PRESSURE ENHANCED AIR DAMPING IN ENCLOSED Laterally OSCILLATING MICROSTRUCTURES

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ABSTRACT

High aspect ratio MEMS devices have been enabled by the development of deep reactive ion etchers. These devices have found use in lateral accelerometers [1], microengines [2], and optical applications [3]. While resonant sensors or scanners may require high- Q operation, closed-loop control of actuators is improved by having the device close to critically damped. Analysis and modeling of the damping for laterally resonant devices have traditionally focused on a Couette or Stokes-based fluid model [4], [5]. For DRIE devices of typical geometry, the damping from these mechanisms may result in a Q of >50 , perhaps suitable for a resonant device, but inadequate for damping a closed-loop actuator. We have developed a technique that uses a cap over the device producing a small air gap between the top and bottom surfaces of the moving part of the actuator to act as a fluid resistor. When air is 'pumped' through this gap by lateral motion of the device, the flow resistance of this air channel provides the needed damping for the system.

INTRODUCTION

One measure of an actuator's performance is its response to externally applied shocks and vibrations. This is especially important for actuators used for optical applications. Sub-micron motion of the actuator or optical element can produce large changes in the optical performance. One method to reduce an actuator's sensitivity to external accelerations is to damp the modes of motion. While closed loop control of the actuator motion is often used, passive damping techniques can also be used to obtain significant improvement in device performance. By damping the motion of the actuator, the response to resonant excitation can be reduced as the Q of the actuator.

For actuators fabricated from single-crystal silicon we can ignore intrinsic sources of mechanical dissipation and focus instead on the external damping caused by the surrounding fluid medium (air) [6]. Figure 1 shows a photograph of the DRIE actuator used in this work. It features two banks of comb drives connected by two orthogonal beams. These beams are attached to two flexures that form the attachment to the surrounding substrate wafer. Details of the fabrication process can be found in previous work [7].

The actuator shown in Fig. 1 will have a number of sources of fluid damping. These sources can be generalized into three terms: damping beneath the motor, damping above the motor, and damping between the comb fingers. Assuming these sources of mechanical energy loss are independent, the total dissipation can be expressed as

$$\frac{1}{Q_{total}} = \sum_i \frac{1}{Q_i} = \frac{1}{Q_{bottom}} + \frac{1}{Q_{top}} + \frac{1}{Q_{comb}}. \quad (1)$$

Depending upon the specific actuator dimensions and geometry, each of the terms in Eq. 1 will contribute different amounts to the mechanical damping in the system. The usual method of quantifying

each of the damping terms is to use either a Couette or Stokes-based flow model between the actuator and its nearby surfaces via the intermediary fluid medium. In most cases the intermediary fluid is air. However, some applications may take advantage of the increased viscosity and dielectric constant of other fluids [8]. In this work, when we discuss fluids, it will be assumed to be air. The equations, however, are general and can be applied to systems involving more viscous materials. Because this work focuses on laterally oscillating structures that have full-deflection ranges of 10's of μm , contributions from squeeze film damping will not contribute large amounts to the system dissipation [9].

Actuator encapsulation is often used as a means to seal an actuator in vacuum for improved quality factor or for improved packaging and handling. In this work we demonstrate that placing a cap over the motor can have an additional damping benefit. Figure 2 shows a diagram of an actuator with a flow-restricting cap placed above it. Without the cap, air is free to move into and out of the motor surroundings as it oscillates. With the cap, however, air is now blocked from entering and leaving the motor area. If the motor has a large enough cross sectional area in the direction that it oscillates, air will now be forced to flow above (between the motor and cap) and beneath the motor (between the motor and underetch cavity). The motor will now act like a piston driving air across its surface. This piston-like effect is what will produce the damping enhancement.

COUETTE AND STOKES-BASED AIR DAMPING

For the actuator shown in Figs. 1 & 2 air flow between the motor and substrate wafer (or cap) can be described using the one dimensional Navier-Stokes equation. Taking the surrounding air to be an incompressible fluid without pressure gradients, the Navier-Stokes equation is reduced to the one dimensional diffusion equation given by

$$\frac{\partial u(y, t)}{\partial t} = \frac{\mu}{\rho} \frac{\partial^2 u(y, t)}{\partial y^2} \quad (2)$$

where $u(y, t)$ is the flow velocity field, μ is the fluid absolute viscosity, and ρ is the fluid mass density. The fluid velocity flow field, $u(y, t)$, is a function of both the distance between the moving motor

Fig. 1. Photograph of one of the electrostatic comb drive actuators used in this work. This actuator features two sets of comb drives anchored to the substrate with flexure beams. The actuator is 85 μm thick.

Fig. 2. Diagram of a capped micromachined motor. The motor has a height h , width w (into the paper), and moves with a velocity amplitude of u_0 . The gap above the motor between the motor and cap is g_1 and the gap between the motor and underetch cavity is g_2 .

and stationary surface and of time. The motor is assumed to move with small amplitude motion that varies as $\sin(\omega t)$. Typical Reynolds numbers, Re , of ~ 0.15 are obtained for our actuators ensuring that the fluid flow will be laminar.

For Couette-based flow, the left hand term in Eq. 2 is taken to be 0 (slowly varying flow) such that the velocity flow field is a linear function of the gap between the motor and stationary surface. However, a more general solution takes into account that the fluid has finite mass and thus will not respond instantaneously to the motion of the actuator. This produces the Stokes-based flow condition described by Cho for a laterally oscillating surface micromachined electrostatic actuator [5]. We can therefore consider Couette flow to be a limiting case of Stokes flow. For the actuators used in this work and those of Tang and Cho [4], [5], it will be shown that there is little difference between the damping forces generated by the two flow conditions.

Whether, the fluid flow is assumed to be Couette or Stokes-like, the quality factor is defined as

$$Q = \frac{2\pi W}{\Delta W} \quad (3)$$

where W is the total energy stored in the oscillator and ΔW is the energy lost per radian. The energy lost is equal to AD where A is the surface area over which the dissipation occurs and D is the energy loss per radian per unit surface area. This dissipation is caused by the fluid shear force on the actuator. Thus, the amount of mechanical dissipation is dependent upon the total shear force on the actuator as it moves in the fluid medium. More specifically, the dissipation per cycle per unit area is given by

$$D = - \int_0^{2\pi/\omega} \tau(y, t) u(y, t) dt \quad (4)$$

where the shear force for a given flow-field is

$$\tau(y, t) = -\mu \left. \frac{\partial u(y, t)}{\partial y} \right|_{y=0} \quad (5)$$

Solving for the quality factor of an oscillator with either Couette or Stokes-based flow produces a Q of the same form. For both flow conditions, the quality factor is given by

$$Q(g_{eff}) = \frac{g_{eff}}{\mu A} \sqrt{km} \quad (6)$$

where g_{eff} is the effective gap between the moving plates, A is the plate overlap area, k is the actuator stiffness, and m is the motional

mass of the actuator. For actuators moving in air, the viscosity, μ , is 180 uP. For Couette-flow the effective gap is simply the distance, d , between the moving motor and stationary plate. Thus, for Couette-based air flow $g_{eff,c} = d$. For Stokes-flow, however, there are two possible effective gaps. For air flow between the moving motor and a stationary surface, the effective gap is given by

$$g_{eff,s} = \frac{1}{\beta} \left(\frac{\sinh 2\beta d + \sin 2\beta d}{\cosh 2\beta d - \cos 2\beta d} \right)^{-1} \quad (7)$$

where $\beta = \sqrt{\rho\omega/2\mu}$, and d is the physical separation between the two surfaces. From this equation for the effective Stokes gap we can identify $\delta = 1/\beta$ as a characteristic length scale for Stokes-based flow [5]. For air flow above an uncapped motor the effective gap is simply the characteristic length scale δ . Therefore, the effective gap above the motor is given by $g_{eff,\infty} = \delta$. For an actuator with a $5 \mu m$ gap operating at 1 kHz, the effective gap ratio, $d/g_{eff,s}$, is 1.00023. This shows that for the actuators we typically use, there is little difference between the Couette and Stokes-based flow models. Therefore, for the remainder of this work, we will use the real physical gap distance when discussing motor gaps.

Eqs. 1 & 6 now provide a means to calculate quality factors in laterally oscillating actuators. The results of these calculations are shown in Table I for two motor styles. Both motors have the same comb drive region as shown in Fig. 1. They do, however, differ in the additional actuator elements attached to the comb drive. This, along with process variation, results in two motors with different damping surface areas, masses, spring constants, and resonance frequencies. These two motors also differ in the motor gap beneath the moving actuator. Motor 1 has a $15 \mu m$ gap while Motor 2 has a $5 \mu m$ gap. For Motor 1 the measured Q of 10.2 is much lower than the Stokes-based calculation of 44.9. Motor 2's measured Q is also much lower than the Stokes-calculated value of 23.8. In order to explain the measured Q 's we introduce the pressure-driven flow model for laterally oscillating micromachined actuators.

PRESSURE ENHANCED DAMPING

For most micromachined actuators, it is assumed that there are no pressure gradients acting upon the actuator. This includes external pressure gradients or gradients produced by the motion of the actuator device. With this assumption, the Navier-Stokes equation becomes the diffusion equation (Eq. 2). For our high aspect ratio structures, however (tall with respect to the gap distances), we may not be able to neglect the $-1/\rho(dp/dx)$ term in the Navier-Stokes equation.

In Fig. 2 a motor of height h moves with velocity amplitude u_0 . This actuator has a cap placed upon it that restricts air flow around the actuator. Without the cap, air would be free to enter and leave the regions surrounding the actuator as it oscillates. With the cap, however, the surrounding air must now be displaced. Because the fluid flow around our actuators is treated as incompressible, the displaced air must travel across the top and bottom of the capped actuator to the region on the left. This piston-like effect produces a combination of pressure driven and Stokes-based flow across the surface of the actuator with the result that the total shear damping force is increased depending upon the geometry of the actuator and gaps above and below the motor. With these assumptions, let us now calculate the amount of expected damping force enhancement due to the pressure-driven flow effect.

First, the Stokes shear force on a plate moving with velocity amplitude u_0 will be calculated. For a capped actuator, the total Stokes shear force acting on it will be the sum of the shear force

Motor Parameter	Motor 1	Motor 2
Motional mass m	8.22×10^{-8} kg	5.26×10^{-8} kg
Spring constant k	1.0 N/m	1.5 N/m
Resonance frequency f_0	555 Hz	850 Hz
Cap gap g_{top}	5 μm	5 μm
Motor gap g_{bottom}	15 μm	5 μm
$\Delta(h, g_1, g_2)$	11.9	52.0
Stokes Q	44.9	23.8
Pressure Q	9.7	1.5
Measured Q	10.2	1.4

TABLE I

TABLE OF MOTOR PARAMETERS FOR THE TWO STYLES OF MOTORS PRESENTED IN THIS WORK. THE COMB DRIVE REGIONS WERE IDENTICAL, HOWEVER, ADDITIONAL STRUCTURES WERE USED TO CHANGE THE RESONANCE FREQUENCY AND AMOUNT OF MOTIONAL MASS. MOTOR 2'S MOTOR GAP, g_{bottom} , WAS FABRICATED AT 5 μm TO VERIFY THAT THE Q WOULD FOLLOW THE PRESSURE ENHANCED AIR-DAMPING THEORY.

above and below the plate. With a gap g_1 above the motor and a gap g_2 below the motor, the total shear force is

$$\tau_{s,g_1,g_2} = \frac{\mu u_0}{g_1} + \frac{\mu u_0}{g_2} = \frac{\mu u_0(g_1 + g_2)}{g_1 g_2}. \quad (8)$$

For pressure driven flow, the velocity field is given by

$$u(y, g)_p = -\frac{1}{2\mu} \frac{dp}{dx} (gy - y^2) \quad (9)$$

where g is the gap through which the air flows (g_1 or g_2) and dp/dx is the pressure gradient formed across the motor as it oscillates. From Eq. 5, the pressure driven shear force on one side of the motor will be

$$\tau_{p,g} = -\frac{1}{2} \frac{dp}{dx} (g - 2y) \Big|_{y=0}. \quad (10)$$

We therefore need to determine the pressure gradient as a function of actuator and gap dimensions in order to calculate the pressure-driven shear force on the moving actuator. The simplest method to determine the pressure gradient for an incompressible fluid is to use a mass-flow calculation to equate the amount of displaced fluid to the fluid flow across the gaps above and below the motor. This flow will be considered to be one dimensional and will ignore edge effects even though the motor has a finite physical depth of w . This mass flow equality can be written as

$$\rho h w u_0 = \int_0^w \int_0^{g_1} \rho u(y, g_1) dy dw + \int_0^w \int_0^{g_2} \rho u(y, g_2) dy dw \quad (11)$$

where on the left side we have the displaced fluid due to the actuator of height h and width w . The right side of Eq. 11 shows the sum of the fluid flow above and below the motor. Using Eq. 9 in Eq. 11 and solving for the pressure gradient gives

$$\frac{dp}{dx} = -\frac{12\mu h u_0}{g_1^3 + g_2^3}. \quad (12)$$

From this expression for the pressure gradient, the total pressure-driven (Poiseuille) shear force can be calculated as the sum of the shear force above and below the motor:

$$\tau_{p,g_1,g_2} = \tau_{p,g_1} + \tau_{p,g_2} = \frac{6h\mu u_0(g_1 + g_2)}{g_1^3 + g_2^3}. \quad (13)$$

We now have an expression for the shear force acting on the actuator due to pressure-driven flow across the motor surfaces. Eq. 13 can be reexpressed as

$$\tau_{p,g_1,g_2} = \kappa(h, g_1, g_2) \mu u_0 \quad (14)$$

where

$$\kappa(h, g_1, g_2) = \frac{6h(g_1 + g_2)}{g_1^3 + g_2^3}. \quad (15)$$

Eq. 15 gives an absolute magnitude for the pressure-based shear force in terms of the motor height, cap gap, motor gap, fluid viscosity, and actuator velocity. This is shown in Fig. 3 where $\kappa(h, g_1, g_2)\mu$ has been plotted for three different motor gap values of 5 μm , 10 μm , and 15 μm . For all three gap values, the motor height is 85 μm . Fig. 3 shows that as the motor gap is reduced, the amount of damping shear force enhancement increases. In addition, for a given motor gap distance, decreasing the cap gap will further increase the amount of damping force enhancement. There is a point, however, at which further reduction in the cap gap will actually reduce the amount of damping force enhancement. From Eq. 15 it can be shown that for a given motor gap, the maximum shear force is obtained when the corresponding cap gap is 1/2 the motor gap value (*i.e.*, when $g_1 = g_2/2$). This reduction in the damping shear force enhancement with further reduction of the cap gap is caused by the motor and cap gap-height asymmetry. This asymmetry produces two competing effects when calculating the shear force: a reduction of the gap heights increases the pressure drop, dp/dx , across the moving actuator while also reducing the total shear force, τ , for that pressure drop. Removing the broken gap symmetry (*i.e.*, forcing $g_1 = g_2 = g$), the κ function will no longer have a local maxima and is now proportional to $1/g^2$. This will produce a continuous increase in shear force as the gap distances are decreased simultaneously. This argument is valid as long as the fluid prefers to flow instead of compress. For a real system, there will be a point at which the shear-flow induced damping begins to be limited by fluid compression.

In order to obtain a reproducible amount of enhanced mechanical damping in a capped actuator, the motor and cap heights must be controlled. For a 5 μm motor underetch cavity depth, cap gaps within the range of 1.1 to 3.9 μm ($2.5 \pm 1.4 \mu m$) will produce at least 90% of the optimal damping value. This is a large acceptance range and allows for less stringent fabrication tolerances.

In order to use the pressure-based damping theory to make quality factor predictions, let us recast Eq. 13 into a different form. Using Eq. 8, Eq. 13 can be rewritten as

$$\tau_{p,g_1,g_2} = \frac{6h g_1 g_2}{g_1^3 + g_2^3} \tau_{s,g_1,g_2}. \quad (16)$$

where we are now expressing the total shear force as a multiple of the corresponding Stokes-based shear force. Because the total shear force acting on the motor can be approximated as the sum of the

Fig. 3. Plot of $\kappa(h, g_1, g_2)\mu$ for 85 μm -tall motors with motor gaps, g_2 , of 5, 10, and 15 μm . The cap gap, g_1 , is varied from 0 to 25 μm . For a given motor gap value g_2 , the maximum amount of damping shear force is obtained when $g_1 = g_2/2$.

Stokes and Poiseuille terms, we can now write the total shear force acting on the motor as

$$\tau_{t,g_1,g_2} = \tau_{s,g_1,g_2} + \tau_{p,g_1,g_2} = \Delta(h, g_1, g_2)\tau_{s,g_1,g_2} \quad (17)$$

where the damping enhancement function, $\Delta(h, g_1, g_2)$, is given by

$$\Delta(h, g_1, g_2) = \frac{6hg_1g_2}{g_1^3 + g_2^3} + 1 \quad (18)$$

A more accurate calculation will also take into account slip effects at the boundary walls. Using the Maxwell slip theory with an accommodation coefficient of 1, the $\kappa(h, g_1, g_2)$ function (Eq. 15) can be reexpressed as

$$\kappa_{slip}(h, g_1, g_2) = \frac{6h(g_1 + g_2)}{g_1^3 + g_2^3 + 6g_1^2\lambda + 6g_2^2\lambda} \quad (19)$$

producing the slip-adjusted Δ function

$$\Delta_{slip}(h, g_1, g_2) = \frac{6hg_1g_2}{g_1^3 + g_2^3 + 6g_1^2\lambda + 6g_2^2\lambda} + 1 \quad (20)$$

where λ is the mean free path of air (~ 69 nm at STP).

Figure 4 shows a plot of $\Delta(h, g_1, g_2)$ and $\Delta_{slip}(h, g_1, g_2)$ for 85 μm -tall actuators with motor gaps, g_2 , of 5, 10, and 15 μm . Note that at smaller motor gaps, the boundary slip term becomes more important. Figure 4 shows that for an 85 μm -tall actuator with a 5 μm motor gap, a damping correction factor (with respect to the equivalent Stokes-based shear force) of >50 can be obtained.

The Q calculations shown in Table I are in excellent agreement with the measured values. Motor 2's measured Q of 1.4 is in excellent agreement with the value calculated using the pressured enhanced damping theory. This shows that the pressure-driven flow model presented in this work can produce the large amount of additional damping required to obtain large amounts of Q reduction. For Motor design 2, with a 5 μm motor gap and a 5 μm cap gap, a damping correction factor of ~ 50 is obtained through the pressure enhanced damping theory. This amount of additional damping is difficult to produce by other passive means. Methods such as increasing the comb overlap area are not as space efficient as the pressure-flow method. The reason for this is that adding damping combs adds additional mass to the actuator and takes up more surface area on the die. To-date, thousands of actuators demonstrating enhanced pressured-based damping have been fabricated and shipped as part of a tunable laser or tunable filter product.

Fig. 4. Plot of $\Delta(h, g_1, g_2)$ (solid lines) and $\Delta_{slip}(h, g_1, g_2)$ (dotted lines) for motor gaps, g_2 , of 5, 10, and 15 μm . For each of these motor gap values, the cap gap, g_1 , is varied from 0 to 25 μm . For each of these cases the motor height, h , is 85 μm .

CONCLUSIONS

The development of DRIE actuators allows for the fabrication of actuators with surrounding sidewalls. When an enclosed cavity is formed through placement of a capping structure, pressure-enhanced damping occurs. This damping mechanism can be described using the Δ function that quantifies the amount of additional shear force on the actuator with respect to the equivalent Couette force. Additionally, the amount of damping can be increased by minimizing the gaps above and below the motor. For a given motor gap (under the motor), the damping can be optimized by selecting the cap gap to be $1/2$ of the motor gap. Finally, quality factors approaching unity have been obtained for laterally oscillating structures with motor and cap gaps of 5 μm .

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REFERENCES

- [1] A. Partridge, *et al.* A high-performance planar piezoresistive accelerometer. *J. Microelectromech. Syst.*, 9(1):58–66, 2000.
- [2] A.H. Epstein, *et al.* Power MEMS and Microengines. *Proc. IEEE Transducers '97*, pages 753–756, 1997.
- [3] H. Jerman and J.D. Grade. A mechanically-balanced, drier rotary actuator for a high-power tunable laser. *Proc. Solid-State Sensor, Actuator, and Microsystems Workshop, Hilton Head Island, South Carolina*, pages 7–10, 2002.
- [4] W.C. Tang, T.-C. H. Nguyen, and R.T. Howe. Laterally driven polysilicon resonant microstructures. *Proc. IEEE Workshop on Microelectromech. Syst., Salt Lake City, UT, Feb 1989*, pages 53–59, 1989.
- [5] Y.-H. Cho, A.P. Pisano, and R.T. Howe. Viscous damping model for laterally oscillating microstructures. *J. Microelectromech. Syst.*, 3(2):81–87, 1994.
- [6] K.Y. Yasumura, *et al.* Quality factors in micron- and submicron-thick cantilevers. *J. Microelectromech. Syst.*, 9(1):117–125, 2000.
- [7] J.D. Grade, H. Jerman, and T.W. Kenny. A large-deflection electrostatic actuator for optical switching applications. *Proc. Solid-State Sensor, Actuator, and Microsystems Workshop, Hilton Head Island, South Carolina*, pages 97–100, 2000.
- [8] K.Y. Yasumura, J.D. Grade, and H. Jerman. Fluid damping of an electrostatic actuator for optical switching applications. *Proc. Solid-State Sensor, Actuator, and Microsystems Workshop, Hilton Head Island, South Carolina*, pages 358–361, 2002.
- [9] P.W. Kwok, M.S. Weinberg, and K.S. Breuer. Fluid effects in vibrating micromachined structures. *J. Microelectromech. Syst.*, 14(4):770–781, 2005.