The sound reduction index applied to automotive problems

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Abstract
In recent years comfort has become an important factor when evaluating the performance of modern automobiles. One important aspect that has negative ramifications on the perception of the quality is the generated noise. Therefore, an important goal in current research activities is to minimize the acoustic noise that is radiated by a combustion engine. It is possible to improve the noise vibration harshness (NVH) behavior by using active or passive methods. One promising passive method is an encapsulation. The paper presents simulation approaches of a thermo-acoustic encapsulation of an engine block, which completely wraps the engine block. In general, thermo-acoustic encapsulations are able to reduce the radiated sound significantly. Furthermore, they also increase the heat storage capacity and thus the efficiency of the engine. As a consequence the exhaust emission is reduced by avoiding so-called cold starts. To evaluate the influence of a thermo-acoustic encapsulated engine regarding the radiated noise appropriate finite element models are commonly employed. Additionally, it is necessary to consider the surrounding air and the air between the engine and the encapsulation. That is to say, a coupled structural-acoustic simulation has to be applied to obtain sufficiently accurate results. One has to take into consideration that fully coupled numerical simulation approaches are very time-consuming and computationally expensive. In early stages of the design process it is desirable to minimize the computational costs. The paper at hand presents first results in that regard. The numerical effort is significantly reduced by applying prediction based approaches known from civil engineering, such as the sound reduction index. The solutions of the prediction based approach are compared and evaluated with results obtained with a full scale finite element model of the encapsulated engine.

Sound reduction index
A detailed overview of different methods to predict the sound reduction index is given in [1]. The formula to compute the sound reduction index \( R \) is given in equation (1)

\[
R = 10 \log \frac{1}{\tau}.
\]  

(1)

The transmission factor \( \tau \) is the ratio of the transmitted \( P_{tr} \) and the incident \( P_i \) sound power of the structure. The transmission factor \( \tau \) is also the ratio of the transmitted \( I_{tr} \) and the incident \( I_i \) sound intensity, or the square of the ratio of the magnitudes of the transmitted \( \hat{p}_{tr} \) and incident \( \hat{p}_i \) sound pressure

\[
\tau = \frac{P_{tr}}{P_i} = \frac{I_{tr}}{I_i} = \left| \frac{\hat{p}_{tr}}{\hat{p}_i} \right|^2.
\]  

(2)

The last part of equation (2) is only applicable, if the fluid in front and behind the wall is the same. This becomes clear, if equation (4) is taken into account. Equation (3) shows the definition of the sound intensity \( I \)

\[
I = \frac{|\hat{p}|^2}{Z \cdot \cos (\varphi)}.
\]  

(3)
In equation (3) \( \varphi \) denotes the angle between the normal of the wall surface and the direction of the propagation of the sound wave.

\[
Z = \rho_{\text{fluid}} \cdot c_{\text{fluid}}
\]  

(4)

The characteristic impedance \( Z \) is the product of the density \( \rho_{\text{fluid}} \) and sound velocity \( c_{\text{fluid}} \) of the fluid, in which the sound waves propagates.

The difference between the measured pressure levels \( L_p \) in front and behind the specimen is used to compute the sound reduction index using experimental data. In general such measurements are executed in a Kundt’s tube, which results in

\[
R = L_{p,i} - L_{p,\text{tr}} = 10 \log \left( \frac{\hat{p}_i}{\hat{p}_{\text{tr}}} \right)^2.
\]  

(5)

A simple possibility to predict the sound reduction index of a plate-like structure for skew sound incidence is the so called Berger’s mass law [2] (see the dash-dotted curve in Figure 1), which results in

\[
R = 10 \log \left[ 1 + \left( \frac{\omega m'' \cos(\varphi)}{2Z} \right)^2 \right].
\]  

(6)

The sound reduction index is frequency dependent. In equation (6) \( \omega \) denotes the circular frequency with \( \omega = 2\pi f \). The area-related mass \( m'' \) is the product of the density \( \rho \) and the thickness \( h \) of the structure as

\[
m'' = \rho \cdot h.
\]  

(7)

Equation (6) is commonly used for thick and heavy walls in civil engineering. According to reference [3] it is possible to consider the influence of the stiffness \( K \) for thin structures, which results in the following sound reduction index

\[
R = 10 \log \left[ 1 + \left( \omega m'' - K \omega^3 \sin^4(\varphi) c_{\text{fluid}}^4 \right)^2 \frac{\cos^2(\varphi)}{4Z^2} \right].
\]  

(8)

The sound reduction index in equation (8) has a simple root, which is shown as the dotted curve in Figure 1. The corresponding frequency \( f_{\text{coincidence}} \) is the so called coincidence frequency [4] given as

\[
f_{\text{coincidence}} = \frac{1}{2\pi} \sqrt{\frac{m''}{K c_{\text{fluid}}^2} \frac{1}{\sin^2(\varphi)}}.
\]  

(9)

Here \( K \) is the plate stiffness as

\[
K = \frac{Eh^3}{12(1-\nu^2)}.
\]  

(10)

where \( E \) denotes the elasticity/Young’s modulus, \( \nu \) the Poisson ratio and \( h \) the thickness of the plate.

\[
R = 10 \log \left[ \left( 1 + \eta \cdot K \omega^3 \sin^4(\varphi) \cos(\varphi) c_{\text{fluid}}^{-4} \right)^2 + \left( \omega m'' - K \omega^3 \sin^4(\varphi) c_{\text{fluid}}^{-4} \right)^2 \frac{\cos^2(\varphi)}{4Z^2} \right].
\]  

(11)
The equation (11) is a more general formulation of equation (8), in which the damping factor \( \eta \) [5] is also taken into account. This damping factor causes decreasing flattening of the sound reduction index curve near the coincidence frequency (see the dashed curve in Figure 1).

The assumptions of equation (11) are the following: the incident waves are plane waves, the transmitted waves propagate in the same direction like the incident ones, structure-borne noise paths are not considered, the front and the back of the structure perform the same motion and the characteristic impedances are identical on both sides of the structure, that means the same fluid can be found on both sides.

\[
\omega_{mn,plate} = \pi^2 \sqrt{\frac{K}{\rho h} \left( \frac{m}{l_x} \right)^2 + \left( \frac{n}{l_y} \right)^2} \tag{12}
\]

For automotive problems it is important to involve the structural eigenfrequencies. Equation (12) shows the resonant frequencies \( \omega_{mn,plate} \) of a rectangular plate with the dimensions \( l_x \) and \( l_y \) and the number of periods \( m \) and \( n \) in x- and y-direction, respectively.

\[
\omega_{mn,room} = \pi c_{\text{fluid}} \sqrt{\left( \frac{m}{l_x} \right)^2 + \left( \frac{n}{l_y} \right)^2 + \left( \frac{l}{l_z} \right)^2} \tag{13}
\]

Furthermore, eigenfrequencies of the air volume arise in a closed cavity like closed rooms and also exert a certain influence on the sound reduction index. Under free-field conditions these air eigenfrequencies \( \omega_{mn,room} \) can be neglected. Equation (13) shows the resonant frequencies \( f_{room} \) of a rectangular room with the dimensions \( l_x, l_y \) and \( l_z \) and the number of periods \( m, n \) and \( l \) in x-, y-, and z-direction, respectively.

In reference [6] a method to consider natural frequencies in the sound reduction index or the transmission factor is proposed. The principle is analogous and quiet similar to the way of handling the attenuation by the coincidence frequency (see Figure 1).

**Figure 1:** Comparison of different possibilities to predict the sound reduction index

Figure 1 shows the sound reduction index computed with the different equations (6), (8), (11) for an arbitrary material. The consideration of the stiffness in equation (8) causes the sharp decline near the coincidence frequency, cf. equation (9). By introducing additional damping in equation (11) the flattening of the curve near the coincidence frequency can be explained. As commonly known the damping has an influence near the resonances or singular events only.
Thermo-acoustical sandwich material

For the encapsulation of the engine a special absorbing sandwich material is used as depicted in Figure 2. Such a material system basically consists of a very soft and highly absorbing foam layer, directed to face the vibrating structure and additionally a much stiffer fiber material on the outside. Both materials are very light and temperature resistant.

Figure 2: High acoustically absorbing sandwich material of the thermo-acoustical encapsulation

Numerically calculated sound reduction index of the sandwich material

In an initial numerical study using the finite element method (FEM) a single specimen is studied. Figure 3 illustrates the applied model. A rectangular specimen of the thermo-acoustical sandwich material is used as a separating wall between two rectangular air filled rooms. One room acts as a source or sending room and includes the sound source and the other room acts as a receiving room.

Figure 3: FE-Model to calculate the sound reduction index of the sandwich material

The basic model in Figure 3 is discretized with quadratic hexahedral elements and used for different investigations. The displacements of the surfaces of the sandwich, which are not in contact with the air volumes are fixed to zero. Thereby different sources, source locations, absorbing boundary conditions (ABCs) and coupled as well as uncoupled simulations have been executed. The results are shown in Figure 4-7. These results are calculated using equation (5). The mean values of the sound pressure level in the source room and the receiving room are used. The incidence angle is 45° to represent a diffuse sound field.

In our case uncoupled means that the effects from the sandwich material to the source room and the effects from the receiving room to the sandwich and source room are not considered.
The sound reduction index computed by equation (11) is drawn as dash-dotted line in Figure 4. At first it is visible, that in general the sound reduction index according to equation (11) underestimates the FE-results in Figure 4 and does not consider the resonances of the rooms or the sandwich structure.

The comparison of the results of the simulation with no ABCs (dotted line in Figure 5), ABCs only at the walls of the air filled room, which are parallel to the biggest surface of the sandwich (dash-dotted line in Figure 5) and ABCs at all surfaces of the air filled rooms (dashed line in Figure 5) shows, that the ABCs prevent room resonances. So the resulting sound reduction index curve is smoother with less and weaker drops in the path of the curve, if ABCs are defined on the outer surfaces of the air volume.
**Figure 6:** Comparison of the sound reduction index of the thermo-acoustical sandwich material, which is calculated by uncoupled and coupled FE-simulations without ABCs.

The result of the uncoupled simulation without ABCs (solid line in Figure 6) compared to the coupled simulation without ABCs (dotted line in Figure 6) shows, that the mutual influence of the air volumes and the sandwich is not negligible. In the coupled case the formation of resonances of a single part is inhibited.

**Figure 7:** Comparison of the sound reduction index of the thermo-acoustical sandwich material, which is calculated by uncoupled and coupled FE-simulations with ABCs at all surfaces of the air volumes and different sound source locations.

Figure 7 shows, that different sound source locations influence the sound reduction index in higher frequency regions only. The same is to be observed for the difference between the coupled and uncoupled simulation with ABCs at all outer surfaces of the air volumes. Below 500 Hz there is almost no influence on the sound reduction index visible in Figure 7.
Sound reduction index of the thermo-acoustical encapsulation

Finite element model of the engine, encapsulation and air
The examined structure consists of the cylinder crankcase, oil pan and oil pan cover, cf. Figure 8. All of them are made of aluminum. By using the FEM these components are discretized. First, the structural vibrations are calculated and then the resulting sound radiation in the far field is analyzed. The surrounding air volume is modeled as a sphere with a coarser discretization to the periphery. Due to the computational costs, the acoustic simulation is carried out exclusively in the frequency domain. The Sommerfeld radiation condition is fulfilled by using special ABCs. This application of ABCs requires the least computational time without a significant loss of accuracy. The acoustic simulation can be performed uncoupled, which means that the influence of the fluid on the vibrating structure is neglected. The engine structure is quite stiff in comparison to the air and is, consequently, not notably influenced by the surrounding air pressure under free field conditions. The fluid-structure-coupling is implemented using special interface elements. These are embodied as shell elements with the contour of the surface structure and transform the surface velocities of the structure to the corresponding pressure values. At the interface the structural mesh and the fluid mesh are conform meaning that their nodes are coincident. In general quadratic tetrahedral FEs are used to discretize the engine, encapsulation and air.

The structural vibrations are excited with the help of bearing reactions, which are introduced into the bearing blocks of the cylinder crankcase. The bearing reactions are obtained from an elastic multi-body simulation of the crank drive dynamics. This requires an experimentally determined cylinder gas pressure curve as input data. Using the method explained in reference [7] cylinder deformation effects resulting from the combustion process and arising forces can also be considered.

Figure 8: FE-model of the engine (left), the encapsulation (middle) and the fluid volume (right)

The influence of the encapsulation on the engine block is neglected with respect to computational costs, because the encapsulation is much softer than the engine block. Therefore the previously calculated structural vibrations are used in analogy to the previously discussed fluid model. Structural displacements and velocities are applied to the internal surface nodes of the encapsulation as boundary conditions. The structural vibrations of the encapsulated surface are used to excite the surrounding fluid.

Numerical results
In Figure 9 the sound pressure distribution with and without the encapsulation is visualized in a cut through the center plane of the spherical air volume. Figure 9 shows a considerable
sound pressure reduction for the case where the encapsulation is surrounding the engine. Furthermore, it can be observed that even if the encapsulation is present, the dominant radiator is still the oil pan.

Figure 9: Simulated sound pressure level of the engine with (right) and without (left) encapsulation in a fixed operating point (2500 min⁻¹, 100 Nm)

The Figure 10 shows as dotted line the sound reduction index, which is extracted from the simulation result depicted in Figure 9. The mean values of the sound pressure level at the outer surface of the spherical air volume for both cases, i.e. with and without encapsulation, are inserted into equation (5). The dash-dotted line in Figure 10 is the sound reduction index computed by equation (11).

Figure 10: Simulated sound reduction index of the encapsulated engine of Figure 8

In contrast to Figure 4 the results of equation (11) in Figure 10 illustrate an overestimation of the sound reduction index of the encapsulation compared to the result of the FE-simulation. There are several reasons for this: equation (11) is independent of any dimensions and this formula was developed for much larger, heavier and stiffer structures as typical in civil engineering applications. The example of the encapsulation in this paper contains a further
problem: equation (11) assumes that the incident waves are plane waves and only air borne noise paths are important for the sound transmission. But the structure born noise paths are very important for this example, because of the many contact surfaces between engine and encapsulation. The final problem is, that real-life applications generally do not fulfill the assumptions of the sound reduction index equations, which were presented in the paper.

Comparison with Measurements
The experimental data of a previous study [8] is used to evaluate the sound reduction index depicted in Figure 10, which is a result of a full scale FE-simulation. The experimental results published in reference [8] are shown in Figure 11, expressed in increments of one-third octave band. Figure 11 shows the measured sound pressure level at a fixed operating point in a distance of 1 m from different sides of the engine. In the simulation the same operating point is used. It can be clearly seen, that the encapsulation has no positive effect below 400 or 500 Hz. At specific locations the encapsulation even increases the sound pressure level in this frequency domain, for example at the top side. In general a reduction of the sound pressure level from 15 to 20 dB is visible in Figure 11.

![Comparison with Measurements](image)

Figure 11: Measured sound pressure level of the engine with and without encapsulation in a fixed operating point (2500 min⁻¹, 100 Nm) [8]

The sound reduction index calculated using FE-results and illustrated in Figure 10 shows comparable behavior. In the lower frequency ranges the encapsulation has either no effect at all or is even counterproductive. Above 500 Hz a smoothed increase of the sound reduction index up to a relatively constant value of 14-18 dB is observed in Figure 10.

Conclusions
The paper at hand presents first results of an investigation to apply the sound reduction index to evaluate the acoustic behavior. The solutions of the prediction based approach are compared and evaluated with results obtained using a full scale FE-model of the encapsulated engine and experimental data from a previous study of the encapsulation. The aim of applying
the sound reduction index is the reduction of the numerical effort by applying prediction based approaches. The sound reduction index known from civil engineering causes some serious problems. The most important problem of the sound reduction index is, that the dependence on the dimensions of the structure and the structure born noise paths are not considered. These points have to be improved in future investigations. Furthermore, one should take the resonances of the plate and the bounding air cavities into account for the sound reduction index. The prediction of the sound pressure level reduction using FE-simulations shows an acceptable agreement compared to experimental measurements.

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References