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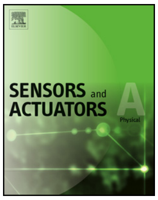

Air damping as design feature in lateral oscillators

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ABSTRACT

The damping of an oscillator is, next to the effective mass and the stiffness, one of the key parameters that determine its frequency response as well as its noise. Since there are numerous different applications for oscillating microstructures, there are also numerous requirements for the shape of the respective resonance peak. While fabricating the right mass and stiffness to obtain a desired resonance frequency is, in general, a basic task, designing a micro-oscillator featuring an intended damping is not trivial. We present a way of utilizing the surrounding air to adjust the passive damping of a laterally oscillating micromechanical system. This is shown to hold in a relatively wide range by comparing analytical models and finite volume method simulations with measurements of a number of micro-electro-mechanical test structures with optical readout.

1. Introduction

Over the last decades a vast number of microelectromechanical systems (MEMS) has evolved which are intended for a wide field of applications. These include micromachined devices such as microphones, gyroscopes, micromirrors, accelerometers or vibration sensors. Since most of them are based on a periodically moving mass, they have in common that they can be described as damped, driven, harmonic oscillators. Apart from an external driving force $\mathbf{F}_{\mathrm{ext}}$, their main properties, therefore, depend on three parameters: the stiffness k, the seismic mass m and the damping parameter d.

In terms of the damped, driven, harmonic oscillator, the stiffness and the mass determine the resonance frequency $\omega_0 = 2\pi f_0 = \sqrt{k/m}$. Mass and damping parameter compose the decay constant $\gamma = d/2m$. All three together determine the so-called quality of the system denoted by the Q-factor $Q = \sqrt{mk}/d = \omega_0/2\gamma$ which is a measure for the oscillator's stored kinetic energy compared to its dissipation. A high quality corresponds to low dissipation and shows up in a high resonance peak and vice versa. The damping is directly proportional to the amount of dissipation. Many applications favor a high quality factor. An example for this is a device based on resonant sensing, where the measurement frequency is equal to the resonance frequency. This kind of sensor depends on a high resonance peak to achieve an optimal output signal. Thus, the damping is required to be small in that case. In other applications, such as vibration sensing, however, one might prefer a low quality factor $Q \gtrsim 1$, i.e. higher damping. This is due to the inherent suppression of ringing and extension of the measurement regime. From the point of view of engineering, one typically wants to provide the device to be fabricated with the optimal set of parameters to assure optimal transfer properties for the given application. Therefore, modeling of these quantities is key.

The mass is usually the most basic parameter to design. It is given by the device geometry and the mass density of the used material and, therefore, no extensive modeling has to be done. The stiffness is also a well-understood quantity. Analytical models such as the Euler–Bernoulli beam theory [1] provide a deep insight. In the case of sophisticated layouts of devices or springs one can fall back on numerical approaches such as finite element method (FEM) simulations.

Accessing the damping, on the other hand, can be very challenging. This is partly due to the many different types of contributions composing the parameter d. The collective term "damping" refers to all mechanisms that lead to a dissipation of the oscillation energy. This includes effects such as anchor losses [2] where the oscillation couples to the surrounding or thermoelastic damping [3,4] where the bending of the spring effects a warming of the material at the compressed side and a cooling on the opposing extended side causing a force acting against the oscillation. More importantly, air damping [5–10] occurs, where the dissipation is caused by the induced flow of the air surrounding the seismic mass. Even if the environmental conditions allow for ruling out some of the

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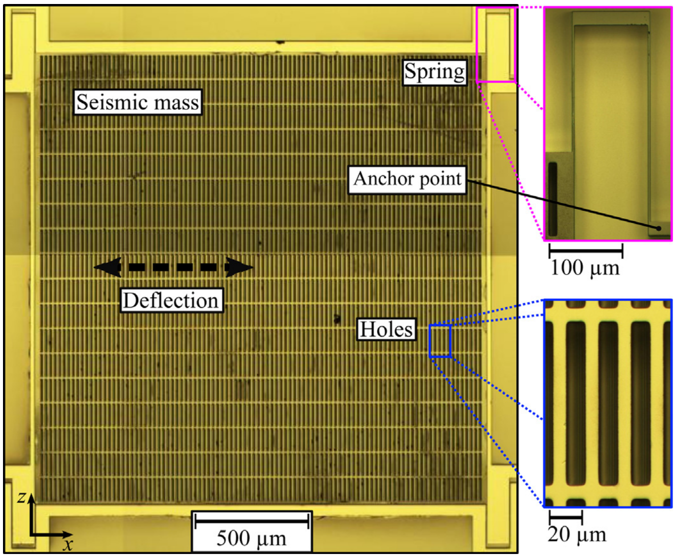

Fig. 1. Micrograph of a test structure. The seismic mass (width $w = 2\,\text{mm}$, length $l = 2\,\text{mm}$) is etched into the 45 µm thick device layer of an SOI wafer. It is suspended on four U-shaped springs. The rectangular holes (10 µm × 100 µm) are required for the optical readout. The mass will be excited only in x-direction. The zooms show an enlarged depiction of a spring and some holes.

dissipation effects, the damping might still be hard to model accurately. In the case of a MEMS operated in air at ambient pressure, the air damping is usually the largest contribution and overshadows the others by far. Due to the non-linearity of the governing equations of fluid flow, accurate modeling can be hard even for basic geometries. Thus, an analytical approach can in most cases only be applied in a limited way. Numerical computations or a combination of numerics and analytics [11] are, thus, usually the method of choice.

In Section 2, we introduce the kind of test structures for which the damping shall be designed. Afterwards in Section 3, the analytical and numerical approaches to the modeling of the air damping as well as their application to the design of a series of lateral microstructures are discussed. Subsequently, the measurement setup is explained and the results are compared to the models (Section 4). Finally, we conclude the paper and give a short outlook in Section 5.

2. The optical vibration sensor

2.1. Sensor principle

The kind of microstructure we want to fine-tune the air damping for is the micro-opto-electro-mechanical system (MOEMS) depicted in Fig. 1. We use this type of MOEMS for vibration sensing. It was introduced in [12] as a promising alternative to standard capacitive sensors. The MOEMS consists of two arrays of rectangular holes, one is made of Chromium patterned by physical vapor deposition (PVD) onto a glass chip, the other is etched into the device layer of a silicon on insulator (SOI) wafer by a standard Bosch deep reactive ion etching (DRIE) process. The resulting two wafers are bonded onto each other in such a way that the two grids form an array of apertures (see Fig. 2). Since the grid in the silicon chip is fabricated into the seismic mass of a suspended plate oscillator, any movement of the plate will increase or decrease the open area of the aperture and, therefore, the light flux passing through. For a displacement δx of the seismic mass, the change in the open area is given by $\delta A_{\text{open}} = \alpha_0\,\delta x = N_{\text{h}}\,l_{\text{h}}\,\delta x$. Note that δA_{open} can be positive or negative, depending on the sign of δx. Due to the large number N_{h} and the dimensions (length l_{h}, width w_{h}) of the holes, this kind of opto-mechanical transduction features a very high sensitivity. The light flux that has passed through the grids $\delta\Phi = \alpha_1\,\delta A_{\text{open}}$ is then converted into an electric current $\delta i = \alpha_2\,\delta\Phi$ and ultimately into a voltage $\delta U = \alpha_3\,\delta i$ by a photodiode or phototransistor and the readout electronics, respectively. The final transduction of the displacement into voltage is then given by $\delta U = \prod_i \alpha_i\,\delta x = g\,\delta x$ which defines the sensitivity or the gain g of the sensor.

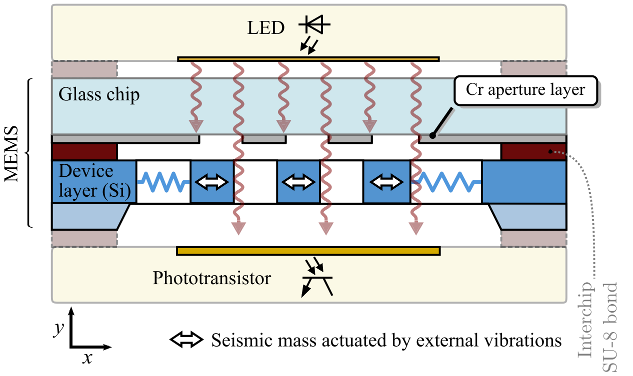

Fig. 2. Schematic showing the mechanism of the light flux modulation. The light coming from the LED passes through the apertures in the Cr layer. Similar apertures are etched into the seismic Si mass (see Fig. 1). If the mass moves, the amount of light registered by the phototransistor will change.

2.2. Transfer characteristics

The setup for testing the MOEMS which is described with more detail in Section 4 involves an external actuation **K**. This is provided by a custom made shaker unit [13]. The actuation is time harmonic with angular frequency ω and unidirectional in x-direction $\mathbf{e}_{\text{x}} = (1, 0, 0)$ and can therefore be written as $\mathbf{K} = K\,\mathrm{e}^{\mathrm{i}\,\omega\,t}\,\mathbf{e}_{\text{x}}$ with $\mathrm{i} = \sqrt{-1}$ being the imaginary unit. In this configuration, the driving force acts onto the suspension which is connected rigidly to the Cr grid on the glass chip and via springs to the seismic mass of the MOEMS. Thus, the driving force at first effects a deflection of the suspension $x_{\text{s}}(t)$ which in turn couples to the deflection of the seismic mass $x_{\text{m}}(t)$ of the harmonic oscillator. Since the change of the open area δA_{open} which is proportional to the difference of the deflections of suspension and mass $\delta x(t) = x_{\text{m}}(t) - x_{\text{s}}(t) = x(t)$ is the quantity we observe, the equation of motion can be written as

$$m\,\frac{\partial^2 x(t)}{\partial t^2} + d\,\frac{\partial x(t)}{\partial t} + k\,x = m\,\frac{\partial^2 x_{\text{s}}(t)}{\partial t^2}. \qquad (1)$$

The term on the right hand side of this equation corresponds to the actuation, i.e. $m\,\partial^2 x_{\text{s}}(t)/\partial t^2 = K\,\mathrm{e}^{\mathrm{i}\,\omega\,t}$. Dividing Eq. (1) by m and applying an exponential ansatz for the relative displacement $x(t) = X(\omega)\mathrm{e}^{\mathrm{i}\,\omega\,t}$ leads to

$$(-\omega^2 + 2\mathrm{i}\,\gamma\,\omega + \omega_0^2)X(\omega) = -K\omega^2. \qquad (2)$$

Multiplying the function $X(\omega)$ with the propotionality factor g which takes into account the whole opto-electro-mechanical transduction explained in the previous subsection and dividing off the actuation force amplitude leads to the complex valued, second order transfer function of the whole system.

$$A(\omega) = \frac{g}{K}X(\omega) = \frac{g}{1 - 2\mathrm{i}(\gamma/\omega) - (\omega_0^2/\omega^2)}. \qquad (3)$$