Thin-Film Thickness Influence on Anisotropic MEMS Damping

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Abstract: The analyzed MEMS oscillator is composed by a spring (beam) supported mass that needs to vibrate with a high Q-factor in the horizontal plane. Nevertheless, the transverse motion needs to be highly damped (low Q-factor). The purpose of the work was to explore different system configuration to determine the most suitable one for the scope of the device. Different parameters were varied to sweep multiple scenarios, namely thin-film gap thickness, ambient pressure and surface texture

Keywords: thin-film, MEMS, damping

1. Introduction

Micro-machined structure, such as MEMS, work in a significantly different environment with respect to larger size machine. As a result MEMS devices are strongly affected by the surrounding air. The air presents a counter reactive force on the moving elements of such devices. According to Yang [1] the damping effect of enveloping air would be enforced if a plate was oscillating close to another plate, so that the air film was squeezed in between the two surfaces. For such sizes, the latter mechanism is what dominates the frequency response of the system, thus it needs to be investigated in order to optimize the system to achieve desired requirements.

2. Model Definition

The described assignment question was tackled by implementing a model able to simulate the thin-film liquid/structure interaction. The model made us of 3D Solid-Mechanics Physics interface; it solved the squeezed film air/structure interaction using the Thin-Film Damping extension within the former domain.
The Thin-Film Damping is a boundary physics, and not a domain one, due to the relative size with respect to the solid structure. Moreover, the model used the zero-pressure edge condition of the thin-film, hence edge pressure was always equal to ambient one. This was selected as boundary condition to better comply with existing researches; this led to a deeper understanding of the author on such constraint and a wider pool of results to compare the model with.
The dynamic behavior of the system was analyzed by applying a body force $F = \rho a$, in which a was equal to half gravitational acceleration.
Several parameters were defined to better handle various analyzed scenarios. More specifically ambient pressure, thin-film gap height and body load magnitude were controlled and changed according to the current aim of the simulation (validation, parametric sweep, etc).

2.1 Geometry

The model consisted of one square proof-mass suspended by a thin cantilever beam. The cantilever beam was fixed at the end to the surrounding environment. The ending part of the assignment consisted in a surface texture optimization; due to lack of computing power a topology optimization could not be performed.
Nevertheless, after some thorough reasoning based on researches, an alternative was simulated to pave the way for the assessment of the best-performing surface texture.
Main geometric parameters are listed below in Tab. 1.

Side [µm]	200
Length [µm]	600
Beam Width [µm]	20
Thickness [µm]	10

Table 1: MEMS device dimension

2.2 Material and Loads

The solid domain was made of Silicon, whereas air was assigned to the boundary thin-film gap. To analyze the step response of the system a body load was applied as mentioned before. The volume force was equal to half the gravitational force, such as the total resultant force could be considered realistic. The body load was simply applied to all the solid structure using predefined density $F = \rho a$.
As it will be seen later, the whole work was focused in determining and investigating the effective damping generated by the interaction

between the structure and the squeeze-film. To do so several derived values were used, namely total displacement of the tip, fluid loads on wall and the damping coefficient.
Viscous effects were responsible for a counteracting force in phase with velocity. From the standard representation of a second order oscillator, the coefficient of viscous damping can be determined as follows:

$$c_d = -\frac{\int_A p_{fluid} dA}{\dot{z}}$$

Then, the damping ratio can be derived as:

$$\zeta = \frac{c_d}{2m\omega_0}$$

2.3 Optimization Algorithm

Nelder-Mead algorithm was chosen to perform the optimization analysis. According to COMSOL Multiphysics® literature [8], one of the most effective algorithm for constrained problems is Nelder-Mead. Indeed it quickly produces satisfactory results, yielding significant improvements in the first iterations.

2.3 Coupling Analysis

The coupling analysis consisted in a particular case of Fluid-Structure Interaction. The modified Reynolds equations were solved to work out the fluid load on surface. To perform such coupled study, Thin-Film Damping boundary condition within Solid Mechanics Physics was used.
The interface created a fictitious pair of wall and base, simulating the relative motion of two surfaces. The wall was set to be coincident with the solid bottom surface, both in position and velocity. This strategy coupled properly the motions and consequent force loads between oscillator and thin-film air gap. The problem is a fully-coupled one. Specifically, the bi-directional coupling extends to different domains (and different fields) and occurs at their interface.

3. Use of COMSOL Multiphysics® Software

In this section non-trivial aspects regarding the COMSOL implementation of the model are discussed.
Two seemingly different approaches are available to couple the fields of fluid dynamics and solid mechanics. On one hand the Fluid-Structure Interaction combining Thin-Film Flow Shell and Solid Mechanics could be used to simulate the interaction; on the other hand, the boundary physics named Thin-Film Damping, within Solid Mechanics worked exactly in the same way as the previous one. The latter allowed a more compact analysis of the component; thus, it was chosen as the most suitable simulation tool.

The Thin-Film Damping physics is a boundary condition, therefore different material had to be assigned to a boundary surface instead of a domain. It is important to define this assignment properly otherwise the model would not calculate fluid flow parameters and consequent loads on structure.

After having determined two principal natural frequencies (X and Z oscillations, 1st order), two boundary probes were included to perform a thorough vibration analysis. The damping ratio is the most important parameter in damped systems; hence, such boundary probes were created to evaluate it during different studies.

Several types of study were adopted throughout the analysis, namely Eigenfrequency, Time-Dependent and Frequency-Domain. Also different additions were implemented, such as Parametric Sweep and Optimization.

4. Model Validation

4.1 Analytical Model

In order to validate the simulation an analytical model was developed according to the extensive literature currently present. The squeeze film is in general governed by both inertial and viscous effect [1]. However, due to the small size of the typical MEMS structures, inertial effects are negligible. Hence the physics behind this phenomenon can be described by Reynolds equation [6]:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial y}\right) = \frac{\partial}{\partial x}\left(\frac{\rho h(u_a+u_b)}{2}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h(v_a+v_b)}{2}\right) + \rho(w_a - w_b) - \rho u_a\frac{\partial h}{\partial x} - \rho v_a\frac{\partial h}{\partial y} + h\frac{\partial \rho}{\partial t}$$

After several manipulations that come from various assumptions ([1], [3]), namely parallel motion and small perturbation, Reynolds equation can be transformed to the form of linearized Reynolds equation for compressible gas:

$$p_a\left(\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial y^2}\right)-\frac{12\mu\omega l^2}{h_0^2}\frac{\partial p}{\partial t}=\frac{12\mu p_a}{h_0^3}\frac{dh}{dt}$$

An important dimensionless parameter that pops up while working out the solution is the squeeze number, defined as:

$$\sigma=\frac{12\mu\omega l^2}{p_a h_0^2}$$

Griffin [7] approached the problem by determining first the film step response to a change in film thickness and then used the superposition principle to determine system response to any perturbation. From this analysis a cut-off frequency of the squeeze film air damping was determined as:

$$\omega_c=\frac{\pi^2 h_0^2 p_a}{12\mu w^2}$$

At the cut-off frequency the elastic force equals the viscous damping force. According to [1], [2] and [3], when a device owns a resonance frequency lower than the cut-off one (typical for high sensitivity MEMS [1]), the coefficient of viscous damping force is approximately a constant, whereas the elastic damping turns out to be negligible. The viscous damping coefficient for a squared plate was found to be:

$$c_d=0.42\frac{\mu L w^3}{h^3}$$

The analytical model was built following the standard second-order differential equation valid for general oscillators:

$$m\ddot{z}+c_d\dot{z}+(k_0+k_e)z=\rho a$$

where the squeeze-film effect has two different contributions: the elastic and damping coefficient. The structural elastic coefficient is represented by k_0. From what has been said earlier, the elastic squeeze-film coefficient was set to 0 and the structural one was assumed to be $k_0 = 3EI/l^3$ (first order approximation for a cantilever beam fixed at one end). Solving for the same step response and plotting total tip displacement, a reasonable agreement was found between the model and the analytical solution, which partially confirmed the validity of the former. The discrepancy between the steady-state displacement might be attributed to the approximation of elastic force component. Indeed the structural term is a first-order estimate, which sums up with the assumption of negligible thin-film elastic coefficient.

Fig. 1 shows the comparison between the simulation and analytical solution (ambient pressure was equal to 1 Atm, gap thickness 10 µm).

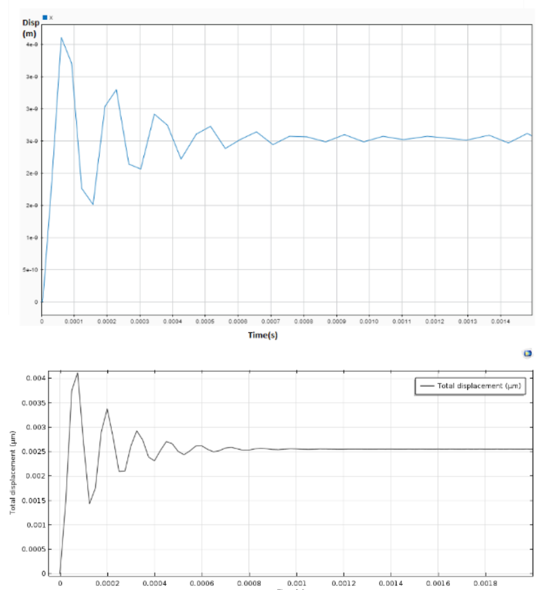

Figure 1: Comparison between simulation and analytical solution

4.2 Mesh Convergence

Mesh convergence technique was used to further validate the results. This approach checked whether the results were mesh-dependent or, as hoped, not. Due to low computational power, the mesh was turned into a less fine one (*extra fine* to *normal*). In Fig. 2 the two curves for tip displacement are overlapped showing acceptable agreement to validate the model.

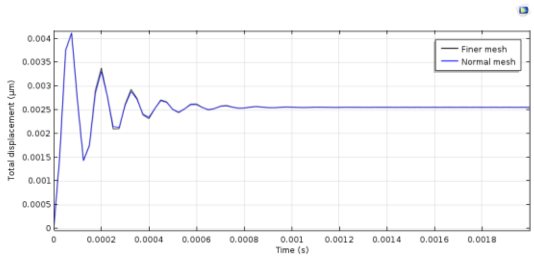

Figure 2: Different meshes results

5. Results and Discussion

The analysis was conducted with the aim of assessing thin-film damping on MEMS device. The hardest challenge of such situation is to find an optimum between high damping in transverse oscillation and very low damping in the horizontal motion.

A very first analysis was conducted upon sweeping ambient pressure from 500 Pa to 1 Atm. This process allowed to get insights on the response of the system; also, it determined the most suitable environmental pressure to adopt. The pressure sweep investigation only concerned the Z-direction because it was regarded as the most sensitive motion to be monitored.

Indeed, it is true that major effects of squeeze-film are present in normal displacement.

In Fig. 3 the different step responses of the system can be seen.

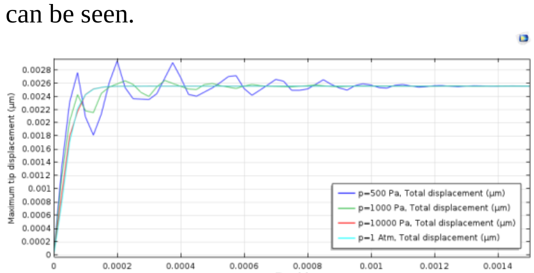

Figure 3: Tip displacement for different ambient pressures

The tip displacement was taken as an indicator of oscillation amplitude. Clearly the smaller the ambient pressure, the less significant is damping. For this reason the ambient pressure should be kept relatively high in order to exploit thin-film damping to block the transverse oscillation. As the pressure rises, the behavior asymptotically reaches the one at ambient pressure, hence a value of 10 kPa could be considered as an optimum threshold. Such ambient pressure yields a critically damped system keeping the pressure 10 times lower than the ambient one, which allows a reduced drag in horizontal oscillations. The same reasoning holds for the total fluid load on wall (integrated over the surface), as in Fig. 4.

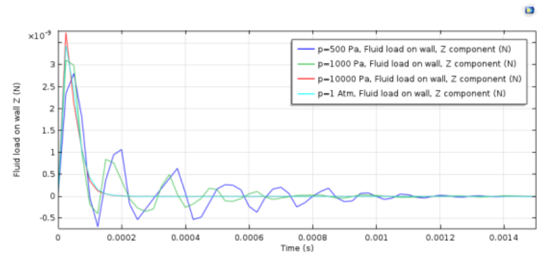

Figure 4: Fluid load on wall for different ambient pressures

The peak intensity, for p = 1 Atm and p = 10 kPA is enough to produce the desired damping an prevent the system from oscillating.

Beside what was done with the pressure sweep, the most significant design parameter to analyze was the thin-film gap height. According to [1], the effects of squeeze film damping becomes dominant when the characteristic size of the device is at least 3 times larger than the gap height. Simulating the step response with a gap height of 50 film led to a very tiny fluid-structure interaction, such that the damping was almost null. For this reason the range that was taken into account swept 3-10 µm. The lower limit was set after a quick estimation of damping in the X-direction as discussed below. The parametric sweep was performed for the body-load step response along the Z-axis (transverse motion): as shown in Fig. 5, the gap height affects significantly the response of the system.

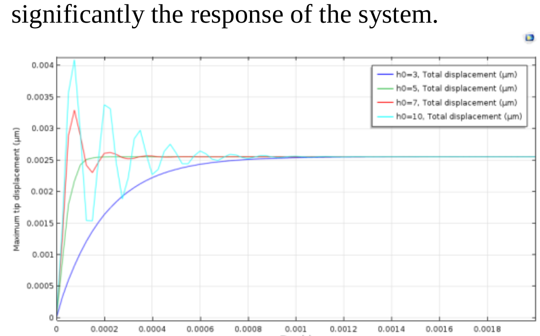

Figure 5: Tip displacement for different gap heights

From a graphical interpretation the oscillation behaves as an overdamped system for h_0 = 3 µm, nearly critically damped for h_0 = 5 µm and underdamped for the last two cases. The MEMS device would certainly take benefit from being critically damped, as this would let the system to be brought back to stable position within the shortest time. An overdamped system would simply react slower than the aforementioned but it would prevent overshooting, which might be present in a critically damped oscillator. In Fig. 6, the intensity of total integrated fluid load on wall is shown.

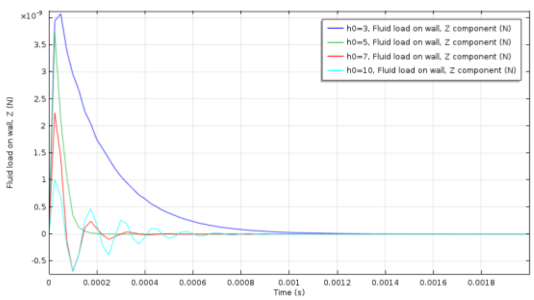

Figure 6: Total fluid load for different gap heights

It is clear that a thinner gap forces the fluid to react strongly to the change in equilibrium. As stated the squeeze-effect is much prominent as the gap height decreases. Again the most damped condition occurs when the fluid reacts with a peak at initial stages.

Furthermore the distribution of the fluid load across the surface was analyzed and typical pattern was found. A radial distribution was expected, in which the inner fluid is trapped by the squeezing-effect resulting in a much higher reaction load on the wall. In Fig. 7 an instant within the transient solution is shown.

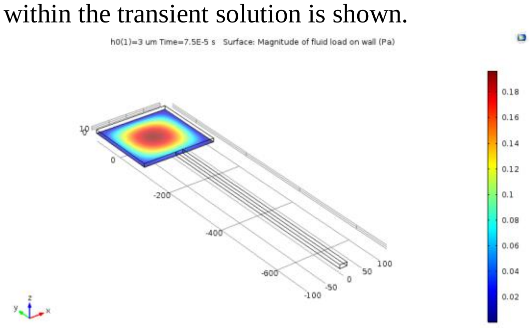

Figure 7: Fluid load distribution across wall surface at t=0.002 s

To further proceed with the analysis, an Eigenfrequency study was performed to determine natural frequencies of the undamped system. The eigenfrequencies were found to be:

$$f_{0Z} = 7771\ Hz$$
$$f_{0X} = 15432\ Hz$$

The damping ratio were plotted for the different gap heights in Fig. 8. Here the average of the data selection was taken because, due to numerical approximation, the variable showed several spikes throughout the time domain.

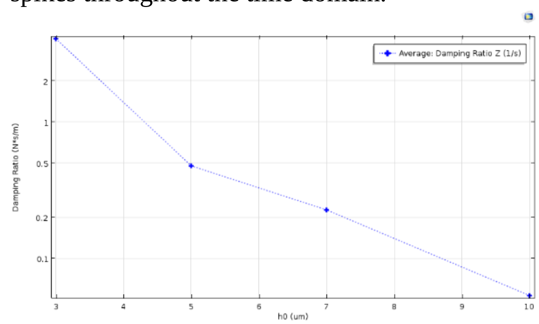

Figure 8: Damping ratio for different gap heights

It is important to note that the quality factor can easily be determined from the damping ratio:

$$Q = \frac{1}{2\zeta}$$

for the sake of simplicity, here the damping ratio was taken into account due to its being directly proportional to the physical phenomenon.

To better understand the system response a Frequency-Domain study was performed. The amplitude-frequency plots, shown in Fig. 9 and 10, demonstrate a little but not negligible effect of the thin-film damping on the X oscillation.

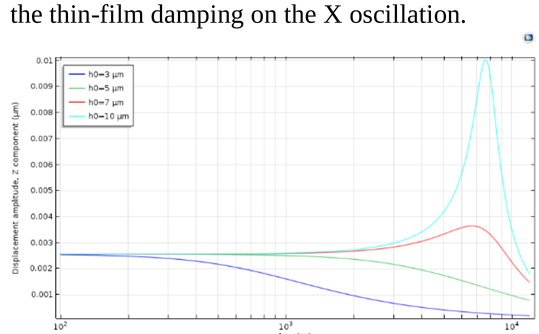

Figure 9: Frequency response in displacement amplitude, Z axis

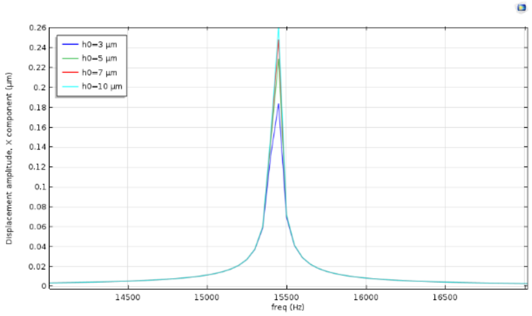

Figure 10: Frequency response in displacement, X axis

Although each frequency response is typical of an underdamped system, the quality factor (proportional to the inverse of the damping coefficient) reduces as the gap height reduces. In the transverse direction, the plot shows a typical set of second-order damped oscillator curves, as expected from the Time-Dependent study. The above-mentioned results led to the conclusion that not much damping was present in the horizontal motion; it is true that further analysis, using a more detailed description of the environment, may yield different and more reliable considerations but this goes beyond the scope of this work. Nevertheless, the minor influence of squeeze-film damping in such plane is definitely realistic. Hence, the device would best-performed if it was critically damped along Z for the reasons mentioned above.

An Optimization study was added to a Time-dependent along the Z direction to find out the gap height that would force the system to be critically damped. Although being a fairly heavy computation step, the optimization resulted in an optimal gap height of 4.48 µm, very close to the expected value drawn by the previous frequency sweep analysis.

Eventually a simple surface texture optimization was performed. The idea was to evaluate the effects of inserting multiple grooves on the interacting surface and vary their sizes to achieve the best-performing configuration. A simple and unique configuration was then taken into account to demonstrate the influence of surface texture on performance; this may lead to further study to be worked out in additional studies. The result, shown in Fig. 11, confirms what was expected.

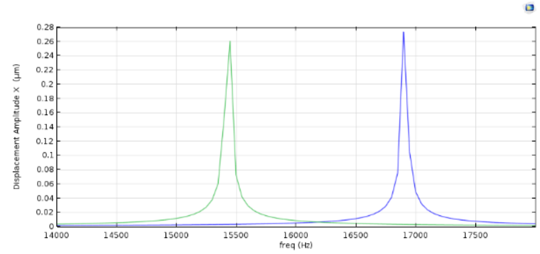

Figure 11: Frequency response along X-axis comparison between optimized surface and original

The maximum amplitude X displacement using $h_0 = 4.5$ µm is higher than the maximum X displacement calculated for the best-case scenario of $h_0 = 10$ µm. It is important to note that, since the shape has changed, the resonance frequency is different.

Finally a mechanical stress analysis was performed to be sure that all oscillations were not disruptive. After a preliminary display of Von Mises stress results, the concerns vanished because the values were below critical ones ($\sigma_y \approx 10^9$ Pa for Silicon). In Fig. 12 and 13 two instant are shown in the transient study along Z with gap height equal to 5 µm.

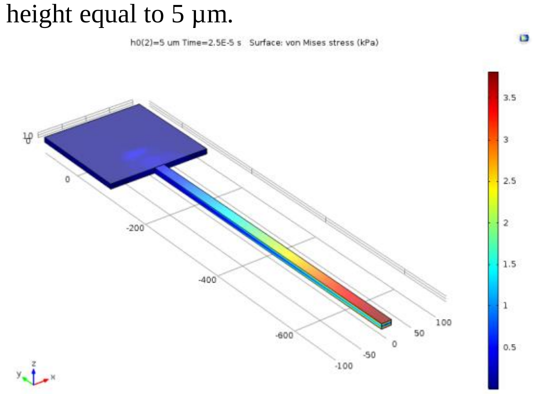

Figure 12: Von Mises stress at t=2.5e-5 s

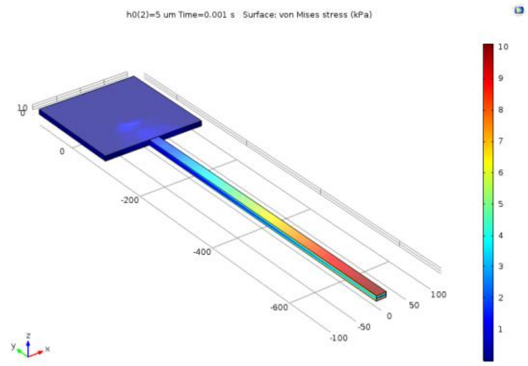

Figure 13: Von Mises stress at t=0.001 s

6. Conclusions

The work consisted in a simulation analysis, validation and optimization of the coupled physics phenomenon of anisotropic thin-film damping in a MEMS device. The analyzed MEMS oscillator was composed by a spring (beam) supported mass that needs to vibrate with a high Q-factor in the horizontal plane. Nevertheless, the transverse motion needs to be highly damped (low Q-factor). Multivariable analysis was performed taking into account the influence of various parameters, such as ambient pressure, gap height and surface texture.
The ambient pressure was lowered down to 10 kPa without altering the damping effect present at atmospheric pressure. Once the little influence on X-damping by squeeze-film was found, the optimization focused on finding the suitable gap height to make the device critically damped along Z direction. The gap height, which fulfilled such requirement, was found to be 4.5 µm.
In the end, an attempt upon surface texture optimization was made. However, inserting grooves parallel to the horizontal motion reduced the damping in such direction, helping the device to achieve an almost drag-free oscillation. Further investigations are certainly needed to work out a more detailed topological optimization.
The model was validated using a simple analytical model, built after having done a thorough literature study on the topic. Furthermore mesh convergence analysis was used as well. The mentioned course allowed the Author to acquire a preliminary knowledge upon the use of the software, undoubtedly a fundamental tool in the design process of all Engineering fields.

9. References

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