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MULTIPHYSICS ANALYSES OF THE EFFECT OF PACKAGE ON THE PERFORMANCES OF PMUT TRANSDUCERS

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Abstract. The paper deals with the multiphysics modeling of piezoelectric micromachined ultrasound transducers (PMUT), that can be used in several practical applications. The model accounts for the multiple couplings between the mechanical fields and the electric and acoustic ones. The numerical solution has been sought by means of the finite element method, for the special case of axial symmetry. The model has been validated with reference to experimental data, that have been obtained by the Authors. The numerical procedure has been applied to carry out a parametric analysis of the effect of package, to extract a set of design guidelines.

1 INTRODUCTION

The application of piezoelectric materials in "smart" microdevices is continuously increasing, with different possible uses of both "direct" (conversion of mechanical energy into electric energy) and "indirect" effect. The latter is applied for actuating purposes, e.g. in the case of micropumps [1]; "direct" effect is now widely used for energy harvesting, namely for obtaining an electric power by exploiting some freely available mechanical energy [2,3].

In the case of MEMS, some recent developments in microfabrication techniques allowed for the introduction of layered structures with thin films of piezoelectric materials (namely, PZT or AlN [4]). After a thorough examination of piezoelectric mechanical properties for thin layers, it is possible to provide a comparative table in terms of "Figure Of Merits" (FOM). In that way, one can select the best material for each possible application, with the use of both direct and indirect effects.

The ultrasound transducers (PMUT) are piezoelectric structures which are actuated with the purpose of emitting and receiving ultrasonic waves [5]. Such devices are used in many practical application: acoustic imaging for medical purposes; hydrophones; finger-printing recognition

[6]; range-finders [7]. Further applications are represented by non-destructive testing, velocity sensing and 3D pattern recognition. The complete simulation of the device's behaviour can be obtained by considering multiple coupling between different fields: electro-mechanical coupling for piezoelectric model; acoustic-structural coupling for understanding the efficiency in the emission and in the sensing phase. The model is complicated by the fact that the PMUT devices are usually packaged, so the interaction between the surrounding medium does not involve directly the vibrating diaphragm only but the package as well.

This paper is focussed on the development of a fully-coupled model in order to carry out parametric studies on the effect of package geometry on the PMUT operation. This has been done with reference to axisymmetric models, with the main purpose of reducing the computational burden. This is perfectly suited for the emission of waves from circular diaphragm in circular package; conversely the sensing phase would require a full 3D model. The achieved results have represented a useful guidance in the design of a PMUT prototype, that has been finally subject to experimental tests. The measured data are in good agreement with the numerical predictions.

2 MULTIPHYSICS MODELLING

The finite element model has been developed in COMSOL Multiphysics 5.2. The numerical model includes three physics and their mutual interactions, namely: solid mechanics model of the membrane, piezoelectric effect in the PZT active layer and pressure acoustics model for the fluid far from the membrane. The thermo-viscous acoustics model for the fluid near to the vibrating diaphragm, that has been considered in [8], is here neglected since focus is on the far-field response.

The materials in the layers of the plate are linear elastic and they are characterized by a certain amount of residual stress in the radial direction of the plate. Such a stress field, that is induced by the fabrication processes, plays an important role in the initial deformation and in the geometric stiffness of the structure. In the piezoelectric active layer, the electro-mechanical coupling is enforced through the linearized constitutive relation (see e.g. [1]). The overall structural damping (which is induced by anchor losses, surface effects and the thermo-elastic losses) is modeled by means of an isotropic structural loss factor, that is the inverse of the structural quality factor Q_{struct}.

The acoustic phenomenon far from the emitting source can be considered as an isentropic process, characterized by zero energy dissipation. The computational burden is reduced by considering a spherical pressure acoustic domain with radius equal to 4 λ, where $\lambda = v_s/f_0$ is the typical wavelength, v_s is the sound velocity in air at the reference state (T_{ref} = 293.15 K and P_{ref} = 1 atm), f_0 is the operational frequency of the system, that coincides with the plate fundamental frequency in order to maximize the acoustic efficiency. In this region the lossy Helmholtz equation for the pressure acoustic field is solved, with the Stokes' attenuation factor for the viscous and thermally conductive case [9]. Finally, outside the pressure acoustic domain, the wave radiation into an infinite medium is simulated by means of a Perfectly Matched Layer (PML) of thickness equal to λ.

The electro-thermo-structural-acoustic response is computed by means of a sequence of simulations. First, a non-linear static analysis is performed to compute the pre-deflected configuration induced by the fabrication residual stresses in the plate. An electro-mechanical static analysis, account taken of the previously computed non-linear deformation, is performed to simulate the effect of the bias voltage that is applied in the actuation phase. Finally, an electro-mechanical-acoustic analysis is performed in the frequency domain to simulate the complete system, under a harmonic voltage perturbation, taking into account the non-linearities introduced before.

3 EXPERIMENTAL VALIDATION

The results of the studies on the single PMUT, without the protecting cap and the package, are shown in the present section along with the comparison with the corresponding experimental data. To compute the pre-deflected configuration of the system, which is characterized by a very high aspect ratio (radius/thickness) of 55, an initial non-linear static analysis is performed, under the residual stresses acting in the radial direction of the plate. The numerically computed profile of the diaphragm along a radius and its comparison with the experimental measured one are presented in Fig. 1.

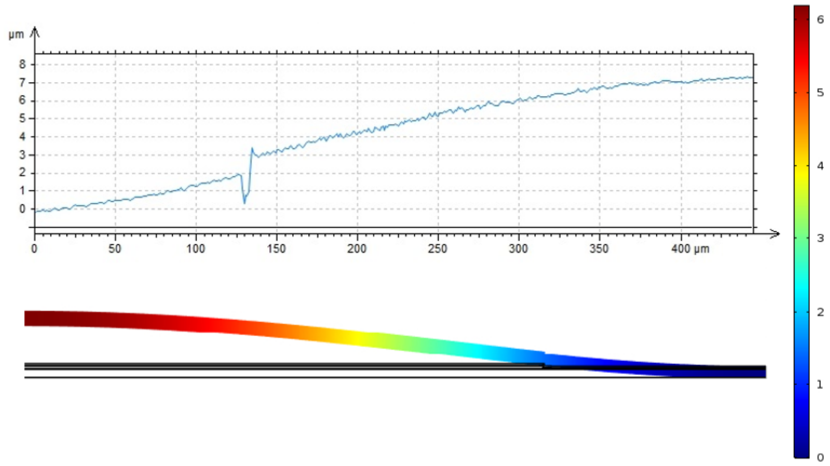

Figure 1: Initial static configuration due to fabrication residual stresses: experimental measured profile by means of MicroXAM-100 3D Surface Profilometer (above), numerically computed [µm] by means of COMSOL Multiphysics 5.2 (below).

The residual stress field determines the initial deformed shape of the layered system characterized by a maximum displacement, which occurs in the center point of the membrane, equal to 7.1 μm for the measured value, while the corresponding numerical value is 6.2 μm. The observed mismatch is attributed to the uncertainty in the experimentally measured residual stresses. The most important effect on the mechanical behavior of the vibrating system, induced by the residual stress state, is the frequency shift for the fundamental eigenfrequency of the membrane, that coincides with the operating one to maximize the acoustic efficiency in the transmission phase. As a matter of fact, the same diaphragm in terms of material and geometry, without the residual stresses, presents a theoretical fundamental eigenfrequency of 111 kHz; taking into account the contribution to the geometric stiffness due to the pre-stress state it decreases to 99.8 kHz in correspondence of an applied static voltage of 0.5 V to the piezoelectric layer of the system. This is correctly captured by the numerical model. To show the shift, a frequency spectrum analysis is performed in the presence of a static voltage of 0.5 V and a harmonic excitation with amplitude of 0.5 V. The numerical electro-mechanical-acoustic simulation runs in 49 minutes with i7 CPU @3.4 GHz and 16 GB RAM for the complete sweep over the system without the protecting package. The comparison between the non-dimensional experimental spectrum and the numerically computed one for the vertical displacement of the center point of the diaphragm is shown in Fig. 2 together with the extended frequency sweep for the packaged system represented in Fig. 3, which allows one to evaluate the frequency associated with the coupled fluid-structure mode induced by the presence of the cap. The quality factor of the device is $Q = 80$ and it is obtained by means of a free vibration decay test.

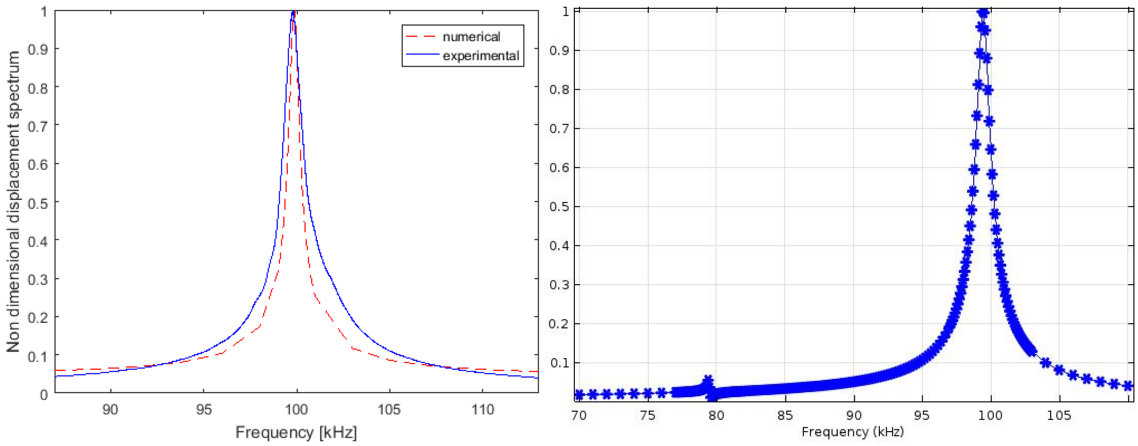

Figure 2: Normalized vertical displacement spectra for the center of the membrane: experimental measurement by means of Polytec MSA-500 Micro System Analyzer and numerical estimation by means of COMSOL Multiphysics 5.2 for the unpackaged PMUT (left), extended frequency sweep for the packaged PMUT (right).

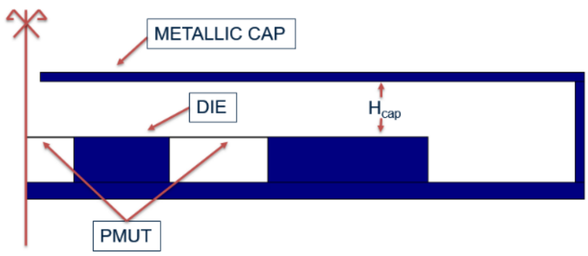

Figure 3: Sketch of the packaged system

4 ANALYSES IN THE PRESENCE OF PACKAGE

4.1 Stand-alone diaphragm vs. packaged device

The acoustic modelling of the surrounding fluid allows to evaluate the acoustic efficiency of the PMUT; the Sound Pressure Level (SPL) can be computed in every point of the fluid domain and outside of it by means of the Helmholtz-Kirchhoff integral technique (*far-field* calculation). The numerically computed SPL frequency response at 15 cm on the vertical axis of symmetry of the diaphragm, for the unpackaged transducer and for the packaged one are presented in Fig. 4. The protecting cap is characterized by a distance from the inner die equal to $H_{cap} = 500\ \mu m$, by a thickness of $t_{cap} = 80\ \mu m$ and by a central hole with radius equal to $R_{hole} = R_{PMUT}/3$ placed over the central vibrating transducer, where R_{PMUT} is the radius of the PMUT (Fig. 3).

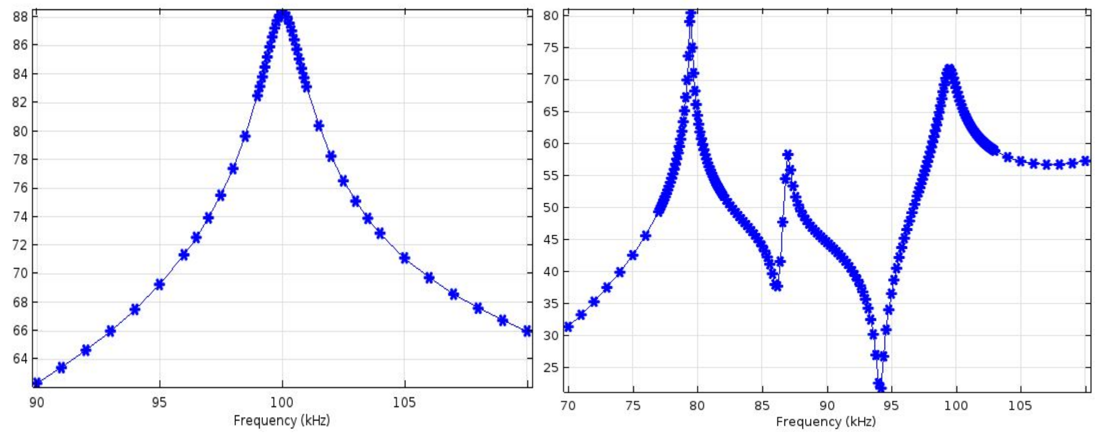

Figure 4: Sound Pressure Level [dB] at 15 cm on the vertical axis of symmetry: unpackaged transducer (left), packaged transducer (right).

It is possible to notice that the maximum value in the SPL at 15 cm occurs, for the system without the protecting cap, in correspondence of the resonance frequency of the membrane (f = 100 kHz), while in the case of the package structure it is 8 dB lower than the previous one and is related to a frequency of 79.5 kHz, which represents the eigenfrequency of the coupled fluid-structure mode associated with the adopted package geometry. Finally, for the device with the protecting cap, it is still possible to notice a local maximum at the corresponding mechanical resonance but it is 16 dB lower than the first one (barrier effect). Both maximum values in the SPL plot are related to a non-directional emitted pressure wave as we can see in Fig. 5 where the polar plot for a distance from the center of the diaphragm equal to 15 cm at the corresponding frequency is shown for the two cases together with the polar plot for the packaged device at the mechanical resonance frequency of the diaphragm.

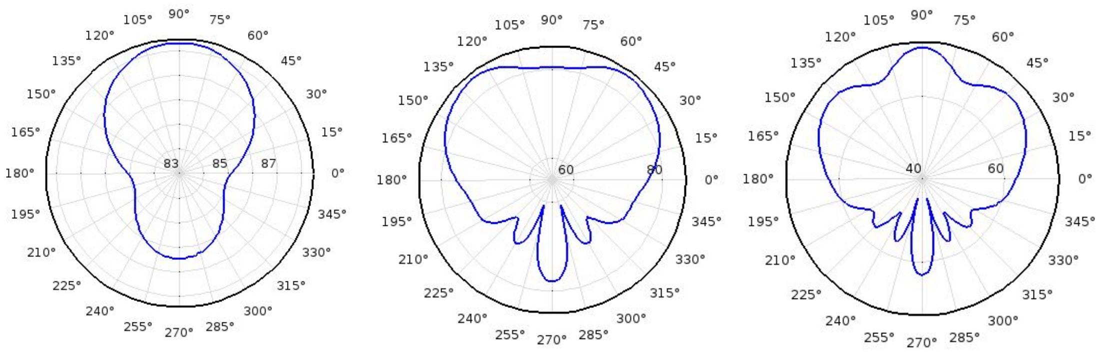

Figure 5: SPL polar plot [dB]: system without package at f = 100 kHz (left), with package at f = 79.5 kHz (central) and 100 kHz (right).

4.2 Simulations for different packages

Going further, several simulations have been performed over devices with PMUTs characterized by a resonance frequency of 63 kHz in the presence of the pre-stress state and a static voltage of 3 V. The numerical procedure has been applied to carry out a parametric analysis over the parameters H_{cap}, R_{PMUT} and t_{cap} to extract a set of design guidelines for the package structure. The results are shown in the Figures 6-11, for a reduced set of parameters, in terms of vertical displacement frequency response, under a harmonic voltage excitation from 0 to 6 V, for the center point of the membrane belonging to the axis of symmetry, SPL at 15 cm on the axisymmetric axis and polar plot at the corresponding peak value.

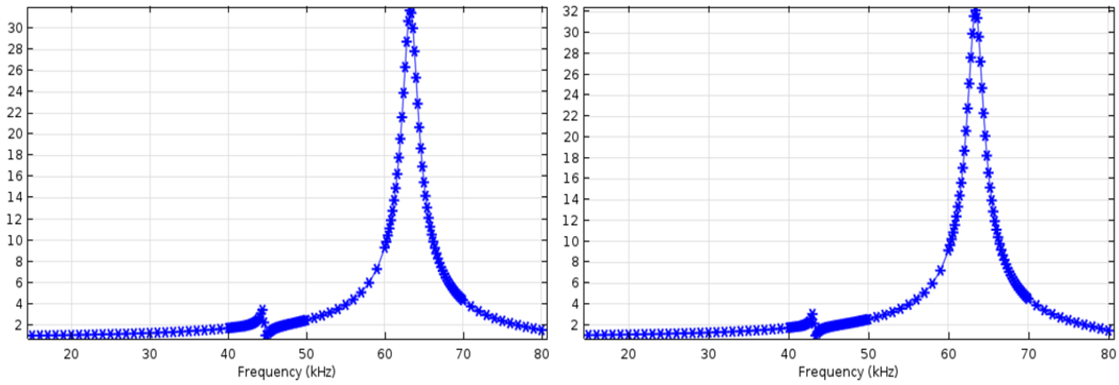

Figure 6: Vertical displacement spectra [μm] for the center point of the membrane belonging to the axis of symmetry by COMSOL Multiphysics 5.2: t_{cap} = 80 μm, R_{hole} = R_{PMUT}/3, H_{cap} = 200 μm (left), 500 μm (right).

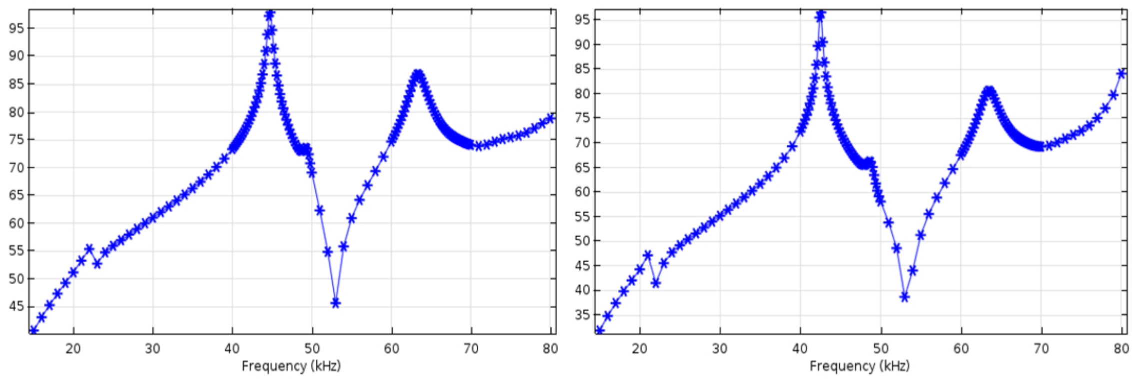

Figure 7: Sound Pressure Level [dB] at 15 cm on the vertical axis of symmetry: t_{cap} = 80 μm, R_{hole} = R_{PMUT}/3, H_{cap} = 200 μm (left), 500 μm (right).

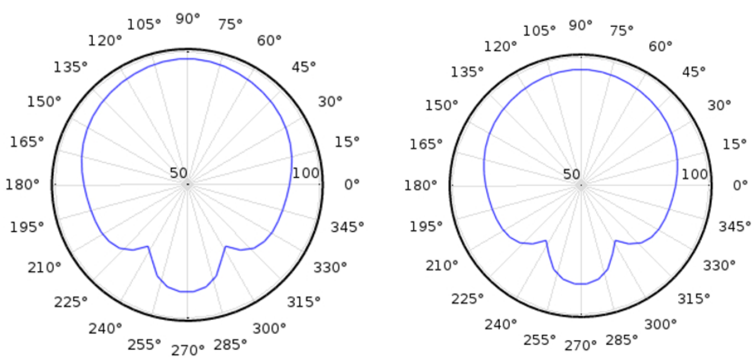

Figure 8: SPL polar plot [dB] at the corresponding peak value in the SPL versus frequency plot: t_{cap} = 80 μm, R_{hole} = R_{PMUT} /3, H_{cap} = 200 μm (left), 500 μm (right).

It is possible to notice that, in all the cases of Figs. 6-8, the peak value in the SPL versus frequency plot appears in correspondence of the frequency of the coupled fluid-structure mode, around 44.5 kHz; moreover, for H_{cap} = 200 μm (Figs. 6-8, left) the maximum in the SPL and the corresponding polar plot is slightly bigger than the case with H_{cap} = 500 μm (Figs. 6-8, right). Generally no great change occurs in the behavior of the system for the adopted package parameters. To emphasize the role of the cap structure, the radius of the hole is increased, keeping H_{cap} = 500 μm and t_{cap} = 80 μm, as it is shown in the Figures 9-11.

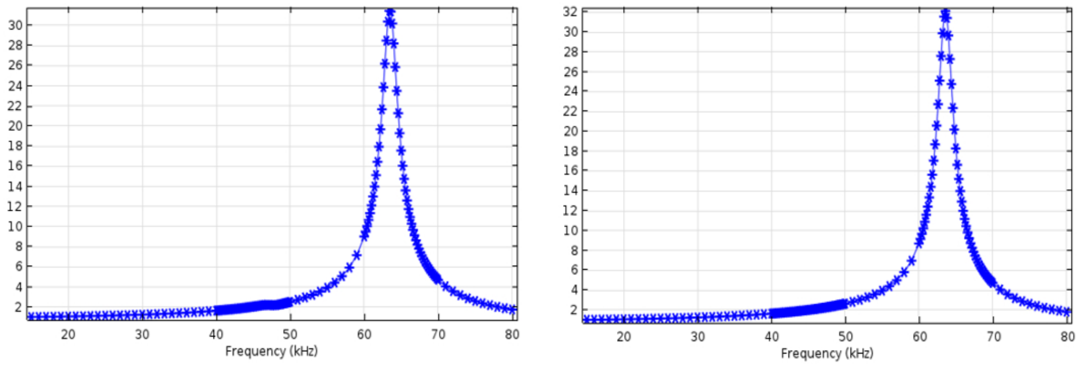

Figure 9: Vertical displacement spectra [μm] for the center point of the membrane belonging to the axis of symmetry by COMSOL Multiphysics 5.2: t_{cap} = 80 μm, H_{cap} = 500 μm, $R_{hole} = R_{PMUT}$ (left), $2R_{PMUT}$ (right).

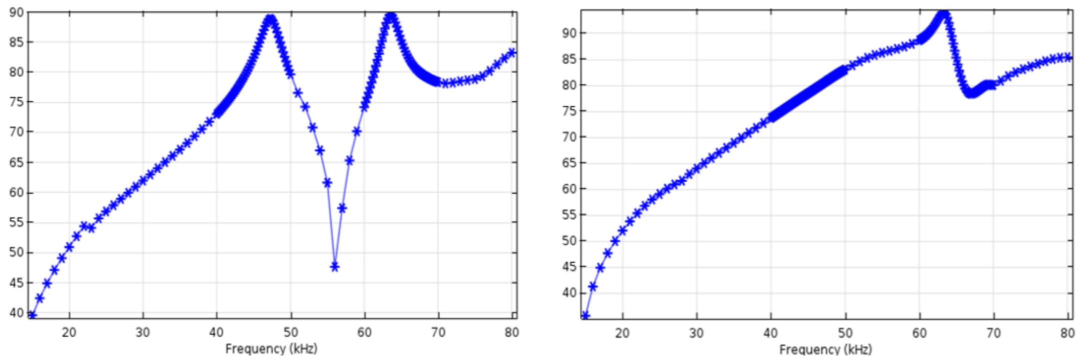

Figure 10: Sound Pressure Level [dB] at 15 cm on the vertical axis of symmetry: t_{cap} = 80 μm, H_{cap} = 500 μm, $R_{hole} = R_{PMUT}$ (left), $2R_{PMUT}$ (right).

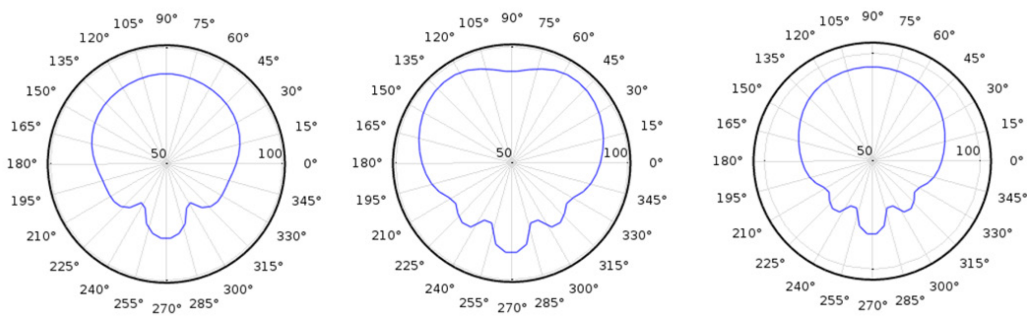

Figure 11: SPL polar plot [dB] at the corresponding peak value in the SPL versus frequency plot: t_{cap} = 80 μm, H_{cap} = 500 μm, $R_{hole} = R_{PMUT}$ at f = 47.5 kHz (left), R_{PMUT} at f = 63 kHz (central), $2R_{PMUT}$ (right).

In this case, it is possible to notice the different behavior of the device for the two values of the hole's radius. For $R_{hole} = R_{PMUT}$ in the SPL versus frequency there are two peaks for the frequency f_1 = 47.5 kHz, which is related to the coupled fluid-structure mode, and f_2 = 63 kHz, that is the fundamental mechanical frequency of the diaphragm. Furthermore, the peaks are characterized by the same maximum value but they are associated with different SPL polar shape, due to the different fluid response in the wave transmission. Finally, for the case $R_{hole} = 2\ R_{PMUT}$ the package influence over the dynamic behavior of the system is negligible; as a matter of fact, the maximum value in the SPL frequency response appears in correspondence of the mechanical resonance frequency of the membrane, as it occurred for the device without the package structure, and it is the highest registered value due to the lower barriest effect, along the positive vertical direction of the axisymmetric axis, determined by the presence of the cap.

5 CONCLUSIONS

The design of PMUTs should be based on a reliable computational model, that include the complex coupling between different physics, the non-linear behavior and the interaction between different parts of the device. In this paper, we have considered a fully-coupled multiphysics model that encompasses the mechanical, piezoelectric and acoustic fields. The effect of residual stresses has been introduced by a suitable non-linear model. The computational results for the stand-alone diaphragm have been compared to experimental data, showing a satisfactory degree of accuracy. The validated model has been used in order to assess the effect of package on the mechanical and acoustic performance of the PMUT. First, the response of the stand-alone diaphragm has been compared to the case of packaged device, with a hole that is a fraction of the PMUT radius: there is no effect on the mechanical response of the diaphragm itself, but the acoustic emission is strongly influenced by the presence of a coupled acoustic-structure mode. The sound pressure level in the far field is obviously reduced by the presence of the package.

The second step of the analyses has been represented by a parametric study for different geometric features of the package. The distance between the diaphragm and the package seems to play a minor role, since neither the mechanical response nor the sound pressure level is affected by a change of that parameter. Conversely, the variation of the radius of the hole induces a noteworthy modification of the acoustic field. In the case of small radius, the fluid-structure mode is dominant and the maximum sound pressure level can be obtained for a frequency that is far away from the resonance frequency of the diaphragm. When the radius is increased over a certain threshold, the obstructive effect of the package is reduced and the wave emission resembles to that of the stand-alone diaphragm.

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