

Flow Induced Interior Noise Prediction of a Passenger Car

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ABSTRACT

Prediction of flow induced noise in the interior of a passenger car requires accurate representations of both fluctuating surface pressures across the exterior of the vehicle and efficient models of the vibro-acoustic transmission of these surface pressures to the driver's ear. In this paper, aeroacoustic and vibro-acoustic methods are combined in order to perform an aero-vibro-acoustic analysis of a Mercedes-Benz A-class. The exterior aero-acoustic method consists of a time domain incompressible Detached Eddy Simulation (DES) and an acoustic wave equation. The method is extended in this paper to account for convection effects when modelling the exterior sound propagation. The interior vibro-acoustic model consists of a frequency domain Finite Element (FE) model of the side glass combined with a generalized Statistical Energy Analysis (SEA) model of the interior cabin. The method is extended in this paper to account for additional geometric details that affect the direct and reverberant fields within the cabin. The overall approach is presented and validated against vibro-acoustics tests and wind tunnel measurements for two separate mirror designs. The aero-vibro-acoustic method achieves very good results and accurately predicts the change in interior noise resulting from an exterior body design change.

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INTRODUCTION

Wind noise is becoming increasingly important during automotive development for improving passenger comfort. This is partly due to significant reductions in road and engine noise in recent years and partly because of the increasing electrification of future vehicles. For many vehicles wind noise is already the dominant contributor to interior noise at higher vehicle speeds.

Unlike external aerodynamics, interior wind noise cannot be estimated from clay models or CFD simulations of rigid structures alone. Small changes in the exterior shape of a vehicle can often cause significant differences in interior noise levels. In some instances, the overall levels on the exterior of a vehicle can decrease while the interior noise level increases. This is due to the complexity of the exterior sources and the spectral and spatial filtering provided by the side glass and interior acoustic space. Accurate assessment of different exterior body design changes therefore requires information about the interior acoustic response at the driver's ear. Physical prototypes are typically not available at an early development stage when a vehicle is far away from series production. Cost reductions and shorter development cycles mean decisions about vehicle shape need to be made as early as possible. It is therefore essential for vehicle development to have reliable and validated simulation methods for predicting interior wind noise.

One of the major challenges for predicting side mirror interior wind noise is the calculation of the exterior acoustic field close to the side glass and mirror. The acoustic field is often the dominant contributor to interior noise at higher frequencies despite being much lower in level than the convective turbulence [1]. The fluctuating surface pressures typically consist of distinct concentrations of energy at the convective and acoustic wavenumbers. In principle, the acoustic field can be estimated from measurements of its wave-numberfrequency spectra [2] made across various lines or patches on the side glass. However, this approach can be challenging due to the limited spatial data available when performing wind tunnel measurements. The acoustic field is also not spatially homogenous across the side glass and so measurements at one particular location on the glass may not be particularly representative of the field in other locations. Direct CAA approaches such as unsteady compressible CFD simulations overcome this limitation but there can often be numerical difficulties [3, 4] associated with the disparity of scales, disturbances and acoustic reflections at boundary conditions and at mesh transitions.

An alternative approach is to instead describe and solve the exterior acoustic and convective pressure fields separately by using an acoustic analogy [5], i.e. a "hybrid" CAA approach. Using a one way coupled acoustic analogy means that it is not possible to model applications in which acoustics modifies the flow field. Such feedback may, for instance, be necessary in order to describe resonant buffeting phenomena but can be neglected for most low Mach number automotive applications.

When using hybrid CAA approaches it is important to choose an acoustic analogy carefully since there can be significant differences in the assumptions of different analogies (particularly when the response is needed in the near field of various aero-acoustic sources). The most important requirement for a hybrid CAA is that it can provide an accurate description of the acoustic near field on the outside of the glass. The aero-acoustic sources are often distributed throughout a volume so a volume based acoustic analogy is typically needed (rather than, for example, a surface based analogy). In principle, a frequency domain acoustic analogy could be used. However, frequency domain acoustic analogies that distribute sources throughout a volume are often not practical since they require the storage and processing of large amounts of time domain volume data (this can lead to the exterior aero-acoustic analysis taking longer than the original CFD analysis [4]). A more efficient approach is to embed a time domain acoustic analogy into a flow code and make use of the high performance computing (HPC) architecture of the flow code when designing this acoustic analogy.

The transmission of noise from the exterior surface of the side glass to the driver's ear requires a vibro-acoustic model. The type of vibro-acoustic methods that are used in this model depend on the frequency range of interest. For side mirror wind noise the frequency range of interest is often from 1 kHz to 8 kHz. The side glass has a relatively small number of structural modes across this frequency range and so can be modeled using FE. In contrast, the vehicle cabin has a very large number of modes across this frequency range and so the use of FE or Boundary Elements to describe the entire interior acoustic space is typically not practical. An SEA model provides an efficient way to describe the interior acoustic space. In a traditional SEA model [6] the vehicle interior is partitioned into a small number of subsystems and the response is described in terms of the average response within each subsystem. Such an approach provides a "lumped parameter" description of the spatial response within the vehicle that can correctly predict the overall space average response (due to conservation of energy). However, the coarse partitioning does not provide information about the spatial gradients in the response within the vehicle interior. As discussed in later sections, these spatial gradients can be significant for side mirror wind noise, particularly in the vicinity of the driver's headspace. In a traditional SEA model, the direct field from the side glass is often not accounted for in detail. In order to accurately model the acoustic response it is necessary to generalize the description of the direct and reverberant fields used in a SEA cavity. This paper describes a new method which combines finite elements, boundary elements and a generalized boundary approach to SEA for accurate and efficient predictions of interior wind noise across a broad frequency range.

The next section describes the hybrid CAA and vibro-acoustic methods that are used in more detail. The experimental setup is then discussed as well as the aero-acoustic and vibroacoustic model of the Mercedes-Benz A-class. Finally the results of the simulation strategy are discussed including the validation of the standalone vibro-acoustic model with vibroacoustic hammer tap tests. The results give insights into the exterior acoustic field and the origins of acoustic waves responsible for the side window excitation, as well as the accuracy of the simulation strategy. In addition, the effects of convection on the acoustic field are investigated. The models are

validated by comparing the predicted interior noise levels associated with an exterior design change with those obtained through wind tunnel measurements.

METHODS

Hybrid Computational Aero-Acoustics

It is essential in automotive applications to accurately model the acoustic near field generated by various aero-acoustic sources (since the side glass is in the "acoustic near field" of these aero-acoustic sources). This section discusses the acoustic analogy that is used in more detail. There are several available acoustic analogies that follow Lighthill's original idea [7] of calculating equivalent noise sources which can be subsequently used for sound propagation.

For near field acoustics the problem with Lighthill's original equation is that it solves a total pressure rather than an acoustic pressure in the vicinity of the car when source terms are present [8]. In order to get rid of the hydrodynamic pressure fluctuations which mask and can distort the acoustic near field content [3, 4] previous studies [3, 9] have shown that one can derive a wave equation for the acoustic pressure that is valid in the entire domain by separating the pressure fluctuations p' in Lighthill's equation into hydrodynamic p^{inc} and acoustic p^a perturbations [10].

One drawback of this acoustic wave equation is that it doesn't take convection effects into account and only describes sound propagation in a medium at rest (like Lighthill's equation does). To overcome this limitation one has to start the derivation from a different point. The final result is a generalized wave equation that accounts for convection (and which collapses to the previous result when convection is negligible).

Based on the Navier-Stokes equations, Ewert and Schroeder [11] derived a set of acoustic perturbation equations by separating the velocity field into a time averaged mean velocity $\bar{U}$, a solenoidal vortical perturbation u^v and an irrotational acoustic perturbation u^a . Additionally a source term filtering ensures that only pure acoustic modes are excited. The APE-1 equation system is given by [11]:

$$\frac{\partial p'}{\partial t} + c^2 \nabla \cdot \left(\bar{\rho} u_i^a + \bar{U}_i \frac{p'}{c^2} \right) = c^2 q_c \approx 0 \quad (1)$$

$$\frac{\partial u_i^a}{\partial t} + \nabla(\bar{U}_i \cdot u_i^a) + \nabla \left(\frac{p'}{\bar{\rho}} \right) = q_m \approx \nabla \Phi_p \quad (2)$$

If one only considers vortex sound at small Mach numbers which is the case in external automotive aero-acoustics the different source terms reduce to the major turbulent induced sound source $\nabla \Phi_p$. For small Mach numbers this source term can be calculated from the incompressible pressure fluctuations by [11]:

$$\nabla \Phi_p \approx \frac{\nabla p^{inc}}{\rho_\infty}$$

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In contrast to a single wave equation the acoustic perturbation equations account for convection and refraction effects. But to get a pure acoustic solution the pressure fluctuations have to be decomposed into a hydrodynamic perturbation pressure and an acoustic perturbation pressure, too.

Starting from the APE-1 system (equation 1 and 2), neglecting all minor source terms except $\nabla \Phi_p$ (equation 3), decomposing the pressure fluctuations p' into p^{inc} and p^a and additionally setting the mean velocity to zero ($\bar{U} = 0$) in a first step, one can derive an equation system close to the APE-2 system without convection effects:

$$\frac{\partial p^a}{\partial t} + c^2 \nabla \cdot (\bar{\rho} u_i^a) = - \frac{\partial p^{inc}}{\partial t} \quad (4)$$

$$\frac{\partial u_i^a}{\partial t} + \nabla \left(\frac{p^a}{\bar{\rho}} \right) = 0 \quad (5)$$

Following Lighthill's derivation, by taking the time derivative of equation 4 and the divergence of equation 5 one can eliminate the acoustic velocity u^a to obtain a scalar wave equation for the acoustic pressure:

$$\frac{\partial^2 p^a}{\partial t^2} - c_0^2 \nabla^2 p^a = - \frac{\partial^2 p^{inc}}{\partial t^2} \quad (6)$$

This results in exactly the same acoustic wave equation derived by directly separating the pressure fluctuation in Lighthill's equation [9]. To overcome the limitation of sound propagation in a medium in rest one can follow the same procedure as sketched above and derive a convective wave equation for the acoustic pressure by not setting the mean velocity field to zero.

In addition to the incompressible pressure fluctuations which can be obtained from an unsteady incompressible CFD calculation, this convective wave equation can use a time averaged inhomogeneous velocity field to account for the convection and refraction effects. For example, the mean velocity field can be computed by a steady state RANS in advance of the unsteady CAA run.

Both acoustic wave equations are implemented in the STARCCM+ software package [12] and can be executed on-the-fly in the time domain along with an incompressible CFD calculation. To limit data transfer and avoid mapping influences, the incompressible flow, the acoustic source terms and the exterior sound propagation are calculated on the same mesh at each time step. The advantages of this strategy are that the hydrodynamic and acoustic solution can be calculated simultaneously. Another advantage is that both solution fields are obtained separately which provides new insights into the near field acoustics, its diagnosis and an analysis of its contribution to the interior noise.

The CFD part is solved by an incompressible Detached Eddy Simulation (DES) which combines the advantages of a Large Eddy Simulation (LES) and an unsteady Reynolds Averaged Navier-Stokes

simulation (URANS). Within the LES region, which requires a highly resolved mesh, the turbulent flow structures and acoustic source mechanisms can be captured accurately. The unsteady RANS part reduces the computational effort and enables the method to be used for larger industrial applications.

Vibro-Acoustics

The vibro-acoustic model used in the current analysis consists of a FE side glass meshed with QUAD8 shell elements along with a single generalized SEA acoustic cavity used to model the interior acoustic space of the vehicle. The cavity model includes various scattering surfaces in the interior of the vehicle and does not require the interior acoustic space to be partitioned into separate SEA subsystems. The model setup and calculations were performed with the commercial vibroacoustic software wave6 [13].

The vehicle analyzed in the current study has a 3.85 mm tempered side glass (ESG) and so a uniform thickness physical property was applied to the elements of the side glass. The damping of the side glass could, in principle, be predicted from a detailed model of the glass run and belt line seals [14]. However, in the current analysis the damping of the side glass was obtained by post-processing mobility measurements performed on an in-situ glass. A frequency dependent glass damping was then used in the model.

The interior cavity absorption could also, in principle, be predicted by including information about the vehicle sound package. However, in the current analysis the cavity absorption was obtained by performing various reverberation time measurements (T60 tests) inside the vehicle.

The fluctuating surface pressures obtained from the hybrid CAA analysis were post-processed and converted to the frequency domain within the vibro-acoustic software. It should be noted that the analysis process described in this paper does not require the fitting of any semi-empirical loads to the fluctuating surface pressure data but instead retains a detailed spatial description of the load. The use of semi-empirical loads homogenized over various spatial regions was found to be not sufficiently accurate to describe changes in interior noise associated with typical design changes.

MODEL SETUP

The aero-vibro-acoustic simulation strategy was applied to a Mercedes-Benz A-class. In order to evaluate future design changes two different side mirror geometries are considered in the measurement campaign. For a detailed comparison of experimental and simulation results, the aero-acoustic and vibro-acoustic models were created to be as close as possible to the experimental setup (and vice versa).

Experiment

The experimental tests were performed in the aero-acoustic wind tunnel of Daimler AG in Sindelfingen. All measurements of the unsteady flow topology and interior noise refer to a free stream velocity of $U_0 = 140$ km/h and a signal length of 30 s. The experimental setup is illustrated in figure 1.