

Coupling CFD with Vibroacoustic FE Models for Vehicle Interior Low-Frequency Wind Noise Prediction

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ABSTRACT

With the reduction of engine and road noise, wind has become an important source of interior noise when cruising at highway speed. The challenges of weight reduction, performance improvement and reduced development time call for stronger support of the development process by numerical methods. Computational Fluid Dynamics (CFD) and finite element (FE) vibroacoustic computations have reached a level of maturity that makes it possible and meaningful to combine these methods for wind noise prediction.

This paper presents a method used for coupling time domain CFD computations with a finite element vibroacoustic model of a vehicle for the prediction of low-frequency wind noise below 500 Hz. The procedure is based on time segmentation of the excitation load and transformation into the frequency domain for the vibroacoustic computations. It requires simple signal processing and preserves the random character as well as the spatial correlation of the excitation signal. The aeroacoustic load is applied on the entire outer surface of the vehicle body. The computed results are validated against wind tunnel measurements on a production vehicle. The numerical and experimental investigations provide some insight into the mechanisms involved in underbody wind noise.

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INTRODUCTION

Nowadays vehicles are mainly the product of a digital development process. Crash, durability and NVH performance are optimized digitally long before any hardware is available for the refinement process. Wind noise prediction is still not a standard discipline in the framework of this computational process. One of the main reasons has long been the computational burden of predicting unsteady load exciting the structure. For the digital process to perform best, it is necessary to be able to tackle all relevant disciplines in parallel and make sure that design decisions which are beneficial for one field are compatible with the other disciplines. Thorough digital testing is the key to avoiding costly redesigns in the hardware refinement phase. This calls for implementing wind noise simulation.

Wind noise, being generated by the flow induced pressure fluctuations on the vehicle, can enter the vehicle by many paths. Noise entering the vehicle through the side windows excited by the flow over the A-pillar and the side mirror in the 2000 to 5000 Hz frequency range was given strong attention in recent years. It was shown that two types of excitation mechanisms should be considered: a short wavelength high energy turbulent convective field which does not couple well with the structure and a lower energy acoustic field involving longer wavelengths that excites the structure in a much more efficient manner. Hartmann et al. [1] highlighted these phenomena based on experimental and numerical investigations of the exciting field.

Several authors have tackled the numerical prediction of the transmission through the side glass based on CFD computations. Moron et al. [2] and Graf et al. [3] used SEA models for the window structure and the interior cavity producing results up to 5 kHz. In these approaches, the hydrodynamic and the acoustic loading present in the compressible CFD results are separated and applied as adequate power inputs. In the description of the hydrodynamic field, simplifications are used "in the form of models to represent the pressure cross spectrum in terms of auto power spectra and coherence decay rates in streamwise and spanwise flow directions" (see [2]).

Van Herpe et al. [4] presented an application on a windshield and two front side windows up to 1.6 kHz using deterministic finite elements both for the glazing structure and the cavity. With the same approach Cabrol et al. [5] demonstrated the feasibility of such a computation for a side window coupled to a car cavity up to 4 kHz. In this case incompressible CFD computations were used, and the acoustic wall-pressure fluctuations were predicted based on Lighthill analogy. Hallez et al. [6] presented a validation on a simplified 1/2 scale setup for the same type of problem up to 4 kHz also using finite elements. This latter paper discusses in details the methodology for precisely and efficiently taking into account the partial correlation of the wall-pressure fluctuations (WPF) spatial distribution. The present work uses the method recommended by these authors. While the reduced size of the CFD/FE coupling interface appears to be manageable, the main challenge in the side-glass problem for

vibroacoustic finite elements resides in the computation of the interior field at frequencies above 1 kHz which requires a very fine discretization of the cavity. This problem can be avoided by a hybrid technique combining finite elements for the window structure with a SEA model of the cavity. Blanchet et al. [7] demonstrated such a methodology on a simplified setup.

Low-frequency underbody noise is the second area of effort in wind noise prediction. It mainly concerns the frequency range below 1 kHz. The CFD computations represent a challenging task due to the complexity of the underbody geometry (spoilers, openings, wheels, wheelhouse, exhaust parts). Structural complexity of the vehicle floor including longitudinal- and cross-members as well as beads makes the use of structural SEA non trivial. On the other hand, the extent of the coupling area over the entire floor and the greenhouse makes the application of a detailed time varying CFD load a challenging task when using finite elements. Powell et al. [8] presented the only application of CFD computations coupled with a vibroacoustic model for underbody wind noise known to the authors. It is based on the same methodology as [2], this time with an SEA model derived from a finite element model of the vehicle.

Our goal is to explore the possibility of staying with a finite element description of the vehicle structure and interior cavity while coupling it with the exciting field without making any assumptions. This approach is expected to require longer computation times but reduces the level of multidisciplinary competence (assumption on flow structure, SEA modeling) necessary for carrying out the investigations in the framework of a development process. The goal was to implement an excitation for a standard commercial FE code (NX Nastran) and be able to compute the response within the code and take advantage of its optimization capabilities.

This paper mainly puts together solutions that were used on a smaller scale, i.e. for transmission through the glazing (see [6]). In order to highlight the benefit of the investigated method, its results are compared with the ones from a simplified approach.

UNDERBODY NOISE

Noise generated in the underbody region is one of the dominant sources in the frequency range up to 1 kHz (see [8, 9]). Measurements carried out with a skirt blocking the underbody flow as shown in Figure 1 exhibit up to 12 dB lower noise levels inside the vehicle (Figure 2).



Figure 1. Example of vehicle setup with a skirt for blocking the underbody flow.

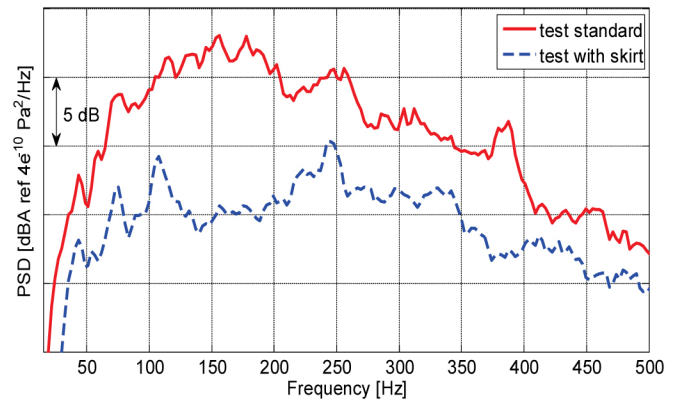


Figure 2. Impact of blocking the underbody flow on the sound pressure level at the driver's outer ear (experimental results at 140 km/h).

CFD SIMULATION

One of the main challenges of low-frequency CFD computations is the prediction of both hydrodynamic and acoustic excitations without the artificial numerical background noise which usually arises in hexahedral meshes at 1:2 cell coarsening boundaries. Achieving this goal requires a good mesh quality and adequate numerical parameters. A major improvement in mesh quality was reached with a polyhedral grid approach. Figure 3 shows the effect of a polyhedral mesh on eliminating spurious sources away from the vehicle's surface.

The complete CFD domain is meshed with 65 million cells with a 4 mm resolution on the vehicle surface. For later mapping on the FE model surface, the pressure time signal is stored for 3 million cells, which cover the entire vehicle surface.

The CFD computations were carried out using the commercial software Star-CCM+ [10] using a compressible solution scheme along with Detached Eddy Simulation (DES) for turbulence modeling.

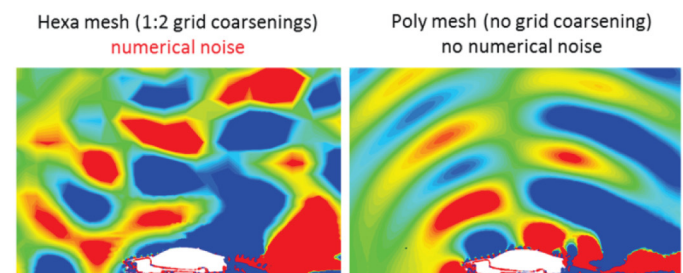
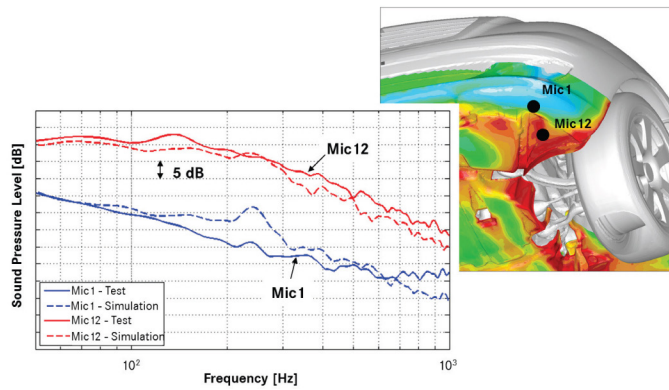
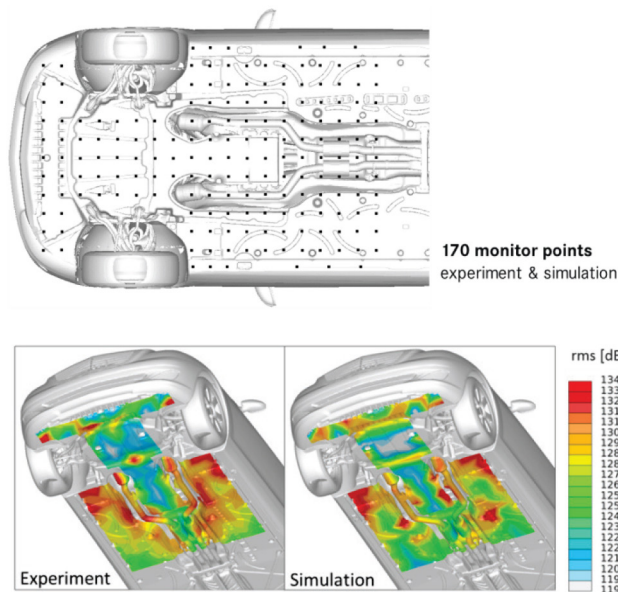


Figure 3. Pressure distribution in the vertical middle plane. Example of the reduction of spurious sources using a polyhedral mesh.

Previous investigations have shown that the pressure prediction on a vehicle body can be achieved with good accuracy at the front of the vehicle as in Figure 4A. This point to point agreement deteriorates in high gradient regions downstream, as small spatial deviations can result in very large variations. Nevertheless, the qualitative agreement of the RMS pressure distribution over the floor seems acceptable for further use of the data in the computational chain (see Figure 4B).



A. Wall-pressure fluctuations in the frequency domain



B. Spatial distribution of RMS averaged wall-pressure fluctuations

Figure 4. Wall-pressure fluctuations in the underbody region. Comparison of experimental and numerical results.

Readers interested in a description of the complex flow phenomena in the underbody region can refer to the investigations carried out by Crouse et al. [11]

FINITE ELEMENT MODELING

The finite element model used in the following computations comes from a development stage NVH trimmed body model of a production car including the complete vehicle structure with closures along with distributed and lumped component mass. Most of the trim parts, including the carpets are modelled as distributed mass. The structural model has about 2.2 million elements, 2.2 million nodes, and 11 million degrees of freedom. About 200,000 surface elements are subject to the wind excitation. The cavity model has about 350,000 degrees of freedom including the passenger space, the trunk and the door cavities.

Frequency response calculations up to 500 Hz involve around 11,000 structural modes (up to 700 Hz) as well as around 600 acoustic modes (up to 1 kHz).

COUPLING CFD EXCITATION WITH A FE-MODEL

The objective of the present work is to investigate the possibility of using CFD time data for computing the interior response to wind loading on the entire vehicle surface using a commercial finite element code (NX Nastran). The number of loaded triangular and quadrangular elements is about 200,000. The available signal length is between 1 and 2 s, with a sampling frequency of 5 kHz.

The vibration amplitudes being very small under wind excitation in the acoustic range, it is not expected that the structure motion will influence the air flow. The CFD computations are therefore carried out over a rigid body.

The mapping from the fine CFD mesh onto the structural finite element mesh is carried out by the CFD software that outputs the spatial average of the pressure on each structural element for each time step [10].

In order to investigate numerous structural configurations and perform optimizations, the finite element computational process should be kept as simple and as fast as possible. The quantities to be computed are mostly the power spectral density of the sound pressure level at a few positions and optionally the power spectral density at some structural response points. We are only interested in the diagonal terms of the cross-spectral density matrix, S_{yy} , of the response signals. The relationship between S_{yy} and the excitation cross-spectral density S_{xx} is given by

$$S_{yy} = HS_{xx}H^H \quad (1)$$

Where H is the transfer function between the excitation nodes (the pressure at the surface elements) and the pressure or displacement at the response nodes.

For $X(t)$, the vector of time signals, the estimator of the cross-spectral density matrix $S_{xx}(\omega)$ is obtained by Fourier transforming N_{seg} overlapping or non-overlapping time segments and averaging the product of the vector with its conjugate transpose.

$$S_{xx}(\omega) = \frac{1}{N_{seg}} \sum_{n=1, N_{seg}} X_n(\omega) X_n^H(\omega) \quad (2)$$

Choosing between Time and Frequency Domain

Vibroacoustic computations could be carried out in the time domain, preferably using a modal model of the structure. This process guarantees the integrity of the signals statistics. The resulting time response can then be Fourier transformed and analyzed in the frequency domain like an experimental signal. However this type of computation tends to be inefficient and makes it impossible to use frequency-dependent elements such as acoustic impedances. In order to keep the computational load reasonable, low-pass filtering of the excitation and adequate downsampling of the time signal are necessary.