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Methodical Base of Development of Damping Coverings

Mohammed A. Alnawafleh
Mechanical Engineering Department, Tafila Applied University College,
Al-Balqa' Applied University, P.O. Box: 179, Tafila (66110)-Jordan

Abstract: In order to reveal the physical possesses at suppression of the metal elements machines vibration, it is necessary to find out the connection nature of the radiating ability reduction for the damped detail with the properties of the applied means of damping.

Key words: Damping, intrinsic losses, laminated, vibrations, elasticity, mass, friction, new materials

INTRODUCTION

Frequently, the used means of damping are inefficient because their inadequate application. So a massive detail it is impossible to damp up by light weight covering with low factor and small of intrinsic losses. Or the vibrations of high frequency character cannot be suppression by rigid damping covering. For a correct estimation of properties of damping coverings and their adequate application this study is intended[1-6].

The analysis of oscillatory system: For understanding of the basic dependences of absorption of vibration energy we shall consider the system consisting of elements of mass m, elasticity C and active mechanical resistance R. The differential equation of the movement of such system making under influence of harmonic force $F_0e^{i\omega t}$ the vibrations $\xi = \xi_0 e^{i\omega t}$:

$$m\frac{\partial^2\xi}{\partial t^2} + R\frac{\partial \xi}{\partial t} + c\,\xi = [-\omega^2 m + c(1+i\eta)]\xi_0 e^{i\omega t} = F_0 e^{i\omega t} \quad (1)$$

where ξ is the amplitude of bias, ξ_0, is the amplitude of bias in an initial instant, ω is the cyclic frequency, F_0 is the initial force, η is the loss coefficient, τ is the time, e is the exponential curve. The value $\eta = \frac{\omega R}{C}$ refers to as the energy losses factor in oscillatory system. The energy spent in system for half of the period of vibrations, is:

$$W_0 = \frac{R\dot{\xi}^2}{4f} = \frac{1}{2}\pi R\omega\xi_0^2 = \frac{1}{2}\pi\xi_0^2 C\eta \quad (2)$$

The maximal potential energy is:

$$W_p = \frac{C\xi_0^2}{2} \quad (3)$$

From expressions (1), (2), we find:

$$\eta = \frac{W_0}{\pi W_p} \quad (4)$$

Factor of losses, hence, determines the absorption of the energy in oscillatory system[3, 6, 5]. The oscillating plate of the final sizes represents oscillatory system with infinitely large number of resonances. After overlaying of oscillating plate by the damping covering the amplitude of vibrations of the plate will decrease. The efficiency of the absorption of vibrations A will make thus

$$A = 101g\frac{\rangle\dot{\xi}_0^2\langle}{\rangle\dot{\xi}^2\langle} = \frac{\eta}{\eta_0} \quad (5)$$

where, $\dot{\xi}_0$, η_0 are the values before overlaying, $\dot{\xi}$, η are the values after overlaying of the plate with the damping covering and A is the vibration level in decibels.

Existing vibroabsorbing coverings are distinguished by character of deformation, occurring in them, of sparingly viscous material into three groups: 1) in coverings the deformations of a stretching of a layer of springly viscous material are used; 2) the deformations of shift of a layer of springly viscous material are used; 3) the deformations arising at the expense of waves, extending in a direction of thickness of a layer of springly viscous material (bend and longitudinal vibrations) are used[7, 4].

Soft vibroabsorbing coverings: The coverings of this group are called soft because soft springy viscous (usually rubber) materials are used for them. To find the efficiency of the vibroabsorption, it is necessary to define the losses factor of a plate with a covering:

Corresponding Author: Mohammed A. Alnawafleh, Mechanical Engineering Department, Tafila Applied University College, Al-Balqa' Applied University, P.O. Box: 179, Tafila (66110)-Jordan

$$\eta = \frac{w_0}{\Pi W_p} = \frac{R_e Z_n}{\omega m} \qquad (6)$$

where m is the mass of oscillatory system, $R_e Z_n$ is the real part of mechanical resistance and Z is the complex resistance.

The losses factor of a plate with a soft covering can be found as follows:

$$\eta = \frac{2sh(v_2\eta_2 - \eta_2 sin(2v_2)}{2\chi v_2[cos(2v_2) + ch(v_2\eta_2)] + \frac{1}{\eta_2}[\eta_2 sin(2v_2) + 2sh(v_2\eta_2)]} \qquad (7)$$

where $v_2 = k_2 h_2$

h_2 is the covering thickness in meters, k_2 is the springy wave number of a compression wave in a covering material, $\chi = \frac{m_1}{m_2}$, m_1, m_2 are the plate and covering area weight respectively, (kg/m), η_2 is the losses factor of a covering material.

As expression (7) is rather bulky and inconvenient for calculations, we use it in certain intervals of frequencies. For low frequencies (V_2<<) expression (7) looks like:

$$\eta = \frac{\eta_2 V_2^2}{3(1+\chi)}$$

With frequency decreasing the losses factor aproaches 0.

For resonant frequencies expression (7) is transformed into:

$$\eta_{pn} = \frac{\eta_2}{1 + \chi\eta_2 v_{pn} th(v_{pn}\eta_2/2)}$$

where

$$v_{pn} = \left(n - \frac{1}{2}\right)\pi, \quad n = (1,2,3\ldots)$$

The expression for resonant frequencies looks like:

$$f_{pn} = \frac{2n-1}{4h_2} \cdot C_2, \quad n = (1,2,3\ldots)$$

where C_2 is the longitudinal wave spreading speed in a covering.

The greatest value of the losses factor η_{pn} attains for the first resonance frequency. And then expression (7) looks like:

$$\eta_{p1} = \frac{\eta_2}{1 + 1{,}23\chi\eta_2^2}$$

Using the covering material application condition with losses factor of $\eta_2 = \eta_{2max}$,

$$\eta_{2max} = 0{,}9\sqrt{\frac{m_2}{m_1}}$$

it could be noticed that the increase of losses factor η above η_{pn} is not meaningful [5].

Maximum value of the losses factor of the covered plate at $\eta = \eta_{max}$ is:

$$\eta_{max} = 0{,}45\sqrt{\frac{m_2}{m_1}}$$

Increasing the maximum of the covered plate losses factor needs increasing the covering weight on its surface unit. For anti-resonant frequencies expression (7) is:

$$\eta_{apn} = \frac{\eta_2}{1 + \chi\eta_2 v_{apn} cth(v_{apn}\eta_2/2)} \qquad (8)$$

where $V_{apn} = \pi n$, n=(1, 2, 3...)

Expression for anti-resonant frequency looks like:

$$f_{apn} = \frac{n \cdot C_2}{2h_2}, \quad n = (1,2,3\ldots) \qquad (9)$$

The covered plate losses factor for the first anti-resonance frequency is described by the following formula:

$$\eta_{ap1} = \frac{\eta_2}{1 + 2\chi} \qquad (10)$$

For high frequencies ($v_2\eta_2>2$) the following expression is valid:

$$\eta = \frac{\eta_2}{1 + \chi\eta_2 v_2} \approx \frac{1}{\chi v_2} = \frac{\rho_2 c_2}{m_1\omega} \qquad (11)$$

where ρ is the density

From expression (11) it is visible that for very high frequencies the covering losses factor as well as when the frequency decreases tends to 0.

Analysis of the found correlations shows that the plate losses factor, reverted by a soft covering, has the greatest value for the first resonance frequency of the elastic waves extending in the covering thickness direction. As it is seen from the analysis the thicker the covering and the less the speed of compression wave within the covering the lower the resonance frequency.

According to (7), the increase of compressibility and density of a material can attain the reduction of speed of compression in springy viscous material. Becomes clear, why for the discussed kind of coverings it is required just soft springy viscous material. This result is necessary to take into account when selecting a material for a soft covering, effective on low frequencies.

For high frequencies the losses factor of the reverted plate does not depend on the losses factor of springy viscous material, if the condition $v_2\chi\eta_2>>1$ is kept. It corresponds to the loading of a plate as endless field with acoustic resistance $\rho_2 c_2$. In this case the energy, which is transmitted from the plate to the covering is fully absorbed by the last.

Putting soft vibroabsorbing covering on a plate making longitudinal vibrations, the shift waves extending on the covering thickness will arise. Losses factor in such a plate will be defined also by formula (7), in which k_{02} should be understood as the wave number of the shift wave in the covering material. The module of a soft springy viscous material shift is close in size to the module of compression. Therefore the frequency characteristic of the covering losses factor of such a material practically will be identical both for longitudinal and for the plate bend vibrations.

Reinforced vibroabsorbing coverings: The energy losses factor of bend wave in the plate reverted with reinforced vibroabsorbing covering, can be found as follows:

$$\eta = \eta_2 \frac{\alpha_2^3\beta_2 + 12\alpha_{21}^2\alpha_2\beta_2 + 12\alpha_{31}^2 g_2\gamma_1 (\alpha_3\beta_3 - \gamma_{21}^2\alpha_2\beta_2)}{1 + \alpha_2^3\beta_2 + \alpha_3^3\beta_3 + 12\alpha_{21}^2\alpha_2\beta_2 + 12\alpha_{31}^2\alpha_3\gamma_1(1 + g_2 + \eta_2^2)} \quad (12)$$

where

$$\alpha_2 = \frac{h_2}{h_1}; \alpha_3 = \frac{h_3}{h_1}; \alpha_{21} = \frac{h_{21}}{h_1}; \alpha_{31} = \frac{h_{31}}{h_1};$$

$$\beta_2 = \frac{E_2}{E_1}; \beta_3 = \frac{E_3}{E_1}; h_{21} = \frac{1}{2}(h_1 + h_2); h_{31} = \frac{1}{2}(h_1 + h_3) + h_2$$

$$\gamma_1 = \frac{1}{(1+g_2)^2 + \eta_2^2 g_2^2}; \gamma_{21} = \frac{k_{n2}}{k_{u1}}$$

The analysis of the structure of the formula (12) shows that the first member of numerator defines losses of energy caused by a bend of springy viscous layer, while the second by a stretching, and third by a shift. It is easy to show that

$$\eta = \frac{\eta_2}{1 + \chi\eta_2 v_2} \approx \frac{1}{\chi v_2} = \frac{\rho_2 c_2}{m_1\omega}$$

and hence, first two members of numerator can be neglected. Besides in the last member of numerator of the formula (12), the inequality $\alpha_3\beta_3>>\gamma^2_{21}\alpha_2\beta_2$ is valid. In view of the made assumptions, the expression (12) will be

$$\eta = \frac{\eta_2\gamma g_2}{(1+g_2)^2 + g_2^2\eta_2^2 + \gamma g_2[1 + g_2(1+\eta_2^2)]} \quad (13)$$

where

$$\gamma = \frac{12\alpha_{31}^2\alpha_3\beta_3}{1 + \alpha_2^3\beta_2 + \alpha_3^3\beta_3 + 12\alpha_{21}^2\alpha_2\beta_2} \quad (14)$$

As it is seen from the formula (13), the value of the losses factor of a plate is higher if the parameter γ is bigger. The expression (14) shows that the increase of the parameter γ is promoted by the observance of a condition

$$\alpha_2^3\beta_2 + \alpha_3^3\beta_3 + 12\alpha_{21}^2\alpha_2\beta_2 << 1 \quad (15)$$

With the observance of this condition, the rigidity of springy viscous and armature layers practically do not influence over rigidity of a compound plate.
The value of the parameter γ is defined geometrical and rigid characteristics of the reinforced covering. If the condition (15) is observed then the parameter

$$\gamma \approx 12\alpha_{31}^2\alpha_3\beta_3 \quad (16)$$

Apart from parameter γ, the value of factor of losses of the reverted plate depends also on parameter g_2 and of the factor of losses of springy viscous layer. From the (15) it is possible to define value g_2, for which the factor of losses of a plate becomes the maximal one.

$$g_{opt} = \frac{1}{\sqrt{(1+\gamma)(1+\eta_2^2)}} \quad (17)$$

Having substituted (13) in (17), we shall find the maximal value of the factor of losses of a plate.

$$\eta_{max} = \frac{\eta_2\gamma}{(2+\gamma) + 2\sqrt{(1+\gamma)(1+\eta_2^2)}} \quad (18)$$

From (18) it is possible to get the extremely possible value of the factor of losses of a plate η_{max} at the given value of the factor of losses η_2. Enhancing γ to infinity, we shall have:

$$\lim_{\gamma\to\infty} \eta_{max} = \eta_2 \quad (19)$$

Thus, the factor of losses of a plate η_{max}, riveted by the reinforced covering, can not be higher than the factor of losses of springy viscous layer.

Taking into account that:

$$k_u^2 \approx k_{u1}^2 = \omega\sqrt{\frac{m_1}{\beta_1}} \quad (20)$$

The expression for g_2 can be:

$$g_2 = \frac{\mu_2}{D_3 h_2 \omega}\sqrt{\frac{\beta_1}{m_1}} \quad (21)$$

where D is the constructional parameter and μ is the viscosity

Having juxtaposed (20) and (21), we shall find the value of the frequency, the η takes a postion in the frequency equation:

$$f_{opt} = \frac{\mu_2}{2\pi D_3 h_2}\sqrt{\frac{\beta_1(1+\gamma)(1+\eta_2^2)}{m_1}} \quad (22)$$

Materials for reinforced vibroabsorbing coverings: To ensure the maximal vibroabsorption at average frequencies interesting for us of the sound range, the module of shift of springy viscous materials for these coverings should be of rather low value. The specified requirements are met by some sorts of a rubber and also special high polymer of plastics.

The last ones are made as extremely viscous sticky mass used also as a material, connecting reinforced layer with damped plate. Such coverings usually refer to as damping tape. The technology of putting on of this covering is extremely simple. It is, however, effective with rather small thickness of damped plate. The reinforced coverings on the base of a rubber can be put on either by gluing, or by vulcanizing of a rubber in connection with metal layers. As reinforcing layers the steel foil of thickness around 10^{-4} m is usually used.

The soft, rigid, laminated and reinforced vibroabsorbing coverings intended for complex noise control of machines have been considered. It is experimentally proved that damping of thin-walled elements of machines, the protections, and casings ensure the effect of noise control of 5 – 8 dB.

Volumetric damping of toothed wheels and frames of machines, of large-sized members, ensures the effect of noise control of 5-12 dB depending on used means and frequency of activation. Thus, the level of oscillatory speed of vibrations can be reduced up to 12-15 dB[1, 2, 6].

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