Evaluation of Sound Transmission Models for Automotive Applications

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Introduction

Acoustics is a distinctive feature for premium class vehicles. However, some of the current trends in the automotive industry, such as weight reduction and engine down-sizing, may have a negative impact on the acoustic performance. Moreover, the introduction of electric and autonomous cars exposes the users to new perceivable noise sources. In order to bring a competitive product to the market, with a satisfying balance of the comfort requirements and the weight and costs limitations, it is necessary to complement measurements with simulative methods. The final objective of an ongoing investigation is the development of a model that predicts the airborne sound propagation phenomena from the engine, through the firewall and the dashboard, to the passenger cabin.

One key factor to achieve reliable results lies on accurate modeling of the several noise control treatments (NCT). A large number of these acoustic treatments is made of poroelastic components, whose acoustical effectiveness is directly related to the amount of energy that they are able to dissipate. Energy losses within a poroelastic material are mainly driven by viscous friction of the fluid inside the pores, structural damping of the solid frame, and thermal exchange between the two phases. A detailed microscopical description of all these phenomena, even if possible, is not applicable for a real size problem, mainly due to its excessive cost. Instead, it is preferable to represent the material as a homogenized compound with a highly complex dissipative behavior.

In this article, some of the most relevant homogenization techniques are reviewed, including different formulations for the finite element method (FEM) and the statistical energy analysis (SEA). After a short recapitulation of the main theoretical principles, the three simplified setups for the investigations are described. Next follows a discussion of the results and the performance of the methods. Lastly, conclusions on their application spectrum and an outlook to ongoing work are presented.

Theoretical background

This section summarizes the analyzed approaches for the representation of poroelastic components.

Finite Element Method

Two main approaches for modeling poroelastic media are found within the scope of the finite element method [1]:

- Full poroelastic description, which solves Biot's poroelasticity equations for both solid and fluid phases of the material.
- Equivalent fluid representation, where the visco-inertial coupling between the phases is neglected and only the fluid inside the pores is calculated.

Equivalent fluid models can be divided in rigid frame (if the stiffness of the skeleton is comparatively large) and limp frame (if the inertial coupling is introduced by modifying the density of the equivalent fluid). In the commercial FE software *LMS Virtual.Lab* different formulations for porous and poroelastic media are available. The Johnson-Champoux-Allard (JCA) model [2], [3] is a semi-phenomenological model and can be used in combination with an elastic frame (for the full poroelastic computation), and with limp or rigid frames. The Delany-Bazley-Miki (DBM) model [4] is based on empirical data and works with limp and rigid frame, and the Craggs model [5] was derived for rigid skeleton materials.

The main drawback associated with FE is its elevated computational effort: each node has 4 degrees of freedom for the full poroelastic model, and the number of required elements increases with the frequency.

Statistical Energy Analysis

To deal with this inherent problem of the FEM, energetic approaches as the SEA [6] may be used. This method predicts average sound and vibration levels. Analogue to the thermal exchange problem, the power balance between subsystems is proportional to the difference in their modal energies and coupling loss factors, whereas the dissipated energy in a subsystem is proportional to its damping loss factor.

In the SEA frame, a noise control treatment is represented by its impedance, calculated from the combination of explicit mathematical models for the behavior of each layer. The introduction of an NCT modifies the coupling and damping loss factor of the subsystem to which it is attached.

Another possibility is the employment of a hybrid approach, where SEA and FEM components are combined. In the following examples, a hybrid model has been built by substituting the plate component in an energetic model by its finite element counterpart.

In the following examples, the models compared are:

- Finite Element: one full poroelastic (*JCA elastic*), one limp frame (*JCA limp*), and three rigid skeleton models (*JCA rigid*, *DBM rigid*, *Craggs rigid*).
- one Statistical Energy Analysis setup (*SEA*).
- the above-mentioned hybrid model (*Hybrid*).

Test setup

In order to assess the efficiency of the chosen methods, a multi-layered system with size $1m\times\ 1m$ has been selected, composed of a $0.8mm$ steel plate, an $11mm$ foam layer and a $4kg/m^2$ heavy layer. Standard values are used for the steel properties, whereas the properties of the other components come from experimental data.

The system has been analyzed in 3 different states, namely the foam layer alone, the foam attached to the steel plate and, lastly, a spring-mass system built by the steel plate, the foam and the heavy layer. This last one could be seen as a simplified representation of a real system used in automotive applications (e.g. firewall with spring-mass). All setups are placed between two semi-infinite fluids and excited by a diffuse acoustic field. The transmission loss (TL, eq. 1) of the system in every case is calculated for frequencies between 80Hz and 2000Hz, a range in which engine radiation through the firewall has its largest contribution.

$$\mathrm{TL} = 10\log_{10}\ 1/\tau = 10\log_{10}\ \Pi_{\mathrm{in}}/\Pi_{\mathrm{out}} \qquad (1)$$

Results

Foam layer

Figure 1 shows the FE results for the transmission loss of the foam layer. The computation where the foam layer is modeled as a full poroelastic component serves as reference, since this formulation is the only one that solves both phases in the medium, and is designated in the graphs by *JCA elastic*. The other results belong to equivalent fluid models.

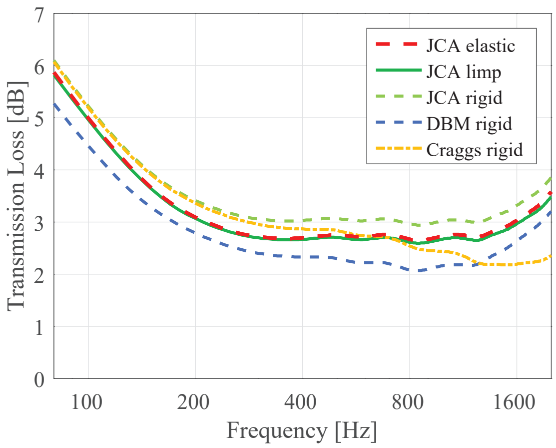

Figure 1: Transmission loss for a foam layer under diffuse field excitation.

One interesting feature when solving the FE equations is the possibility to compute the dissipated power inside the poroelastic material and to separate it into the different dissipation mechanisms taking place, namely viscous effects in the fluid (*Fluid*), structural damping (*Struct*), thermal losses (*Thermal*), viscous effects in the solid (*Solid*), and viscous effects due to the fluid-structure coupling (*F-S*). For this application, most of the sound energy is dissipated due to viscous losses in the fluid phase (see Figure 2), whereas the percentage due to thermal losses increases with the frequency. These two dissipation mechanisms are mainly dependent on the fluid phase, which explains the good match between the elastic and limp models. The rigid frame formulations show a small offset, but follow the right tendencies. The *Craggs* formulation does not capture the thermal losses, therefore the results diverge at higher frequencies.

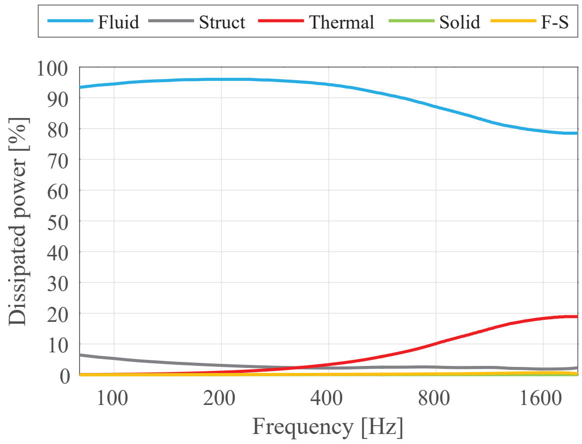

Figure 2: Percentage of dissipated power by each mechanism inside the foam layer bonded on to a plate.

The model analyzed has a 31% of nodes corresponding to the foam layer, while the rest belong to the sender and receiving rooms. Since the equivalent fluid formulations only solve the pressure degree of freedom (in contrast to the four degrees of freedom of the full poroelastic), the computation time is reduced by a factor of 3.

Plate with foam layer

Next, the foam layer is attached to a steel plate, on which the diffuse sound field is applied. Figure 3 displays the FE transmission loss results for this configuration. For reference, the results for the bare steel plate are added (*Plate*). Comparing firstly the solutions for the plate and the full poroelastic formulation, one can observe that, at the low frequency range, the introduction of the foam layer produces a shift of the resonances towards lower frequencies and an increase of the TL values. This effect is equivalent to the foam being replaced by an increase of the plate density corresponding to the additional mass provided by the foam.

Under these conditions, the frame of the foam is directly excited and most of the power is dissipated by structural damping in the frame (see Figure 4). Limp and rigid models cannot reproduce this impact because the losses in the frame are neglected, therefore their use is not appropriate. Here, the contribution of the dissipation in the fluid phase is comparatively small, mainly because the chosen material has a large structural damping coefficient, $\eta_s \approx 0.3$. When this factor is reduced, the energy dissipation distributes more uniformly between the structural damping and the viscous losses in the fluid phase: another test showed that for $\eta_s \approx 0.1$ the percentage dissipated by the structural damping reduces up to 60%.

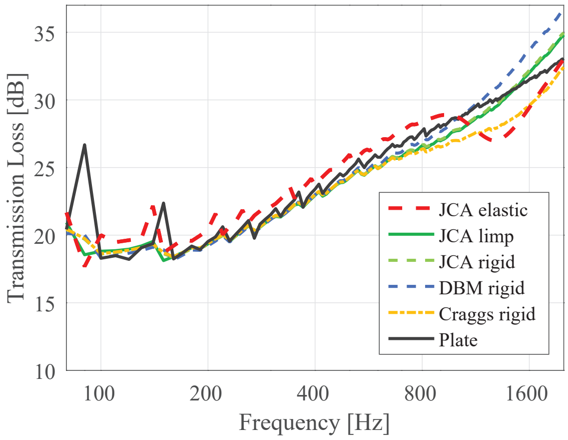

Figure 3: Transmission loss for a foam bonded on to a plate under diffuse field excitation.

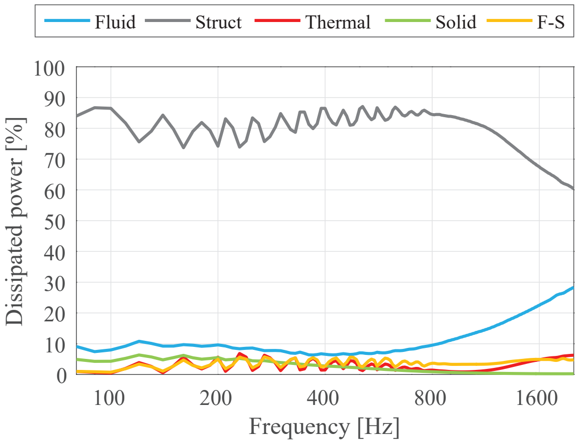

Figure 4: Percentage of dissipated power by each mechanism inside the foam layer bonded on to a plate.

Plate with spring-mass system

Now, the spring-mass system attached to the steel plate is discussed. Finite element rigid and limp frame formulations are compared to the full poroelastic solution in the Figure 5. The equivalent fluid models show a shift of the resonance frequency of the spring-mass system towards lower values. This effect is due to the fact that the presence of the solid frame increases the dynamic stiffness of the material, but this cannot be reproduced in the equivalent fluid formulations. The resonance frequency for the *JCA* and *DBM* equivalent fluids correlates with the analytical value for an air layer placed between two plates with the surface densities of the plate and the heavy layer.

The spring-mass resonance peak is less pronounced for the full poroelastic model, which is caused by the high damping coefficient of the studied foam, but its effect can only be captured with *JCA elastic*. Another proof of the large influence of the frame is found in the distribution of the dissipation mechanisms in Figure 6, where again the losses due to structural damping are dominating. If, for example, the damping factor of the frame is reduced by half, the resonance peak in the TL drops by $4dB$, and the percentage of viscous dissipation in the fluid doubles over the whole frequency range.

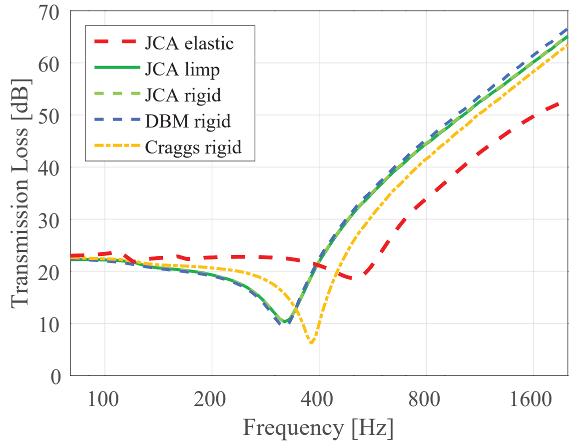

Figure 5: Transmission loss for a spring-mass system bonded on to a plate under diffuse field excitation.

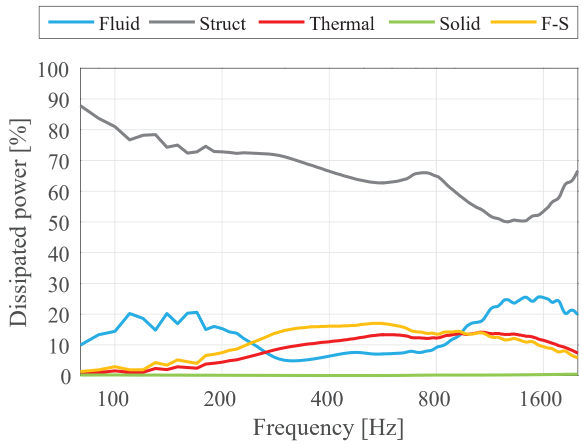

Figure 6: Percentage of dissipated power by each mechanism inside the foam layer bonded on to a plate.

For this system, SEA and hybrid results have also been calculated using *ESI VA One*. In this case, the results have been averaged in third octaves to ensure that a minimum modal density in the band is present, and they are displayed in the next section together with the FE computations for the *JCA elastic* formulation (Figure 7). The comparison with the energetic and hybrid models proves that the asymptotic behavior at the higher frequencies is the same for different formulations. On the lower frequency range some discrepancies are observed, since the modal behavior of the plate is neglected by the intrinsic averaging process of the SEA. The solving step of the SEA simulation takes only a few seconds, whereas the hybrid computation is approximately 15 times faster than the Finite Element analysis.

Comparison with measurement results

As a next step, the accuracy of the proposed modeling approaches is tested by confronting the simulation results with some experimental data. To this effect, the spring-mass system used in the numerical investigations was placed in a window test bench. In an attempt to reduce the influence of the experimental mounting conditions, the insertion loss (equation 2) is calculated in this case.

$$\mathrm{IL} = \mathrm{TL}_{\mathrm{NCT}} - \mathrm{TL}_{\mathrm{plate}} = \Pi_{\mathrm{out,plate}} - \Pi_{\mathrm{out,NCT}} \qquad (2)$$

The results obtained with *JCA elastic*, the *SEA* model, the *Hybrid* approach and the *Measured* results are plotted in Figure 7. As mentioned before, there is good match among the simulative results. However, the correspondence with the measured values is not yet satisfactory and further investigations are being carried out.

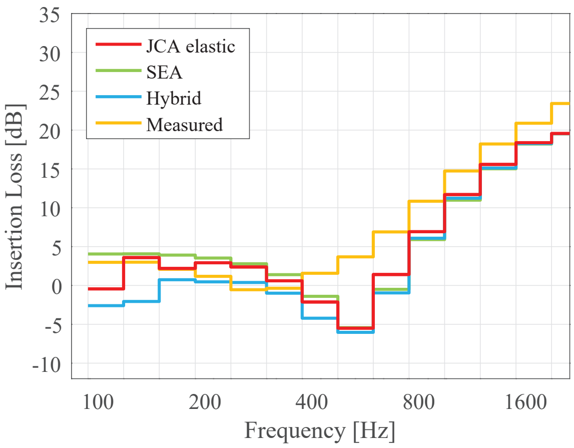

Figure 7: Insertion loss for a spring-mass system bonded on to a plate. Comparison with experimental data.

Conclusions and outlook

Though poroelastic materials are widely applied as noise control treatments, there exists no definite approach for their modeling that has a complete range of validity.

Equivalent fluid models imply a significant reduction of the required computation time due to two factors: first, only the interstitial pressure needs to be solved, such reducing the number of degrees of freedom from four to one; second, the equivalent fluid wavelength is always larger than the shortest wavelength in the poroelastic media, therefore it is possible to use larger elements in the FE model. However, the suitability of the equivalent fluid approach is limited in cases where the porous material is directly coupled to an excited structure, because it directly induces vibrations in the skeleton and, therefore, the assumption of a motionless frame would neglect the dissipation due to the structural damping. As seen in the proposed examples, the power loss in the frame can be the main dissipation mechanism for some materials. Therefore, the application of equivalent fluid formulations is recommended only for absorbers, that is, components directly excited by an airborne wave.

Approaches based on statistical methods, such as the statistical energy analysis, are advantageous in the higher frequency range, as the size of the model is independent of the frequency. Moreover, the modeling process is much faster because no geometrical details are considered (only a minimum size of the subsystems is needed, so that their modal density is high enough for the statistical method). However, since they are based on frequency and component averages, their applicability is limited if local results are necessary, or in the lower frequency range, where only few eigenmodes are dominating the system's response.

A good compromise may be found in the use of hybrid approaches, either by combining discretized and energy-based components (as shown in the previous examples), or by dividing the frequency range in two parts and selecting the most suitable method for each case [6].

Further analysis, including studies on the influence of the material parameters, the mounting and coupling conditions, and excitations with other sources on both the numerical and the experimental setups, will be conducted.

Although not within the scope of this article, it should be mentioned that another difficulty regarding the proposed methods arises from the identification of the material parameters. The characterization of most poroelastic properties requires special measurement methods, and the boundary conditions under which the tests are carried out may have a big influence on the results. For this reason, other approaches to describe these materials could also be of interest, for instance experimentally determined transfer matrices [8].

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