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MODAL ANALYSIS OF A COMPOSITE SANDWICH PANEL USED IN THE STRUCTURE OF AN HYBRID BUS

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Summary. *The rear module of an hybrid bus was redesigned in a joint effort by Volvo and IDMEC – in the framework of the european project HCV – Hybrid Commercial Vehicle (Grant Agreement No 234019)–, replacing most of the metallic frame and other components with composite sandwich panels. The redesigned bus overall weight was reduced by approximately 900kg. This paper presents the numerical and experimental modal analysis of a sandwich panel that was produced using the same materials and production process used to produce the main prototype rear component.*

1. INTRODUCTION

The HCV project is an European Collaborative Project (Large Scale Integrating Project) with a duration of 48 month, from January 1st, 2010 to December 31st, 2013, approved under the 7th Framework Programme. The HCV project aims to develop urban buses and delivery vehicles with advanced second generation of energy efficient hybrid electric power-trains in line with objectives in topic SST 2008.3.1.5 - Urban buses and delivery vehicles using second generation hybrid electric technology. The final result will be the demonstration of a passenger bus and a distribution truck with this advanced technology.

An 18-ton passenger bus demonstrator and a 6-ton distribution truck demonstrator will be fully developed in the project. A low weight body will be demonstrated for the 18-ton bus and some weight optimisation will be made on the distribution truck. The hybrid electric concepts in the HCV project aims to further integrate components and to improve the hybrid system with new gearboxes, electrification of auxiliaries and advanced control system. A total of 12 demonstrators will be exposed within the project. New advanced technologies will be tested and made public across Europe from east to west and north to south in a variation of climates and seasons.

The rear module of an hybrid bus was redesigned, produced and assembled into a prototype, replacing most of the metallic frame and other components including panels with composite sandwich panels. The chassis was modified to fit the sandwich panels and to allow further weight reduction. Figure 1 shows the final configuration of the sandwich panels.

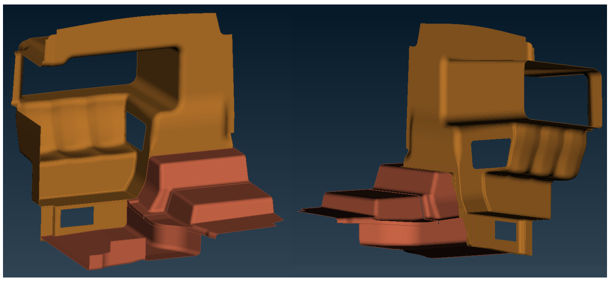

Figure 1. Sandwich panels in the rear module.

Other parts of the structure were also redesigned by Volvo, using metal and foam based sandwich structures and other materials, although this particular part of the work is not addressed in this paper. For the rear module, four independent composite sandwich panels were produced by vacuum infusion: the largest rearmost panel, two panels which received two seats each and a floor panel just in front of the rear door. The largest sandwich panel replaces all transversal structural elements in the rear module and supports totally three seating passengers and several equipments and systems (exhaust system and all equipments at roof level, which also introduce thermal and vibration loading) and to a great extent the longitudinal beams where the engine is mounted and the cooler.

Every proposal of modified bus must satisfy specified axel loads, which were verified subjecting the whole structure to unit gravity in the vertical direction.

The overall weight reduction was approximately 900kg, 80% above the initial objective. This reduction was confirmed by weighting the prototype.

It was calculated using the numerical model that, without inserts, the main rear sandwich component weights 62.3kg, and the floor panel and the two podesters, together, weight 25.8kg. effective reduction of weight of the structure was thus estimated to be 308kg (taking only the rear module modifications into account). Figure 2 shows the prototype main sandwich panel soon after being demoulded.

One important achievement is that the panels are produced in one single process and are directly assembled into a more simplified structure. The accuracy in the production of the sandwich panels proved to be adequate for a direct assembly without the need of forced adjustments. It also represents the abolition of a number of components which were integrated into a single one and the consequent reduction of assembly operations and time consumed.

Last but not least, despite the experience of the workforce involved in the production of the panel, margin still exists for process optimization. When adopted for industrial production, several improvements and more detailed planning will contribute to an increased efficiency.

Figure 2. Main sandwich panel finished

Nevertheless, the implemented prototype process proved competitive given the mentioned integration and assembly simplification.

The technology and experience are currently being used by Volvo and IDMEC in the formulation of an alternative bus concept in HCV, where conventional steel frames are replaced by structural sandwich to an even larger extent. Furthermore, the number of passengers seating in the new rear sandwich component is increased: in the current stage, 9 passenger seats are considered. The design is also complicated by the fact that the energy cells for a potential variant with electric drive should be possible on the roof, close to the rear door, and several suspension components are directly mounted onto the composite sandwich.

Prior to the production of the sandwich components, a plate was produced for testing. Although the materials used were the same, the lay-up is of the sample sandwich is different. Tests were performed by IDMEC to measure the properties of the real laminates to assess the veracity of the numerical model used in the design of the rear module. The experimental modal analysis tests were performed at the Laboratory of Vibrations of the Departement of Mechanical Engineering, FEUP. The complete report of the experiments is presented in this paper.

The plate is 522mm long and 142mm wide. The lay-up was assessed by microscopy and the thickness of the individual plies was averaged out from several measurements at different locations. The lay-up considered representative of the real sandwich and used in the numerical models is:

0.35mm of carbon fibre fabric at 0/90° (0° is the longitudinal direction of the plate)

0.30mm of unidirectional carbon fibre at 0°

0.35mm of carbon fibre fabric at 0/90°

29mm of PVC foam core (40kg/m^3)

0.40mm of carbon fibre fabric at 0/90°

0.30mm of unidirectional carbon fibre at 0°

0.40mm of carbon fibre fabric at 0/90°

The differences in ply thickness on both skins were considered to be due to slight differences in volume content of fibres in the laminates. The material properties used in the numerical model were not changed to account for this difference.

The composite sandwich plate received from Volvo for testing is shown in Figure 3.

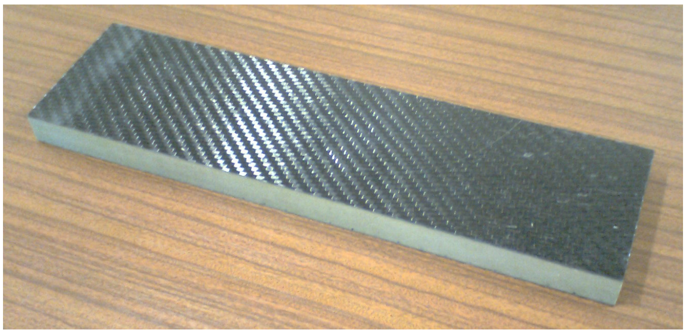

Figure 3. Plate cut out from rear sandwich component

An undamped model was used.

2. PRELIMINARY MEASUREMENTS

Preliminary measurements were made to determine the natural frequencies and mode shapes in free-free boundary condition. To materialize this condition, the plate was supported by two soft foam pads which were expected to significantly reduce the upper limit value of the frequency range containing the rigid body modes of the assembly, while minimizing the external ground noise effects.

To obtain some frequency response functions of the plate an ICP accelerometer (*Brüel & Kjaer* 4507) was attached to one of the faces of the plate using special purpose mounting beewax and an instrumented impact hammer (*Brüel & Kjaer* 8202) was applied to provide the excitation in several locations of the same skin.

Both transducers were connected to a dynamic signal analyser for signals conditioning and analysis using a *Brüel & Kjaer* 2035 spectral analyzer. The signal analysis was performed inside the spectral analyzer, providing the set of frequency response functions of type accelerance for a coarse measuring mesh considered in this study. The set-up is shown in Figure 4.

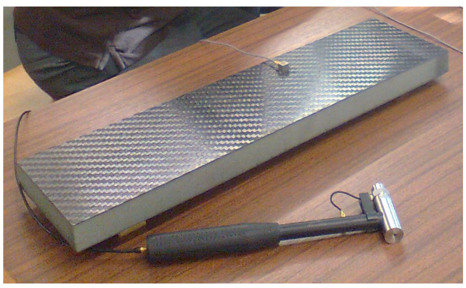

Figure 4. Preliminary measurements set-up

In parallel, the natural frequencies and mode shapes were extracted from a numerical model using Abaqus. The model consists of a flat rectangular shell which was modelled using the same element types (S4), material properties (for the carbon fabric) and section lay-up(except for ply thicknesses, which were modified to correspond to the case of the tested sandwich) as used when modelling the rear sandwich in the numerical model of the complete bus used in the design phase. The layers, the thicknesses and the material properties are not the same in the tested plate and in the sandwich components produced for the prototype bus.

These preliminary measurements and studies were concluded with the preparation of the next measurements: the nodal lines of the different modes of vibration were mapped and overlapped in a single map to allow the correct positioning of the accelerometer. This map of nodal lines is shown overlapped by the mesh of the plate in Figure 5. This procedure allowed for the identification of mesh nodes where the accelerometer could be positioned and the respective coordinates were taken to be used in the measurements. It should be noticed that the dark areas correspond to areas with small normalised diplacement (forcedly including the nodal lines) which were avoided for the positioning of the accelerometer.

Figure 5. Nodal lines map overlapped by the mesh of the plate

Figure 6 shows the magnitude of the frequency response function (FRF) determined directly on the excitation node – one of the nodes identified using the nodal lines map.

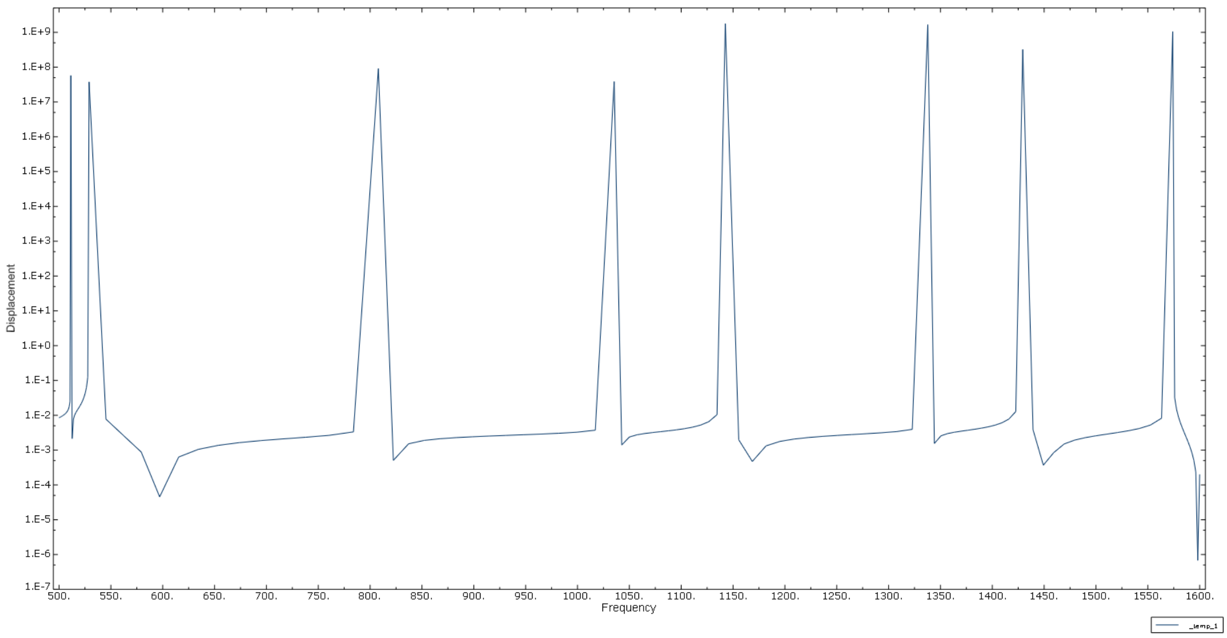

Figure 6. direct FRF obtained with the model

3. EXPERIMENTAL MODAL ANALYSIS

3.1 Experiment using an instrumented impact hammer

An experimental modal analysis was performed on the same plate. Using the data from the previous analysis a measurement mesh with 55 points was defined as illustrated in Figure 7.

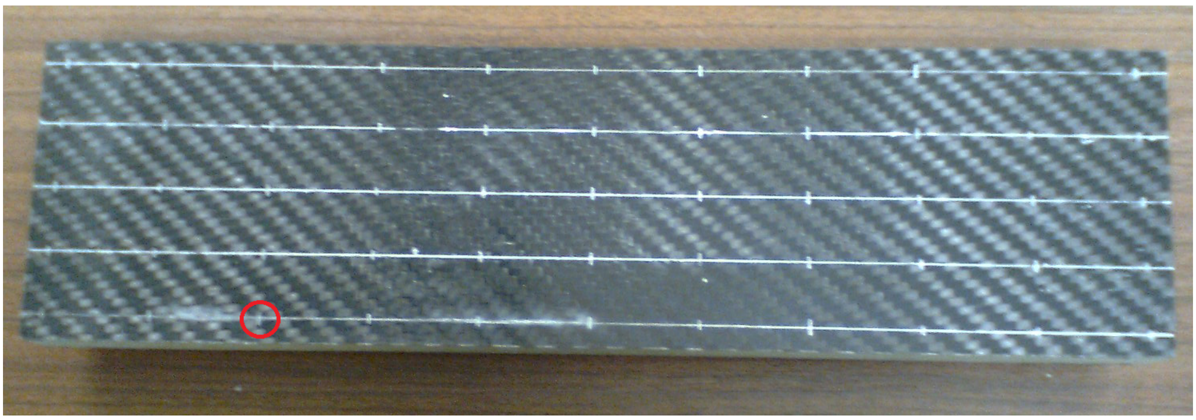

Figure 7. Grid marked on the plate

Like in the preliminary measurements, to materialize free-free boundary condition, the plate was supported by two soft foam pads. An ICP accelerometer (*Brüel & Kjaer* 4507) was attached to the face of the plate opposite to the measurement mesh using special purpose mounting beewax, in a location just opposite to the mesh node 3, marked with a red circle in Figure 7, and an instrumented impact hammer (*Brüel & Kjaer* 8202) was applied to provide the excitation at each point of the measurement grid.

Again, both transducers were connected to a dynamic signal analyser for signals conditioning and analysis using a *Brüel & Kjaer* 2035 spectral analyzer. The signal analysis was performed inside the spectral analyzer, providing the set of frequency response functions of type accelerance for the entire measuring mesh considered in this study. The measurements were made in the frequency range 0–1600Hz. The set-up is identical to the one shown in Figure 4.

From the measured FRFs, in a first stage, a multi-degree of freedom (MDOF) technique, which uses the ”least squares complex exponential” time domain algorithm, eas applied in order to identify the natural frequencies and damping ratios values for each mode.

In a second stage, the residues were identified with a ”least squares frequency domain” technique and the mode shapes were obtained. The modal parameters were identified by using the modal analysis software *LMS CADA-PC*.

3.2 Experiment using a shaker

An second experimental modal analysis was performed on the same plate. The same measurement mesh with 55 points illustrated in Figure 7 was used.

To materialize free-free boundary condition, the plate was suspended using soft nylon ropes. A shaker **(?????)** was applied to provide the excitation at point just opposite to the mesh node 3, marked with a red circle in Figure 7, and an ICP accelerometer (*Brüel & Kjaer* 4507) was sequentially attached to every node of the measurement mesh using special purpose mounting beewax.

The set-up is shown in Figure 8.

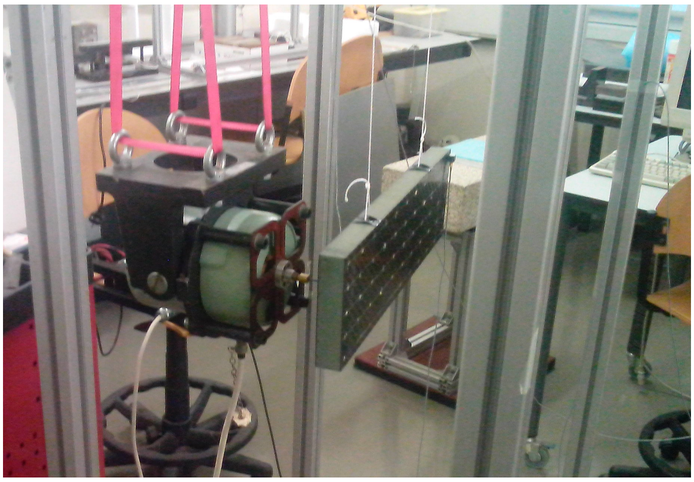

Figure 8. Comparison of natural frequencies

Again, both transducers were connected to a dynamic signal analyser for signals conditioning and analysis using a *Brüel & Kjaer* 2035 spectral analyzer. The signal analysis was performed inside the spectral analyzer, providing the set of frequency response functions of type accelerance for the entire measuring mesh considered in this study. The measurements were made in the frequency range 0–1600Hz. The set-up is identical to the one shown in Figure 4.

From the measured FRFs, in a first stage, a multi-degree of freedom (MDOF) technique, which uses the "least squares complex exponential" time domain algorithm, eas applied in order to identify the natural frequencies and damping ratios values for each mode.

In a second stage, the residues were identified with a "least squares frequency domain" technique and the mode shapes were obtained. The modal parameters were identified by using the modal analysis software *LMS CADA-PC*.

4. RESULTS AND DISCUSSION

Figures 9 and 10 respectively represent the Bode's diagram and the Nyquist's diagram of the driving point accelerance (A_{33}).

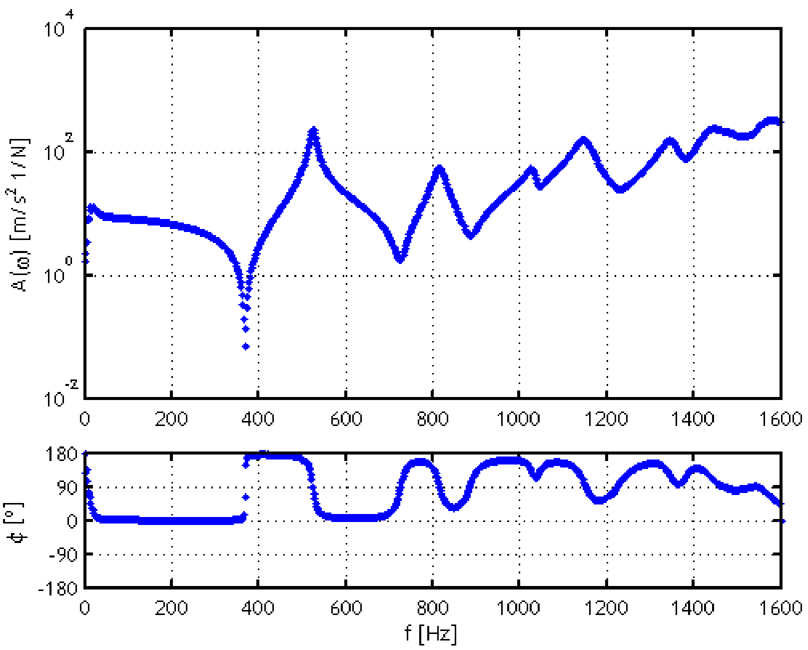

Figure 9. Bode's diagram of the driving point accelerance (A_{33})

The graphic in Figure 11 represents the magnitude of the 55 frequency response functions of the type accelerance obtained with the linear average of five independent measurements for each point during the experiment using an instrumented impact hammer, where special care was taken to ensure a good signal/ noise ratio and a good coherence level for the entire frequency analysis band (0:1600 Hz).

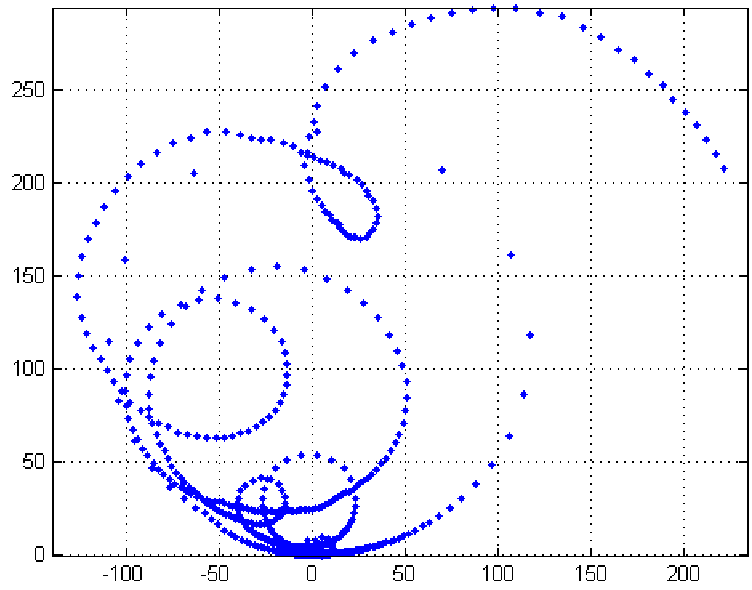

Figure 10. Nyquist's diagram of driving point's FRF $A_{33}\,(w)$

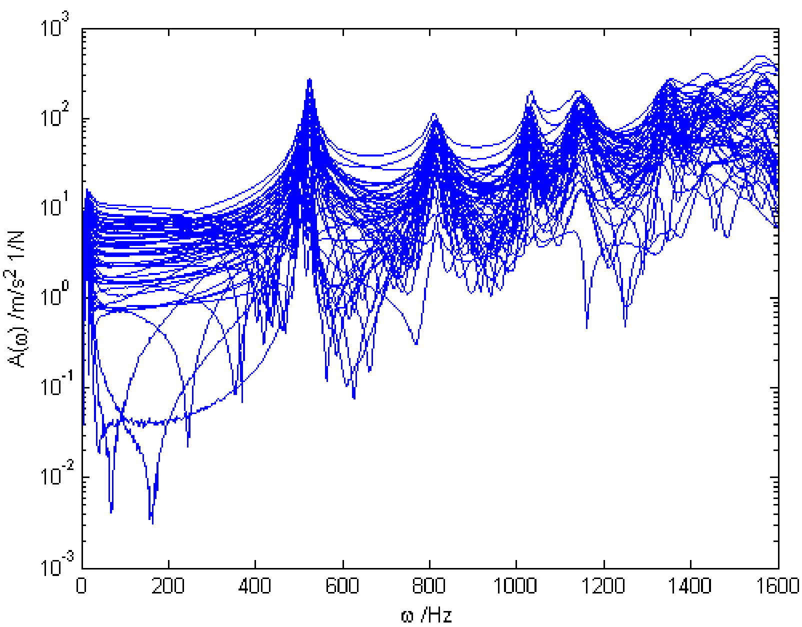

Figure 11. Measured frequency response functions

Table 1 presents the natural frequencies and modal damping ratios identified during the experiment using an instrumented impact hammer.

Table 1. Identified natural frequencies and modal damping ratios

Mode	ω (Hz)	Damping ratio (%)
1	503	
2	523	
3	815	
4	1030	
5	1149	
6	1347	
7	1436	

The graphic in Figure 12 shows the comparison of the natural frequencies extracted in the preliminary measurements, in the experimental modal analysis with hammer and from the numerical model.

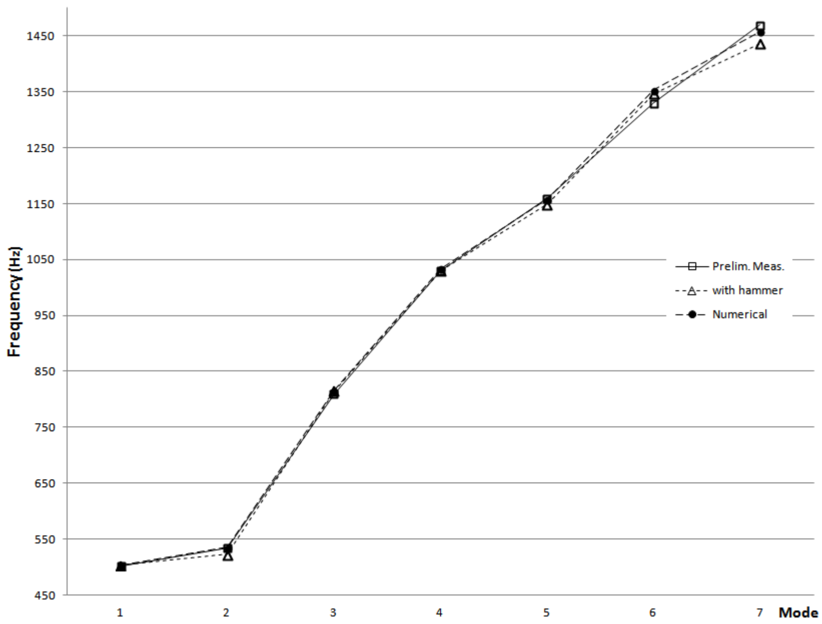

Figure 12. Comparison of natural frequencies

Figures 13 through 19 show the comparison between experimental mode shapes (above)and the numerical ones (below). Note that the first six modes extracted with the numerical model were not presented since they are rigid body modes and that the plots of the results from experimental analysis do not include the areas of the plate outside the grid limits (approximately 10mm wide margins around the plate).

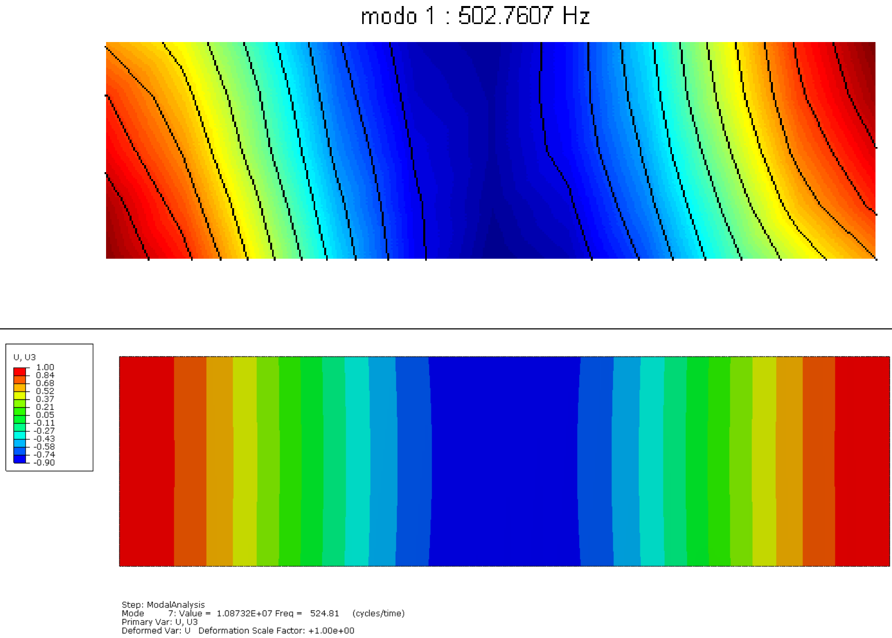

Figure 13. Comparison – mode 1

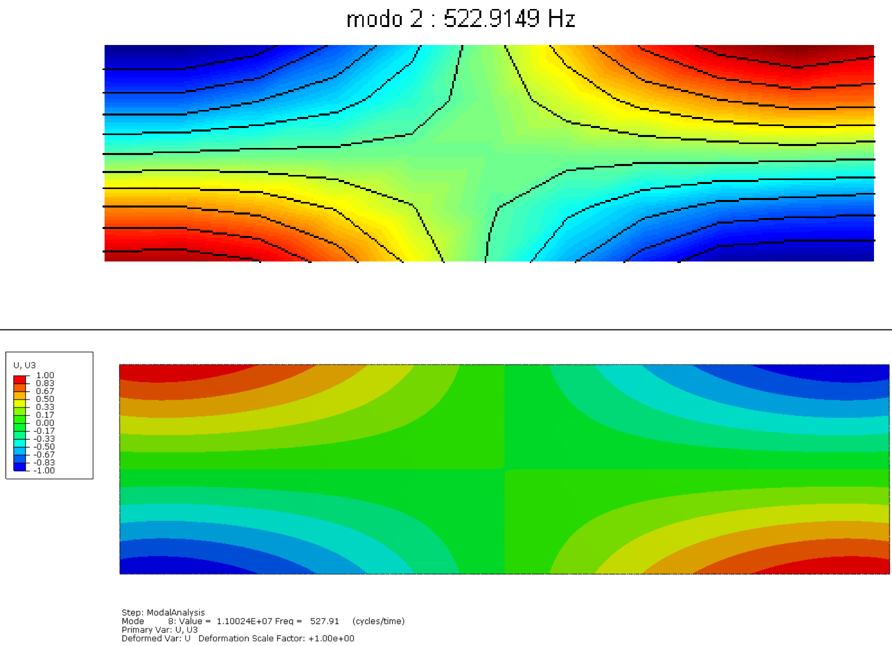

Figure 14. Comparison – mode 2

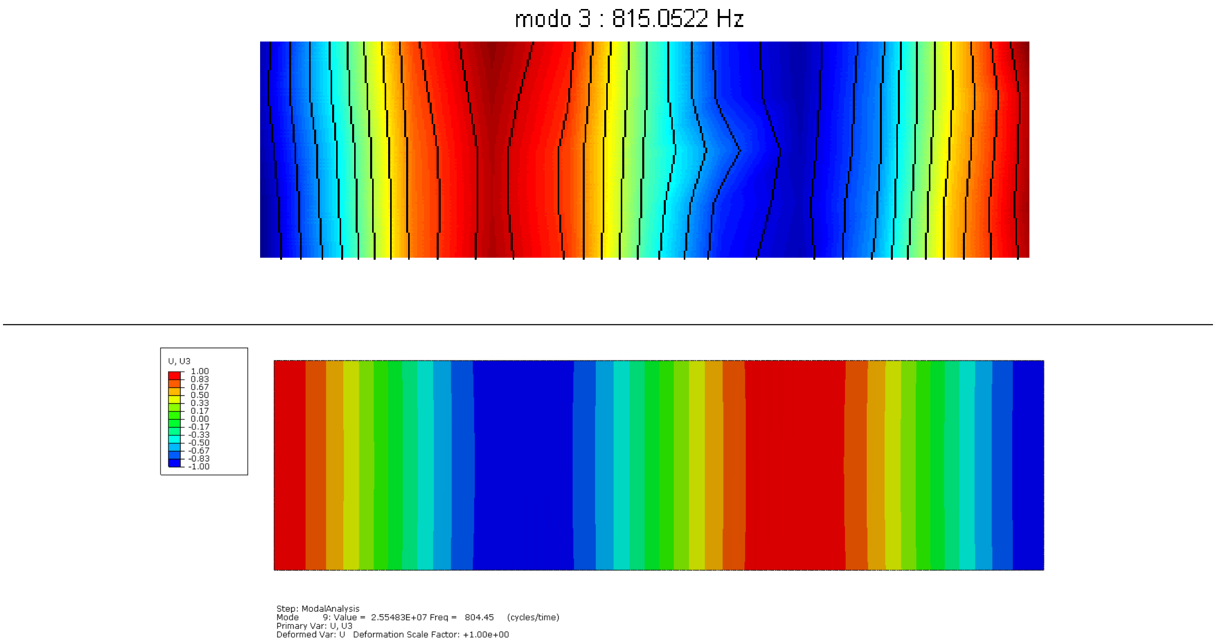

Figure 15. Comparison – mode 3

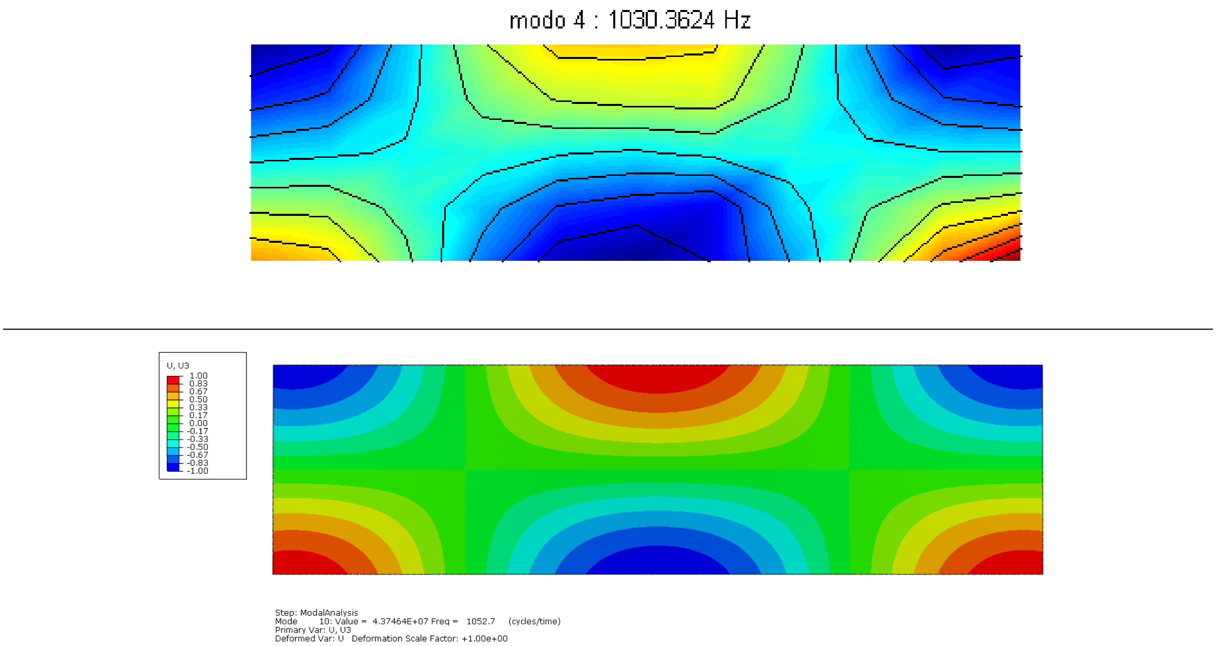

Figure 16. Comparison – mode 4

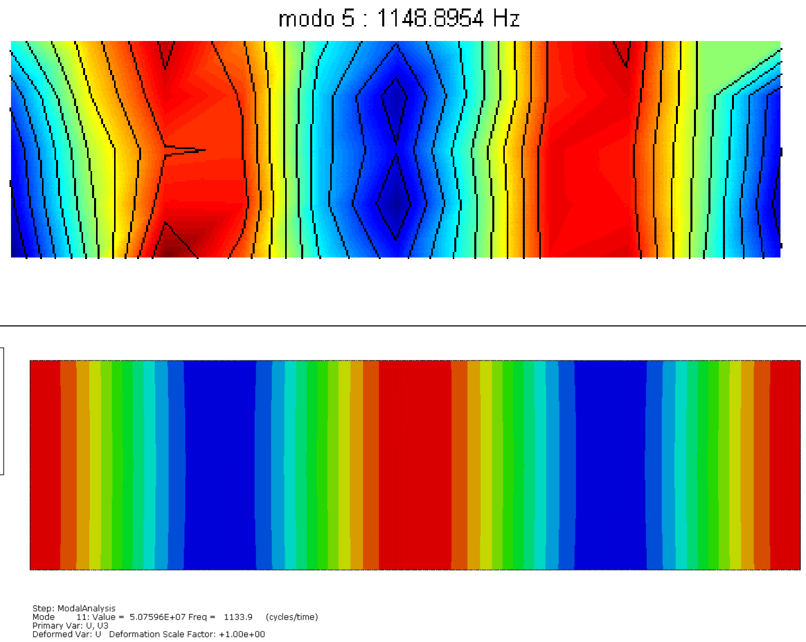

Figure 17. Comparison – mode 5

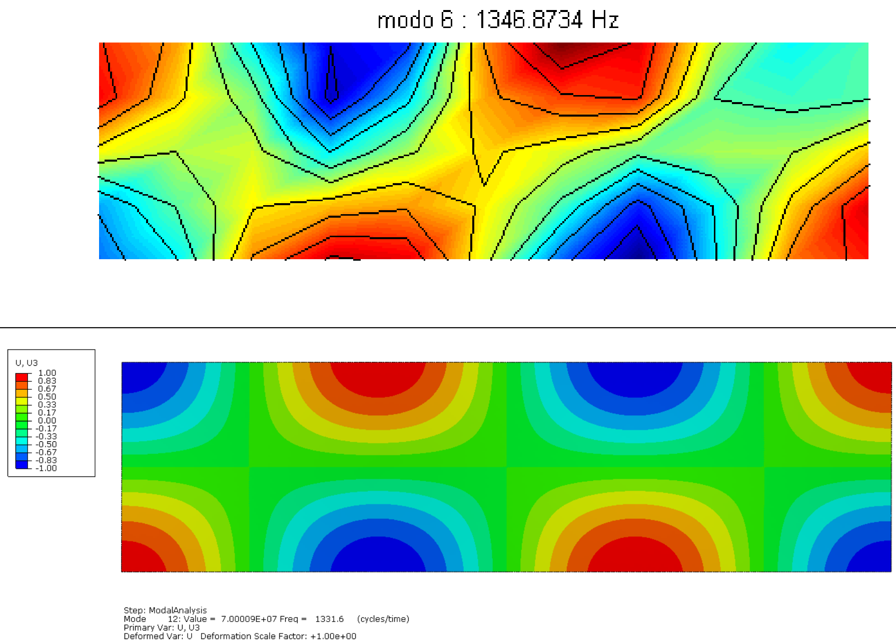

Figure 18. Comparison – mode 6

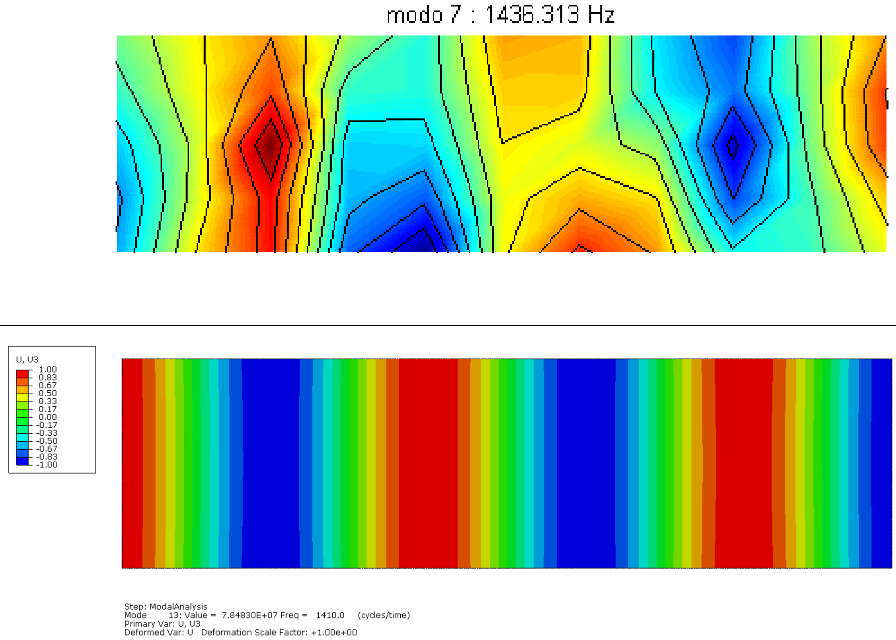

Figure 19. Comparison – mode 7

5. CONCLUSIONS

The numerical models were able to extract modal parameters that were very approximate to those effectively identified by experimental modal analysis of the sandwich plate. In fact, the first five mode shapes are very approximate. The sixth and, even more so, the seventh mode exhibit more relevant differences despite the fact that some similarity can still be identified.

Although it was possible to observe an almost constant proportional relation between the natural frequencies extracted with the numerical models and those experimentally identified, the necessary factors to correct the densities or the stiffness properties are to high to be admissible.

The modelling strategy and material properties used in the bus structure design was assessed when used in the modal analysis of the sample plate produced using the same materials and production process. The results obtained in this assessment do not indicate any limitation when using this approach to model the bus structure.

References

[1] Grant Agreement No 234019: Annex 1 – Description of Work; November 18th, 2009