Radiation from partially open enclosures

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ABSTRACT
An approach is investigated to describe the radiation coupling between sources in an engine compartment and the air gap beneath. Emphasize is put on analytical treatment in order to gain insight into the salient physics. In this contribution, the results will be compared with those of experiments.

1 INTRODUCTION

Irrespective of vehicle type, the mechanical and combustion related sources inside the engine compartment contribute substantially to the pass-by noise. The description and modelling of the radiators, the coupling to the adjacent air gaps and the sound transmission via the cavity between the bottom of the vehicle and the road surface to the farfield has only received limited attention and is therefore inadequately understood.

Intuitively, the characterisation of the automotive engine compartment as a rectangular room in a building acoustics sense would lead to improper results. The engine compartment is a densely packed enclosure with tight walls and a highly reactive acoustical environment inside. Most likely, it contains several coupled rooms and the installations inside can be seen as diffracting elements.

Efficient analyses employing deterministic methods such as the Finite Element Method (FEM) are currently limited to the low wavenumber range. The elevated number of resonant modes at high wavenumbers generally precludes deterministic methods. In this range, statistical methods such as Statistical Energy Analysis (SEA) are often applicable. This gives a time-averaged statistics of the vibro-acoustic energy but doesn't reflect aspects such as strong correlation between the various source mechanisms of the engine noise radiation problem.

The objective of the following is to gain insight into the physics of sound radiation from motor vehicles. Focus is on the sound propagation from a source like the engine or an auxiliary unit to any point at the interface between the engine compartment and the gap between the vehicle and the road surface. Therefore, different analytical methods will be investigated with respect to their ability to calculate the point to point transfer impedances in a simple way.

In order to validate the theoretical approaches, the results will be compared with those of measurements in a generic model.

2 THEORETICAL INVESTIGATIONS: MODAL SYNTHESIS OF THE ENGINE COMPARTMENT

To simplify, the engine compartment can be seen as a box with some installations inside and complex wall impedances. For the first investigations the appliances inside and the soft walls were neglected and the transfer impedances was calculated in terms of a modal expansion of the field inside.

With all walls rigid, the mode shape of the rectangular enclosure is given by [1]

$$\Psi_N = \cos(\pi n_x x)\cos(\pi n_y y)\cos(\pi n_z z)\,.$$

The eigenfrequencies are

$$k_N^2 = \left(\frac{\pi n_x}{l_x}\right)^2 + \left(\frac{\pi n_y}{l_y}\right)^2 + \left(\frac{\pi n_z}{l_z}\right)^2; \qquad n \in N\,,$$

and with the acoustic impedance, as ratio of pressure to volume velocity, calculated from

$$\frac{p}{q} = jk\frac{\rho c}{V}\sum_N \frac{\Psi_N(S)\Psi_N(R)}{k_N^2 + j\gamma_N k_n - k^2}$$

Figure 1 shows the calculated impedance for those simplified boundary conditions.

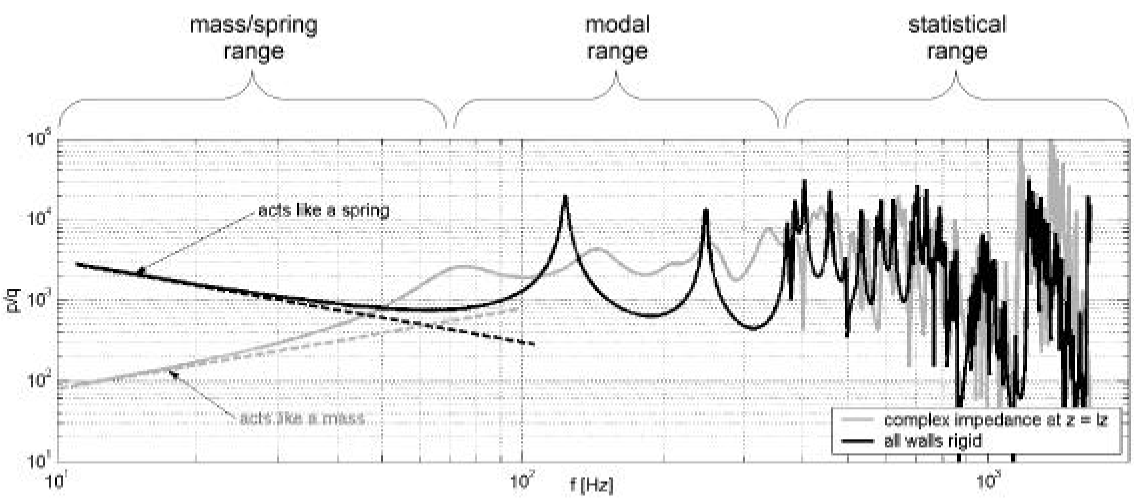

Figure 1: Ranges of transfer impedance for closed and open box, see figure 2 a) and b)

To introduce a general impedance, one wall of the box under consideration was removed. This simulates a more general case like the surface impedance at the interface between the compartment and the air gap underneath the car.

It is assumed that the impedance Z_a at the open side of the box ($z = l_z$) can be approximated by that seen by a baffled piston radiating into free space. In [2] the impedance for a circular piston is developed. To transform the results to the rectangular system under consideration two different routes appear possible, same area or perimeter. The variation due to the inhomogenities at the edges of the piston and due to the assumption of an infinite baffle at low frequencies are summarized in a frequency dependent correction term.

Using the complex impedance, the modal pattern can be calculated by means of an adapted modal function [3] with a complex component in z direction

$$\Psi_N = \cos(pn_x x)\cos(pn_y y)\cosh(pg_z z)\ .$$

The complex eigenfrequencies are given by

$$k_N^2 = \left(\frac{pn_x}{l_x}\right)^2 + \left(\frac{pn_y}{l_y}\right)^2 + \left(\frac{pg_z}{l_z}\right)^2;\ \ n \in N;\ \ g \in C\ ,$$

and the acoustic impedance is obtained as

$$\frac{p}{q} = jk\frac{rc}{V}\sum_N \frac{\Psi_N(S)\Psi_N(R)}{k^2 - k_N^2}\ ,$$

yielding the complex eigenvalue problem

$$\tanh(pg_N) = j\left(\frac{2hl_z}{lg_N}\right).$$

Therein, the impedance is introduce as

$$h = \frac{rc}{Z_a}\ .$$

Figure 1 demonstrates the transfer impedance for the analytic model with complex impedance Z_a .

For the two cases calculated above, three regions can be distinguished. In the first, the system behaves like a spring for the closed box and a mass for the open box. The mass is that seen as the radiation impedance of the piston. The second range termed the modal range comprises the first few distinct eigenfrequencies. At high frequencies, the field is increasingly diffuse in nature and the methods of statistical acoustics apply.
The trough at about 1 kHz results from reflection interferences at source and receiver position.

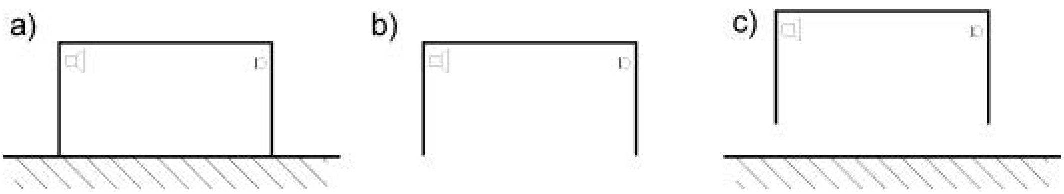

Figure 2: Different configurations of the lab rig

3 EXPERIMENTS: MEASUREMENTS OF TRANSFER IMPEDANCES

Shown in Figure 2 are the different configurations of the experimental set-up. In these set-ups any transfer impedances from a loudspeaker driven volume velocity source to a receiving microphone can be measured. As can be seen in Figure 3 and 4, the experimental results capture the theoretical trends for the rigid boundary condition as well as for the case with one side open.

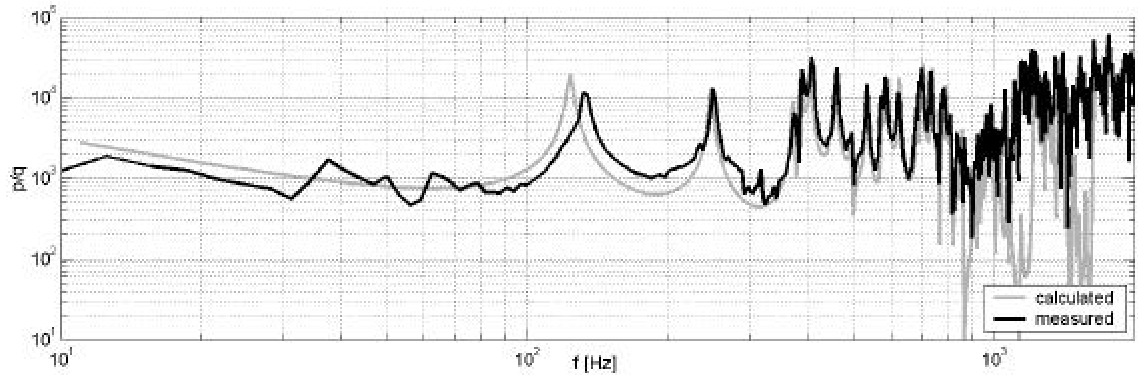

Figure 3: Measured and calculated impedance when all walls rigid

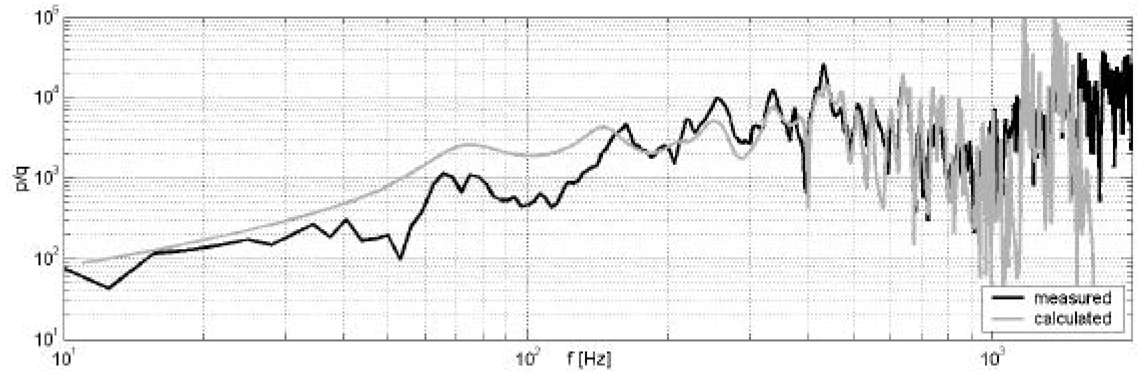

Figure 4: Measured and calculated transfer impedance when one wall removed (complex wall impedance)

In the subsequent experiments, the significance of the subsystem properties and geometrical dimensions of the system under consideration were investigated. Three main topics have been the objectives of experimental investigations:

1. Difference between a sufficiently wide gap (like the gap between car and road surface) and free field as seen in Figure 2b) and c),
2. influence of partitioning the engine compartment and
3. arrangements of the installations inside the enclosure.

The results of the impedance measurements with different air gaps are shown in Figure 5. It is observed that there is no significant difference between a case with a sufficiently wide gap like the gap between car and road surface and free field.

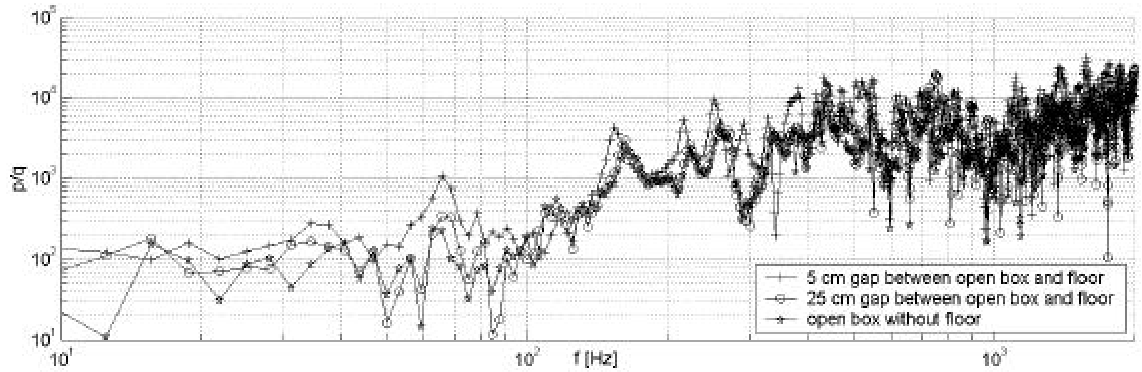

Figure 5: Measured transfer impedance with different air gap, see figure 2 b) and c)

In reality, the engine compartment is very densely packed and contains many subcavities. To simulate the subcavities, partitioning walls are introduced in the box (Figure 6). From the measured transfer impedance it is seen in Figure 7 that such hindrances significantly alter the first few eigenfrequencies. This is most likely because of the longer way from one side wall to the other such that there is a downwards frequency shift of the eigenfrequencies.

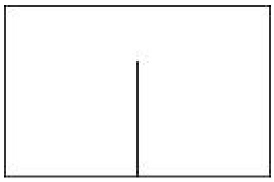

Figure 6: Enclosure subdivided with partition wall

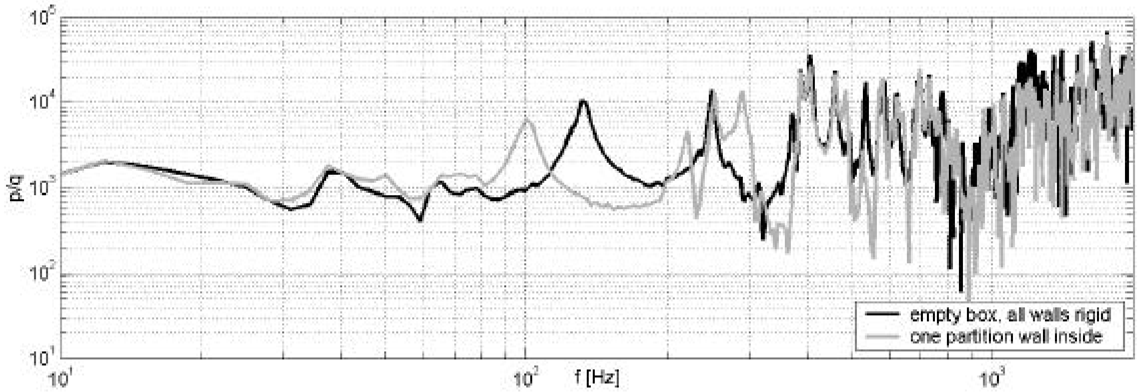

Figure 7: Measured transfer impedance without/with partition wall

Mostly the installations inside the engine compartment of different vehicles are the same but with different layout and with different dimensions. For this reason, transfer impedances of different cars with source points inside the engine compartment and receiver points at the interface to the underneath air gap was measured. In Figure 8 a selection of measurement results is shown. Although the cars considered are different in dimensions and had different levels of packing density inside, there is no significant difference in transfer impedance of nearly the same source and receiver points.

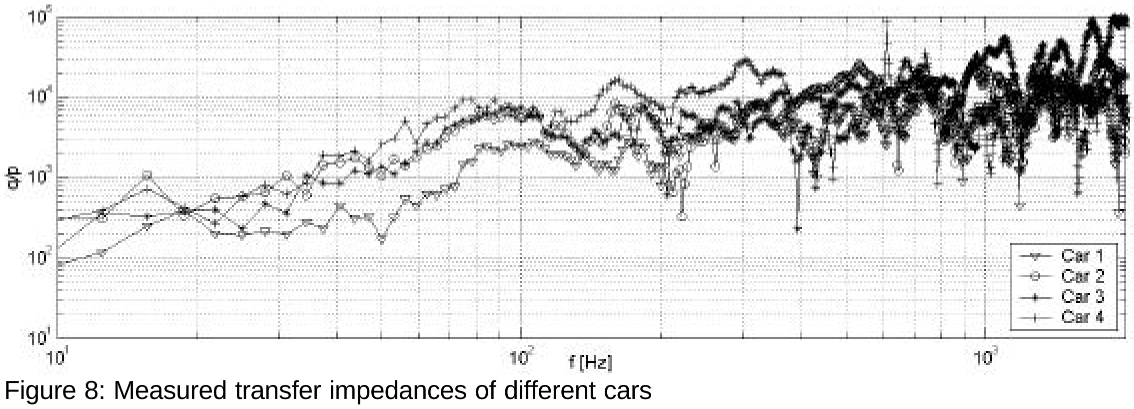

Figure 8: Measured transfer impedances of different cars

5 CONCLUDING REMARKS

From the work presented above it is demonstrated that the sound propagation form a source inside the engine compartment to a point at the interface to the air gap between vehicle and road surface doesn't significantly depend on dimensions and layout of the inside installations. Although the engine compartment is a very complex system of coupled volumes, always three basic regions of the transfer impedance are recognizable. Only at an intermediate frequency range where the first few eigenfrequencies appear, an evident influence of structural differences can be observed. At low frequencies the cavity inside the compartment behaves mass-like in analogy with a perforated absorber. At high frequencies, the methods of statistical acoustics apply.

These results provides an opportunity to determine the sound propagation to the interface. Given the pressure field at the interface, the radiation from the engine compartment into the far field can be calculated using e.g. the method proposed in [4].

5 ACKNOWLEDGEMENT
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