Experimental Evaluation of the Antivibrating Damping Capacity in Case of Elastomers used for Tram Railway Supporting

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This paper is focused on the main static and dynamic characteristics in case of rubber elements designated to damp the vibrations transmitted to the surrounding buildings by means of the railway. Thus, the experimental methods performed on a stand having harmonic excitation in case of actual loading under dynamic regime both stationary and transient attaining the resonance point are presented. Based on this, the system dynamic parameters such as: dynamic rigidity, critical damping fraction, dissipation coefficient (hysteresis) can be determined. Aiming this the free damped vibration method and the controlled excitation method with monotone increasing variation for the excitation frequency have been used. In this way can comparatively be determined both the rigidity and the damping dynamic characteristics of the rubber for various manufacturing receipts taking into account the chemical composition as well as the manufacturing process. Finally, the paper presents experimental results obtained by our research institute ICECON Bucharest in cooperation with University "Dunărea de Jos" of Galaţi during the works carried out for the tram rails in Bucharest, Romania.

Keywords: elastomer, rigidity, damping, vibration, structural noise, antivibrating insulation

The constructive solution designated to insulate vibrations generated while the tram moves along the railway consists in embedding the railway in a rubber elastic system. The rubber elastic system is constructed so that the damping and vibration effects in the horizontal and vertical direction as well as the structural noise transmitted to the environment should be diminished within the permissible limits.

Aiming this, the manufacturing companies have constructed rubber elements displayed along the railway on both internal and external side and at the bottom also, in order to obtain a continuous support.

In order to assess the damping and insulating efficiency for the transmitted vibrations, this paper presents the dynamic methods of analyse. Based on these methods the antivibrating material that has been used can be characterized. So, the tests performed on the stand provided with cylindrical rubber elements excited by a given rotating inertia force are presented. The eigen vibrations and damping parameters have been determined on the stand while maintaining the moving mass and the rotating force constant and changing the number of rubber elements parallel connected.

The experimental method offers the parametric assessment possibility in order to obtain an objective comparison between various antivibrating rubber materials. These materials are used as rubber damping systems for the railway type 49 on the external side as well as for the internal side.

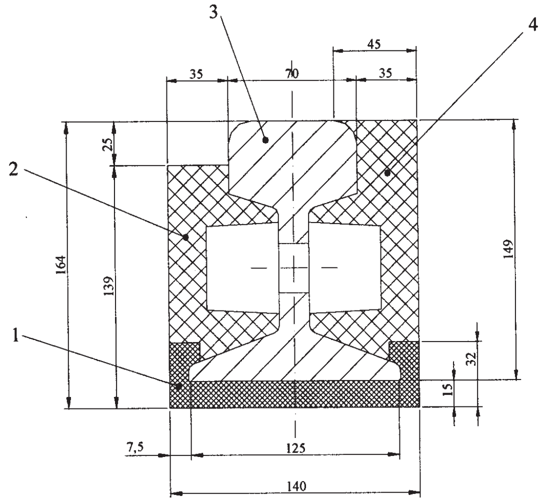

Fig. 1. Example of tram railway: 1- Base plate; 2 - Internal damping element; 3 - Railway type 49; 4 - External damping element

Experimental part

The tests have been carried out on a stand having the inertial vibrator contained in the upper mass. The mass is three points supported by groups of elements made of the antivibrating rubber subjected to test. Each group can consist on a single element or more identical elements parallel connected.

Elastic Parameters

Figure 2, illustrates the stand consisting on two frames having three arms. The electric motor 1 performs the actuation and the vertical harmonic force is generated by the vibrator 2, mounted on the plate 3, fixed on the upper star 4 by means of three arms.

Each arm transmits the vertical harmonic force by means of the load cell 5, to the fixing system 7. The relative displacement between the two supporting points of the rubber element is measured by the moving–coil transducer 8.

The stand allows the testing under dynamic regime for free and imposed vibrations of 50 Hz and the harmonic force magnitude $P_o = 3 \cdot 10^4$ N, where $P_o = m_o r\omega^2$.

The static rigidity factor has been determined by the dynamic response method for the free damped vibration and for the stationary vibration having harmonic excitation at resonance. In both situations the eigen frequency f_0 has

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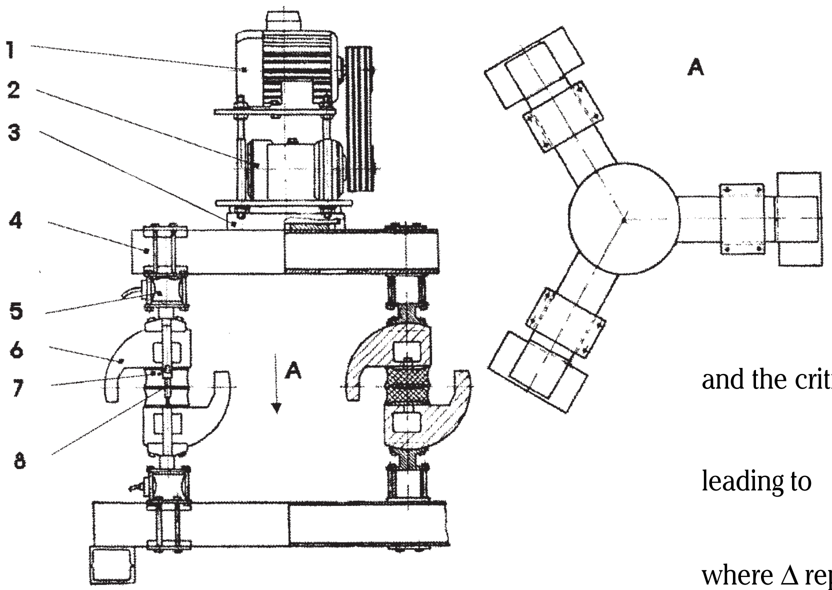

Fig. 2. Testing stand

been determined. The dynamic rigidity factor k has been calculated as follows [1, 6, 7, 11]:

$$k = 4\pi^2 f_0^2 m, \tag{1}$$

with m – the mass and f_0 – the eigen frequency.

Damping Parameters

Taking into account the linear viscoelastic behaviour of the antivibrating rubber under given conditions for the testing stand, the essential parameters are ζ, the critical damping fraction and ψ- the dissipation factor of the hysteresis loop. Thus, on the same testing stand two methods can be used, namely:

-response of the system subjected to free vibrations excited by an inertial shock [1, 2];

-response of the system subjected to forced vibrations under steady regime with the force $F = P_0 \sin \omega t$ [2, 6, 11].

<u>Free Damped Vibration Test</u>

The damping logarithmic decrement δ in case of an integer number of complete cycles $j \geq 2$ is determined and ζ is given by relation [2, 3, 5, 7, 9]:

$$\zeta = \frac{1}{2\pi}\delta \tag{2}$$

The specific damping capacity is defined as follows [6, 7]:

$$\Psi = \frac{\Delta W}{W} \tag{3}$$

where:

$\Delta W = \pi c \omega_0 A_0^2$ represents the energy variation corresponding to a hysteresis cycle;

$W = W_{el} = \frac{1}{2} k A_0^2$ is the maximum elastic energy corresponding to a hysteresis cycle.

with

ω_0 - the system eigen frequency;

A_0 – the displacement amplitude under free regime for frequency ω_0;

C – the viscous damping coefficient.

Thus, we have:

$$\Psi = \frac{\pi c \omega_0 A_0^2}{\frac{1}{2} m \omega_0^2 A_0^2} = 2\pi \frac{c}{m\omega_0}$$

and the critical damping fraction is expressed as:

$$\zeta = \frac{c}{c_{cr}} = \frac{c}{2\sqrt{km}} = \frac{1}{2}\frac{c}{m\omega_0}$$

leading to

$$\Psi = 4\pi\zeta = 2\pi\Delta \tag{4}$$

where Δ represents the loss angle.

<u>Harmonic Forced Vibration at Resonance Test</u>

The resonance frequency $f_{rez} = f_0$ and the displacement amplitude in resonance A_{rez} are measured.

Taking into account the static moment of the vibrator $m_0 r$, where m_0 is the unbalanced mass in rotation with the circular frequency ω, and r is its eccentricity related to the rotation axis as well as the global mass being in motion m, we have:

$$\zeta = \frac{1}{2}\frac{m_0 r}{m A_{rez}} \tag{5}$$

where:

$m_0 r$ represents the static moment of the exciting inertial system with the force $P_0 = m_0 r \omega^2 \sin \omega t$ and A_{rez} stands for the amplitude at resonance.

The specific damping capacity under forced regime is [4, 6 ,7]:

$$\Psi^* = \frac{\Delta W}{W},$$

with

$\Delta W = \pi c \omega A^2$ having the circular frequency ω of the exciting force;

$W = W_{el} = \frac{1}{2} k A^2$ the maximum elastic energy and A - the amplitude of the forced vibration under steady regime.

Thus, one could write:

$$\Psi^* = \frac{\pi c \omega A^2}{\frac{1}{2} k A^2} = 2\pi \frac{c\omega}{k}$$

Taking into account $\omega_0 = 2\pi f_0$ and $k = m\omega_0$, it results in:

$$\Psi^* = 2\pi \frac{c\omega}{m\omega_0^2} = 2\pi \frac{\omega}{\omega_0} \cdot \frac{c}{m\omega_0}$$

where at resonance $\omega/\omega_0 = 1$ and $c / m\omega_0 = 2\zeta$, so that we obtain:

$$\Psi^*_{rez} = \Psi = 4\pi\zeta \tag{6}$$

The loss hysteretic or structural factor $h = c\omega$ under forced regime is defined as follows [6, 7]:

$$\beta = \frac{1}{2\pi}\frac{\Delta W}{W} \tag{7}$$

Table 1

Sample	Number of rubber elements	Dynamic rigidity k_{din}, N/m	Eigen frequency f_0, Hz	Logarithmic decrement δ	Damping ζ	Dissipation ψ	Loss factor β
1	3	$20{,}5 \cdot 10^5$	10,74	0,33	0,0525	0,66	0,105
2	6	$44{,}1 \cdot 10^5$	15,78	0,463	0,0737	0,92	0.146
3	9	$63{,}5 \cdot 10^5$	18,91	0,570	0,0912	1,14	0,181

Taking into account the complex rigidity k*=k+jh=k(1+jg), where $j=\sqrt{-1}$, $g=\Delta=\frac{h}{k}$we have:

$$\Delta W = \pi h A^2$$

$$W = \frac{1}{2} k A^2$$

So that we obtain:

$$\frac{\Delta W}{W} = \frac{\pi h A^2}{\frac{1}{2} k A^2} = 2\pi \frac{h}{k} = 2\pi g$$

and

$$g = \frac{1}{2\pi} \frac{\Delta W}{W} = \beta = \Delta \qquad (8)$$

and finally

$$\beta = g = \frac{\Delta W}{2\pi W} = \frac{4\pi\zeta}{2\pi} = 2\zeta \qquad (9)$$

Table 1 presents the experimental results obtained for rubber elements parallel connected [8, 10], for three grouping versions - three by three, six by six and nine by nine.

Conclusions

The rigidity and damping experimental results have been determined by both testing methods pointing out a high accuracy conformity between the two methods.

We conclude that for the same value of the moving mass the dynamic rigidity and the material damping are specific physical quantities that are situated in the frame of the laws presented.

The testing method performed on the stand both for free vibration and forced vibration at resonance is useful to characterize antivibrating materials necessary to insulate the tram railways.

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Manuscript received: 15.12.2008