# STRUCTURAL AND FUNCTIONAL REQUIREMENTS IMPOSED TO VIBRATION INSULATOR SYSTEMS

PACS Reference: 43.40. Tm VIBRATION ISOLATORS, ATTENUATORS AND DAMPERS

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## ABSTRACT

This work presents the essential requirements concerning both the structure and the behavior of vibration insulator systems by introducing the performance concept. Thus, some of the constructive solutions useful for vibration insulation in case of stationary equipment being in contact with building structural elements are analyzed.

## INTRODUCTION

The rubber antivibrating element certification basing on specific tests confirms the conformity with essential requirements and the index values in numerical expression.

The tests performed upon the rubber sample have characterized only the basic material without clarifying the aspects concerning the finite elements such as rubber insulating elements.

In order to characterize the structural and functional quality in case of rubber elements the test have to be performed on special stands at natural scale and under real loading conditions. Thus, this paper puts into evidence both the necessity and the procedures of the tests so that the elastic and damping parameters can be experimentally determined.

## SPECIFIC REQUIREMENTS FOR VIBRATION INSULATING SYSTEMS

Let consider the elastic systems consisting o rubber elements connected in parralel.

The main structural requirements are as follows:

a. the resistance so that the stress under static and dynamic regime frames in the permissible interval;
b. the specific deflection under static regime;
c. the system stability while passing through resonance;
d. the structural damping capacity evaluated for the whole system as a function of the component elements.

The functional requirements for the elastic system are the following:

a. the equivalent rigidity under static and dynamic regime;
b. the critical damping factor for the whole system;
c. the elastic system deflection under the mass system static action;
d. the rubber internal temperature under dynamic regime.

## THE INSULATING SYSTEM TESTING

In the frame of the Research Institute ICECON Bucharest has been conceived and realized a testing stand for the natural scale rubber antivibrating elements under harmonic excitation and static loading having discrete rise in the range 360 kg to 2500 kg. The stand can modify the rubber elements repose angle from $0^0$ to $90^0$, from compression loading to shear loading and for angles equal to 15, 30, 45, 60 and 75 degrees the composed compression-shear loading is obtained.

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

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

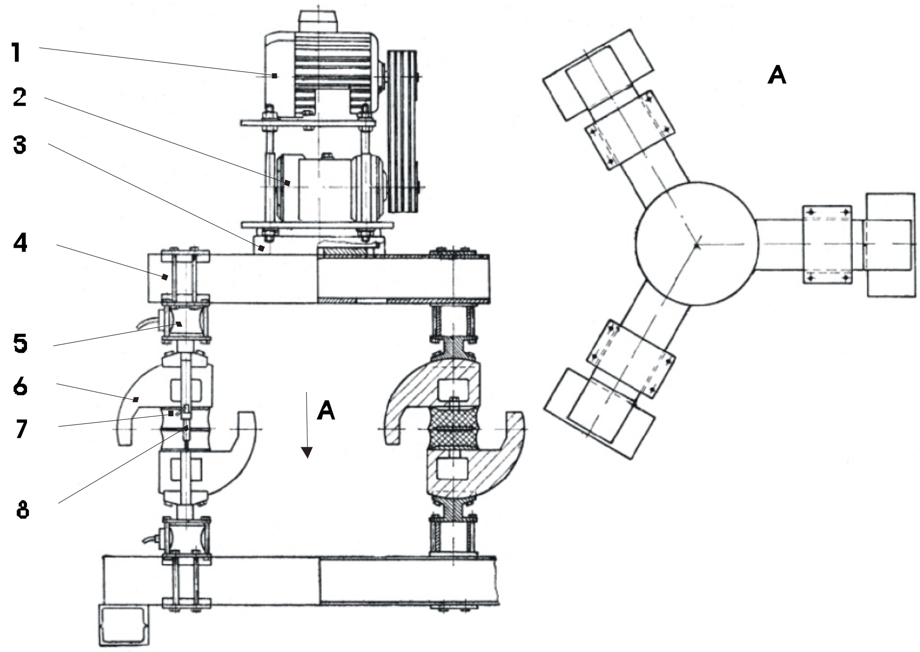


Figure 1

The stand permits the testing under dynamic regime for free and imposed vibrations for 50 Hz and the harmonic force magnitude $P_0 = 3\ 10^4$ N.

In figure 2 is presented a free vibration example recorded for a group of three antivibrating elements S 120; rubber AB 31, connected in parallel, subjected to shear loading = $90^0$, the static load 360 kg.

Figure 3 represents the displacement–time sequence in case of resonance for the same antivibrating system consisting of three elements S 120. Remarkable is the eigenperiod $T^*$=2,2 seconds and the eigenfrequency $f_0$ = 4,5 Hz, obtained in both situations.

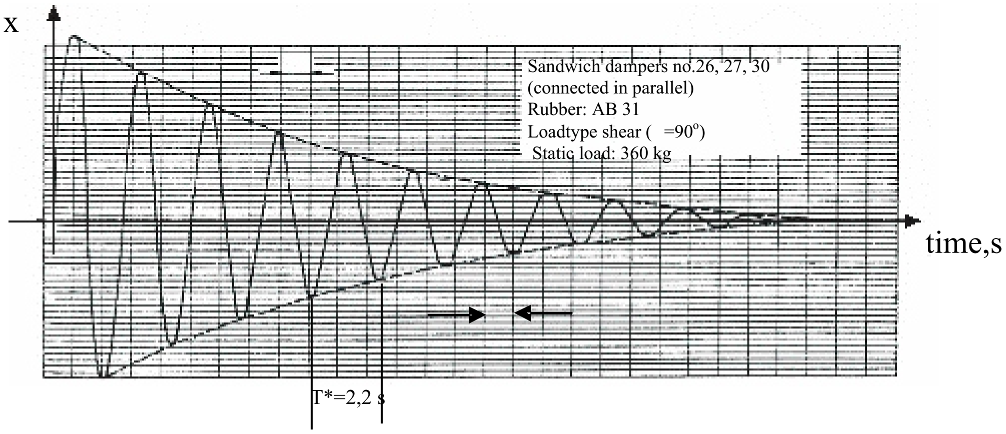


Figure 2

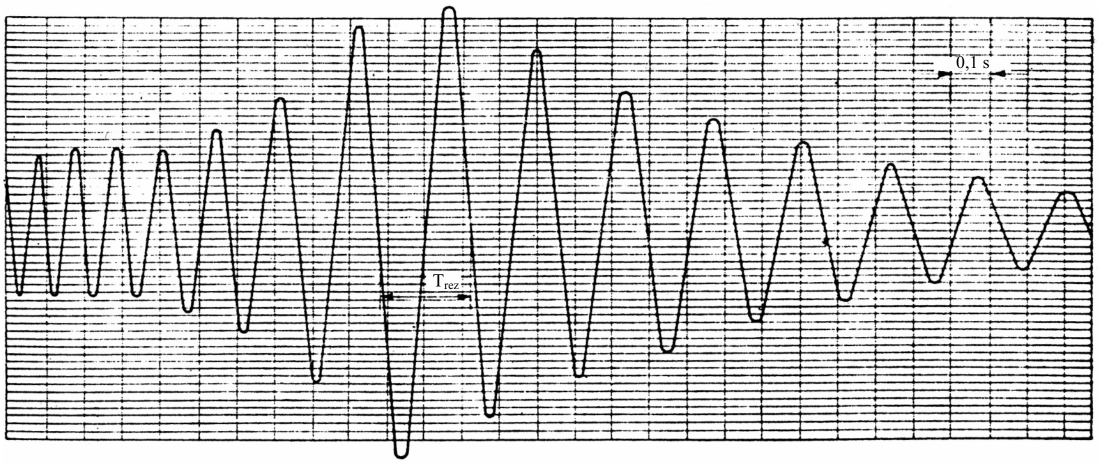


Figure 3

## THE EXPERIMENTAL DETERMINATION OF THE ELASTIC AND DAMPING PARAMETERS

## THE SYSTEM DYNAMIC RIGIDITY

The system linear elasticity is defined by means of the dynamic rigidity factor $k_{din}$ for repose angle $\alpha$. Thus, it results in:

$$k_{din} = \frac{4\pi^2}{T_{rez}^2} m (1-\zeta^2) \qquad (1)$$

where:

m represents the global mass;

$T_{rez} = T^*$ - the natural eigenperiod, without damping;

$\zeta$ - the critical damping factor.

The relation between $k_{din}$ for a repose angle α and the rigidity factors under dynamic regime for compression $k^c_{din}$ and shear $k^f_{din}$ is given by:

$$k_{din} = k^c_{din} \cos^2\alpha + k^f_{din} \sin^2\alpha . \qquad (2)$$

Figure 4 presents the dynamic rigidity variation in case of a system consisting of three rubber elements for different repose angles.

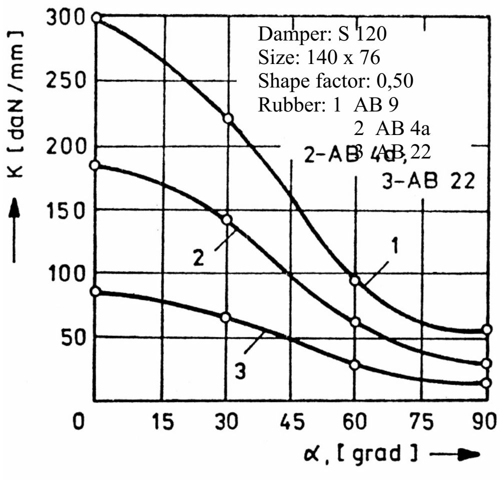


Figure 4

## THE SYSTEM DAMPING

The system linear viscous damping can be evaluated basing on the experimental tests using free vibration method and stabilized stationary forced vibration method.

### The System Free Vibration Method

The stand has been used aiming to determine the displacement signal for an initial impulse applied on the three arms platform and different repose angles. Thus, on the basis of damping signal vibro-record, for j consecutive displacements $x_j$, j = 1, 2, …, n the physical quantities , , n, are determined as follows:

The logarithmic decrement is given by relation:

$$\frac{1}{j \quad 1}\ln\frac{x_1}{x_j}, \quad j \quad 1, 2, \; ....n . \qquad (3)$$

The critical damping factor is:

$$\frac{1}{2} \quad . \qquad (4)$$

The viscous damping factor **n** is:

$$2n \quad \frac{b}{n} \quad \frac{4}{T^*} m \quad , \qquad (5)$$

where b is the linear damping factor.

$$T \quad \frac{2}{{}_{n}}, \quad \text{with} \quad {}^{2}_{n} \quad \frac{k}{m} \quad n^2$$

The energy dissipation factor :

$$2 \qquad (6)$$

**The System Forced Vibration Method**

By means of load cells the global force F is measured as a result of the linear elastic force $F_e$ = kx and the linear viscous force $F_v \quad b\dot{x}$, under the form $F^2 \quad k^2x^2 \quad b^2\dot{x}^2$. The viscous damping factor b is defined by:

$$b \quad \frac{1}{\dot{x}} F^2 \quad k^2x^2 \;^{1/2} \qquad (7)$$

where the instantaneous displacement x, the instantaneous speed $\dot{x}$ and instantaneous force F = F(t) are simultaneously measured.

The following parameters can be determined:

The damping factor n is:

$$2n \quad \frac{b}{m}. \qquad (8)$$

The critical damping factor is:

0 177,00 2,03 69,50 15,0 23,0 3,0 0,75 1,70

$$\frac{1}{4}\frac{b}{m^2}T^*. \qquad (9)$$

The frequency range at resonance f is:

$$f \quad \frac{1}{2}\frac{b}{m}. \qquad (10)$$

The system quality factor Q is:

$$Q \quad \frac{n}{2} \quad \frac{1}{4}\frac{b}{m} \quad \frac{1}{2}. \qquad (11)$$

Table 1 presents some of the test results obtained for rubber sandwich elements AB 31, in case of three repose angles equal to 0, 60 and 90 degrees, introducing the notations $\,_0^2 \quad \frac{k}{m}$ and $x_{st} \quad \frac{mg}{k}$ where mg is the global weight of the sprung mass on the three rubber elements of the antivibrating system.

Table 1

| (degree) | k(daN/mm) | $x_{st}$(mm) | $_0$(rad/s) | $10^{-3}$ | $10^{-3}$ | $10^{-1}$ | $T^*(10^{-2}$ s) | n($s^{-1}$) |
|---|---|---|---|---|---|---|---|---|
| 0 | 177,00 | 2,03 | 69,50 | 15,0 | 23,0 | 3,0 | 0,75 | 1,70 |
| 60 | 55,50 | 6,48 | 38,80 | 20,0 | 31,0 | 4,0 | 16,15 | 1,23 |
| 90 | 15,63 | 23,03 | 20,60 | 35,0 | 55,0 | 7,0 | 30,40 | 1,15 |

## CONCLUSIONS

The vibration insulating systems are characterized by structural requirements regarding the constructive structure and functional requirements concerning the eigenfrecquencies, the structural damping as well as the durability under long term dynamic regime.

The experimental results have been obtained by means of a stand for dynamic tests under actual loading conditions.

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