Modeling of Acoustic-Structural Coupling in a MEMS Hydrophone

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ABSTRACT

The coupling of acoustics and structural dynamics is of primary importance in the design of MEMS based microphones for use in liquid environments. A three-dimensional finite element study of acoustic-structural coupling is presented here for such a MEMS hydrophone system. The experimental system under consideration, currently under development at Boston University, consists of a horn to amplify incoming sound, a sub-micron thin plate resonator, and a Helmholtz resonator cavity to amplify the measurable response. The finite element results, which consist of steady state pressure fields in the fluid medium and steady state displacements of the resonating plate, compare favorably to experimental results. The effects of acoustic and structural resonances in each of the components of the system are explored, and it is shown that a nonuniform composite plate resonator provides a smoother broadband response in the frequency range of interest than a uniform plate.

Keywords: Mems, acoustics, hydrophone, resonator, silicon

1 INTRODUCTION

The work presented here is part of the design process for developing an acoustical sensor at Boston University using microelectromechanical systems (MEMS) technology. The hydrophone, shown in Figure 1, is used to sense acoustical pressure fluctuations by direct optical measurement of a vibrating sub-micron thin plate immersed in water. The motion of the plate is intended to follow the motion of the fluid, so structural physical phenomena such as plate resonances should be avoided in the operating frequency range of 1MHz to 3MHz. Thus, the understanding of fluid-structural coupling effects on a vibrating plate in liquid is of major importance in the design of an underwater microphone of this kind.

The device under consideration is geometrically complex, and the equations governing the dynamics of the fluid and structural components of the system are fully coupled. These complications directly influence the behavior of the system, particularly in the desired operating environment, where the acoustical wavelengths and the geometrical dimensions are of same order of magnitude, and the frequencies to be measured are on the order of the structural and acoustical natural frequencies of the system. Since analytical models are inadequate in this regime, a finite element model is developed, using a commercial software package.

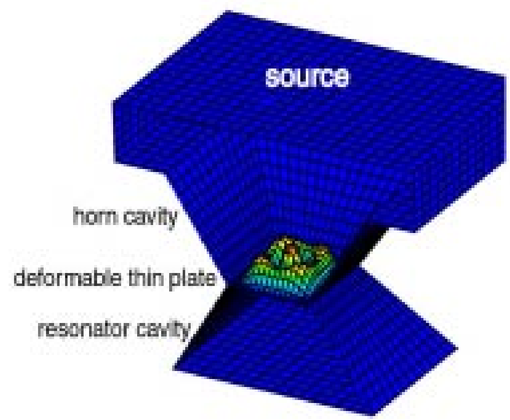

Figure 1: Schematic of the MEMS hydrophone

Although there has apparently been little finite element analysis of acoustical MEMS applications, there have been some computational studies in recent years of other fluidic MEMS devices.[1],[2] Finite element simulations are common for large scale coupled structural and acoustics problems,[3],[4] and the radiation of sound has been extensively investigated; underwater acoustics studies include results on the radiation of sound emitting devices such as piston shaped transducers.[5] However, the response of immersed structures subjected to acoustical excitation for use as pressure sensors has not been considered.

This paper presents an analysis of the complete hydrophone system, including effects of the plate resonator as well as the horn and Helmholtz resonator cavity, both of which are intended to amplify the measurable response of the plate. The numerical results compare favorably to experimental data. A system based on a nonhomogeneous plate is then proposed as a means to avoid the undesirable effects of structural resonances in the frequency range of interest.

2 EXPERIMENTAL AND COMPUTATIONAL SETUP

The system to be modeled is fabricated experimentally[6] by etching preferentially through a Si wafer bonded to a sub-micron Si_3N_4 layer on the opposite side. The newly created free surfaces can be used alternatively as a horn for an incoming acoustical signal, or, if sealed and turned over, as a Helmholtz resonator cavity. Figure 2 shows cross-sectional views of the "horn" configuration, the "resonator" configuration, and the "plate" configuration, which are combined to form the "complete device" configuration. To make contact with the experiments, each of these configurations is modeled using the finite element program ABAQUS.[7]

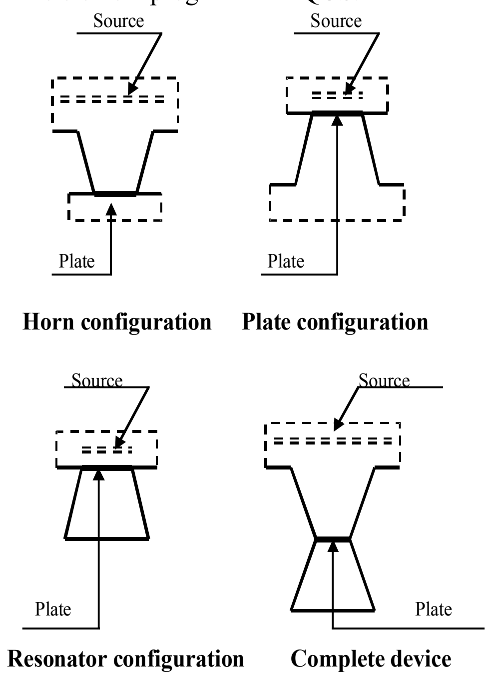

Figure 2. – Cross sectional views of the system

The acoustic-structural finite element analysis of the system requires the solution of the equation governing displacements in the fluid, given by

$$ {}^2u^f + \frac{\rho^f}{K}\ddot{u}^f = 0 $$

where u^f is the fluid displacement, ρ^f is the fluid density, and K is the bulk modulus of the fluid. This equation is coupled to the solution for the transverse displacements of the thin plate resonator, governed by

$$ \frac{Eh\char`^3}{12(1-\upsilon^2)}\ {}^4u^p + \rho^p h\ddot{u}^p = -q $$

where E and ν are the plate elastic constants, u^p is the transverse displacement of the plate, ρ^p is the density of the plate, h is the thickness of the plate, and q is the force per unit area acting on the plate. The load q acting on the plate is due to the fluid pressure difference across the plate, given by

$$ q = -K\left(\frac{fu^{f+}}{fx} - \frac{fu^{f-}}{fx}\right) $$

Furthermore, the plate and fluid particle accelerations are continuous on the interface, so that

$$ n\,?\ddot{u}^f = n\,?\ddot{u}^p $$

while on the other walls of the system, which are considered rigid and acoustically reflective, the fluid acceleration is zero. The single dashed lines in Figure 2 represent open, or dissipative, acoustical boundaries, from which there is no reflection of normally incident plane waves. Finally, the double dashed line represents the source, at which a series of point volume accelerations are prescribed.

The effects of material and structural damping are not considered. The material response in the plate is assumed to be linearly elastic, and the fluid is treated as inviscid and compressible. Some error is expected in the modeling of the plane wave source as a series of point sources. Furthermore, some interference effects between the source and the system would be expected when the source is separated from the system by approximately half of one wavelength. A related limitation comes from the nonreflecting boundaries surrounding the fluid domain outside the device.

3 RESULTS

3.1 Plate without horn or resonator

A uniform plate hydrophone, in the plate configuration shown in Figure 2, is first considered. The square plate consists of a 0.5µm thick layer of Si_3N_4. A range of plate widths is considered, from 20µm to 45µm. Computed and experimental resonance frequencies for the plates are in close agreement, as shown in Figure 3. Furthermore, the results are consistent with the analytical model of Morfey[8], which accounts for the effect of the fluid on the plate as that of an added mass. This leads to the conclusion that the plate resonances should vary with plate width as $r^{-5/2}$. For comparison, a vibrating plate in vacuum is characterized by resonances that vary with plate width as r^{-2}.

Individual response spectra of the plates further illustrate the agreement between computation and experiment. The measured and predicted response of a 34µm wide plate, for example, is shown in Figure 4. The difference in the displacement amplitudes between computation and experiment is reasonable given the uncertainty in the amplitude of the experimental sound source. Furthermore, all sources of material and structural damping are neglected in the computation. Thus, calculated amplitudes at resonance are expected to be larger than experimentally observed amplitudes.

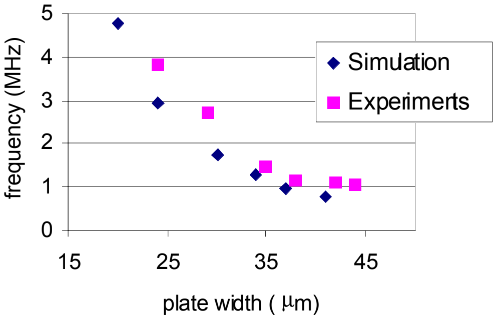

Figure 3. - Lowest mode plate resonance frequency as a function of plate length.

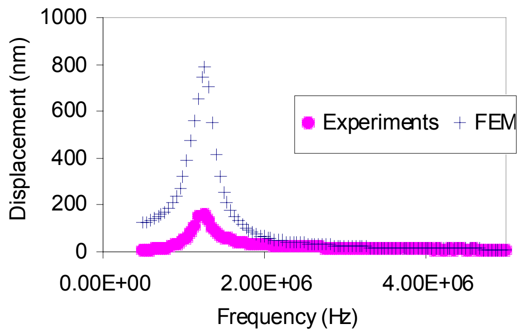

Figure 4. – Midpoint displacement of a 34 µm plate as a function of source frequency.

3.2 Effect of the horn

In the device operating range, in which the acoustical wavelength is of the same magnitude as the physical length scales of the structure, no analytical result is available for predicting the effects of the horn. In the small wavelength limit, the amplification or gain effect, *n*, of the horn on fluid particle displacement would be given by

$$n = \sqrt{h/p}$$

where *h* is the horn opening width and *p* is the plate width. In the present case, however, the horn gain is expected to be smaller than in the large structure limit, but to increase with increasing acoustical frequency.

To estimate the horn gain effect, plate displacement is studied for the horn configuration as shown in Figure 2. For fixed plate size a range of systems, with horn depths varying from 30µm to 70µm, is considered. Figure 5 shows both that plate displacement at a given excitation frequency increases with horn depth, and that the gain effect is more pronounced at higher frequencies, as expected. At 5MHz, for example, plate displacement for a system with a 70µm horn is approximately three times that of a system with a 30 µm horn.

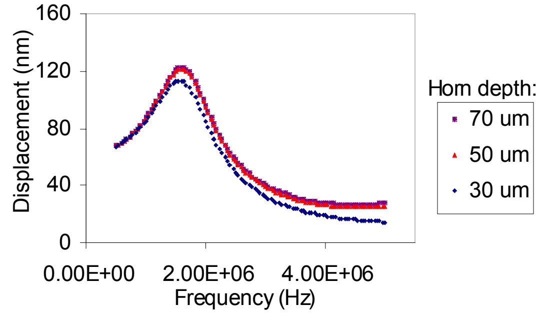

Figure 5. – Midpoint displacement of a 30 µm wide plate as a function of horn depth.

3.3 Effect of the resonator

The resonance of an acoustic cavity behind the thin plate is used experimentally to further amplify the measurable displacement. This classical Helmholtz resonance effect is studied using the resonator configuration shown in Figure 2. A Helmholtz resonator, which is typically a rigid-walled fluid-filled container isolated from the external environment except through a narrow opening, or neck, has a resonant frequency given by[9]

$$f = c_0 \sqrt{A/lV}$$

where c_0 is the sound velocity, *A* is the area of the aperture, *V* is the volume of the fluid contained in the cavity, and *l* is the neck length. The resonant frequency thus varies as $V^{1/2}$ if the neck length remains constant. To investigate the presence of this effect, a series of resonators of varying depth is studied for a 30µm wide plate system. Figure 6 shows the dependence of resonance frequency on resonator volume, based on computations and the analytical result given above. The computed resonant frequencies vary roughly as $V^{-0.14}$. This deviation from classical Helmholtz resonator behavior is likely due to the effect of the plate on the motion of the fluid.

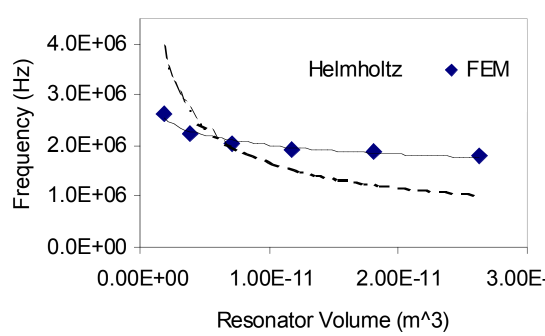

Figure 6. – System resonant frequency as a function of resonator size. Cavity depth varies from 60-160µm.

3.4 Effect of a nonuniform plate

In the frequency range of interest, the behavior of the experimental hydrophone system is clearly marked by resonant frequencies of the plate. This is undesirable, since a smooth broadband frequency response is needed, and because nonuniform deformation due to excited modes creates optical sensing problems. Thus, the use of a nonuniform plate, consisting of layers of Si_3N_4, Si, and Au, is proposed. (Figure 7) The thin Si_3N_4 edges lead to a flat deformed shape, reduce the fundamental mode frequency of the plate, and increase the resonant frequencies of the higher modes. Furthermore, the reflectivity of the Au layer aids in the optical detection of plate displacements.

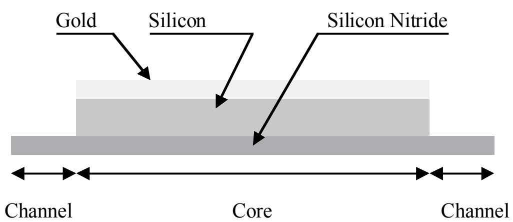

Figure 7. – Cross-sectional view of the square nonuniform plate.

The deformed shape of the nonuniform plate at low frequencies, which is dominated by the fundamental mode, is shown in Figure 8. This resonance is below the 1MHz to 3MHz operating frequency range, as shown in Figure 9. Above this range, higher modes are excited in the plate.

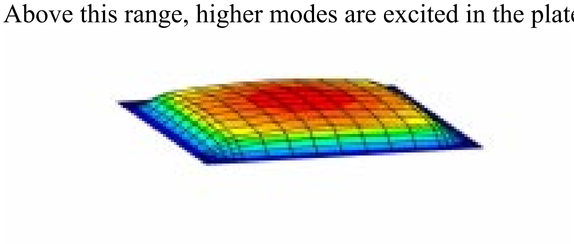

Figure 8. – Deformed shape of the nonuniform plate with 10μm channel and 50μm core at 1.7MHz.

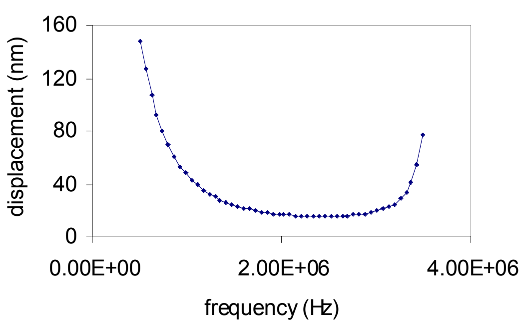

Figure 9. – Midpoint displacement of the nonuniform plate with 25μm channel and 50μm core as a function of source frequency.

4 CONCLUSIONS

The response of a MEMS hydrophone system to an acoustical excitation is numerically studied using steady-state finite element analysis. Comparisons to available data show close agreement with experiments. Because analytical results are not available for the case where wavelengths are on the order of structural length scales, the effects of an amplifying horn and an acoustical resonator cavity are considered. Finally, the use of a nonuniform plate resonator is proposed in order to avoid excitation of higher modes in the plate and to provide a smooth, broadband response.

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