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ENERGETIC METHOD FOR NON LINEAR DYNAMIC CHARACTERIZATION OF MEMS SQUEEZE FILM DAMPING

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INTRODUCTION

Design of Micro Electro Mechanical Systems (MEMS) often includes the presence of many oscillating elements and components, which are usually represented by perforated plates suspended on elastic beams. Both elastic and dissipative effects of the fluid surrounding the structure are observed to be variable with the frequency of actuation [1]. This phenomenon is well known as ‘squeeze film damping’. Many analytical [2] and numerical [3] works have been presented about this topic. The aim of this work is to define an energetic approach for the experimental evaluation of dynamic parameters. The coupling behaviour of the mechanical versus electrostatic domain is linearized by the assumption of small oscillation amplitudes.

THE ENERGETIC METHOD

This method is well suited to be supported by experimental measurements in the time domain. The optical interferometric and the laser techniques can be used at this proposal. Damping and stiffness parameters are demonstrated to be evaluated from the hysteresis loop between the driving force and the displacement of the structure.

In fig. 1 is presented the scheme of a typical MEMS perforated damper plate [3]. In a model order reduction approach the structure can be simplified, for the purpose of the present work, to the single degree of freedom system (spring-mass-damper) where: m is the mass of the plate, c is the fluidic damping and k_{tot} is the sum of the fluidic (k_f) and the structural (k_s) stiffnesses. The driving electrostatic (AC) force is indicated as

$$F(t) = F_0 \cos \omega t \tag{1}$$

and the frequency of actuation is $f = \omega/2\pi$.

The resulting second order differential governing equation is

$$m\ddot{z} + c(\omega, h)\dot{z} + \{k_f(\omega, h) + k_s\}z = F(t). \tag{2}$$

As indicated in the eq. (2), both dynamic coefficients c and k_f generated by the fluid presence are dependent from the frequency of actuation and from the gap thickness (h) extension; this represents a double source of non-linearity of the system produced by the structural-fluidic interaction. This evidence is represented in the figs. 2a and 2b, where the fluidic damping and stiffness are estimated through a harmonic FE analysis at different values of ω and g.

The solution of the eq. (2) can be written as

$$z(t) = z_0 \cos(\omega t - \varphi) \tag{3}$$

where $z_0 = F_0 \Big/ \sqrt{\left(k_{tot} - m\omega^2\right)^2 + \omega^2 c^2}$ and $\varphi = \tan^{-1}\left\{c\omega \big/ \left(k_{tot} - m\omega^2\right)\right\}$.

By plotting the values of $F(t)$ and $z(t)$ from eqs. (1) and (3) it is possible to trace the hysteresis loop of fig. 3 [4]. Some fundamental quantities can be easily calculated as indicated in the figure. The value of k_s can be experimentally measured with a static load as the ratio between the static force F_{st} and the corresponding displacement z_{st}. The energy dissipated per cycle of vibration due to the fluidic damping can be calculated by integrating the product of the force and the displacement over one cycle, obtaining

$$E = \oint F \cdot dz = \int_0^{2\pi/\omega} F\dot{z} \cdot dt = \pi c \omega z_0^2 \,. \qquad (4)$$

APPLICATION ON A MEMS DAMPER

As an example, the hysteresis loop of the structure reported in fig. 1 with a = 40μm, s_0 = 1μm, $s_1 = s_2$, h_c = 2μm, h = 1μm and number of holes N = 64 is plotted for three different values of the frequency of actuation (1, 10 and 100kHz) by means of a FE harmonic analysis. The harmonic analysis is performed by imposing a uniform velocity to the structure; a structural stiffness of the elastic beams k_s = 0.294N/m is assumed. The values of damping and stiffness calculated from the hysteresis loops are reported in the tab. I as c_{loop} and $k_{tot,loop}$. In the same table the corresponding values calculated by the FE fluid pressure force acting on the lower surface of the damper (c_{FEM} and $k_{f,FEM}$) are reported. The damping values result to be identical, as the stiffness values, if the structural contribution k_s is added to the FE result.

CONCLUSIONS

The presented method can be used to extract damping and stiffness parameters from experimental measurements on MEMS devices in the time domain and trace as result the non-linear dependency of the 'squeeze film damping' parameter in the frequency range; driving force and displacement histories can be used to trace the corresponding hysteresis loop.

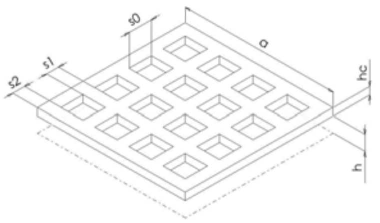

Fig. 1 General shape of a MEMS damper

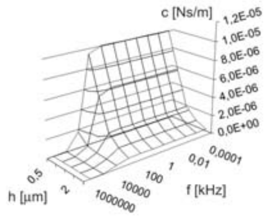

Fig. 2a Damping versus gap and frequency variation

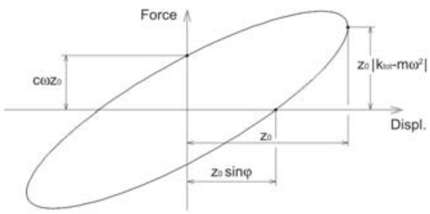

Fig. 3 Force-displacement hysteresis loop

frequency [kHz]	c_{loop} [Ns/m]	$k_{tot,loop}$ [N/m]
1	$6.12 \cdot 10^{-6}$	$2.94 \cdot 10^{-1}$
10	$6.12 \cdot 10^{-6}$	$2.95 \cdot 10^{-1}$
100	$6.11 \cdot 10^{-6}$	$4.18 \cdot 10^{-1}$
frequency [kHz]	c_{FEM} [Ns/m]	$k_{f,FEM}$ [N/m]
1	$6.12 \cdot 10^{-6}$	$1.24 \cdot 10^{-5}$
10	$6.12 \cdot 10^{-6}$	$1.24 \cdot 10^{-3}$
100	$6.11 \cdot 10^{-6}$	$1.24 \cdot 10^{-1}$

Tab. I Damping and stiffness evaluated by histeresis loop and FE pressure distribution

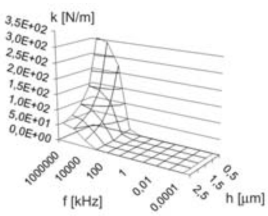

Fig. 2b Stiffness versus gap and frequency variation

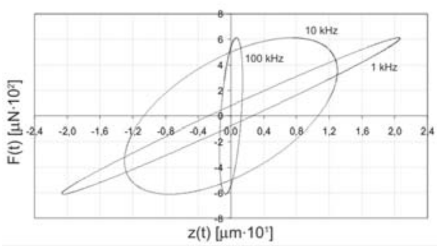

Fig. 4 Hysteresis loop of a MEMS damper at different frequencies obtained from FE model

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