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CONSIDERATIONS ON THE VEHICLE FLOOR CARPET VIBRO-ACOUSTIC MODELLING

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Abstract: *In the present paper the carpet structure covering the firewall area of a vehicle is observed during the firewall vibration. The carpet structure is modelled as a vibrating spring-mass-damper system excited from the firewall which is in contact with the foam layer. The rest of the layers toward the habitacle interior are modelled as the mass of the system. By exciting carpet samples on the dynamic shaker the foam dynamic parameters and the first resonance of the carpet system are determined from the standpoint of the vibration transmitted from the firewall to the habitacle. This frequency is perceived like a weak point of the acoustic carpet insulation and has to be carefully managed when considering the vibration spectra coming from the engine and the power train.*
Keywords: *vehicle floor panel, carpet, resonance, sound absorption, vibration isolation.*

1. INTRODUCTION

The floor carpet of a car has to accomplish a list of basic functions such as to offer comfort in terms of thermal, acoustic and shock insulation, to provide suitable cover of the metal floor pan and hence aesthetics of the car interior. The floor carpet is composed of multiple layers. The visible part of the carpet is the tufting usually made from woven materials (natural or synthetic). The second set of layers are forming the carpet backing, in contact with the floor pan, comprising layers like the felt, foam or latex.

The cabin acoustics and the reduction of the noise level at the occupant ears is sustained by the design, modeling and the validation of new vehicle floor carpets with better acoustic performance. In parallel the reduction of the associated cost and the weight of the absorbing structure are under observation. Two of the main characteristics of the vehicle floor carpet are the transmission loss and the sound absorption coefficient. An important target of the floor carpet is to diminish the engine noise and the tyre-road noise. In order to have the description of the acoustic field, the noise transmission path and the noise isolation capability, a set of measuring spots located above the carpet covering the floor, has been chosen. In parallel the modeling of the acoustic and vibration parameters and the experimental validation of these parameters are of interest, when observing the whole habitacle acoustics.

In order to evaluate the acoustic comfort and the sound pressure at the floor carpet level, measurements has to be done on the road and on the chassis dynamometer in various driving conditions. Sound pressure water fall spectrum

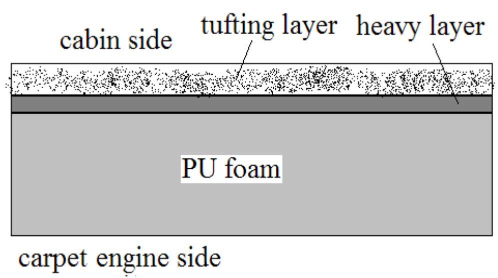

Fig. 1. Carpet layers

with the microphone at the driver ear is another useful indicator on the carpet quality.

Conclusions from the measurements are indicating that the front portion of the floor carpet is subjected to a high sound pressure level coming from low frequency power train noise. On the contrary the rear part of the carpet is subjected to sound pressure level of high frequency originating from the tyre – road reciprocal action. The floor carpet assembly has to act like a countermeasure in order to block low frequency at the front area and to block or absorb the high frequencies of the impinging sound waves or noise for the rear portion. A special attention has to be paid to the sound quality parameters like the articulation index, loudness, fluctuation strength, tonality, roughness and sharpness for the proper evaluation of the carpet structure and efficiency.

A common floor carpet is composed of three layers. A tufting layer toward the inside of the cabin, a thick foam layer on the engine side of the carpet and a heavy layer in between the previous two layers. The PU foam layer has the roll of vibration decoupler and vibration absorber, the middle layer is acting like a sound barrier and the tufting like an acoustic absorber.

In the sequel the carpet is modeled as a spring-mass-damper system attached to the vibrating floor.

2. THE CARPET EXCITED BY THE FIREWALL

The carpet under study is covering the floor with the foam layer in contact with the firewall. The dynamics of the spring-mass-damper system is observed while the support is exciting the carpet by harmonic loads in the frequency band of interest.

2.1. The heavy layer response to the support excitation – analytical approach

The carpet simplified model is shown in Figure 2, where *m* is the mass of the heavy layer and a part of the foam mass. A harmonic force coming from the engine and the transmission is transferred through the firewall and is exciting the foam outer side.

By applying Newton's second law, the differential equation of motion of the mass *m* is:

$$m\ddot{x} = -c(\dot{x}-\dot{y}) - k(x-y) \qquad (1)$$

or

$$m\ddot{x} + c\dot{x} + kx = c\dot{y} + ky \qquad (2)$$

where *x(t)* is the motion of *m* with respect to an inertial reference, *y(t)* is the imposed harmonic displacement of the support which in this case is the firewall, *c* and *k* are the damping and respectively the stiffness coefficients of the foam. The exponential form of the excitation $y(t) = Ye^{j\omega t}$ and the proposed response with a phase delay $x(t) = Xe^{j(\omega t-\varphi)}$ are substituted in relation (2), resulting:

$$(-m\omega^2 + cj\omega + k)Xe^{j(\omega t-\varphi)} = (jc\omega + k)\cdot Ye^{j\omega t} \qquad (3)$$

On the right side of (3) the excitation force acting on the support $F\sup(t)$ is observed. The response/displacement of the mass is:

$$Xe^{j(\omega t-\varphi)} = \frac{(jc\omega+k)\cdot Ye^{j\omega t}}{(-m\omega^2 + jc\omega + k)} \qquad (4)$$

The complex frequency response function of compliance type is:

$$\frac{x(t)}{F\sup(t)} = \frac{Xe^{j(\omega t-\varphi)}}{(k+jc\omega)Ye^{j\omega t}} = \frac{1}{k - m\omega^2 + jc\omega} \qquad (5)$$

The acceleration of the mass *m* is:

$$\ddot{x}(t) = (-\omega^2)Xe^{j(\omega t-\varphi)} = -\omega^2\frac{(jc\omega+k)\cdot Ye^{j\omega t}}{-m\omega^2 + jc\omega + k} \qquad (6)$$

The system inertance becomes:

$$\frac{\ddot{x}(t)}{F\sup(t)} = -\omega^2\frac{1}{-m\omega^2 + jc\omega + k} \qquad (7)$$

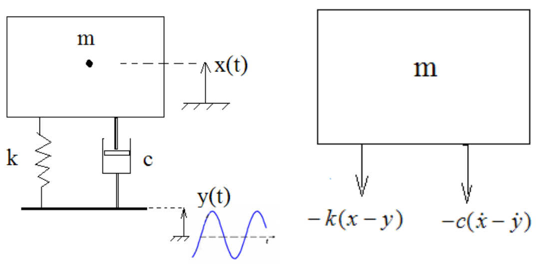

Fig. 2. Support excitation

After some calculations in which the definition of the damping ratio and the formulas for critical damping and the undamped natural frequency $\zeta = c/c_0$, $c_0 = 2m\omega_0$, $\omega_0 = \sqrt{k/m}$, are considered, results:

$$\frac{\ddot{x}}{F\sup} = \frac{-\omega^2/m}{-\omega^2 + 2j\zeta\omega\omega_0 + \omega_0^2}$$

By introducing the dimensionless variable $r = \omega/\omega_0$, the frequency response function, inertance type, FRF_{a_f}, becomes:

$$FRF_{a\_f} = \frac{\ddot{x}}{F\sup} = \frac{-r^2}{1 - r^2 + 2j\zeta r}\frac{1}{m}$$

The inertance transformed to have a real denominator is:

$$FRF_{a\_f}(r,\varsigma) = \frac{-r^2[(1-r^2) - 2j\zeta r]}{(1-r^2)^2 + 4\zeta^2 r^2}\frac{1}{m} \quad (9)$$

The real and imaginary components of the inertance are:

$$\mathrm{Real}(FRF_{a\_f}) = \frac{-r^2(1-r^2)}{(1-r^2)^2 + 4\zeta^2 r^2}\frac{1}{m},$$

$$\mathrm{Imag}(FRF_{a\_f}) = \frac{2j\zeta r^3}{(1-r^2)^2 + 4\zeta^2 r^2}\frac{1}{m}$$

For r=1, the real component equals zero and the complex part presents a minimum. The inertance modulus is:

$$A = |FRF_{a\_f}| = \frac{r^2}{\sqrt{(1-r^2)^2 + 4\zeta^2 r^2}}\frac{1}{m} \quad (10)$$

The *FRF_a_f* modulus and phase graphs for several different damping ratios, are similar to the graphs depicted in Figure 3.

By differentiating the modulus expression with respect to *r*, the peak of the FRF modulus is found:

$$\frac{d\,FRF_{a\_f}(r,\zeta)}{dr} = 0, \text{ resulting:}$$

$$r_{peak} = \frac{1}{\sqrt{1 - 2\zeta^2}} \quad (11)$$

which is placed on the right of the abscissa r=1 for various ζ values (Fig. 3).

2.2. Swept sine test force excitation of the carpet support

The material probe under study, of a specific area, is mounted on the shaker, in order to be submitted to vibration, similar to the carpet on the firewall.

A linear swept sine excitation has been applied to the structure, getting the Bode diagram which is covering the frequency band from 10 to 180Hz [6].

The absorbing structure composed by the PU foam layer and the heavy layer (HL) on top of it, is excited by the shaker. The excitation force is measured by a force sensor. The heavy layer response is registered by using a mini-accelerometer placed on top of the structure (Fig. 4).

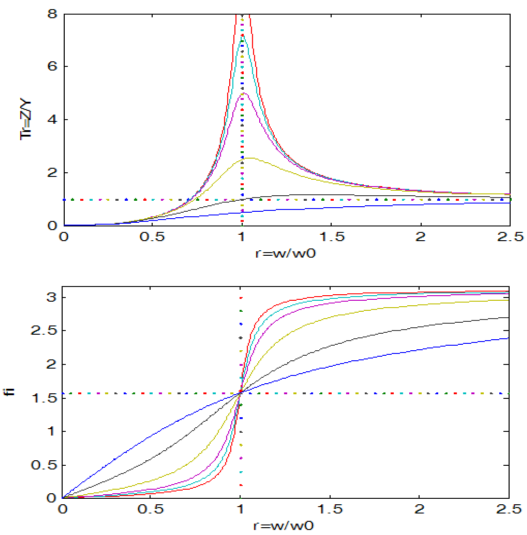

Fig. 3 FRF magnitude and phase

The Bode diagram (Fig. 5) where the magnitude and phase versus frequency are superposed has been recorded for the frequency band where the resonance is expected to be.

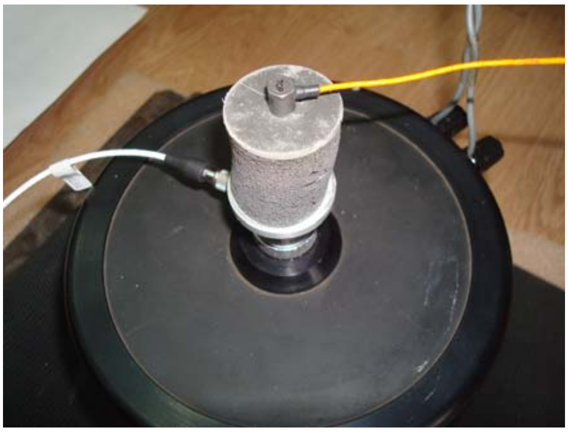

Fig. 4. Experimental set-up

The peak of the magnitude versus frequency graph of the frequency response function is associated to the first structure resonance:

$$\omega_{d\,test} = 2\pi \cdot 94.6\ [rad/s] \qquad (12)$$

In the phase versus frequency graph, the phase lag at the resonance magnitude peak, is registered to be close to -90 degree. The peak of the FRF magnitude (11) is at r_v:

$$r_v = \frac{\omega_{d\,test}}{\omega_{0\,test}} = \frac{1}{\sqrt{1-2\zeta^2}} \qquad (13)$$

The resulted undamped angular frequency is:

$$\omega_{0\,test} = \omega_{d\,test} \cdot \sqrt{1-2\zeta^2} \qquad (14)$$

From the peak the damping ratio (linear axes) can be extracted assuming small damping:

$$\zeta = \frac{\omega_D - \omega_C}{2\omega_0} \qquad (15)$$

where ω_D and ω_C are the half power angular frequencies. The resulted damping ratio, for small foam elastic deformations, is $\zeta \cong 0.074$.

$$\omega_{0\,test} = 2\pi \cdot 93.6 \qquad (16)$$

The stiffness constant k of the foam for small elastic deformations of the sample, is determined from the relation:

$$\omega_{0\,test} = \sqrt{\frac{k}{m_{HL} + 1/3 \cdot m_{foam} + m_{acc}}} \qquad (17)$$

where m_{HL}, m_{foam}, m_{acc} are the heavy layer, the foam and the accelerometer masses. A third of the spring (foam) mass has been added.

The sample spring constant determined from the experimental approach is $k = 1388$ [N/m], resulted from the following relation:

$$k = \omega_{0\,test}^2 m = \omega_{0\,test}^2 \cdot (m_{HL} + 1/3 \cdot m_{foam} + m_{acc}) \qquad (18)$$

The carpet can be excited from the other side in order to observe the same resonant frequency of the structure for small elastic deformations of the foam layer.

3. THE STRUCTURE EXCITATION FROM THE HEAVY LAYER SIDE

In this approach the damped resonant frequency has been measured by using free vibrations of the specimen.

A light accelerometer is attached to the heavy layer of the structure. The structure is slightly compressed and suddenly released (Fig. 6). The rate of decay of damped free oscillation is determined from the recorded acceleration. Then the damped period and damped frequency are determined. A similar approach is to apply the Fast Fourier transformation to the decay of the free oscillation and deriving the damped frequency by using the peak picking method. For the viscously underdamped single degree of freedom system the following relation is valid: $\omega_d = \omega_0\sqrt{1-\zeta^2}$, from which results:

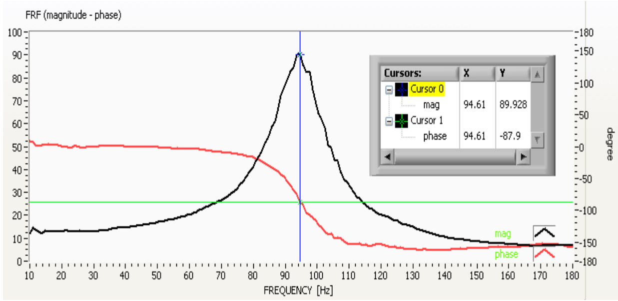

Fig. 5. Experimental FRF inertance

$$f_{d\,test2} = f_0\sqrt{1-\zeta^2} \qquad (19)$$

The r value for the peak registered when the support is excited is also recalled:

$$r = \frac{1}{\sqrt{1-2\zeta^2}} = \frac{\omega_{0d}}{\omega_0}$$

from which, results:

$$f_{d\,test1} = \frac{f_0}{\sqrt{1-2\zeta^2}} \qquad (20)$$

Based on the relations (19) and (20) and on the resonant frequencies experimentally determined, the natural frequency f_0 and the

Fig. 6. Free vibrations start-up

damping ratio ζ , can be found.

The analytical approach of the harmonic excitation of the structure at the mass location, in particular at the heavy layer, is recalled.

When the mass of a single degree of freedom system is harmonically excited the following differential equation states:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(\omega t) \tag{21}$$

By assuming a harmonic response and some adjacent calculations the relation for the damped frequency results:

$$r_v = \sqrt{1 - 2\zeta^2}$$

from which the damped angular frequency results:

$$\omega_v = \omega_0 \sqrt{1 - 2\zeta^2} < \omega_0 \tag{22}$$

The peak abscissa is smaller than the natural frequency.

4. WHOLE CARPET STRUCTURE ON THE SHAKER

A tufting sound absorbing layer is added to the structure. The accelerometer is glued in a special prepared area in the middle of the disk (Fig. 7). The frequency response function is

Fig. 7. The carpet structure

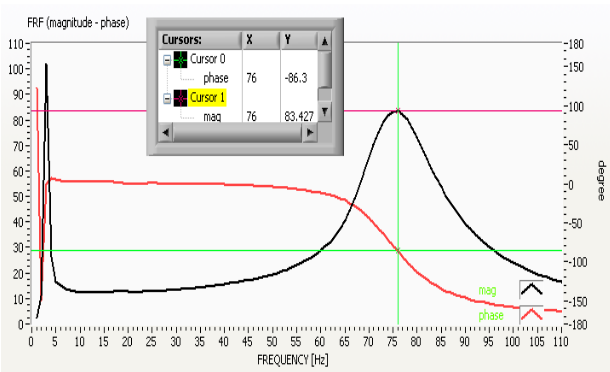

Fig. 8. FRF of the whole structure

recorded during a sine swept force excitation applied to the whole carpet structure. The acceleration response and the excitation force are simultaneously sampled and recorded by the acquisition system [4]. A low resonant peak associated to the elastic deformation of the foam is registered on the frequency response function. The spring constant should be the same but the mass is increased because of the extra acoustic layers added on top of the carpet heavy layer. The free vibration of the whole structure can be again onset in order to verify the features already determined for the foam elasticity.

5. CONCLUSIONS

A portion of the carpet, covering the firewall area of a sedan vehicle, has been observed analytically and experimentally, during the firewall vibration. This is interesting from the point of view of sound radiation of various vehicle body in white panels, contributing to the sound pressure level at the interior cabin comfort points. By covering the floor with the carpet the sound radiation can be alleviated. The carpet structure sample is modelled as a spring-mass-damper system excited from the support. The foam layer in contact with the firewall is modelled as a spring, while light or small amplitude vibrations take place. The rest of the layers toward the habitacle interior are modelled as the mass of the system. By exciting a carpet sample on the dynamic shaker the first resonance of the carpet sample is determined from the point of view of the vibration transmitted from the firewall to the habitacle. This frequency is considered a weak point of the associated acoustic carpet and has to be carefully managed when considering the vibration and acoustic spectra coming from the engine and the power train.

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Consideratii asupra modelarii vibro-acustice a carpetului vehiculului

Rezumat: In prezenta lucrare este studiat carpetul acoperitor al scutului metalic care desparte camera motorului de habitaclu, acesta având rolul de a atenua vibrațiile transmise spre interior pe de o parte şi de a absorbi sunete din habitaclu. Eşantioane de carpet au fost supuse vibrației folosind un excitator dinamic. Prima rezonanță a structurii este observată, aceasta constituind un punct slab al protecției interiorului la vibrațiile provenite de la motor.

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