VISCOUS DAMPING OF LATERAL MEMS DEVICE WITHIN A FULLY-ENCLOSED CAVITY

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ABSTRACT

A new viscous damping model has been proposed for analyzing damping performance of a laterally moving MEMS device inside a fully-enclosed cavity. Trench Pressure Damping, identified as a new damping term and not included in previous models, becomes dominant in the total viscous damping in a MEMS device. The cavity height inside the enclosed package is a critical dimension in determining the damping performance. The analytical results from the new model agree well with the measured results.

INTRODUCTION

Fully enclosing MEMS devices has been common practice in many applications, including accelerometers, gyroscopes, and resonators, where Couette flow (Stokes) and squeeze-film damping models have been used extensively to estimate air damping. Several analytical models have been suggested applicable for MEMS devices with uncapped packages, where air is assumed free to move in and out to the outside environment as the device moves. Previous work by Yasumura et. al. [1] has shown that viscous damping in an enclosed MEMS device can be significantly altered from that observed in an open environment. Damping is enhanced inside the chamber by a pressure gradient formed during device movement that changes the dominant dissipative mechanism from the Stokes or Couette flow of slide-film damping [2, 3] to Poiseuille flow damping. Evidence in our study shows that the pressure difference between two sides of the device resulting from forced air movement inside the cavity adds another damping term not included by Yasumura. A new analytical damping model has been proposed for analyzing viscous damping performance of a laterally moving MEMS device inside a fully-enclosed cavity. The analytical results agree well with the measured results. This new term plays a significant role in the damping of a MEMS device under test and is the dominant term compared to either Stokes flow damping or pressure driven damping enhancement.

DEVICE

Figure 1a shows a schematic section view of the device in this study. The moving component of the device (*TRANSLATOR*) is sandwiched between a *CAP WAFER* and *STATOR WAFER* which form a fully enclosing cavity. The dimensions of the *TRANSLATOR* and enclosing cavity are shown in Figure 1b. The *TRANSLATOR* has a height (*h*) ~100μm, width (*W*) and length (*L*) ~1800μm. Typical gap dimensions were ~2 μm for the bottom cavity gap, *g₂*, and ~30 μm for the upper cavity gap, *g₁*. The width of side trenches, *W_t*, was about 20 μm. The pressure inside the device cavity could be adjusted and monitored using the variable pressure system shown in Figure 2. A venting hole was introduced that connected the internal device cavity to the external volume of the 68-pin CLCC package whose pressure could be controlled between 10⁻⁴ Torr and 1 atm (760 Torr).

THEORY

Classic Damping Theory

According to traditional damping theory, without an enclosing top cover, sliding film damping between the *TRANSLATOR* bottom surface and the fixed cavity floor should be a dominant

damping source, since the device cavity has a small bottom gap, *g₂*, in microns, while other cavity dimensions were in 10's of microns. Considering damping from the upper gap, *g₁*, the total sliding film shear force is:

$$\tau_{s,g_1,g_2} = \frac{\mu u_0}{g_1} + \frac{\mu u_0}{g_2} = \frac{\mu u_0 (g_1 + g_2)}{g_1 g_2} \quad (1)$$

where μ is air viscosity and u_0 is the velocity of the *TRANSLATOR* when moving.

Squeeze-film damping losses in the side trenches due to vertical flow out of the trenches may be included using a model from Veijola et. al., which includes border flow effects [4]. Since the length and width of the proof mass are much larger than its height, a one-dimensional model may be employed. The total squeeze length, will include the twice the proof mass Length, *L*. The squeeze-film damping coefficient is given by:

$$b_{sq} = \frac{\mu_{eff,trench} L (2h + 1.3W_t)^3}{W_t^3} \quad (2)$$

Figure 1 Schematics of the device package and damping physical model, (a) Device model, (b) Damping model. The moving structure has a height of *h*, width and length of *L*. The bottom gap is *g₂* and the upper gap is *g₁*. The side trenches between the device sidewall and the package have width of *W_t*. Lateral resonant frequency, *f_r*, of the device was ~1100 Hz.

Yasumura's Correction

Yasumura's work reported that the damping inside an enclosed volume was enhanced due to the forced air movement inside the cavity [1]. The air flow physics in the bottom and top

gaps is changed from Couette flow to Poiseuille flow and an extra damping term (Shear Pressure) must be added in this case. Yasumura et. al. assumed incompressible flow such that the full displaced volume resulting from lateral motion was assumed to flow through the upper and lower gaps from one side of the rotor to the other [1]. The $-1/\rho(dp/dx)$ term can no longer be neglected and the pressure driven velocity is then given by [1]:

$$u(y, g)_p = -\frac{1}{2\mu} \frac{dP}{dx} (gy - y^2) \quad (3)$$

where g is the gap (g_1 or g_2) through which air flows and dP/dx is the induced gradient. Pressure driven shear force is then:

$$\tau(y, t)_p = -\mu \frac{\partial}{\partial x} u(y, t) \Big|_{y=0} = -\frac{1}{2} \frac{dP}{dx} (g - 2y) \Big|_{y=0} \quad (4)$$

By equating mass flows as:

$$\rho h w u_0 = \int_0^w \int_0^{g_1} \rho u(y, g_1) dy dw + \int_0^w \int_0^{g_2} \rho u(y, g_2) dy dw \quad (5)$$

And combining Equations 3 and 5, one can solve for dP/dx in terms of h , g_1 , g_2 and u_0 .

Rarefaction effect correction

In order to investigate pressure dependent damping, correction for rarefaction effects must be included. A typical method, in this case for lateral motion, assumes an effective viscosity that depends on the Knudson number, K_n , for each gap:

$$\mu_{eff,1} = \frac{\mu}{(1 + 2K_{n,1} + 0.2K_{n,1}^{0.788} \exp(-\frac{K_{n,1}}{10}))} \quad (6)$$

where $K_{n,1}$ is the Knudson number for gap 1, $K_{n,1} = \lambda/g_1$, and λ is the gas mean-free-path [2]. An alternative effective viscosity that was formulated for “squeeze” gaps subject to Poiseuille flow is given by (for gap 1) [5]:

$$\mu_{eff,1} = \frac{\mu}{(1 + 9.638K_{n,1}^{1.159})} \quad (7)$$

Since pressure-induced flow in upper and lower gaps is expected to be Poiseuille-like, Equation 7 is considered more applicable.

Combining Equations 3 and 5 and solving for dP/dx , we have:

$$\frac{dP}{dx} = \frac{12hu_0}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} \quad (8)$$

The total pressure-driven shear force is:

$$\tau_{p,g_1,g_2,V} = \frac{6hu_0(g_1 + g_2)}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} \quad (9)$$

Correction to the Stokes shear force would take the form:

$$\tau_{s,g_1,g_2,V} = \frac{\mu_{eff,1}u_0}{g_1} + \frac{\mu_{eff,2}u_0}{g_2} \quad (10)$$

Using the Veijola rarefaction correction, the total shear force is assumed to be the sum of Stokes and Pressure terms as:

$$\tau_{total,V} = \tau_{p,g_1,g_2,V} + \tau_{s,g_1,g_2,V} = \frac{6hu_0(g_1 + g_2)}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} + \frac{\mu_{eff,1}u_0}{g_1} + \frac{\mu_{eff,2}u_0}{g_2} \quad (11)$$

Trench Pressure Damping

In addition to the shear forces in top and bottom gaps, an additional pressure term resulting from a pressure difference between trenches must be included. This term was not included in Yasumura’s formulation, but its necessity was made clear by FEA results. The resulting trench pressure term is:

$$F_T = (P_{T1} - P_{T2})A_T = \frac{dP}{dx} L(h) = \frac{12L^2h^2u_0}{\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}} \quad (12)$$

where F_T is the net force acting between trenches, P_{T1} and P_{T2} are trench pressures, A_T is the area of the Translator ends, and dP/dx is from Equation 8. The damping coefficient term is given by:

$$b_{trench,V} = \frac{12L^2h^2}{\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}} \quad (13)$$

Using Veijola rarefaction corrections (Equations 6 and 7), the corresponding damping coefficient are then:

$$b_{T,V} = \frac{6Ah(g_1 + g_2) + 12L^2h^2}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} + \frac{\mu_{eff,1}A}{g_1} + \frac{\mu_{eff,2}A}{g_2} \quad (14)$$

Note:

$$b = \frac{\tau A}{u_0} \quad (15)$$

where A is the area of the translator bottom surface in contact with the air gap.

Device Damping and Quality Factor

Using the Veijola rarefaction corrections, the total damping coefficient combines Equations 2 and 14 as:

$$b = \frac{6Ah(g_1 + g_2) + 12L^2h^2}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} + \frac{\mu_{eff,1}A}{g_1} + \frac{\mu_{eff,2}A}{g_2} + \frac{\mu_{eff,trench}L(2h + 1.3W_t)^3}{W_t^3} \quad (16)$$

For the micro-mover with a square geometry, the total damping coefficient can also be written as:

$$b = \frac{6Ah(g_1 + g_2 + 2h)}{\left(\frac{g_1^3}{\mu_{eff,1}} + \frac{g_2^3}{\mu_{eff,2}}\right)} + \frac{\mu_{eff,1}A}{g_1} + \frac{\mu_{eff,2}A}{g_2} + \frac{\mu_{eff,trench}L(2h + 1.3W_t)^3}{W_t^3} \quad (17)$$

The quality factor, Q , is then given by:

$$Q = \frac{m\omega}{b} \quad (18)$$

where b is the total damping coefficient, ω is the resonant angular frequency (radians/sec), and m is the moving mass.

TESTING SETUP AND RESULTS

A vacuum test system has been built and constructed to test the packaged device that has been vented by laser drilling into the device cavity over a pressure range from ~0.1 mTorr to 760 Torr. The schematics of the vacuum test setup are shown in Figure 2. The device under test (DUT) is mounted and wire bonded into a Kyocera ceramic open-top 68 pin CLCC package. The DUT is inserted into a CLCC socket on a special test board so that a vacuum housing can be clamped over the top making an O-ring seal to the face of the CLCC package, hence, forming a vacuum cavity with the package. Two ports allow this volume to be evacuated and the pressure sensed. Because the pressure gauges are upstream from the test volume, measured pressures should set an upper bound on the actual device pressure. The system has been designed to minimize pressure differences between gauge and device. Pressure control above base vacuum is accomplished using a combination of a pump throttling valve and a leak valve, both downstream from the test volume. By this means, we hope to insure minimal flow in the gauging arm of the test chamber and therefore minimal pressure difference between gauge and device.

In Figure 3, a typical measurement at a pressure of 1 mTorr is shown by physically tapping the package. The measurement shows that it takes more than 60 s for the proof mass to ring down (dissipate the energy). The quality factor, Q , is about 135,000. This is significant for a device with a resonance frequency of ~1 kHz.

Figure 2 Schematic of the vacuum test setup. A vacuum cavity is formed between a vacuum housing on top of a Kyocera ceramic open-top 68 pin CLCC package with an O-ring to seal the vacuum. The pressure can be adjusted from 10⁻⁴ Torr to 1 atm.

A square voltage signal is input into the electrodes to move the proof mass laterally, so that the translator oscillates eventually decaying to a new equilibrium position. As the trailing edge of the pulse arrives, the electrostatic force is removed. Therefore the translator moves back to its original position by the restoring force of the spring, decaying exponentially as well. The decay rate of the resulting oscillations provides a measure of the quality factor, Q , and through Q , the damping coefficient of resistance can be found. Figure 4 shows a Q versus pressure measurement of a device packaged as described above.

Figure 3 X-axis ringdown curve for the Translator at 1 mTorr. The observed Q is 135,000.

Analytical results from several models are shown in Figure 4 as well. Both the results from the traditional Couette flow damping and pressure-enhanced-damping from Yasumura's work underestimated the damping. The dash-dot curve in Figure 4 shows the analytical result after adding the new damping term, Trench Pressure Damping, which agrees well with measured results. The deviation from theory at low pressure (i.e., < 10 mTorr) indicates that other damping phenomena, likely thermo-elastic damping [6, 7], would need to be included in this pressure regime.

ANALYSIS AND DISCUSSION

The encapsulation of the device in a sealed chamber has changed the damping physics from seemingly Couette flow dominant damping to pressure driven dominant damping. At 1 atm, the Quality factor of the device is about 166 from Couette flow damping theory and 70 from Yasumura's pressure-enhanced-damping theory, compared to 14 in actual device. The encapsulation package dramatically enhanced air damping in the device. As the device moves, a pressure difference is formed between trenches from both sides of the device in the moving axis. This pressure difference not only increases damping due to the pressure driven flow in the upper and lower gap, but the existence of the pressure different in the trenches is also an obvious damping source for the device.

According to the Equation 17, air damping coefficient is a function of the depth of the cavity in the cap wafer. Therefore by varying the depth, air damping of the device can be tuned. Increasing the cavity depth causes the corresponding pressure drop to decrease. Figure 5 shows the calculated damping coefficient and quality factor dependence on the cavity depth from Equation 17 at 1 atm (760 Torr). Figure 5 also shows three testing results of damping coefficient at cavity gap of 33, 41, and 99 μm , which agree with the analytical results.

In an enclosed volume, the total height of the device cavity becomes a critical dimension controlling viscous damping. Utilizing this design parameter would allow individual damping adjustment of multiple devices which share the same cavity volume (and hence, cavity pressure). Quality factor (Q) can be improved by increasing the cavity, until it "saturates" at certain depth, where further increase has little effect on damping. In this case, the damping of the device is governed by traditional damping theory.

Figure 4 Damping Coefficient (b) and Quality Factor (Q) versus pressure of a MEMS device including Trench Pressure Damping. Also shown are Damping Models based on Couette Flow damping (top, dash curve) and on Couette+Shear Pressure (solid curve). Device Parameters: Resonant Frequency (f_r) $\sim$ 1100 Hz, Translator Mass (m) = 780 μ g, Length (L) = 1800 μ m, Width (W) = 1800 μ m, Height (h) = 100 μ m.

CONCLUSION

Air damping of a MEMS sensor or actuator inside an encapsulated chamber can be quite different from uncapped package. Traditional Couette flow and squeeze film damping theories are no longer enough to accurately characterize the damping. Yasumura improves the theory by including a pressure enhanced damping term, but still overestimates the Quality factor of the device under test. In this work, pressure buildup in the side trenches of the translator as it moves was observed and proved to form an extra drag to the movement of the proof mass. Trench Pressure Damping has been identified as a new and dominant damping term, with analytical results that agree well with the experimental results in the devices in this work. The device damping in an encapsulated package has a strong dependence on the cavity depth in the cap-wafer and can therefore be tuned. An accurate analytical damping model has great utility in the design and optimization of lateral resonator-based MEMS sensors allowing for improved performance of accelerometers [8], gyroscopes, and mass sensors.

Figure 5 Damping Coefficient (b) and Quality Factor (Q) versus Upper Cavity Gap, g_1 , for the MEMS device in this study at one atmosphere. All damping terms in Equation 17 are included. Solid triangles show the testing results of damping coefficient at cavity gap of 33, 41, and 99 μ m respectively.

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