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Analytical and Experimental Research of Compressive Stiffness for Laminated Elastomeric Structures

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Abstract

In this work stiffness characteristics of multilayer elastomeric packages are discussed. Package consists of alternating thin metallic and elastomeric layers jointed by vulcanization or gluing. Such packages are used as compensators, shock- absorbers, vibroisolators. The analytical expression of "compression force - displacement" dependence is derived for the flat thin-layered rubber-metal element (TMRE) on the basis of the variational principle. Force is directed flatwise; reinforcing metallic (steel) plates-layers are assumed to be perfectly rigid. Analytical solution was confirmed by experimental data for flat packet of circular cylinder shape. Based on the "force-displacement" dependence the expression of static compressive stiffness as the function of displacement was derived for TRME; it may be used in the equation of motion of single-mass object protected from low frequency vibration by means of TMRE packet.

Keywords: elastomers; multilayers paskage; compressive stiffness; variational principle; Ritz's method; vibration damping

1. Introduction

Elastomers (rubber and rubberlike materials) are a unique family of materials which offer many engineering advantages. Physical properties of elastomers, as polymeric materials, are qualitatively different from traditional construction materials because of their ability to maintain large elasticity deformation and small volume compressibility under deformation [1, 2, 3].

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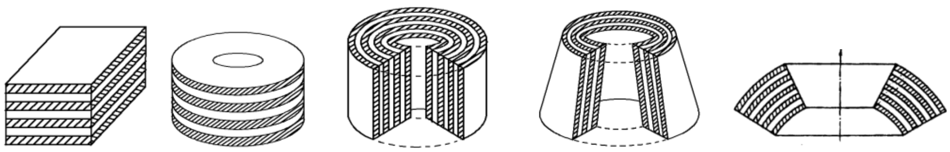
Laminated elastomeric structures, or reinforced elastomers, consist of a large number of alternating thin layers of elastomeric and reinforcing layers of other much more rigid material, usually metal. This allows obtaining the structures, which axial compression stiffness is in several orders greater than the shear stiffness. The connection of elastomeric with reinforcing layer is usually done by means of vulcanization or gluing. Packages of thin-layered rubber-metal elements (TRME) successfully replace traditional technical systems, such as bearing, joints, compensating devices, shock-absorbers because of its important advantages: improving of machines dynamics, vibration and noise reducing, low shear and compression stiffness ratio [4-8]. These structures are used in machine building, shipbuilding, civil engineering, aviation and aerospace due to its unique mechanical properties.

In practice the TRME packages of different geometric shapes are used: flat, cylindrical, conical and others; number of layers may be different, at least three (Fig. 1).



a) b) c) d) e)

Fig. 1. Multilayer elastomeric structures examples: a) flat of rectangular shape, b) flat ring, c) cylindrical, d) conical, e) spherical.

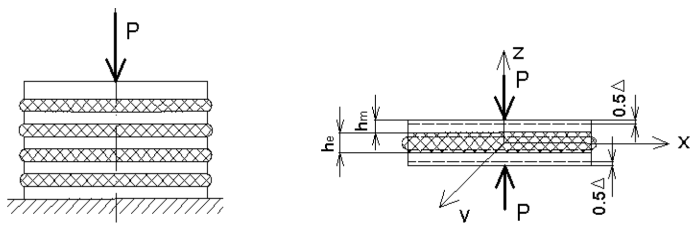
In many applications of TRME structures it is necessary to know its stiffness characteristics, in particular, if TRME packet is used for vibration isolation of the object from vibrating base. The elastic compensation device mounted between the vibrating base and protected object is the main element of any passive vibration protection system. In this case the amplitude of the protected body oscillations depends on the excitation frequency and on the possibility of resonance phenomenon occurrence. Natural circular frequency λ and linear frequency f of vibrating mass may be estimated if stiffness characteristics of vibroisolator are known [5, 6].

The objective of this work is flat TRME stiffness characteristics analytical determination and its comparison with experimental data from literature [7-9]. Usually stiffness characteristics (compressive or shearing) are determined by the solutions or boundary-value problems of the theory of elasticity, but it is very complicated problem. In given work stiffness characteristics are defined by using of variational methods, in particular the principle of minimum of total potential energy of deformation, having applied of Ritz's procedure.

2. Analytical investigation of compressive stiffness of laminated elastomeric structures

For vibration protection of objects with large masses the compensating TRME isolators, consisting of a sufficiently thin elastomeric layers and steel layers ($\rho = a / h > 20 \div 30,$ where a and h - the width and thickness of the elastomeric layer respectively) find expanding applications in recent years.

For these structures in the low-deformation domain (up to 5% ÷ 10%) high intensity of the external load (up to 200 MPa) may be exerted in practice. Experimental studies [7 ÷ 14] indicate that under these loads a significant non-linearity of the "force-displacement» stiffness characteristics associated with the physical nonlinearity of elastomeric materials take place. Traditional methods of calculation [5] do not allow to describe non-linear stiffness characteristics of these elements. Deformation of TRME under axial force normal to flat surface is shown in Fig. 2.



a) b)

Fig. 2. Deformation of flat TRME: a) deformation of packet, b) deformation of bond rubber layer.

For determination of stiffness characteristics for laminated elastomeric structures with regard to the physical nonlinearity it is more suitable to use variational methods. It is assumed that the geometry of the elastomeric layer, which allows imposing a significant external load, provides small deformation, i.e. problem remains geometrically linear For boundary value problems of static theory of elasticity for low volume compressible materials only the physical group of equations - the ratio between stress tensor σ_{ij} and strain tensor ε_{ij} changes [5]. Taking into account experimental data from [1, 3-6] σ_{ij} - ε_{ij} dependence may be expressed as [3]:

$$\sigma_{ij} = K\varepsilon_{ll}\chi(\varepsilon_{ll})\delta_{ij} + 2G\eta(\varepsilon_{ll})\left(\varepsilon_{ij} - \frac{\varepsilon_{ll}\delta ij}{3}\right), \quad \text{where} \quad \varepsilon_{ll} = \frac{3(1-2\mu)}{2(1+\mu)}\frac{s\varphi(s)}{G} \tag{1}$$

where: G and K - shear modulus and bulk modulus of the physically linear material; ε_{ll} - volume strain; δ_{ij} - Kronecker symbol; $\chi(\varepsilon_{ll})$, $\eta(\varepsilon_{ll})$ - functions of volume change; $i, j, l = x, y, z$; s - average normal stress (or hydrostatic pressure), $s = (\sigma_{xx} + \sigma_{yy} + \sigma_{zz})/3$; $\varphi(s)$ - a hydrostatic pressure function; μ - Poisson's ratio of elastomeric for linear Hooke's law.

Volume change functions $\chi(\varepsilon_{ll})$ and $\eta(\varepsilon_{ll})$ allow to describe the physical nonlinearity of bulk and shear deformations depending on volumetric strain (or hydrostatic pressure). Here and further the summation over repeated indices is performed. In solving boundary value problems functions $\eta(\varepsilon_{ll})$, $\chi(\varepsilon_{ll})$ and $\varphi(s)$ is represented by power series:

$$\eta(\varepsilon_{ll}) = 1 + \eta_n\varepsilon_{ll}^n, \ \chi(\varepsilon_{ll}) = 1 + \chi_k\varepsilon^k, \ \varphi(s) = 1 + \varphi_k s^k, \quad n=1, 2, ...; \ k=1,2,.... \tag{2}$$

$$lim\,\chi(\varepsilon_{ll}) = 1, \ lim\,\eta(\varepsilon_{ll}) = 1, \ \text{if} \ \varepsilon_{ll} \to 0; \ \ lim\,\varphi(s) = 1, \ \text{if} \ s \to 0. \tag{3}$$

For stiffness characteristics of rubber elements definition the variational methods are used, in particular, the principle of minimum total potential energy of deformation Π, using Ritz's procedure [5, 8]. For equation (1) ÷ (3) the expression for Π is written as:

$$\begin{aligned} \Pi &= U_V + U_f - A = (U_{1V} + U_{1f}) + (U_{2V} + U_{2f}) - A = \\ &= \int_V \left(\frac{K\chi_k\varepsilon_{ll}^{2+k}}{2+k} + G\eta_n\varepsilon_{ll}^n\left(\varepsilon_{ij} - \frac{1}{3}\varepsilon_{ll}\delta_{ij}\right)^2 \right) dV + \int_V \left(\frac{1}{2}K\varepsilon_{ll}^2 + G\left(\varepsilon_{ij} - \frac{1}{3}\varepsilon_{ll}\delta_{ij}\right)^2 \right) dV - \int_F P_i u_i dF, \end{aligned} \tag{4}$$

where: U_v – strain energy of the elastomeric layer due to volume change; U_f - strain energy of elastomeric layer due to distortion; $(U_{1V} + U_{1f})$ - part of potential energy of deformation, which describes the contribution of physical nonlinearity, $(U_{2V} + U_{2f})$ - part of potential energy of deformation, if Hooke's law is valid (no physical nonlinearity); P_i - given external forces; u_i -the components of sought displacement function; V - volume of the elastomeric layer; F - body surface on which form the loading is specified.

Approximate solution may be simplified if solution of corresponding linear problem already exists or may be simply obtained. Required displacement vector $\boldsymbol{u}$ of physically nonlinear problem may be expressed by available solution of a linear problem $\boldsymbol{u}^*$ with correction factor C: $\boldsymbol{u} = C\,\boldsymbol{u}^*$. Correction factor C depends on the element geometry, on the mechanical properties of elastomeric and loading parameters of isolator. Since the boundary value problem is geometrically linear, the same dependence takes place both for the strain ε_{ij} and for the required stiffness properties $\Delta(P)$ of rubber elements: $\varepsilon_{ij} = C\varepsilon_{ij}^*$, $\varepsilon_{ll} = C\varepsilon_{ll}^*$, $\Delta = C\Delta^*$ (ε_{ij}^*, ε_{ll}^*, Δ^* - known solutions of physically linear boundary value problems). After integrating the potential energy Π will depend on only one unknown parameter – factor C:

$$\Pi(C) = D_k C^{k+2} + B_n C^{n+2} + 0.5C^2 A^* - CA^*, \tag{5}$$

where: $$D_k = \int_V K \frac{\chi_k \varepsilon_{ll}^{*k+2}}{2+k} dV, \quad k=1, 2,... \qquad B_k = \int_V G\eta_n \varepsilon_{ll}^n \left(\varepsilon_{ij} - \frac{1}{3}\varepsilon_{ll}\delta_{ij} \right)^2 dV, \quad n=1, 2, ... \tag{6}$$

Parameter C is found from non-linear equations: $\dfrac{d\Pi(C)}{dC} = 0$:

$$(k+2)D_k C^{k+1} + (n+2)B_n C^{n+1} + CA^* - A^* = 0 \ . \tag{7}$$

From experimental data analysis [5,6,7,8] it succeeds that required accuracy of calculation allows to limit ourselves by one or two expansion coefficients in the functions of equation (4). Thus, for the case $k = n = 1$ we obtain:

$$D_1 = \frac{1}{3} K\chi_1 \int_V \varepsilon_{ll}^{*3} dV, \qquad B_1 = G\eta_1 \int_V \varepsilon_{ll}^{\ *} \left(\varepsilon_{ij}^* - \frac{1}{3}\varepsilon_{ll}^*\delta_{ij} \right)^2 dV. \tag{8}$$

If the element is loaded with point axial force P the work A* is equal:

$$A^* = P\Delta^*, \quad \text{where} \quad \Delta^* = \frac{P}{G\gamma^*(a,b,h,\mu)}. \tag{9}$$

where: Δ^* - displacement of rubber elements along the line of force P action; γ^*- rigidity of the element, the analytical expression of which is known from the solution of the linear problem; *a, b, h* - the geometrical dimensions of the elastomeric layer.

From equation (7) we obtain an expression for factor C and required characteristics "displacement- force" for rubber elements under axial compression, with the physical nonlinearity can be written as:

$$C = \left(\sqrt{1 + 12\frac{D_1 + B_1}{A^*}} - 1 \right) \frac{A^*}{6(D_1 + B_1)}, \qquad \Delta = C\Delta^* = \left(\sqrt{1 + 12\frac{D_1 + B_1}{A^*}} - 1 \right) \frac{A^*}{6(D_1 + B_1)} \frac{P}{G\gamma^*(a,b,h,\mu)}. \tag{10}$$

It is easy to verify that if $\chi_1 = 0$ and $\eta_1 = 0$ (i.e. there is no physical nonlinearity), then $D_1 = 0$, $B_1 = 0$ and $C = 1$, and from equation (10) well-known solution of the physically linear problem follows. From equation (10) a special case follows: if $\chi_1 \neq 0, \mu \neq 0,$ and $\eta_1 = 0$, this equation describes the dependence of the "force - displacement" of thin elastomeric layers, when the main contribution to the displacement gives the volume strain elastomeric layer and form changing of elastomeric layer can be ignored (in the extreme case - this compression elastomeric layer in a perfectly rigid "matrix").

As an example, the axial compression of a solid cylindrical isolator is considered (Fig.3). Solution of physically linear problems with the weak compressible elastomeric layer has the form [5]:

$$u^* = \frac{0.75\Delta^*}{N_0}\left(1 - \frac{4z^2}{h^2}\right)\frac{r}{h}, \qquad w^* = -1.5\Delta^*\left(z - \frac{4z^3}{3h^2}\right)\frac{1}{h}, \qquad N_0 = 1 + (1 + 4.5\rho^2)\frac{G}{K}, \tag{11}$$

$$\Delta^* = \frac{P}{G\gamma^*(a,b,h,\mu)}, \quad \gamma^*(a,b,h,\mu) = \frac{\pi b^2 N_1}{hN_0}, \quad N_1 = 3.6 + 1.5\rho^2, \quad \rho = \frac{b}{h}.$$

where: *b, h* - the radius and the height of the elastomeric layer; u^*, w^* - displacement in the direction of the axes r and z.

From equation (7) ÷ (11) after simplifying transformations, we obtain the dependence of the "force - displacement" $(\Delta(P))$ for the isolator with one elastomeric layer:

$$\Delta = \frac{sh\left[1+\chi_1 s+\left(1+4.5\rho^2\right)\left(1+\varphi_1 s\right)\dfrac{G}{K}\right]}{G\left(1+\varphi_1 s\right)\left(1+\chi_1 s\right)\left(3.6+1.5\rho^2\right)}, \qquad s=\frac{P}{A} \tag{12}$$

For vibroisolator with N number of elastomeric layers dependence "force-displacement" is equal: $\Delta_\Sigma = \Delta \cdot N$.

3. Experimental verification of compression stiffness of TRME

In technical literature [7, 8] there are a large number of experimental data of *K(s)* modulus at volume compression of the packets with thin ($\rho = b/h >>1$) flat elastomeric layers, for which we can assume that $s \approx P/A$.

In Fig. 4 plots of *"K (s) / K - s"* and *"G (s) / G - s"* is given for the elastomeric 2959 ($K \approx 2760$ MPa -bulk modulus, calculated at low pressures $s \approx P/A$, where *A* - area of the loaded surface of elastomeric layer, $G \approx 1.17$ MPa - shear modulus, calculated at low pressures) [8,10].From Fig. 4 it follows that in equation (3) we can limit ourselves by linear approximation. For the bulk modulus: $K(s) = K(1+\chi_1 s)$, $\chi_1 \approx 7\cdot 10^{-3}$ MPa^{-1}; for the shear modulus: $G(s) = G(1+\varphi_1 s)$, $\varphi_1 \approx 2\cdot 10^{-2}$ MPa^{-1}.

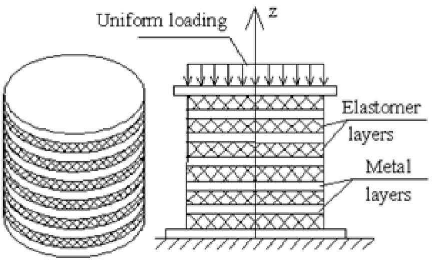


Fig. 3. Scheme of testing object.

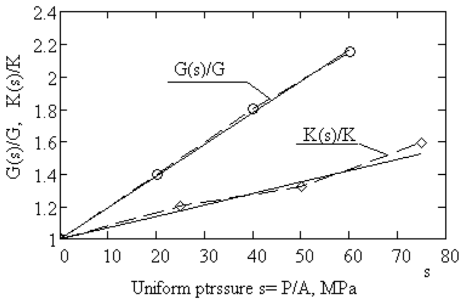


Fig. 4. Plot of dependence *G(s)/G – s*, -experimental points; plot of dependence *K(s)/ K – s*, - experimental points , -approximating curves.

In work [9] the experimental results are given to describe "force – displacement" characteristics of 12- layers cylindrical TRME package under axial compression. Radius of the elastomeric layer is $b=27.5$ mm, thickness of the elastomeric layer $h=1$mm. In table 1 the vertical displacement of the 12-layer thin rubber-metal element received experimentally, calculated in accordance with linear and nonlinear models are presented for data: $\chi_1=7\cdot 10^{-3}$ MPa^{-1}, $\varphi_1 \approx 2\cdot 10^{-2}$ MPa^{-1}, $G=1.17MPa$, $K=2760$MPa, G/K=$4.2.10^{-4}$, b=2.75cm, A=23.74cm^2, $h=0.1$cm, number of layers $N=12$.

Table 1. Vertical displacement of the 12-layer thin rubber-metal element

Vertical force, P	Uniform pressure, s =P/F	Experimental data	Results in accordance with	
			Linear solution	Nonlinear solution
kN	MPa	mm	mm	mm
10.0	4.20	0.089	0.092	0.088
20.0	8.50	0.161	0.185	0.173
30.0	12.60	0.230	0.278	0.240
40.0	16.90	0.300	0.368	0.310
60.0	25.70	0.410	0.570	0.420
80.0	34.00	0.520	0.730	0.530
100.0	42.00	0.600	0.920	0.630
140.0	58.80	0.740	1.290	0.790

The experimental results and calculations using formulas of equation (11) and equation (12), corresponding to the physically linear solution (if $\chi_1=\varphi_1=0$) and the solution taking into account the physical nonlinearity of elastomeric, are graphically shown in Fig. 5 – vertical displacement dependence on axial force.

In Fig. 6 compressive stiffness C=P/z (kN/mm) dependence on vertical displacement z (mm) is presented in accordance with experimental data and according to approximated curve. Stiffness curve may be approximated by linear dependence on displacement with formula:

$$c = c(z) = c_0 + zc_1 = 103.12 + z108.67\,. \tag{13}$$

For the dependence "force-displacement" in accordance with linear theory $c=108.45$ kN/mm.

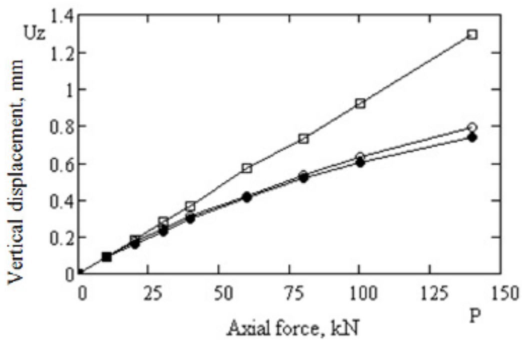


Fig. 5. Plots of dependence the vertical displacement on axial force: experimental data, - in accordance with linear theory.

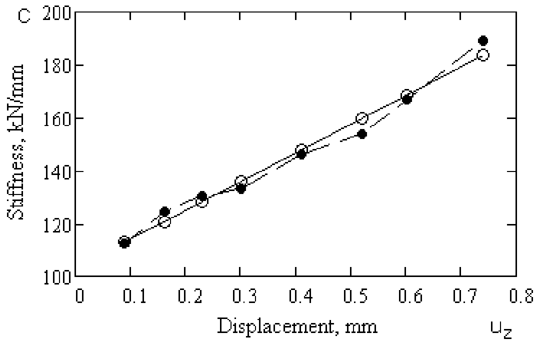


Fig. 6. Plots of dependence the compressive stiffness on vertical displacement: experimental data, - in accordance with approximating curve.

4. Analytical model of behavior of laminated elastomeric vibroisolator and its numerical solution

Laminated elastomeric vibroisolator with discussed above characteristics is placed between the object to be protected and a vibrating base (Fig. 7). The lower plate of the vibroisolator is subjected to kinematic excitation. In this paper the periodic excitation $\xi(t) = \Delta\sin(\omega t)$, is taken for numerical solution. It is assumed that the external excitation is independent of motion of the system to which it is applied. Determination the law of motion of the upper plate, on which the protected object is located, is the important problem of passive systems. In this case excited vibration amplitudes of plate depend on the excitation frequency and on the possibility of resonance phenomenon occurrence in an oscillating system "protected objet - vibroisolator".

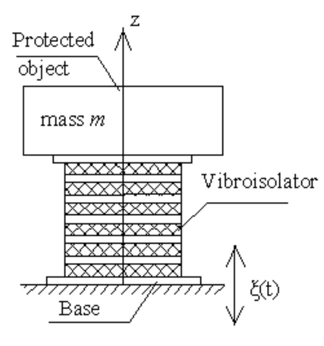


Fig. 7. Scheme of vibration protection of object.

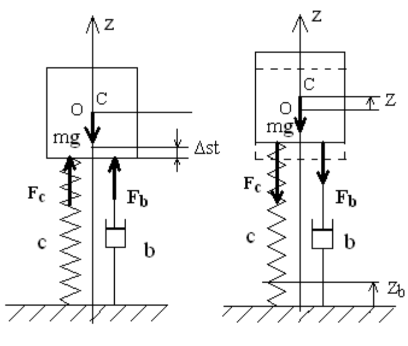


Fig. 8. Analytical model of vibrating object.

The dynamic component of vibroisolator response D(z, ż) composed of elastic force *cz* and the resistance force *bż*, proportional to the strain rate ż: $D(z,\dot{z}) = cz + b\dot{z},$ where: *c* - compressive stiffness; *b* - damping coefficient. The analytical model of this system is represented in Fig. 8, differential equation of protected object motion under kinematic excitation is:

$$m\ddot{z} = -c(z - z_b) - b(\dot{z} - \dot{z}_b), \qquad (14)$$

where z - mass center displacement in respect to static equilibrium center.

As it was established above, the compressive stiffness of laminated vibroisolator may be defined as:

$$c = f(z - z_b) = c_0 + c_1(z - z_b), \quad \text{then} \quad m\ddot{z} = -c_0(z - z_b) - c_1(z - z_b)^2 - b(\dot{z} - \dot{z}_b). \qquad (15)$$

Since harmonic law of vibratory motion of the base is specified as $\xi(t) = \Delta sin(\omega t),$ equation (15) will be as the following:

$$\ddot{z} + \frac{d}{m}\dot{z} + \frac{c_0}{m}z + \frac{c_1}{m}z^2 = \frac{d}{m}\omega\Delta cos(\omega t) + \left(\frac{c_0}{m} + \frac{2c_1}{m}z\right)\Delta sin(\omega t) - \frac{c_1}{m}\Delta^2 sin^2(\omega t). \qquad (16)$$

In this system after some transient motion the forced oscillation established in accordance with: $z = \alpha_0 + \alpha\, sin(\omega t + \beta),$ where the first harmonic is dominate. Here α_0 - deviation of the middle of swing amplitude from the position of static equilibrium, α- vibration amplitude, β- phase shift between oscillations of protected object and vibration action $\xi(t)$.

Equations of motion may be simplified if we denoted coordinate of the relative motion of mass center in respect to the base as $z^r = z - z_b$, then $\ddot{z}^r = \ddot{z} - \ddot{z}_b$, and $\ddot{z} = \ddot{z}^r + \ddot{z}_b$; substitute this to equation (16):

$$\ddot{z}^r + \frac{d}{m}\dot{z}^r + \frac{c_0}{m}z^r + \frac{c_1}{m}z^{r2} = \omega^2\Delta sin(\omega t).$$

In this work differential equation (16) is solved numerically by Euler method using MathCAD program, results are presented in Fig. 9 ÷ 13. Damping coefficient is taken from reference literature on rubber products recommendations: $\delta = d/2\lambda = 0.02 \div 0.25$, where $\lambda = f(z)$ - natural vibration circular frequency; in presented bellow examples $\delta = 0.25$. The comparison with linear model is executed.

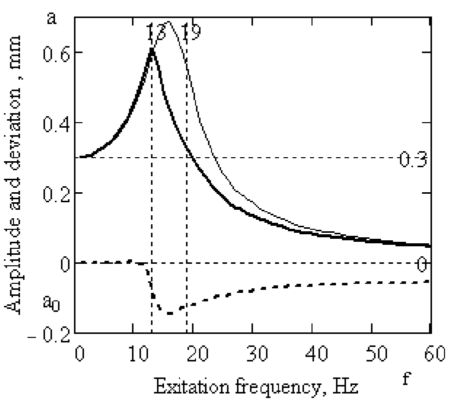


Fig. 9. Plot of dependence of amplitudes and deviations of steady-state motion on excitation frequency for vibrating mass m = 10 t under amplitude of excitation Δ = 0.3 mm, in accordance with: —— linear model, ▬▬ nonlinear model, - - - - deviation of the middle of swing amplitude in case of nonlinear model.

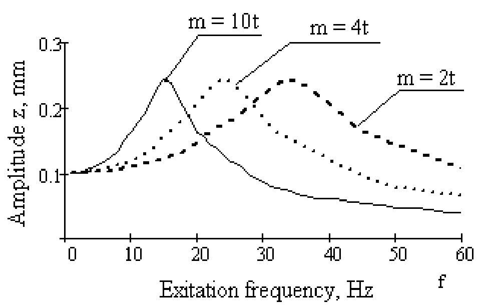
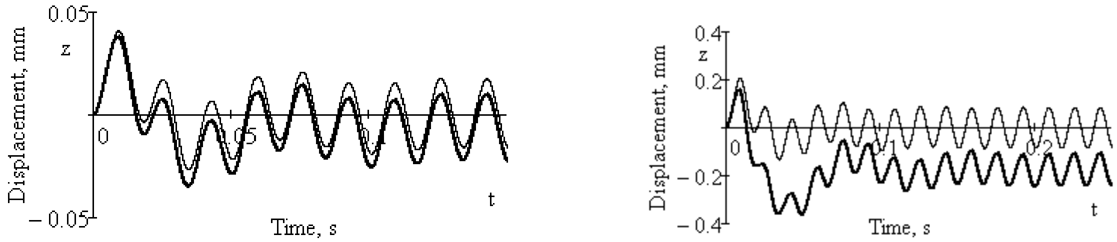
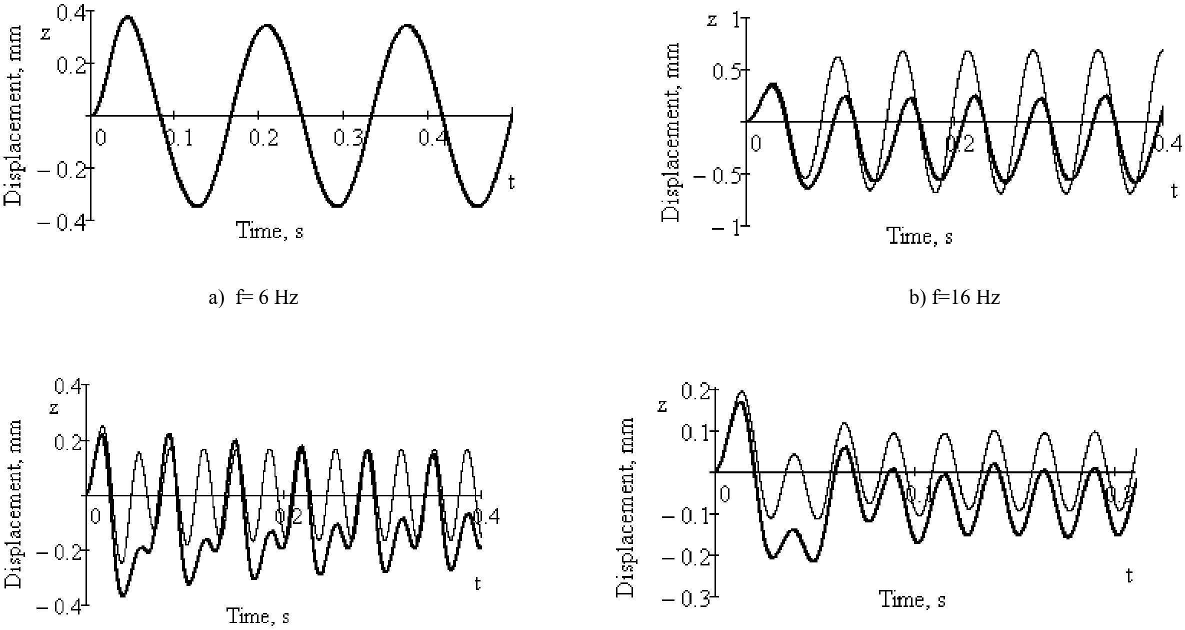


Fig. 10. Plot of dependence of maximal displacement on excitation frequency for vibrating mass m = 2t, 4t,10t, under amplitude of excitation Δ =0.1 mm for nonlinear model.



a) Δ = 0.1 mm b) Δ = 0.5 mm

Fig. 11. Plot of dependence displacement on time for vibrating mass m = 10t under frequency f=60Hz in accordance with: —— linear model, ━━━ nonlinear model.



a) f= 6 Hz b) f=16 Hz

c) f=30 Hz d) f=40Hz

Fig. 12. Plot of dependence displacement on time for vibrating mass m =10t with excitation amplitude Δ =0.3 mm in accordance with: —— linear model, ━━━ nonlinear model for different excitation ftequency.

In pre-resonance frequency range operation of linear and non-linear insulators coincides, but in post-resonance zone amplitudes of the steady - state motion of nonlinear insulator are smaller, however, with frequency of forced action increasing, the amplitude converging. Value of deviations of mid-span oscillation of the position of static equilibrium increases with the amplitude of forced action increasing.

Static stiffness in the dynamics equations can be used for preliminary calculations in case of low-frequency excitation because the parameter p:

$$p = \frac{\pi v_c}{h_e},$$

where v_c - shear waves propagation velocity, is several kHz and the orders of real frequencies is tens Hz.

5. Summary

The thin-layered rubber-metal structures applying is still limited because of the complicated theoretical calculations and the lack of simple calculation models. This paper presents a simple model of a flat TRME package, which takes into account the physical nonlinearity of elastomeric, provided that the task remains geometrically linear. Experimental studies indicate that, under these loads a significant non-linearity of the "force-displacement" stiffness characteristics associated with the physical nonlinearity of elastomeric materials takes place. Traditional methods of calculation do not allow to describe non-linear stiffness characteristics of these elements.

In this work the analytical research of the compression stiffness characteristics of flat TRME structures under action of the axial force in normal to the layers direction is presented.

The analytical expression for “force-displacement” characteristics equation (12) for flat TRME of circular cylindrical form is derived on the basis of the variational principle; metallic plates-layers are assumed as perfectly rigid.

Analytical solution in accordance with the received formula shows the good coincidence with experimental data. This proves that the posed problem is completely solved and corresponds to the research of other scientists.

Approximating equation “stiffness - displacement” of linear type equation (13) was derived and used in the equation of motion of the object, protected against vibrating by means of TRME packet.

In future investigations it is necessary to clarify the dissipative properties of the rubber and to develop a model of dissipative forces; to clarify the limits of application of the formulas for calculating of static stiffness of the TRME dampers; to develop a model of dynamic compression stiffness and its verifying.

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