Physical and numerical simulation of the performance of close-fitting partial engine shields

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Introduction

In order to comply with the recently tightened European regulations to be put into force later this year, manufacturers of heavy vehicles need to take effective measures to reduce exterior noise to the required level. Due to the noise characteristics of currently manufactured diesel engines and also as a consequence of the applied standard pass-by measuring procedure, the necessary noise reduction can usually be reached only if the powertrain is equipped with an appropriately tailored close-fitting sound shield, a partial or even a full enclosure. Vehicle manufacturers need reliable but affordable computational tools to enhance their design and optimisation process while designing these noise control elements.

Since the acoustic efficiency of engine enclosures is strongly influenced by various and relatively large openings, no classical calculation methods seem to be appropriate for predicting their performance [1,2]. Numerical, mainly BEM acoustic prediction methods in turn have the potential to be useful for sound shielding prediction applications [3-7]. In order to establish the applicability and the practical limitations of these advanced techniques for the engine sound shielding problem, an extensive study involving parallel FEM and BEM calculations and verification measurements was initiated [8]. The aim of this paper is to compare the results of BEM predictions and laboratory verification measurements for the determination of the insertion loss (IL) of various realistic sound shields, applied on a simple mechanical engine simulator.

Development and characteristics of the engine simulator

In order to enable easy comparison of calculated and measured results, a simple mechanical engine mock-up has been developed and used throughout the laboratory experiments. (Note that full-scale verification experiments with real-life truck engines are in preparation.) The mock-up, a relatively small, welded steel structure, was designed such that it has more or less similar modal behaviour like a real-life engine block. For the sake of realistic modal frequencies a number of trial calculations were performed by using a standard structural FE software package (MSC NASTRAN). As a result, the final dimensions were selected to be 400 mm x 300 mm x 150 mm, plate thickness 5 mm, with one symmetric bulkhead inside and a closing steel plate on top but not below (thus leaving the bottom edges of the structure freely vibrating). According to FE predictions this structure has 14 vibration modes between 300 Hz and 1000 Hz. This is assumed to be appropriate for simulation purposes, even if the vibration of real engines is composed of much higher numbers of forced harmonic components in this frequency range.

The development and fabrication of the mock-up was completed by performing an extensive experimental modal analysis (EMA) test. Normal accelerations were measured in 296 measurement points. The FRFs, referenced to the input force provided by an electrodynamic shaker, were calculated and stored for later use, both as input data for modal extraction by using standard EMA methods and as direct inputs (converted into velocity boundary conditions) for the acoustic predictions. The measurements and the subsequent modal analysis were performed by means of an LMS CADA-X measurement system. A typical structural mode shape is shown in Fig. 1.

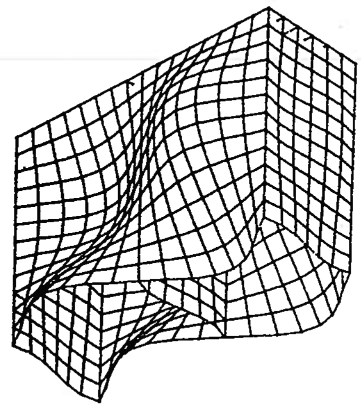

Fig.1. *Mode shape of the simulator*

Shield/enclosure models

The sketch of the used measurement set-ups, together with the measurement planes consisting of the field points used for sound measurements and calculations are shown in Fig. 2.

Two essentially different shields were applied: a simple steel plate, having identical dimensions as the largest side panel of the test source, and a modular shield system, accommodating a wide range of various shielding elements around the source. The *steel plate* (with a thickness of 3 mm, provided with 10 mm thick structural damping material) was placed at 50 mm distance from the source (see Fig. 2.a). The basis of the *modular set-up* is a rigid steel frame fixed to a wooden base plate, onto which various shield elements (rigid steel plates, flexible plastic or aluminium plates, untreated or covered with sound absorbing lining, with or without apertures) can be screwed at a constant distance of 75 mm from the source.

Special attention was paid to a configuration where the two smallest sides of the modular shield are left open. This combination, aimed at modeling a real-life engine shield that is very often open in front and at the back to ensure good cooling, will be referred to as a *tunnel-type shield* below (Fig. 2.b.),

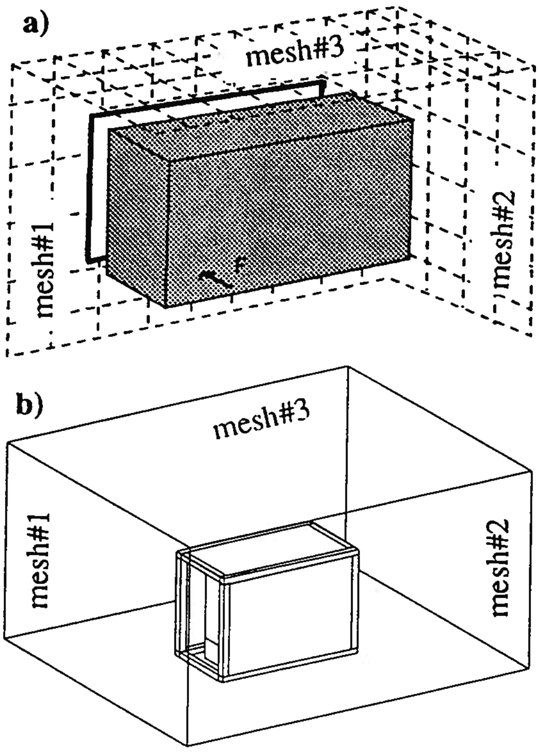

Fig.2. *Shield/enclosure models and planes of the applied field point meshes*

Acoustic experiments

The effect of the sound shields was evaluated in terms of Insertion Loss values, based on measurements of sound pressures and/or sound intensities averaged along measurement planes with and without the shield. In order to be in accordance with the numerical calculations where the sound field was calculated for 1 N input force, the measured sound field quantities were normalised to unity autospectrum of the input force, too. The excitation, provided by an electrodynamic shaker, was band limited random noise between 200 and 3000 Hz.

All measurements were performed in a semi-anechoic room, either in discrete points selected along various plane field point meshes or by using manual surface scanning along the same or similar meshes. The insertion loss values were calculated both in terms of sound pressures and intensities. No essential differences were observed in between, therefore only IL curves obtained from sound intensities are reported on throughout this paper.

Numerical predictions

The peculiarity of the sound shielding problem is that it is both a radiation and a scattering problem at the same time. The radiation part of the problem can be easily solved by using either the collocational or the variational BEM approach. However, in order to allow for the presence of a thin, non-closed obstacle being non-contiguous to the radiator, the indirect approach is the only viable selection. The results reported herein have been obtained by using the indirect variational BEM option of the commercial software package SYSNOISE [9].

Our simple source could be efficiently described by a modal database, but a more general source description can be obtained if the measured data are used directly, converted to the form of prescribed velocity boundary conditions on the surface of the radiator. As to the shield, the BE problem was solved by imposing zero velocity (rigid shield) or finite impedance (acoustic absorption) boundary conditions. Limited by the first structural mode from below and by the applied numerical BEM surface mesh resolution from above, the selected frequency bandwidth was 350 Hz to 1420 Hz with a frequency step of app. 3.8 Hz. During the post-processing phase the normal sound intensity components and the sound pressures were calculated in discrete points of, and then averaged along, the same field point meshes as used in the course of measurements. Eventually, the IL spectra were calculated by using a separate, small user program.

Comparison of measured and predicted insertion loss spectra

Three different sound shields were investigated in more details: a plane sound shield, a tunnel-type shield and a closed enclosure with a small rectangular aperture on the side.

The measured insertion loss of the *rigid plate*, averaged for mesh#1 of Fig. 1a, is shown in Fig. 3a, together with its calculated counterpart. As one can expect, the performance of the shield is rather low, and even negative Insertion Losses are observed for certain frequency ranges. This effect is more pronounced for some simple low frequency structural modes of the source (not reported herein) and can be satisfactorily be explained by the increased radiation impedance of the source [6]. Generally speaking, the accuracy of the prediction is reasonable except for some sharp dips, e.g., at 910 and at 1020 Hz. These irregular frequencies are caused by the numerical non-uniqueness problem, inherent to the applied variational technique.

The effect of some *absorption* applied on the inner surface of the shield plate (a 20 mm thick rockwool plate) is shown in Fig. 3b. The IL is definitely increased, though there are frequencies where the shield still amplifies rather than reduces the power flowing through mesh#1, in spite of its absorbing surface.

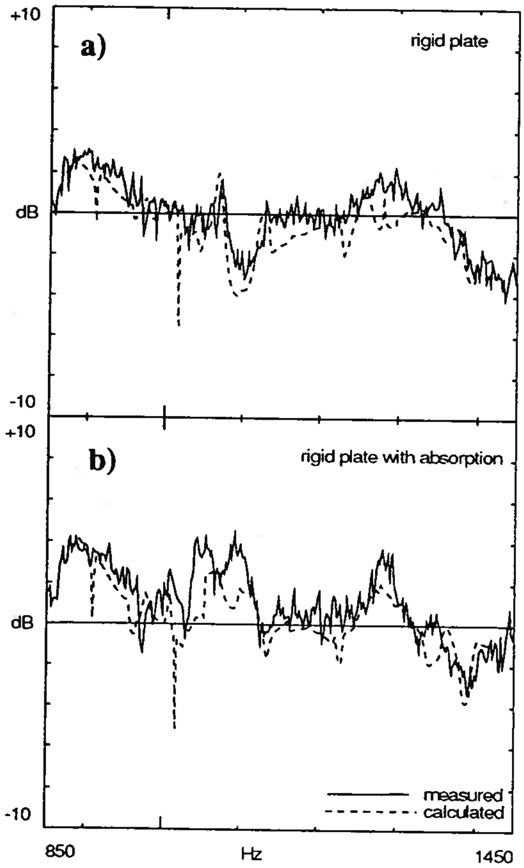

Fig.3. *IL spectra of various shield plates*

The numerical model of the *rigid tunnel shield* case is shown in Fig. 4. and the obtained IL spectra in Fig. 5a. Due to the lack of absorption inside the shield a number of constructive and destructive interferences take place, resulting in a rather frequency dependent IL curve. The detailed analysis of the modal behaviour of the source (a structural subsystem) and the gap between the source and the shield (an acoustical subsystem) has revealed that the performance of the shield is influenced both by its constituent subsystems and a strong interrelationship thereof [7]. The acoustic energy is clearly redirected by the shield, but the total amount of energy radiated from the source is strongly influenced by the presence of the shield, too. This means that the

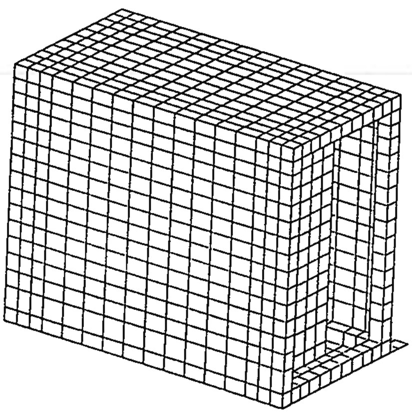

***Fig.4.** Numerical model of the tunnel shield case*

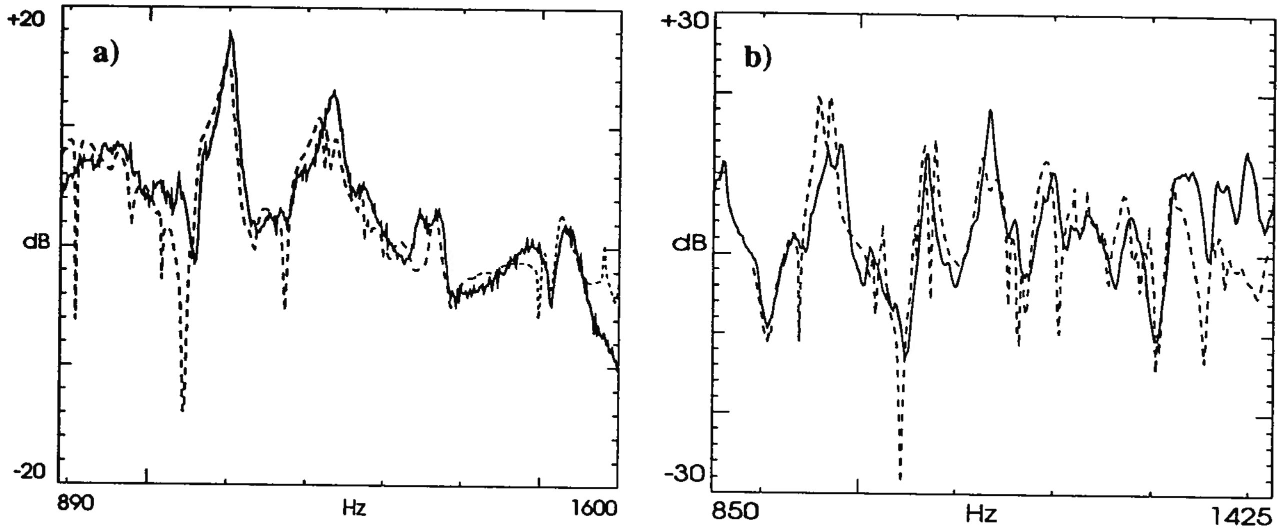

Fig. 5. Insertion Loss spectra of the ***a)*** *tunnel shield case;* ***b)*** *closed enclosure with aperture (Solid line: measurement, dashed line: BEM prediction)*

main operating mechanism of such a shield is changing the coupling between the source and its surrounding fluid, rather than changing the direction of the propagation or reducing the amount of radiated energy by dissipation. The BEM prediction of the tunnel shield is rather good, though the irregular frequencies are more pronounced as the complexity of the model increases.

Eventually, the obtained IL spectrum of a closed rigid enclosure with a *single aperture* of size 100*100 mm on the larger side of the enclosure is shown in Fig. 5b. The frequency dependence of the IL spectrum is even stronger, but the accuracy of the prediction is still reasonable.

Evaluation and conclusions

On the basis of the performed mock-up experiments the applied BEM method seems to be appropriate to predict the insertion loss of realistic engine enclosures. Two major parameters limit the practicability of the technique: the occurrence of irregular frequencies (thus hampering meaningful third-octave band calculations) and the necessary computing effort to run full-size engine calculations.

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