Dynamic Measurement of Gas Damping Effects in MEMS

David S. Epp *
Sandia National Laboratories
MS 0557, P.O. Box 5800
Albuquerque, NM 87185

O. Burak Ozdoganlar
Department of Mechanical Engineering
Carnegie Mellon University
Pittsburgh, PA 15213

Hartono Sumali
Sandia National Laboratories
MS 1310, P.O. Box 5800
Albuquerque, NM 87185

ABSTRACT

The study of fundamental structural dynamic properties is critical for improving Micro-electromechanical System (MEMS) design and functionality. A new facility for measuring the structural dynamics of microsystem parts with a base-excitation technique is described and experimental results from a study of the effect of gas damping on the structural dynamics of MEMS are presented. The new facility includes a piezo-actuator to provide stepped-sine excitation to the bases of the test part and a laser Doppler velocimeter coupled to a microscope to sense velocity response. An environmental chamber enables examination of the effects of air pressure from atmospheric to near vacuum. Eliminating any resonances of the facility and any unintended excitation in the frequency range of interest were particularly important in this design. One of the tests that has been accomplished with the new facility is an experimental study of gas damping in MEMS. Gas damping has a disproportionately large effect on microsystems because of their large surface areas and their relatively small gaps from the substrate. This paper presents experimental results from measurements of micro-cantilever beams vibrating in ambient pressures from atmospheric down to 0.4 Pa. The results from these tests illustrate the strong dependence of energy dissipation on geometry and gas pressure. Effective gas damping at atmospheric pressure is shown to be as much as three orders of magnitude greater than structural damping. Some nonlinearity was evident in the test results at lower pressures where damping was reduced. The results from these tests will be used for future model validation.

1 INTRODUCTION

Micro-electromechanical systems (MEMS) research is a new frontier in the field of engineering. Technology exists now to create systems on the sub-micrometer scale, such as acceleration, pressure, and chemical sensors, actuators, and mirror arrays. Realization of commercially viable MEMS products requires a thorough understanding, and eventually prediction, of their performance and reliability characteristics. Currently, much of the experimental work on MEMS is focused on investigating whether new designs function properly. However, as design complexity increases and reliability becomes more critical, detailed knowledge about structural dynamics and coupled-physics phenomena that affect these structures will be necessary.

As expected from any relatively new technology, the research in microsystems has concentrated on fabrication techniques and design[1]. Numerous products have been designed and fabricated within research organizations and universities[2]. However, only a few have been realized as commercially viable products. In addition to long-term reliability issues, one of the primary impediments to commercialization is the lack of tools for understanding, testing, modelling, and performance prediction for microsystems, including those for microsystem dynamics[3]. Characterization of mechanical and dynamic properties, testing capabilities, and experimentally validated high-fidelity predictive modelling are required to assure that the microsystems reliably function.

*Corresponding Author, dsepp@sandia.gov
Structural dynamics behavior of microsystems that include moving, overhung, and compliant subcomponents plays a pivotal role in determining system performance and reliability. Traditionally, experimental modal analysis is used to characterize dynamic behavior of structures, as well as to develop, validate, update and correct analytical and numerical models. When performing modal testing, especially in the presence of system nonlinearities as is the case with MEMS, it is necessary to dynamically isolate the system from the environment so that the measured system response is that due to the prescribed input only. Furthermore, the structural dynamics of the input transducer must be carefully tuned to exhibit no resonances within the frequency range of interest. Previous work in testing nonlinear phenomena in MEMS devices has shown the adverse effects of resonance frequencies of the facility and environmental noise. The problems encountered in that work motivated the facility design in this paper.

Gas damping has a disproportionately large effect on microsystem structural dynamics because of their large surface areas and their relatively small distances from the substrate. Recent work in both experimental characterization and theoretical modeling indicate that understanding gas damping in MEMS is still an inadequately understood phenomenon. Current experimental work in this area uses electrostatic excitation, which introduces nonlinearities associated with the excitation. Previous work by Ozdoganlar et al. has made preliminary attempts at measuring gas damping effects with base excitation in an effort to mitigate nonlinearity issues associated with the excitation. They had difficulties, though, since resonances of their test facility biased the system response. The gas damping data presented here was collected with a redesigned experimental facility that eliminated those problems.

This paper presents an experimental facility that uses base excitation to test nonlinear structural dynamics and coupled-physics phenomena in microsystems. As an example of its utility, the facility is used to experimentally explore gas damping in micro-cantilever beams. Results from modal testing with base excitation require some processing to maintain consistency with conventional modal results. The new experimental facility uses a piezo-actuator that provides stepped-sine excitation to the base of the MEMS device and a laser Doppler velocimeter to sense velocity response. A seismic mass and suspension provide isolation from environmental noise, and a vacuum chamber enables examination of the effects of air pressure from atmospheric to near-vacuum. Eliminating resonances of the facility and unintended excitation in the frequency range of interest are particularly important in this design. Micro-cantilevers of various geometry are used to investigate the effects of gas damping.

2 MODAL TESTING WITH BASE EXCITATION

Modal testing typically involves a measurable excitation, such as that from a modal hammer or an electrodynamic shaker, and a response, which is simultaneously measured by an independent transducer. The most common combination of excitation and response measurements is that of force and acceleration. The ratio of the response to the excitation can be represented in the frequency domain as a frequency response function (FRF). When the excitation is force and the response is acceleration, the resulting FRF is referred to as an accelerance FRF. Most modal extraction algorithms assume that the measured response of the structure is an accelerance FRF.

Due to the small size of the MEMS structures, non-intrusive excitation and response measurement is required. Base-excitation provides a simple means of non-intrusive excitation. Testing via base-excitation typically involves measurements of the base exciter motion and the structural motion. These measurements can be combined into a transmissibility, which is the ratio of output motion to input motion in the frequency domain. For the apparatus in this paper, the velocities of the MEMS structure and the top face of the miniature shaker have been measured.

To be used with standard modal analysis techniques, the transmissibility data must be cast in the form of an accelerance FRF. The relationship between transmissibility and accelerance is illustrated in Figure 1. Figure 1 represents a second-order, single-degree-of-freedom system where the motion of the mass relative to the base is

$$z(t) = x(t) - x_0(t).$$  

Equation 1

In Equation 1, $x(t)$ is the displacement of the test mass as a function of time and $x_0(t)$ is the base displacement. Then, using
the relative coordinate \( z(t) \), the system dynamics can be expressed as

\[ \ddot{z}(t) + 2\zeta \omega_n \dot{z}(t) + \omega_n^2 z(t) = -\ddot{x}_0(t), \]  

(2)

where \( \omega_n \) is the natural frequency and \( \zeta \) is the damping ratio. Taking the Fourier transform of 2 with zero initial conditions yields

\[ \frac{Z(j\omega)}{X_0(j\omega)} = \frac{\omega^2}{-\omega^2 + 2j\zeta \omega_n \omega + \omega_n^2}. \]  

(3)

The right-hand-side of this equation is now in the form of an accelerance. Next, the left-hand-side of 3 must be expressed in terms of the base and system velocities that are measured during the testing by substituting the Fourier transform of Eq. (1) for \( Z(j\omega) \).

\[ \frac{X(j\omega) - X_0(j\omega)}{X_0(j\omega)} = \left( \frac{X(j\omega)}{X_0(j\omega)} - 1 \right) = \frac{\omega^2}{-\omega^2 + 2j\zeta \omega_n \omega + \omega_n^2}. \]  

(4)

Equivalently,

\[ (H_{VT}(j\omega) - 1) = \left( \frac{V(j\omega)}{V_0(j\omega)} - 1 \right) = \frac{\omega^2}{-\omega^2 + 2j\zeta \omega_n \omega + \omega_n^2}, \]  

(5)

where \( H_{VT} \) is the velocity transmissibility, \( V(j\omega) \) and \( V_0(j\omega) \) are the velocity spectrums at the locations of \( X(j\omega) \) and \( X_0(j\omega) \), respectively. Thus, the accelerance can be calculated through a simple operation on the velocity transmissibility, thereby allowing modal properties to be extracted from velocity data.

3 EXPERIMENTAL FACILITY DESIGN

The experimental facility is designed to minimize the influence of the test apparatus and external noise on the measured response of the MEMS. A piezo-actuator is used to excite the MEMS device with a sinusoidal base motion, and the velocity response is measured using a Laser Doppler Vibrometer (LDV) coupled with a microscope. To ensure that the response will be predominantly in a given direction, the piezo-actuator is attached onto a small seismic mass. A suspension for the seismic mass is designed to insure that the external noise is filtered. Moreover, a vacuum chamber is designed to completely enclose the miniature shaker and test piece. The facility is designed for a test frequency range of 1 kHz to 100 kHz. The software development for this facility is covered in a companion paper [24].

3.1 The Seismic Mass

The main purpose of the seismic mass is to direct the motion induced by the piezo-actuator in a given direction, i.e., away from the seismic mass. For a specified motion amplitude ratio, \( \frac{v_{pa}}{v_{sm}} \), the required mass ratio of the seismic mass and piezo-actuator
can be determined from the conservation of momentum expressed as
\[ m_{sm}v_{sm} + m_{pa}v_{pa} = 0, \]  
(6)
where \( m_{sm} \) and \( v_{sm} \) are the respective mass and velocity of the seismic mass, and \( m_{pa} \) and \( v_{pa} \) are the respective mass and velocity of the piezo-actuator. Then, the mass ratio
\[ \frac{m_{sm}}{m_{pa}} = \left| \frac{v_{pa}}{v_{sm}} \right|. \]  
(7)
Therefore, if one requires the velocity of the piezo-actuator to be four times larger than that of the seismic mass, the seismic mass needs to be at least four times more massive than the piezo-actuator.

It is also required that the miniature shaker does not possess resonance frequencies within the frequency range of interest. The dimensions and material of the seismic mass must be selected such that its elastic resonances are above the highest frequency of interest. The first elastic resonance of the seismic mass can be calculated as
\[ f = \frac{c}{2h}, \]  
(8)
where \( c \) is the speed of sound in the material and \( h \) is the largest dimension of the seismic mass.

Based on the above considerations, a stainless steel seismic mass with \( 0.7 \times 1.0 \times 1.0 \) cm dimensions has been selected for the test facility. This seismic mass is more than an order of magnitude more massive than the piezo-actuator, which yields a mass ratio greater than 10. Using Equation 7, it is determined that at least 90% of the actuator motion will be away from the seismic mass. Using Equation 8, the first elastic mode is calculated as 250 kHz, which is well above the frequency range of interest.

### 3.2 The Suspension Mechanism

One way to isolate the system being tested from external noise is to implement a mechanical filter. The mechanical filter is tuned such that all its resonances are below the frequency 1 kHz. Since the response rapidly diminishes after a resonance, any external noise above the resonance will be filtered.

This facility uses foam as the suspension mechanism based on the configuration represented in Figure 2. A 2 cm square block of 1.25 cm thick open-cell neoprene foam rubber is used. This foam is rated for 25% compression under 13.8 to 34.5 kPa and fails at 414 kPa of tension. To determine an acceptable means of attaching the seismic mass to the foam, Eastman 910 (superglue) and ordinary double-sided tape were compared. It was determined that neither material altered the dynamic response of the seismic mass by introducing new resonances within the frequency range of interest. For simplicity, the double-sided tape was chosen as the method of attachment.
To evaluate this design, the suspension and seismic mass assembly was connected to a commercial Wilcoxon shaker driven with a true random (white noise) signal from 50 Hz to 10 kHz. The data were collected from the four corners of the seismic mass. Figure 3 shows a typical response of one corner of the seismic mass. It can be seen that the suspension design does not exhibit resonances between 1 kHz and 10 kHz. The evaluation of this suspension design at higher frequencies using a piezo-actuator will be described in the next section.

3.3 The Piezo-actuator

Piezo-actuators are capable of providing dynamic excitation up to very high frequencies. Since we want to directly control the piezo motion with the input voltage to the amplifier, a flat frequency response is required between the input and output voltages of the amplifier through the frequency range of interest with the piezo-actuator attached. The method for attaching the piezo-actuator to the seismic mass was also considered.

A single layer of PIC151, from Polytec PI, Inc, is used as the base exciter in this facility. This 10 mm square actuator with 1 mm thickness is rated to produce up to 1 \( \mu \text{m} \) displacement at 1000 V. It was driven with a Krohn-Hite 7500 power amplifier which has a bandwidth of 1 MHz when it drives a 200 \( \Omega \) load. To evaluate the effect of the capacitive load on the frequency response of the amplifier, an FRF between input and output voltages of the amplifier was collected. This showed that the amplifier and piezo-actuator combination does not introduce any unwanted dynamics inside the 1 kHz to 100 kHz bandwidth.

To rigidly attach the piezo-actuator to the seismic mass Eastman 910 (superglue) was used, however this resulted in the corners of the piezo-actuator exhibiting higher displacements than its center during testing. It was suspected that this behavior arises from the nature of the attachment of the piezo-actuator onto the seismic mass. The actuator was glued down to the seismic mass by applying Eastman 910 over the entire contacting face. This attachment essentially produced a clamped boundary condition on the lower face of the actuator and a free boundary condition on the upper face.

A finite-element model was developed to investigate the deformation of the piezo-actuator. Linear piezoelectric finite elements were used for a 10 mm by 10 mm by 1 mm block of a PIC151 piezoelectric ceramic. The base of the actuator is rigidly attached (i.e., clamped) to a stationary foundation by specifying zero displacement and zero rotation on the nodes that belong to the bottom surface of the model. Material properties (elastic, dielectric, and piezoelectric constants) are taken from the specifications provided by Polytec PI, Inc. The model was meshed with 6900 brick elements. An electric field was applied to the model by specifying a 0 V boundary condition on the upper nodes and a 200 V boundary condition on the bottom nodes, which resulted in a 0.2 MV/m electric field, representative of that applied during experiments.

Figure 4 shows the deformed shape of the piezoelectric actuator predicted by the finite-element model. A positive electric field results in the material elongation in the direction of the field and contraction in both transverse directions. The clamped boundary surface generates reaction stresses that restrict the transverse contraction of the bottom, which results in larger displacement at the corners than at the center of the piezo-actuator forming a bowl shape. It can also be seen, however, that there exists a large plateau. So long as the test article is attached within this plateau, this behavior should not be of concern.

3.4 Vacuum Chamber

To test MEMS dynamics under various levels of ambient pressure a vacuum chamber was constructed. The vacuum chamber was designed to completely enclose the miniature shaker to minimize distortion in the new facility. The desired vacuum limit for the chamber was approximately 0.4 Pa.

Figure 5 shows the assembled vacuum chamber with the seismic mass, piezo-actuator, and a test specimen inside. A blank flange was used as the base of the chamber. A hollow flange with a 5 cm inner diameter was placed on top of the blank flange, and a viewport was attached on the top of the hollow flange. Ports were fabricated to allow electrical and vacuum connections while Viton gaskets were used in all connections to provide a vacuum seal. All of the parts used are rated for pressures well below the intended working pressure.

Figure 6 shows the response of the final design configuration from the top of the piezo-actuator. Assuming the input voltage into
Figure 4: Analysis results for the piezo-actuator with a clamped base in (a) cross-section and (b) three-dimensional displacement.

Figure 5: Assembled vacuum chamber with seismic mass, piezo-actuator, and test specimen inside.

Figure 6: FRF of the final design showing velocity relative to actuator drive voltage. Final design combines the single layer 10 mm square by 1 mm piezo-actuator with the foam suspended seismic mass.

the piezo-actuator is linearly proportional to its displacement, when all exogenous disturbances are eliminated, the FRF between the amplifier input and the velocity output of the piezo-actuator would approximate that of a first-order derivative in the frequency domain. Since the FRF in Figure 6 approximately follows this form, it was concluded that all the design requirements have been fulfilled.

4 GAS DAMPING EXPERIMENTS

Due to the large surface-to-volume ratios of microsystems, surface forces play a critical role in determining the characteristics and performance of microsystems. Gas forces include air drag and squeeze-film effects, the latter of which is usually dominant for surfaces moving relative to one another in close proximity. The goal of the experiments was to determine the amount of energy dissipated due to ambient gas forces as measured by the viscous damping ratios for various cantilever beam geometries, pressures, and gap heights.

Gas forces are nonlinear functions of instantaneous gap height and beam velocity. In an effort to capture nonlinear behavior in the parts tested, special considerations were made. A stepped-sine excitation was used rather than a broadband random
excitation to capture the transmissibilities from the base to the MEMS part. Used with the Hanning window and spectrum averaging, broadband type of excitation effectively linearizes the measured FRFs\cite{9,10,25}. Furthermore, by using stepped-sine excitation, we can capture nonlinearities such as those arising from multi-physics phenomena in the FRF\cite{26}. Since there was a possibility of amplitude-dependent nonlinearities, each test was repeated at two excitation levels. The second level was selected to be four times higher than the nominal level.

4.1 MEMS Micro-Cantilever Specimens

To demonstrate the capabilities of the test facility, the effect of beam dimension, beam-to-substrate gap height, gas pressure, and excitation amplitude on the damping characteristics of micro-cantilever beams was investigated. Polysilicon beams were used with varying lengths from 100 \( \mu \text{m} \) to 1 mm, and a 18 \( \mu \text{m} \) width fabricated in the SUMMiT V\textsuperscript{TM} (Sandia Ultra-planar, Multi-level MEMS Technology) process. Two nominal gap heights (the distance from the bottom of the beam to the base surface) were considered: 2 \( \mu \text{m} \) and 6.3 \( \mu \text{m} \). It should be noted that the gap height varies along the length of the beam from the residual stresses due to the fabrication processes. The height variation is more pronounced for the longer beam lengths. The 2 \( \mu \text{m} \)-gap beams had a design thickness of 2.5 \( \mu \text{m} \), whereas the 6.3 \( \mu \text{m} \)-gap beams had a design thickness of 2.25 \( \mu \text{m} \). Figure 7 is an SEM image of the 100 \( \mu \text{m} \) through 500 \( \mu \text{m} \) beams for both gap heights. The beams at the top of the image have a 6.3 \( \mu \text{m} \) nominal gap height, whereas those at the bottom of the image have a 2 \( \mu \text{m} \) nominal gap height.

4.2 Experimental Results and Analysis

The experiments reported here concentrate on the 200 \( \mu \text{m} \), 400 \( \mu \text{m} \), 500 \( \mu \text{m} \), and 700 \( \mu \text{m} \) beams with a 6.3 \( \mu \text{m} \) nominal gap height and the 200 \( \mu \text{m} \) and 400 \( \mu \text{m} \) beams with a 2 \( \mu \text{m} \) nominal gap height. Transmissibility data for each beam was collected from the beam tip and beam base. The tip was located by positioning the outer radius of the observable laser spot at the edge of the beam, which means that the motion was measured from 3 \( \mu \text{m} \) inside the edge\footnote{Although the actual laser spot diameter is known to be less than 2 \( \mu \text{m} \), it looks like a 6 \( \mu \text{m} \) spot due to the light source and CCD camera visualization}. The data collected for the base of the beam was taken from the substrate close to beams anchor. Two excitation levels were used for each test location.

Figure 8 shows the effect of varying ambient pressure on the measured transmissibility for the 400 \( \mu \text{m} \) beams at both gap heights. For all of these tests the range for the stepped-sine excitation was designed to capture the resonant peak and surrounding dynamics in the \( 1/10 \)th power band (i.e., the frequency range where the FRF amplitude is more than \( 1/\sqrt{10} \) times the resonant peak amplitude). Also, the frequency steps in the excitation were narrowed with reduction of the ambient pressure. These two factors combine to narrow the frequency range used for each resonant peak at lower pressures, as shown in Figure 8.

The FRFs analyzed in this investigation were calculated by considering the data only at the excitation frequency and stepping through the frequency range. For an ideally linear structure, it is expected that the steady-state response only exhibit the excitation frequency. However, nonlinear structures may shift the energy to different frequencies, which means that the response may
Figure 8: Transmissibilities at various pressures of the first mode for a 400 µm cantilever beam with a (a) 6.3 µm gap height and (b) 2 µm gap height.

Figure 9: Amplitude-dependent nonlinearities are evident only at very low pressures.

As seen in Figure 8, the natural frequencies do not vary significantly at different pressures, indicating that stiffness and mass characteristics seem to be pressure invariant regardless of the nonlinear gas forces. Also, the peaks did not lean towards higher or lower frequencies, meaning that no jump-discontinuity is present. In addition, the transmissibilities from two different input levels were seen to yield very small difference in the amplitude typically. The only evidence of nonlinear behavior was seen in the second resonance of the 500 µm beam with a 2 µm nominal gap height at the lowest pressure used. Figure 9 shows that the transmissibility magnitude for that beam changes significantly with the excitation amplitude. In addition, the response to the higher excitation level exhibits the jump phenomenon.

To nominally capture the energy dissipation effect of the gas forces, the effective viscous damping ratios from the data were determined using a linear curve-fit for various beams at a range of pressures. Since the nonlinear effects that could have been captured by this FRF data were minimal, this was a reasonable approach. The results are shown in Figures 10 and 11.
It is clearly seen from these plots that longer beams have higher damping and that this is true regardless of the ambient pressure. The strong dependence of the damping ratios on ambient pressure indicates that the energy dissipation arises almost completely from the gas forces. Another observation is that the behavior of the damping ratio with pressure, as seen by the shape of the fitted curves, is similar for different modes and beam geometries. Also, the damping of the second resonances shown in Figure 11 is significantly less than that of the first resonance for the same beams. Interestingly, the damping ratio measured in these tests does not find a plateau at low pressures as has been reported by other authors [8, 13]. Thus we cannot make the claim that, at the lowest pressure, the effects of gas-damping are minimized and the structural damping of the beams becomes more dominant, as has been done in the past. However, we can conclude that the structural damping of these micro-cantilevers is less than 0.01% of critical, since it appears that the trend of the tests would indicate continued decreases in damping with further decreases in pressure.

5 SUMMARY AND CONCLUSIONS

An experimental facility for testing nonlinear structural dynamics and coupled-physics phenomena for microsystems is presented. A miniature shaker that is composed of a piezo-actuator, a seismic mass, and a suspension mechanism is designed to excite MEMS modules using the base excitation technique. The measurements are conducted with an LDV coupled to a microscope.
A vacuum chamber that encloses the miniature shaker and the test piece was also designed to allow varying ambient pressures. The main goal of the design has been to eliminate the influence of the apparatus dynamics and external noise on the modal response of the test article. Experimental studies attempting to further understanding of gas damping effects in MEMS have been conducted. The results presented illustrate the major dependence of damping in MEMS on geometry and environmental pressure.

The following specific conclusions can be stated for this work:

- The test facility designed here allows effective dynamic testing of MEMS components within 1 kHz - 100 kHz frequency range for pressures between 101 kPa and 0.4 Pa.
- Deduced from the low pressure data, structural damping in surface micro-machined polysilicon-based Microsystems is much lower than that for most conventional (macro) systems.
- Gas damping, arising from air drag and the squeeze-film effect, was experimentally observed to be the main mechanism of energy dissipation for Microsystems under consideration (when there exists vertical motion in close proximity to the “ground”) at atmospheric ambient pressure.

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