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Comparison of numerical methods for miniature loudspeaker modeling

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ABSTRACT

This work compares the results of obtained using conventional acoustics FEM simulation models to CFD based results for miniature loudspeakers. The methods discussed here include acoustical models with different approaches to fluid properties (linear, nonlinear, lossy) using a moving geometry, and a stationary geometry CFD model. The results indicate that even if the conventional acoustical models are highly successful in predicting the pressure distribution in an enclosure, and in predicting the resonance frequencies, the full features can be revealed only through a more detailed fluid dynamic analysis.

Keywords: Electroacaoustics, Transducers, Modelling

1. INTRODUCTION

Finite element modelling is a standard tool in the design of any audio transducer. The small-scale devices, such as headphones, mobile phones, and hearing aids pose challenges not found in large form factor transducers (such as full-scale loudspeakers):

- The distance from almost any point in the interior to the nearest boundary is short enough so that point is within viscous and thermal boundary layers
- The displacement of a microspeaker in normal usage conditions is a significant fraction of the total air volume of the system, so the nonlinearity of the air is a possibility
- The geometry of the system varies due to the diaphragm displacement
- The system contains narrow internal gaps and ports close to boundaries so that the geometry of the system can lead to the formation of laminar vortices already at low signal levels

2. THE COMPUTATIONAL MODEL

2.1 Basic features and assumptions

The geometry chosen represents a typical microspeaker. The features shared by most designs include an airspace behind the diaphragm that is coupled to the external space through narrow gaps. The geometric nonlinearity in the surround is included (or rather, could not be easily excluded without making the model geometry otherwise unrealistic), but the materials as such are assumed to have linear elastic properties.

The driver electromagnetic coupling is assumed to be linear, and the electromagnetic damping is modelled through a velocity-dependent component in the force. The electromagnetic aspects of the driver are not otherwise included in the FEM model. Although electromagnetic nonlinearity is a significant factor in real-life drivers, this choice was made in this study to focus more clearly on the nonlinearity features due to the fluid flow and the chosen fluid models. All the examples were computed using Comsol Multiphysics, versions 5.3a and 5.4.

The cases studied include:

- Harmonic solution for a linear transducer without flow losses (benchmark, represents the most commonly used method)

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- Time-domain solution of a linear transducer with lossless acoustics (approximated as a very low signal level simulation)
- Time domain solution of a transducer with geometry variation due to large diaphragm displacement but with linear lossless air inside the cavities
- Geometry variation and nonlinear lossless air (no viscosity or thermal conduction), computed using the ideal gas state equation instead of the linearized equation
- Geometry variation and thermal and viscous losses in the fluid, but not on the boundaries. The propagation losses in the fluid itself are small over the distances and frequencies in this study, so this should also be considered more as a benchmark case
- Non-isothermal CFD solution with stationary geometry

The interesting case which unfortunately had to be left outside this study is using acoustic equations with boundary effects, which should yield better estimates for internal acoustic damping. This is a topic for further study.

2.2 The challenges and benefits of time-domain modelling

The usual choice for solving finite element problems is to use the frequency domain solutions. The solvers can be written efficiently, and the frequency resolution is determined only by the maximum size of the mesh elements. Time domain solutions, on the other hand, require more memory, are less efficient to compute, and due to the numerical behavior of the finite element solvers the time discretization has usually to be much finer than the $1/2f_s$ (where f_s is the sampling frequency) that the Nyquist theorem would suggest. Thus, time-domain solutions are usually avoided in acoustics. However, capturing nonlinearity in frequency domain computations removes to a large extent the numerical efficiency advantage, since frequency-domain nonlinearity methods, such as harmonic balance, are iterative and work best only with weak nonlinearities. The explicit time domain formulation, on the other hand, can be basically extended to any level of nonlinearity, although practical considerations limit the usability of deformed geometry solutions. The practical limit for nonlinearity comes mostly from the ability to sufficiently deform the mesh geometry, since if the geometry changes so much that a remeshing is needed, the discontinuity caused in the results by the change in mesh geometry renders any distortion analysis results unusable.

2.3 Test configuration

The driver-enclosure combination used in the simulations represents the typical dimensions found in miniature loudspeakers or headphone drivers. The diaphragm is 0.1 mm aluminum dome, and the surround material parameters correspond to typical silicone rubber. The thickness of the surround is adjusted to 0.2 mm to yield the desired resonance frequency of 400 Hz with the cavity size used in this simulation. The voice coil is modeled as a solid copper ring.

The electromagnetic force on the voice coil is approximated by an annular force acting on one edge of the voice coil. In case of voltage drive, the force F is obtained from expression

$$F = Bl \cdot (E_{in} - Bl \cdot u)/R_E \tag{1}$$

where Bl is the force factor, E_{in} the input voltage, u the velocity of the edge, and R_E the electrical resistance of the voice coil. In reality the force acts across the entire voice coil surface, and is place dependent due to the inhomogeneous magnetic field B, but this approximation is accurate enough for the purposes of this work, since the electromagnetic nonlinearity is left outside the scope of the analysis.

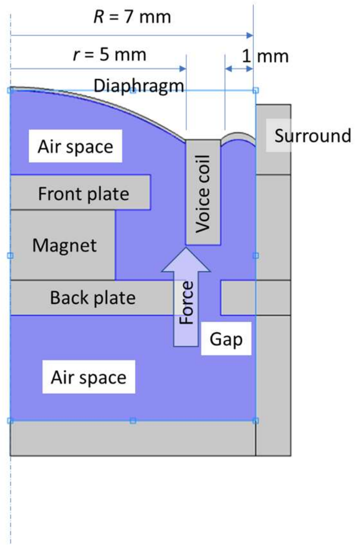

Figure 1 – The structure of driver and back cavity the model.

The simulations were run with two types of excitation: for acoustics models with deformable geometry using an annular force acting on one corner of the voice coil, and for the CFD studies with stationary geometry a flow velocity source on the diaphragm, voice coil bottom surface, and the surround, and with non-slip moving wall boundary condition for the side surfaces of the voice coil.

3. VARIABLE GEOMETRY SIMULATION

The first cases in this study describe the behavior of the loudspeaker when the geometry is time-variant and the fluid is described using conventional acoustics equations without thermal conduction or viscous effects at the boundaries. The test signal used for the simulations is a logarithmic sweep, since it will allow the analysis of harmonic distortion in the post-processing. Three different fluid models: linear fluid, adiabatic fluid with nonlinear compressibility, and ideal gas (with internal thermal conductivity)

The cases studied include no electromagnetic damping (i.e. current drive), which here mostly serves as a stability test for the method, as current drive is rarely used with microspeakers, moderate electromagnetic damping (force factor $Bl = 1$), yielding the total Q factor of approximately 1 with the material parameters used here, and high damping ($Bl = 5$). The moderate damping corresponds to practical designs, and the high damping case is for computational testing, but unfeasible in micropspeakers with the currently available magnetic materials. As and additional confirmation, the small-signal frequency responses were computed using a frequency domain formulation. The results below are for the thermally conducting and viscous formulation. Lossless fluid models perform well when electromagnetic damping is present, but they are unstable in the test case with current drive (i.e. without electromechanical damping), since the driver surround is assumed to have no mechanical losses. The computation time differences between various fluid models is small, but the computing time for any time domain model is about two orders of magnitude higher than for frequency domain models with the same number of frequency or time points.

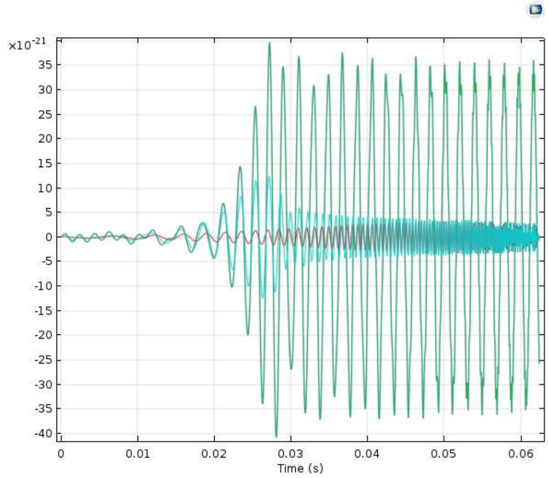

Figure 2 – Average outer surface acceleration of a logarithmic sweep signal with no electromagnetic damping (light green), moderate damping (dark green), and high damping (red).

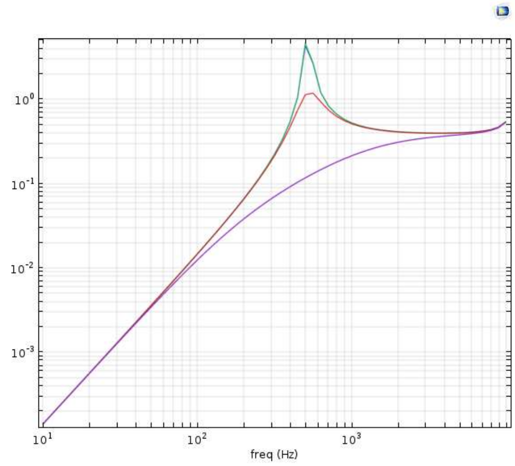

Figure 3 – Frequency responses corresponding to the different values of electromagnetic damping. The rise of the responses near 10 kHz is due to the emerging diaphragm mechanical resonance.

4. STATIONARY GEOMETRY CFD SIMULATION

4.1 CFD simulation

As opposed to the wave equation used in conventional FEM solutions, the CFD equations do not solve directly for particle velocity, but rather for energy variables of the field, which are in turn derived from the velocity and pressure variables. From practical perspective the most critical difference between the acoustic wave equation and CFD equations is that the CFD solutions allow the existence of vortices in the field, while the classical derivation of the acoustic wave equation explicitly assumes that the field is irrotational, and even viscothermal extensions to the acoustical wave equation do not assume a vortex component.

The computing time for stationary geometry time dependent CFD model is about one order of magnitude higher than that of a variable geometry acoustical model. This difference is due to both the complexity of the equations and the denser mesh needed. Practical experience indicates that the CFD model should include turbulence. The amount of turbulent energy in the field is not large, but apparently the turbulent energy transport helps to stabilize the model.

The behavior of the field in the loudspeaker assembly is illustrated by examining the sound field generated by sinusoidal vibration at 450 Hz. The data is plotted at the velocity maximum, i.e. displacement zero crossing.

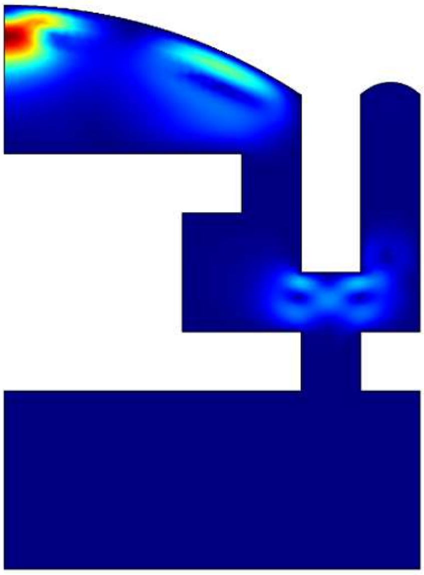

Figure 4 – Flow velocity distribution at the peak of the diaphragm velocity.

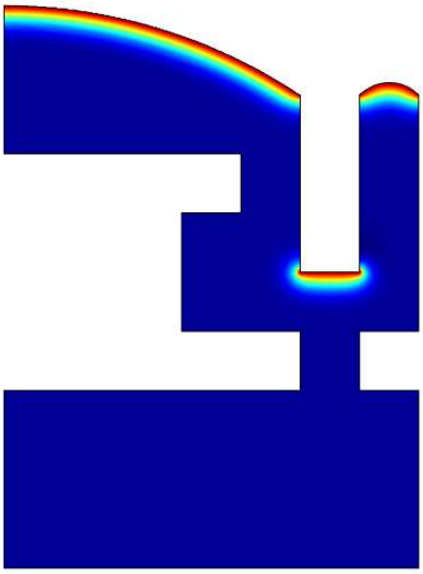

Figure 5 – Pressure distribution at the peak of the diaphragm velocity. The actual pressure differences in the cavity are quite small, a few perecent, but the figure is scaled to illustrate the slight pressure inhomogeneity near the moving boundaries.

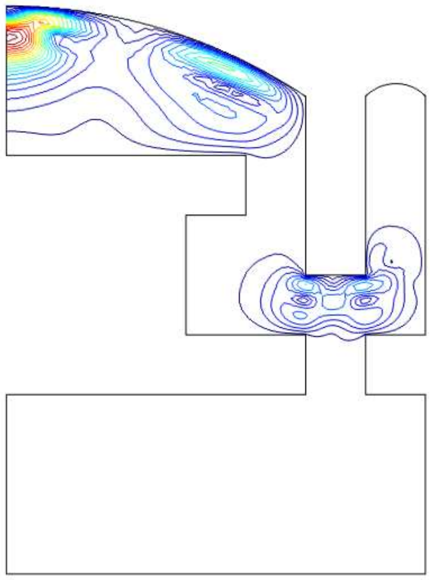

Figure 6 – Equal-velocity magnitude contours corresponding to Figure 4.

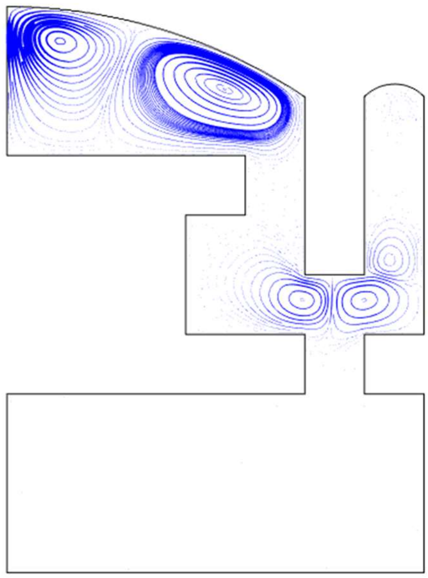

Figure 7 – Velocity field streamlines, line thickness scaled proportional to velocity to illustrate the dominant features of the flow field.

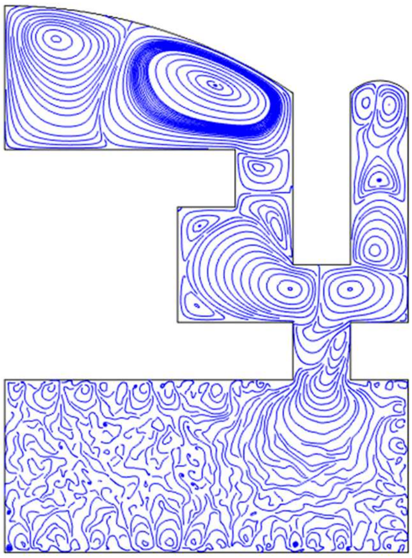

Figure 8 – Normalized streamline plot of the velocity field, all streamlines shown at the same level to illustrate the actual complexity of the flow field. It should be noted from Figure 7 that the flow velocity, and thus the energy in the vortices, is quite low in most of air space.

The important conclusion that can be drawn from the figures above is that the flow field in transducers is according to CFD results much more complex than the acoustical theory results would lead to expect. The vortex formation is important especially in the air space between the diaphragm and the front plate and around the voice coil.

The acoustical models are excellent when estimating the resonance frequencies and pressure distribution, but as the vortex formation cannot appear in the acoustical theory predictions, the local flow velocities in the air space get underestimated, which in turn will result in underestimating the viscous losses.

5. CONCLUSIONS

The preliminary results in this work confirm that the behavior of miniature loudspeakers and related transducers is best understood using CFD models. However, as the models are computationally significantly heavier than wave equation based analysis, and the convergence of solutions is by no means guaranteed especially in more complex geometries, the use of lossy acoustics models is still better suited for practical design work, especially when using time-dependent geometries, which are essential for proper analysis of transducer nonlinearities. Further work is needed on how to combine the approaches needed for the various parts of the loudspeaker assembly into a computationally efficient system model.