Precise numerical simulation of electrostatically driven actuators with application to MEMS loudspeakers

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Summary. A recently developed Finite-Element (FE) model for the precise computation of electrostatic driven MEMS (Micro-Electro-Mechanical-Systems) is presented. Thereby, we fully take geometric nonlinearity, contact in the snap-in mode, nonlinear electrostatic coupling forces as well as the moving of charged thin membrane-plate structures (electrodes) in an electric field into account. Furthermore, since we concentrate on MEMS loudspeaker applications, we can also compute the radiated sound. In addition to this, an optimization method towards sound pressure level increase is presented, namely the buckling back plate method. In this method stress induced self raising is applied to increase the volume flow of the speaker.

Introduction

Micro- Electro- Mechanical- Systems (MEMS) loudspeakers, fabricated in complementary metal oxide semiconductor (CMOS) compatible technology merge energy efficient driving technology with cost economical fabrication processes. The most conventional driving technology for MEMS speaker is the electro-dynamic principle, followed by piezo speakers. A detailed overview about maximum sound pressure level of various MEMS speakers with various driving technologies can be found in [1]. The electrostatic driving concept and its optimization method is discussed in a prior work [2]. In most cases the design process of these MEMS devices is a lengthy and costly task. Therefore, the need for computer modeling tools capable of precisely simulating the multi-field interactions is increasing. The accurate modeling of such an electrostatic driven MEMS device results in a system of coupled partial differential equations (PDEs) describing the interaction between electrostatic, mechanical and the acoustic field. A finite element (FE) method is applied to the system of coupled PDEs to obtain an efficient and accurate solution. Thereby, we fully take the nonlinear effects into account. In this paper we present the three field coupled model, describing the interaction of the electrostatic-mechanical and acoustic fields.

Modeling Chain:

The full speaker modeling chain is separated into an electrostatic-mechanical and a mechanical-acoustic model, where the first model transforms an input voltage into a membrane movement with the help of the electrostatic attracting force. The second part projects the membrane movement into a propagating sound wave to compute the sound pressure level (SPL). The modeling approach for designing the MEMS speaker is depicted in Figure 1. The electrostatic mechanical model is totally modeled with ANSYS in APDL. The acoustic block is modeled with CFS++ (see [3]) for single transducer investigations and small speaker arrays up to 16 elements. In addition, a MATLAB tool solving the integral form of the wave equation is used to compute large arrays and investigate digital sound reconstruction (DSR). Detailed information about the acoustic model using Kirchhoff Helmholtz and DSR can be found in [4].

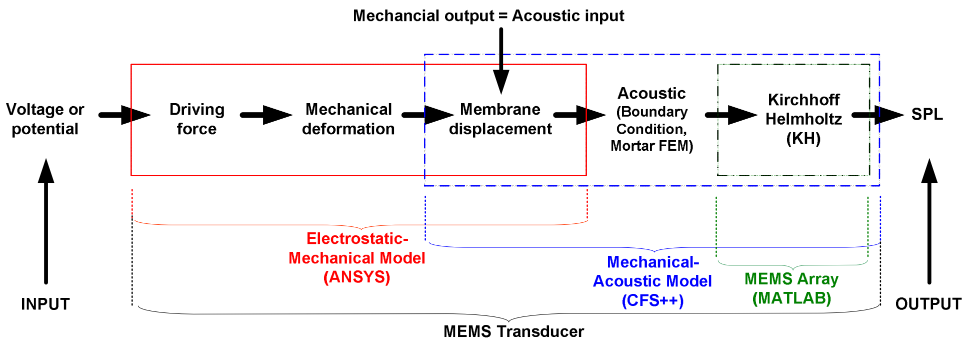

Figure 1: Major parts for modeling the MEMS speaker. Starting on the left side with an applied voltage resulting in a sound pressure level on the right side of the modeling chain.

Both models, firstly the electrostatic-mechanical and secondly the mechanical-acoustical model are separated at the membrane displacement, because the counteracting force of the surrounding air can be neglected [5]. To optimize computational time and amount of memory, the complex three dimensional structure was reduced into a two dimensional axis symmetric model. Material parameters for modeling the perforated back plate in axis rotational domains and characterization of fabrication processes was determined by Füldner [6] in a prior work on the MEMS microphone.

Modeling Methods

Electrostatic – Mechanical Model:
It is important to understand the effects of the nonlinear electrostatic forces driving the MEMS speaker. The electrostatic force is the sum of the charge forces (Coulomb forces) acting on the charged electrode and the interface force caused by the insulation layer on the membranes as described in [5, 7]. There are two commonly used implementation methods for electrostatic-mechanical coupling in ANSYS. First, the use of a reduced order electro-mechanical transducer element (EMT) like TRANS126 and second, an ansatz for separately modeling the electrostatic field and the mechanical field. The latter is not suitable for the snap-in driving mode, where the electrode is mechanically in contact with the insulation layer of the back plate and as a result FE are distorted and squeezed to zero. Additional information about the multi field method and EMT method referring this MEMS speaker can be found in [8].

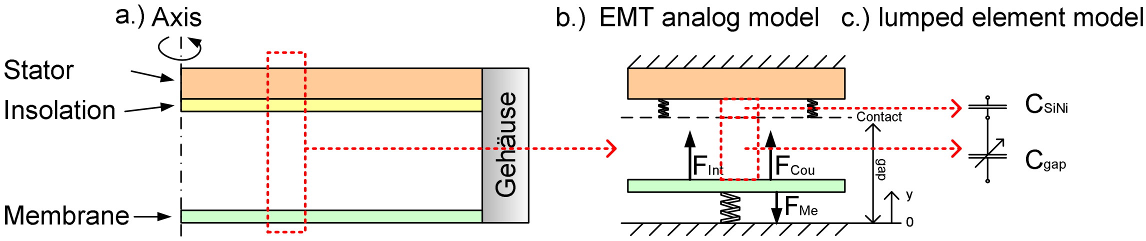

Figure 2: Simplified axis-symmetric MEMS speaker model with SiNi coated stator and flexible membrane (a) with mass spring analogy (b) and reduced order lumped element model for implementation in FEM (c).

To analyze stable and instable characteristics, a simple one-dimensional mass-spring analogy is used as analytical model, as depicted in Figure 2(b) [9]. The mechanical force is represented by a non-linear spring, and can be written as

$$F_{\mathrm{me}} = k(u)\cdot(d-u)\ , \tag{1}$$

where k is the stiffness, d the initial gap and u the displacement. As depicted in Figure 2(b), the electrostatic force can be split up to Coulomb's rate and the interface rate, where Coulomb's rate is represented by

$$F_{\mathrm{Cou}} = \frac{1}{2}C^2U^2\frac{1}{\varepsilon_{\mathrm{SiNi}}A}\ . \tag{2}$$

A comfortable way to compute the interface force is the use of the electrostatic pressure on the interface, which is represented as the normal component of the electric displacement vector, as described in more detail in [8, 10]. Tangential components can be neglected because of the geometry of the speaker, therefore the electric field intensity is homogeneous along the plate, as well as the resulting displacement is only one dimensional (in our case only in y-direction). This must be taken into account by choosing reduced order elements like TRANS126. If tangential components have to be taken into consideration, the TRANS109 element or multi-field ansatz would be the correct choice [4, 11]. Since the electrostatic pressure defines the ration between interface force and surface, we can obtain the interface force by

$$F_{\mathrm{Int}} = \frac{1}{2}C^2U^2\left(\frac{1}{\varepsilon_{\mathrm{SiNi}}A} - \frac{1}{\varepsilon_{\mathrm{Air}}A}\right), \tag{3}$$

and finally the total electrostatic force acting on the plate type capacitor can be written as the sum of (2) and (3)

$$F_{\mathrm{el}} = F_{\mathrm{Cou}} + F_{\mathrm{Int}} = \frac{1}{2}C^2U^2\left(\frac{\varepsilon_{\mathrm{SiNi}} - 2\varepsilon_{\mathrm{Air}}}{\varepsilon_{\mathrm{SiNi}}\varepsilon_{\mathrm{Air}}A}\right). \tag{4}$$

As a result, the electrostatic force is a function of the capacity C between the electrodes (see Figure 2(c)).

From mechanical field point of view, it is important to consider geometric nonlinearities like large deformation, stress stiffening and mechanical contact. Large deformation occurs, because the thickness of our membrane structure is about 330 nanometer and the displacement can be up to two micrometer, hence (4) must be computed on the deformed geometry. Stress stiffening must be taken into account, because the poly-silicon layer (electrodes) and the silicon-nitride layer (insulation) show up different intrinsic pre-stresses and constitute the mechanical set-back force (1). In the snap-in case, the membrane is rapidly forced towards the stator and abruptly stopped there by mechanical contact, which defines the last geometric nonlinearity.

Mechanical – Acoustical Model:
We consider linear acoustics, and therefore longitudinal waves, where the particles in the media are only able to move back and forth towards the propagation. If a solid is moving in a fluid it generates sound, caused by the normal component of its surface velocity. A general setup for a fluid-solid-interface (FSI) can be seen in Figure 3.

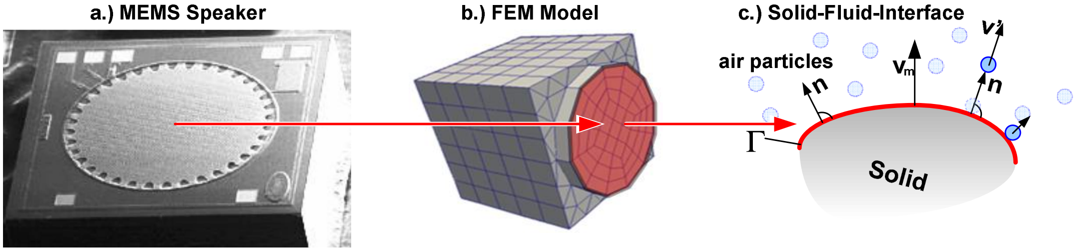

Figure 3: MEMS speaker cell (a), with FEM mesh (b) and Solid-fluid interface (Γ) for mechanical-acoustic coupling (c).

We see, that the normal component of the solid velocity $\mathbf{v}_\mathbf{m}$ and the normal component of the particle velocity **v'** are equal on the interface **Γ**

$$\mathbf{n} \cdot (\mathbf{v}_\mathbf{m} - \mathbf{v}') = 0\,. \tag{5}$$

This is called forward coupling, since mechanical displacements have an impact on the acoustic result. The impact of the air particles on the structure is called backward coupling, whereas this effect can be neglected in our case, since we are using a MEMS speaker and air as propagation medium [5].
The challenge for acoustically modeling the MEMS micro speaker with FE method is given by the small membrane size in comparison to the wavelength and the large wave propagation region. Since the MEMS structure defines the fine mesh at the interface and the wavelength the coarse mesh at the propagation region, a proper mesh size adaption routine must be used to avoid distorted elements. The total deformation ratio of one element should not get higher than 1:10 ratio [12]. Hence, there is a tradeoff between amount of elements and distortion rate. By the help of Mortar FEM using non-conforming grid technology both problems can be avoided. As depicted in Figure 4, the mechanical interface defines the smallest grid and the perfectly matched layer used to model open domain characteristics defines the largest (or coarsest) grid. Further information on Mortar FEM with non-conforming grids can be found in [13, 14].

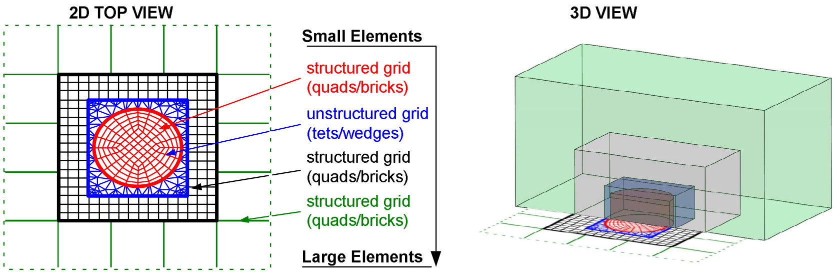

Figure 4: Non-conforming grid for mechanical-acoustic interface and stepwise adaption from fine (membrane) to coarse (open domain).

Introducing a natural or Dirichlet pressure boundary condition on the outer regions of the acoustic domain, would lead to reflections in the computational domain. Therefore, open domain treatments like absorbing boundary conditions (ABC) or perfectly matched layers (PML) must be applied. Since standard (first order) ABC only absorb normal components of impinging waves, PML are very tolerant on any impinging angles. The use of surrounding PML elements opens up the opportunity to truncate the propagation region and therefore reduces the amount of elements. Numerical tests on have shown that the PML method is more suitable for our MEMS speaker model. The encasing structures as well as the mechanical acoustic interface and the perfectly matched layer are displayed in Figure 5. Further information on PML technique can be found in [15-17]. There is also the possibility to use an infinite finite element method, which is mentioned only for completeness, additional information can be found in [18].

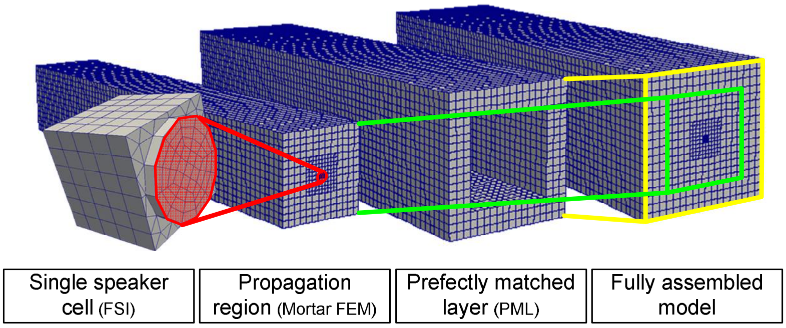

Figure 5: Mechanical acoustic model with single speaker cell and fluid structure interface (FSI), encasing propagation region assembled with Mortar FEM and PML to model open domain characteristics.

MEMS Speaker Optimization

There are three major parameters to increase the sound pressure level (SPL) for our MEMS speaker. First parameter is the frequency, where the limitation is given by human auditory, between 20 Hz and 20 KHz [19]. Second parameter is the surface area, where the limitation is given by dimensions and size as well as cost economical effects from the manufacturer side. Third parameter is the stroke level of the membrane, where the limitation is also given by manufacturer side due to technology restrictions. The challenge is to increase the gap size or stroke level with the same production steps to keep the device cheap from manufacturer side. This optimization is called buckling back plate [20] (BBP), where previously explained stress stiffening effects lead to self raising structures.

Buckling Back Plate:
The aim is to combine layers with various intrinsic pre-stress, so that the resulting MEMS speaker can be fabricated as cost economically flat design with stress induced back plate raising – hence the name buckling back plate. The discussion of this principle for thin film layers can be found in [21] and the fabrication process for our MEMS speaker is treated in [22] in more detail.

The design process of the buckling back plate (BBP) can be seen in Figure 6,. In the first MEMS draft a.) the flat structure is displayed, where the poly silicon layer is fully covered with the nitride layer. This results in a flat structure with high stiffness but little gap size. The second design process b.) shows first stages of corrugation rings. The disadvantage of this setups is the possibility of short circuits during mechanical contact of these regions without nitride coating. The third case c.) displays an optimized structure with deep corrugation rings for softening the poly-silicon layer on the one hand and increasing the leverage of the silicon-nitride layer. Additionally, the short circuit problem is reduced due to the corrugation ring structure. The parameter optimization procedure towards maximum buckling and stiffening investigations can be found in [4].

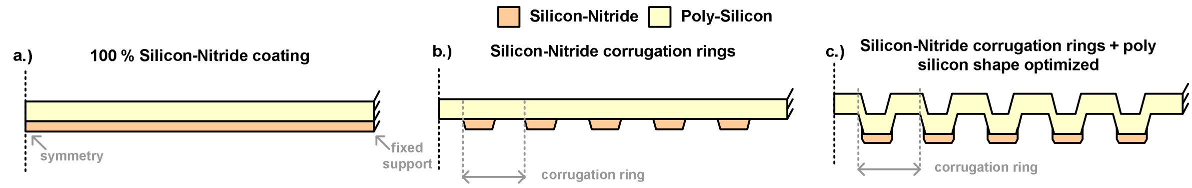

Figure 6: Buckling back plate with full silicon-nitride coating a.), with corrugation rings b.) and an approach for an optimized corrugation ring shape c.).

Results

Electrostatic – Mechanical Model Verification:
To verify the electrostatic-mechanical model, previously discussed nonlinearities like large deformation, stress stiffening, contact and the nonlinear force displacement relation are included in the model. Figure 7 displays the MEMS speaker membrane loaded linearly from zero to 30volt. The left side shows the center point stroke level where the quadratic distortion can be seen below the snap-in point. The right side displays the membrane displacement over the radius.

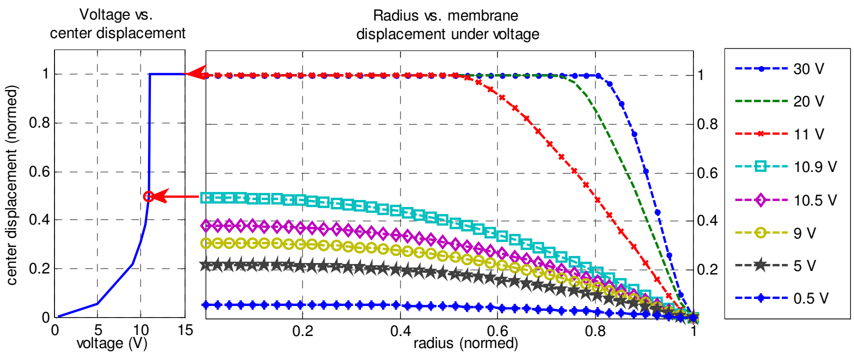

Figure 7: Voltage driven MEMS speaker in snap-in mode. Membrane center displacement (left) and axis rotational overview (right).

In Figure 8, the simulated results are compared to measured data. The measurement was performed with a single point laser vibrometer and the measurement point was not ideal aligned at the center position of the membrane. Therefore, the amplitude of the measurement is smaller compared to the simulated results of Figure 8. Comparing static FEM results with measurements is only valid for electrical stimuli up to 33 kHz. Increasing the driving frequency over 33 kHz leads to inertial effects and transient mechanical behavior, which is not modeled in the static FE model.

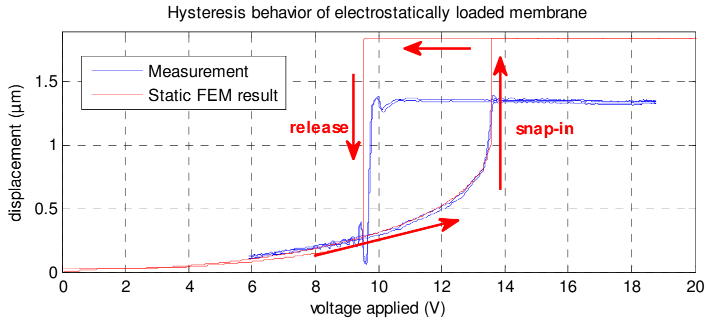

Figure 8: Comparison between FEM result und measurement of hysteretic behavior of the membrane driven in snap-in mode.

Mechanical – Acoustic Model Verification:

To verify the acoustic model with the fabricated speaker system, a small four by four array was driven at simultaneously sinusoidal stimulus. For this test, the array was loaded with a bias voltage of 10 V and an audio signal level of 1 V to 4 V. In Figure 9 the measured sound pressure levels are compared to the simulation results.

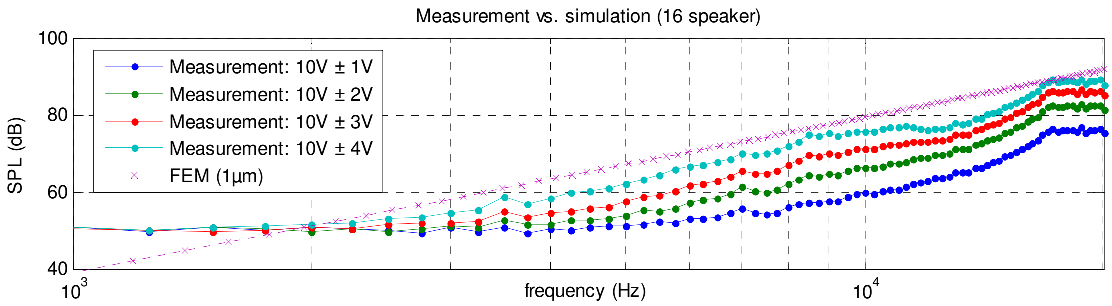

Figure 9: SPL measurements for non snap-in mode with sinusoidal characteristics, loaded with bias voltage and superimposed with acoustic audio signal compared to FEM results.

Because of small array dimensions and the relatively low stroke level (below snap-in), the resulting audible SPL starts at 1 kHz. In addition to this, the measurements were taken in standard laboratory setup without an acoustic box, hence the ambient noise is about 50 dB.

Buckling Back Plate Verification:

The reference system or initial buckling back plate system was characterized by measurements at CTR GmbH. A white light interferometer measurement setup was used, to determine the three dimensional shape of the buckling back plate. A x-z plane was extracted from measurement data to verify the simulated axis rotational results. Figure 10 displays the first buckling back plate prototype shape in comparison to the FEM result. Measurements as well as FEM results show that the basic idea of various tensile pre-stress layers is working for MEMS speakers with buckling back plate systems.

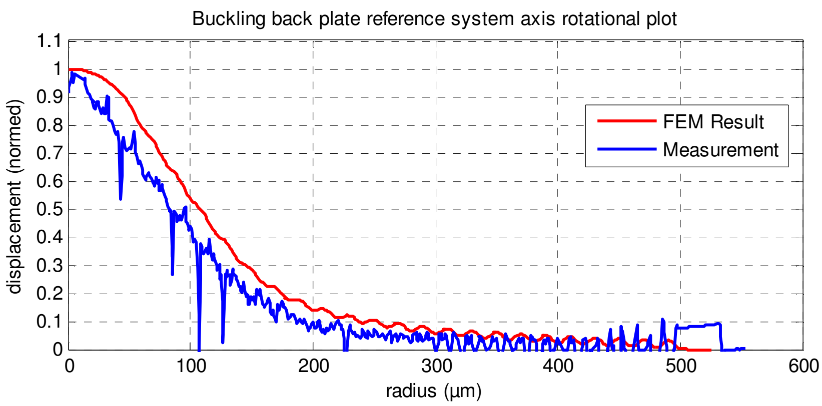

Figure 10: FE model result versus measurement of buckling back plate static deformation.

Conclusion

A multi field finite element model has been developed to investigate on electrostatically driven plate type MEMS speaker. In the electrostatic- mechanical model, reduced order elements were used to model the nonlinear electrostatic force consisting of the Coulomb's rate and the interface force on the insulation layer. Geometric nonlinearities as large deformation and stress stiffening effects were taken into account. The latter was used to optimize the flat speaker system towards sound pressure level by the help of stress induced self raising structure, called buckling back plate. In the mechanical acoustic domain, the tradeoff between mesh distortion and amount of elements was challenged with Mortar FEM by the use of non-conforming grids. Open domain characteristics were modeled with perfectly matched layers. All finite element results were verified with measurements and establish a solid base for further investigations.

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