CALCULATING SOUND RADIATION FROM LOUDSPEAKER ENCLOSURES USING THE FINITE ELEMENT ANALYSIS

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Walls of a loudspeaker enclosure exhibit vibrations and resonances, creating unwanted acoustic output. These vibrations are excited by the mechanical forces from the drivers and by the acoustic pressure inside the enclosure. Enclosure vibrations cannot be satisfactorily simulated using a mechanical model that only includes the mechanical conduction of vibration in the enclosure. Also acoustical phenomena have to be considered to get accurate results. In this paper we show how Finite Element Analysis using a coupled model of mechanics and acoustics can predict enclosure vibrations and their causes with good resolution.

INTRODUCTION

Ideally loudspeaker enclosure walls do not contribute to sound radiation. Enclosure wall movements can cause audio colorations and otherwise change the performance. As materials have finite stiffness and mass, loudspeaker enclosure walls tend to vibrate and can produce some acoustical output. Loudspeaker manufacturers have been trying to minimize enclosure vibrations with a number of means. [1]

There are different mechanisms that generate acoustic output from secondary sources of a loudspeaker, mainly the enclosure walls and reflex port. The loudspeaker enclosure itself has finite mass and stiffness and therefore exhibiting resonances and deformations under load. The reflex port introduces resonances also above the desired Helmholtz frequency. Standing waves inside the enclosure affect the excitation of the enclosure walls and thus also their vibrations. Internal acoustical resonances can also radiate audio through the reflex port opening. In some conditions the coincidence effect can decrease the sound attenuation through the enclosure walls.

Calculating loudspeaker enclosure vibrations using finite element analysis is not a new idea. First references date back to over 30 years ago [2]. However, until recently, the lack of computing power has limited the use of finite element analysis. Traditionally enclosure vibrations are studied empirically [3-5]. Numerical methods for estimating loudspeaker enclosure vibrations have been studied for example by Karjalainen [6] et al. and Luan [7,8] showing the potential in solving loudspeaker enclosure vibrations numerically. When calculating sound fields created by vibrating surfaces boundary element method has often been used with finite element method because of the lower computational cost of the boundary element method [9]. Even then, simplifications and idealizations of the structures and boundary conditions have often been necessary to conduct analyses. The models presented have usually been significantly simpler than a full actual loudspeaker. Factors limiting the accuracy of finite element simulations are reported to be the accuracy of CAD geometry and material parameters as well as the assumptions made for the analysis. [10-12].

The aim of this paper is to study the benefits of using a coupled model over using a mechanical vibrational model or acoustical modal analysis model alone. The complexity of the modeled loudspeaker closely corresponds to that of a real loudspeaker.

1 THE ANALYSIS MODELS

In this work a case study of the resonant behavior in a two way loudspeaker enclosure is studied with acoustical and mechanical eigenfrequency analyses and mechanical analysis with loading, as well as a coupled model combining both acoustical and mechanical models. The analysis models were built with cleaned-up CAD models of the enclosure and the woofer frame. The analyses are conducted on a model of a real prototype loudspeaker and the results are compared with acoustical measurements.

For the mechanical vibrational model the woofer magnet and the tweeter were replaced with homogenous masses as their inner features were considered to have negligible effect to the behavior of the whole system.

Connections between parts were considered rigid and the enclosure was rigidly supported from the back wall.

The acoustical model consists of the air volume inside the enclosure and a finite volume or air surrounding the enclosure.

The coupled and mechanical analysis models were halved at the symmetry plane to reduce computational load (Figure 1). In principle the coupled model combines mechanical and acoustical analysis models.

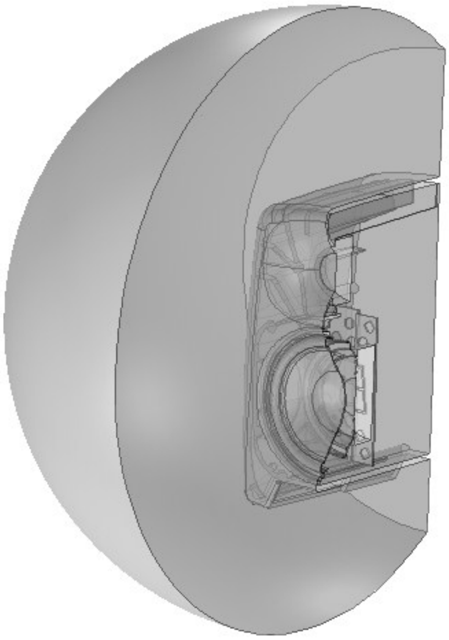

Figure 1. The analysis model for coupled analysis.

2 MECHANICAL ANALYSIS

Both eigenfrequency and frequency domain analysis were made. The amount of radiated sound cannot be directly evaluated from the displacements so a rough estimation was made by integrating velocities on the enclosure external surfaces to estimate volume velocity.

2.1 Eigenfrequency analysis

The eigenfrequency analysis resulted 30 nominal frequencies between 128 and 870 Hz. A closer look reveals that many of them appear not to be easily excited, given the loading in the loudspeaker, and/or because they do not seem to have the potential to generate much acoustic output. Normal velocities integrated over top, side and front surfaces of the enclosure at eigenfrequencies can be seen in Figure 2. The eigenmodes are RMS scaled. The analysis does not in any way predict how the modes get excited in reality. This processing does demonstrate, however, the potential of a mode to radiate sound.

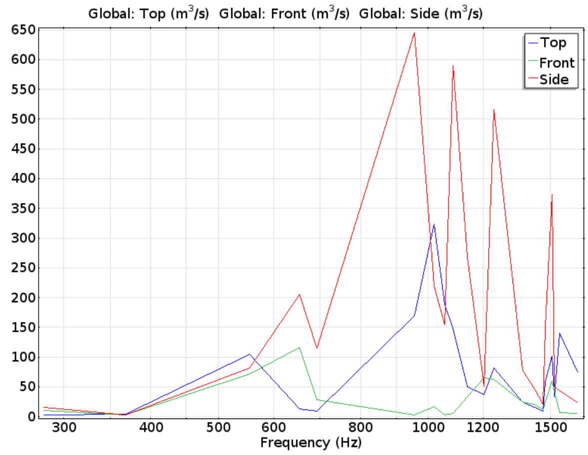

Figure 2. Normal velocities of eigenmodes integrated over top, side and front surfaces. RMS scaling of eigenvectors.

2.2 Loaded mechanical analysis

The loading in the mechanical analysis is the recoil force from the voice coil and a uniform pressure inside the enclosure. Figure 3 shows on a log scale the normal velocities for the same case as in Figure 2 but considering loading. The data is calculated at discrete frequencies with 50 Hz spacing. These figures indicate that the eigenmodes can have an impact on the vibration patterns of the enclosure when acoustical load is considered.

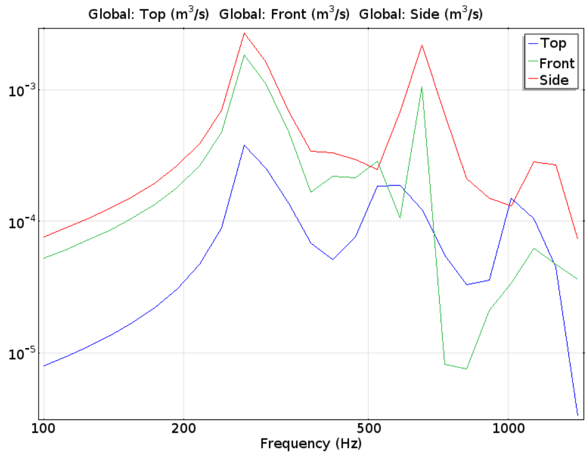

Figure 3. Normal velocities with loading considered, integrated over top, side and front surfaces. 50 Hz frequency steps.

3 ACOUSTICAL ANALYSIS

The acoustical model consisted of the air volume inside the enclosure and a finite air volume surrounding the enclosure. Enclosure walls were considered rigid. An eigenmode analysis (Table 1) found eigenfrequencies in the inside air volume and in the reflex ports.

Table 1. Eigenfrequencies *fe* of the air inside the enclosure.

fe [Hz]	possible cause
61	Helmholtz resonance
548	Standing wave in the enclosure (top – bottom)
750	Standing wave in the enclosure (left – right)
762	Standing wave in the enclosure (duct)
815	Open pipe resonance in reflex ports
858	Standing wave in the enclosure (diagonal)
940	Standing wave in the enclosure (diagonal)

The modes found in the analysis are typical but the basic equations for acoustical resonances do not accurately predict the resonant frequencies with a geometry this complex. The lowest mode frequency found is the Helmholtz resonance. The standing waves are determined by the enclosure dimensions, and in the example case, the folded duct formed inside the enclosure structure.

Many of the eigenfrequencies found were in the region showing problems also in the acoustical measurements. Also this analysis fails to directly predict which modes may cause audible problems, i.e. audible resonances, in reality.

4 COUPLED ANALYSIS

The coupled model was loaded acoustically with the cone acceleration and its recoil force on the driver. The cone was set to radiate only inside the cabinet to segregate direct sound from results.

Average sound pressure level at equal distance from the loudspeaker (Figure 4) represents the sound radiated from secondary sources through enclosure walls and reflex ports.

The peak frequencies and deformations of the structure were compared to mechanical and acoustical eigenfrequencies to resolve the contribution of acoustical and mechanical resonances on the whole.

At lower frequencies the acoustic output originates mostly through the reflex ports. There are peaks visible at 550, 750, 950 and 1050 Hz. In this case example, both mechanical and acoustical resonances coincide near 550 Hz leading to a peak in the simulated response. The peak at 750 Hz was caused by an acoustical resonance, and radiated through the enclosure walls. Also the reflex port pipe resonance at 800 Hz is visible, but at a lower amplitude than the peak at 750 Hz. The peaks at 950 and 1050 Hz can be linked with mechanical resonances. In this case example, there was no damping material inside the enclosure, and this tends to emphasize acoustical resonances.

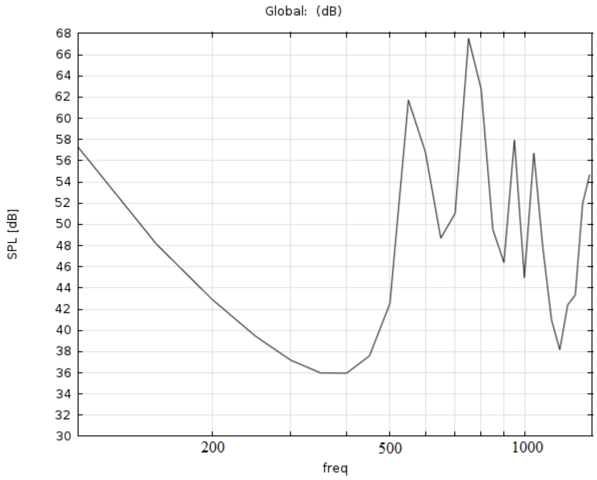

Figure 4. Simulated average sound pressure level on the external surface of the air domain.

5 COMPARISON TO MEASUREMENTS

An actual prototype loudspeaker matching the construction used in simulations was built and measured. The measured cumulative spectral decay plot of the prototype loudspeaker (Figure 5) demonstrates that although some problem frequencies match the simulations, a mere mechanical model does not predict the real system response. Resonances at 550 and 750 Hz match well with what was simulated in the coupled model.

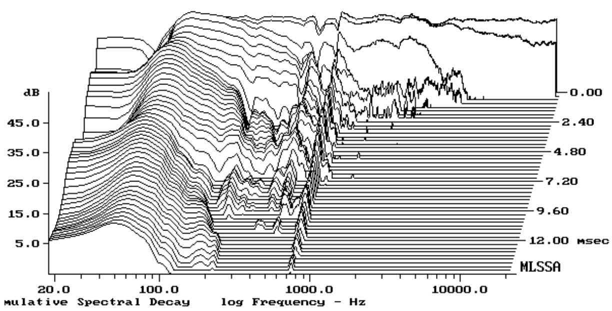

Figure 5. Cumulative spectral decay plot of a prototype speaker.

6 CONCLUSIONS

The results show that finite element analysis can predict enclosure vibrations with good accuracy also in practical cases. Both mechanical and acoustical simulations of a loudspeaker enclosure yield substantial information of the enclosure's resonance characteristics. Both can pinpoint some problem frequencies with good accuracy. However, running separate mechanical and acoustical analyses cannot predict whether it is mechanical or acoustical phenomena that dominate the sound radiated from the enclosure at a given frequency.

A coupled model considers both mechanical and acoustical excitations with correct relative magnitudes, and the sound radiation can be calculated directly from such model. The coupled analysis shows good match with acoustical measurements on a real system and gives a good overall view to enclosure vibrations.

Overall, the results can be considered very informative. It becomes clear that when looking at the total performance of a loudspeaker, the mechanical and acoustical resonances cannot be handled separately.

In our case examples, simulations indicate that there are three main mechanisms contributing to the secondary sound radiation. Depending on the frequency, the source mechanism is a mechanical resonance, an acoustical resonance inside the enclosure and mediated out by radiation through the walls, an open pipe resonance in the reflex port, or a combination of these.

Meaningful simulations of a loudspeaker enclosure require including all the parts that contribute significantly to the mass and stiffness of the structure. In practice this usually includes at least the loudspeaker drivers in addition to the enclosure. The parts in the loudspeaker construction form a system and it should be analyzed as that. Performing separate analyses on different parts separately makes evaluating the whole system very difficult. Our results are sufficiently accurate to enable identification of the potential problem frequencies before building a real prototype. This can lower the number of design iterations and prototypes needed. Postprocessing the simulation data can enable understanding the causes for problems easier than measurements of a physical prototype. In our case example, the coupled analysis made it possible to get a clear picture of the impact of different design options and further improvements of the enclosure's characteristics and therefore the quality of acoustical response of the loudspeaker system. In our case example, it became evident that the overall stiffness of the enclosure dominates the performance and that there was little to be gained from small tweaks such as, for example, optimizing the locations of stiffening ribs.

With an efficient way to simulate the vibration and unwanted sound output from the enclosure walls, the levels of these source mechanisms can be reduced below the threshold of audibility without making excessively heavy-built structures. Another benefit from reliable simulations is the decreased need for physical prototypes. This can save time and money especially when the enclosure is complicated to manufacture.

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