Experimental Characterization of Innovative Viscoelastic Foams

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ABSTRACT. The evolutionary trend in the automotive industry has produced over time numerous performance and aesthetic innovations, however, the exponential development related to transportation technologies also introduced new requirements concerning the environmental impact [1]. The awareness of ecological issues has led to a reorganization of the evaluations and the vehicle design, currently aimed at reducing the problems that have emerged in empirical investigations and the parallel increase in environmental solutions. The vehicle renewal process involves targeted technical mutations both to observance of ecology as to the safety and comfort of the driver. New recyclable materials and more resistant have been developed in order to minimize the environmental impact of the vehicle even at the end of the operating life of its components, as well as solutions relating to the reduction of noise pollution generated as a response to the requirements of comfort. Modern research programs on a global scale have set themselves the objective of exploiting the potentiality of innovative technologies in the optimization of vehicles efficiency, the noise reduction and in the consequent reduction of fuel burn. One of the crucial topics in the greening of the new generation automotive sector is therefore the use and development of high vibro-acoustic performance materials. The goal of this research is properly focused on the analysis of viscoelastic materials appointed to increase the damping of the vibrations generated in a vehicle. The use of a viscoelastic material in this context is due to its high property to convert vibrational energy into heat, providing a significant dissipation of the vibrations. Trade-off analyses are performed in order define the stiffness and damping capacity of several viscoelastic foams with different thickness and density.

Introduction. The purpose of the present work is the experimental investigation of new materials and technologies to reduce the noise and vibrations produced inside motor vehicles. It is known, that there are several noise causes of the vehicles, both in the cockpit and in the environment: those of great impact are related to the engine and to the tyre/road interaction processes, that induce noise into the cockpit both through a structure-borne path contribution (mainly the vibration of the car floor induced by the mechanical forcing of the car body floor) and an air-borne path contribution. In Figure 1 these noise sources are evident for a moving car.

	Nomenclature
δ	Logarithmic decrement
FE	Finite Element
F	Force
F	Frequency
FRF	Frequency Response Function
g	g-force
k	Static stiffness
t	Time
w	Static deflection
ζ	Damping ratio

The new low-consumption engines also provide a design, which causes a higher specific noise than the previous ones: new soundproofing solutions are therefore necessary. In this research contest, 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]. The investigation of this class of viscoelastic material in this context is due to the high ability to convert vibration energy into thermal energy, providing a significant increase of vibration damping, hence this aspect will be study in terms of both static and dynamic stiffness and damping factor [3-6].

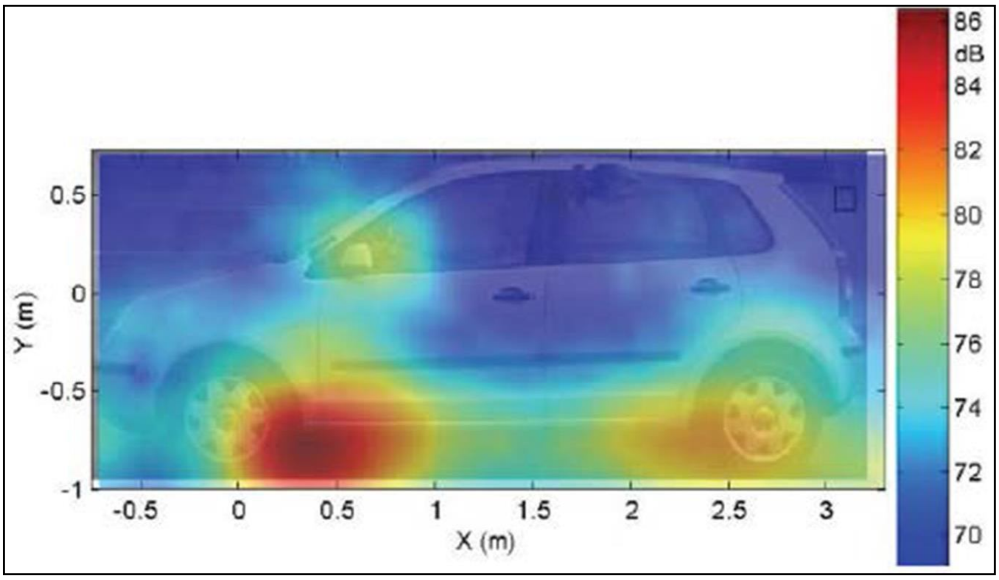

Fig. 1. Vibration distribution color map

Fig. 2. Technological application

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.

Within the paper, viscoelastic foams with different physical properties (density, thickness, structure) will be compared. The first experimental step concerns static stiffness measurements of the several viscoelastic foams.

In the second phase, damping characteristics of each foam have been carried out by the use of modal testing. Furthermore, in some cases, the dynamic stiffness has been measured for comparison with the static one.

Experimental measures. The experimental measurements performed in this research have allowed estimating the properties of stiffness and damping of innovative viscoelastic foams with different densities and thicknesses.

Static stiffness. In this section, the results of laboratory tests in order to measure the stiffness coefficient (1) will be explained.

$$k = \frac{F}{w} = \frac{[N]}{[mm]}. \qquad (1)$$

The experimental investigation concerns the comparison of original foam (intended as the standard foam already used in most of the automotive applications) 65-30 and two other viscoelastic foams 65-30, 75-30 where these two digit represent respectively density (Kg/m^3) and thickness (mm).

The static stiffness of each viscoelastic foam was evaluated in correspondence of different compression load settings by means of a test facility, shown in Fig. 3.

Fig. 3. Static test facility

In Figure 4 the trends of load versus static displacement have been plotted with reference to each case of investigation.

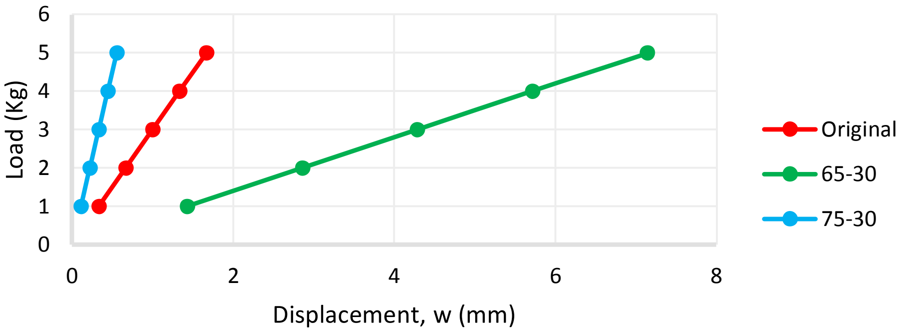

Fig. 4. Load-Displacement Curve

Table 1. Static stiffness measure

ID Foam	Slope (Kg/mm)	Static stiffness, k (N/mm)
Original 65-30	3	29.43
Visco 65-30	0.7	6.867
Visco 75-30	9	88.29

Dynamic stiffness. The dynamic stiffness is defined as the ratio between the dynamic force and the dynamic displacement: it is the quantity that expresses the elastic capacity of a material subjected to a harmonic stress. An excitation source, an accelerometric transducer, a test material (viscoelastic foam), and a rigid support plate are necessary to perform a dynamic test. The dynamic stiffness is comparable with the static value as a result of spectral analysis, as can be seen in Figure 5 for the Visco 75-30 foam.

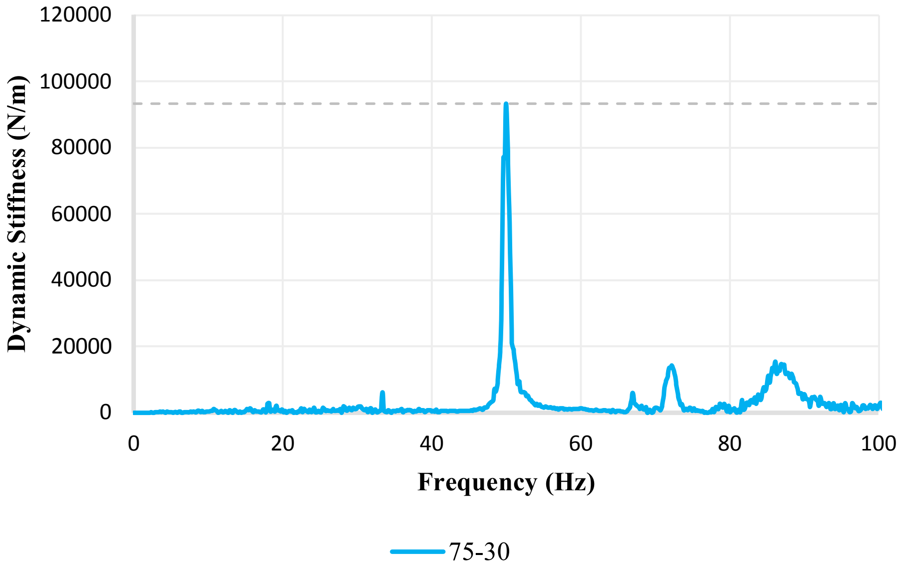

Fig. 5. Dynamic stiffness, Viscoelastic Foams 75-30

For the characteristic curve, 1024 points were chosen in a 0-100 Hz frequency range. It is clear that the stiffness of the foam 75-30 in correspondence of 50 Hz (first mode of vibration of the structure), is about 93500 N/m, next to the respective static stiffness value.

Modal Analysis. For the purposes of the frequency response measurement an LMS TestLab system has been used; 9 acquisition points have been defined for the mode shape reconstruction by the use of the rowing hammer technique. The goal of the spectral test is to compare the structure without material (Baseline) with the coated structure (viscoelastic foam) so as to discriminate the dynamic response of each foam. In Figure 6 the setup made for the dynamic test, is shown.

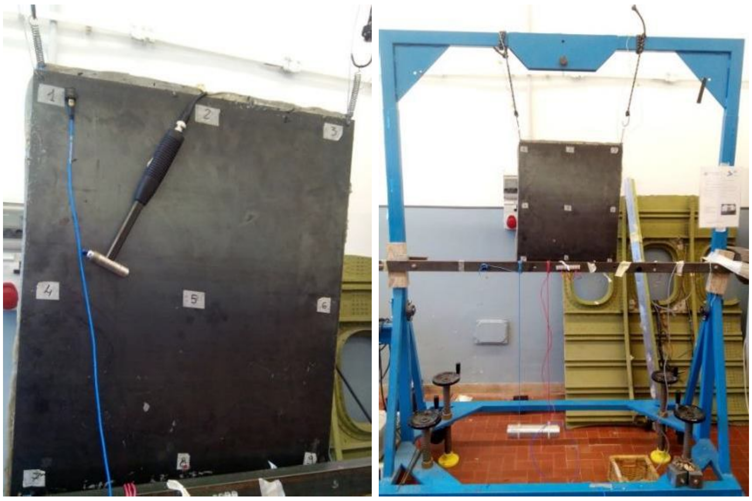

Fig. 6. Riding plate in free-free conditions

In Figure 7 the first elastic mode shape of the metal plate, which the resonance frequency is, about 48 Hz is represented.

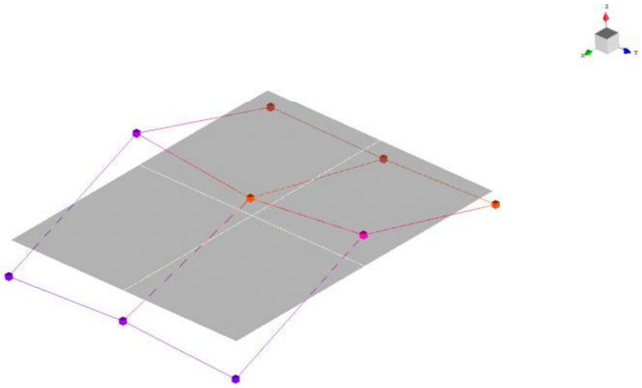

Fig. 7. Baseline configuration first mode shape, f = 47.8 Hz

The following figures show the most significant frequency responses (FRF) of the tested materials.

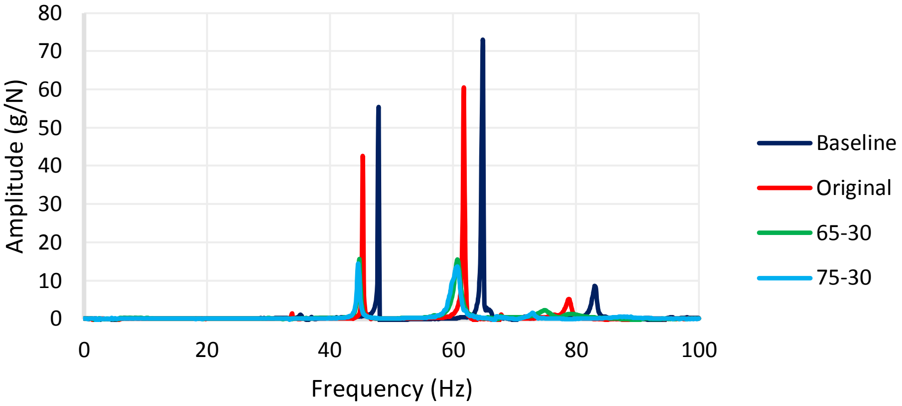

Fig. 8. Frequency Response Function (FRF)

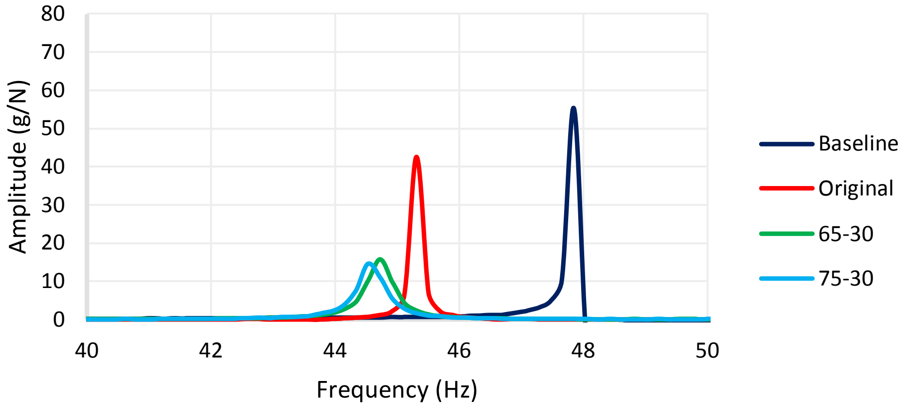

Fig. 9. Zoom about the first resonance frequency bandwidth

The results processing leads to observe a significant reduction of the resonance peak mainly due to the 65-30 and 75-30 viscoelastic foams. The added mass is perceived as a shift of the transfer function curve in virtue of (2):

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{2}$$

Half-Power bandwidth method. The system response close to the resonance region is strictly dependent on the damping. To estimate damping factor from frequency domain, the half-power bandwidth method is usable. In this method, two point corresponding to 3 dB down from the resonance peak are considered.

The damping factor ζ, is so defined as:

$$\zeta = \frac{f_2 - f_1}{f_n} \tag{3}$$

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_n is the value of the natural frequency. In Table 2 the damping coefficients obtained by that method for some of the foams are reported.

Table 2. Trade-Off damping coefficients, Half-Power Bandwidth

	Baseline	**Original Foam**	**Visco Foam 65-30**	**Visco Foam 75-30**
Frequency (Hz)	47.8	47.3	44.7	44.5
Max Amplitude (g/N)	55.4	43.1	15.7	14.5
Damping Ratio, ζ	0.11	0.19	0.56	0.60

Logarithmic decrement method. To estimate the value of damping factor in time domain through logarithmic decrement method it is necessary to know peak amplitude in two consecutive points, Y_1 and Y_2.

$$\delta = \ln\left|\frac{Y_1}{Y_2}\right|. \quad (4)$$

$$\zeta \approx \frac{\delta}{2\pi}. \quad (5)$$

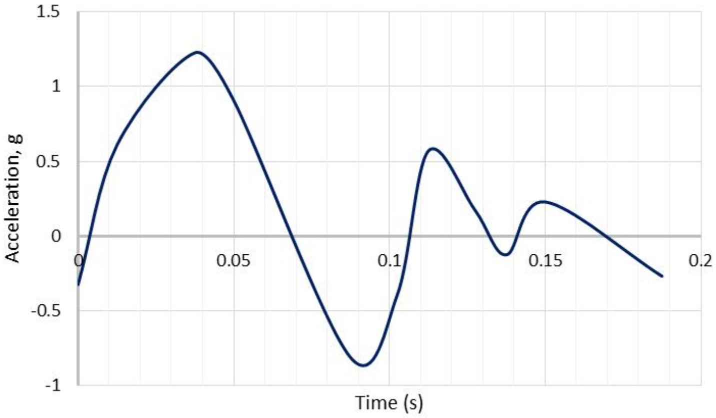

Fig. 10. Time History, Baseline

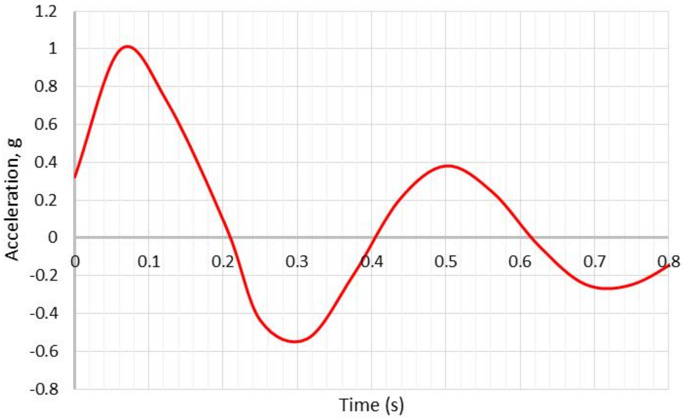

Fig. 11. Time History, Original Foam

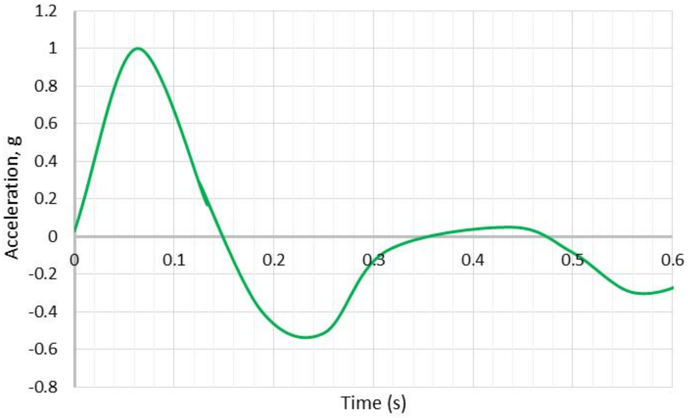

Fig. 12. Time History, 65-30

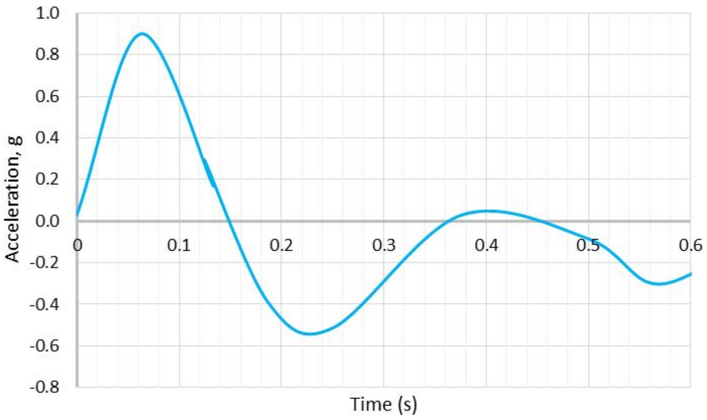

Fig. 13. Time History, 75-30

Table 3. Trade-Off damping coefficients, Logarithmic Decrement

	Baseline	Original Foam	Foam 65-30	Foam 75-30
Logarithmic decrement, δ	0.77	0.96	3.01	3.52
Damping Ratio, ζ	0.12	0.15	0.48	0.56

Summary. Acoustic and vibrational aspects are becoming central in many engineering field as those including automotive application and many transportation systems where the research of light-weighting construction solutions is a demanding aspect. Along the presented research, some visco-elastic foams have been studied as a possible mean to reduce the vibration induced noise inside a vehicle; the use of these foams could lead to the overall weight reduction because of the elimination of extra treatment nowadays used for this specific target.

It has been assessed that the use of viscoelastic materials brought significant benefits in terms of vibration's damping that has been approximately measured four times than the standard commercial solution. As regard the weight aspects, these viscoelastic foams are porous by nature, they have low density, and therefore they are particularly suitable for light-weighting application.

For the next developments, a numerical FE model will be developed to be correlated with the experimental one. Also direct acoustic measurements (to directly evaluate the radiated power from the panel) will be performed through a piezo-electric as an input source and a microphone as a transducer for the acquisition.

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