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Multiphysical Simulation Methods for Loudspeakers - Advanced CAE-based Simulations of Vibration Systems

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ABSTRACT

This is the second in a series of papers on the details of loudspeaker design using multiphysical computer aided engineering simulation methods. In this paper, the simulation methodology for accurately modeling the structural dynamics of loudspeaker's vibration systems will be presented. Primarily, the calculation of stiffness, or its inverse, the compliance, in the virtual world will be demonstrated. Furthermore, the predictive simulation of complex vibration patterns, e.g. rocking or break-up, will be shown. Results will be presented, correlating the simulated model results to the measured physical parameters. From that, the important aspects of the modeling which determine its accuracy will be discussed.

1. INTRODUCTION

While most loudspeakers can be treated as axisymmetric systems, and thus simplified 2D models can be applied, its multidimensional vibration pattern can have a significant impact on the acoustical performance. Depending on the final application, e.g. non-axisymmetric enclosures, non-axisymmetrical behavior can be important, and advanced 3D models need to be used. Thus, typically finite element models for detailed vibration system design and optimization are highly valuable and efficient design tools.

For system or subsystem level simulations (without the goal of designing a vibration system) 1D lumped models are highly efficient. A lumped parameter model can be found in [1]. The basic differences between 1D, 2D and 3D models will be described in the current work.

2. THEORY

2.1. Mathematical and Physical Background of Structural Dynamics

A loudspeaker is driven by a time-harmonic voltage, $V = V_0\exp(i\omega t)$, applied to the voice coil. The following mathematical background section excludes a description of the electromagnetic analysis of the current in the voice coil and the driving force that this current generates, and focuses purely on the mechanical domain. We refer to [2] for details on the relation between the driving voltage and the force exciting the vibration system. Additionally, we will at this time focus on linear effects, and leave the nonlinear simulation approach for future publications.

The governing equation for the mechanical vibrations in the frequency domain, discretized by means of FEA (Finite Element Analysis), can be written as follows:

$$(K^m + i\omega D^m - \omega^2 M^m)u^m = f^m \qquad (1)$$

At a first glance there seems to be only a little difference in the governing equations by matrix methods and by lumped parameter models. However, the big difference is the dimension of the system. In the finite element governing equation stiffness, mass and damping are being described via matrices. K^m is the stiffness matrix, D^m is the damping matrix and M^m is the mass matrix. Furthermore, u^m is the vector of displacements and f^m is the vector of mechanical forces exciting the system. ω is the angular frequency. Typically the dimension is of several of thousands degrees of freedom. In fact the governing equation is a system of equations describing the mechanical vibrations with respect to a detailed definition of the geometry (CAD model) discretized via finite elements. Thus it is possible to use these models for the whole audible frequency range which is typically from 20 Hz up to 20 kHz where a lot of non-pistonic and non-axisymmetric motion patterns occur.

For $\omega = 0$ we get a static solution, and thus the lumped stiffness of the system, typically referred to as K_{ms}, can be calculated, or its inverse, the compliance C_{ms}.

3. SIMULATION MODEL

3.1. 2D Model Setup

Setting up a model is straight forward and relatively simple (at least in 2D). Starting point is a 2D (cleaned-up) cross section as given in the following figure (we will later discuss 3D models):

Figure 1 2D cross-section of a typical vibration system

For details on general CAE modeling, such as geometry and material definition, input parameters, applying physical laws, and parametric studies, we refer to [3].

3.2. 2D Solutions

By applying a force of 1 [N] at the voice coil, and assuming $\omega = 0$ (i.e. a static solution) we get the following displacement pattern of the vibration system:

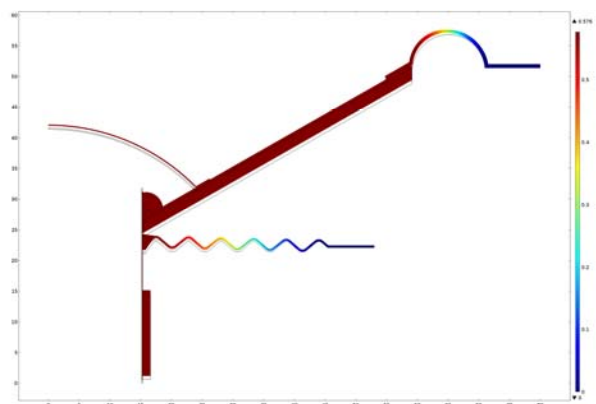


Figure 2 Displacement of vibration system at 1 [N]

The displacement at the voice coil is 0.576 [mm]. Thus the lumped stiffness K_{ms} = 1 [N] / 0.576 [mm] = 1.74 [N/mm], which is in good agreement with a measured value of 1.67 [N/mm].

By performing a dynamic eigenvalue analysis, we get the first natural frequency (or eigenfrequency) in vacuum (i.e. without the influence of the surrounding air) at 37.0 [Hz]. This again is in excellent agreement with a measured value of 36.9 [Hz]. The following figure shows the displacement pattern of the 1st eigenfrequency, the so-called piston mode of a loudspeaker.

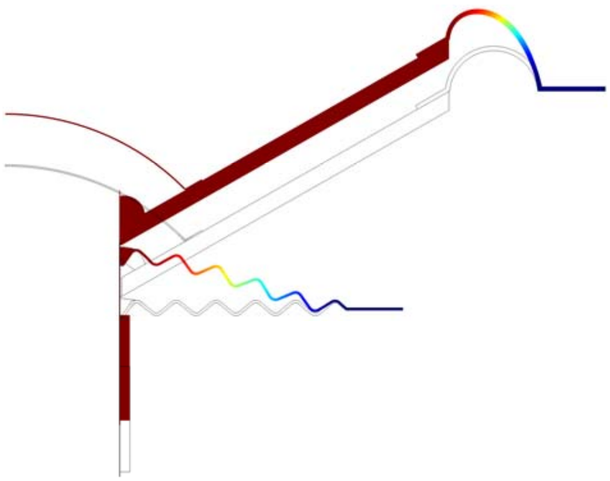


Figure 3 1st natural frequency

In a further step, a forced response analysis is performed, where a constant force of 1 [N] in the frequency domain is being applied.

By evaluating the displacement at a point of the voice coil we actually get a description of the lumped dynamic stiffness as a function of frequency (see the following figure). Below 1 [kHz] we see a pretty smooth variation, showing very prominent the 1st natural frequency. However, above 1 [kHz] we see a couple of significant variations. The first variation is often referred to as the break-up frequency. The name actually comes from the fact that a lumped parameter solution breaks-up at that frequency, i.e. it simply delivers wrong results.

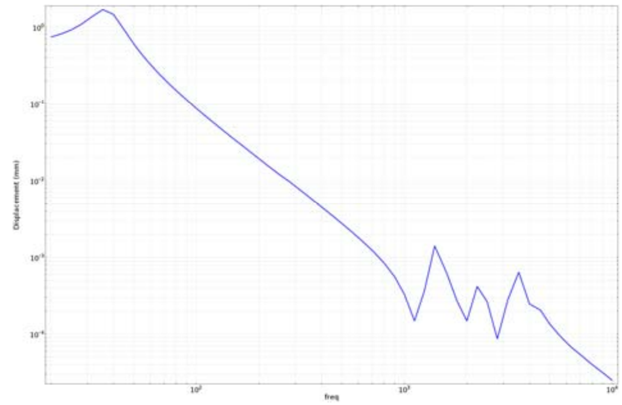
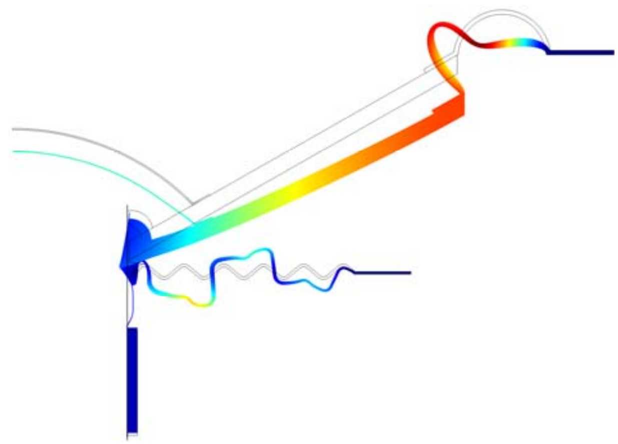


Figure 4 Displacement of voice coil over frequency

The following figure shows the operational deflection shape of the vibration system at 1,400 [Hz]:



The bending in the cone is dominating a disturbance in the frequerncy response.

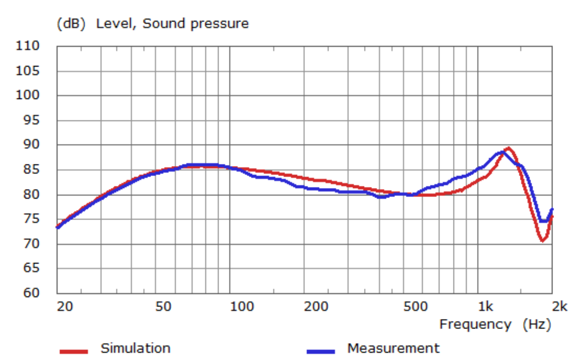


Figure 5 SPL on-axis over frequency

The figure shows the on-axis SPL in [dB] in a distance of 1 [m] for simulation and measurement as well. Details of coupling to the surrounding air are not given here, but will be presented in a future paper.

3.3. 3D Solutions

In a similar way a 3D model can be used to derive additional, non-axi-symmetric, results. However, simply generating a full 3D geometry based on a sweep of the 2D cross-section model does not lead to satisfying models. Moreover, a surface based (instead of 2D solid based) model using shell finite elements leads to highly efficient 3D simulation models (for details see [3]).

Here the starting point is a (cleaned-up) CAD based surface model of the vibration system as given in the following figure:

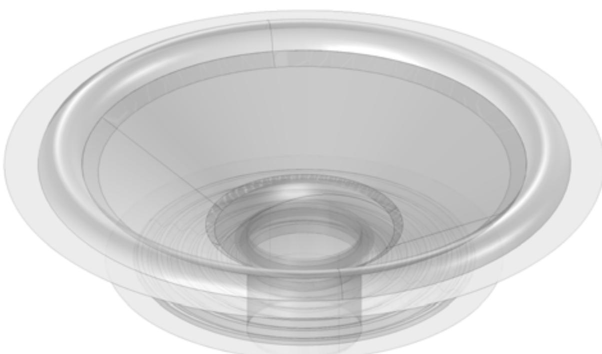


Figure 6 3D surface based CAD model of vibration system

While all results showed in the previous section about 2D modeling are very similar in 3D, we especially get additional results, in terms of non-axisymmetric deflection shapes, that are typically caused by non-axisymmetric enclosures (which is usually the case).

Most important is the so-called “rocking” of loudspeakers, showed in the following figure. This effect leads to strong variations in radiated sound pressure, and can lead to heavy distortions in the extreme case, when the voice coil hits the magnet.

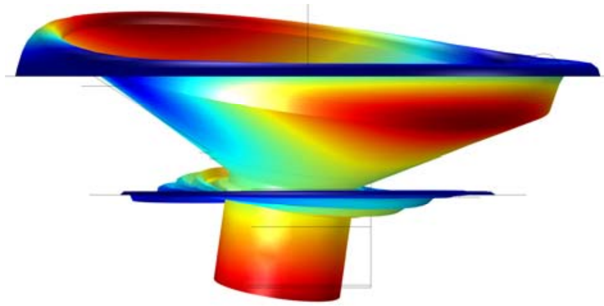


Figure 6 “Rocking” of a vibration system

4. FINAL COMMENTS

The audio industry, as well as most industries today, is challenged by the need to constantly increase engineering efficiency. Computer-Aided Engineering (CAE) based on simulation and analysis of the functional performance of products has already played a key role for more than two decades. CAE methodologies are today typically used at every stage of the development cycle, from first concept studies up to detailed engineering for final product development to be released to the market place (including modeling of the manufacturing processes as well). During the last years a strong trend for moving CAE upfront in the design process (to be applied already in the concept phase) can be monitored. Thus the term frontloaded Virtual Product Development (VPD) is often used. The advantages for moving CAE upfront are:

- more freedom in the design decisions
- design changes can be made at lower costs

These advantages additionally fulfill the above mentioned basic requirements to increase engineering efficiency. Similar thoughts have been applied to the automotive industry [4].

This frontloaded approach was first (successfully) introduced for the development of automotive and aerospace key components at the Automotive Original Equipment Manufacturers (OEMs). A good example for a first application of VPD is the development of car body structures with respect to crashworthiness, fatigue or NVH behavior. Today nearly every industry is finding the need to follow the path of a frontloaded VPD cycle as well.

In this paper, we can precisely see that advanced CAE methods can accurately predict the structural dynamics of loudspeaker's vibration systems. Thus we can optimize designs in a very early design stage where no physical prototypes exist. Ultimately these results can be used to see the behavior of the loudspeaker when coupled to an enclosure (to be published in a future4 paper), whether that enclosure it is simple or complex geometry. Complex geometries are more and more commonplace as the need to innovate new designs and integrate loudspeakers in unique ways grows rapidly. Here again, CAE methods are key technologies to optimize product development in terms of performance and cost, resulting in optimization of engineering efficiency in general.

5. REFERENCES

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