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Acoustic optimization of a high-speed train composite sandwich panel based on analytical and experimental Transmission Loss evaluation integrated by FE/Test correlation analysis

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Abstract

Present work purpose is to optimize the acoustic attenuation properties of a composite sandwich panel used for a high-speed train structure, choosing the best panel configuration which allows to improve the performances. Firstly, Nilsson's analytical formulation for Transmission Loss (TL) evaluation has been implemented and experimentally validated on a typical material used for high-speed railway applications, highlighting the opportunity to use a different material to satisfy the new required design specifications. Different materials and stratifications have been then considered and TL parameter of each configuration have been calculated using Nilsson's formulation, characterizing acoustic behavior in the frequency domain. Once found the composition which ensures the best compromise between high acoustic insulation and low weight, the panel has been physically realized. Finally, an experimental and a numerical modal analysis have been performed on it. Starting from both FE simulation and impact testing outcomes, a correlation study through the computation of the Modal Assurance Criterion (MAC), has been carried out. A good agreement between numerical and experimental analyses has been found, obtaining a reliable FE model for future improvements.

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Keywords: Sandwich panel; Acoustic optimization; Transmission Loss; Nilsson's formulation; FE/Test correlation

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1. Introduction

In almost any vehicle application, maximum performance is strongly required. Various objectives may be used as a measure of performance, such as maximum speed, maximum payload, maximum safety, minimum energy consumption, or some combination of these and other objectives [1].

Global environmental issues like global warming and the reduction of fossil fuels usage, increase generalized interest to reduce the energy consumption in the transportation system. Besides the development of more efficient propulsion systems, one of the more recognized ways to reach a reduced energy consumption is to reduce the structural weight of vehicles. This weight reduction can be used to increase the payload or to increase the speed or just to reduce the energy consumption with maintained loading capability and top speed.

For these reasons, the interest in lightweight design in general and for sandwich constructions in particular for railway vehicles has increased over the last years. Sandwich structures represent a viable candidate for lightweight design, both from a mechanical and from an acoustic point of view.

There are many noise sources in and around a train. Particularly high speed trains generate a lot of noise and vibrations [2]. The largest contributor to the noise of a high speed train is the contact between wheels and rails. Close to the wheel-sets, sound pressure levels of 110-120 dB has been measured at 200 km/h [3] [4]. For even higher speeds, above 300 km/h, the aerodynamic noise becomes as important as the noise generated at the wheels [5]. Other noise sources are internal and more or less independent of speed, such as the air conditioning and the electrical systems.

Today's railway vehicle designs rely on a required weighted average sound reduction index for various parts of the body, which are combined to give a satisfactory acoustic environment.

Specifically, this work deals with the design of sandwich panels of a high speed train driver's cabin, ensuring minimum weight under simultaneous stiffness, strength, and acoustic constraints. The acoustic constraints are defined as a required sound reduction index for air-borne sound.

For this purpose, many variants of a typical high-speed railway material will be considered and acoustically characterized by using Nilsson's analytical formulation for calculating TL parameter, [6]. Once established the best plate stratification, a panel specimen will be realized and used for implementing a correlation procedure between panel experimental and numerical modal shapes. The good accordance found between experimental and numerical results will allow to consider the defined panel numerical model accurate and reliable for implementing further improvements.

Nomenclature

R	Sound Reduction Index
TL	Transmission Loss
P_i	Incident Acoustic Power
P_r	Reflected Acoustic Power
P_t	Transmitted Acoustic Power
f_c	Critical frequency
m_f	Skin mass
m_c	Core mass
$L_{p,s}$	Sound pressure level in source room
$L_{p,r}$	Sound pressure level in receiving room
R'_w	Weighted Apparent Sound Reduction Index
DOF	Degree Of Freedom
FFT	Fast Fourier Transform
FRF	Frequency Response Function

2. Sandwich panels

The term “sandwich panel” refers to a structure in the form of a lightweight core with thin faces bonded to each side of the core (see Fig. 1) [7]. The faces are usually made of an high performance material such as steel, aluminum or fiber composite, whereas the core is usually a structural solid foam, honeycomb or balsa wood. The efficiency of the sandwich construction derives from the proper choice of both face and core materials and on the adhesion between them [1].

A sandwich panel can improve some acoustical and structural requirements decreasing mass.

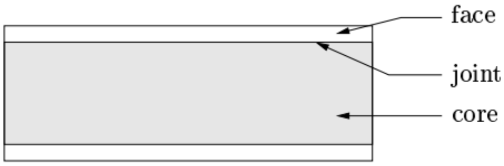


Fig. 1. The configuration of a typical sandwich structure.

The properties of the sandwich panel are a combination of skins’ and core’s properties. The skins are subject to tension/compression and provide a large contribution to the strength of the “sandwich”. The function of the core is to support the thin skins so that they don't deform and stay fixed relative to each other. The core experiences mostly shear stresses (sliding) as well as some degree of vertical tension and compression. Its material properties and thickness determine the stiffness of such a panel.

One of the major advantageous features of sandwich design is the good stiffness.

In Fig. 2 the problem of sound transmission through sandwich panels is schematized. Incident acoustic power P_i (sound waves) hits a sandwich panel. Some of P_i is reflected back, this is denoted by P_r, and some of it excites a wave motion in the panel. During this motion some power is lost through dissipation in the material P_d and some causes radiation of power P_t in the form of transmitted sound from the opposite side of the panel.

The dominating wave type in this excitation-radiation system is the bending wave, since it has a relatively large velocity component perpendicular to the panel surface and thus interacts efficiently with the surrounding medium. The Sound Reduction Index (R), or Transmission Loss (TL) in Anglo-Saxon literature, can be defined in its most general form based on the incident and transmitted acoustic power as

$$R = 10\log\left(\frac{P_i}{P_t}\right) \tag{1}$$

The Sound Reduction Index depends on the incidence angle and on the frequency of the sound waves. The sound field is usually variable both in frequency content and the incident direction and it is difficult to predict the exact properties of the incident field. Thus the assumption of a diffuse field is often used to predict the sound reduction of a construction when there is no available information about the real conditions. In the diffuse field, sound waves arrive with equal probability from all directions and the reduction index is integrated for all incidence angles.

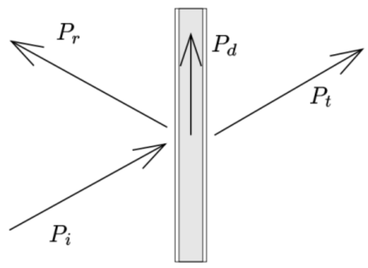


Fig. 2. Schematic description of sound transmission through a sandwich panel.

In Fig. 3 a "classic" reference curve that describes the trend of R as a function of the frequency of the incident sound waves, is shown.

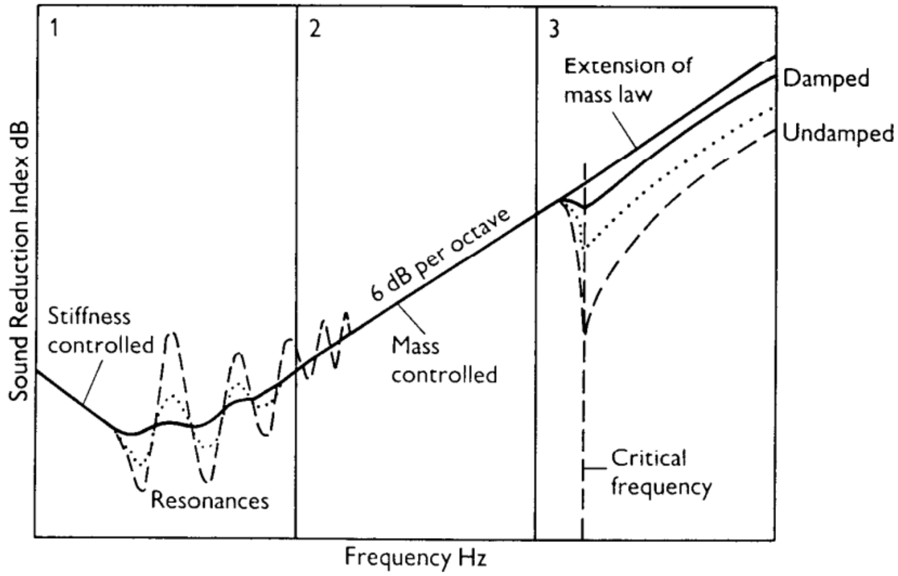


Fig. 3. Sound Reduction as a function of frequency.

In the graph three relevant regions can be distinguished. In the low frequencies region the main contribute to R is given by the panel stiffness. In the resonance region there is not a clear distinction between the contributions of the mass and the stiffness of the panel. In the third one the main input to R comes from the panel mass according to the mass law, that can be usually written as:

$$R = 10\log(mf) - 48 \tag{2}$$

where m is the panel mass for unit area [kg/m^2] and f is the frequency of the sound wave [Hz]. This region extends as far as the frequency at which there is a sudden reduction of R. In particular, at the frequency where the speed of bending waves coincide with the speed of sound in the surrounding medium, resonance occurs and the panel becomes more or less transparent to sound waves. This phenomenon is called coincidence and the corresponding frequency is called the critical frequency, f_c. The Sound Reduction Index at or close to this frequency is usually low. For frequencies above f_c, the R trend follows the mass law again. The structural damping of the material has a great influence on the reduction index close to the critical frequency.

For sandwich panels the wave speed depends on the geometry of the constituents and on the materials used. Fig. 4 shows the calculated phase velocities for plane bending waves in a sandwich panel c_s and for a single-skin panel c_p, having the same bending stiffness and mass per unit area as the sandwich panel. The lower asymptote, c_f, is the bending wave speed in one of the sandwich faces acting as a single skin panel and c_a is the speed of the sound waves in air [1].

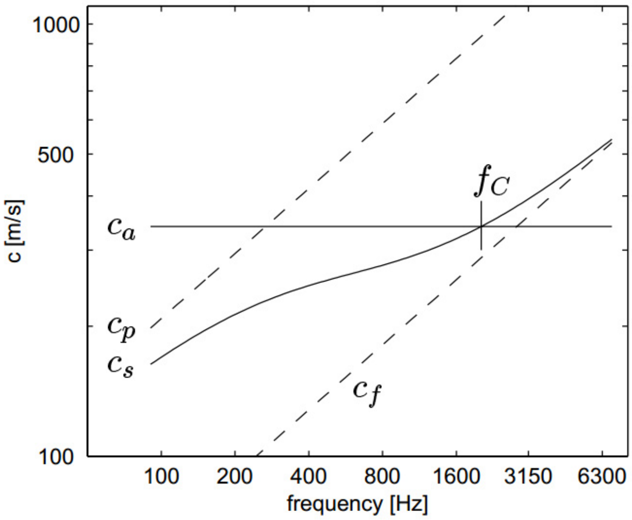


Fig. 4. Phase velocities of sound waves.

In general, at low frequencies the whole sandwich create bending waves, while at high frequencies the skin start to bend independently. It is possible to demonstrate [8] that critical frequency can be obtained by

$$f_c = \frac{c_a^{\ 2}}{2\pi}\sqrt{\frac{m_f + 0.5m_c}{D_f}} \tag{3}$$

where m_f and m_c are the skin and the core mass per unit surface respectively, and D_f is the skin stiffness.

Generally, Sound Reduction Index of a plate structure is experimentally evaluated mounting the panel in the wall between a source room and a receiving room, see Fig. 5. The source room is usually a reverberation room and the receiving room is an anechoic room.

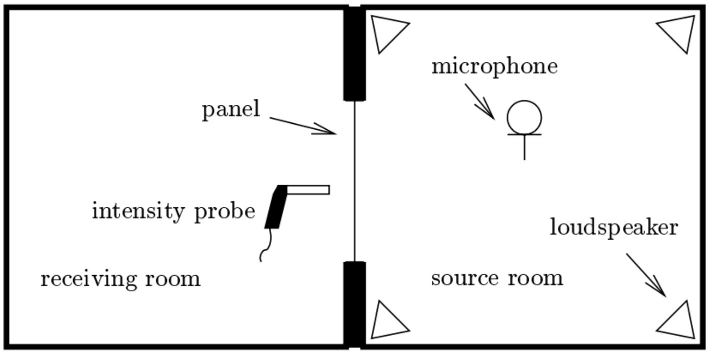


Fig. 5. Test set-up for measurements of Sound Reduction Index.

A diffuse incident field is commonly used for experimental work on sound reduction, since this is representative of many applications. Experiments with directional incident fields are much more cumbersome to perform. Under laboratory conditions it is common to measure the sound intensity level on the receiving sound and the sound pressure level on the source side. The relation between Sound Reduction Index and the measured quantities is

$$R = L_{p,s} - L_{I,r} - 6 \qquad \text{[dB]} \tag{4}$$

where $L_{p,s}$ is the sound pressure level in the source room and $L_{I,r}$ is the sound intensity level in the receiving room. The subtraction of 6 dB is because the receiving room is anechoic, and thus the sound radiates from the panel purely in one direction without reflection in the receiving room.

3. Analytical formulation

Analytical approach has been implemented to evaluate R'_w of a typical material used for high-speed railway applications, starting from the structural/mechanical properties of core and plies.

R'_w is the Weighted Apparent Sound Reduction Index, a single-number rating, expressed in decibels, of the laboratory or field frequency dependent measurement of airborne sound insulation between rooms, that may include the influence of flanking sound.

The analytical approach refers to Nilsson's equation [7] [8] [9]. The equation is based on mass law and is adjusted to keep in account some effects due to modal behavior, coincidence and damping. Nilsson's equation is expressed as:

$$R = 20\log_{10}(m) + 20\log_{10}(f) - 10\log_{10}(\Gamma\Delta + G) - 48 \qquad if\ f < f_c \tag{5}$$

$$R = 20\log_{10}(m) + 30\log_{10}(f) - 10\log_{10}(f_c) + 10\log_{10}(\delta) + 5\log_{10}(1 - f/f_c) - 47 \quad if\ f > f_c \tag{6}$$

where m is the mass for unit area of the plate, f the frequency and fc the coincidence frequency calculated using (3).

The parameter Δ is a function of the boundary conditions for a plate with length and width equal to b and c; in particular, for a simply supported panel is

$$\Delta = 1 + \frac{3 \cdot 10^4}{4\delta f^{0.5} f_c^{1.5}} \left(\frac{1}{b^2} + \frac{1}{c^2} \right) \tag{7}$$

whereas for a clamped panel is

$$\Delta = 1 + \frac{3 \cdot 10^4}{\delta f^{0.5} f_c^{1.5}} \left(\frac{1}{b^2} + \frac{1}{c^2} \right) \tag{8}$$

δ represents the total damping of the plate.

Γ is a function of baffle and plate dimensions. If the plate is mounted without baffle then Γ is equal to 1.

G describes the resonant transmission trough the panel

$$G = \int_0^{\frac{\pi}{2}} \frac{d\gamma \sin(\gamma)}{\left\{ \left[1 - (f/f_c)^2 \sin^4(\gamma) \right]^2 + (f/f_c)^4 \sin^8(\gamma \delta^2) \right\}} - 1 \tag{9}$$

and can be numerically solved considering f/f_C as parameter.

In Fig. 6 Transmission Loss parameter calculated using Nilsson's equation is compared to experimentally determined Sound Reduction (R) data.

A good accordance between analytical and experimental results (both performed by our Lab and Berlin University Lab) can be observed.

For the considered panel configuration R'_w, obtained according to the standard UNI EN ISO 717-1 starting from both analytical and experimental R data, assumes the value of 34 dB.

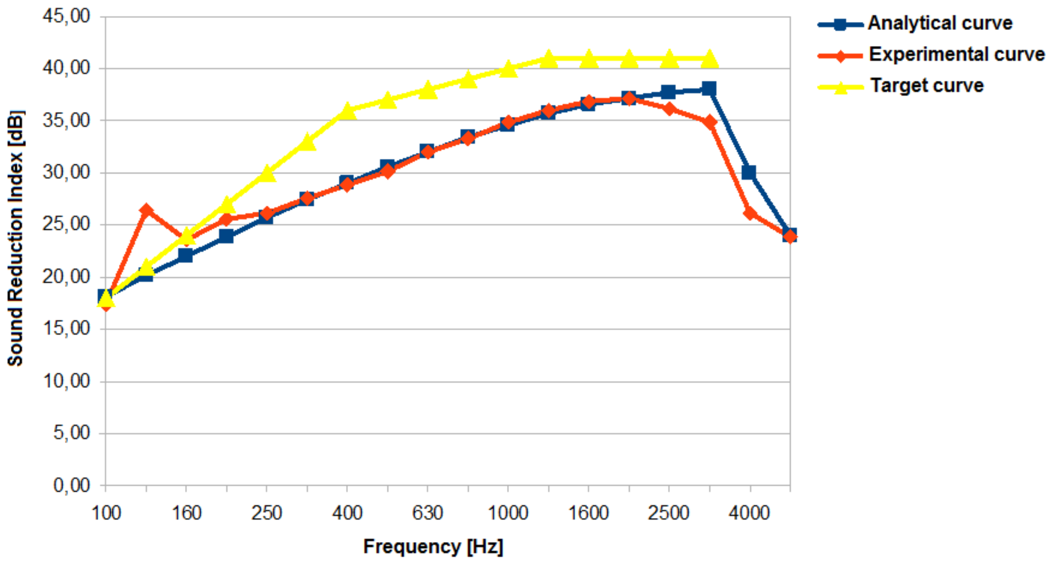


Fig. 6. Nilsson's formula data vs experimental data.

Since new design specifications have to be satisfied to improve material acoustic attenuation properties (see above yellow marked target curve), different kinds of layup stratification have been considered choosing the configuration which will represent the best compromise in terms of high R'_w value and low material weight per unit area.

As Nilsson’s formulation seems to well approximate R experimental results, optimization process has been implemented using the analytical approach, so to avoid costs that could derive from an experimental campaign or from a numerical simulation (in the latter case it would be a computational cost).

4. Optimization

New structure configurations have been established changing the type of foam and varying the core’s thickness value. The different stratifications are analyzed calculating R'_w starting from Nilsson’s formulation results. For sake of brevity only some of the obtained results will be depicted in the following figures (Figg. 7-10).

Table 1. Characteristic values of some of the considered stratifications.

	R'_w [dB]	Weight [kg/m^2]
Layup 1	35	18
Layup 2	36	15.8
Layup 3	42	25.0
Layup 4	41	20.5

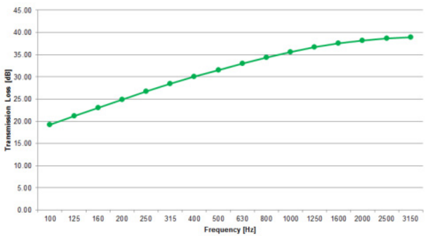


Fig. 7. R of layup 1.

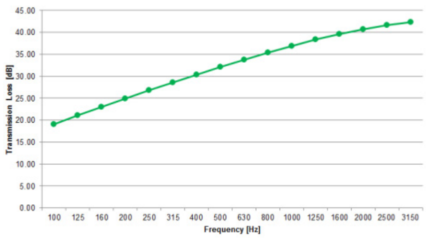


Fig. 8. R of layup 2.

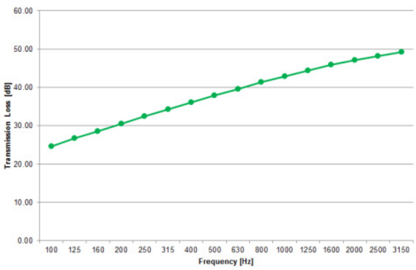


Fig. 9. R of layup 3

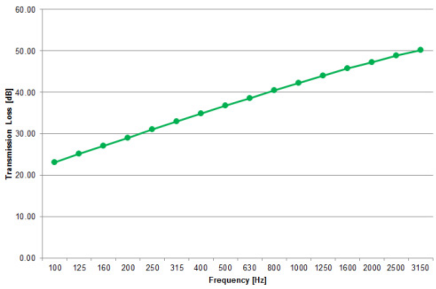


Fig. 10. R of layup 4.

Layup 4 (whose stratification is shown in Tables 2-3) seemed to be the best stratification to be used, as in addition to represent the right compromise between high sound insulation and low weight, it satisfies also stiffness and strength requirements.

Table 2. Stratification of layup 4.

	Material	Thickness [mm]
1	MAT	1
2	BIAX	1
3	BIAX	1
4	FOAM	25
5	CORK	3
6	FOAM	25
7	BIAX	1
8	BIAX	1

Table 3. Mechanical properties of layup 4's materials.

Material	ρ kg/ m^3	E_L N/ mm^2	E_T N/ mm^2	G N/ mm^2	ν
MAT (Orthotropic)	150	7830	7600	2818	0.27
BIAX (Orthotropic)	530	13750	14000	3090	0.27
FOAM (Isotropic)	80	104	104	40	0.3
CORK (Isotropic)	120	5.1	5.1	0.9	0.3

Legend:

E_L	Young's Modulus in the fiber long direction (0°)
E_T	Young's Modulus in the fiber transversal direction (90°)
G	Shear Modulus
ν	Poisson's Ratio

As a consequence, a panel characterized by this particular composition has been physically realized and, once available, it has been used to implement an experimental/numerical modal analyses correlation, which will be described in the following paragraph.

5. Experimental/Numerical Correlation

Panel dynamic behavior is generally determined through modal analysis [10], performed in two different ways: the first one, consists in a numerical modelling involving the use of Finite Element Method (FEM) and through the employment of a commercial software; the second one is represented by an experimental test carried out by using the well-known Impact Test procedure [11]. The main objective is to correlate the numerical model to the test model, so to check their dynamic behavior in the frequency domain in terms of transfer functions (FRF), mode shapes, total mass and vibrational response in operating conditions. Fig. 11 shows the flow diagram of the procedure that is generally implemented.

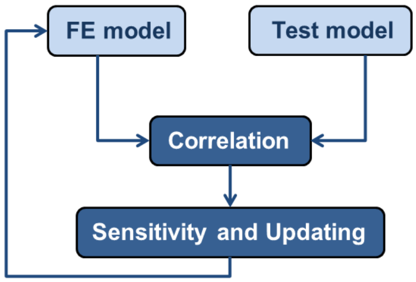


Fig. 11. Flow diagram of the FE-Test model correlation procedure.

After the two analyses implementation, the definition of an accurate model is obtained step by step through the identification of the uncertain model parameters which mostly influence the correlation targets and the progressive optimization and updating of the FE model. The typical model uncertainties could regard the materials, the geometry, the boundary conditions and connections and the mesh characteristics.

5.1. Experimental modal analysis

Experimental modal analysis allows to determine modal parameters (frequencies, damping factors, modal vectors and modal scaling) of a system by way of an experimental approach. Four basic assumptions are necessary to perform an experimental modal analysis: (a) the structure is linear, that is the response of the structure to any combination of forces, simultaneously applied, is the sum of the individual responses to each of the forces acting alone; b) the structure is time invariant, that is the parameters that are to be determined are constants; (c) the structure obeys Maxwell's reciprocity, which states that a response R_{ab}, measured at location a, when the system has an excitation signal applied at location b, is exactly equal to R_{ba} which is the response at location b, when that same excitation is applied at a (if H_{ab} is the transfer function between a and be then $H_{ab} = H_{ba}$); (d) the structure is observable: input/output measurements contain enough information to generate an adequate behavioral model of the structure.

The tested panel satisfies all these four assumptions.

The methodology adopted in present work to acquire modal data is represented by impact testing, the most popular modal testing method used today which allows to compute FRF measurements in a FFT (Fast Fourier Transform) analyzer. In this specific circumstance, a roving hammer impact test has been performed, fixing the output and measuring FRFs for multiple inputs..

The equipment used to perform the impact test (see Fig. 12) was composed of:

1. an impact hammer with a load cell attached to its head to measure the input force;
2. a PCB tri-axial accelerometer to measure the response acceleration at a fixed point (DOF) and directions;
3. a multi-channel FFT analyzer to compute FRFs (Scadas III Acquisition System);
4. a pre and post-processing modal software for identifying modal parameters and displaying the mode shapes in animation (LMS Test.Lab).

In LMS Test.Lab "Impact Testing" module measurement setup has been defined in terms of geometry and orientation of the structure, positions of acquisition points (nodes) with respect to the chosen coordinate system, sensitivities of the tri-axial accelerometer (one for each direction) and of the load cell attached to the hammer, frequency range, trigger point.

For testing the panel structure, 81 acquisition points were chosen on it (Fig. 12).

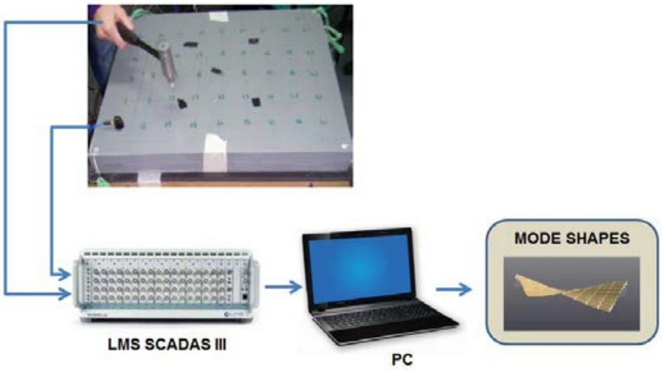


Fig. 12. Data acquisition through roving hammer impact test.

For this purpose the tri-axial accelerometer has been simultaneously sampled together with the force data. The chosen frequency range was fixed up to 3200 Hz, so that a Teflon impact hammer head could be used.

Each DOF has been hit 3 times, therefore each FRF was calculated averaging over 3 instantaneous FRFs [11].

After testing results analysis has been carried out, specifically the first ten modal shapes up to 1000 Hz have been computed and virtually animated.

5.2. Numerical modal analysis

As regard the numerical modal analysis, the software Femap (acronym of Finite Element Modeling And Post-processing) has been used in order to build the finite element model of the panel [12] [13] [13]. Solution results, obtained with internal MD Nastran solver, have been then utilized for FE/Experimental correlation through the employment of LMS Virtual.Lab module.

Once created CAD geometry, in the menu Material the properties of each lay-up of the panel have been set. The sandwich panel is composed of different layers as better described in previous chapters. In order to create an accurate model, the core that can be modeled as an isotropic material (foam) have to be assembled to the skin that can be modeled as a 2D orthotropic material (matrix + reinforcement) [14] whose characteristics may be calculated by the use of the lamination theory, starting form the properties of fiber and matrix, taking into account the orientation of the consecutive plies and then calculating the characteristics of the entire laminate. BY the use of the sandwich theory, laminate and core have been assembled to define an equivalent orthotropic material that has been used to define the shell element in the FEM model.

The entire panel surface has than been divided in 81 (9x9) QUAD elements, delimited by 100 (10x10) nodes (see Fig. 13), to guarantee accurate results up to 1000 Hz..

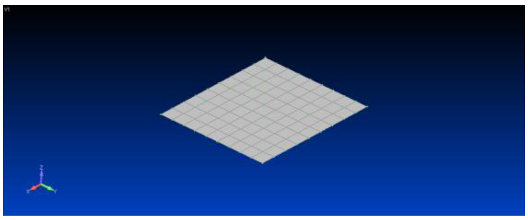


Fig. 13. Panel shell model.

In this case the panel has been modelled according to a “free-free” configuration, in the same conditions in which experimental analysis, has been performed. The output results have been then imported in the Virtual.Lab environment to define the FE correlation analysis to be implemented in.

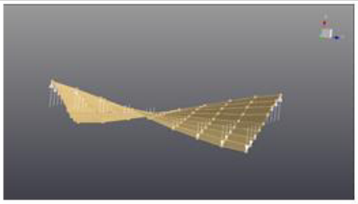
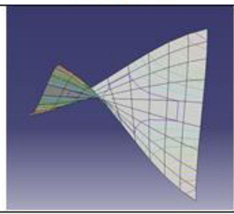
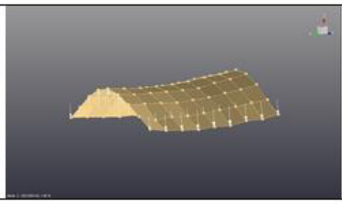
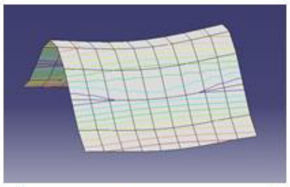
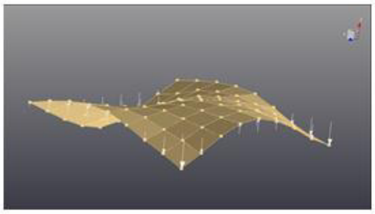
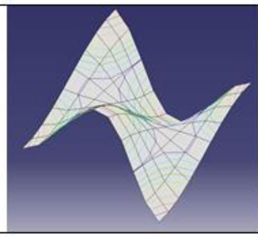
5.3. Correlation procedure

Both numerical and experimental results in terms of natural mode shapes and frequencies of the panel, are shown and compared in Table 4. For sake of brevity only the first three mode shapes experimentally and numerically determined are shown in Table 5. The first one is (1,0) mode that represents the flexural shape in x-direction, the second one is (0,1) mode that is, vice versa, the flexural mode in y-direction, the third one is (1,1) mode representing the flexural mode in both plane directions.

Table 4. Comparison between sandwich panel numerical and experimental modal analysis.

Mode no.	**Modal frequency [Hz]**		***m,n***
	Numerical	**Experimental**	
1	286.5	288.3	1,0
	0.62 %		
2	419.3	393.8	0,1
	6.48 %		
3	508.2	516.8	1,1
	1.66 %		
4	579.2	588.2	2,2
	1.53 %		
5	649.2	665.9	0,2
	2.5 %		
6	709	758.5	1,2
	6.53 %		
7	794.9	833.8	2,0
	4.67 %		
8	829.9	899.7	2,1
	7.76 %		

Table 5. First three experimental and numerical modal shapes.

Mode no.	Experimental	Numerical	m,n
1			1,0
2			0,1
3			1,1

It is evident that a good agreement is found up to a frequency of 1000 Hz because of the low error percentages (<10%). However, in order to validate the FE model a modal shapes correlation study, has to be performed.

The correlation phase is focused on comparing, understanding and evaluating correlation between Test and FE data. The final scope is eventually to modify design parameters improving the correlation and to progressively update the model, also in terms of material properties.

The MAC (Modal Assurance Criterion) index represents a possible correlation criterion to be used.

MAC index is defined as:

$$MAC\left(\{\Psi\}_{test},\{\Psi\}_{FE}\right)=\frac{\left|\{\Psi\}_{test}^{*t}\{\Psi\}_{FE}\right|^2}{\left(\{\Psi\}_{test}^{*t}\{\Psi\}_{test}\right)\left(\{\Psi\}_{FE}^{*t}\{\Psi\}_{FE}\right)} \tag{10}$$

where $\{\Psi\}_{test}$ $\{\Psi\}_{\mathrm{test}}$ and $\{\Psi\}_{FE}$ $\{\Psi\}_{\mathrm{FE}}$ are the modal vectors computed respectively in the experimental and numerical analyses, too.

MAC allows to quantify the modal shapes correlation (from 0 to 1), pointing out the possible presence of missing modes (not square matrix) and/or switching mode (matrix diagonal inversion), and expressing the orthogonality (0) or the parallelism (1) of two any vectors.

Firstly, in order to verify the correlation of the experimentally determined modal shapes, an Auto-MAC analysis [15] has been carried out (Fig. 14).

Obtained results show that the computed modes are not combined between them, because the extra-diagonal values are lower than the critical threshold value of 20%.

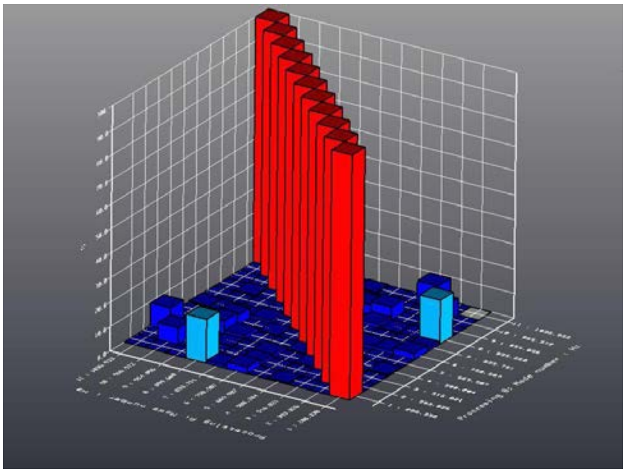


Fig.14. MAC - Modal Assurance Criterion of the experimental modes (Auto-MAC)

MAC methodology is used also for Test-FE data correlation.

Fig. 15 demonstrates that modes correlation is mostly good, in fact only MAC diagonal values are different from zero and greater than 0.7, which is indicative of a good correlation between the two compared models. In particular, MAC lower diagonal values are those highlighted.

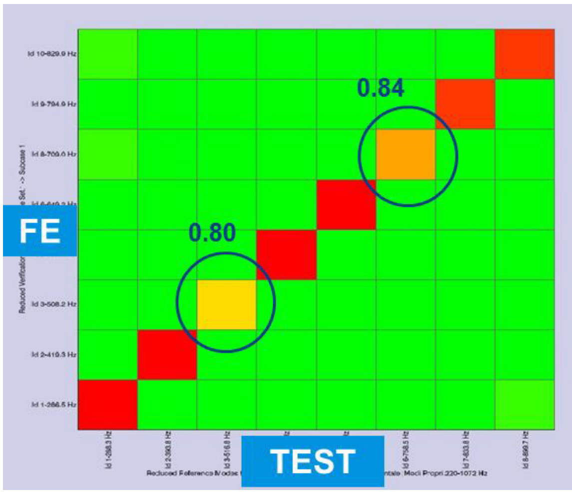


Fig. 15. MAC for Test-FE correlation.

These results allow to validate the defined FE model, without the necessity to implement a successive sensitivity and updating procedure.

6. Conclusion

In the present work an acoustic optimization process of a high-speed train sandwich panel has been carried out, through the implementation of Nilsson's analytical formulation for the determination of the structure acoustic attenuation properties. The use of Nilsson's formula for TL calculation has been possible since a good accordance has been firstly found between analytical and experimental TL results of panel initial configuration.

Different sandwich panel stratification have been considered changing the type of foam and varying the core's thickness value, and so calculating corresponding R'_w index and weight per unit surface. The various configuration have been evaluated also in terms of strength and stiffness. The configuration which has been finally chosen ensures the best compromise between high sound insulation and low weight, allowing to satisfy the prescribed strength and stiffness requirements too. After the choice of the most suitable structure, a panel specimen has been physically realized in order to use it for performing a correlation procedure between experimental and numerical modal shapes and natural frequencies. As a very good agreement has been found between experimental and numerical results, the defined sandwich panel FE model could be used for further material acoustic performance improvements. Transmission Loss parameter, in fact, could be numerically assess through a BEM simulation, avoiding the costs which may derive from an experimental test.

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