Finite Element Analysis of CMUTs with Pressurized Cavities

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Abstract – We propose using CMUTs with pressurized cavities in environments with extreme pressure variations. By controlling the pressure inside the cavity, the pressure differential across the CMUT plate can be kept low, ensuring a stable operating point and preventing mechanical failure. In such CMUTs, a squeeze film is formed between the plate and the substrate, which provides additional damping as well as stiffening. The damping from the squeeze film helps increase the bandwidth of the CMUT. We present a new method for performing a finite element analysis for such structures using ANSYS. We fabricated a variety of vented CMUTs in the frequency range of ~100-200 kHz, which exhibited a quality factor of 25-30 in air at 1 atm pressure. Our finite element model successfully predicts the center frequency and quality factor for these devices.

Keywords: CMUT, FEA, varying pressure, squeeze-film damping

I. INTRODUCTION

Capacitive Micromachined Ultrasound Transducers (CMUTs) are versatile devices which have applications in varied fields such as medical imaging, ultrasonic flow metering, ranging, chemical sensing, etc. Some of these applications, *e.g.* flow metering often involve operation in a varying ambient pressure. Conventional CMUTs are inherently unsuitable for operating in such environments. A conventional CMUT consists of a fixed substrate (bottom electrode) and a moving plate (top electrode). The cavity between the plate and the bottom substrate is hermetically sealed during fabrication of the CMUT, typically under vacuum [1]. The vacuum cavity offers the advantage of reduced loading and increased transduction efficiency, however having a fixed pressure inside the cavity limits the operating pressure range for the CMUT. The static deflection of the CMUT plate depends on the pressure differential across it and the transmit and receive sensitivity of the CMUT depend on this static deflection. In a varying ambient pressure, the static deflection would vary considerably, and the CMUT would have an unstable operating point.

To overcome this problem, M.-C. Ho, *et. al.* proposed operating CMUTs in a permanent contact mode even under 1 atm pressure [2]. This would enable a more stable operating point over a wider operating pressure range. However, even such a CMUT would still be limited by the mechanical strength of the structure. Beyond a certain pressure, such a CMUT would fail mechanically.

We propose making CMUTs with cavities which are vented to an external fluid source. The cavity could simply be vented to the ambient environment thus ensuring a zero differential pressure across the plate, or it could be pressurized using a variety of gases / liquids, with the pressure being controlled independently. The pressure differential can be kept constant irrespective of the absolute ambient pressure. This will ensure a stable operating point for the CMUT. Also, with this approach the pressure differential across the plate will be limited and hence such a CMUT should be able to operate in any pressure condition.

In such vented CMUTs, a fluid squeeze film will be formed inside the cavity. This squeeze film will provide additional stiffening as well as damping for the plate. The additional damping will help increase the fractional bandwidth at the cost of some loss of transmit and receive sensitivity. An improved bandwidth would be especially useful for applications like airborne ultrasound imaging. Also in flow metering applications, a broad bandwidth would help relax the frequency matching requirement for the transmit and receive transducers.

Squeeze films are often seen in MEMS devices like accelerometers [3], micromirrors, etc. In such cases, the entire moving structure is usually vented to ambient pressure at the edges. However in a CMUT, most of the cavity is sealed at the edges by the post supporting the plate and the cavity needs to be vented through channels made through the side (post) or through the substrate [Fig. 1]. The location, number and size of these channels will affect the stiffening and damping, and can be optimized for the design of such CMUTs.

II. SQUEEZE FILM THEORY

The squeeze film theory comes into picture whenever there is squeezing of a thin fluid film between two surfaces moving normal to each other. Reynolds equation from lubrication theory is used to analyze the fluid structure interaction when squeeze films are involved [4,5].

$$\frac{\partial(\rho d)}{\partial t} = \nabla . \left(\frac{\rho d^3}{12\eta} \nabla \mathrm{P} \right)$$

Where,
d = local gap separation
ρ = density
t = time
η = dynamic viscosity

P = absolute pressure

For an ideal gas $\rho = \frac{P}{RT}$

For an isothermal case we get

$$\frac{\partial(Pd)}{\partial t} = \nabla.\left(\frac{Pd^3}{12\eta}\nabla P\right)$$

This equation is then solved together with the governing equations for the structure, typically using finite element analysis.

III. Finite element model

Finite element analysis is a powerful tool for simulating devices like CMUTs where multiple physics are involved. Several commercial software packages are available for performing finite element analysis for CMUTs. We chose ANSYS14.0 (ANSYS Inc., Canonsburg, PA) since it provides all the required elements for this analysis, and since it provides a scripting language which makes it easier to automate many tasks and perform parametric modeling.

In ANSYS the FLUID136 element is used to model viscous fluid flow in thin squeeze films. This is a 3-D element based on the generalized Reynolds equation and has pressure and displacement degrees-of-freedom (DOFs). For analyzing the effect of the squeeze film, either a transient or harmonic analysis can be performed. However a transient analysis requires significantly more computing resources, especially since we have to analyze a full 3-D model. Hence we perform a harmonic analysis to analyze the stiffening and damping effect of the squeeze film. The FLUID136 elements can only be used with a pressure DOF in a harmonic analysis. This necessitates separation of the squeeze film model from the structural model since the displacement DOFs from the structural elements cannot be coupled with the squeeze film elements. ANSYS recommends using a modal projection method for such analyses [6]. In this method, a modal analysis is performed on the structure and the resulting eigenvectors are impressed on the squeeze film elements to compute the pressure in the squeeze film. The stiffness and damping parameters are derived from this pressure distribution and applied back to the structure. In the case of CMUTs the model includes fluid elements representing the medium, and a modal analysis cannot be performed on fluid elements. To overcome this, we separate the model and perform harmonic analyses on both models instead of a modal analysis.

We build a full 3-D model for the CMUT plate using SOLID186 elements. The fluid medium is represented by FLUID30 and FLUID130 elements. After the plate and the medium are meshed, the FLUID136 squeeze film elements are overlaid on the bottom surface of the plate. The lumped transducer TRANS126 elements which are used to model the electrostatics are defined at the bottom nodes of the plate. Along with the transducer elements, spring-damper COMBIN14 elements are also defined at the bottom nodes of

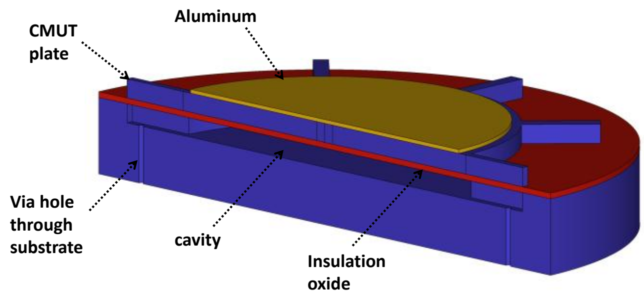

Fig. 1 3-D cross section schematic of vented CMUT structure

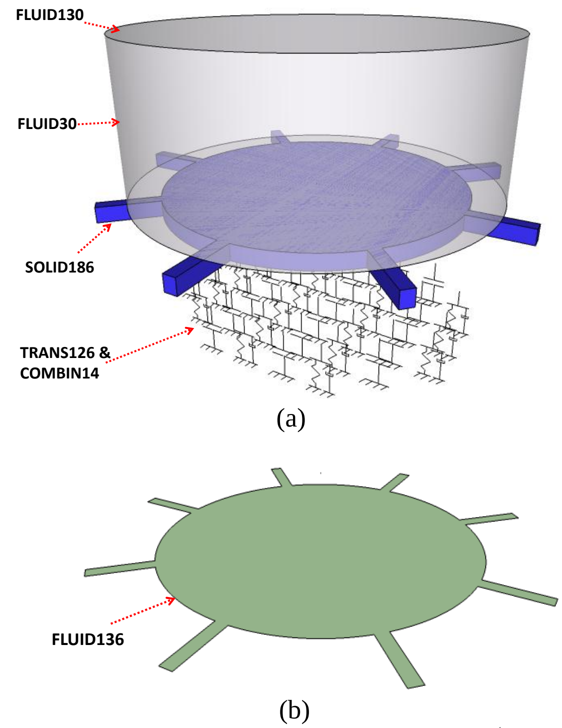

Fig. 2 Schematic of the ANSYS finite element models (a) 1st sub-model which includes the CMUT plate, fluid medium, transducers and spring-damper elements (b) 2nd sub-model which consists of the squeeze film elements overlaid on the bottom of the CMUT plate

the plate to represent the stiffening and damping effect of the squeeze film on the plate. Although a single finite element model is built, the squeeze film elements are analyzed separately from the rest of the elements [Fig 2(a), (b)].

We begin with a static analysis to find the deformation of the plate under the biasing voltage and any pressure differential. Then a pre-stressed linear harmonic analysis is performed on the first sub-model at a single frequency, excluding the squeeze film elements. The velocities of the plate nodes are then applied as fluence loads on to the squeeze film elements. Next, a harmonic analysis is performed on the second sub-model which consists of the squeeze film elements to find the pressure distribution. The real and imaginary forces at the nodes are computed from the pressure distribution and are converted into equivalent damping and stiffness coefficients.

$$C = \frac{F^{Re}}{v_z} \qquad K = \frac{F^{Im}.\omega}{v_z}$$

The spring-damper elements in the first sub-model are then updated using these values, and the first sub-model is analyzed again. The harmonic analyses for the two models are repeated till the damping and stiffness coefficients converge. Typically the damping and stiffness coefficient at each node converge to within 2% in less than 5 iterations. This process is then repeated for other frequencies to determine the center frequency and bandwidth of the CMUT.

IV. VENTED CMUT FABRICATION

To verify our finite element model, we fabricated a variety of CMUTs with vented cavities, with varying plate radius, plate thickness and cavity. The CMUTs were designed to be in the ~100-200 kHz range which would make them suitable for both airborne imaging as well as flow metering. The CMUTs have vent channels on the sides which are exposed by etching vias through the substrate [Fig. 3]. These vias allow us to introduce different gases inside the cavity as well as to control the pressure inside the cavity independent of the ambient pressure.

V. FEM VERIFICATION AND DISCUSSION

In our initial characterization of these CMUTs, we kept the cavities vented to the ambient air, and took all measurements under 1 atm pressure. The devices were excited by ac voltage (continuous wave mode), which was superposed on a dc bias voltage. The frequency of the ac excitation was varied and the amplitude of the CMUT vibration was measured using a laser Doppler vibrometer (LDV). These measurements were compared to the vibration amplitude predicted by our finite element model. Three of the designs with a plate thickness of 20 µm, gap of 3.5 µm and radii varying from 750 µm to 1500 µm were measured. As seen in fig. 4, the measurements show very good agreement with the finite element simulations in terms of center frequency and fractional bandwidth. The center frequency of the vented CMUTs differs significantly from unvented CMUTs of similar size. It is observed that the stress stiffening effect depends heavily on the gap height. For CMUTs with similar gap height, the effect of stress stiffening on the center frequency is more dominant on the larger, low frequency devices, as can be seen in table I.

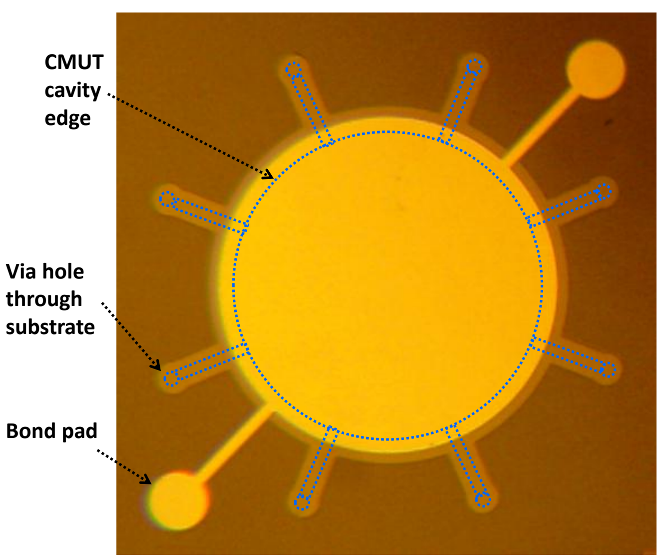

Fig. 3 Optical picture of a vented CMUT

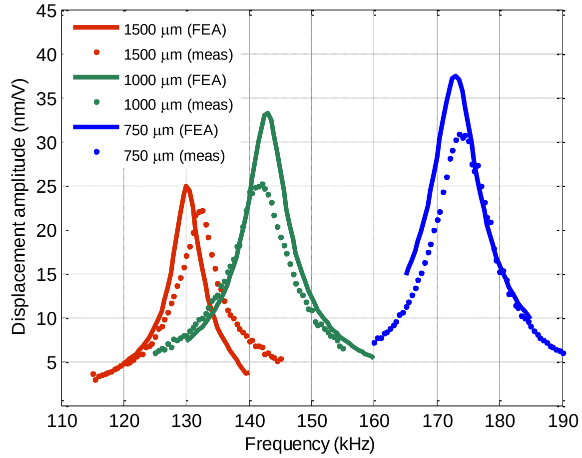

Fig. 4 vented CMUT amplitude measurements and comparison to FEA

TABLE I

CENTER FREQUENCIES OF CMUTS WITH 20 µm THICK PLATE AND 3.5 µm GAP AT 1 ATM AMBIENT PRESSURE, COMPUTED USING FEA

	Plate radius		
	1500 µm	1000 µm	750 µm
Vacuum in cavity	34 kHz	76 kHz	135 kHz
Cavity vented to ambient air	132 kHz	143 kHz	173 kHz

The CMUTs with vented cavity are designed to operate in a varying ambient pressure, while controlling the cavity pressure. To understand the effect of the ambient medium and the squeeze film, we simulated a CMUT by independently varying the ambient pressure and the cavity pressure. We used a CMUT design with a 19 µm thick plate, 750 µm radius, and 20 µm gap. In fig. 5(a) we kept the cavity pressure fixed at 2 atm and varied the medium pressure from 1 atm to 3 atm. Under a pressure differential of 1 atm, the plate deflects by 5.8 µm at the center. For a conventional CMUT with evacuated cavities, such a deflection would cause an appreciable change in the response of the CMUT. However, with a pressurized cavity the CMUT response stays more stable. The stable response is due to two balanced competing mechanisms; the higher ambient pressure decreases the gap height, thus increasing the electric field, but the higher electric field gets nullified by the increased squeeze film damping. This means

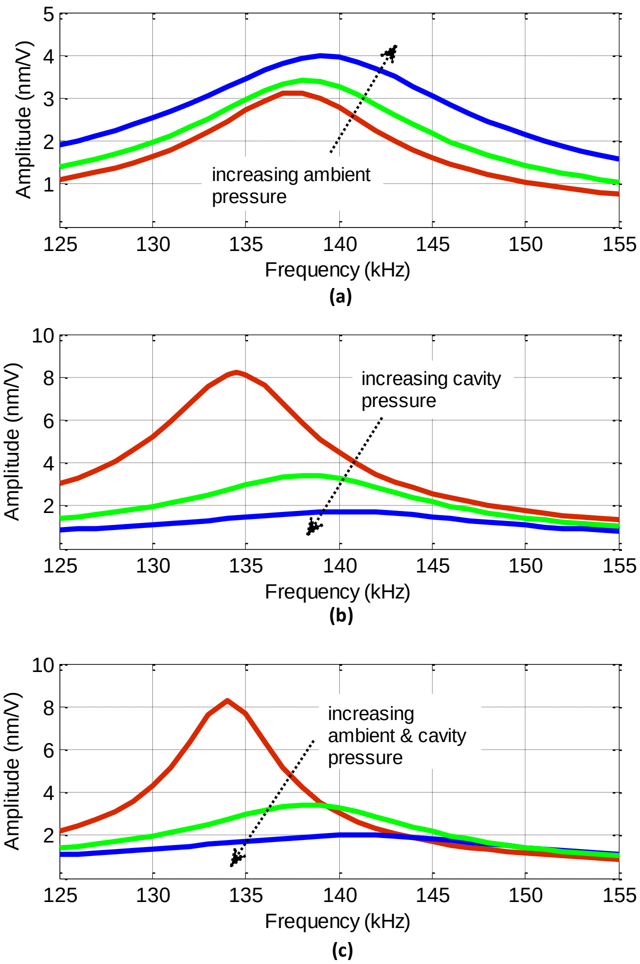

Fig. 5. Response of a vented CMUT with 20 µm gap under varying ambient and cavity pressure (a) constant cavity pressure (2 atm) and increasing ambient pressure (1-3 atm) (b) constant ambient pressure (2 atm) and varying cavity pressure (1-3 atm) (c) ambient pressure = cavity pressure varying from 1 atm to 3 atm

that for small enough pressure variations, the cavity pressure can be fixed at a certain value during operation. Obviously, for large pressure variations, the cavity pressure cannot be kept constant, and needs to vary with the medium pressure.

Next we varied the cavity pressure from 1 atm to 3 atm under a constant ambient pressure of 2 atm. The CMUT response changes significantly on changing the cavity pressure as seen in fig. 5(b). An increased cavity pressure not only causes a loss in the electric field by deflecting the plate, but it also increases the squeeze film damping and stiffening effect. We also simulated a CMUT with its cavity vented to ambient air, such that there is no pressure differential across the plate [Fig. 5(c)]. We analyzed this CMUT from 1 atm to 3 atm. Comparing this to the previous case of [Fig. 5(b)], it is obvious that the CMUT response is dominated by the squeeze film.

VI. Conclusion

We have presented a method for using CMUTs in varying pressure conditions by pressurizing the cavities. A method for analyzing such CMUTs using finite element analysis is also presented. The finite elements simulation results match well with measurements and the CMUTs exhibit squeeze film damping and stiffening when vented to ambient pressure. Further these devices need to be characterized under varying pressures and with different gases.

Acknowledgments

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