ACCURACY ISSUES IN LOUDSPEAKER SIMULATION

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1 INTRODUCTION

Finite element software for vibroacoustic simulation of loudspeakers has been around for many years. It is being used more widely used now, because of improved solver functionality, faster hardware and improved CAD links together with other preprocessing improvements. Using simulation is advantageous because it allows the design cycle to be shortened, and cost-effective assessment of more radical design changes to be made[1]. Furthermore it can give the user more insight into the physical phenomena occurring and this can be helpful in coming up with design modifications. The analyst has choices to make from different computational techniques, what approximations to make and how much detail to include.

Lumped parameter methods[2,3] are limited to low frequency, where the diaphragm is behaving as an ideal piston. Finite element (FE) approaches can be used to model both structural components and regions of acoustic medium. They can be used to predict diaphragm breakup, interior cavity modes, load/displacement characteristics and other effects. Often FE are combined with boundary element (BE) techniques[4] for modeling the exterior acoustic domain, permitting the solution of transducer radiation, enclosure diffraction, horn design and other types of analysis. Theoretically this approach could consistently produce very accurate results. However there are potential sources of inaccuracy, including insufficient mesh refinement, ill-conditioning, inaccurately known material properties, failure to include important structural details, use of approximate fluid/structure coupling, use of approximate radiation conditions, and other phenomena such as viscothermal losses or enclosure venting which may not be included sufficiently accurately.

As a finite element mesh is refined, using smaller elements, more accurate results are generally obtained. To achieve a specified level of accuracy, a finer mesh is required at higher frequencies, where there is a more rapid spatial variation of the pressure and displacement fields. It is in principle straightforward to check for errors of this type, by rerunning with a refined mesh, or perhaps using the software to compute the degree of continuity in the gradient of the primary field variable. If ill-conditioning is present, it can often be detected by the solver. However the analyst can do other checks, such as solving the problem in a different manner to determine whether the computed results change. For someone starting to use FE analysis to model audio transducers, insufficient material data can be a serious problem. However by testing simple specimens or correlating simulation and experiment, many companies are accumulating a knowledge of material properties of the components they commonly use, over a period of time. The other sources of error listed above, are investigated in this paper, by using an accurate transducer model as a basis, and varying the conditions, properties or solution.

2 REFERENCE MODEL DESCRIPTION

A realistic, but non-proprietary, design was used as the reference model. In the current work, only radiation from a baffled transducer, of axisymmetric design is considered. The driver used was typical of a 165mm automotive unit. The finite element model was created from a CAD file, using a preprocessor. The axisymmetric mesh of the generator plane is shown in figure 1. Quadratic solid elements of revolution were used for both structural and acoustic finite elements. The air immediately in front of the diaphragm and extending out slightly into the half space was modeled with acoustic FE. Acoustic BE line elements were used to model the remainder of the half space. The air behind the dustcap and inside the former was also modeled with acoustic FE. This region is vented at the rear. There is a thin cylindrical region, inside the former, connecting to a smaller cavity which links via another thin region outside the voicecoil to the air behind the cone. The cavity outside the voicecoil was ternminated by a zero pressure condition, as was the end of the vent of the cavity behind the dustcap. The spider and surround were clamped on their outer sections. The effects of viscosity in the thin gaps was modeled as described in section 3.5. The structural components included in the simulation were the surround, cone, former, dustcap, spider and glue. Figure 2 shows a detailed view of part of the structural model.

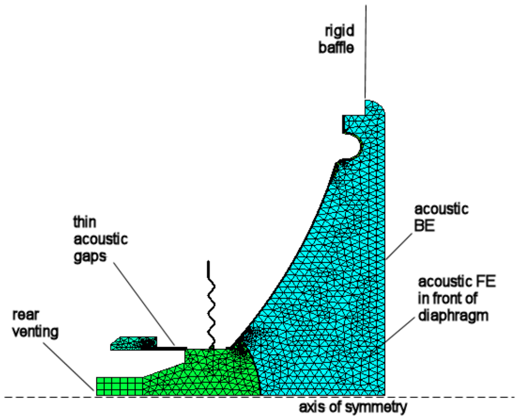

Figure 1: structural and acoustic meshes

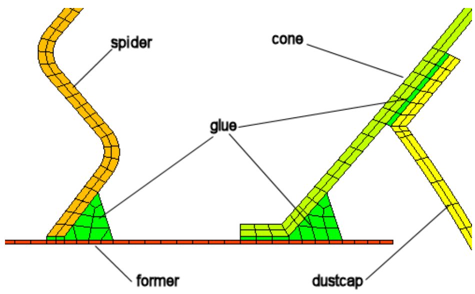

Figure 2: detailed view of part of structural mesh

A fully coupled analysis with a generalized coupling scheme, permitting dissimilar meshes along the fluid/structure interface, was used. There were 4100 structyural degrees of freedom, 3850 acoustic FE degrees of freedom and 95 acoustic BE degrees of freedom in the model. The maximum instantaneous front size was 395.

The system was analysed over the frequency range 50 Hz to 10 kHz. Initially excitation was by a unit axial force applied to the voicecoil. The results were then postprocessed to transform to a constant voltage drive excitation, using a simple lumped parameter model for the motor, with an assumed blocked impedance

$$Z_{blocked} = R + i\omega L \tag{1}$$

for fixed resistance R and inductance L. The force factor Bl was assumed to be independent of the voicecoil position.

To provide some background information to assist interpretation of the fully coupled results below, an in vacuo natural frequency analysis was performed. The main piston mode is at 77.4 Hz. There are cone breakup modes at 695 Hz, 2044 Hz, 2806 Hz, 3452 Hz. Spider-related modes occur at 566 Hz, 1417 Hz, 2540 Hz. The surround has modes at 910 Hz, 1970 Hz.

3 FACTORS AFFECTING ACCURACY

The reference model was modified and reanalysed as described in the sections below to investigate potential sources of inaccuracy. In all cases the SPL at 1 metre is compared between the modified and reference solutions.

3.1 Radiated Sound

For steady state vibration the pressure distribution in the acoustic domain must satisfy the Helmholtz equation.

$$\nabla^2 p + k^2 p = 0 \qquad (2)$$

At the boundaries, where the acoustic domain is in contact with a vibrating surface the pressure gradient is related to the normal velocity.

$$\frac{\partial p}{\partial n} = -i\omega\rho V \qquad (3)$$

In an exterior unbounded domain, the pressure field should satisfy the Sommerfeld radiation condition for outgoing waves

$$\lim_{r\to\infty} r\left(\frac{\partial p}{\partial r} + \frac{i\omega p}{c}\right) = 0 \qquad (4)$$

The acoustic finite element method is “exact” for solving finite domain problems, in the sense that, if the mesh is recursively refined, then the result computed, with exact arithmetic, will converge on the solution, satisfying (2) and (3). The boundary element method can be used for either interior or exterior problems and is similarly “exact”. For exterior problems it also satisfies (4).

Some analysts both in the audio industry and in other fields, use a Rayleigh integral to estimate the radiated sound field.

$$p(\underline{x}) = \int_D 2i\rho\omega V g(\underline{x}, \underline{y})\, dA(\underline{y}) \qquad (5)$$

Where the integration takes place over the diaphragm D, V is the normal velocity and g is the free space Green’s function. The Rayleigh integral is exact for the case of a flat diaphragm embedded in an infinite rigid baffle, radiating into a half space. If the diaphragm is not flat, then conditions (2) and (4) are satisfied but not (3). It would be expected that as the radiating surface becomes less flat, the Rayleigh integral becomes more inaccurate. Some studies comparing Rayleigh integral and BE results for problems radiating into a full 3D space from the automotive industry have been published[5]. The appendix of this paper examines the accuracy of predicting radiation from a hemispherical dome using the Rayleigh integral.

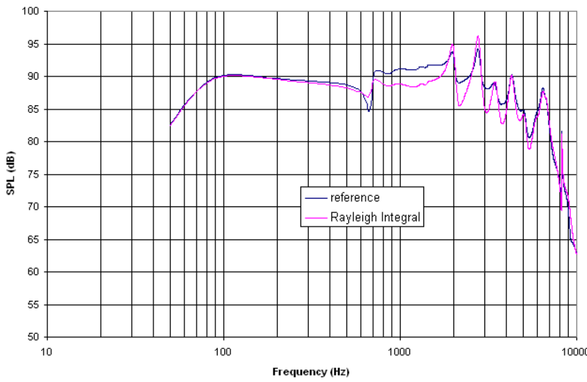

Figure 3: SPL on axis with Rayleigh Integral

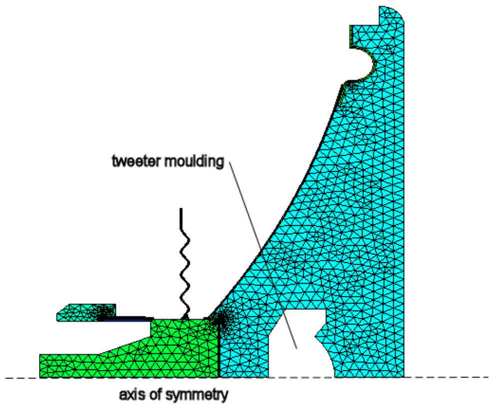

Figure 4: design with tweeter moulding

Figure 3 compares Rayleigh integral pressures computed using the correct surface normal velocities with the FE/BE results. This represents a "best possible" case in some sense for the Rayleigh integral, since if the method were being used to compute the radiated pressure field without a proper vibroacoustic solution, the fluid loading on the structure would be only approximately known, and hence the structural vibration would be less accurate to start with. Based on these results it seems that the Rayleigh integral method becomes less reliable as frequency increases.

The Rayleigh integral takes no account of any obstruction or diffraction type phenomena. Hence if a cone-based transducer has a tweeter mounted in front, as shown in the modified model in figure 4, the Rayleigh integral would be of no use in designing the position and shape of the tweeter. A comparison between radiation from the reference design and the case with tweeter is shown in figure 5.

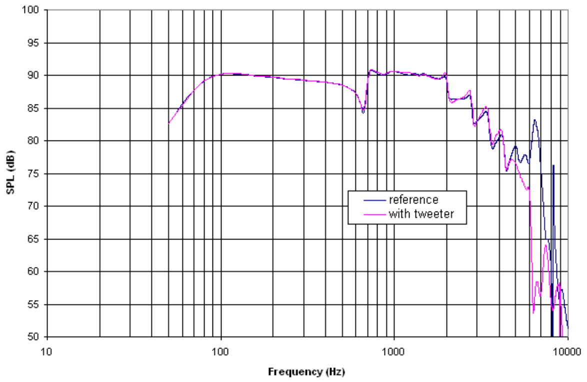

Figure 5: SPL 40 degrees off axis with/without tweeter

3.2 Acoustic/Structure Coupling

On the acoustic/structure interface there should be continuity of both normal velocity and pressure/(-ve normal stress). As it vibrates the diaphragm radiates waves into the acoustic medium and the pressure on the fluid/structure interface applies loading to the diaphragm, affecting the way it vibrates. Thus to include all the phenomena, it is necessary to solve simultaneously for structural displacements and acoustic pressures. in a fully coupled problem. A simplified procedure, with only one-way coupling, is to solve for the structural vibration in vacuo, i.e. without fluid loading, and then solve a secondary pure acoustic problem taking the normal velocity as a boundary condition on the fluid domain.

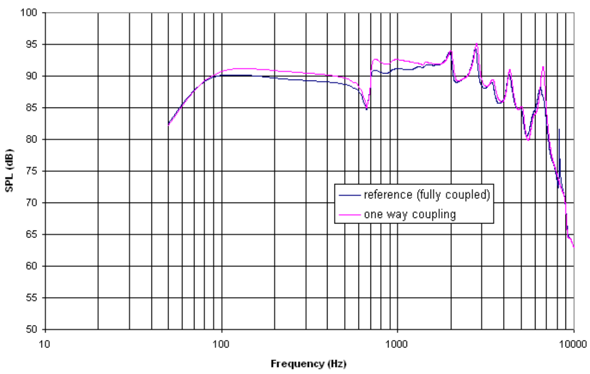

Figure 6: SPL on axis fully/partially coupled

Figure 6 shows the effect of the fully coupled assumption. At low frequencies, in the stiffness controlled region there is little difference. In the mass-controlled region there is a difference caused by the added mass effect of the air. At higher frequencies the radiation damping of the external air appears to reduce the amplitude of some of the resonances slightly. And there is a slight frequency shift caused by the additional air loading in the fully coupled model.

3.3 Joint Accuracy

Some analysts model junctions between components in detail, including layers of glue, whereas others use shell elements with pin-jointed connections. The effect of modelling the stiffness of the joint correctly is investigated here by increasing the Young's modulus of the epoxy from 0.5 GPa to 5.0GPa.

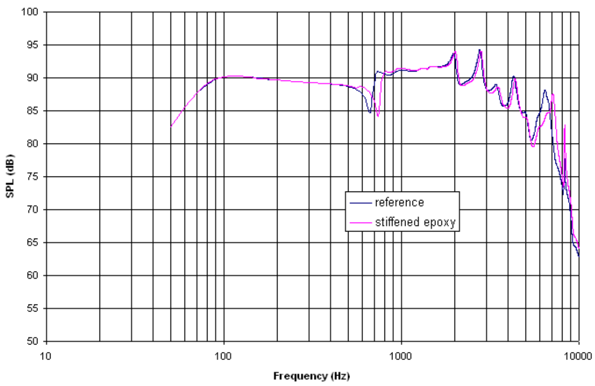

Figure 7: SPL on axis for epoxy variations

The stiffness and mass controlled regions are relatively unaffected. At higher frequencies some of the resonances increase in frequency. It would be expected that on a 65 or 110 mm unit, not modeling glue beads accurately would have a more drastic effect, because typically there is more overlap between cone and dustcap resonance and the system is more sensitive to minor changes.

It is much easier to generate a finite element mesh for a model which is fully based on shell elements. However the complexity of the joints, as shown in figure 2, is then difficult to represent. One approach would be to use a mixture of solid and shell elements, however the analyst then needs to ensure that the different element types are coupled correctly, and has lost most of the simplicity of a 2D line-based model. Alternatively it would be possible to use a shell model with artificially stiffened structural properties in the neighborhood of the junctions. Such properties would be difficult to determine without having experimental results to compare against.

3.4 Venting of Cavity

The air behind the dustcap is vented in the reference model. Figure 8 shows the effect if the vent is blocked. The main difference is at low frequency, where there is an acoustic spring effect if the vent is closed.

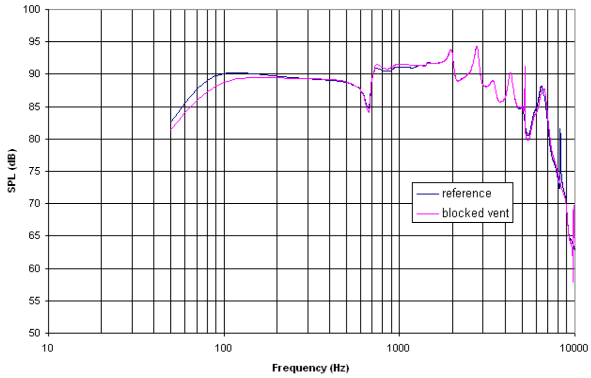

Figure 8: SPL on axis, back vent open/closed

3.5 Viscothermal Effects

Viscous effects may potentially be significant in narrow gaps. In the reference problem, the air gaps inside the former and outside the voicecoil are a fraction of a millimetre. To solve for this phenomena rigorously would probably require some CFD analysis. In the current study a simpler approach was used. In the simple geometry of an air gap of constant width between two parallel, rigid, isothermal planes , the linearized Navier Stokes, continuity, energy and ideal gas equations can be solved to first order, predicting a complex, frequency dependent, speed of sound for acoustic propagation in this region[6]. In the reference problem the gaps were 0.2mm and 0.3mm. The air in these regions was assigned properties based on the infinite case. In this simplied approach no “end effects” have been included. If the air flow through the gaps becomes turbulent, then the theory would not be valid. Furthermore viscosity is a temperature dependent property, and the temperature in the neighbourhood of the voicecoil is not known. The dynamic viscosity of air used for the reference model was 18.2 x 10^{-6} Ns/m^2. Figure 9 shows the effect of varying the viscosity.

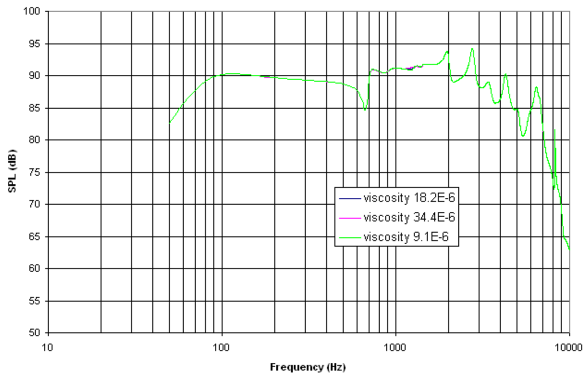

Figure 9: SPL on axis, variation with viscosity

The viscosity variation causes very little difference. This needs further investigation.

4 CONCLUSIONS

From the results above it is concluded that to obtain accurate results it is advisable to use a fully coupled model with an accurate radiation computation and model the details of component connections, in addition to requiring material properties of the components.

5 ACKNOWLEDGEMENT

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6 REFERENCES

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7 APPENDIX – RADIATION FROM AXIAL VIBRATION OF A HEMISPHERICAL DOME

Consider a hemispherical dome of radius R, vibrating with an axial velocity of amplitude V, radiating into a half space. Only the normal component of velocity is coupled to the acoustic vibrations. Also the baffled condition is identical to the condition for a plane of symmetry. Hence the radiated pressure field is identical to that radiated by a spherical surface of radius R vibrating with surface velocity

$$V_r(\theta) = \begin{cases} V\cos\theta & \text{for } 0 \le \theta \le \frac{\pi}{2} \\ -V\cos\theta & \text{for } \frac{\pi}{2} \le \theta \le \pi \end{cases} \tag{6}$$

and radiating into full 3D space. Using standard separation of variable techniques on the Helmholtz equation, the pressure field can be expressed in the form

$$p = \sum_{n=0}^{\infty} a_n P_n(\cos\theta) h_n^{(2)}(kr) \tag{7}$$

where P_n is the Legendre polynomial of order n and $h_n^{(2)}$ is a spherical Bessel function. Multiplying (7) by $P_n(\cos\theta)\sin\theta$, integrating from 0 to π and using the orthogonality properties of Legendre polynomials results in

$$a_n = \frac{-(2n+1)i\rho cV}{2h_n^{(2)\prime}(kR)} \int_0^{\pi} V_r(\theta) P_n(\cos\theta)\sin\theta d\theta \tag{8}$$

Figure 10 compares the closed form solution of (7) & (8) and a Rayleigh integral for a dome of radius 0.01m, vibrating with unit amplitude axial velocity, radiating into a half space of acoustic medium with properties ρ=1.2 kgm^{-3} and c=340ms^{-1}.

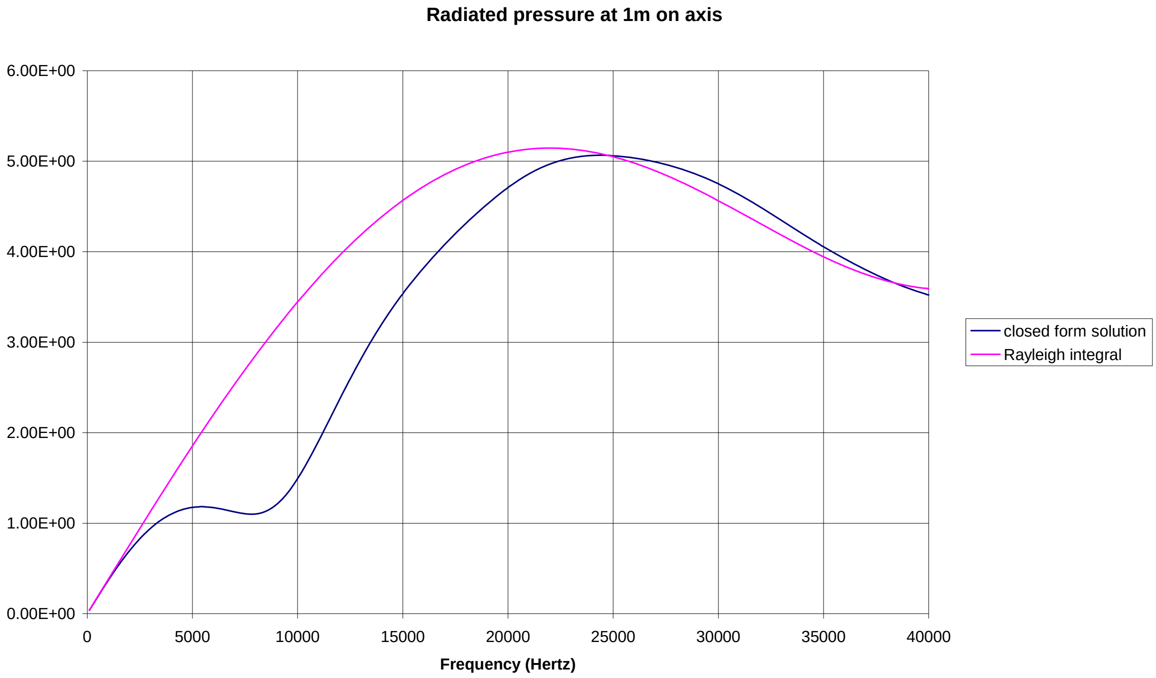

Figure 10: Rayleigh Integral and exact solution for radiation from hemispherical dome