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NUMERICAL MODELLING OF SOUND RADIATION AND TRANSMISSION LOSS OF A HONEYCOMB PANEL USING MEASURED MATERIAL PROPERTIES

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Abstract

Facing the increasing demand of passengers for comfort and well-being, low noise levels inside the aircraft cabin become more and more an important development objective. Current practice for optimising the cabin noise are mainly cost- and time-intensive experimental techniques. Therefore in order to cut down development costs, efficient computational methods to predict the acoustical behaviour of aircraft fuselage and lining are highly demanded. In particular, suitable numerical models for honeycomb sandwich plates – the major design material for components of the cabin lining – are needed.

In the current paper, a computational method based on a coupled model of Finite Elements and Boundary Elements to predict the sound radiation and transmission loss of honeycomb panels is used. The honeycomb core is thereby represented by means of 3D solid elements, while 2D thin shell elements are used for taking into account the facing skins.

A successful forecast of the acoustical behaviour requires not only accurate element models, but also a detailed knowledge of the material parameters which often is not available. Thus the Young’s and shear modulus of honeycomb plates found in technical data sheets usually lead to an underestimation of the real values, as they are minimum quantities guaranteed by the manufacturer for dimensioning tasks. To get reliable data for the current study, the elastic properties of the facing material have been determined by standard tensile testing. Furthermore, the honeycomb core was investigated using a dynamic method to measure the complex Young’s and shear modulus.

The determined material properties are finally used as input data for the numerical model to predict the acoustical behaviour of a honeycomb panel. The obtained results show an excellent agreement with transmission loss and sound radiation measurements of the panel.

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INTRODUCTION

Combining high mechanical stiffness with low weight honeycomb sandwich plates are the major design material for the cabin lining of today's passenger aircrafts. Interior parts like ceiling panels, sidewall panels or hat racks often consist of honeycomb cores covered on both sides by composite laminate face skins. However, one of the drawbacks of honeycomb panels is their poor acoustic performance which is mainly caused by the high stiffness and the low mass per unit area rates of typically 1-2kg/m^2.

To allow prediction and optimisation of the vibroacoustic behaviour already in the early design phase, different methods have been developed in the last years. Beside analytical approaches like, e.g., suggested by E. Nilsson, A.C. Nilsson et al. [9][10] or by Huang et al. [5], numerical methods become more and more important due to steadily rising computer performance. Buehrle, Grosveld et al. [2][4][6][12] showed a procedure to calculate transmission loss and sound radiation of a curved honeycomb panel based on Finite and Boundary Element models. A similar approach has been suggested by von Estorff and Wandel for the modelling of cabin lining elements, based on a baffled formulation of the Boundary Element Method (BEM) and the Finite Element Method (FEM) [3].

The numerical models used in [3] for the calculation of transmission loss and sound radiation have shown their potential to provide reliable results when being compared with measurement data. However, for predicting the acoustic behaviour of a virtual prototype instead of an existing component, an accurate knowledge of the material properties must be provided to gain dependable results.

This paper proposes a method to determine the necessary material properties of a flat honeycomb panel and compare those with the values found in literature. Moreover, a FEM modal analysis of the panel is made using both literature and measured material parameters. With the calculated modes, the transmission loss as well as the sound radiation is computed and compared with measurements.

DESCRIPTION OF THE HONEYCOMB PANEL

For the purposes of this study, a flat honeycomb panel consisting of an aramid honeycomb core with a phenolic resin two-ply woven glass facing skin on each side has been manufactured. The quadratic panel has an edge length of 0.9m, a thickness of 8.7mm after curing, and a mass per unit area of 1.82kg/m^2. The untreated core has a thickness of 9.4mm, a cell size of 3mm, and an average spatial density of 48kg/m^3, while the facing skins have a thickness of 0.36mm and a density of 1900kg/m^3. Both materials, the core and the facing skins, can be regarded as typical representatives that are commonly used for the creation of components of the cabin interior lining.

MATERIAL PROPERTIES

Various material properties of the honeycomb core and the facing skins, like density, thickness, Young's and shear modulus, can be found in literature. However, these values often do not provide sufficient information needed for vibroacoustic calculations. While density and geometric measures generally picture the real values, this may not automatically apply for the specification of Young's or shear modulus. Since the elastic properties are used for dimensioning tasks, the values that are specified by material manufacturers or claimed in customer material standards often can be regarded as "minimum demanded quantities". Especially fibre reinforced composite materials generally show a wider spreading of the elastic properties compared to homogenous materials such as metals, thus resulting in bigger differences between values from literature and real measurement data.

Facing skins

To get reliable input data for the numerical model, 6 samples of the facing skin material have been produced by separately curing the prepregs without a core. The samples of 250mm x 50mm (according to EN ISO 527 type 2) were investigated by standard tensile test. The results are shown in Table 1.

Table 1 – Facing skin: Measured Young's modulus from tensile test

	Minimum value [GPa]	Maximum value [GPa]	Average value (6 test samples) [GPa]	Difference to literature value [%]
Young's modulus	18.99	20.23	19.70	+40.71

The average measured Young's modulus of the samples amounts to 19.7GPa. As expected, it is thereby clearly above the required mean value of 14.0GPa specified by the material standard [17]. The different samples vary more than 1.2GPa.

Honeycomb core

In contrast to static calculations, the knowledge of material damping and a possible frequency dependence of the elastic properties can be important when performing vibroacoustic calculations. Especially the honeycomb core is assumed to have a significant damping, therefore complex Young's and shear modulus were examined.

The method used to determine the complex Young's modulus harmonically excites the lower end of a material specimen, while the upper free end is loaded with a mass (Figure 1). To calculate the complex modulus, the transfer function between accelerometer 1 and 2 is evaluated as a frequency function. With the resonance peak, the eigenfrequency ω_0 and the width Δ at half transfer function maximum is known. For low ratios of specimen mass m to loading mass M, the magnitude E_d and the loss factor η_E of the complex modulus at resonance frequency can be calculated by

$$E_d = M \cdot \frac{h}{A} \cdot \omega_0^2 \qquad \text{and} \qquad \eta_E \approx \frac{\Delta}{\omega_0}, \tag{1}$$

where h denotes the thickness and A the area of the specimen [7][8].

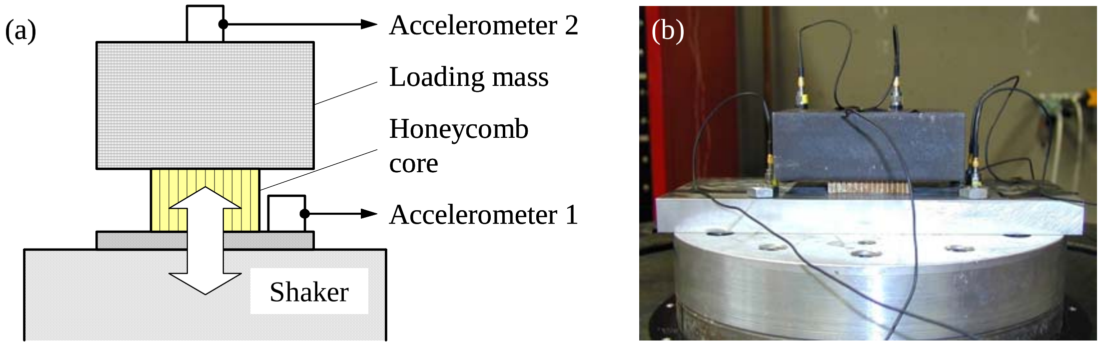

Figure 1 – (a) Schematical experimental setup (b) Real test setup

Furthermore, using the transfer function method [11] it is possible to calculate magnitude and loss factor of the complex modulus in a larger frequency interval without changing the loading mass. With an approximation of the transcendental equations that is valid in the frequency interval around $0.5 \cdot \omega_0$ to $2 \cdot \omega_0$ [11] one gets

$$E_d = M \cdot \frac{h}{A} \cdot \omega^2 \cdot \left(1 + \frac{m}{2M}\right) \cdot \frac{|T|^2 - |T| \cdot \cos\varphi}{|T|^2 - 2|T| \cdot \cos\varphi + 1} \qquad \text{and} \qquad \eta_E = \frac{\sin\varphi}{\cos\varphi - |T|}, \tag{2}$$

where T denotes the magnitude and φ the phase of the transfer function. Measurement results for the Young's modulus of the honeycomb core for both the resonance and the transfer function method are shown in Figure 2.

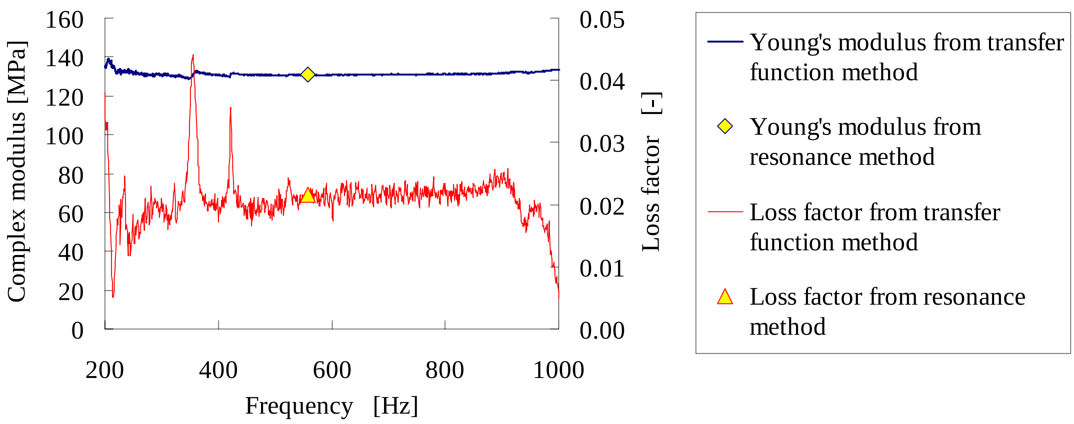

Figure 2 – Honeycomb core: Measured Young's modulus and loss factor

Rotating the excitation direction defined in Figure 1a by 90 degrees, similar measurements can be done for the shear modulus [1][7]. Analogous to the Young's modulus, a frequency dependence of the shear modulus was not observed for frequencies up to 1kHz, while the loss factor was also constant at around 0.02 in both L- and W-direction. The measured values are given in Table 2 and show a very good agreement with the ones found in literature [13].

Table 2 – Honeycomb core: Measured Young's and shear modulus at resonance frequency

	Loading mass [kg]	Resonance frequency [Hz]	Modulus [MPa]	Difference to literature value [%]
Young's modulus	2.840	557	130.80	+0.62
Shear modulus W-direction	3.954	426	26.63	+6.52
Shear modulus L-direction	3.971	522	40.15	+0.38

NUMERICAL MODEL

To describe the vibrational behaviour of the panel, a structural Finite Element (FE) model was created in MSC.PATRAN with a global edge length of 10mm. The honeycomb core was thereby modelled with 8.100 solid HEX8-elements using 3D-orthotropic material properties to consider the different elastic characteristics of the core in the different directions. Due to their low thickness, the two facing skins were discretised each with 8.100 shell QUAD4-elements and isotropic material properties. The use of 2D-orthotropic material properties for the facing skins of the investigated panel was not necessary, since the woven fabric shows a symmetric texture.

The normal modes of the panel were computed in MSC.NASTRAN using the solution SOL103 solver and exported for acoustic simulation within LMS.SYSNOISE. For the calculation of transmission loss and radiation efficiency, a coupled FEM/BEM model and an indirect variational approach is used. While the radiation efficiency was determined for "free/free" boundary conditions, the transmission loss was calculated with translative supported boundary conditions (simple support) at the four edges of the panel and a baffled formulation of the BEM.

Normal modes have been evaluated up to 8kHz, thus allowing for acoustic calculations up to 4kHz [16]. A detailed description of the implemented FE and BE modelling and the baffled approach can be found in [3][16].

MODAL ANALYSIS

In order to show to what extend the predicted structural behaviour is affected by using literature or measured material properties respectively, the normal modes for free boundary conditions have been calculated for each of the two sets of material properties. To assess the quality of the results, an experimental modal analysis of the real panel suspended “free” on elastic bands has been performed as well. The measured Frequency Response Function (FRF) of acceleration and excitation force at the driving point and the simulated FRF for literature as well as measured material properties are shown in Figure 3.

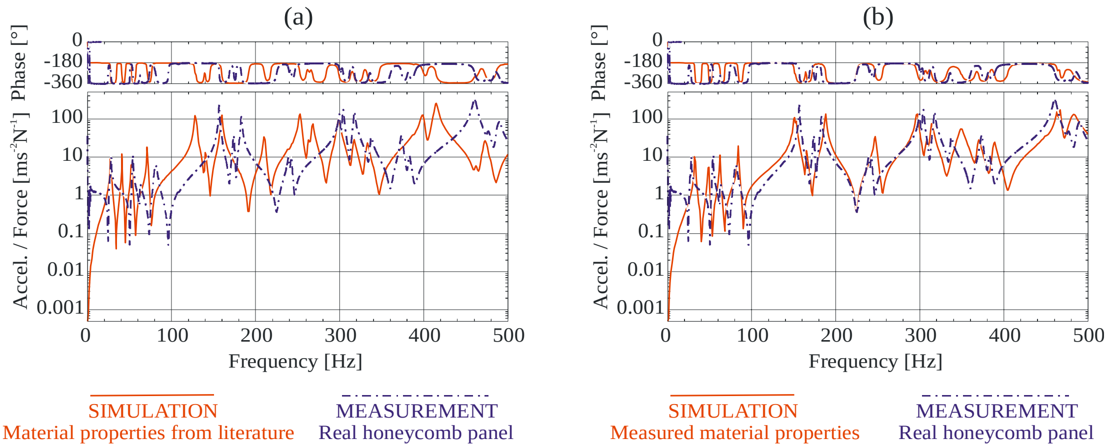

Figure 3 – Driving point FRF of acceleration and excitation force from FE simulation for (a) literature and (b) measured material properties compared to measured FRF of the real panel

It can be seen that the FE model with material properties obtained from literature does not reflect the real panel behaviour adequately (Figure 3a). A comparison of the first 20 measured modal frequencies with the ones of the corresponding calculated mode shapes shows an averaged deviation of 14.1%. On the other hand, the FE model based on measured material properties reduces the deviation to 3.4% and represents the frequency response of the real panel quite satisfactory (Figure 3b).

TRANSMISSION LOSS AND RADIATION EFFICIENCY

To study the influence of the different material parameters on the predicted acoustic behaviour, the transmission loss and the radiation efficiency have been calculated using the software LMS.SYSNOISE. As illustrated in [3], the transmission loss was determined by means of the simply supported modes employing one after the other both sets of material properties as input data. The simulation results are compared to the transmission loss measurements conducted at a test facility according to the intensity method of the ISO 15186 norm [15]. The transmission losses of simulation and measurement have been evaluated in third octave band steps up to 4kHz. Due to the size of the reverberant room, the test facility provides reliable data for frequencies above 250Hz. The results are shown in Figure 4.

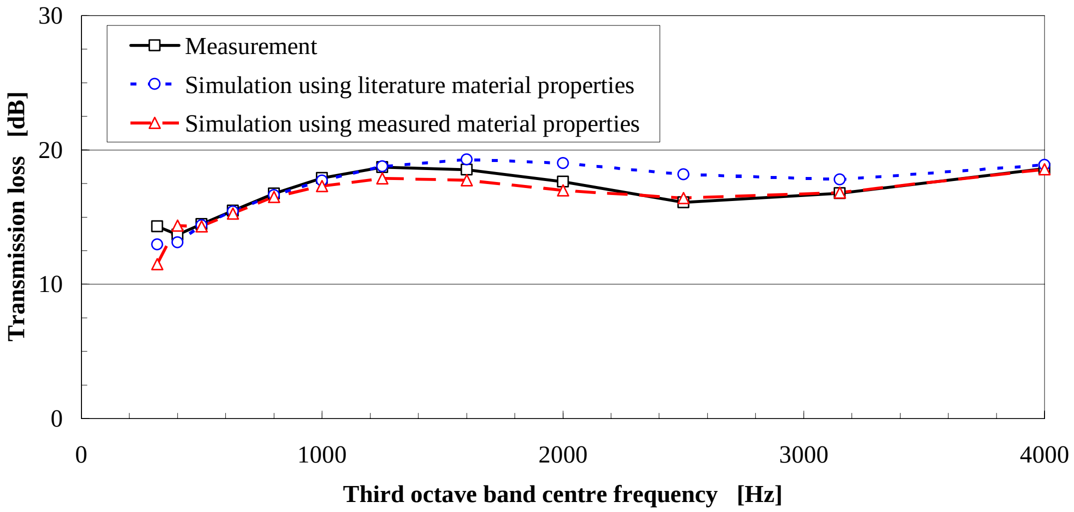

Figure 4 – Calculated transmission loss for literature and measured material properties compared to measured transmission loss of the real panel

The maximum deviation between the measured and the calculated transmission loss is in both cases less than the limit *R* of 3.5dB according to ISO 140 [14], which was used as a performance index for the assessment of the simulation results. However, the results obtained with measured material properties describe the panel characteristics more accurate. Especially the coincidence frequency of 2.5kHz is predicted exactly by using measured material properties, while being overestimated to 3.15kHz when applying the lower literature values to the elastic properties.

To allow a further assessment of the simulation models, the results of radiation efficiency calculations for the freely suspended panel are superimposed with the according measurement in Figure 5. In compliance with the transmission loss calculations, the model using measured material properties shows the better agreement with the radiation measurement of the real panel.

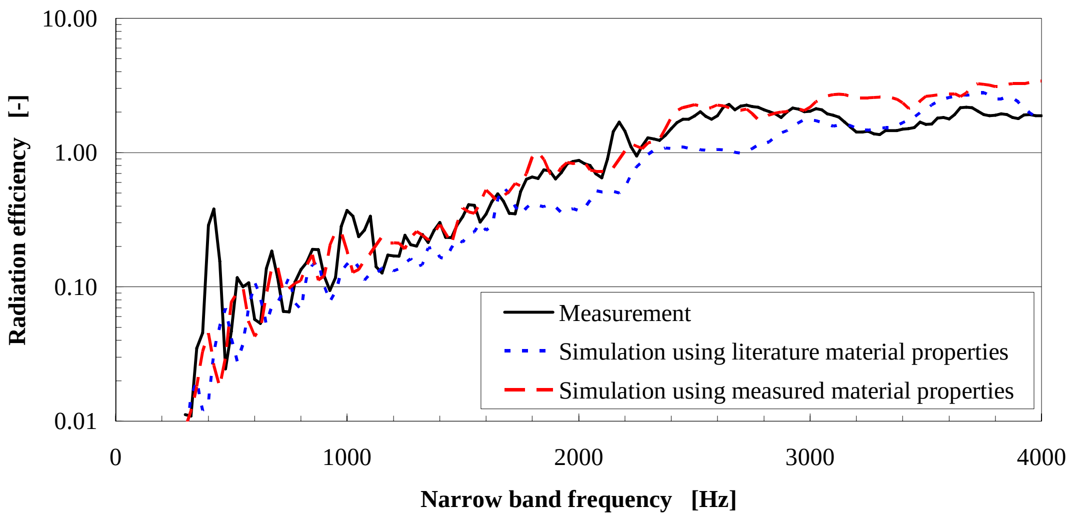

Figure 5 – Calculated radiation efficiency for literature and measured material properties compared to measured radiation efficiency of the real panel

CONCLUSIONS

The facing skin material of a honeycomb panel was investigated by tensile testing. The comparison between literature and measured material properties showed a significant difference in the elastic properties. However, the honeycomb core, which was examined with the resonance method and the transfer function method, agreed very well with the theoretical values. A frequency dependence of Young's modulus, shear modulus and damping could not be observed for the core material.

Different FE and BE calculations have been performed to determine the effect of applying measured material data instead of relying on literature values. The results of the modal analysis, the predicted transmission loss and the radiation efficiency show that using measured material properties leads to a more accurate representation of the vibroacoustic characteristics of the real panel. Especially the eigenfrequencies and the coincidence frequency were not predicted sufficiently, if literature values are applied. When aiming for reliable models, measured elastic properties for the fibre reinforced facing skins are one of the basic requirements, while literature values are sufficient for adequately modelling the honeycomb core.

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