Squeeze film damping in CMUTs with air-filled cavities

A. T. Galisultanov, P. Le Moal, V. Walter, G. Bourbon
Department of Applied Mechanics, FEMTO-ST Institute, Besançon, France
ayrat.galisultanov@femto-st.fr

Most of CMUTs applications require high bias voltage in order to improve either the sensitivity of the receiver mode, or the radiated acoustic pressure of the transmitter mode. Such high bias voltage strongly affects the dynamic characteristics of CMUTs in terms of the resonant frequency and the bandwidth.

The behavior of gas, contained between couple of vibrating parallel plates, is defined by frequency of vibrations [1]. In the case of low frequency oscillation plates moves with low velocity and the gas has time to "leak" out from the gap. For this case viscous damping force dominate. Opposite, for high frequency oscillation, the gas film mainly compressed and has no time to "leak". In this case the elastic force dominates and air film acts as a spring. A solution for circular plates in parallel motion was given by Blech [2]. He used simplified boundary conditions and vanished acoustical pressure at the edge of the moving plates. The dimensionless viscous damping $f_d(\sigma)$ and elastic $f_e(\sigma)$ forces are:

$$f_d(\sigma) = \left(\sqrt{\frac{2}{\sigma}}\frac{ber\sqrt{\sigma}\left(ber_1\sqrt{\sigma}+bei_1\sqrt{\sigma}\right)+bei\sqrt{\sigma}\left(bei_1\sqrt{\sigma}-ber_1\sqrt{\sigma}\right)}{\left(ber\sqrt{\sigma}\right)^2+\left(bei\sqrt{\sigma}\right)^2}\right) \quad (1)$$

$$f_e(\sigma) = \left(1-\sqrt{\frac{2}{\sigma}}\frac{ber\sqrt{\sigma}\left(bei_1\sqrt{\sigma}-ber_1\sqrt{\sigma}\right)-bei\sqrt{\sigma}\left(ber_1\sqrt{\sigma}+bei_1\sqrt{\sigma}\right)}{\left(ber\sqrt{\sigma}\right)^2+\left(bei\sqrt{\sigma}\right)^2}\right) \quad (2)$$

where $\sigma = \frac{12\mu\omega a^2}{P_a g^2}$ is the squeeze number, μ is the viscosity of the air, ω is radial frequency of vibration, a is the radius of circular plate, P_a is atmospheric pressure and g is air-filled cavity thickness.

Darling et al. [3] used Green's function method for calculation the effects of compressible squeeze film damping with different venting conditions. It was shown, that decreasing of the venting perimeter length for rectangular structure decreased critical squeeze number σ_c, for this value of squeeze number viscous damping and elastic force are equal. Moreover in the same work they demonstrated that squeeze film damping equal to zero for sealed structure with parallel plates approximation. The same assumption was used in [4] for flexible membrane.

Our work deals with the study of the resonant frequency and the squeeze film damping of CMUTs with air-filled cavities. In particular, the "stiffening" effect of the squeeze air film in the back cavity and the electrostatic "softening" effect lead to contradictory influences on the resonant frequency. A simplified analytical approach was developed with parallel plate approximation for the electromechanical behavior and the squeeze film effects for $\sigma \gg \sigma_c$. Finally, an experimental dynamic characterization was conducted on fabricated CMUTs (from 60μm to 160μm radius) with an anodic bonding technology. The evolution of the resonant frequency with respect to the bias voltage for different ratio squeeze film stiffness constant (k_e)/mechanical stiffness constant (k_0) is shown in Figure 1. The noteworthy point is a given stiffness constant ratio leading to stable resonance frequency with the bias voltage.

Also we used Navier-Stokes equation within COMSOL Thermoacoustic interface to calculate the dimensionless viscous damping $f_d(\sigma)$ and elastic $f_e(\sigma)$ forces. The thickness of air-filled sealed cavity varied from 1 to 50 μm and rarefaction effect did not take into account. The radius and thickness of circular plate equal to 150 and 2.3 μm respectively. The top of the movable membrane was not loaded by any fluid or gas, so radiation loss was equal to zero. The difference between resonant frequencies of the plate vibrating over a uniform thin air gap ω and in vacuum ω_0 determined by squeeze film stiffness constant. We coud defined dimensionless elastic force constant as:

$$f_e(\sigma) = \frac{k_e}{P_a S/g} = \frac{m(\omega^2 - \omega_0^2)}{P_a S/g}. \quad (3)$$

At the same time the dimensionless viscous damping forces $f_d(\sigma)$ could be defined as:

$$f_d(\sigma) = \frac{\Delta\omega}{\omega} \cdot \frac{m\omega^2}{P_a S/g}. \quad (4)$$

The analytical (1-2) and FEM calculated (3-4) dependencies of dimensionless viscous damping and elastic force on squeeze number for circular membrane are presented in Fig. 2.

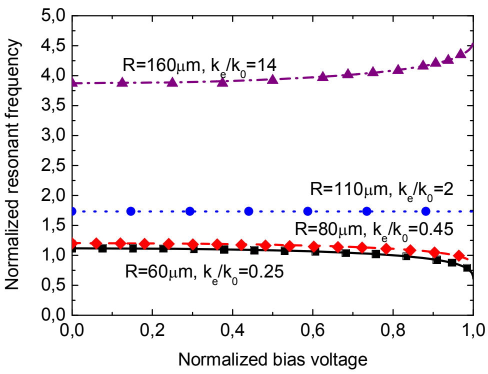

Fig. 1. Analytical (-) and experimental (*) dependencies of normalized resonant frequency on the normalized bias voltage

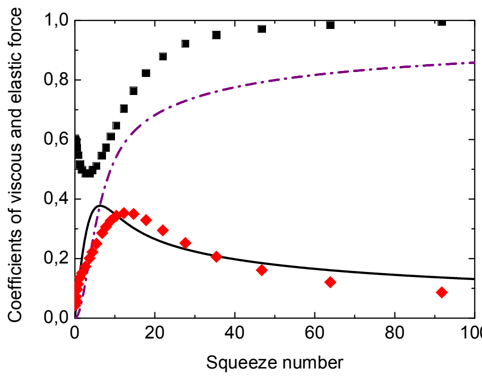

Fig. 2. Analytical (-) and FEM (*) calculated dependencies of viscous and elastic force on squeeze number for circular membrane

The analysis of FEM calculated data showed that for flexible circular membrane with $\sigma \gg \sigma_c$ the squeeze film stiffness constant is equal to $k_e = P_a S/g$. We observed that for this type of sealed system viscous damping force was greater than zero and the behavior of viscous damping and elastic force was the same as for the simple piston model with open boundary condition. The loss originated from non-uniform amplitude of deflection of vibrating membrane. This nonuniformity cause gas lateral motion and consequently viscous and thermal loss.

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References

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