Effect of Reducing the Vibrations Transmitted with High-Performance Elastomeric Devices

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Abstract: - It is presented the effect of reducing the vibrations transmitted from the source to the body of the machine is presented so that the functional conditions can be ensured at the level of predictable performance. Thus, the research results highlight the influence of hysteretic damping expressed by the angle of internal loss of the energy of the forced vibrations.

Keywords: - elastomer, transmissibility, complex module, dynamic system

1. INTRODUCTION

It is considered the elastic system consisting of two concentrated masses separated from each other by a rubber anti-vibration connection.

The level of vibrations transmitted from the active mass element m_1 to the mass element m_2 to be protected is assessed by transmissibility. Figure 1 shows the model of the system, which contains a viscoelastic step made of anti-vibration rubber elements. Out of the category of vibrating machines that can be studied on the basis of this model includes the following: vibro-pokers for piles and sheet piling, vibratory compactor rollers (towed), vibratory plates. [1-5]

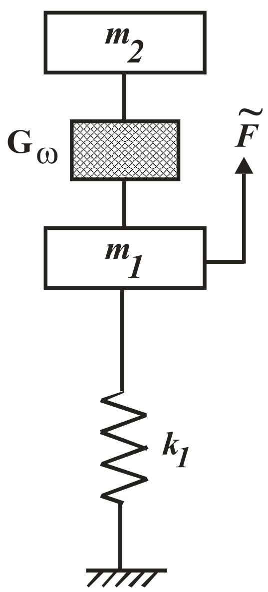

Figure 1. Dynamic model

It is mentioned that the calculation model may also be applicable to some subassemblies that are part of the structure of some machines that do not have vibrating action, but which by their nature of operation generate harmful vibrations [6-10].

2. EVALUATION OF VIBRATIONS TRANSMISSIBILITY

The calculation scheme in Figure 1 is the dynamic module for some categories of vibrating action machines, whose constructive scheme is presented in [11-13].

In order to assess the vibration isolation effect, the transmissibility coefficient shall be used so that it may express the ability to reduce the vibrational motion transmitted from the working element (roller, plate, clamping element) to the subassembly containing the drive engine, and the other auxiliary mechanical elements [12-15].

The rubber elements are characterized by the complex longitudinal modulus of elasticity E * which is expressed as follows:

$$E^* = G^* (1+\beta\Phi^2)$$

where:

G^* is the complex transversal elasticity module
β-multiplication coefficient ($\beta = 5 \div 20$)
Φ - form coefficient of the rubber element.

Complex transversal module G^* is expressed according to the elastic and viscous component, as:

$$G^* = G(1+j\delta) \quad (1)$$

where : G is the transversal elasticity module which depends on the movement pulsation (for the rubber used, the influence of the pulsation on the viscous-elastic properties is insignificant, therefore the G* notation will be used); δ is the internal mechanics loss angle $j = \sqrt{-1}$ is the imaginary unit.

In this case, the rigidity coefficient equivalent of the system consisting of several anti-vibratile elements, geometrically identical and based on the same rubber recipes, is expressed as follows:

$$k^* = \frac{S}{h} \cdot E^* \quad (2)$$

or, taking into consideration the relations (1) and (2) it is obtained

$$k^* = \frac{S}{h}\,(1+\beta\varphi^2)\cdot G^* \quad (3)$$

where: S is the total transversal area of the rubber elements corresponding to the anti-vibratile step; h – active height of a rubber element.

It is noted with $\lambda = (1+\beta\varphi^2)\cdot\frac{S}{h}$ geometric multiplication factor and it is obtained:

$$k^* = \lambda \cdot G^*$$

The differential movement equations for the calculation model in figure 1 written down in complex, are as follows:[16-17]

$$\begin{cases} m_1\ddot{\widetilde{x_1}} + \lambda G^*(\widetilde{x_1} - \widetilde{x_2}) + k_1\widetilde{x_1} = \widetilde{F_1} \\ m_2\ddot{\widetilde{x_2}} + \lambda G - (\widetilde{x_1} - \widetilde{x_2}) = 0 \end{cases} \quad (4)$$

Force $\widetilde{F_1}$ transmitted to the superior mass by the anti-vibratile elastic step characterized by, is given by the relation :

$$\widetilde{F_2} = \lambda G^* \cdot \widetilde{x_2} \quad (5)$$

From the first relation of the system (4) it emerges:

$$\widetilde{F_1} = (-m_1\omega^2 + \lambda G^* + k_1)\,\widetilde{x_1} - \lambda G^*\widetilde{x_2} \quad (6)$$

or taking into account (5) it is obtained

$$\widetilde{F_1} = (-m_1\omega^2 + \lambda G^* + k_1)\,\widetilde{x_1} - \widetilde{F_2} \quad (7)$$

From the second relation of the system (23.5) we have:

$$\widetilde{x_1} = \widetilde{F_2}\frac{\lambda G^* - m_2\omega^2}{(\lambda G^*)^2}$$

which inserted in relation (8) leads to the formula:

$$\widetilde{F_1} = \frac{\widetilde{F_2}}{(\lambda G^*)^2}[(\lambda G^* - m_2\omega^2) - (\lambda G^*)^2](-m_1\omega^2 + \lambda G^* + k_1) \quad (8)$$

The transmissibility coefficient, in complex, is defined on the bass of the relation (8) as follows:

$$T^* = \frac{\widetilde{F_2}}{\widetilde{F_1}} = (\lambda G^*)^2\,[m_1m_2\omega^4 - m_2k_1\omega^2 + \\ +(k_1\lambda - m_2\omega^2\,\lambda - m_1\omega^2\,\lambda)G^*]^{-1} \quad (9)$$

By expressing the elasticity module G* as $G^* = G(1+j\delta)$, for the Romanian rubber used, relation (9) appears as:

$$T^* = \frac{A+jB}{C+jD}$$

or (10)

$$T^* = \frac{AC+BD}{C^2+D^2} + j\,\frac{BC-AD}{C^2+D^2}$$

where:

$$A = (1-\delta^2)\lambda^2\,G^2$$
$$B = 2\lambda^2 G^2\,\delta$$
$$C = m_1\,m_2\,\omega^4 - (k_1\,m_2 + \lambda G m_2 + m_1\lambda G)\omega^2 + \delta G_{ki}$$
$$D = \lambda\,G\,\delta\,(k_1 - m_1\,\omega^2 - m_2\,\omega^2)$$

The size of the transmissibility coefficient $T = |T^*|$ is obtained from (10) as:

$$T = |T^*| = [(A^2+B^2)/(C^2+D^2)\,]^{1/2} \quad (11)$$

with the dephasing angle given by the relation:

$$\Phi = \arctan\,[(B\,C - A\,D)\,/\,(A\,C + B\,D)] \quad (12)$$

Table 1 – Mechanical characteristics

Rubber hardness in ^{0}ShA	30	40	45	50	60	65	70	75
Mechanic angle of loss δ	0.04	0.05	0.08	0.15	0.20	0.25	0.30	0.40
Module of transversal elasticity G, in daN/cm^2	3.30	4.60	5.40	6.50	9.40	11.60	16.00	21.0

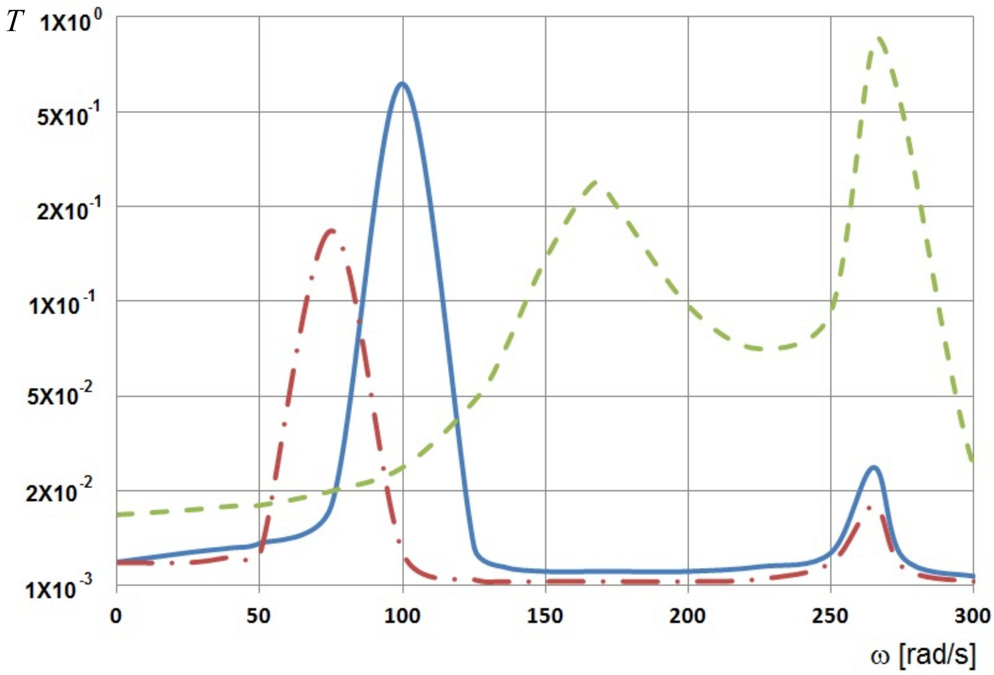

Figure 2. Vibration transmissibility

For the anti-vibration rubber currently used in the manufacture of amortization elements, depending on their hardness, the value pair δ, G is shown in Table 1, so that the relationships (10) become parameterized. In this regard, the transmissibility variation curves can be drawn according to the current variable and the pair of parameters δ, G (figure 2) for a machine with the following data: $m_1 = 1700$ kg, $m_2 = 270$ kg, $k_1 = 120 \cdot 10^6$ N/m.

For a rubber element, type S 1201 the geometric characteristics are: $\Phi = 0.5$; $\beta = 10$; $\lambda = 70$, and G and δ are chosen from table 1.

It is specified that the anti-vibration system of the machine is composed of eight rubber elements connected in parallel [18-21].

3. CONCLUSIONS

Based on the calculation model adopted, which contains an anti-vibration rubber insulation step shaped as a viscoelastic medium defined by the complex module of elasticity, the transmissibility coefficient was determined, depending on the current variable (pulsation of the disturbing force) and the physical-mechanical parameter of the rubber defined by the value pair (δ, G).

For the anti-vibration rubber with discrete hardness varying from 30^0 ShA to 75^0 ShA, characterized by the values of the parameters (δ, G) defined in Table 1, the families of curves T = (ω, δG) were drawn, which show the following:

a) the modification of the parameters of the anti-vibration rubber of the second stage, through the pair of values of parameters (δ, G), leads to the displacement of the first resonance zone, while the second zone remains unchanged.

b) the maxima of the first resonance zone, parameterized by the pair (δ, G) are characterized by a point with the lowest value in the whole spectrum of corresponding curves.

c) the working regime of the machine can be chosen between the two resonance zones, or in the most favorable case after the second resonance zone, where the transmissibility coefficient is much diminished, below the recommended values of 0.05...0.10.

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