Design and testing of a prototype foam for lightweight technological applications

M. Viscardi, M. Arena and D. Siano

***Abstract*—**Over recent years, in the automotive field, numerous performance and aesthetic innovations have been produced thanks to the development process of the manufacturing technologies gained mainly in the aerospace industrial context. The automotive industry is currently experiencing relevant technology changes in the design of the engines, transmission and total drivetrain, induced by increasing customer demand for fuel efficiency and more stringent government requirements in emissions and safety. One of the problems relating to environmental impact concerns the noise emitted by the vehicle, for which various solutions have been experimented: new and more resistant materials have been worked out in order to minimize noise pollution and the environmental impact of the vehicle, even at the end of the operating life of its components. Several research programs are currently running or recently terminated worldwide to explore the feasibility of smart materials. The increasingly dominant role of lightweight materials in many technological sectors is motivated by the multitude of benefits that they could offer like the weight optimization and the reduction of the fuel burn and noise levels. This research illustrates a solution as a response to those requirements, as well as being a response to the targets of comfort: a viscoelastic material, appointed to increase the damping of structures involved in vibro-acoustic phenomena generated in a vehicle. The performance of these innovative materials have been analyzed both from a numerical standpoint that experimental. Static mechanical properties and modal parameters carried out in the laboratory, pertinent to each configuration were arranged into a rational database for further studies on the vibro-acoustic behaviour of the coupled cavity-structure system. The main goal of this research project has been reached in the design, manufacturing and testing of an innovative viscoelastic prototype got out by the best compromise of structural and acoustic characteristics of pre-existing trim materials.

***Keywords*— Automotive, Damping, Finite Element Model, Non-contact Measurement, Noise Vibration Harshness, Viscoelasticity.**

I. INTRODUCTION

THIS work has the purpose to investigate new materials and technologies aimed to reduce the noise and vibrations produced inside vehicles cabins. In recent years, the acoustic problem has represented a rate of the quality of human life. Thanks to the appropriate "know-how" matured in the aerospace sector, many technological solutions have been extended to all transport engineering fields. Aside legal obligation, all the big automotive manufactures are actually directed to solutions for better passenger's acoustic comfort. The reduction of vibration and noise in and across several components and modules of the automotive, such as the panels, doors, engine covers, seats, and others, is of primary importance. The NVH (Noise Vibration Harshness) performance may be a crucial factor in the purchase decisions of numerous buyers [15]. The considerable use of trim materials is supported by the benefits offered in terms of weight reduction and satisfactory mechanical performance. Constrained viscoelastic treatments have been considered as a possible technology to enhance damping in new aircraft BLISK engines (integrally bladed disk): the localization of the constraining layer within the main shaft, avoids any interaction with the airflow and imposes very little shear in the constrained layer thus limiting the risk of delamination [23]. An innovative approach in manufacturing fuselage skin panels has been evaluated, where a thin layer of viscoelastic material has been added to the lamination sequence of the skin, in the middle plane. This allows for increasing the panel damping performance with a very neglectable added weight [6]. The "low-noise" design concept consists so in the development and implementation of smart solutions in other sectors of transportation engineering too. In this framework, innovative materials targeted at the reduction of the car-body floor, will be analysed and compared to "standard one"; in the specific, viscoelastic foams will be investigated as a valid alternative to conventional add-on damping element as generally used in these applications [3]. Standard foams are already used as a part of the car-body carpet element, but their role is mainly the decoupling of the carpet from the floor; basic idea of the research is to force this element to strongly contribute to the vibrational energy dissipation. In the present paper, both numerical and experimental procedures have been addressed to evaluate the dynamic performances of viscoelastic foams with different physical properties [4]. Such foam, with a small thickness, is able to absorb strong shocks and impacts, to disperse about 97% of that energy without breaking. The particular structure of the foam allows for having an excellent sound absorption coefficient; specifically, it was designed to demonstrate the maximum absorption in the frequency range where the human ear is more sensitive. The role of the viscous damping term by means of several experimental techniques will be discussed. The test results have been collected into a well-validated numerical database, which could be adopted for potential fluid-structural simulations. By the static and dynamic test results of reference samples, the input to the manufacturing of an improved viscoelastic carpet prototype has been given. The foam used for its production is a compromise: it has the stiffness of a standard foam and the vibro-acoustic characteristics of those viscoelastic analyzed. The weight of the prototype foam is the same of the standard one. It is obtained a foam with better insulation characteristics but with the same weight of the standard one.

II. VISCOELASTIC MATERIALS

Viscoelasticity is a property of materials that exhibit an intermediate response between those viscous and elastic ones. The viscoelastic materials are opposed to both the tangential stresses and the normal stress, generating a distribution of balancing efforts. Same examples of viscoelastic materials are synthetic polymers, wood and human tissues. A polymer is a macromolecule, or a molecule with a high molecular weight, consisting of a large number of molecular groups (said repeating units), united "chain" through the repetition of the same type of bond (covalent). The pure polymeric materials are divided into thermoplastics, thermosetting and elastomers. An ideal elastic material has a defined shape and subjected to external forces tends to a form of equilibrium. Upon removal of the force, the body returns to its original shape. For small deformations, there is a law of proportionality between stress σ and strain ε, independent of time (Hooke's Law). Instead, the fluids do not show an elastic response: there is not proportionality between stress and deformation applied. Their behavior is expressed by a relationship describing irreversible deformation processes (Newton's Law). The viscous and the elastic behaviours can be combined together, giving rise to viscoelastic response. Such characteristic is typical of the polymers: they can show all the features of a glassy solid, or of an elastic rubber, or of a viscous fluid, depending on the operating temperature and the strain rate. A viscoelastic material combines the characteristics of an elastic solid with those of a viscous fluid. The application of a load for a relatively long time may induce a permanent deformation. A transverse load can induce a response of a resilient type in certain polymeric fluids. From scientific and mathematical standpoint, two important theoretical ways are acknowledged: linear viscoelasticity and nonlinear viscoelasticity; the first one is an approximation highly used in polymer engineering ('small' strain theory) while the second one is used when strains are so large as to change material's properties. To describe viscoelastic materials with good approximation are used different physical models, commonly called viscoelastic mechanical models, such as Maxwell model and Kelvin-Voigt one, which represent material's response subject both to stresses and strains. Polyurethane foams are produced through the formation of the polyurethane polymer and the concomitant development of the expanding gas. When these two processes are well balanced, the gas bubbles are trapped with the polymer matrix during its formation. If gas development is too fast, the foam initially grows well, but then collapses because there is not an enough strong array to hold the gas; if the polymerization is too fast, the foam does not grow properly. The flexible foams are characterized by a glass transition temperature below the room temperature. The rigid foams have higher temperatures. Thanks to the viscoelasticity, to a non-elastic return and to the initial shape, the main property of the viscoelastic polyurethane foams is to absorb energy [2-3].

III. STRUCTURAL NUMERICAL MODEL

The FE model has been conceived to be fully representative of the test articles adopted in the laboratory in order to validate their vibro-acoustic behavior [20]. Each foam has been discretized using HEX8 elements glued on a 2-D (CQUAD) mesh [5], which simulates the metal support plate, Fig. 1. The main characteristics of FE (Finite Element) model, performed within Patran – MSC Nastran® environment, are summarized in Table I.

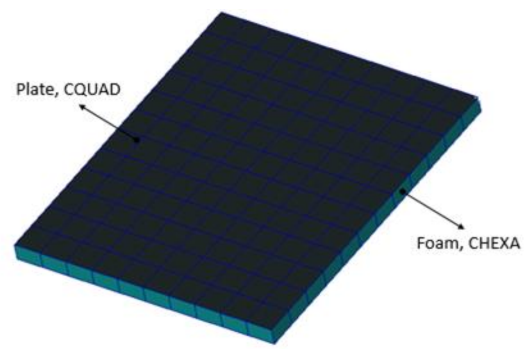

Fig. 1 Structural system mesh

TABLE I. FE MODEL CHARACTERISTICS

FE entity	n°
Nodes	286
CHEXA	120
CQUAD	120

The material properties in structural model (density ρ, Elastic Modulus E, Shear Modulus G and Poisson ratio ν) have been assigned considering a linear behavior for the original foam, while viscoelastic characteristics were defined for 65-30 and 75-30 foams, Table II.

TABLE II. ISOTROPIC MATERIAL PROPERTIES

Data	Original Foam	Foam 65-30	Foam 75-30
ρ [Kg/m^3]	65	65	75
E [MPa]	0.012	0.017	0.525
G [MPa]	Linear	0.0057	0.1236
ν	0.3	0.3	0.3

IV. DYNAMIC ANALYSIS AND FE MODEL UPDATING

The vibration test has been performed by a scanning Laser Doppler Vibrometer (LDV), Polytec 400® [21]. Such non-invasive technique allowed for acquiring the microscopic vibrations at 63 monitoring points as well as the operational deflection shapes (ODS) of the entire surface in the bandwidth 0-2048 Hz. The structure was excited by a piezoelectric actuation device, which generated a white noise signal, Fig. 2.

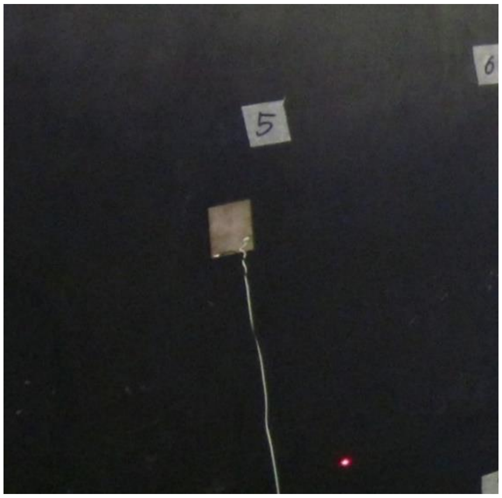
Fig. 2 Driving point and laser beam

In order to simulate the free-free condition, the prototype was suspended by means of springs as showed in order to get a proper frequency separation between pendulum vibration and structural elastic modes.

Free-free boundary conditions were targeted. This condition has been realized suspending the panels to a rigid structure through two springs, in order to avoid panel rotation around suspension point. The springs have been designed in a way that proper natural frequency is one order of magnitude less than first natural frequency of panels [6]. The link between the springs and the panel is realized by two small holes in the shortest side of the panel.

Good quality experimental data were obtained, as demonstrated by the consistency of the coherence and FFT (Fast Fourier Transform) functions, Fig. 3.

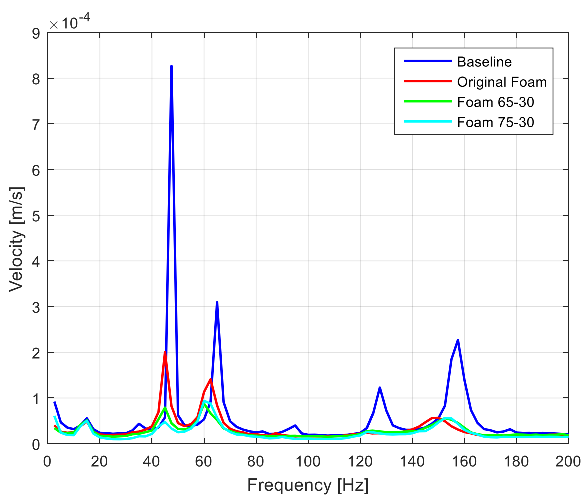

Fig. 3 Vibration velocity spectrum, Laser Test

The sequence of the extracted RMS (Root Mean Square) velocity, associated with each configuration is listed in the Table III. The results show the significant damping induced by the foams, on the vibration velocity: it is clear a reduction of one order of magnitude compared to the baseline case.

TABLE III. RMS VIBRATION VELOCITY

	Plate	Original	65-30	75-30
RMS $[m/s]^2$	$3.04*10^{-4}$	$1.02*10^{-4}$	$1.09*10^{-4}$	$1.13*10^{-4}$

FE model has then been validated by a correlation with the test results, Fig. 4, 5. The validation has been conducted for configuration in free-free condition, correlating modal analysis results in terms of modal frequencies and mode shapes referring to Lanczos method implemented in SOL 103 [5]. A quick model updating has been carried out mainly to match better the first and second flexural modes with the experimental results, which were used as reference conditions in previous studies to estimate the structural damping [4], [20].

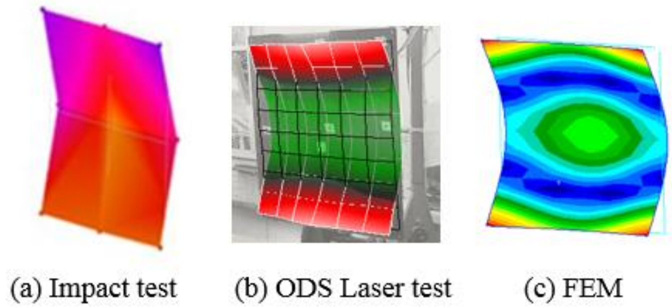

Fig. 4 First mode shape correlation

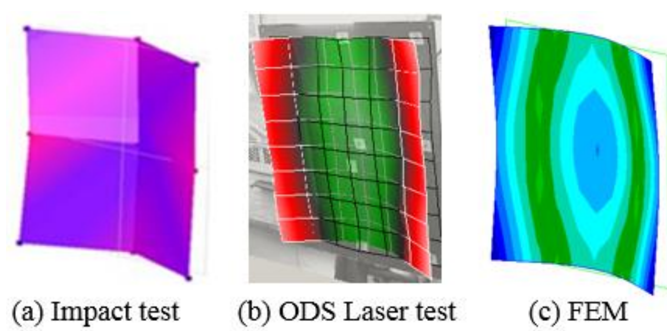

Fig. 5 Second mode shape correlation

The comparison between the numerical results, obtained by MSC Nastran®, and experimental ones both by the laser ODS (Operational Deflection Shapes) that hammer test, is reported in Table IV - V.

TABLE IV. NATURAL FREQUENCY CORRELATION, FIRST MODE SHAPE

	LMS TestLab	Polytec 400	MSC Nastran
Plate	47.8 Hz	48 Hz	47.8 Hz
Original	45.160 Hz	45.2 Hz	45.556 Hz
65-30	45.607 Hz	45.6 Hz	45.649 Hz
75-30	45.951 Hz	45.9 Hz	45.991 Hz

TABLE V. NATURAL FREQUENCY CORRELATION, SECOND MODE SHAPE

	LMS TestLab	**Polytec 400**	**MSC Nastran**
Plate	66 Hz	66 Hz	65.35 Hz
Original	63.2 Hz	63 Hz	63 Hz
65-30	62.1 Hz	62.3 Hz	62.3 Hz
75-30	62.4 Hz	62.6 Hz	62.5 Hz

Relying upon these results, it can be concluded that the updated numerical model can be considered validated for further vibro-acoustic analysis.

V. DAMPING RATIO MEASUREMENT

The identification and characterization of the different noise sources play a key role in the NVH (Noise Vibration Harshness) assessment of a vehicle [22]. The experimental measurements performed in this research have allowed for estimating the damping properties of innovative viscoelastic foams by means of different proven techniques. In the vibration test, Fig. 6, the frequency response measurements lead to observe a significant reduction of the resonance peak mainly due to the 65-30 and 75-30 foams, Fig. 7.

(a) (b)

Fig. 6 Vibration test set-up: instrumentation (a), test rig (b)

The half-power bandwidth method (HPB) is very suitable to evaluate the modal damping factor ζ from frequency domain close to the resonance region: two point corresponding to 3 dB down from the resonance peak are considered for the calculation of (1):

$$\varsigma = \frac{f_2 - f_1}{f_0} \tag{1}$$

Where f_1 and f_2 represent the cut-off frequencies at the two points with an amplitude of 3 dB under the resonance value, f_0 is the value of the natural frequency, Fig. 8.

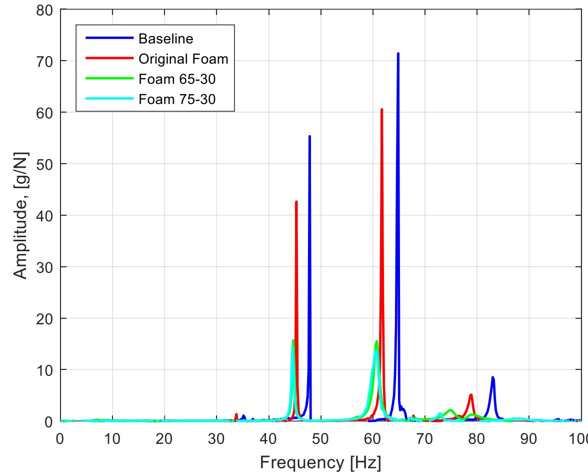

Fig. 7 Frequency Response Function (FRF), range [0; 100 Hz]

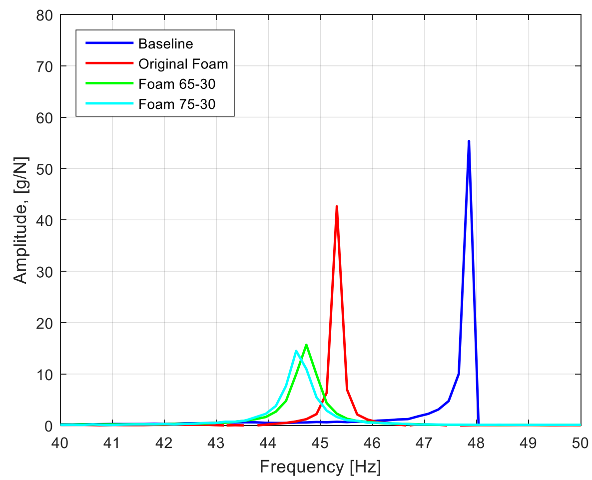

Fig. 8 Frequency Response Function (FRF), first mode detail

The maximum level of response for each resonance frequency is obtained immediately after the impact, while the amplitude decays thereafter at a speed proportional to the structural damping factor ξ, which can be calculated according to the relationship (2) in time domain, Fig. 9:

$$\xi = \frac{\delta}{\sqrt{\delta^2 + 4\pi^2}} \tag{2}$$

In which δ is the logarithmic decrement (LD) calculated between two consecutive peaks, g_i and g_{i+1}, of the acceleration time history (3):

$$\delta = \ln\left(\frac{g_i}{g_{i+1}}\right) \tag{3}$$

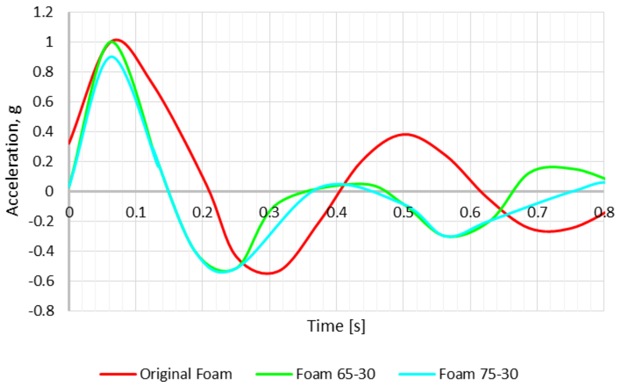

Fig. 9 Acceleration time response

Therefore, the dynamic test has been simulated on the numerical model, considering a force spectrum applied to the plate central node in free-free condition, Fig. 10 [5].

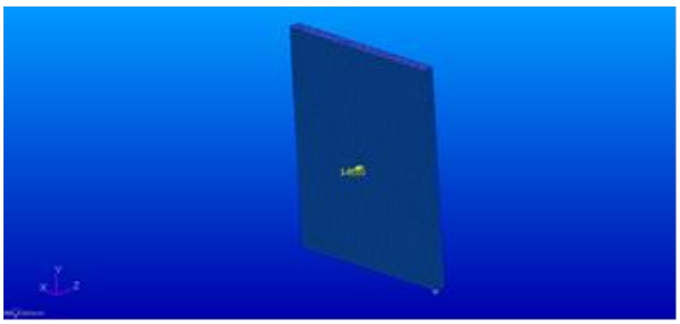

Fig. 10 Driving point on FE model

So the transfer function g/N and the vibration speed spectrum have been computed by means of SOL 111 [5], in correspondence of the Z d.o.f. (degree of freedom) of the monitoring point, Fig. 11, 12. The important outcomes observables from a frequency response analysis usually include the displacements, velocities, and accelerations of grid points as well as the forces and stresses of elements. The computed responses are complex numbers defined as magnitude and phase with respect to the applied force or as real and imaginary components [5].

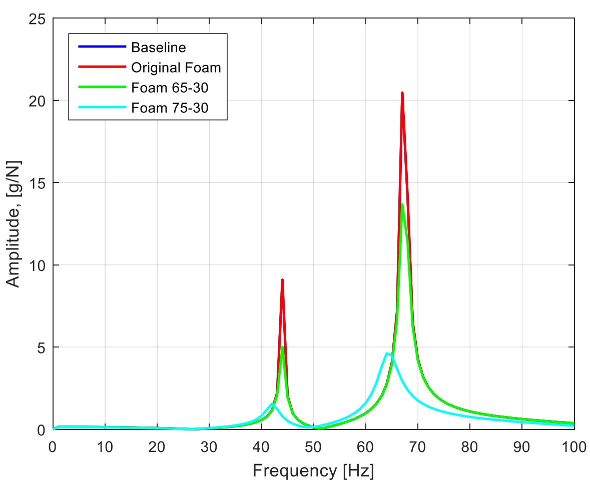

Fig. 11 Frequency Response Function (FRF), SOL 111

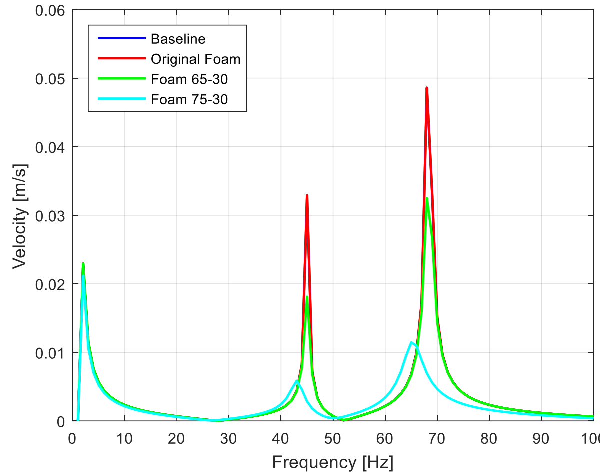

Fig. 12 Vibration velocity spectrum, SOL 111

The acoustic testing have been performed to assess the dissipative effects at high frequency range too. A white noise signal has been used as input by means of a piezoelectric exciter, obtaining a pressure excitation on the panel measured then by a microphone. All the pressure measurements are reduced at a fixed distance about 50 cm from the panel, as shown in Fig. 13 – 16, very close to this one in order to avoid all the environment influence: so the average SPL (Sound Pressure Level) has been estimated [6].

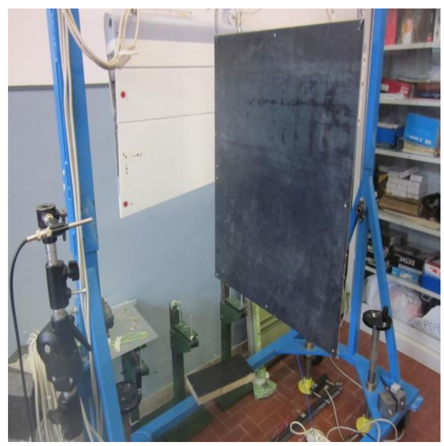

Fig. 13 Acoustic test set-up, Plate

Fig. 14 Acoustic test set-up, Plate with Original Foam

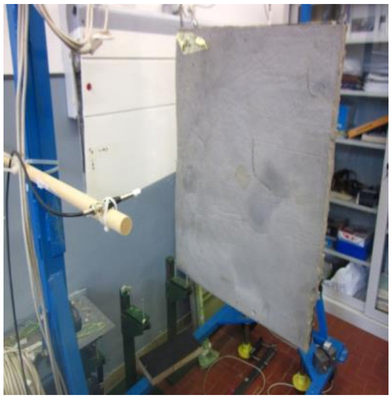

Fig. 15 Acoustic test set-up, Plate with Foam 65-30

Fig. 16 Acoustic test set-up, Plate with Foam 75-30

The SPL measure, whose spectral pattern is represented in Fig. 17, was made according to the expression (4):

$$SPL = 20 * Log\left(\frac{P}{P_0}\right) \qquad (4)$$

In which:

- P: acoustic pressure [Pa];
- P_0: reference pressure [$20*10^{-5}$ Pa].

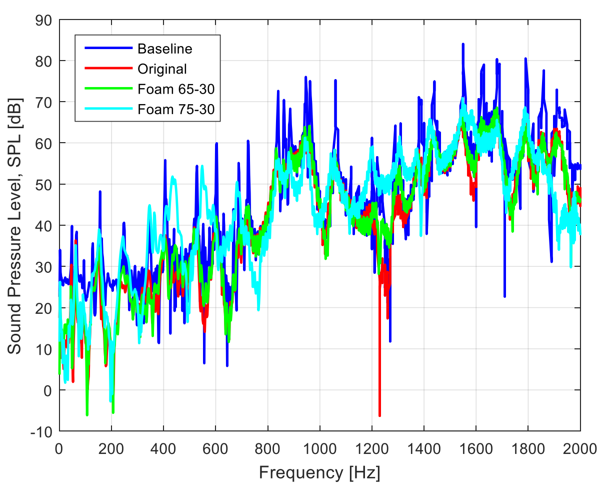

Fig. 17 Sound Pressure Level (SPL) spectral distribution

It is then possible to evaluate the loss factor by the analysis of SPL time history, Fig. 18. The loss factor η, which is related to damping ratio ζ by a linear function, has been calculated according to the expression (5), on the basis of the reverberation time RT_{20}. The results, listed in Table VI, show a very good correlation with the modal damping ratio, estimated in [4], both by half-power bandwidth that by logarithmic decrement methods.

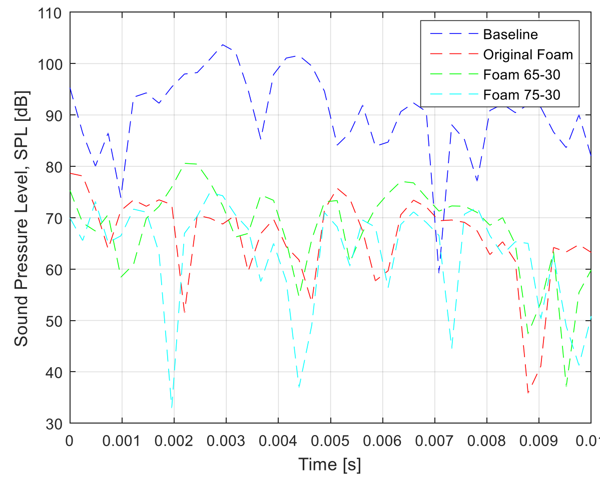

Fig. 18 Sound Pressure Level (SPL), time domain

$$\eta = \frac{2.2}{f_0 * RT_{20}} = 2\varsigma \qquad (5)$$

TABLE VI. DAMPING EFFECT ESTIMATION

Method	Original	65-30	75-30
HPB, ζ	0.19	0.56	0.60
LD, ξ	0.15	0.48	0.56
RT_{20}, ζ	0.131	0.52	0.58

VI. RADIATED SOUND POWER ESTIMATION

Noise and vibration control is a demanding task in many engineering applications such as automotive engineering. Computation of the equivalent radiated power (ERP) is a simplified method to gaining information about maximal possible dynamic radiation of components and panels for specific excitations in frequency response analysis [27]. The sound power radiation of each structural configuration has been predicted as follows. Radiation efficiency σ is defined in (6) as the proportionality between radiated sound power Π_{rad} and the square of surface normal velocity $\langle v^2 \rangle$ averaged over time and radiating surface S:

$$\Pi_{rad} = \sigma \rho_0 c S \left\langle v^2 \right\rangle \qquad (6)$$

in which ρ_0 is the density of the acoustic fluid, and c is the speed of sound [7]. In this preliminary assessment, a radiation efficiency of the modes of a steel simply-supported plane plate has been selected by empirical datasheet, Fig. 19.

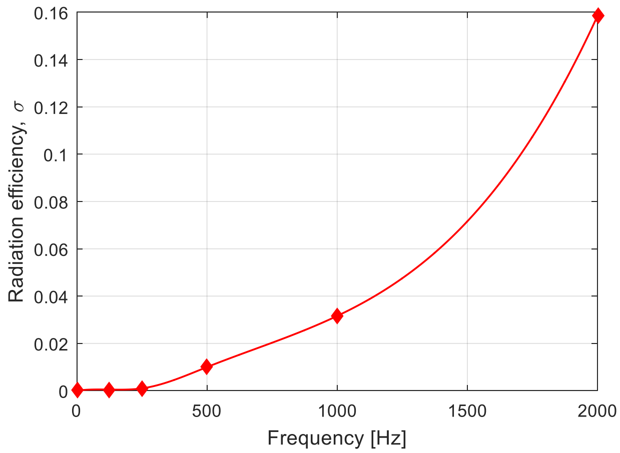

Fig. 19 Radiation efficiency: steel plate (0.5*0.5*0.0015 m)

The RMS vibration velocity were obtained within the non-contact dynamic measures carried out by means of laser scanning vibrometry.

The spectral trend of the radiated power has been therefore extrapolated by data post-processing in Matlab® environment. Fig. 20 shows the global comparison of linear acoustic power in the dynamic range [0; 200 Hz], while Fig. 21 shows a direct comparison of each configuration in the range [200; 2000 Hz].

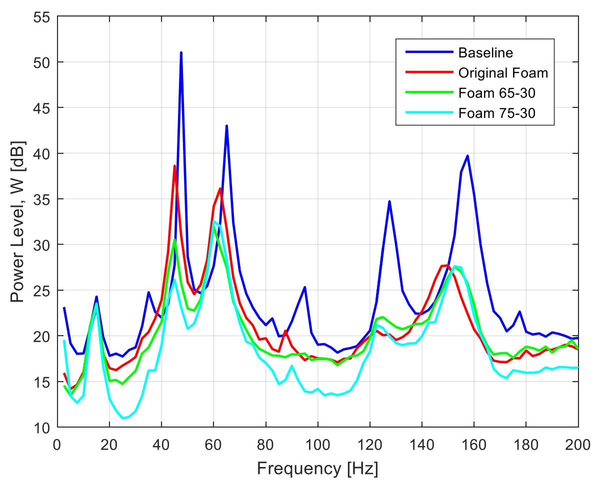

Fig. 20 Radiated sound power, range [0; 200 Hz]

The relevant outcome in Fig. 20 is the identification of the actual resonant frequencies of each configuration. By the spectral analysis, it is still evident a greater damping effect of viscoelastic foams. In this research context, innovative materials, targeted at the reduction of the car-body floor, have been analysed and compared to "standard one"; in the specific, viscoelastic foams will be investigated as a valid alternative to conventional add-on damping element as generally used in these applications.

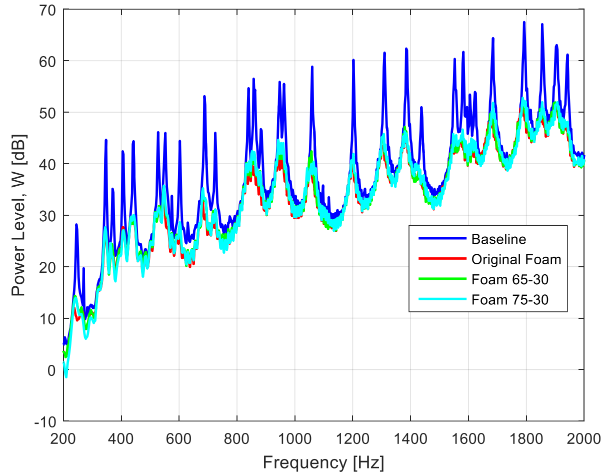

Fig. 21 Radiated sound power, range [200; 2000 Hz]

It is important to underline that these data extrapolation are referred to the plate vibration level under the effect of different foams measured at the free side (where were not glued the foams); so, they only take into account the damping effect of foams and not eventual insulating effect on the noise propagation. In any case, these results are only indicative and must be properly correlated with "near-field" acoustic measurements by a PU (pressure-velocity) probe, which permits to get with a better resolution degree, the sound power level emitted. The effect of add-on damping may be also highlighted by the comparison of structural/acoustic frequency response functions as those reported in next Fig. 22. They have been obtained by a hammer impact excitation of the structure and normalized to the impact force. They can be read as an indicator of the body acoustic sensitivity of different configurations. In addition, this comparison explains the extra damping effect that viscoelastic foam introduces into the structural system.

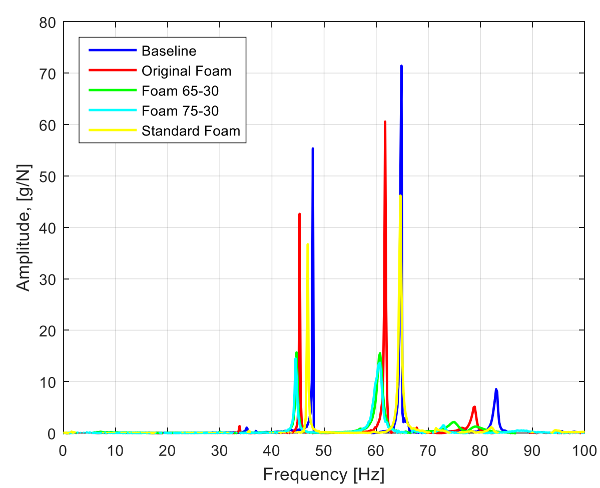

Fig. 22 Frequency Response Function (FRF) of different tested configurations

VII. VISCOELASTIC FOAM PROTOTYPE DESIGN

The vibro-acoustic behavior of viscoelastic foams has been verified to be better than the original one. On the otherside, the latter has good stiffness characteristics, which can increase the structural stability of the whole floor covering system. It has been targeted the development of a prototype with static performances as close as possible to the original foam and with an improved damping coefficient. In Fig. 23, the prototype design performed in CATIA V5® is showed. Within the testing phase, it has been analyzed in the same way of the other ones, Fig. 24.

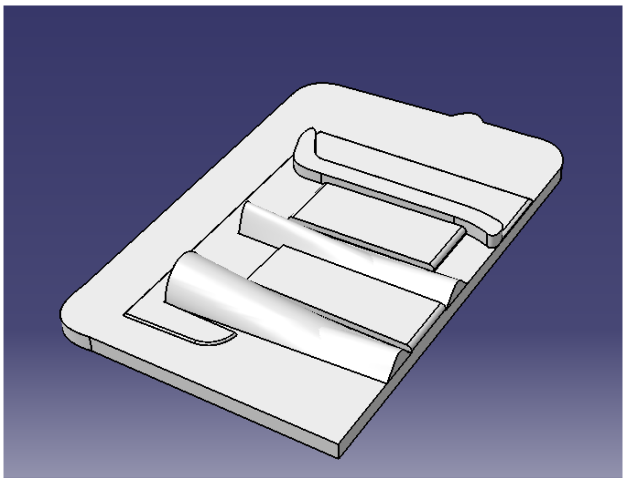

Fig. 23 Prototype 3-D CAD

It considered the pressure measurements recorded by the microphone placed at a distance of 50 cm from the plate-foam system. Applying the relationship (4) indicated is obtained the SPL (Sound Pressure Level) spectral trend, Fig. 25, 26.

Fig. 24 Acoustic test set-up, Plate with foam prototype

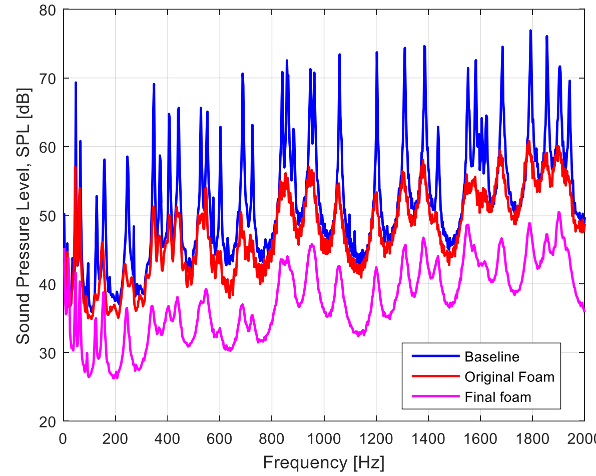

Fig. 25 Sound Pressure Level (SPL) spectral distribution

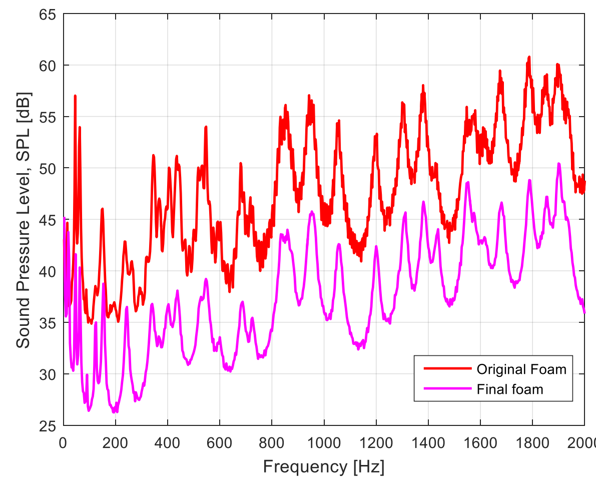

Fig. 26 Sound Pressure Level (SPL) spectral distribution

Starting by the vibration velocity it can be seen the improved response of the final foam. Through these speed values, the radiated sound power has been assessed from the plate-side by implementing the equation (6), with the same radiation efficiency trend, Fig. 27, 28.

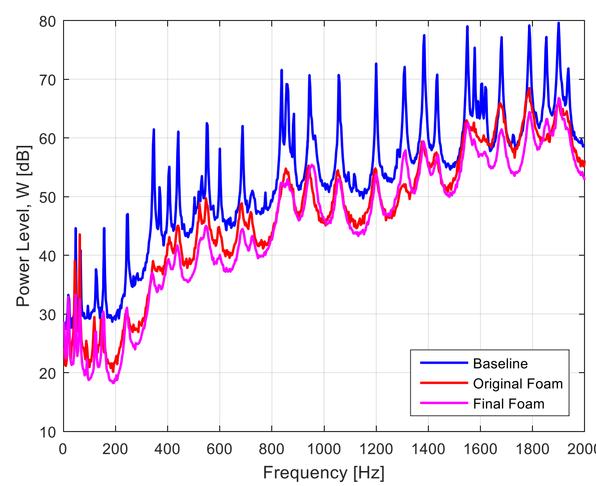

Fig. 27 Radiated sound power

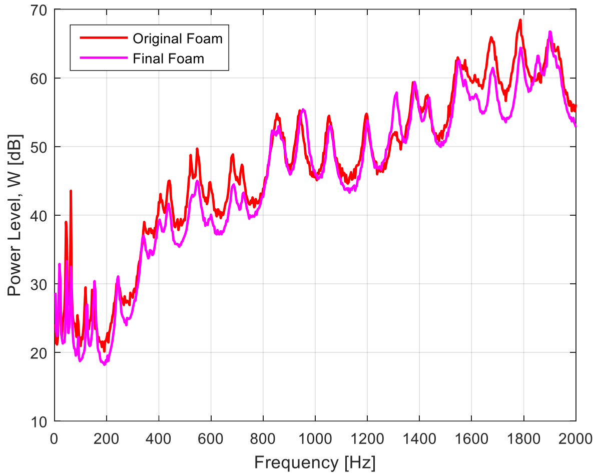

Fig. 28 Radiated sound power: foams behavior trade-off

Trade-off results may be summarized in the next Table VII, in terms of overall radiated power.

TABLE VII. DAMPING EFFECT ESTIMATION

Baseline	**Original Foam**	**Final Foam**
66.98 dB	59.62 dB	57.73 dB

The numerical model has then been validated by a correlation with the experimental results. The validation has been performed for configuration in free-free boundary condition, correlating modal analysis results in terms of modal frequencies and mode shapes referring to Lanczos method [5]. So the transfer function g/N and the vibration speed spectrum have been computed by means of SOL 111 [5], in correspondence of the Z d.o.f. (degree of freedom) of the driving point, Fig. 29, 30. This method uses the mode shapes of the structure to reduce the size, uncouple the equations of motion and make the numerical solution more efficient. Since the mode shapes are typically computed as part of the characterization of the structure, modal frequency response is a natural extension of a normal modes analysis [5].

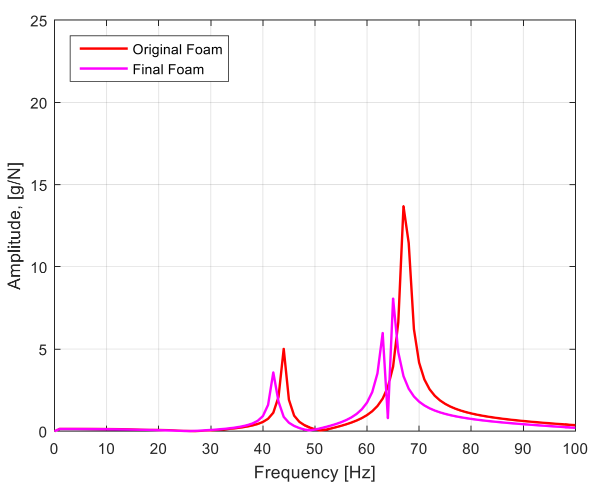

Fig. 29 Frequency Response Function (FRF), SOL 111

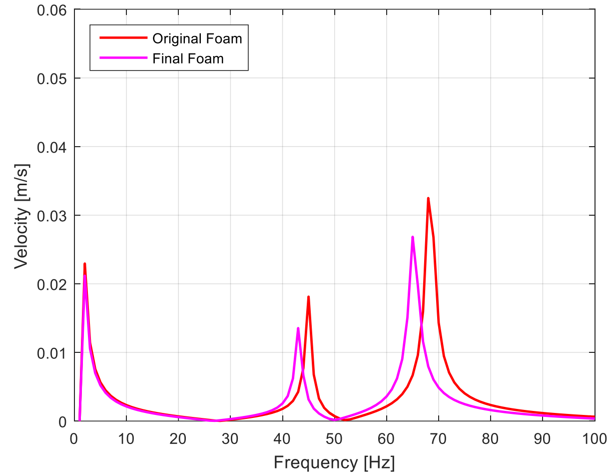

Fig. 30 Vibration velocity spectrum, SOL 111

In the acoustic "near-field", particle velocity is the dominant acoustic property. In the "near-field", the acoustic particle velocity and the structural velocity coincide. This makes the Scanning Probe very suitable for non-contact measurements of vibrations, Fig. 31. The Scanning Probe is designed in such a manner that the probe can be used very close to the surface and capable of reaching measurement locations that are difficult to access. Measuring acoustic particle velocity in the "near-field" can be beneficial for more than only non-contact vibration [17].

Fig. 31 PU probe, Microflown Technologies®

The particle velocity of prototype is compared with the original one, in the case of white-noise excitation provided by a piezo-electric input source, Fig. 32.

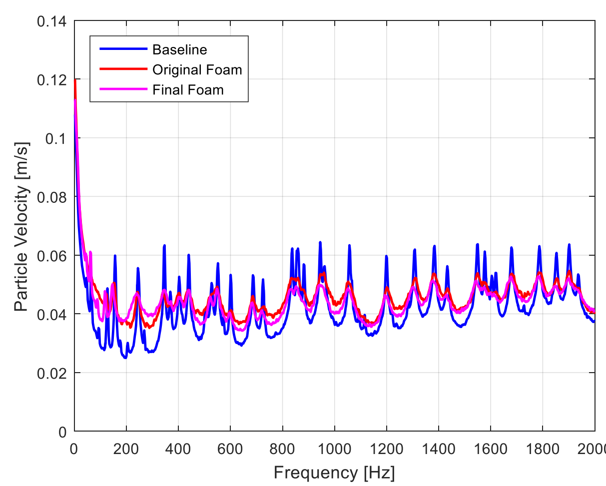

Fig. 32 Particle velocity by means of PU probe

Scan & Paint is a quick and easy method to determine the position and strength of sound sources even in reverberant conditions. The Scan & Paint method is straightforward; the surface is scanned with the PU probe while a camera films the scanning procedure. The recorded video and audio data are automatically synchronized by the software and the measurements are directly ready to be processed. In the post-processing sequence, the position of the probe is extracted from each video frame with an auto-tracking function in the software that automatically recognizes the position of a color marking on the probe. At each measurement point, acoustic quantities are calculated from the corresponding audio data section, Fig. 33 [17-18].

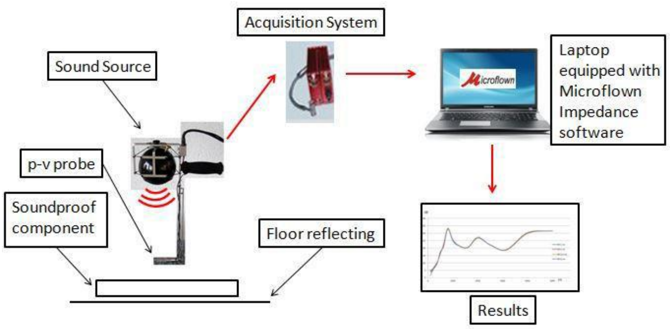

Fig. 33 PU probe, test setup

Finally, a high-resolution sound color map is generated as a result, Fig. 34, 35.

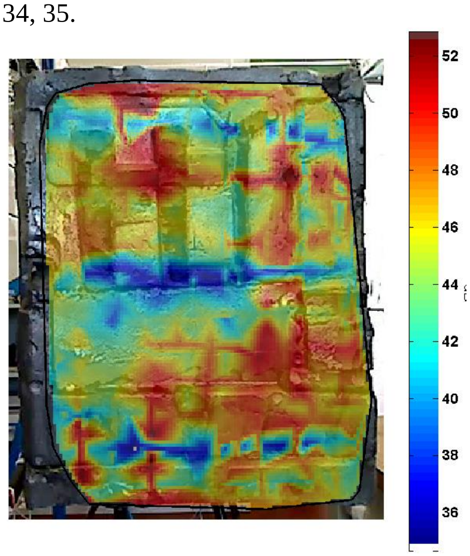

Fig. 34 Acoustic intensity, Final Foam

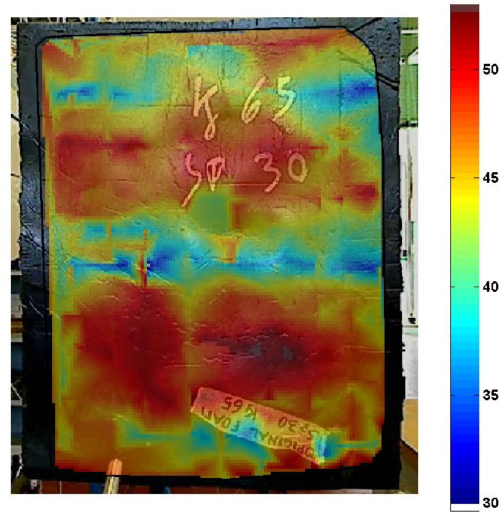

Fig. 35 Acoustic intensity, Original Foam

The "near-field" analysis confirms the better acoustic behaviour of the prototype compared to the original foam.

For the experimental measurement of the static stiffness, a mono-directional compression test has been carried out on the foam prototype by different load conditions, in the same way described in [4]. In Fig. 36, the trend of load versus static displacement has been plotted with reference to such case of investigation.

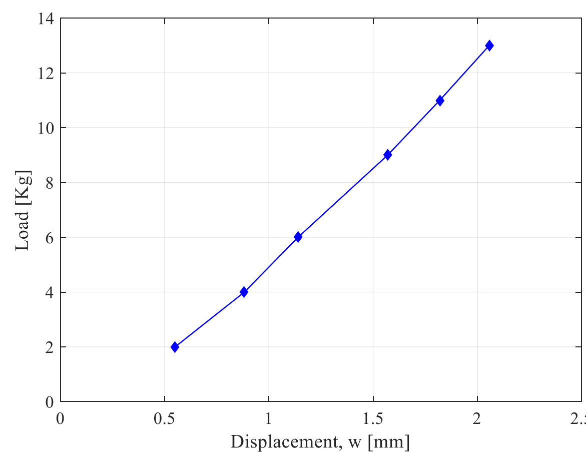

Fig. 36 Load-Displacement curve, Compression test

The test outcomes are listed in the next Table VIII, in terms of static stiffness (7):

$$K = \frac{F}{w} = \frac{[N]}{[mm]} \qquad (7)$$

TABLE VIII. STATIC STIFFNESS MEASURE

Load [Kg]	w [mm]	K [N/mm]
2	0.55	35.67
4	0.88	44.46
6	1.14	51.63
9	1.57	56.24
11	1.82	59.29

Load [Kg]	w [mm]	K [N/mm]
13	2.06	61.89

Therefore, it has been developed a prototype by a compromise of the mechanical characteristics of standard foams that of viscoelastic ones. The prototypes of the front and rear passenger compartment, Fig. 37, were made using the same production equipment with which the company realizes the standard passenger compartment carpet. The foam has been coupled with "heavy layers" of EPDM (Ethylene-Propylene Diene Monomer) to enhance vibro-acoustic absorption as well as contributing to assembly efficiency [19].

Fig. 37 Prototype carpets

VIII. Application in the Automotive Engineering

The EPDM rubbers are a family of synthetic rubbers of the group M according to the classification DIN / ISO 1629 and ASTM D 1418-19. The M group includes elastomers containing polymer chains of polymethylene type without unsaturations (double or triple bonds).
For a greater completeness of the results, both the final foam that the original one have been tested in such way as described until now, with the EPDM layer in order to simulate the interior floor coating of a potential vehicle. The experimental set-up of each configuration is represented in Fig. 38, 39.

Fig. 38 Final Foam with EPDM layer

Fig. 39 Original Foam with EPDM layer

So it has been acquired the vibrational velocities in the windowing range 0-2000 Hz by means of the laser technique (Polytec PSV-400®), Fig. 40.

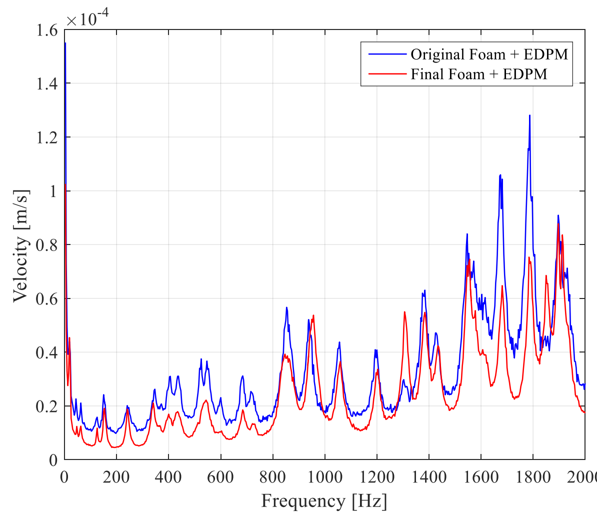

Fig. 40 Vibration velocity spectrum, Laser Test

Considering the vibration velocity spectrum, the final foam gives the best dynamic behavior. Starting from these values, the radiated sound power, Fig. 41, has been estimated from the plate-side by implementing the equation (6), with the same efficient trend represented in the Fig. 18.

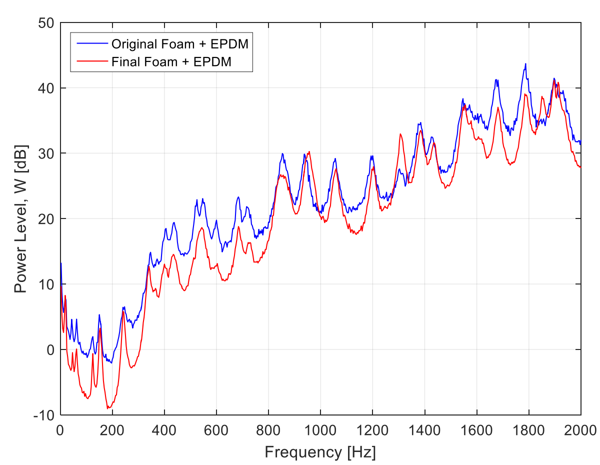

Fig. 41 Radiated sound power

The Sound Pressure Level (SPL) in Fig. 42, calculated by the relationship (4), has been deduced to compare the trend of Insertion Loss of the prototype foam with the original one.

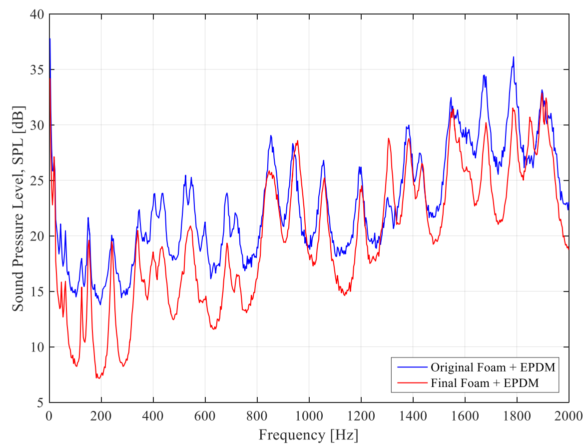

Fig. 42 Sound Pressure Level (SPL) spectral distribution

At the monitoring point, Insertion Loss (IR) has been calculated as the difference between the reference SPL and the SPL of the coupled system, by the equation (8), Fig. 43:

$$IR = SPL(Baseline) - SPL(Coupled) \tag{8}$$

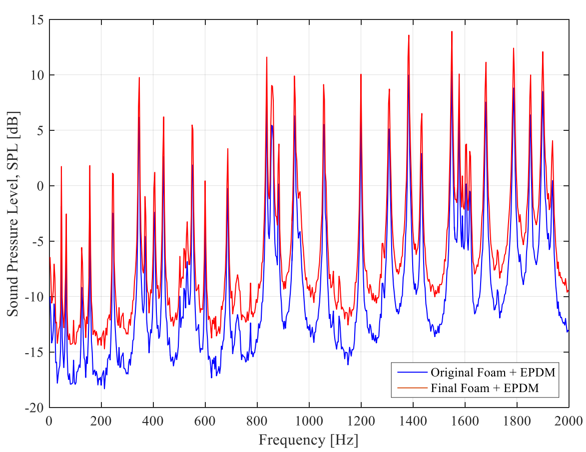

Fig. 43 Insertion Loss evaluation

From the graph in Fig. 43, it is possible to assess the improvement in the acoustic performance due to the insertion of viscoelastic patch into the structural system.

IX. Conclusion and Future Developments

In recent years, the acoustic problem has represented a rate of the quality of human life. Aside legal obligation, all the big automotive manufactures are directed to solutions for better passenger's acoustic comfort. The reduction of vibration and noise in and across several components and modules of the automotive, such as the panels, doors, engine covers, seats, and others, is of primary importance [15]. The increasing role of lightweight materials in many engineering fields is related to several advantages, which provide like the fuel burn reduction and noise abatement. The importance of aspects relating to environmental protection and to the tutelage of comfort inside the vehicle have made it increasingly necessary in-depth research on issues such as noise and vibrations. This research has therefore presented a possible and viable solution to the problems related to noise, showing the results relating to the use of innovative viscoelastic foams. The adoption of these foams could lead to the reduction of the overall weight due to the elimination of the extra treatment used today for this specific purpose, also these materials led to significant benefits in terms of vibration damping, which was measured approximately four times compared to the standard commercial solution [1]. Within this research project, an integrated experimental-numerical procedure to evaluate the vibro-acoustic performances of an innovative viscoelastic carpet has been outlined and implemented. Particularly, the experimental tests have been performed to investigate the dynamic parameters of the system and to measure the sound pressure level and vibration velocity in order to predict the radiated power [15]. Moreover, "near-field" acoustic measurements have been carried out according to directly estimate the noise intensity from the panel, by means of a piezo-electric excitation and a PU (pressure-velocity) probe as a transducer for the acquisition. The experimental results have been collected into a well-correlated rational database for further studies on the fluid-structural interaction between the damping carpet floor and air by means of MSC Actran® software and VA One® Acoustic BEM (Boundary Element Method) Model. For an investigation closer to a potential automotive application, the prototype has been tested coupling it with the EPDM layer in order to simulate the interior floor coating of the vehicle. The damping enhanced performance of final foam reflects into a lower acoustic noise transmitted inside the environment, all over the investigated frequency range [28].

References

[1] M. Viscardi, M. Arena, D. Siano, "Experimental and numerical assessment of innovative damping foams," *International Journal of Mechanics*, Vol. 10, pp. 329-335, 2016.

[2] D. Roylance, "Mechanical properties of materials," pp. 37-40, 2008.

[3] D. Roylance, "Engineering viscoelasticity," pp. 8-14, October 24, 2001.

[4] M. Viscardi, M. Arena, "Experimental Characterization of Innovative Viscoelastic Foams," *Mechanics, Materials Science & Engineering*, Vol. 4, pp. 7-14, doi:10.13140/RG.2.1.5150.6325, May 2016.

[5] MSC-MD/NASTRAN®, Software Package, Ver. R3-2006, "Reference Manual".

[6] M. Di Giulio, G. Arena, A. Paonessa, "Vibro-acoustic behaviour of composite fuselage structures using embedded viscoelastic damping treatments," *AIDAA Congress*, Milan, June 29-July 3, 2009.

[7] C.H. Oppenheimer, S. Dubowsky, "A radiation efficiency fon unblaffed plates with experimental validation," *Journal of Sound and Vibration*, pp. 473-48, 1997.

[8] D. Siano, M. Viscardi, R. Aiello, "Experimental and Numerical Validation of an Automotive Subsystem through the Employment of FEM/BEM Approaches," *Energy Procedia*, Vol. 82, pp. 67-74, doi: 10.1016/j.egypro.2015.11.884, 2015.

[9] L. Beranek, "Noise Reduction," Mc Graw-Hill, 1960.

[10] L. Beranek, I. L. Ver, "Noise and Vibration Control Engineering: Principles and Applications," Whiley & Sons, New York, NY, USA, 1992.

[11] D. I. G. Jones, "Handbook of Viscoelastic Vibration Damping," Whiley & Sons, New York, NY, USA, 2001.

[12] ASTM E 756-04e1, "Standard Test Method for Measuring Vibration-Damping of Materials," February 22, 2005.

[13] A. D. Nashif, D. I. G. Jones, J. P. Henderson, "Vibration Damping," John Wiley and Sons, Inc., New York, NY, USA, 1985.

[14] M. Di Giulio, A. Paonessa, E. Monaco, S. Migliore, "Aircraft composite structures modelling and vibro-acoustic optimisation analysis," *AIDAA Congress*, Volterra, Italy, September, 19-22, 2005.

[15] D. Siano, M. Viscardi, R. Aiello, "Sensitivity analysis and correlation Experimental/Numerical FEM-BEM for Noise Reduction assessment of an engine beauty cover," *Energy Procedia*, Vol. 81, pp. 742-754, 2015.

[16] D. Siano, M. Viscardi, M. A. Panza, "Acoustic optimization of a high-speed train composite sandwich panel based on analytical and experimental Transmission Loss evaluation integrated by FE/Test correlation analysis," *Energy Procedia*, Vol. 81, pp. 689-703, 2015.

[17] www.microflown.com.

[18] The Microflown E-Book.

[19] W. Fung, M. Hardcastle, "Textiles in Automotive Engineering," Woodhead Publishing, 2001.

[20] M. Viscardi, M. Arena, G. Barra, L. Guadagno, "Structural Performance Analysis of Smart Carbon Fiber Samples Supported by Experimental Investigation," *International Journal of Mechanics*, Vol. 10, pp. 376-382, 2016.

[21] M. Viscardi, P. Napolitano, M. Arena, "Simulation and Experimental Validation of Fatigue Endurance Limit of Copper Alloy for Industrial Applications," *International Journal of Mathematical Models and Methods in Applied Sciences*, Vol. 10, pp. 340-346, 2016.

[22] G. Vanore, A. Paonessa, G. Arena, A., "Improved testing methods for structural damping evaluation," *AIDAA Congress*, Milan, June 29–July 3, 2009.

[23] E. Balmes, M. Corus, S. Baumhauer, P. Jean, J. P. Lombard, "Constrained viscoelastic damping, test/analysis correlation on an aircraft engine," *Proceedings of the IMAC-XXVIII*, Vol. 3, pp. 1177-1185, 2011.

[24] M. D. Rao, "Recent Applications of Viscoelastic Damping for Noise Control in Automobiles and Commercial Airplanes," *Journal of Sound and Vibration*, Vol. 262, No. 3, pp. 457-474, 2003.

[25] E. Balmes, "Frequency domain identification of structural dynamics using the pole/residue parametrization," *International Modal Analysis Conference*, pp. 540-546, 1996.

[26] A. Plouin, E. Balmes, "A test validated model of plates with constrained viscoelastic materials," *International Modal Analysis Conference*, pp. 194-200, 1999.

[27] K. Wiechmann, J. Hiller, "Evaluation and visualization of equivalent radiated power in μETA," *4th ANSA & μETA International Conference*, Thessaloniki, Greece, June 1-3, 2011.

[28] M. Klaerner, S. Marburg, L. Kroll, "FE based measures for structure borne sound radiation," 43rd International Congress on Noise Control Engineering, November 16-19, 2014.

Massimo Viscardi was born in Naples (Itay) on the 28th of January 1970**,** where he graduated in Aerospace Engineering. Assistant professor of Aerospace Structural Testing and of Experimental Vibroacoustic at University of Naples, his research is mainly dedicated to innovative measurement and control technologies for acoustic and vibration phenomena.
He has been involved in many EU project within the 6th, 7th framework as well H2020 contest.
Expert evaluator for the Ministry of Economic Development, Italy. He is member of several association and has been author of about 70 scientific papers as well as referee of many scientific journals.

Department of Industrial Engineering
University of Naples
massimo.viscardi@unina.it

Maurizio Arena was born in Naples (Itay) on the 28th of December 1989**,** where he graduated in Aerospace Engineering. He is now involved in the Ph.D. program at University of Naples, where he work within the Smart Structures group. His main area of interest are: smart structures, noise and vibrations control to improve the flight conditions. He is tutor of several thesis and junior lecturer of advanced aircraft structures.

Department of Industrial Engineering
University of Naples
maurizio.arena@unina.it

Daniela Siano was born in Naples - Italy, and graduated in Aeronautical Engineering at the University of Naples "Federico II", Italy in 1994. Until 2001, she was researcher in acoustic and vibration department at C.I.R.A. (Italian aerospace Research Center). From 2001 until now, she is a Researcher at National Research Council of Italy (CNR) in the field of Acoustic and Vibration in transport field. She is responsible of Acoustic and Vibration Laboratory in her Institution. Expert evaluator within the EU 6th and 7th Framework Research Programme, in Transport-Aeronautics in 2006 and 2007. Project expert evaluator in Ministry Economic Development, Italy. Referee for some International Journals and session organizers collaborating with SAE conferences. She is author of about 75 Scientific Papers published on International Journals and Conferences Proceedings and editor of two Scientific Books : 'Noise Control, Reduction and Cancellation Solutions in Engineering', ISBN 978-953-307-918-9, 308 pages, DOI: 10.5772/1375, and 'Fuel Injection', ISBN 978-953-307-116-9, 262 pages, DOI: 10.5772/55402. She is tutor of several thesis and Ph.D. thesis, as well.

CNR (National Research Council of Italy) - Istituto Motori
d.siano@im.cnr.it