Experimental and numerical assessment of innovative damping foams

M. Viscardi, M. Arena, D. Siano

***Abstract*—**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. 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 vibroacoustic phenomena generated in a vehicle. The performance of these innovative materials have been analyzed both from a numerical standpoint that experimental. Starting from the empirical results of tests carried out in the laboratory, finite element models have been developed in order to have a suitable numerical database for further vibro-acoustic simulations.

***Keywords*— Automotive, Finite Element Model, Non-contact Measurement, Viscoelasticity.**

I. Introduction

THIS work has the purpose to investigate new materials and technologies aimed to reduce the noise and vibrations produced inside motor vehicles. The meaningful growth process and the exponential development related to automotive industry has currently introduced new requirements concerning the fuel consumption and the noise emitted [1]. The awareness on meeting the comfort targets implies a significant evolution of the assessments in vehicle design, aimed to deliver best fuel efficiency with no loss of powers and refinements. In this research context, 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 [2]. 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 [3]. In the next paragraphs will be then describe the main characteristics of the numerical model, the results from tests by non-contact techniques and finally the numerical-experimental correlation of the processed data.

II. 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 behaviour. Each foam has been discretized using HEX8 elements glued on a 2-D (CQUAD) mesh [4], which simulates the metal support plate, Fig. 1. The main characteristics of FE (Finite Element) model, performed in PATRAN – MSC NASTRAN©, are summarized in Table I.

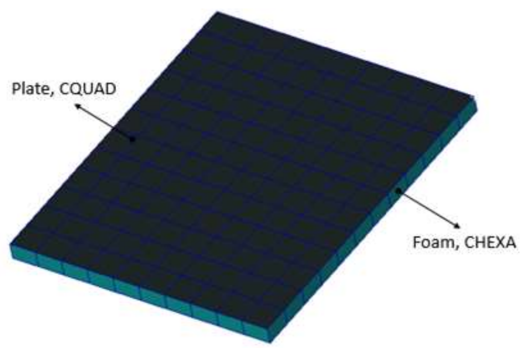


Fig. 1 Structural system mesh

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	Plate	Original	65-30	75-30
ρ [Kg/m^3]	8110	65	65	75
E [MPa]	1161	0.012	0.017	0.525
G [MPa]	Linear	Linear	0.0057	0.1236
ν	0.3	0.3	0.3	0.3

III. MODAL ANALYSIS AND FE MODEL UPDATING

The vibration test has been performed by a scanning Laser Doppler Vibrometer (LDV), Polytec 400©. Such non-invasive technique has allowed the acquisition of the microscopic vibrations at 63 monitoring points and the deflection shape of the entire surface in the bandwidth 0-2048 Hz. The structure was excited with a piezoelectric device, which generated a white noise signal, Fig. 2.

Fig. 2 Monitoring points

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 [5]. 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.

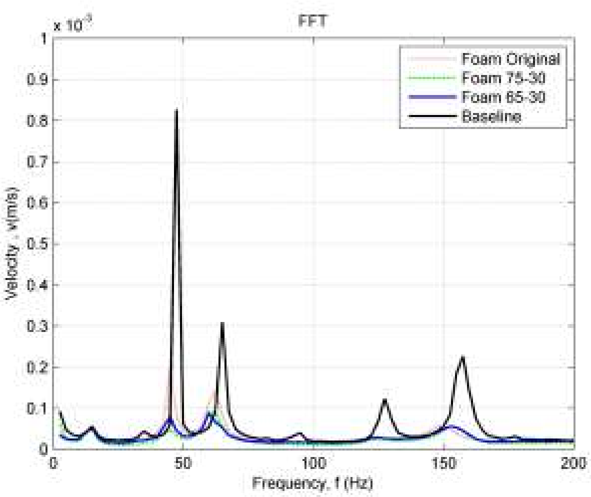


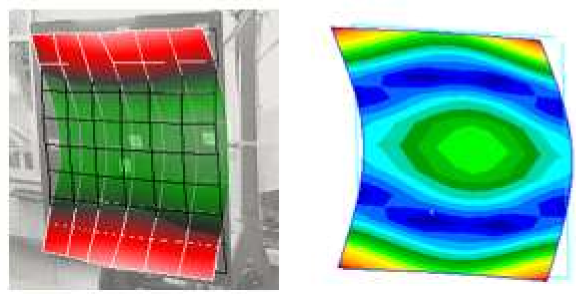
Fig. 3 Vibration velocity spectrum, Laser Test

The sequence of the extracted RMS (Root Mean Square) velocity, associated with each configuration 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

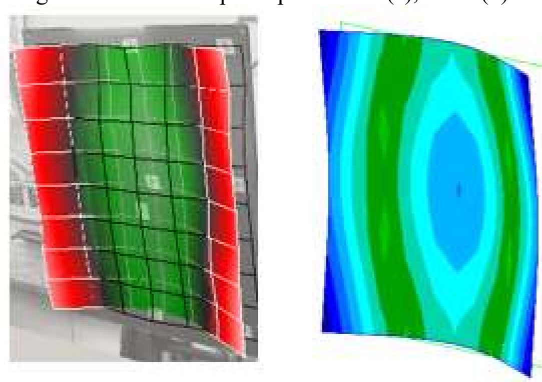
	Plate	Original	65-30	75-30
RMS $[m/s]^2$	$3.04*10^{-4}$	$1.02*10^{-4}$	$1.09*10^{-4}$	$1.13*10^{-4}$

FE model has then been validated by a correlation with the test results, Fig. 4. 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 [4]. 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 [3].



(a) Laser test (b) FEM

Fig. 4 First mode shape: experimental (a), FEM (b)



(a) Laser test (b) FEM

Fig. 5 Second mode shape: experimental (a), FEM (b)

The comparison between the numerical results, obtained by MSC Nastran©, and experimental ones both by the laser 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

On the basis of the results, it can be concluded that the updated numerical model can be considered validated for further vibro-acoustic analysis.

IV. DAMPING RATIO MEASUREMENT

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. 5, 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. 6.

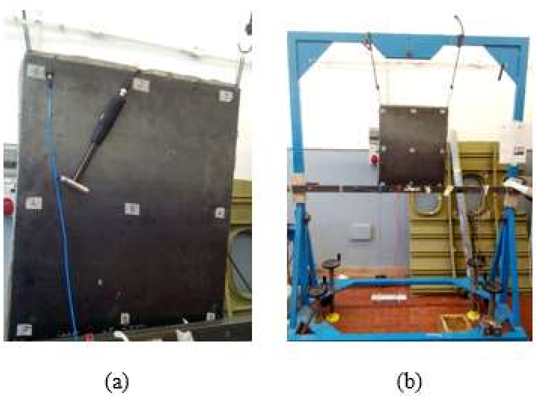


Fig. 5 Vibration test set-up: instrumentation (a), test rig (b)

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. 7.

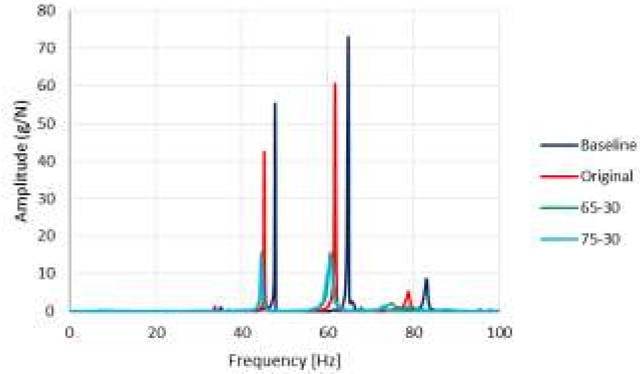


Fig. 6 Frequency Response Function (FRF), range [0; 100 Hz]

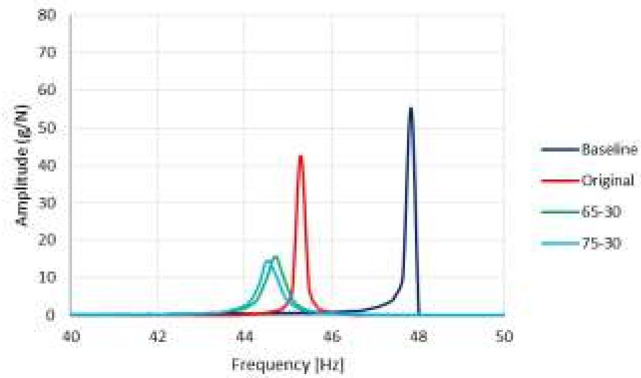


Fig. 7 Frequency Response Function (FRF) about the first mode

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. 8:

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

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):

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

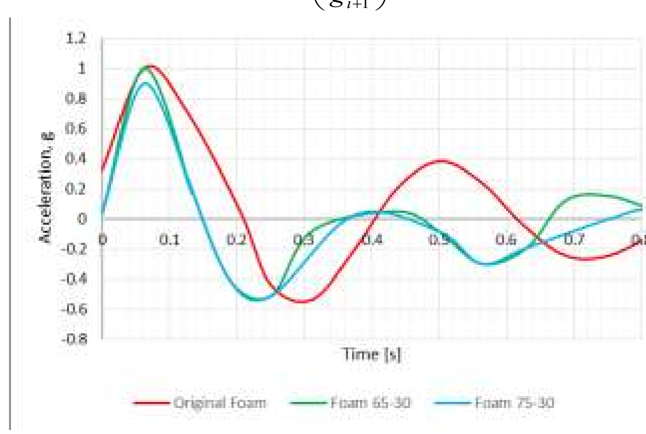


Fig. 8 Acceleration time response

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 by the microphone. All the pressure measurements are reduced at a fixed distance about 50 cm from the panel, as shown in Fig. 9 – 12, very close to this in order to avoid all the environment influence, and the average SPL (Sound Pressure Level) is calculated [5].


Fig. 9 Acoustic test set-up, Plate


Fig. 10 Acoustic test set-up, Plate with Original Foam


Fig. 11 Acoustic test set-up, Plate with Foam 65-30


Fig. 12 Acoustic test set-up, Plate with Foam 75-30

The SPL measure, whose spectral pattern is represented in Fig. 13, was made according to the expression (4):

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

In which:

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

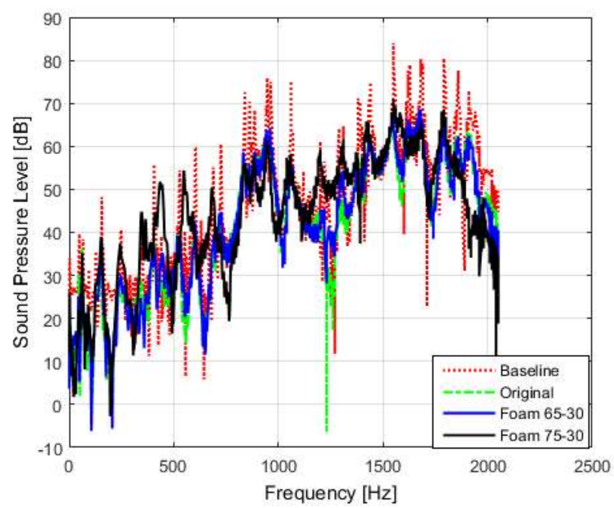


Fig. 13 Sound Pressure Level (SPL) spectral distribution

It is then possible to evaluate the loss factor by the analysis of SPL time history, Fig. 14. 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 [3], both by half-power bandwidth that by logarithmic decrement methods.

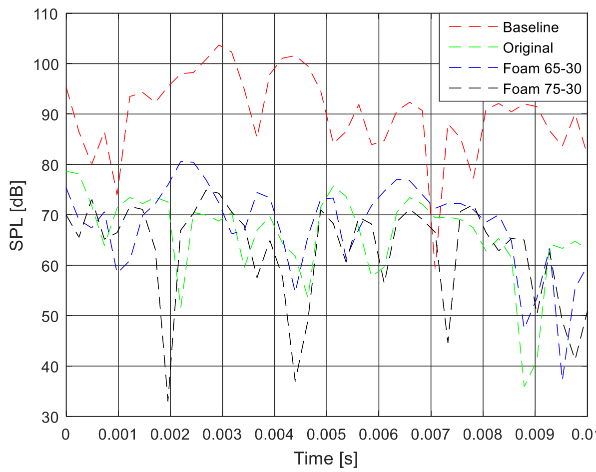


Fig. 14 Sound Pressure Level (SPL), time domain

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

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

V. RADIATED SOUND POWER ESTIMATION

The sound power radiation of each structural configuration has been predicted as follows. Radiation efficiency σ is defined in (5) as the proportionality between radiated sound power Π_{rad} and the square of surface normal velocity $<v^2>$ averaged over time and radiating surface S:

$$\Pi_{rad} = \sigma \rho_0 c S \langle v^2 \rangle \qquad (5)$$

In which ρ_0 is the density of the acoustic fluid, and c is the speed of sound [6]. In this preliminary assessment, a radiation efficiency of the modes of a steel simply supported plane plate has been selected by empirical datasheet, Fig. 15.

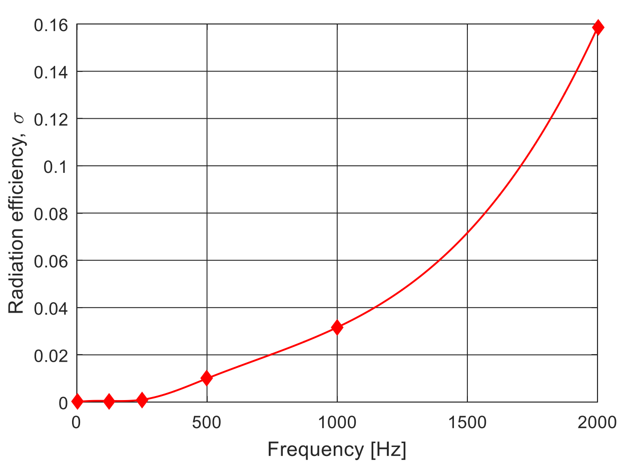


Fig. 15 Radiation efficiency: steel plate (0.5*0.5*0.0015 m)

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. 16 shows the global comparison in linear acoustic power , while Fig. 17 shows a direct comparison of original and viscoelastic foam in dB power.

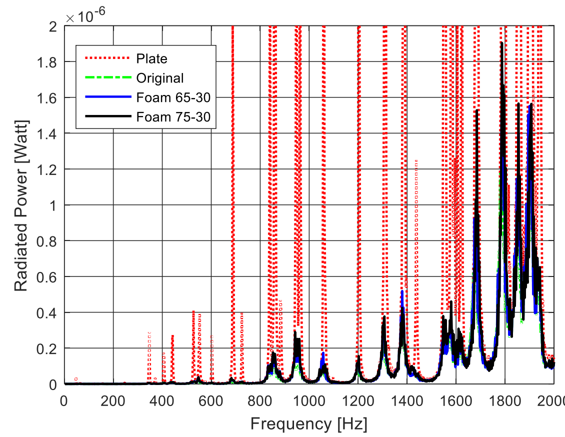


Fig. 16 Radiated sound power

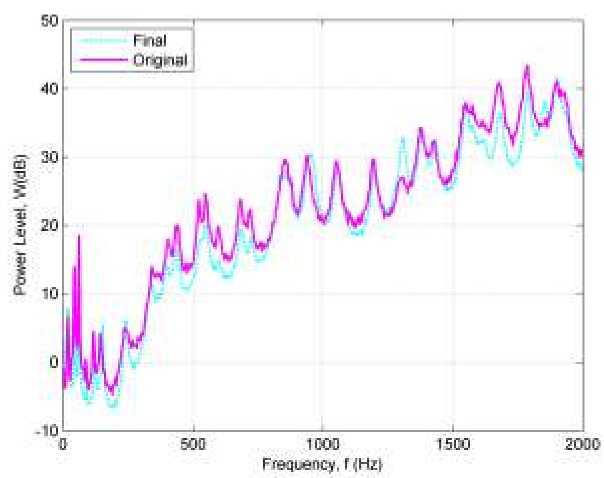


Fig. 17 Radiated sound power: foams behavior trade-off

Data 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

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 p-v (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 function as those reported in next Fig. 18. They have been obtained through an 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 configuration.

Also this comparison give explanation of the extra damping that viscoelastic foam introduce into the system.

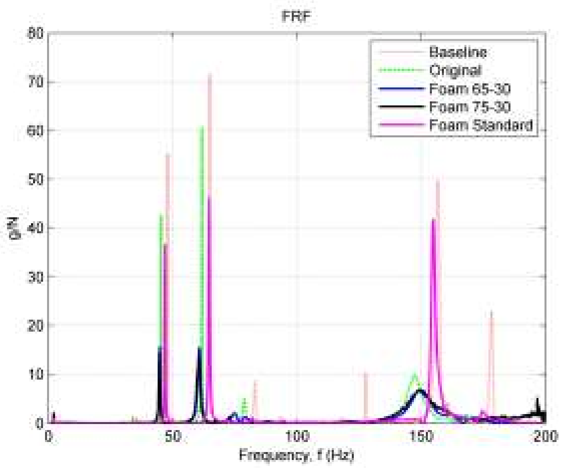


Fig. 18 Vibro-acoustic FRF of different configuration

VI. Conclusion and Future Developments

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 vibration. 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. Lastly, these viscoelastic foams are porous by nature, therefore they have a low density and thus are particularly suitable for lightening applications, representing a solution in the "light-weight" next generation of vehicle. To continue the search and subsequently developing these materials, a numerical acoustic FE model will be developed to be correlated with the experimental and it will be made an acoustic measurements (to directly evaluate the radiated power from the panel), through a piezo-electric as an input source and a p-v (pressure-velocity) probe as a transducer for the "near-field" acquisition.

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