A STUDY OF GAS FRICTION IN A NONLINEAR ENERGY HARVESTER

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Abstract: Internal gas damping can be a limiting factor to the performance of micro devices scavenging vibration energy from the environment. We have studied the effect of gas at a pressure between 2 and 1000 mBar in the cavity of a shuttle type vibration harvester where the seismic mass moves within a confined volume with narrow gaps. Based on the assumption that the mass spring system is linear at atmospheric pressure we develop a relation between the electric output and the mechanic displacement of the mass that is valid over the entire pressure range. When we use this relation as the basis for our analysis of the experimental data, we find that the gas damping force is proportional to the velocity of the seismic mass in the pressure range studied. The outcome of the analysis is the damping coefficient and the mechanical Q value as function of gas pressure.

Keywords: Vibration harvester, gas friction in narrow gaps.

INTRODUCTION

A vibration energy harvester will always have a seismic mass that is excited by vibrations in the operating environment. In micro fabricated devices the mass is most often confined in a sealed cavity. In an electrostatic harvester, which we studied, the electrodes on the vibrating mass interact with electrodes on the fixed frame or housing to produce electric power in an external load. A sketch of our device is shown in figure 1 and description of the device is found in [1]. Gas in the cavity of this device will be squeezed back and forth as the as the mass moves as a shuttle. Unless the pressure in the cavity is very low, this gives rise to damping and reduced performance. The loss will depend on the width of gaps between mass and frame as well as the cavity volume not filled by the mass. Small gap widths and efficient use of the device volume are, however, positive design features that should not be scarified.
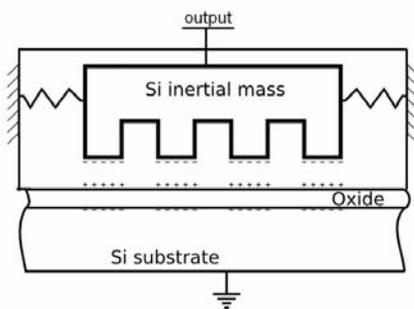

Figure 1: The basic structure of the vibration scavenger.

In an earlier study we have identified gas damping in a vibration scavenger and observed the relation between damping force at a constant output voltage and the cavity pressure. [2] While these results showed that the gas damping was a dominant factor over a wide pressure range, they did not refer to standard parameters, such as Q value and damping coefficient, which can be compared between devices of different design.

In the present work we address this problem. We separate the response of the mass spring system of the harvester from the observed response of the device, which also includes the nonlinear transduction. To do this we must establish an empirical relation between the output power and the displacement of the vibrating mass. This relation will not depend on pressure and we can use it to convert the electrical output, which can be measured, to proof mass displacement over a wide pressure range.

Since the effectiveness of our device is very low, the gas damping force totally dominates over the damping force due to the transduction. The gas damping force will therefore equal the fictitious input force, $m_{sh} \cdot a$, of the harvester. m_{sh} is the mass of the shuttle or proof mass and a is the acceleration caused by a vibrating environment.

We are able to show that the assumption of a linear mass spring system and a non linear transduction relation, derived from the output characteristics at atmospheric pressure, is consistent with viscous gas damping in the range 20-1000 mBar. The gas friction can then be expressed by a damping coefficient or a Q value.

NONLINEAR TRANSDUCTION

Our energy harvester has a pronounced nonlinear behavior. This is illustrated by figure 2. The figure shows the output waveform and the acceleration at 4 different levels. The electric output is directly related to the rate of change of the induced charge caused by variations in the overlap between the fixed electret
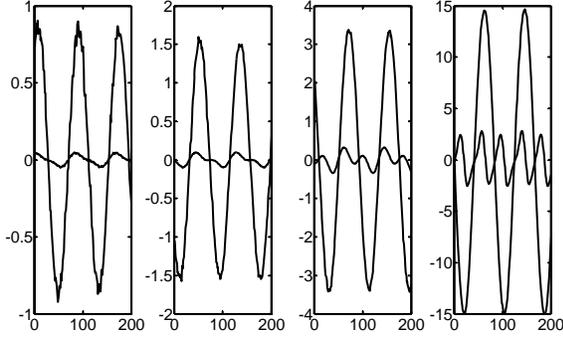
pattern and the vibrating electrode structure.

Figure2: The output waveform and excitation signal at 4 different acceleration amplitudes.

The plots tell that the electrodes imperfectly align with the electret stripes in the equilibrium position. This gives a linear response for small excitations. As the acceleration is increased, second order terms of the spatially modulated potential become dominant. This gives rise to an output of twice the mechanical frequency.

From the discussion above it is quite obvious that we must expect a nonlinear input-output relation in the electric transduction. Figure 3 shows how the output voltage first increases at a linear rate with the displacement of the shuttle. As the excitation increases this rate first increases and then decreases at high input.
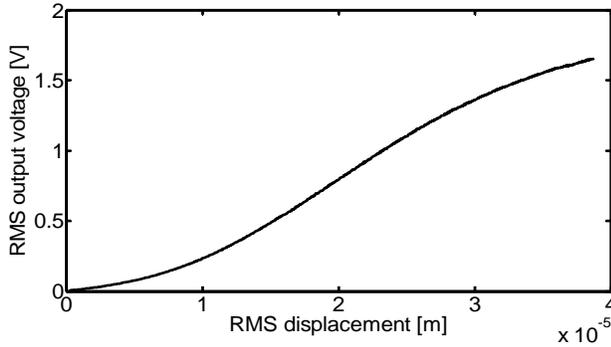

Figure 3: The relation between the RMS displacement of the shuttle and the output voltage.

The figure is based on experimental data measured at the resonance frequency, 573 Hz, at atmospheric pressure. The measured x-axis variable was the applied acceleration. To change from acceleration to mass displacement as in figure 3, we must make the assumption that the mass spring system of our device behaves as a linear 2. order system at high pressure. This is well justified since the gas damping is strong and nonlinearities in the

springs, which may be evident at low pressure, are not seen in the frequency responses recorded at atmospheric pressure. Higher order resonances are above 6 kHz and they will not interfere with our measurements. In a linear 2. order system there will be a fixed ratio between the excitation, a_{frame} and v_{frame}, and the corresponding variables of the seismic mass, a_{sh} and v_{sh}. The ratio is given by the quality factor, Q_{mech}. We determine Q_{mech} by fitting a standard 2. order frequency response plot, characterized by f_0 and Q_{mech}, to a measured frequency response plot. This is shown in figure 4.
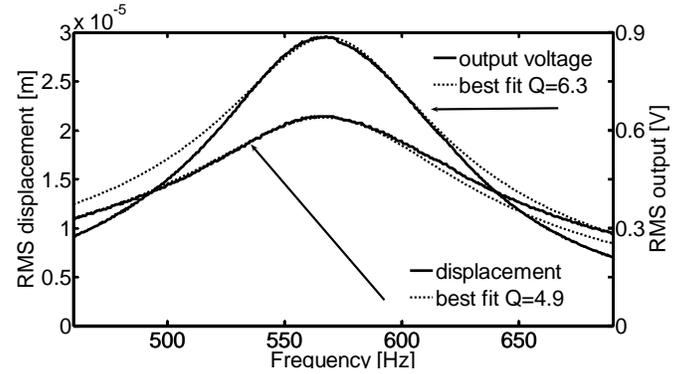

Figure 4: A linear, 2.order response fitted to the linearized frequency response. (The displacement scale is based on Q=4.9)

The figure shows two plots. When we fit the standard response to the upper, non adjusted, output voltage, the two curves are not very close. When we try the same on the lower, mechanical displacement curve, the result is much better. The displacement as function of output voltage is determined by the relation in figure 3. At this stage the scale of the displacement axis is not known, but we know that the scale is linear. The unknown scale factor does not affect the curve fitting and we arrive at the best result with a Q_{mech} of 4.9. We can now use this value to properly scale the displacement axis in figure 3 and figure 4. The relation used to convert the applied acceleration into the shuttle displacement on the x-axis of figure 3 is given by:

$$x_{sh} = \frac{Q_{mech}}{\omega_0^2} a_{frame} \qquad (1)$$

We will now proceed to analyze experimental transfer relations recorded at pressure between 2.5 mBar and 1000 mBar, where the relation in figure 3 plays an important role.

ANALYSIS OF DAMPING

A practical method to study the nature and pressure dependence of gas damping in an energy harvester is to determine the relation between damping force and shuttle velocity at different pressure values. The procedure starts with measurement of the relation between applied acceleration of the frame and output voltage of the harvester. The procedure is described in [2]. Measurements were done at the resonance frequency of the mass spring system which goes from 550 Hz to 570 Hz as the pressure increases from 2.5 to 1000 mBar. The raw data are seen in figure5.
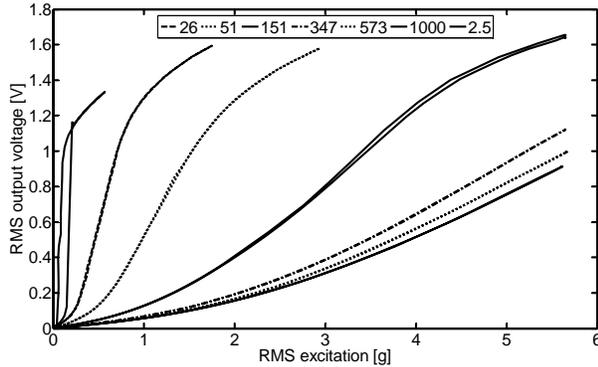

Figure 5: Measurement results. Output voltage as function of excitation at 7 different pressure values.

At the resonance frequency spring- and inertial forces cancel out, and the applied fictitious force, $m_{sh} \cdot a_{frame}$, is equal to the gas damping force. This is used to convert the x axis values in figure 5 into damping force. The y axis data are converted from output voltage into velocity by the transduction relation in figure 3 and the relation between displacement and velocity:

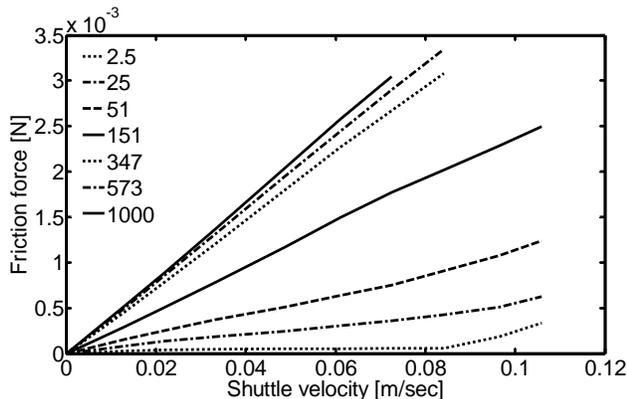$$v_{mass} = \omega_0 \cdot x_{sh} \tag{2}$$

Figure 6: Friction force as a function of the velocity of the seismic mass at 7 different pressure values.

When we use this procedure on the seven plots in figure 5, we arrive at the damping force versus shuttle velocity plots shown in figure 6.

The plots show that a nearly linear relation exists between the velocity and the damping force. This implies that as far as the initial assumption of a viscous damping and thereby a linear relation at 1000 mBar is correct, the damping is also linear in the pressure range we investigated, down to 2.5 mBar. From the plots in figure 6 we can therefore extract the damping coefficient, B, as a function of the pressure. This is done in figure 7. In this figure we have also plotted the mechanical Q_{mech} value that is related to B as:

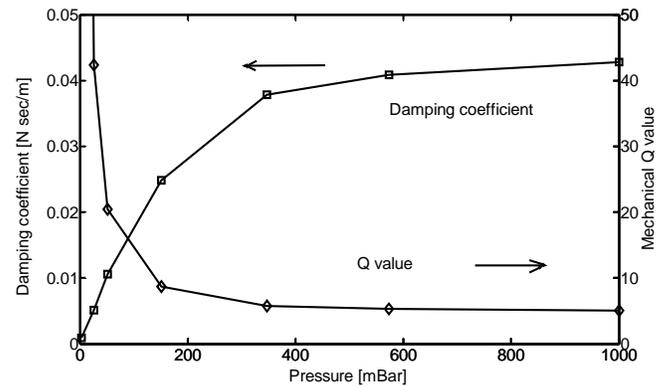$$Q_{mech} = \frac{\omega_0 \cdot m_{sh}}{B} \tag{3}$$

Figure 7: The damping coefficient B and Q_{mech}, as functions of the ambient pressure.

We see that the damping increases with pressure at the low end of the region while it has almost saturated at atmospheric pressure.

DISCUSSION

Figure 7 gives a good picture of the extent of degradation of the harvester performance due to gas damping. While damping force and damping coefficient are closely related to the device size, the Q_{mech} value is dimensionless and well suited as a figure of merit that can be compared between different designs. The two important features of the curve are its value, compared to what can be accepted in an efficient harvester, and its pressure dependence. We see from the figure that B increases from a low value which corresponds to a Q_{mech} of 175 at 2.5 mBar, to saturate at a level which corresponds to Q_{mech} close to 5 as the pressure approach atmospheric. The transition is in the range 50-200 mBar. It has not been our intention to study in any detail the gas flow in our

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device. We believe that the damping force is set up by the pressure difference between the end faces of the shuttle when gas is squeezed from one side of the cavity to the other side as the shuttle moves back and forth. The gas has to move through channels with a height in the 10 μm range on both top and bottom of the shuttle. On the side faces transverse to the motion, the height is larger, but the width is only the wafer thickness The flow resistance to viscous flow is not pressure dependent, but the resistance will decrease when the flow goes into a regime where the mean free path between collisions approaches the channel height. The transition between the two regimes has been observed in the region 10-100 mBar pressure [3], which agrees well with our observation.

The other aspect of our results is how low value of Q_{mech} that can be accepted and the values of residual pressure that can be expected in a sealed MEMS cavity. The acceptable range of Q_{mech} should be well above the equivalent Q value representing the transduction. In most of the harvesters that have been fabricated in the past, the efficiency has been poor and the Q_{trans} has been accordingly high.[4] On the other hand, there is not much to gain by having so low electric and mechanic attenuation in the device that the seismic mass will have its travel much limited by the cavity walls or stopper structures when it is operating in its normal vibration environment.

All problems associated with gas damping would be solved if the MEMS cavity could be made completely evacuated and leak proof. While there are processes that result in very good seals, there will always be some outgasing that leaves a residual gas in a sealed cavity. From figure 7 we see that even residual pressure in the few mBar region will lead to a significant reduction of Q_{mech} in our specific device.

CONCLUSION

We have studied the effect of gas damping in a vibration energy harvester. We find that the effect acts as viscous loss in the mechanical system, and it can therefore be characterized by a mechanical Q_{mech} or viscous friction coefficient. The friction depends strongly on the pressure. In our device it becomes a limiting factor to the performance of the harvester at a cavity pressure of less than 10 mBar.

ACKNOWLEDGEMENT

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