Interior vehicle acoustics up to a high frequency range using a combination of advanced deterministic and statistical techniques

Koen De Langhe[1], Gerard Borello[2] , Koen Vansant[3]

[1] *SIEMENS PLM Software, 3001 Leuven Belgium, E-Mail:koen.delanghe@siemens.com*

[2] *InterAC, 31240 L'Union France, E-Mail: gerard.borello@interac.fr*

[3] *SIEMENS PLM Software, 3001 Leuven Belgium, E-Mail:koen.vansant@siemens.com*

Abstract

The paper addressess the use of advanced simulation technologies in context of vehicle interior and exterior acoustics. At first, a new efficient approach for modeling the exterior sound field of a car body is described and illustrated. The new approach is based on Adaptive Order technology within a Finite Element based framework. This innovative technique is in particular interesting as it provides the possibility to perform full vehicle exterior noise studies much faster than any other method around.

Secondly, an innovatie approach to model high frequency interior acoustics in the framework of Statistical Energy Analysis is introduced as well. The innovation relates to the modeling of the so-called mass law behavior when studying noise transmission through complex panels. Both approaches are illustrated through examples.

Introduction

When implementing vibro-acoustic simulation in a product development process, for example in view of improving interior noise of a vehicle, it is key to do such predictions in an accurate and efficient way. Accuracy relates both to the way the loading (the excitation, sources) is described but also the way the system itself – the vehicle - is modeled.

As far as the excitation is concerned, when dealing with airborne noise problems in particular, then several types of source are to be considered: Engine and components,Tire and Exhaust.

These noise sources are scattered within the complex environment they are installed in. For example the engine bay with its many systems and components. The noise from these sources could be amplified (e.g. through resonances in the cavities) before it loads the vehicle body. In view of body and trim optimization for improved acoustic comfort, being able to have an accurate loading description is key. Such a simulation model should be able to predict the pressure loading onto the body panels across a wide frequency band. It should be able to capture accurately the complex scattering (components, pipes,..) and the sophisticated physics inside the engine bay such as porous materials. In addition, solutions should be efficient, and fast i.e. not taking weeks to solve.

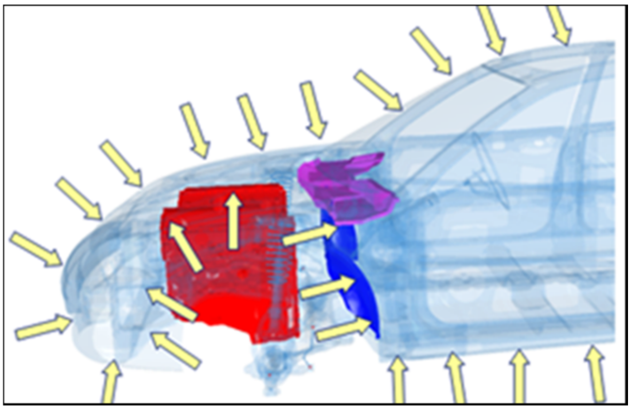


Figure 1: schematic representation of the loading of the vehicle body due to powertrain sources

The first objective of this paper is to present a new technique allowing to carry out such simulations accurately and efficiently.

Now, once the pressure loading onto the vehicle body is obtained, then the next step is to apply that excitation onto a SEA-based vehicle body model in order to simulate and analyse the airborne interior noise up to high frequencies. A key aspect in the modeling of the interior noise with SEA is related to transmission of sound through the individual body panels and more specifically, the mass-law dominant transmission.

The 2nd part of this paper describes a new technique in the SEA framework to allow for indirect transmission in a novel way. Instead of explicitly modeling an indirect transmission path, the new approach works with so called, non-resonant energies‘ within the subsystems.

Part 1: Exterior Vehicle Acoustics and Panel Loading using FEMAO

When applied to wave propagation problems, the conventional FEM method is known to suffer from the so-called pollution effect, which is linked to cumulative dispersion errors. Since the dispersion error increases with frequency, the mesh resolution required to obtain a reasonable accuracy also increases with frequency and the use of the conventional low-order FEM is restricted in practice to low frequencies. It is a well-known fact that high-order FEM or p-FEM, which resorts to higher-order approximations, allows diminishing the resolution requirements and therefore the total number of degrees of freedom to solve for a particular Helmholtz problem with a targeted accuracy.

Beriot et al. (1) have therefore tried quantifying the real benefits of a p-FEM approach on a full three-dimensional Helmholtz problem. They found that the solving time required to compute their test case (a square duct section with a plane wave propagating through it) with a given required accuracy was diminishing with increasing element order. In other words, for the duct example at hand, fewer higher order elements seemed to be more efficient compared to more lower order elements to predict accurately the pressure field response. The total number of degrees of freedom (DOF) being less in the higher order element case. Next to this superior efficiency of higher order elements, the even more important idea behind FEMAO is to adjust the order P_e^f of each element automatically prior to the computation depending on the frequency f , the local speed of sound c(M) , itself depending on the mean flow speed, and the element dimension h , in order to guarantee a predefined accuracy.

Essentially, higher orders are used at high frequencies and/or for large elements and low orders will be employed at low frequencies and/or for small elements. Figure 2 illustrates some higher order shape functions on a hexagonal element. It makes clear that FEMAO allows for a much coarser discretization compared to conventional FEM methods which use only first or second order shape functions.

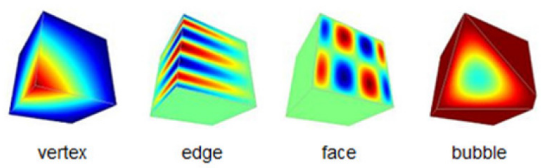


Figure 2 higher order shape functions**,**

A performance comparison between FEM and FEMAO is carried out on the rectangular duct Helmholtz problem for a full frequency sweep. A full frequency range of f = [100,4000]Hz is considered which corresponds to a non-dimensional Helmholtz number range of [1.84,74] , with

d =1m . A thin uniform linear FEM mesh, valid up to f 3500 Hz max using a 8 elements per wavelength rate, is generated as shown in Figure 3, top-left. A uniform coarse FEMAO mesh is also generated with the same upper frequency limit (the upper max f of a FEMAO mesh is reached when adaptive rule indicates that f > 10e P).

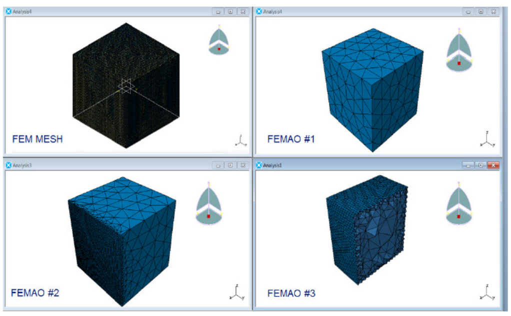


Figure 3 FEM cube mesh (upper left), FEMAO cube mesh #1 (upper right), mesh #2 (lower left) and mesh #3 (lower right).

To test the ability of the FEMAO solver to cope with strong local mesh refinements, two other meshes were generated as displayed in Figure 3. These have the same upper frequency limit of f 3500 Hz max , but have one or all faces with refined elements, to represent the situation in which such a refinement is required to accurately capture the geometry of a more complex structure (a car for instance).

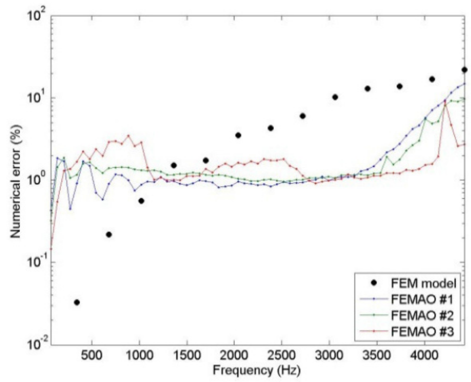


Figure 4 - Comparison with FEM and FEMAO

The numerical results obtained for the four meshes are compared with the analytical solution and the numerical error is displayed along the full frequency range in Figure 4.

The FEM model yields larger errors at higher frequencies. On the other hand, the error for FEMAO is stable and remains close to 1% on the full frequency range. This result is quite remarkable for the FEMAO meshes #2 and #3 and indicates that the FEMAO solver can guarantee a close to constant accuracy even in the case of highly non-uniform meshes.

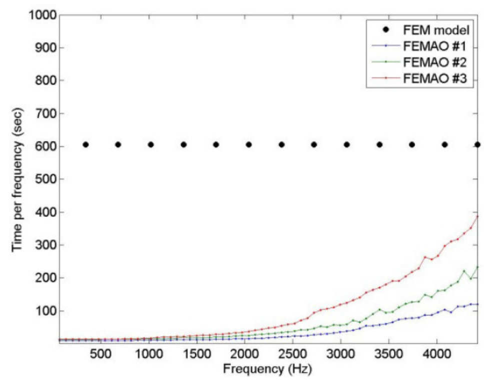


Figure 5 - Comparison of the timings.

In Figure 5, the time required per frequency is displayed for the four models. As expected, it is constant for the FEM, while it increases with frequency for the FEM AO models.

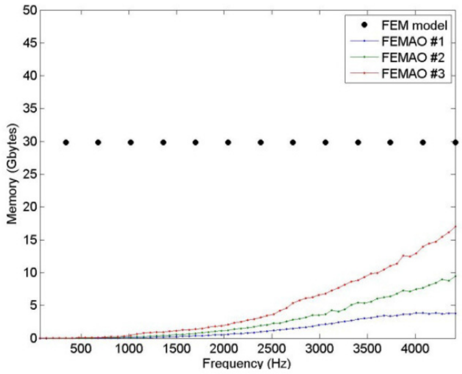


Figure 6 - Comparison of the memory required

Application to full vehicle exterior

The previous technology, FEMAO, is now applied to a full vehicle model. 2 models are made: one with standard FEM elements and another one using FEMAO.

The model with standard FEM elements resulted in 30 Million TETRA 4 elements, corresponding to roughly 5.2M Nodes. The model is shown below.

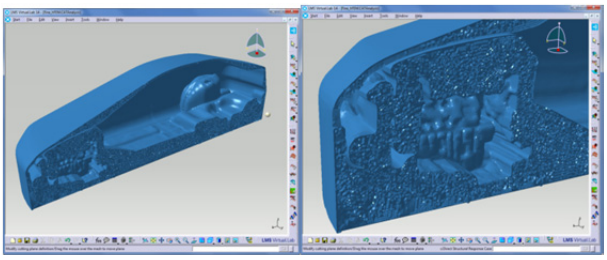


Figure 7 – FEM model using standard FEM

The FEMAO model is build with very coarse elements, and for this model, during the mesh creation, it is not important to take into account the ‚*6 elements / wavelength*' criterion as one typcially does when meshing with standard FEM elements.

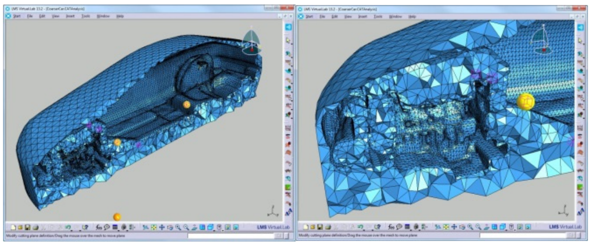


Figure 8 – FEMAO mesh

The FEMAO model contains roughly 500 000 Tetra elements and 130 000 physical nodes. Clearly such a model is relatively small and therefore, handles very well.

For both models, a point source (acoustic monopole) is placed at the lower part of the tire. Analysis is done for every 12th octave between 100 and 4000Hz, resulting in 64 frequencies to calculate. The Acoustic FEM solver integrated in Virtual.Lab Acoustics uses MUMPS technology (SYSNOISE solver). Whileas the standard FEM models takes around few days to calculate the full spectrum, the *FEMAO model does the job in* ***just over one hour*** on a 20core system. The lower frequencies, for which only lower orders needs to be considered for the elements, run extremely fast in the case of FEMAO. Whileas the lower frequencies in the FEM approach takes a similar CPU time for all the frequencies.

When it comes to accuracy, both models provide similar results as regards to the pressure loading on the body panels.

Part 2: modeling of SEA mass law transmission

Introduction

Air-borne high frequency interior acoustics based on SEA has the particular advantage of efficiency (fast simulations) and the possibility to address a wid frequency band, typically from the first eigenmodes onwards to high frequencies. SEA-based STL has proven to be accurate, provided the SEA modelling is properly done for the particular structure. While as with FEM, an engineer mostly focuses on the geometry and meshing, the situation is different for SEA. For SEA, it is more about modelling the physics.

Mass Law

Considering sound transmission through a flat panel, below critical frequency, the STL is dominated by a Mass Law curve, corresponding to the non-resonant energy path between the incident and receiving room. Above the critical frequency, the STL is dominated by the resonant path as shown in figure 9 below.

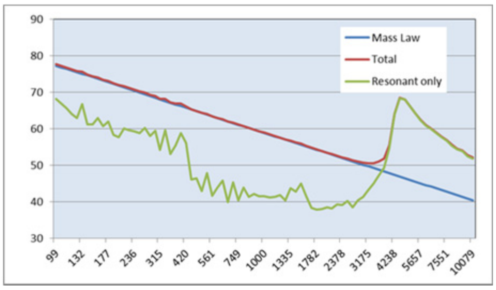
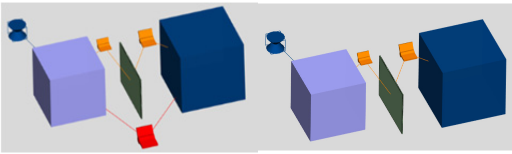


Figure 9: Contribution of mass law and resonant path

The mass dominated path (blue curve) is often referred to as the indirect path, because the energy exchange between the 2 rooms happens indirectly though non-resonant motion of the partition. Such phenomenon does not only occur for Sound Transmission through panels but also happens between pure structural connections. While as the indirect path in the case of a simple partition is easy to formulate (straight 'mass' line in blue in the graph above), this is not necessarily the case for complex partitions (double walls, treated panels, corrugated panels) or structurally connected subsystems. In an effort to generalize the indirect connections and hence extending the scope of the indirect transmission from standard mass law to generalized connections, the concept of non-resonant energy of subsystems is introduced. Conceptually it means that when a partition is connected to 2 rooms, the energy of the plate radiating into either cavity is the combination of the resonant energy, the non-resonant energy from the plate modes excited above their eigenfrequency (mass controlled) and the non-resonant energy from the plate modes excited below their eigenfrequency (stiffness controlled). Essentially the plate has 3 distinct energies in the frequency band of interest: the resonant energy, mass controlled non-resonant energy and stiffness controlled non-resonant energy. In standard SEA, only the mass law is typically considered and mostly and only for panel transmission. This is simplified to a straight line (mass law line). A more general formulation is where the 3 types of energies are considered through calculating also the effective radiation efficiencies of these off-resonant excited modes as opposed to approximating the mass law by a single straight line. Not only does this approach provide higher accuracy but also a major benefit is that the indirect connections should not be modelled explicitly as they are implicitly taken into account through the non-resonant energies of the subsystems.

This technique is implemented in SEA+, the SEA modeling software from InterAC.



Figuure 10: left: standard SEA model with explicit modeling of the indirect path (colored in red). right: new model with non-resonant energies.

For a single panel, the STL calculated by Standard Mass law versus the new approach based on non-resonant energies is shown below. Both models match well.

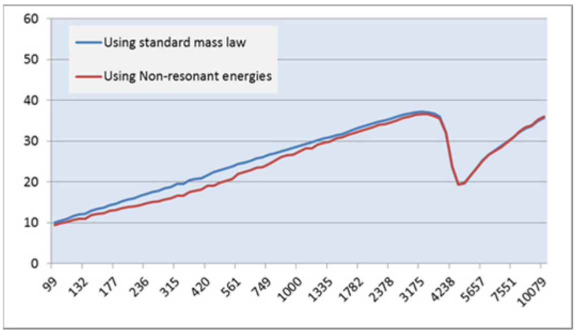


Figure 11: STL curves comparing mass law with new approach based on non-resonant energies

The new technique is applied on a model of a truck, for airborne noise. The objective is to compare the interior SPL obtained by the standard method (mass law connection) versus the new approach (non-resonant energies of subsystems).

The model contains roughly 80 subsystems. Exterior pressure loading is applied onto the panels and the interior SPL is calculated. For this model, we were able to have very good correlation with measurements as illustrated below:

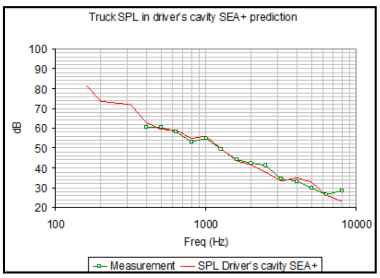


Figure 12: Comparing SPL of the truck interior: measurements versus simulation (SEA+).

The question is how the new technique (non-resonant energies compares with the standard method (mass law connection). For this study, the existing truck SEA model is re-used. The model is adapted such that the non-resonant energy paths are explicitly removed and the non-resonant energy of the subsystems is explicitly added.

The interior SPL is then compared between the standard SEA Model (including explicit mass law connections) and the new SEA approach (with non-resonant energies of the subsystems).

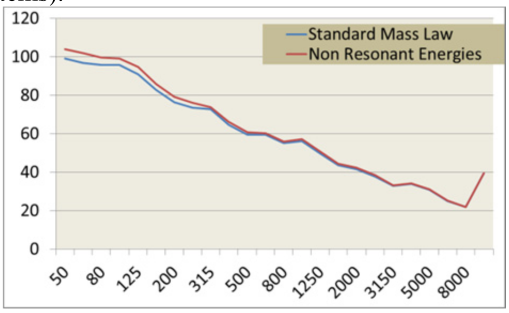


Figure 13: Comparing SPL between the 2 approaches

As can be seen from figure 13, the 2 approaches provide approximately the same results and all compare very well with measurements.

Summary

This paper presented 2 new technologies that can be used in context of airborne cabin interior acoustics. On the one hand, the modeling of the exterior noise field in view of panel loading (airborne excitation) using FEM Adaptive Order technology. Whileas in the past such calculations took days on a cluster, today with the new technique one can obtain a full spectrum in a matter of hours.

Secondly, the mass law transmission, a key energy flow path for airborne noise, is modeled using a new technique based non-resonant energies. This new technique is much simplier from modeling viewpoint because it is not needed to model explicitly the indirect connection. The latter is automatically take into account by consideration of the non-resonant energy of the subsystems.

Literature

[1] Beriot H, Prinn A, Gabard G.: On the performance of high-order FEM for solving large scale industrial acoustic problems, ICSV 20, 2013

[2] Bériot H, Tournour M.: On the locally-conformal perfectly matched layer implementation for Helmholtz equation, NOVEM Noise and Vibration: Emerging Methods, 2009

[3] De Langhe K, Vansant K.: Advanced techniques for vibro-acoustic panel loading calculations, including Automatically Matched Layer (AML) and Frequency Domain Multipole Boundary Element Methodologies (FMBEM); Automotive Acoustics Conference ATZ, 2011