FEASIBILITY OF THE NVH SIMULATION OF REPRESENTATIVE CFRP AUTOMOTIVE BODY STRUCTURES

T. Courtois[1*], O. Chesa[2*], R. Stelzer[1], D. Golombowski[3], M. Zogg[2]

[1] Autoneum Management AG, Schlosstalstrasse 43, CH-8406 Winterthur,
[2] Inspire AG / ICS, Technoparkstrasse 1, CH-8005 Zürich
[3] Dow Europe GmbH, Bachtobelstrasse 31, CH-8810 Horgen
*: both authors contributed equally to the results of this publication

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ABSTRACT

To reach the future CO_2 emission targets of cars, a significant weight reduction is required. As a consequence OEMs apply more and more lightweight carbon fibre reinforced polymer (CFRP) elements to their body in white structures [1]. Beside the weight and stiffness advantage, this introduction of light weight structures has a direct impact on the performance of vehicle bodies, with respect to requirements such as crash, handling, but also body NVH. Particularly the plurality of number of integrated materials, as well as the introduction of necessary bonding solutions at the interfaces will result in increasing complexity of the design of structure borne noise reduction solutions, such as damping. Therefore, it is necessary to establish the know-how to reliably and accurately simulate the acoustic vibration behaviour of multi-material lightweight vehicle structures [2-4]. For this, a research activity was performed in order to put in evidence and to simulate the structure borne noise performance of CFRP structures, with final demonstration on a representative vehicle floor mock-up. A material technology was selected based on a benchmarking activity. The necessary material properties were acquired with dedicated characterization methods. Then, the NVH performance of simple characteristic structures was validated against a correlated FEA model, taking into account several composite variants. The study includes the influence of adhesive bonding solutions, in order to take into account the realistic influence of this major and determinant variable in the integration of CFRP elements in vehicle bodies. Finally a couple of demonstration vehicle floor structures have been prototyped based on the learning of the characteristic structures. This mock-up is used to demonstrate that it is hence possible to apply the existing simulation methodology, together with appropriate material properties, to assess the NVH performance of a realistic vehicle component featured with CFRP elements and adhesive bonding boundary conditions.

1 INTRODUCTION

During vehicle development, the simulation by means of CAE is commonly used to support the reduction of development times, especially when the developed parts are concerned by a multitude a targets influencing each other. Moreover, the constant reduction of necessary vehicle prototypes during the development time, challenges the usage of CAE processes for decision making prior to the availability of hardware. In this context, a trend consists in applying the CAE during early phase, in order to provide design decisions as early as possible before design freeze, while taking into account various parameters, like structure borne and air borne targets, body panel stiffness, dampers location, acoustic packaging space as well as weight of insulators, and so on. In particular, body NVH design at low-medium frequency is traditionally carried out with respect to panel vibration targets [5], and the CAE processes based on FEA simulation must be able to calculate accurately the vibration level of the bare or trimmed body as response to a structural excitation, as represented figure 1. Based on this CAE capability, NVH part suppliers have developed numerous of efficient optimization and design methodologies, which support the cost and weight effective design of treatments, such as damping pads or noise reduction trims. For example, the methodologies referred in the following references are good examples of design engineering tools, which require the appropriate harmonic movement response prediction of the car structure [5-6].

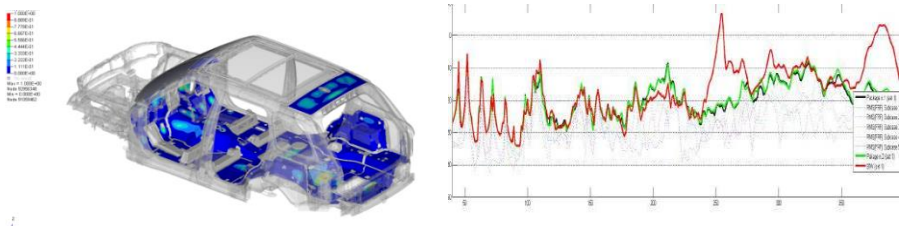


Figure 1: Simulated vibration response of vehicle body during development phase

The simulation of body vibration response must take into account all structural elements contributing to the transmission of the structure borne noise, including not only the primary panels, but also beams, reinforcements, moving elements such as doors. It is important to say that all those elements are often specifically designed to fulfil requirements like crash or handling, despite the fact that they all contribute to the final NVH result. In particular the introduction of light-weight CFRP elements aiming at reaching the CO_2 emission targets of cars by means of weight reduction generates a serious challenge for NVH CAE design engineers. Indeed, since a few decades, the car manufacturing industry considers the potential application of lightweight CFRP elements to their body in white structures [1]. More and more vehicles with primary structures made of CFRP are available today, starting from (very expensive) high performance sport cars to sporty cars in medium price segment, e.g. the Alfa Romeo 4C. Additionally some OEMs started to integrate CFRP elements into the metallic primary vehicle structure (e.g. Audi R8, Lamborghini Huracan, BMW 7-series) or have a FRP body on a metallic chassis (e.g. Chevrolet Corvette, BMW i3, BMW i8). The majority of these primary vehicle structures or structural FRP components and bodies is made of carbon fibres with epoxy matrix systems today. The fibres typically are locally arranged in roving leading to typical fibre volume contents of 40 - 60 % and therefore they have a good lightweight potential and can be oriented according to load requirements. The design of these structures and components contains monolithic, hollow and sandwich elements.

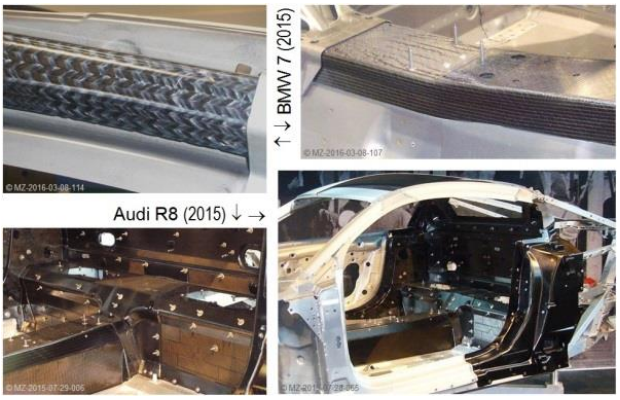


Figure 2: Example of CFRP elements bonded to the metal primary structure

Beside the weight and stiffness advantage, this introduction of light weight structures has a direct impact on the performance of vehicle bodies, with respect to requirements such as crash, handling, but also body NVH. Particularly the plurality of number of integrated materials, as well as the introduction of necessary bonding solution at the interfaces will result in increasing complexity of the design of low frequency noise reduction solutions, such as damping. Therefore, the purpose of this article is to demonstrate that it is possible to take into account in a realistic way the CFRP elements in the FEA simulation of automotive structure vibration, which is the necessary condition to be able to define and design the corresponding best NVH countermeasures. This demonstration is made by selecting a representative CFRP technology, then performing its characterization for the definition of the relevant material properties, then obtaining the correlation on a representative test rig featuring characteristic structures, and finally by applying the methodology in a realistic vehicle floor component.

2 MATERIAL CHARACTERIZATION

2.1 The selected FRP composite

This paragraph describes a methodology to obtain all the required material data in order to simulate the vibration behaviour of CFRP materials. Those materials can typically be modelled as orthotropic shell elements (except for the core in sandwich constructions), so only the in-plane material properties are taken into account. The CFRP material was prototyped by Inspire ICS and the later required adhesive interfaces for the integration of the structures were prototyped at the Bonding and the "R&D and TS&D Center" of DOW Automotive Adhesive Systems Switzerland. The selected technology consists in a wet lamination process, were the fibres are saturated by an epoxy resin mixed to a hardener, and the application of a specific process associating vacuum and accelerated curing at temperature. The composites consist in three different layups. The first one embeds 4 layers of a woven fabric of 445 $^{g}/_{m^2}$; the second embeds 8 layers of Unidirectional (UD) fabric of 230 $^{g}/_{m^2}$, and the third is a sandwich with CFRP facings (2 layers of a woven fabric of 445 $^{g}/_{m^2}$ each) and a PET foam core of density 100 kg/m^3. Before beginning with the material characterization, a reproducibility study was performed in order to verify the robustness of the manufacturing process and to quantify the possible deviations of the material properties. Thickness and fibre volume content as well as bending properties were compared. When comparing the three different plates, deviations were below 8.5%.
The material characterization is done in a way, so that the composite can be represented by means of homogenised shell elements. For that, the static properties focus on bending and membrane characteristics. The tests used are tensile and bending tests according to the standards EN 2561/2562, and in-plane shear tests according to standard DIN 65466. For the sandwich characterization, one may characterize the layers (facing and core) separately. However, due to the difficulty of representing correctly the core shear, it was made used of an equivalent method based on a 3-point bending test (Fig. 3) of the sandwich sample, where the tensile modulus of the facings and the shear modulus of the core are calculated from the stiffness of sandwich specimen tested with 2 different support lengths in a 3-point flexural bending test, based on the following equation calculating the deflection W of a sandwich specimen at a certain load P:

$$W_{(x = l/2)} = \frac{P \cdot l^3}{24 \cdot b \cdot E_f \cdot t_f \cdot d^2} + \frac{P \cdot l \cdot t_c}{4 \cdot b \cdot G_c \cdot d^2} \tag{1}$$

where E_f is the unknown Young's Modulus of the facing, G_c the Shear Modulus of the core.

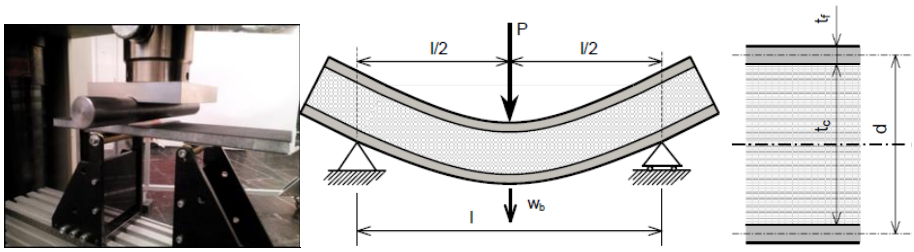
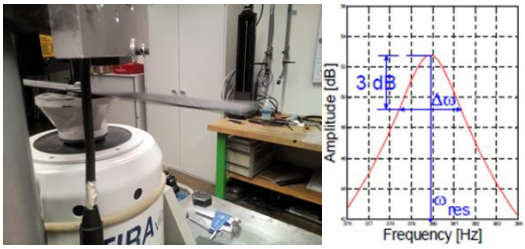


Figure 3: 3-point flexural bending test based on standard EN 63 for the determination of bending and shear sandwich properties

Those static properties are meant to be used later on for dynamic harmonic displacement simulations. Due to the important role played by dynamic properties for the NVH performance [7-8], a specific methodology based on the measurement of modified horizontal Oberst [9] beam method has been exploited for the definition of composite damping factor For this, a horizontal beam sample of the material is mounted on a shaker, and the damping is evaluated based on resonance bandwidth of the displacement frequency response function of the excited beam. See setup figure 4. Table 1 shows the summarized properties for the woven and UD samples, summarizing all the necessary inputs data for FEA simulations.



$$\eta = \frac{\Delta\omega}{2\omega_n}$$

Figure 4: Damping measurement on resonant horizontal beam setup

Properties		Woven	Unidirectional	Sandwich Core
Tensile E_1	[GPa]	54.0	94.3	29.3
Tensile E_2	[GPa]	52.1	4.9	–
Bending E_1	[GPa]	39.2	80.6	–
Bending E_2	[GPa]	40.0	4.6	–
ν_{13}	–	0.10	0.333	*0.45**
ν_{23}	–	0.17	–	–
Tensile G_{12}	[GPa]	2.3	2.0	–
Bending G_{12}	[GPa]	2.9	4.0	–
G_{13}	[GPa]	–	–	26.8
ρ	[kg/m^3]	1273	1279	106.5
η_1 (res.1/res.2)	–	0.018/0.005	0.012/0.003	
η_2 (res.1/res.2)	–	0.02/0.006	0.025/0.015	

Table 1: Mechanical properties of the CFRP material for simulation

2.2 The selected bonding adhesive

Regarding the bonding, the industrial suppliers are currently making efforts at differentiating adhesive technological product solutions adapted to the challenge of multi-material interfaces such as the combination of aluminium and CFRP. Such adhesive must be taken into account in the evaluation of NVH performance of composite structures, due to the fact that those adhesives must primarily fulfil severe non-NVH requirements in terms of temperature sustainability, crash resistance, elongation requirements, and manufacturing application tolerance, making it important to determine their properties, and create awareness of what is their influence or potential on the final system NVH performance. The selected adhesive for this study is 2-component polyurethane adhesive with fast curing rate at room temperature, specially developed for structural bonding. It is a perfect compromise for the tested case, as it features a good adhesion to composites and painted surfaces as well as to coated metal surfaces, it has a high mechanical strength and elongation at break, it has a low temperature dependency of the modulus and a glass transition temperature outside the application range, and features a very long open time allowing easy bonding at prototype level. The mechanical characterization results will also show that it features a rather good damping, which is predisposing it to NVH applications. The mechanical characterization of glues is a complex domain, involving different rheological tests. The choice has been done here for a test that reproduces the behaviour of the glue, in the same conditions as when applied at the boundary of a flexural flat structure bounded on its contour. In this case the properties of the glue are assimilated to an isotropic elastic material. This approach reduces the material characterization to a Young's Modulus, a damping factor (assumed Poisson Ratio). However, this limits the usage of the identified material model to limited assembly conditions. Figure 6 shows the test, a dynamic DMA 3-point bending test (figure 6.b), in which the glue is loaded in similar conditions as in assembly. To identify the unknown elastic properties of the glue in such conditions, a genetic optimization procedure is applied to a FE model representing the test setup (figure 7), using both the tested displacement and the measured $tan\delta$ (where δ is the stress-strain phase lag) as targets. Finally, figure 8 shows the identified damping and stiffness values of the glue in

function of the frequency. It is worth commenting that the damping value of the glue appears to be rather high at about 20%, for a rather high stiffness value having a Young's Modulus at around 2500 MPa.

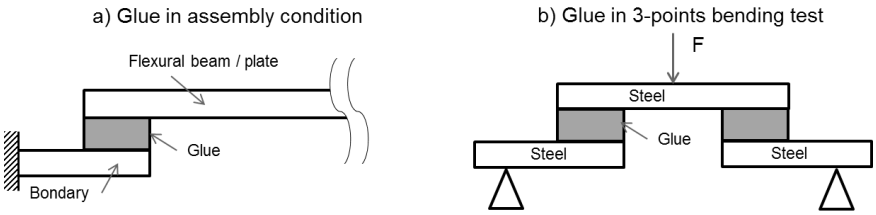


Figure 6: Sketch of 3-point bending test to characterize the glue in same conditions as in assembly

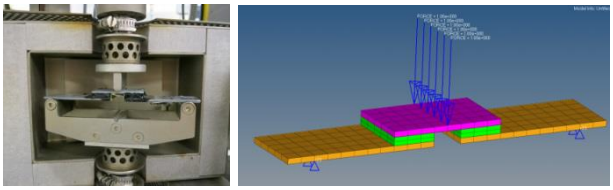


Figure 7: DMA test and FE model used to identify the equivalent properties of the glue

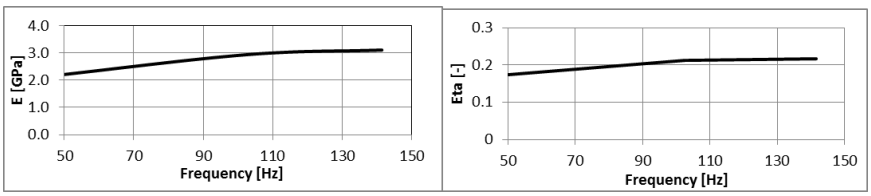


Figure 8: Equivalent elastic properties of the glue, identified in assembly conditions

3 STUDY BASED ON ACADEMIC TEST SETUP

3.1 Setup and correlation of the FE model

The material properties are used for the definition of composite FE shell elements, aiming at performing the necessary NVH simulations of structures made of CFRP. The software used for the simulation is NX NASTRAN. For the woven "monolithic" shell structure made of 4 layers of fabric, resulting into a 1.84 mm total thickness, PSHELL and MAT8 cards are used. For the unidirectional (UD) "monolithic" shell structure made of 8 layers of fabric, resulting to 1.92 mm total thickness, and symmetric lay-up *(0/+45/-45/0)s*, PCOMP and MAT8 cards are used. For the sandwich structure made of woven facings of 1 mm each, and a PET foam core of 10 mm resulting to 12 mm total thickness, a PCOMP and MAT8 were used, with orthotropic PET core. After a first static validation of the modeling on 3-point bending plates, a classical suspended free plate vibration response was used to correlate the model. Table 2 shows the results of the 3-point bending test with satisfactory static correlation against FE simulation.

Tested metrics		Woven		UD		Sandwich	
		Meas.	Simu.	Meas.	Simu.	Meas.	Simu.
Static force	*[N]*	*100*		*150*		*500*	
Deflection (0°)	[mm]	23.7	23.9	22.5	21.7	322.9	318.7
Static force	*[N]*	*100*		*35*		*600*	
Deflection (90°)	[mm]	9.3	9.4	9.8	9.4	603.0	636.9

Table 2: Results of the static 3-point bending plate correlation test (plate dimension is 600 x 480mm)

Figure 9 shows the setup of the free-free plate. The plates were excited dynamically with a shaker, with a sweep signal from 10 to 400Hz. An impedance head mounted between the shaker and the plate measured the input force. The vibration response of the plate was measured with a laser at 49 points equally distributed on the plate surface. The correlation was performed on the modal analysis, as well as on the *rms* average of the 49 points mobility frequency response function of the plate. Table 3 shows the results of the free-free plate test with satisfactory dynamic correlation against FE simulation. This is a good transition from static correlation to dynamic correlation.

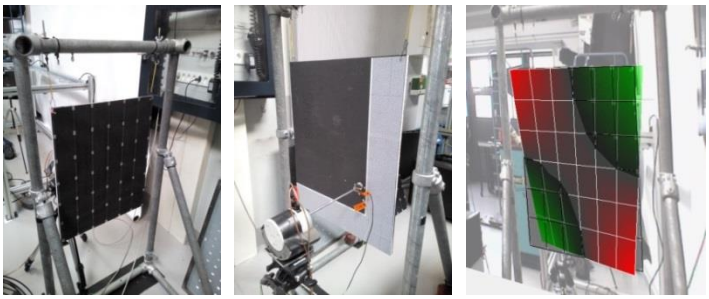


Figure 9: Dynamic free-free plate correlation test (plate dimension is 600 x 480mm)

Resonance Modes		Woven		UD		Sandwich	
		Meas.	Simu.	Meas.	Simu.	Meas.	Simu.
Mode 1		16.5	10.4	15.3	15.8	89.0	84.2
Mode 2		30.7	29.4	22.8	21.1	233.2	214.0
Mode 3		40.0	36.3	41.3	40.6	286.7	258.6
Mode 4		48.7	45.9	47.2	50.1	328.3	325.6
Mode 5	[Hz]	53.9	50.2	49.1	58.0	361.4	351.9
Mode 6		–	70	71.4	71.1	–	–
Mode 7		80.9	81.6	82.2	73.2	–	–
Mode 8		90.6	87.1	91.4	94.3	–	–
Mode 9		119.4	114.9	102.7	105.8	–	–
Mode 10		130.9	126.0	118.3	116.6	–	–

Table 3: Results of the dynamic free-free plate correlation test (plate dimension is 600 x 480mm)

3.2 Validation of NVH design feasibility on characteristic structures

The usage of CFRP materials in the design of automotive structures involves wide design options for the introduction of panel structures in the body architecture. A benchmarking study carried out on vehicles currently on the market, as well as the observation of current trends, have shown that monolithic, hollow and sandwich structures can be used. In some cases, the vehicle dynamic or crash requirements lead to the insertion of reinforcement beams or specific elements with associated specific properties in the panel construction. The combination of different materials in a single structure involves also the introduction of mechanical interfaces between them, such as bonding by means of adhesives. All those different solutions are expected to have a direct impact on the dynamic behavior. Therefore the NVH performance of flat panel based structures, namely characteristic structures, with a selection of different relevant design features was studied, aiming at demonstrating that they can be taken into account in the FE simulation. This validation process is necessary if ones want to successfully exploit FE simulation methodologies for the performance and design of body NVH countermeasures. The testing setup used for the NVH performance test (shown figure 10) was consisting in a plate, clamped on its boundary to a rigid structure, such as representing the idealized conditions of a vehicle body panel stiffened on its contour by thicker or shaped elements. The clamping frame was designed so that the first eigen frequency was above 1000 Hz. The structure is excited by a shaker at one point on the surface of the plate. The panel vibration response is measured by means of a vibro-meter laser.

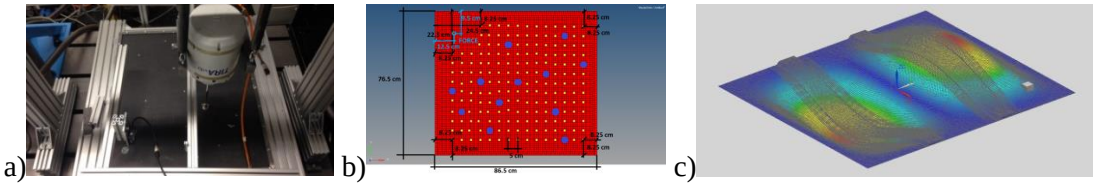


Figure 10: Characteristic structure panel test setup (dimension 760 x 860mm)
a) Photo - b) Excitation and measurement positions - c) Vibration test

Table 4 summarizes all the different characteristic structures considered in this activity. The reference structure is a steel plate of 1mm thickness. Then the monolithic structure, corresponding to a typical flat vehicle floor panel, is made of 6 layers of woven carbon fibers (each 445 $^{g}/_{m^2}$fiber area weight) for a total thickness of 2.7 mm. The sandwich structure corresponds to a nearly-flat sandwich floor, such as the floor solution adopted by Lamborghini in the Aventador model. It is made of face-sheets manufactured with two layers of woven carbon fiber (each 445 $^{g}/_{m^2}$fiber area weight) for a total of 1 mm each facing, and a 10 mm thick PET foam core with a density of 100 $^{kg}/_{m^3}$. The beam-reinforced structure is a combination of monolithic construction made of 4 layers of woven carbon fibers for a total thickness of 2 mm, together with omega profiles made as a sandwich construction. Finally, the last configuration corresponds to a solution inspired from BMW and adopted on the model i3. It consists in a flat carbon panel with bonded aluminum beam reinforcement, where some elements of the vehicle are fixed, such as the seats. It is made of a combination of monolithic construction made of 4 layers of woven carbon fibers for a total thickness of 2 mm, together with hollow profiles made of 1 mm thick aluminum, filled with PET foam. Moreover, the bonded configuration corresponds to the monolithic configuration, where the clamping condition is replaced by a 2-3 mm thick and 15 mm wide adhesive glue line. This represents the realistic influence of one major and determinant variable playing an important role in the integration of CFRP elements in vehicle bodies.

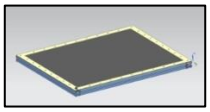
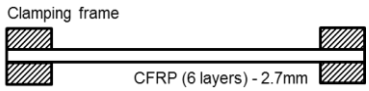
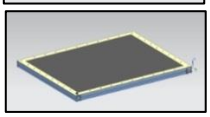
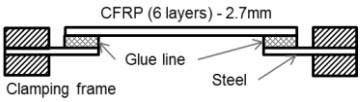
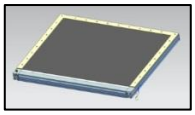
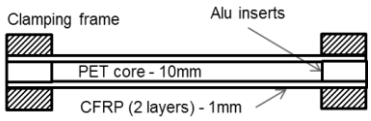
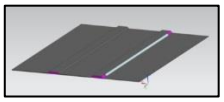
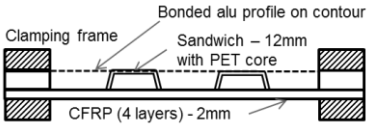
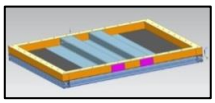
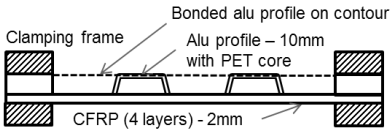
Characteristic Structure	Picture	Aw [kg/m^2] *@iso-area*	Sketch
Steel		7.8	
Monolithic		3.4	
Monolithic *Bonded*		3.45	
Sandwich		3.9	
Beam-reinforced		3.5	
Alu-reinforced		4.2	

Table 4: Overview of the different characteristic structures

Each of the characteristic structures have been simulated by means of FEA, and validated against the

test. Figure 11 shows the validated *rms* averaged mobility frequency response function acquired on the respective panels. The correlation of the steel plate shows the level of correlation that can be obtained for a conventional structure. Such correlation is known to be difficult to be obtained, when the structure is not damped. It is interesting to see that the CFRP monolithic structure leads to a better correlation than with steel, due to its lower modal density. The correlation is even better, in the configuration where the structure is bonded by means of adhesive, for a simple reason: without glue, it is difficult to obtain a perfect clamping at experimental level, while the boundary with the glue is realized experimentally in a way that the boundary condition can be easily reproduced. The correlation in the case of the sandwich is not satisfactory. The correct representation of the complex interface between the sandwich and the clamping frame at its boundary, in the condition of the experiment, would require more investigation. In the case of beam and aluminum reinforced structures, the mobility is strongly driven by the beams stiffness, leading to a very low modal density. Figure 11, the common y-axis allows an interesting comparison of the levels between the different structures, where we can see that CFRP structures are not necessarily worse (higher) than steel reference.

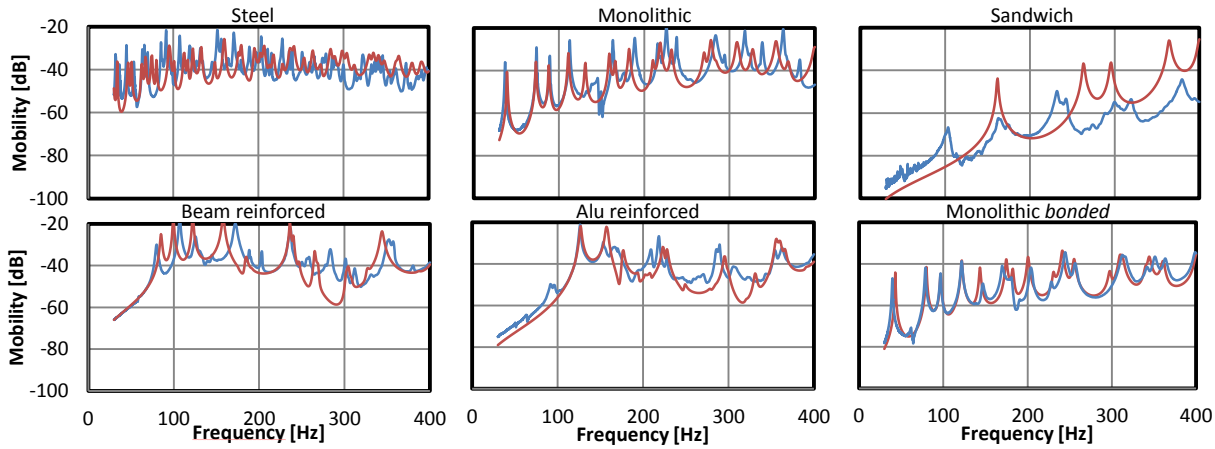


Figure 11: Mobility frequency response functions of the characteristic structures (in m/s/N (dB))

In a later stage, we show that the Finite Element modeling can be exploited is a similar way as in the case of metal structures, in order to take relevant decisions for the application of vibration reduction countermeasures. In the following example, a methodology based on Finite Element simulation, is applied to the monolithic CFRP structure to identify the optimal location of damping pads. Such damping pads are a conventional solution used in body vehicle NVH to reduce the structure borne noise of vehicles due to vibration transmission. In particular, we are considering here a light weight (3.2 kg/m^2) high performance (η = 0.34) constrained layer damping made of 1.5 mm of bitumen layer covered by a 0.25 mm aluminum foil. The methodology [6] consists in calculating on a map the contribution of each location of the structure to the peaks of the averaged mobility. This leads to the identification of the location, where the application of the damping is the most efficient. Figure 12 shows the calculated color-map, and the corresponding engineered damping design, and figure 13 shows the simulated versus measured vibration reduction of the damping application (in what follows, the simulations are done using ACTRAN). We validate in this way the feasibility of a conventional NVH engineering method applied on a non-conventional CFRP structure.

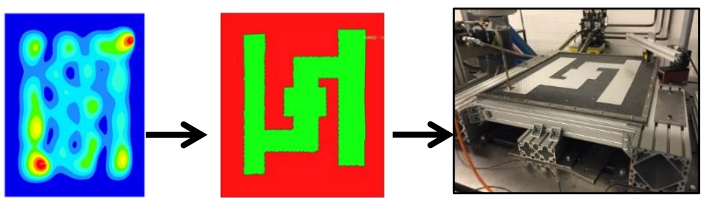


Figure 12: Application of Finite Element based engineering methodology for application of damping on CFRP monolithic structure. Damping weight = 0.7kg (plate weight = 2kg)

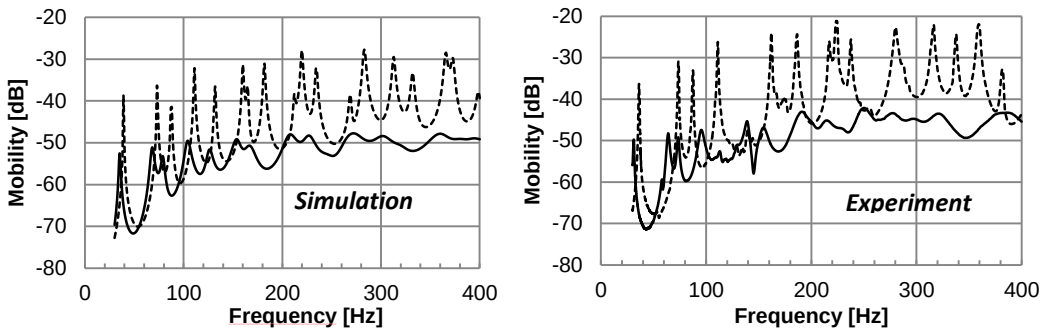


Figure 13: Validation of the designed damping (0.7kg) on CFRP monolithic structure
—— Plate with damping; ····· Bare plate ; Mobility in m/s/N (dB)

An additional experiment is performed in order to compare the efficiency of such damping when applied on the CFRP structure compared to application on conventional steel. In that case, we use a different design of the damping pads as in the previous example, so that it is potentially efficient for both the CFRP structure (monolithic) and the 1 mm thick steel structure. Figure 14 shows the simulated versus measured vibration reduction induced by the damping application on both the monolithic CFRP and the steel structures.

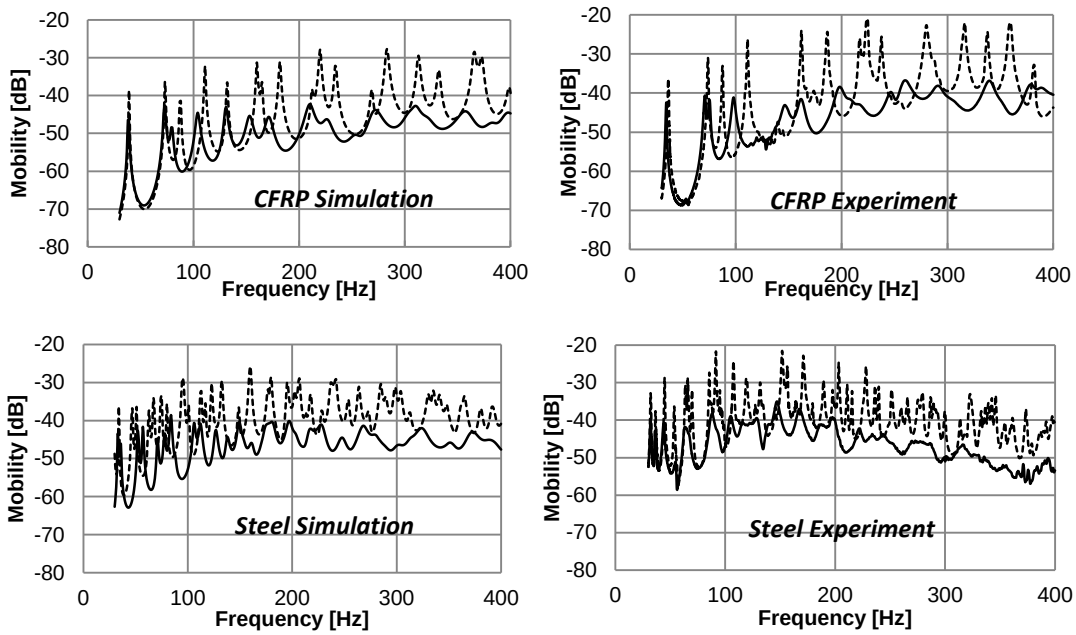


Figure 14: Validation of the designed damping (0.4kg) on CFRP monolithic plate
—— Plate with damping; ····· Bare plate ; Mobility in m/s/N (dB)
Weight steel structure Bare = 4.6kg ; Damped = 5.0kg (+9%)
Weight CFRP structure Bare = 2.0kg ; Damped = 2.4kg (+21%)

In the case of the steel structure, the peaks of the mobility spectrum are reduced of about 15 dB according to measurement (10 dB according to simulation). In the case of the CFRP plate, the peaks of the mobility spectrum are reduced of up to 20 dB according to measurement (15 dB according to simulation). This might lead to the conclusion that the damping is more efficient on the CFRP than on the steel structure. At the same time, it must be observed that the modal density and therefore the number of peaks to be damped in the case of CFRP is lower than in the case of steel. Moreover, the relative weight increase due to the application of damping in the CFRP case is more than 20%, while it is less than 10% in the steel case.

4 APPLICATION ON VEHICLE MOCKUP

Finally a couple of demonstration vehicle floor structures have been prototyped based on the learning of the characteristic structures. This mock-up is used to demonstrate that it is hence possible to apply the existing simulation methodology, together with appropriate material properties, to assess the NVH performance of a realistic vehicle component featured with CFRP elements and adhesive bonding boundary conditions. For this, the body in white of three recent C-segment vehicles have been cut-out at the level of the pillars, and prepared in the following way. One vehicle has been kept in original steel conditions. The second vehicle had the right floor panel cut-out and substituted by a CFRP monolithic panel of the same shape as the original one. The third vehicle had the entire tunnel, as well as one transversal beam substituted by a shaped CFRP monolithic panel. The CFRP material used was of the same construction as the monolithic panel made of 6 woven carbon fiber layers, as in the previous section. Figure 15 shows the two prototyped configurations, respectively called the “tunnel” case, and the “floor” case. The steel configuration is called “reference”.

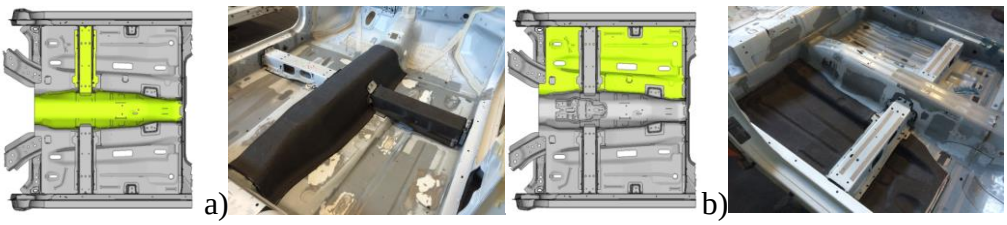


Figure 15: Floor mockup featuring respectively tunnel / transverse beam and flat panel CFRP structure
a) “Tunnel” case: 3.4 kg steel replaced by 2.5 kg CFRP and 0.3 kg glue
b) “Floor” case: 2.5 kg steel replaced by 1.8 kg CFRP and 0.4 kg glue

In both cases, the CFRP panel has been bonded to the rest of the steel structure by means of the adhesive glue described in the previous sections. The application of the CFRP panel has been done in the same way as it was in the case of the characteristic structure. The obtained structure was instrumented in the same way as it is done for vehicle BIW mobility measurements, like it can be done for example during body NVH development. The structure is suspended by decoupling springs (Fig. 16), and excited by means of two different shakers ; one shaker at a front right suspension point, and one shaker at a rear right sub-frame connection point, such representing the vibrational input at realistic locations. The vibration response of the structure is measured by means of a vibro-meter laser.

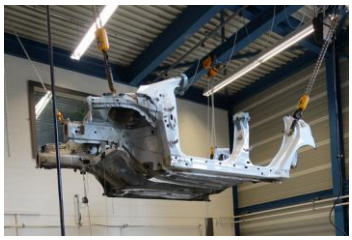


Figure 16: Floor mockup in testing condition

In parallel to the measurement, a FEM model of the vehicle is used for the simulation of the complete floor mockup. Three variants of the model have been created, corresponding to the respective reference, tunnel case and floor case prototyped variants. The FEM model is shown Figure 17. It is mainly made of shell elements. The CFRP shell properties are the same as the one of the “monolithic” characteristic sample. The bonding glue is modeled by means of 3D elements, and the software used for the simulation is ACTRAN. The simulation is based on the material modeling and on the NVH FEM practice applied previously on the characteristic structures. The validity of the simulation for NVH purpose, consists in observing against testing, the prediction of the alteration of the mobility between the reference case, and respectively the “floor” and “tunnel” cases. For this purpose, figure 18 shows the simulated CFRP versus Steel panel mobility, as well as the measured CFRP versus Steel

panel mobility. The mobility is averaged on the area, where the steel has been substituted to CFRP (respectively the dark areas on figure 17).

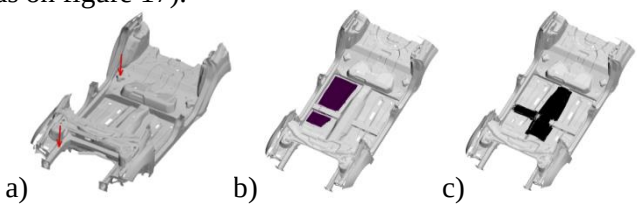


Figure 17: FEM model of the floor variants; a) Reference – b) Floor case – c) Tunnel case

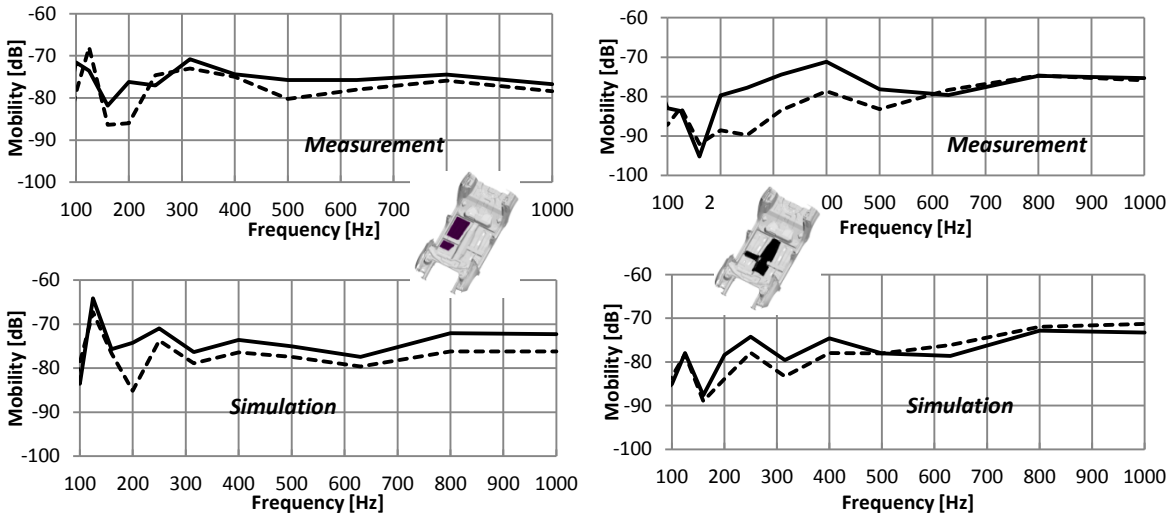


Figure 18: Vibration (3rd octave average) of the CFRP structure against steel structure
Reference (steel) structure ; - - -CFRP structure ; Mobility in m/s/N (dB)

Figure 18 shows that the simulation can fairly predict the effect on the vibration response of the introduction of a CFRP structure in a steel structure. Some discrepancies exist here between absolute simulated and measured levels, which can be explained by some structural details introduced by the prototyping operations, which are not represented in the simulations. Nevertheless, the simulation of body mobility is able to catch the overall structural change, which is the prerequisite for CAE designed based body NVH engineering.

4 CONCLUSION AND OUTLOOK

In this paper, it was possible to demonstrate the validity of the simulation of vehicle floor mobility to be used for body NVH, in the case where the vehicle structure integrates CFRP elements or components. Based on such simulation, it is indeed possible to design, in the same way as it is done for conventional metal vehicle bodies, the optimal damping, or even the optimal acoustic treatment, such as carpet silencer, by coupling poro-elastic FE methodologies to the body FEM [10]. To illustrate this, the last figure (Fig.19) shows the measured effect on body vibration due to the application of a carpet insulator prototype onto the floor mockup. The evaluation of such effect of the trim insulator on the body is typically part of the structure-borne NVH design, which suppliers of acoustic part or car manufacturers must be able to take into account during vehicle and part development. More precisely, we see in this experimental example that the application of the trim on the CFRP vehicle structure modifies the body vibration level in a different way, than when the trim is applied on the reference steel vehicle structure. This is highlighting the importance to dispose of body NVH simulation method, which must be valid for specific CFRP body scenarios.
Beyond this, the integration of the properties of the bonding adhesive in the simulation model structure including CFRP elements opens the door to the definition of appropriate targeted glue properties for an optimal NVH performance.

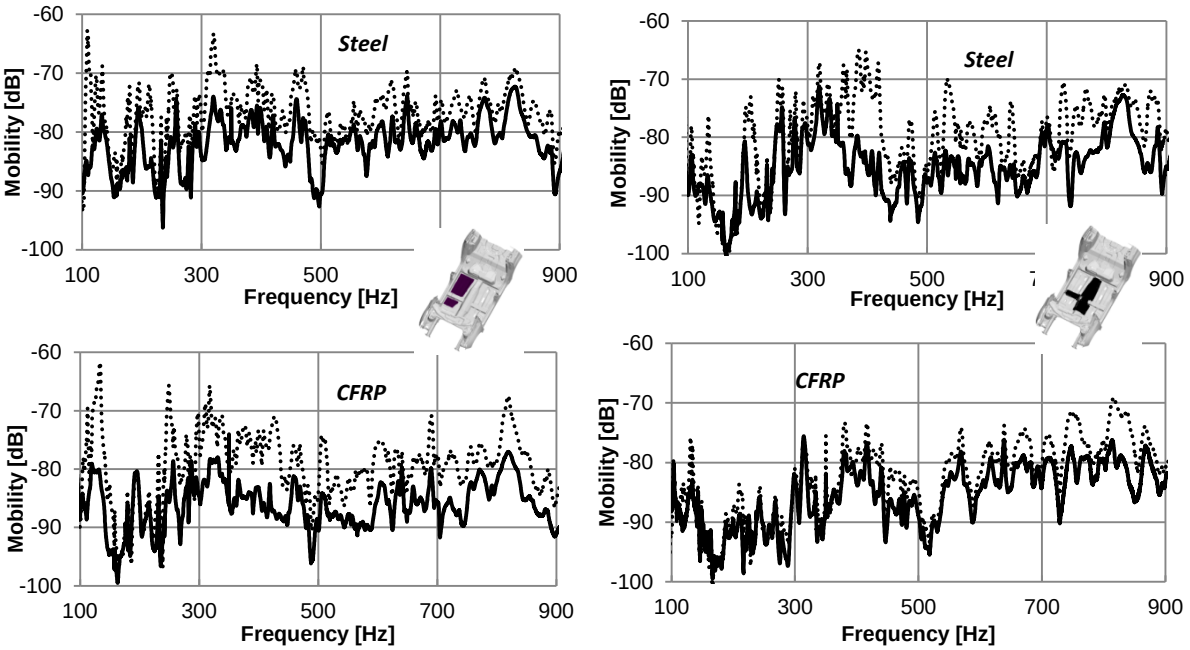


Figure 19: Vibration of the vehicle body in bare and trimmed conditions
——— trimmed (foam and 3kg heavy-layer) ; ····· bare; Mobility in m/s/N (dB)

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