Lightweight structures represent an appealing solution in a large number of very different technology areas and they are therefore implemented in a wide range of applications. The high stiffness over weight ratio, typical of lightweight sandwich structures, can however give rise to a poor overall NVH behaviour. Hence, the designer has to ensure that the advantages of a significant weight saving are not nullified by a collateral and undesired reduction of functional vibro-acoustics performance. The present deals with the testing and design optimisation of a composite sandwich panel. NVH performance is evaluated using a dedicated experimental setup for the vibro-acoustic characterisation of lightweight panels. Experimental modal analysis and acoustic insertion loss tests are carried out in order to verify the structural and acoustic numerical models. The predicted improvements in NVH behaviour are experimentally verified with good accuracy. The vibro-acoustic test rig is shown to be a valuable support for the design of composite structures when NVH performance is a key indicator attribute.

1. INTRODUCTION

Lightweight structures find their application in various technology sectors, from automotive to marine, aerospace, railway and building industries. Typically, the main advantage offered by a significant weight saving results in unsatisfactory NVH behaviour, as collateral effect of a high stiffness over weight ratio. A comprehensive understanding of noise reduction characteristics of unconventional constructions and materials becomes fundamental already in the first design stages. This work presents a design optimisation of a composite structure and is carried out in the scope of a joint research program between KU Leuven and Toyota Motor Europe (TME-Toyota), Brussels. In this framework a lightweight structure able to satisfy specific structural and noise reduction specifications is sought. Classical numerical modelling methods and experimental tests are used to obtain a final design which meets the project requirements. The study starts from an already existing solution (a composite sandwich panel) implemented for similar purposes in other applications. Modal behaviour and acoustic Insertion Loss (IL) of this reference panel are tested in order to build up and check a structural Finite Element (FE) model and a fully coupled vibro-acoustic FE-Indirect Boundary Element (FE-IBEM) model. In a next step, the former structural model is included in an optimisation loop aimed at the identification of a new design able to offer the desired dynamic stiffness, while minimising the panel weight. The problem variables are chosen to be the core thickness and the stiffeners configuration. The final layout is experimentally verified. All NVH tests are carried out using a dedicated setup for vibro-acoustic characterisation of lightweight panels at KU Leuven. The expected NVH behaviour is accurately confirmed with measurements, proving the potential of this test rig as a helpful tool for design of composite structures with specific NVH requirements.
2. RESEARCH GOAL

A new experimental test rig for vibro-acoustic characterisation of trim materials is developed. The setup makes use of an acoustic receiver cavity for sound pressure measurements. For practical reasons (handling and (dis)mounting of the cavity) this cavity component needs to be as light as possible while still providing sufficient acoustic and structural stiffness, such that it is sufficiently rigid up to 1 kHz. Based on an extensive literature review it was found that some lightweight materials are sometimes used instead of a conventional concrete walls solution [1] to obtain rigid acoustic cavities. As reported in [2] when describing the main guidelines for constructing reverberation chambers for automotive testing purposes, the authors explicitly mention that, apart from the traditional constructions of solid masonry, some alternative prefabricated acoustic panels can be used to ensure both a proper room sound isolation and notable reflecting surfaces inside the cavity. They refer to panels typically fabricated of heavy gauge sheet steel backed with gypsum drywall able to produce a massive, sound reflecting enclosure. In the present study however, the authors focus on lightweight sandwich solutions due to the strict design specification of a low maximum allowed weight of the acoustic cavity. In order to guarantee an easy dismantling and installation of the cavity and the overall ease of operation of the setup as a whole, the weight of the structure is limited to about 40kg. Starting from a composite sandwich construction already implemented in similar applications, first a testing campaign is carried out to evaluate its vibro-acoustic properties in terms of dynamic behaviour and noise reduction characteristics. Results from this first phase are used to build up and verify a corresponding structural FE model, which is afterwards optimised to meet the project requirements. The NVH properties of this final design are numerically analysed and furthermore experimentally verified.

2.1 NVH BEHAVIOUR OF THE FIRST PANEL: EXPERIMENTAL STUDY

The composite sandwich panel (660 x 860 x 55 mm³) consists of glass fibre (GF)-epoxy resin face sheets, in which several layers are stacked according to a specific layout and a foam filled core in which unidirectional stiffeners spaced 400 mm from each other are present, see Figure 1.

2.1.1 EXPERIMENTAL MODAL ANALYSIS

An experimental modal analysis (EMA) is carried out on the freely suspended panel in order to verify the composite and core material properties and to validate the FE model. Some of the nominal geometric and material values specified by the manufacturer are not precisely known, due to variability and uncertainty in the material production process and product assembly. The most important ones are the density and thickness of the foam used to fill the core and thickness of the composite layers used for the face sheets. Other material parameters are taken from literature, like the shear modulus and Poisson ratio of the face sheet material. Using these initial nominal values, discarding any variability, a correlation analysis is made between the FE and the experimental results. A good correlation of mode
shapes up to 1 kHz is observed, although a shift in terms of natural frequencies is apparent. In general, the numerical model appears to be too flexible when compared to the actual structure resulting in lower predicted natural frequencies. Figure 2 and Figure 3 show the Modal Assurance Criterion (MAC) values for the first ten experimentally identified modes and their numerical counterpart. Despite a frequency shift of about 10% observed for most of the modes up to 1 kHz (although the frequencies of the second and fourth modes, with one half wavelength in the width and length direction respectively, are off by 27%) the model is kept unvaried for subsequent simulations, in view of the uncertainties regarding the aforementioned parameters.

Lightweight accelerometers PCB 352A24, a PCB 086C03 load cell, an LMS SCADAS III acquisition system, LMS Test.Lab9B and Virtual.Lab Acoustics Rev.8b are used to carry out and post-process the experimental tests results and numerical analyses.

2.1.2 EXPERIMENTAL VIBRO-ACOUSTIC ANALYSIS

An experimental setup, designed and realised at KU Leuven [1], is used to evaluate the vibro-acoustic behaviour of the baseline panel. This test rig is specifically developed for the experimental investigation of noise reduction and acoustic absorption characteristics of lightweight structures. It consists of a
concrete cavity of 0.83 m$^3$ irregularly shaped to ensure an even distribution of the acoustic natural frequencies at low frequency below the diffuse field region (no pair of parallel walls is present inside the cavity, [1]). The acoustic volume is excited by means of a loudspeaker set in one of the corners and it provides a flat spectrum from 100 Hz to 15 kHz. The cavity can be fully closed by a rigid front wall for the acoustic absorption test configuration, or an aperture can be made in the front wall to host test specimens of different sizes and thicknesses. In the latter configuration the setup allows the evaluation of the acoustic Insertion Loss (IL) in a wide frequency range, typically from 50 Hz to 20 kHz. The experimental procedure necessitates two measurements of the sound power to determine the IL, i.e. once in the open cavity configuration and once when the panel is inserted to close the aperture, see Figure 4 (left and right respectively). Additionally this setup allows the accurate identification of vibro-acoustic coincidence frequencies.

![Figure 4 Vibro-acoustic setup at KU Leuven. Open configuration on the left. Closed configuration on the right, by insertion of the A2 test panel.](image)

A pink noise signal is sent to the system with frequency content from 0 Hz to 1.6 kHz. The radiated acoustic power intensity is recorded outside, by scanning the vibrating surface of the test panel and the open window with an intensity probe (B&K type 3584). The spacer set in between the probe’s microphones is 50 mm, restricting the measurement range to [20 Hz – 1.25 kHz]. The cavity front wall has an A2 sized window. An additional frame clamps the sample all around its edges (Figure 4 on the right). The measured third octave band average IL is shown in Figure 5, together with the numerical prediction that will be discussed in details in the next section. Bands at and below 100 Hz are discarded due to the low frequency limitation of the loudspeaker.

Before introducing the structural and acoustical numerical models, some additional remarks on the use of the experimental setup are given. The radiated acoustic intensity measured over the panel surface and the numerical prediction of the acoustic power radiated by the latter are plotted in Figure 6. On the same graph the natural frequencies of the uncoupled acoustic cavity (green squares) and those of the clamped panel (red diamonds) are indicated. As evident from Figure 6, the acoustic power intensity scanning allows an easy and full understanding of the coupled behaviour between the structural component and the acoustic cavity. The three structural modes inside the considered frequency range are coupled to the acoustic field inside the cavity. The intensity maps clearly indicate this coupling in terms of acoustic radiation around the uncoupled natural frequencies.
The source input in the numerical model is an acoustic monopole emitting a flat unit power spectrum up to 1000 Hz. The shape and the level of the input signals have no influence on the final IL, since this quantity is the difference of the acoustic powers radiated with and without the sample, as long as the assumption of linearity is valid. This explains the difference shown in Figure 6 between the trend of numerical radiated power and the measured acoustic intensity.

With the use of a smaller spacer (12 mm) installed in the intensity probe, the intensity measurement provides acoustic IL up to 7 kHz. Figure 7 also clearly shows the coincidence frequency of the examined sample. Looking at the narrow band results, a strong coupled modal behaviour exhibited by
the system (panel and cavity) is evident up to 900 Hz. Above this region, similar as for Transmission Loss (TL), the IL is controlled by the structural mass and it adopts a 2dB/octave slope behaviour. The frequency band at 4 kHz is characterised by the coincidence phenomenon (at around 4.3 kHz), which causes a clear dip in the IL curve. Above coincidence the IL starts again rising, with an increase of 6dB/octave.

Figure 7 Narrow band and third octave band average IL.

The nature of the coincidence is related to the symmetric motion of the sandwich structure and can be estimated as described in [3]. Symmetric motion involves deformation of the core in the thickness direction (Figure 8).

Figure 8 Symmetric (on the top) and antisymmetric (on the bottom) motions in a sandwich panel.

This kind of coincidence occurs near the double wall frequency, defined by the mass of the face sheet and the stiffness of the core and which can be approximated as reported in [4]

\[
\tilde{f}_{DoubleWall} = \frac{1}{2\pi} \sqrt{\frac{E_c}{t_c} \left( \frac{1}{m_{f_1}} + \frac{1}{m_{f_2}} \right)} \quad (1)
\]

where \(E_c\) and \(t_c\) are the Young’s modulus and thickness of the sandwich core and \(m_{f_1}\) and \(m_{f_2}\) the surface mass of the two face sheets respectively. Approximating the Young’s modulus of the core with the Young’s modulus of the foam material (neglecting the stiffeners contribution) the formula above gives \(f_{DoubleWall} = 3.80\) kHz, which is still a good indication for the measured value.
3. NUMERICAL MODEL AND OPTIMISED DESIGN

The FE model of the structural component already introduced showing the EMA-numerical correlation, is now implemented to study the panel’s vibro-acoustic behaviour. No numerical updating and structural damping is performed on the first model, implementing the nominal geometric and material values. A geometrically reduced model, meeting the front wall window size of the test setup (A2) is built. It consists of 12600 linear 8-noded solid elements to model the foam core, 2520 cubic 4-noded shell elements for each of the faces and 420 for the stiffeners (ref. Figure 1). A numerical modal analysis on the isolated panel case with clamped edges is carried out using MD Nastran 3Rb. The resulting modal structural FE model is then coupled with an indirect BE model of the acoustic cavity and a fully coupled FE-IBEM analysis is carried out in LMS Virtual.Lab Acoustics Rev.8b out to evaluate the IL.

The relative simplicity of the geometry allows accurate numerical modelling (i.e. the geometry is not a limiting factor, although the non-exact clamping of the panel in the front wall is not straightforward to model) and a Wave Based (WB) model can be adopted to overcome the limitations of standard numerical techniques (FEM/IBEM) at higher frequencies [5]. The latter however is out of the scope of the present work and discussed in more detail in [6].

As mentioned before, the source input in the numerical model is an acoustic monopole emitting a flat unit power spectrum up to 1000 Hz and located at the central point of the actual speaker inside the cavity, neglecting the speaker’s directivity. The shape and the level of the input signals have no influence on the final IL, since this quantity is the difference of the acoustic powers radiated with and without the sample, as long as the assumption of linearity is valid. The third octave band IL is compared to the experimental results in Figure 5. To increase the fidelity of the numerical model a real valued acoustic normal impedance boundary condition (14E+4 kg/m²/s) is applied at the cavity walls to take into account their finite (although very small) acoustic absorption. Even though the structural damping is neglected in the present simulation, the numerical prediction is close to the measured IL in the whole examined frequency region with a maximum deviation of 4dB.

Based on the numerical model described above, a model optimisation is carried out with the aim of designing a stiffer structure able to meet some project specifications, with minimum weight and satisfactory NVH performance.

3.1 OPTIMISED DESIGN

A new experimental test rig for vibro-acoustic characterisation of trim materials is developed. The setup makes use of an acoustic receiver cavity for sound pressure measurements. A design optimisation study is carried out, aiming at modifying the structural layout of the baseline panel such that it can be used as walls of a receiving room for acoustic tests. The total air volume inside this cavity is limited to approximately 0.3 m³ and the dimensions of the base are fixed. The optimised structure is designed to exhibit satisfactory structural rigidity combined with good wall reflectivity and sound insulation properties to avoid sound leakage in and out of the acoustic cavity which may adversely affect the accuracy of the setup. In order to guarantee an easy dismantling and installation of the cavity and the overall ease of operation of the setup as a whole, the weight of the structure is limited to about 40kg. This is translated in an optimisation problem with the cost function being the minimum global weight, subjected to the constraints of satisfactory acoustic insulation and structural rigidity (wiz. a
difference of less than 3 dB in the mean sound pressure level inside the acoustic volume with perfectly rigid walls and sandwich panel walls).

A skewed sidewall cavity configuration is preferred over a perfectly parallelepiped shape (Figure 9) both for a weight minimisation issue and for structural performance enhancement. This way, the implementation of reduced size panels at the roof and all sidewalls increases their first bending eigenfrequencies. Nevertheless, from a design point of view, such a configuration is more challenging than a regular parallelepiped solution, since the stiffener layout needs to be adapted to the parallelepiped shape of the sidewalls. Since the roof of the cavity still is rectangular and approximately square, a regularly spaced stiffener configuration is selected for this side. On the contrary at the lateral walls they follow the trapezoidal shape, ending in equally spaced endpoints along the base of the cavity. Horizontal stiffeners are also considered in the sidewalls, running parallel to the roof surface and equally spaced. The proposed stiffener layout is shown in Figure 10.

With the general layout fixed, a parametric study is performed taking into account the following design variables:

- sandwich panel thickness - due to manufacturing constraints the thickness of the walls may vary between 50mm and 78mm;
- stiffener spacing in both the roof and sidewall panels;
- layout and thickness of the GF-epoxy materials used for the stiffeners and skins - both the number of plies and their orientation angle can be chosen freely. To avoid bending-torsion coupling effects and since the wall panels do not exhibit a clearly dominant direction of flexibility, only symmetric laminates are considered.

The design study is carried out in two stages. First a large number of design alternatives is assessed on panel-level based on their coupled vibro-acoustic performance compared to a perfectly rigid cavity, their weight and their manufacturability and production cost. In a next step, a limited number of candidate configurations are integrated in a model of the full box to assess the impact of coupling effects of the sidewalls on the acoustic performance. The resulting layout consists of a 50mm spaced roof reinforcements configuration, combined with 8 horizontal stiffeners on each of the sidewalls. The skin panels are made of a 2.5 mm thick quasi-isotropic layup of GF-epoxy material and all the walls are 78mm thick, while twice the former layout is used to produce the stiffeners (which are 5 mm thick). The resulting stiffener configuration at the sidewalls is similar to that shown in Figure 10, which is relative to an A2 sized flat sample. The latter is used to experimentally validate the predicted NVH behaviour.
Design parameters of the initial and optimised panels are compared in Table 1.

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Table 1. Design parameters of the initial and optimised panels.

4. EXPERIMENTAL VALIDATION

A similar numerical analysis as described for the first panel is carried out for the final design. The stiffeners layout of this panel is shown in Figure 10. An FE-IBEM model is built and the acoustic IL is calculated (with the same boundary conditions applied). Numerical and experimental results are compared in Figure 11. Good agreement is found with the experimental dataset, with a maximum error less than 4dB (absolute value) in the considered frequency range, similar to the initial panel.

As shown by the IL comparison illustrated in Figure 12, the optimised panel solution indeed offers better noise reduction performance than the initial panel. Stiffening the structure has an especially strong effect on those frequency regions which are affected by the presence of resonance frequencies of the first panel (400 Hz, 500 Hz and 630 Hz bands). This gain is almost nullified at 800 Hz due to the presence of the first resonance frequency of the optimised sandwich around this frequency. The stiffening effect is also visible in the coincidence frequency which shifts to the 3.15 kHz band. Approximating the Young’s modulus of the core component only with that of the foam material
(neglecting the stiffeners contribution, as done for the initial system), equation 1 gives $f_{\text{DoubleWall}} = 2.98$ kHz, which is again a good indication for the measured value.

It is notable to mention that due to the weight optimisation constraint included in the numerical optimisation problem, although the mass of the optimised design is less than the double of that of the starting system (exactly 1.75 times higher), the IL gain is more than 5 dB (i.e. prediction coming from the TL mass law) in the frequency bands from 160 Hz to 1000 Hz (except for the band at 200 Hz and 630 Hz, where it drops to about 4 dB).
5. CONCLUSIONS

A composite sandwich structure is tested and analysed to identify its dynamic and NVH behaviour. A finite element numerical model is built up on the base of nominal geometrical and material properties, provided by the manufacturer and partially found in literature. The model is coupled to a boundary element model of an irregularly shaped acoustic cavity representing the experimental test setup. The test rig allows fast acoustic insertion loss measurements. Good agreement is found between simulations and experiments, although no numerical updating and structural damping was performed. The model is then used in a structural optimisation in order to improve its stiffness, minimising the global weight and keeping satisfactory noise reduction performances. Using the same experimental and numerical approaches the acoustic IL of the optimised design is verified with good accuracy. The vibro-acoustic test rig also allows controlling the structural-acoustic coupling and location of the coincidence frequency. It is shown to be a valuable support for the design and optimisation of composite structures when NVH performance is a key design parameter.

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