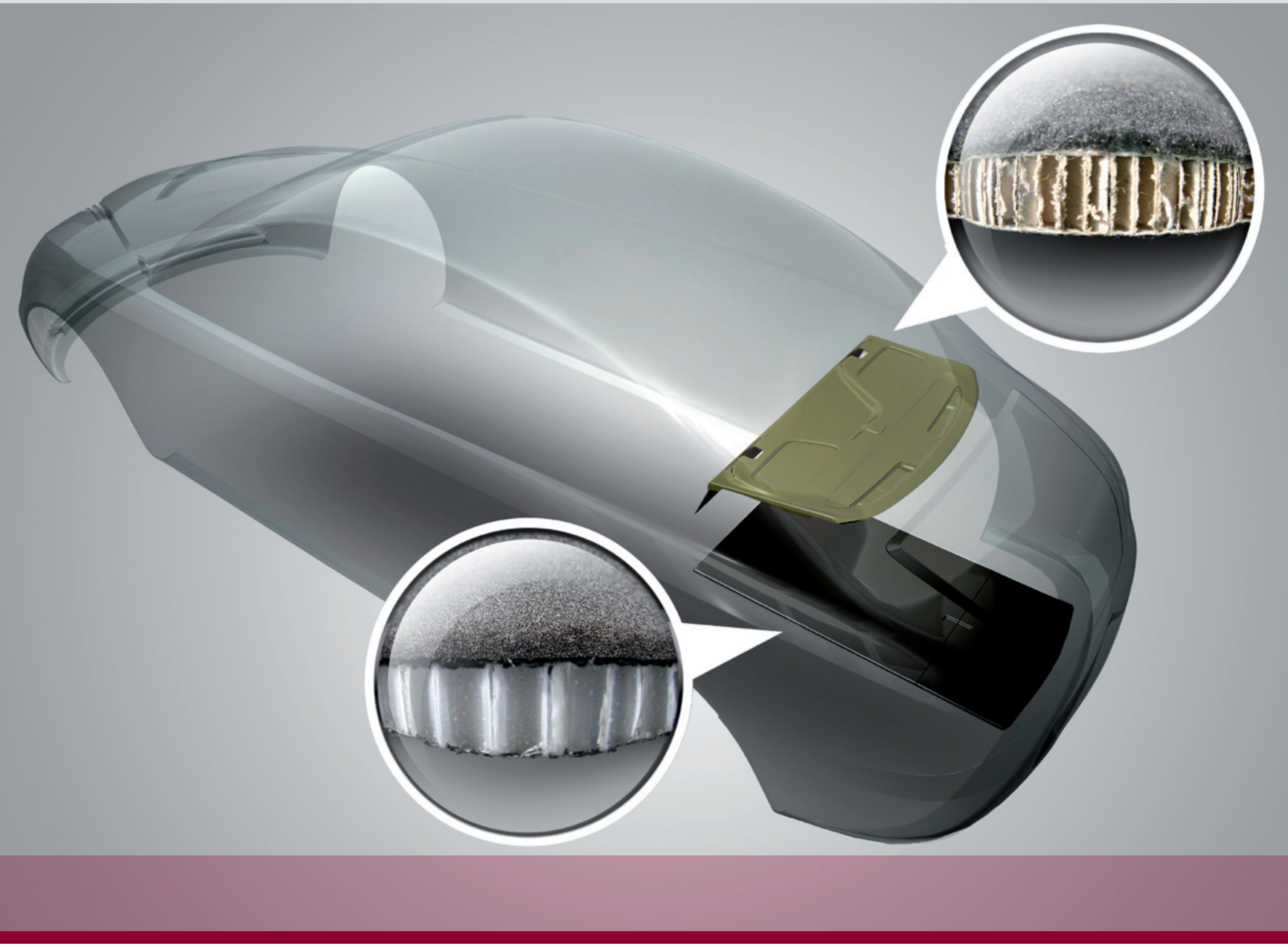


FE Simulation of Honeycomb Core Sandwich Panels for the Body

Lightweight honeycomb sandwich structures have been increasingly employed in the automotive industry: from parcel-shelf to load-floor applications. There can be an infinite variety of possible solutions adopted for these parts depending on the choice of the core geometries and of the materials. In the design phase an efficient prediction tool is needed – Rieter has developed a mathematical model based on a multi-scale asymptotic technique.

1 Introduction

Two main factors drive the progressive introduction of sandwich structures in the automotive industry: value of stiffness to weight ratio and, not surprisingly, material and processing cost. The problem of weight reduction, common to all forms of transportation for reasons of fuel economy and driving performance, demands solutions that maximize structural stiffness with less material. In the automotive industry, sandwiches have already made their first steps in replacing conventional solutions for both non-structural components – such as parcel shelf or load floor panels – in most of the automotive market, and with structural function in high performance sports models.

Certainly, the versatility of composite materials with cell structure, **Figure 1**, is revolutionizing the architecture of sandwich structures, and designs with multiple choices of periodic spatial core arrangements are becoming a reality. Based on demands of mechanical behaviour, cost, weight and acoustic performance, the design can be rationally optimized when certain theoretical guidance is available; however the availability of analytical solutions appears to be lagging. It must also be realized that even with current high-speed computers, it is still difficult to tackle a full description of periodic sandwich structures by employing Finite Elements commercial codes: the number of degrees of freedom necessary to build up a detailed model of the desired panel make its simulation very expensive (of the static or dynamic behaviour) especially if these models have to be embedded for example inside a complete vehicle model.

2 Proposed Procedure

The method essentially proposed here consists of a homogenization process and goes through the following steps: identifying the different scales characterizing the physical problem; studying the influence of each scale on the others; developing a technique to model this influence. Once the homogenized process is developed, it is possible to analyze the dynamics of the largest scale without considering the other scales.

Rieter has succeeded in extending the validity of the two-scales perturbation asymptotic technique, developed in recent literature [1], to determine the constitutive dynamic equations of the equivalent homogenous medium representing the honeycomb cellular structure: the outcome of this process is the determination, for any honeycomb core geometry, of all the nine compliance matrix terms of the equivalent homogenous orthotropic material representing the considered honeycomb cellular structure.

The derivations here described start from the general partial differential equations that represent the dynamic behaviour of the honeycomb structure (Cauchy's equations) and re-formulate them in a weak form, which is as a variational problem. This type of approach is made necessary by the fact that, in any

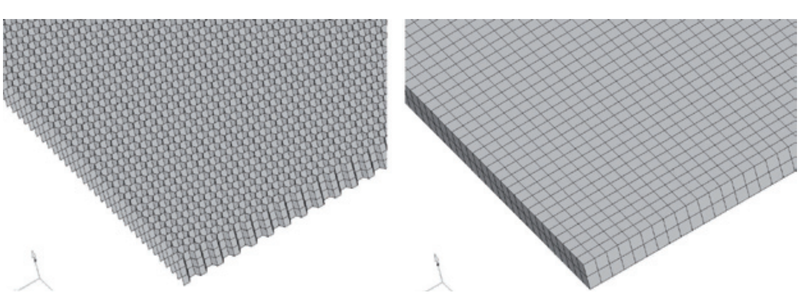
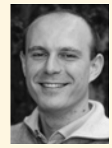


Figure 1: Comparison (FE modelling) of honeycomb validation model (left) to homogenized model (right) of a sandwich structure

Author



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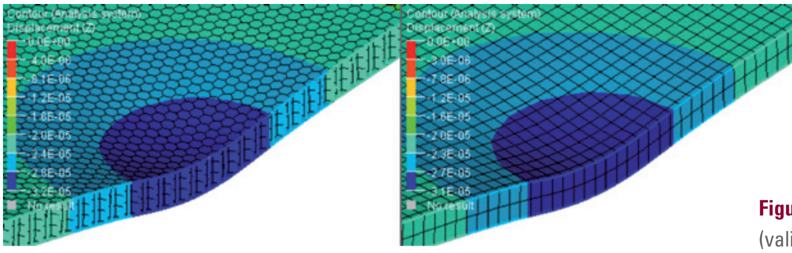

Figure 2: Static deformation comparison (validation model vs. homogenized model)

honeycomb structure, the transition from the cell wall to the air around it determines discontinuities in the local density and in the local elasticity tensor of the honeycomb structure. Therefore one considers a 3D-periodic body occupying a bounded region Ω in the Euclidean R^3 space, defined by coordinates x_1, x_2 und x_3. Assuming a linear behaviour, the equations that describe the dynamics of the body can be written as follows:

$$\rho \frac{Dv_i}{Dt} - \frac{\partial}{\partial x_j}\left(C_{ijkl}\frac{\partial u_l}{\partial x_k}\right) = \rho f_i \ \ in\ \Omega \qquad \text{Eq. (1)}$$

Here u is the displacement field, v is the velocity field, $\frac{D}{Dt}$ is the material derivative, ρ is the density, C_{ijkl} are the terms of the elastic tensor, and fi are the terms relative to the mass loads. For a linear elastic body the tensor C_{ijkl} must be positive definite and must satisfy the following symmetry conditions:

$$C_{ijkl} = C_{jikl} = C_{ijlk} = C_{klij} \qquad \text{Eq. (2)}$$

For any periodic structure the elasticity tensor C_{ijkl} and the density ρ will be periodic functions of the spatial coordinates.

As a consequence of this, it will be possible to isolate a unit cell of the global structure, intended as the region that covers a single spatial period of the elasticity tensor and of the density. By defining, on this domain, local coordinates (micro-scale coordinates) (y_1, y_2, y_3) one has $u = u(x_1, x_2, x_3, y_1, y_2, y_3, t)$, $v = v(x_1, x_2, x_3, y_1, y_2, y_3, t)$ and $C = C(y_1, y_2, y_3)$. The micro-scale coordinates are able to describe the small oscillating perturbations of the solution, while the macro-scale coordinates are able to describe the average spatial trend of the solution.

Using the periodicity of the perturbations and the Green theorem, two of the endless variational equations are extracted: the first represents the micro-scale equation while the second is the macro-scale equation.

The macro-scale equation provides the solution of the equivalent material identification, since it gives the definition of the averaged density and stiffness tensor, which are the equivalent properties needed to be identified. Indeed analyzing the expression of the average stiffness tensor, it can be seen that together with known properties such as the volume of the 3D periodic cell geometry, and the stiffness tensor of the material constituting the honeycomb, the expression presents a third order tensor χ_q^{kl}, which is unknown.

The identification of this unknown micro-scale tensor comes from the variational equation system expressing the micro-scale problem. This equation system can be solved by means of numerical computation on the cell domain. And thanks to the symmetry of the equivalent elastic tensor only the solution of ten terms of the unknown micro-scale tensor χ_q^{kl} (y_1, y_2, y_3) have to be performed. Once all the χ_q^{kl} have been obtained by numerical solution on the cell domain, for the computation of the equivalent elastic tensor we can directly apply the macro-scale dynamic equation already obtained from the endless variational equations deriving from Eq. (1).

3 Numerical/Numerical Validation

To confirm the results of the proposed formulation, we compared, for the most common honeycomb core configurations, a detailed FE model versus our homogenized approach. With this type of validation, since no problem regarding the material and geometry uncertainty arises, between the compared models, it is possible to better appreciate the accuracy of the proposed approach. Here the results obtained for a typical honeycomb core structure with hexagonal cell (with 4 mm side length and 8 mm width) are considered. In particular considering a honeycomb sandwich plate (1000 mm × 900 mm) with 1 mm aluminium skins, and 2 mm core height made with polypropylene.

The validation of the multi-scale technique is carried out employing the commercial FE code M.S.C. Nastran. The validation process consists in the comparison between the dynamic behaviours of two different finite element models of a sandwich structure with honeycomb core, Figure 1:

- the validation model, where the thin-walled core of the selected honeycomb is described with a fine mesh made of linear plate elements (CQUAD4), to which the polypropylene material properties attributed. This model ended up having nearly 218.000 nodes.
- the homogenized model, where the volume bounding the core layer has been meshed with 3D elements. This involved a much lower modelling effort compared to the validation model, leading to an FE description with only approximately 5500 nodes. To these brick elements (CHEXA) the orthotropic material (MAT 9) behaviour with the equivalent homogeneous core properties have been applied.

Both the FE models present two skins, which are simulated via isotropic linear plate elements (CQUAD4). A first comparison between the two mentioned numerical models was performed with respect to a static solution, loading the plate with a uniform load of 10 N on a central square area, and with simple support condition on two opposite panel sides, on the skin opposite to the load. The solution of the validation model took 19 min while the solution of the homogenized model took approximate-

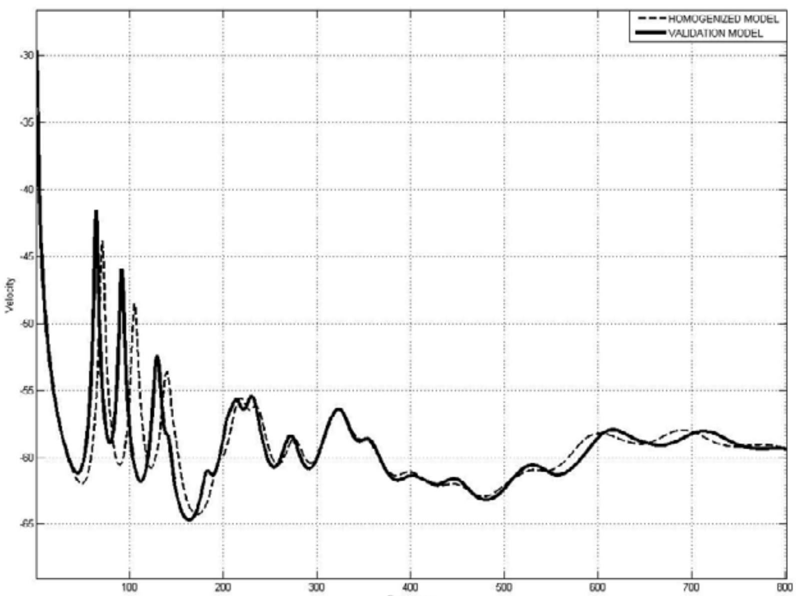

Figure 3: Root mean square (RMS) of mobility comparison (validation model vs. homogenized model)

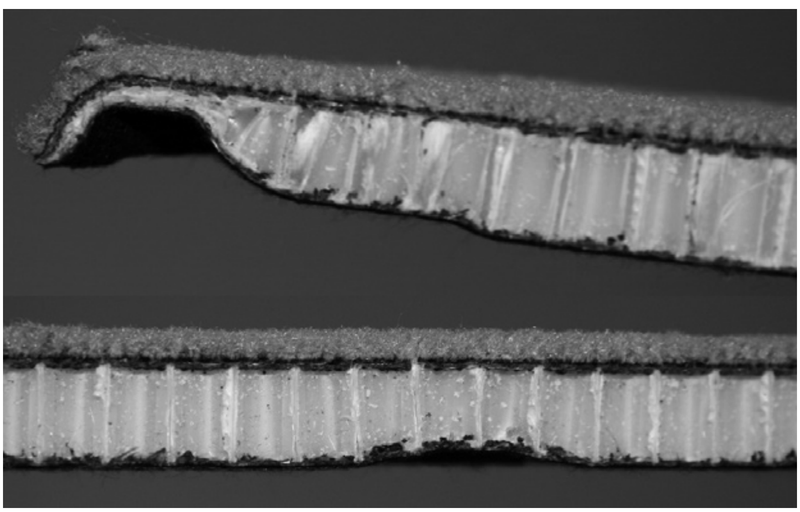

Figure 4: Detail of a trunk load floor section with honeycomb cores

ly 1.5 min. The results show that not only is the deformation profile of the full model reproduced correctly by the homogenous one, but the local deformation is also correctly represented leading to a maximum 3.5 % difference in the displacement field of these two models, **Figure 2**.

A further comparison between the two models involved the forced response of the sandwich panel, when exciting the sandwich panel, in free-free condition, on one point with a force normal to the skin surface. The average mobility frequency response function is performed over the simulated FRFs, by summing up the contributions of all the nodes belonging to the skin opposite to the one where the loads are applied. Several excitation points have been considered. **Figure 3** shows the results where one of them is applied.

Also in this behaviour comparison, despite the difference for both the modelling effort and solution time (approximately 6 h for the validation model and 43 min for the homogenized model), the proposed homogenized model is able to provide a response, which is pretty similar to the one of the validation model.

4 Numerical/Experimental Validation

Similar results as mentioned in Chapter 3 have been obtained with numerical validation of other core shapes, and with numerical and experimental validation of flat sandwich plates [2]. Here for brevity the application of the proposed homogenization method to the simulation of a load floor panel in the trunk will be reported, **Figure 4**.

This load floor is composed of sandwich carrier material, named RHOC (Rieter Honeycomb), which is made up of the following starting layers: two skin layers made out of EAC material (glass fibre based material), a polypropylene tubular honeycomb core (17 mm thickness), and on top a non woven carpet. In the production process, this starting pileup undergoes a hot moulding process, which delivers the final shape: as a result of it the part presents several areas where the core has been squeezed between the skins, but also many local skin curvatures, so that the component presents a non uniform thickness. Concerning the modelling of the core, it has been meshed with 3D linear tetra elements in order to adapt to the skin geometry: this has lead to an FE model having 17.000 nodes. The model was divided into two parts: one zone where the thickness variation could be considered negligible, and a region where a high compression ratio of the core was evident.

Both these parts inherited the stiffness equivalent macro characteristics, using the proposed methodology; the only difference was on the equivalent density given to these parts. For the numerical/experimental validation, comparison the forced response computed with the FE model just described, with the spectra on two load floor samples are measured. These were suspended with rubber bands on the upper corners, acquiring the vibrations on one floor skin by means of a laser scanner on 221 points, **Figure 5**.

Figure 6 shows how well the proposed modelling can correlate to the measurements. Despite the high geometrical and material complexity, the numerical model can find a pretty good agreement with measurements up to nearly 500/600 Hz. Only at higher frequencies does it seem

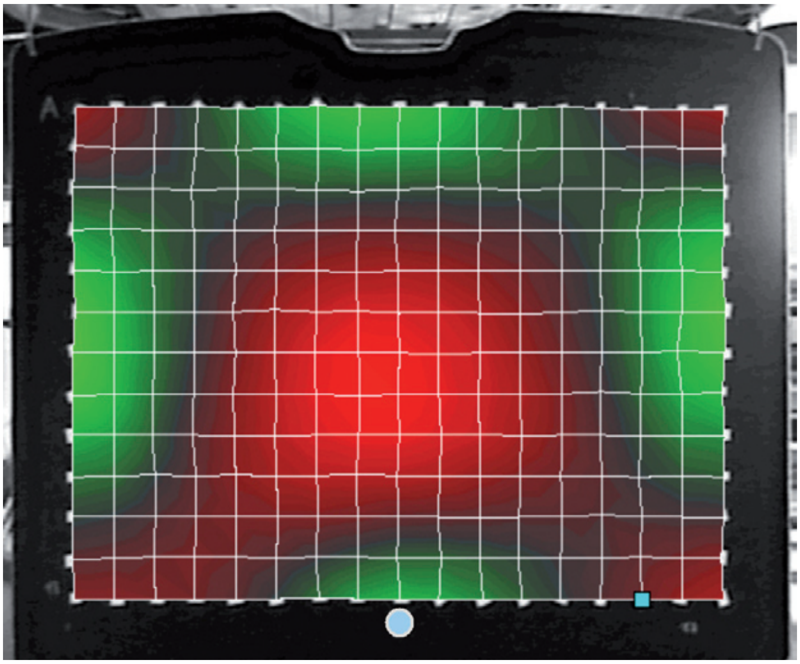

Figure 5: Snapshot record with a laser scanner of a load floor deformation mode

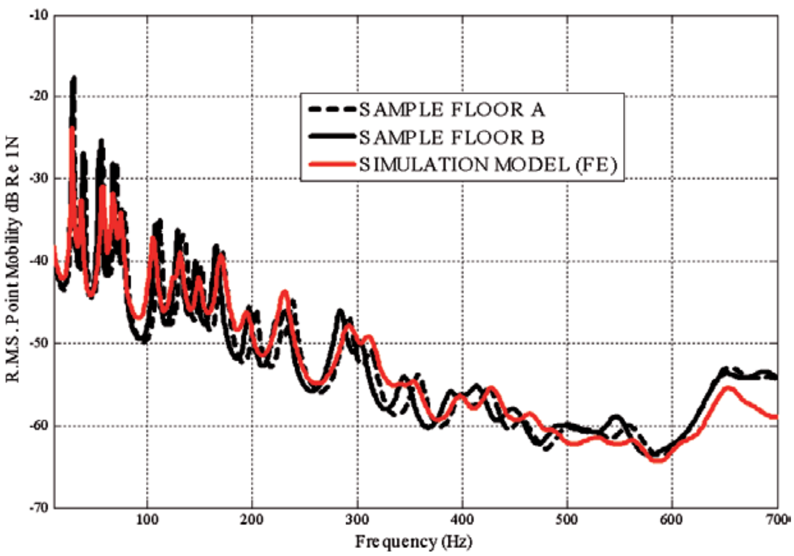

Figure 6: Numerical/experimental comparison of root mean square (RMS) values for mobility velocity

to lose its accuracy even though the trend of the experimental behaviour appears to be respected.

5 Outlook

The work of FE simulation done by Rieter in the past years has shown that the proposed homogenization approach can be efficiently used. It is thus possible to predict the equivalent elastic tensor of honeycomb cores with any general core configurations.

The numerical approach has been validated using industrial finite element software, onto several sandwich constructions having various core configurations, with respect to static and dynamic analysis types. This simulation has shown a great accuracy, as well as direct implementation in the design phase of honeycomb sandwich car parts like for trunk load floor panels. It can seamlessly handle parts with variable core thickness and curved surfaces, requiring a reduced FE meshing effort.

Moreover the honeycomb core solution can be changed either by varying parametrically the core shape or by changing the core material, without having to re-mesh the FE component. Therefore it can be applied in the design of honeycomb sandwich and in the identification process of efficient cellular core layouts with respect to NVH targets.

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