finite difference formula with constant subdivisions Δx characteristics, the foregoing equation was discretized and by solving obtained liner system of algebraic equations, ψ can be calculated at a given applied voltage.

4.2. Dynamic Equation

Different methods are available for solving the linear and non-linear differential equations. Among them weighting residuals approach is considered. In this method, for solving a dynamic equation firstly, the shape functions must be obtained, and then, along with this Galerkin-Bubnov procedure, must be applied. An approximate N-term solution for deriving small vibrations of the beam is defined as:

$$\varepsilon_N(x,t) = \sum_{n=1}^{N} u_n(t)\varphi_n(x) \qquad j = 1,..N\,, \tag{18}$$

where, $u_n(t)$ and $\varphi_n(x)$ are the function of time and shape, respectively. The shape function must satisfy of the geometrical boundary conditions. Therefore, solving the small vibration equation to the formula (9) the following error function will be obtained:

$$L(\varepsilon_N(x,t)) =$$
$$EI\sum_{n=1}^{N}\varphi_n^{IV}(x)u_n(t)+\left(\rho A+\left(m_0\delta(x-l)\right)\right)\left[\left(\sum_{n=1}^{N}\varphi_n(x)\ddot{u}_n(t)\right)+a\right]+c\sum_{n=1}^{N}\varphi_n(x)\dot{u}_n(t)-\frac{\varepsilon_0 bV^2}{(g_0-w(x))^3}\sum_{n=1}^{N}\varphi_n(x)u_n(t)=E_r(x,t) \quad (19)$$

where $E_r(x,t)$ the represents some residual obtained by substituting of the approximate solution into the differential equation. Using the $\varphi_j(x)$ weighting function the same as shape function and applying Galerkin-Bubnov procedure a set of N nonlinear ordinary differential equations with respect to time can be obtained as:

$$\int_0^l \varphi_j(x).L(\varepsilon_N(x,l))dx = 0 \qquad j = 1...N \quad (20)$$

By using Eq. (20), ordinary dynamics equation can be obtained as follow:

$$\sum_{n=1}^{N} M_{jn}\ddot{u}_n(t)+\sum_{n=1}^{N} C_{jn}\dot{u}_n(t)+\sum_{n=1}^{N} k_{jn}u_n(t)=F_{jn} \quad (21)$$

Here, M_{jn} is the mass:

$$M_{jn}=m_0\varphi_j(l)\varphi_n(l)+\rho A\int_0^l \varphi_j(x)\varphi_n(x)dx \quad (22)$$

and C_{jn} , is the damping:

$$C_{jn}=c\int_0^l \varphi_j(x)\varphi_n(x)dx \quad (23)$$

K_{jn} is the stiffness of the system which equals to the difference between mechanical and electrical stiffness:

$$K_{jn}=K^m{}_{jn}-K^e{}_{jn}=EI\int_0^l \varphi_j(x)\varphi_n^{IV}(x)dx-\int_0^l \frac{\varepsilon_0 bV^2}{(g_0-w(x))^3}\varphi_j(x)\varphi_n(x)dx \quad (24)$$

And, finally, F_{jn} is the force vector, as

$$F_{jn}=m_0 a\varphi_j(l)+\rho Aa\int_0^l \varphi_j(x)dx \,. \quad (25)$$

5. Numerical Results and Discussion

Physical properties and geometrical characteristics of the capacitive accelerometer are shown in Table 3.

Table 3. Physical properties and geometrical characteristics of the capacitive accelerometer.

$l[\mu m]$	250	$E[Gpa]$	169
$b[\mu m]$	100	$\varepsilon_0[F/m]$	8.8541878×10^{-12}
$h[\mu m]$	3	$\rho[Kg/m^3]$	2300
$g_0[\mu m]$	1.5	$c[Ns/m^2]$	48.26
$m_0[Kg]$	10×10^{-7}	N	1

The static and dynamic solutions of the model were carried out by programming in Matlab software and unstable voltage and charge of the capacitor were resulted in step, impact and harmonic accelerations. The pull-in or unstable voltage is defined as the voltage which causes the microbeam to jump towards the next electrode and stick to it, thus limit the function range of the system. The stability of the model can be evaluated by static solving it. Also the best damping value which can measure 1-80 Hz frequencies was recognized.

5.1. Results of the Static Behavior of the Model

Using the SSLM and Finite Differential Methods, the unstable voltage is determined. The best condition for voltage gradual change and the number of elements on the microbeam are shown in Tables 4 and 5. According to these tables, with voltage change equals to 0.08 and number of elements equals to 11, the system unstable voltage is 11.2 volt.

Table 4. Calculation of unstable voltage with 50 elements on the micro beam for different voltage change.

Voltage change (V)	2	0.1	0.08	0.06
Pull-in voltage (V)	12	11.3	11.2	11.2

Table 5. Calculation of unstable voltage with the voltage change of 0.08 for different elements.

Number of element	7	9	11	13
Pull-in voltage (V)	11	11.1	11.2	11.2

In this microbeam the deflection at the last point, for determining unstable voltage is very important. The behavior of this point to the increasing of the voltage has been shown in Fig. 2. In Fig. 3, the effects of voltages lower than the unstable voltage have also been shown. It is clear that, as the voltage increases the deflection of the beam increases.

Because of the importance of pull-in voltage determination, scholars had used different methods for specifying it. One of these methods is, using of closed form formulations which determine pull-in voltage with high accuracy. In Fig. 4, the obtained pull-in voltage by FDM has compared with the results of closed formulas and experimental findings [18-19].

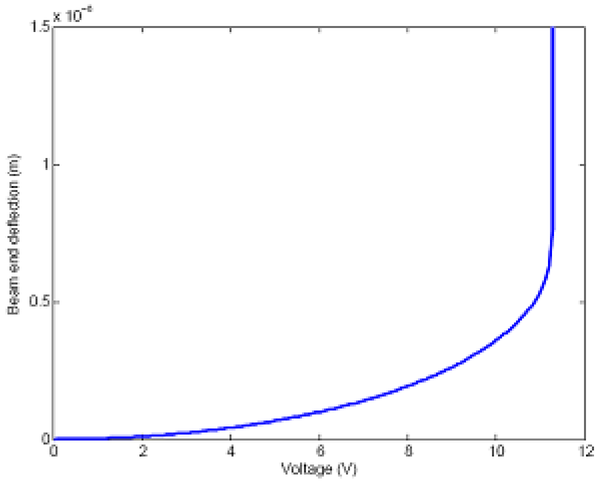


Fig. 2. Deflection of the last point of micro beam versus increasing of voltage.

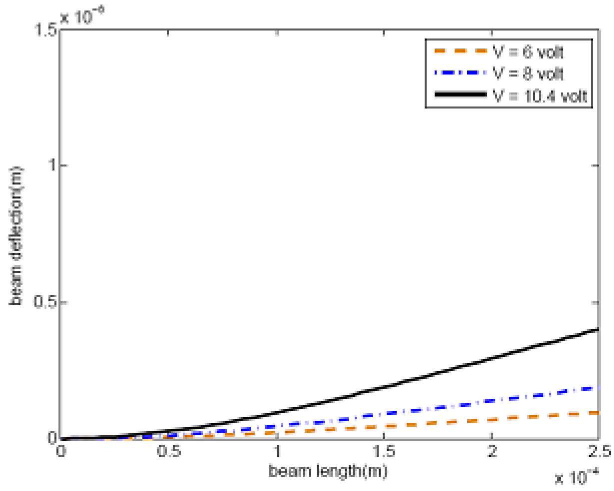


Fig. 3. Micro beam deflections at different applied voltage.

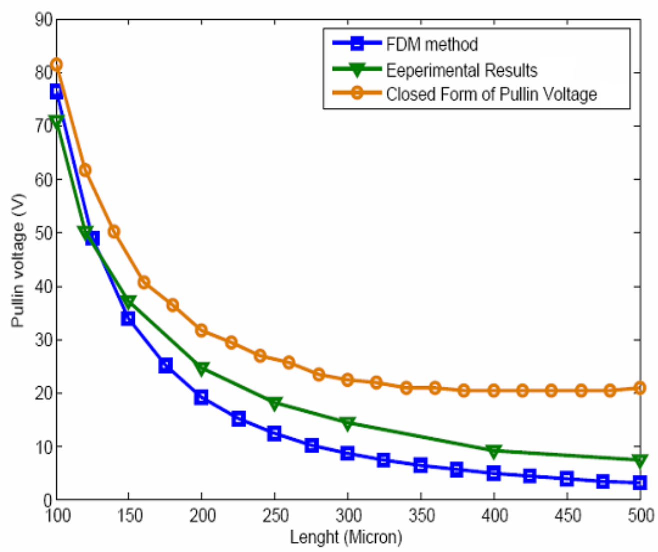


Fig. 4. Pull-in voltage comparison with experimental measurements.

The compared microbeam have width 40 µm, thickness 2.1 µm, gap 2.34 µm and module of elasticity 155 Gpa as its length change between 100-500 µm. The computed Pull-in voltages using FDM method has a strong similarity with the experimental results [18].

5.2. Results of Dynamic Behavior of Model

For evaluation of dynamic behavior of capacitive micro-accelerometer, accelerations were applied at the zero time. These accelerations are at the range shown in Table 2. w_s, which is the static deflection of the beam, due to the base voltage, is considered as a function of shape. The system, under the applied accelerations, oscillated with the time, and because of the damping, its oscillations declined during the time.

5.2.1. System Response to the Step Excitation

Different step accelerations were suddenly applied to the system at zero moment and microbeam displacement and its capacitive variations at the applied voltage of 9.6 volt, are shown in Figs. 5 and 6. As it is indicated in these Figs., at the defined damping for the system, the amplitude of the oscillation and charge variation of the capacitor will stabilize within a short period of time.

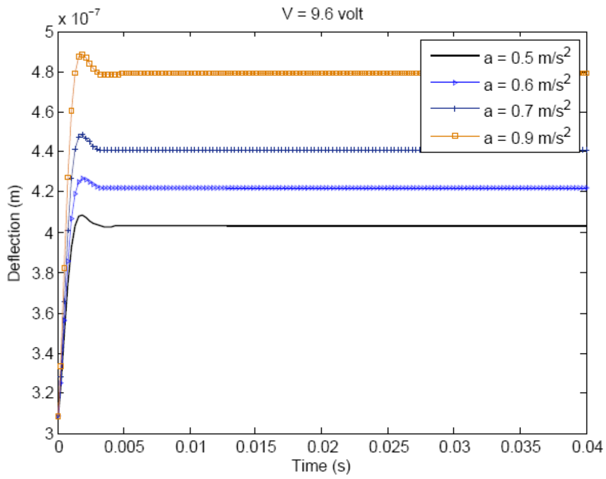


Fig. 5. Micro beam deflection versus different step acceleration.

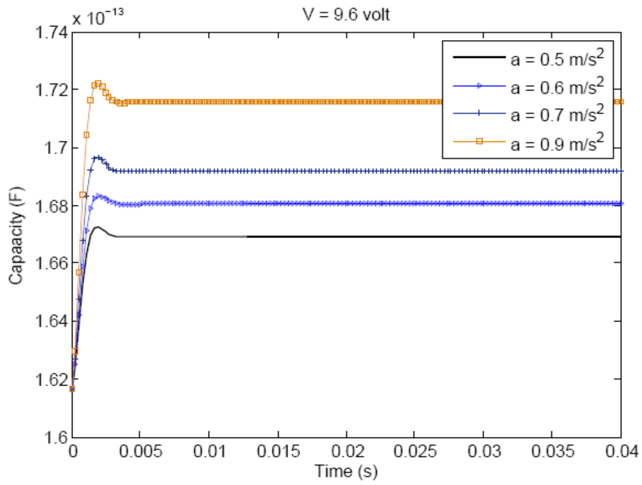


Fig. 6. Micro beam capacity versus different step acceleration.

5.2.2. System Response to the Impact Excitation

The system responses to the impact excitation are shown in Fig. 7. These excitations are applied to the system in a moment. As it is shown, the system presents responses proportional to the amplitude of excitation. The charge variations of the microbeam, at different accelerations are also shown in Fig. 8.

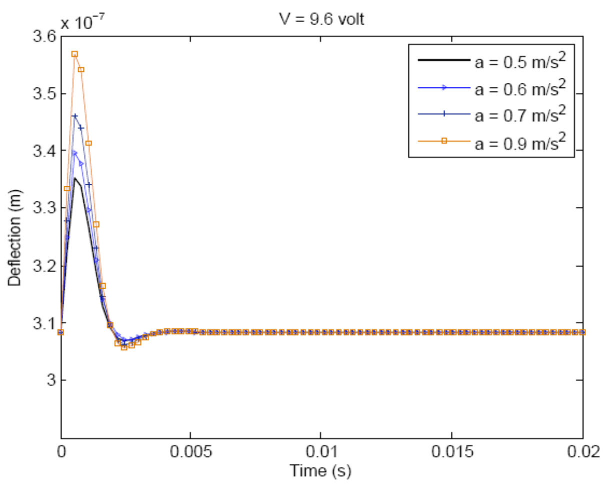


Fig. 7. Deflection of micro beam versus different impact acceleration.

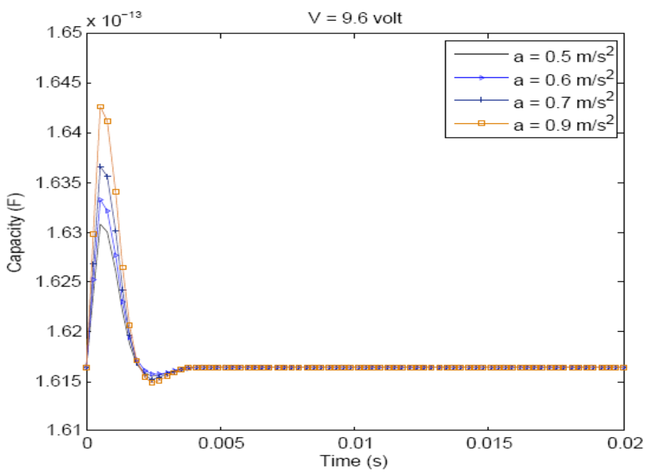


Fig. 8. Micro beam capacity versus different impact acceleration.

5.2.3. System Response to the Harmonic Excitation

The harmonic excitation applied to the system is in the form of $a = a_0 \sin(wt)$, where a_0 is the amplitude of the applied acceleration and w is an applied stimulation frequency that is 50 Hz. The system responses to different excitations and the charge variations of the microbeam at the applied voltage of 9.6 volt are shown in Figs. 9 and 10.

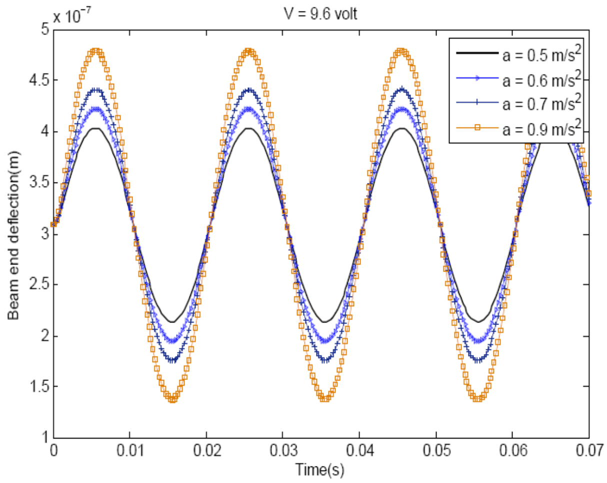


Fig. 9. Deflection of micro beam versus different harmonic excitation in 50 Hz.

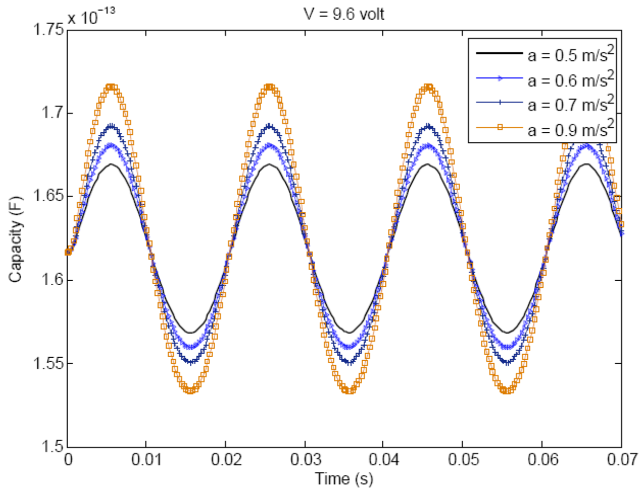


Fig. 10. Micro beam capacity versus different harmonic excitation in 50 Hz.

In Fig. 11, capacity of the microbeam versus increasing acceleration at two different applied voltages has been shown. Results show that, the system is more sensitive at higher voltages.

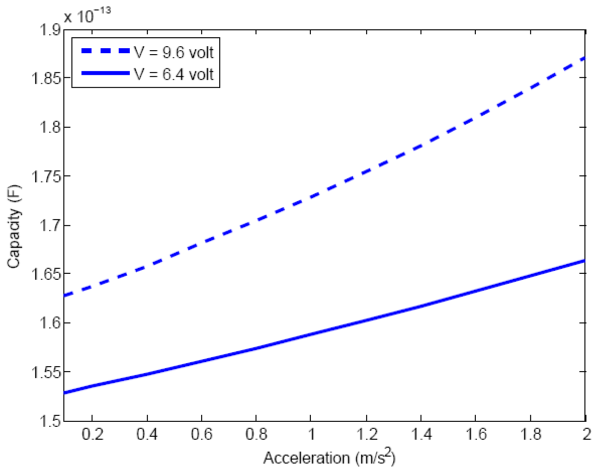


Fig. 11. The variation of micro beam capacity versus increasing acceleration.

5.2.4. Frequency Response

The frequency responses of the system in different damping and at applied voltage of 9.6 volt are shown in Fig. 12. By attention to Fig. 12, the natural frequency obtained for the accelerometer is 336 Hz. These graphs show that in the area of frequencies close to resonance, the damping has a great effect on the magnitude and the phase angle. As it is shown, with increasing system damping, the magnitude ratio variations at frequencies close to the resonance frequency reduce and the range of proper measurable frequencies increases.

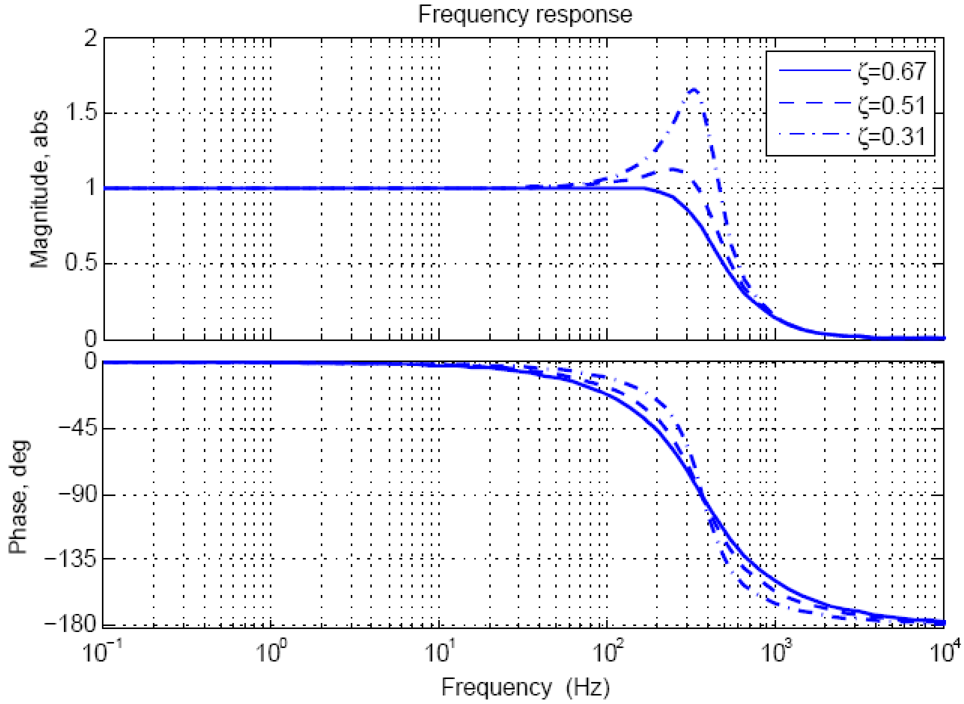


Fig. 12. Variations of magnitude and phase angle versus frequency in different damping.

The damping value of 0.67 which provides frequency range of 0.1-130 Hz for the accelerometer that would be desirable. This value covers the important frequency range of 1-80 Hz.

In Fig. 13, effects of applied voltage variation at a damping of ζ=0.32 have been evaluated. According to the Fig. 13, by increasing voltage the natural frequency of the system decreases and magnitude ratio variation at the frequencies close to the resonance frequency decline.

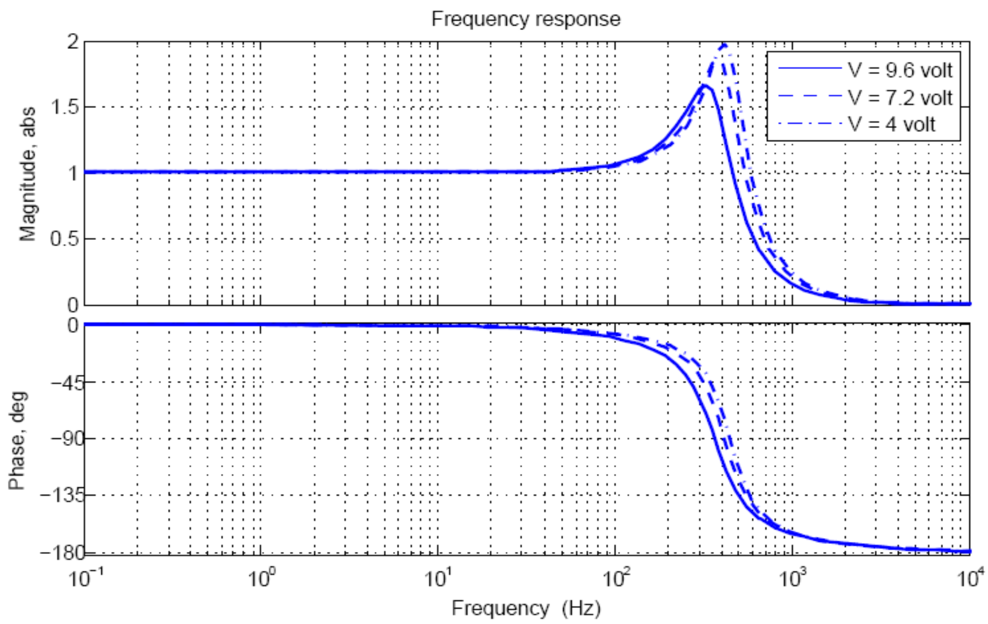


Fig. 13. Variations of magnitude and phase angle versus frequency at different applied voltages.

6. Conclusions

In this paper a capacitive accelerometer is designed for measuring applied acceleration to tractor seat. In this study, for solving the static and dynamic non-linear equations of suggested model, we used SSLM and Galerkin-Bubnov approaches. The accelerometer static and dynamic behaviors in step and impact and harmonic accelerations applied to the system had modeled and the microbeam capacity had obtained for these stimulations. Results indicated that the accelerometer has a very good response for the small range of vibrations applied to the tractor seat. Moreover, by attention to frequency response graph, at the damping of 0.67, the amplitude of the accelerometer, for the excitation frequencies lower than 130 Hz, was independent of this frequency range and the accelerometer has a useful frequency range between 0.1-130 Hz.

Reference

[1]. D. C. Viano, D. V. Andrzejak, Research issues on the biomechanics of seating discomfort: an overview with focus on issues of the elderly and low-back pain, *Society of Automotive Engineers*, Technical Paper Series 920130, 1992.

[2]. ISO 2631-1, Mechanical Variation and Shock-Evaluation of Human Exposure to Whole-body Vibration, Part I: General Requirements, International Organization for Standardization, Geneva, 1997.

[3]. M. Bovenzi and C. T. 1. Hulshof **.**An Updated Review of Epidemiologic Studies on the Relationship between Exposure to Whole-Body Vibration and Low Back Pain (1986-1997), *International Archives of Occupational and Environmental Health*, Vol. 72, 1999, pp. 351-365.

[4]. A. Kumar, P. Mahajan, D. Mohan and M. Vargjese, Tractor Vibration Severity and Driver Health: A Study from Rural India, *Journal of Agricultural Engineering Research*, Vol. 80, Issue 4, 2001, pp. 313-328.

[5]. J. Gniady and J. Bauman, Active seat isolation for construction and mining vehicles, *in SAE Tech, Paper Series*, 1991, pp. 1-6.

[6]. M. J. Griffin. Handbook of Human Vibration, *Academic Press*, London, 1990.

[7]. G. S. Paddan and M. J. Griffin, Effect of Seating on Exposures to Whole-Body Vibration in Vehicles, *Journal of Sound and Vibration*, Vol. 253, Issue 1, 2002, pp. 215-241.

[8]. A. Marsili, L. Ragni, G. Santoro, P. Servadio and G. Vassalini, Innovative System to Reduce Vibrations on Agricaltural Tractors: Comparative Analysis of Acceleration Transmitted Through the Driving Seat, *Biosystem Engineering*, Vol. 81б Issue 1, 2002, pp. 35-47.

[9]. A. J. Scarlet, J. S. Price and R. M. Stayner, Whole-Body Vibration: Initial Evaluation of Emissions Originating from Modern Agricaltural Tractors, Silsoe Research Institue and RMS Vibration Test Laboratory for the Health and Safety Executive (Contract Research Report 413/2002), 2002.

[10].M. J. Madou, Fundamentals of Micro Fabrication, *CRC Press,* Boca Raton, Fla, 1997.

[11].J. Eloy, Status of the MEMS industry in 2006, *Journal of Microsystem Technologies,* Vol. 66, Issue 4, 2006, pp. 521-525.

[12].B. Bais and B. Y. Majlis, Mechanical Sensitivity Enhancement of an Area Changed Capacitive Accelerometer by Optimization of the Device Geometry, *Springer Netherlands*, Vol. 44, Issue 2, 2005, pp. 175-183.

[13].N. Yazdi, A. Salian and K. Najafi, A High Sensitivity Capacitive Microaccelerometer with a Folded-Electrode Structure, *Tech. Dig. IEEE Micro Electro Mechanical Systems Workshop (MEMS'99)*, 1999.

[14].L. K. Baxter, Capacitive Sensors, *IEEE Press,* Piscataway, N. J, 1997.

[15].E. Abdel-Rahman, M. Younis, and A. Nayfeh, Characterization of the mechanical behavior of an electrically actuated microbeam, *Journal of Micromechanics and Microengineering*, Vol. 12, 2002, pp. 759-766.

[16].M. Younis, E. Abdel-Rahman, A study of the nonlinear response of a resonant microbeam to an electric actuation, *Journal of Nonlinear Dynamics,* Vol. 31, 2002, pp. 91–117.

[17].Gh. Rezazadeh, A. Tahmasebi and M. Zubtsov, Application of Piezoelectric Layers in Electrostatic MEM Actuators, Controlling of Pull-in Voltage, *Journal of Microsystem Technologies,* Vol. 12 Issue 12, 2006, pp. 1163- 1170.

[18].R. K. Gupta, P. M. Ostberg and S. D. Senturia, Material Property Measurement of Micromechanical Polysilicon Beam, *SPIE 1996 Conference,* 1996.

[19].M. A. Crisfield, Nonlinear Finite Element Analysis of Solids and Structures, Advanced Topics, Vol. 2, *Wiley*, Chichester, 1997.

Guide for Contributors

Aims and Scope

Sensors & Transducers Journal (ISSN 1726-5479) provides an advanced forum for the science and technology of physical, chemical sensors and biosensors. It publishes state-of-the-art reviews, regular research and application specific papers, short notes, letters to Editor and sensors related books reviews as well as academic, practical and commercial information of interest to its readership. Because it is an open access, peer review international journal, papers rapidly published in *Sensors & Transducers Journal* will receive a very high publicity. The journal is published monthly as twelve issues per annual by International Frequency Association (IFSA). In additional, some special sponsored and conference issues published annually. *Sensors & Transducers Journal* is indexed and abstracted very quickly by Chemical Abstracts, IndexCopernicus Journals Master List, Open J-Gate, Google Scholar, etc.

Topics Covered

Contributions are invited on all aspects of research, development and application of the science and technology of sensors, transducers and sensor instrumentations. Topics include, but are not restricted to:

- Physical, chemical and biosensors;
- Digital, frequency, period, duty-cycle, time interval, PWM, pulse number output sensors and transducers;
- Theory, principles, effects, design, standardization and modeling;
- Smart sensors and systems;
- Sensor instrumentation;
- Virtual instruments;
- Sensors interfaces, buses and networks;
- Signal processing;
- Frequency (period, duty-cycle)-to-digital converters, ADC;
- Technologies and materials;
- Nanosensors;
- Microsystems;
- Applications.

Submission of papers

Articles should be written in English. Authors are invited to submit by e-mail editor@sensorsportal.com 8-14 pages article (including abstract, illustrations (color or grayscale), photos and references) in both: MS Word (doc) and Acrobat (pdf) formats. Detailed preparation instructions, paper example and template of manuscript are available from the journal's webpage: http://www.sensorsportal.com/HTML/DIGEST/Submition.htm Authors must follow the instructions strictly when submitting their manuscripts.