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Constants of Metal Rubber Material

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Abstract

Damping materials made of pressing wire, such as Metal Rubber (MR material), “metal-flex”, “spring cushion” etc. are used widely in vibration protection systems. They have high strength and damping, however they are non-linear and anisotropic. To use contemporary finite element software for calculation of vibration insulators made of wire damping material one should know constants of this material. Till this time this problem almost isn’t researched, there is only a few data in linear approximation. A Young modulus for pressure and bending, shear modulus, Poisson ratio, friction force for pressure and for shear, friction coefficient between wire material and steel plate are obtained for MR material made of stainless steel wire for different load directions by static experiment in the present research. An influence of deformation, relative density, wire diameter, pressing force, deformation in other direction on these constants is considered. Peculiarities of pressing wire material deformation process are discussed: a difference of Young modulus for pressure and bending, a stabilization of hysteretic loop for one direction during load in another direction, an energy dissipation coefficient for different load directions. Results of the present research not only allow calculation of vibration isolator made of MR material by finite element software, but give a method for research of other pressing wire damping materials.

Keywords: MR material, Young modulus, shear modulus, friction force, Poisson ratio

1. Introduction

MR material (MR – Metal Rubber – is manufactured by cold pressure of wire spiral) are used widely for elastic-damping elements of vibration insulators (Ao et al, 2005; Jiang et al, 2008; Xia et al, 2009). It has high strength, high energy dissipation coefficient, ability to work under hugh and low temperature, in agressive media, in vacuum etc. Analogous wire materials are “metal-flex”, “spring cushion”, “wire mesh damper” (Gildas, 1989; Kozian and Schmoll, 1998; Al-Khateeb, 2002). These wire damping materials may be used in combination with squeeze film damper (Jiang et al, 2005). However these materials have large non-linearity and anisotropy (Ao et al, 2003). The only way to calculate elastic-damping element made of wire material by contemporary finite element software is to consider this material as anisotropic continuous media (Ulanov and Ponomarev, 2009) and to know constants of this material for different axis direction and dependencies of these constants on parameters of wire material and load. A present paper continues works (Yan et al, 2010a) and (Yan et al, 2010b) and contains a number of constant which is maximally possible in the present time.

2. Method

To obtain constants of MR material a static load experiment was used in the present research. Elastic-damping element with size $25\times25\times25$ mm was used. To research deformation of shear and tension this element was glued to plane surface. X axis is a direction of pressing during MR elastic-damping element manufacturing. Y and Z axis are perpendicular to direction of pressing. Compression direction is positive.

Parameters of wire material are diameter of wire d_w and relative density of material $\bar{\rho}$. The relative density is ratio of density of manufactured elastic-damping element to density of material of wire (for MR it is stainless steel usually). Pressure of manufacturing of elastic-damping element depends on its relative density:

$$\sigma_{press} \approx 576\bar{\rho}^{1.7} \tag{1}$$

Hysteretic loop of damping elements made of MR material is significantly nonlinear (Fig. 1). To make results more common it is better to use relative coordinates such as stress and relative deformation.

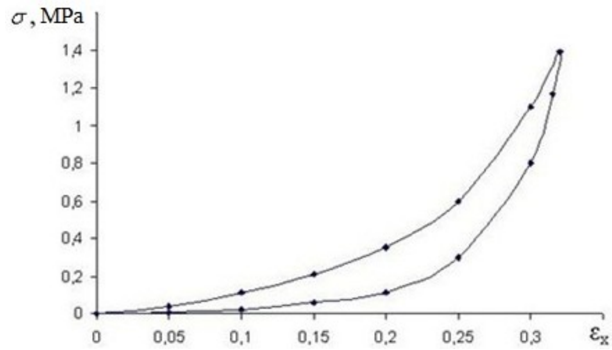


Fig. 1. Hysteretic loop of elastic-damping element used in the present research for X axis direction. $\overline{\rho}$=0.18, d_w=0.1 mm

Full stress consists from elastic stress σ_L and friction stress σ_H .

$$\sigma = \sigma_L \pm \sigma_H \tag{2}$$

Here "+" is for load process with stress σ_1 and "-" is for unload process with stress σ_2. Elastic stress is a middle line of hysteretic loop described by equation

$$\sigma_{\mathrm{L}} = (\sigma_1 + \sigma_2)/2 \tag{3}$$

Friction stress is difference between load (or unload) process and the middle line, thus

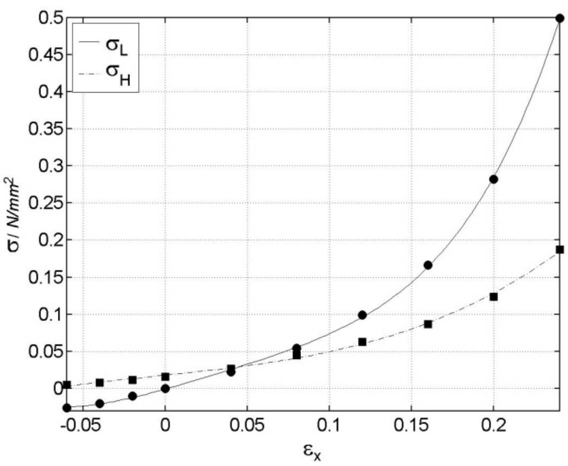
$$\sigma_{\mathrm{H}} = (\sigma_1 - \sigma_2)/2 \tag{4}$$

Equations (3) and (4) are correct if all wire contacts slip. For the beginning of unloading process it is not so. (Ulanov and Lazutkin, 1997) gives a range for correctness of (3) and (4) as $[0.7A_{\min};0.7A_{\max}]$ (here $A_{\max}$ and $A_{\min}$ are maximal and minimal amplitude of deformation respectively). For one time loading in experiment $A_{\max}$ and $A_{\min}$ may be much more than usual working amplitude of vibration isolator, so range $[0.7A_{\min};0.7A_{\max}]$ is sufficient for practice. For X axis a researched range of strain is $\varepsilon_x \in [-0.06;0.24]$, for Y and Z axis $\varepsilon \in [-0.06;0.16]$. Usual working strain of MR material is less (Yan et al, 2013). Range of relative density is $\overline{\rho} \in [0.18;0.3]$. Range of wire diameter is 0.1…0.25 mm.

3. Results

3.1 Young Modulus and Friction Stress for Pressure and Tension

The elastic stress and friction stress for $\overline{\rho} = 0.18$ and d_w = 0.1 mm for X and Y axis direction are presented at Fig 2 (Yan et al, 2010a). Equations (5) – (8) describe dependencies of stress on strain and relative density with an error about 5%.



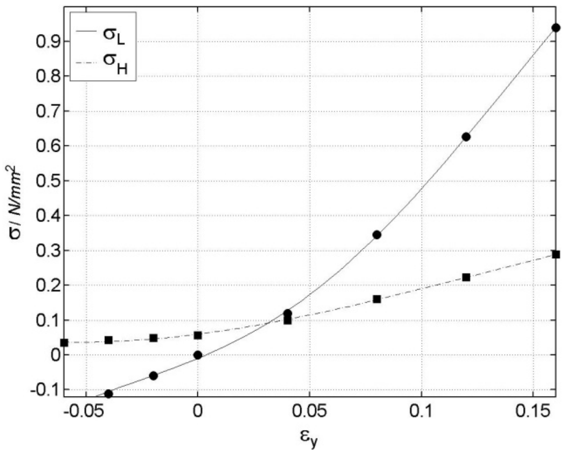


Fig. 2. Elastic stress and friction stress for $\overline{\rho} = 0.18$, d_w = 0.1 mm for X and Y axis directions

$$\sigma_{Hx} \approx (0.167 + 2.15\varepsilon_x + 0.641\varepsilon_x^2 + 71.6\varepsilon_x^3)\overline{\rho}^{1.3}, \text{ MPa;} \tag{5}$$

$$\sigma_{Hy} \approx (0.56 + 7.62\varepsilon_y + 60.5\varepsilon_y^2 - 156\varepsilon_y^3)\overline{\rho}^{1.3}, \text{ MPa.} \tag{6}$$

$$\sigma_{Lx} \approx (11 + 33\varepsilon_x - 376\varepsilon_x^2 + 2952\varepsilon_x^3)\varepsilon_x\overline{\rho}^{1.7}, \text{ MPa;} \tag{7}$$

$$\sigma_{Ly} \approx (49.6 + 262\varepsilon_y + 2560\varepsilon_y^2 - 11660\varepsilon_y^3)\varepsilon_y\overline{\rho}^{1.7} \text{ MPa.} \tag{8}$$

First efficient in equations (7) and (8) is analogous to Young modulus E.

Because Z axis is similar to Y axis, it is possible to find stress σ_{Hz} as σ_{Hy} and stress σ_{Lz} as σ_{Ly}.

To obtain dependency of Young modulus on wire diameter the elastic force and friction force for wire diameters 0.15 – 0.25 mm were divided by force for d_w = 0.1 mm. These ratios are required coefficients of influence. They are described by functions

$$f_{Hx} = -87.8d_w^2 + 30.8d_w - 1.19 \tag{9}$$

$$f_{Hy} = -27.2d_w^2 + 9.56d_w + 0.322 \tag{10}$$

$$f_{Lx} = -47.1d_w^2 + 16.5d_w - 0.174 \tag{11}$$

$$f_{Ly} = -7.1d_w^2 + 1{,}90d_w + 0.881 \tag{12}$$

Thus to take into account an influence of wire diameter it is enough to multiply the function (5) by coefficient (9), function (6) by coefficient (10) etc.

3.2 Young Modulus for Bending

To obtain Young modulus for bending a process of deformation of ring-shape elastic-damping element was considered (Xia et al, 2005). For a deformation of ring made of linear material under load *P* is known (Timoshenko S. and J.N. Goodier, 1951):

$$E_b = 0.149\frac{P \cdot R^3}{\delta \cdot J_x} = 0.149C_p \cdot \frac{R^3}{J_x} \tag{13}$$

here δ is deformation, R is radius of ring middle line, J_x is ring section inertia moment, C_p is ring stiffness. For little deformation $C_p \approx (|T_1| + |T_2|)/(|a_1| + |a_2|)$. It is possible to obtain segments T_1, T_2, a_1 and a_2 experimentally by a hysteretic loop (Fig 3).

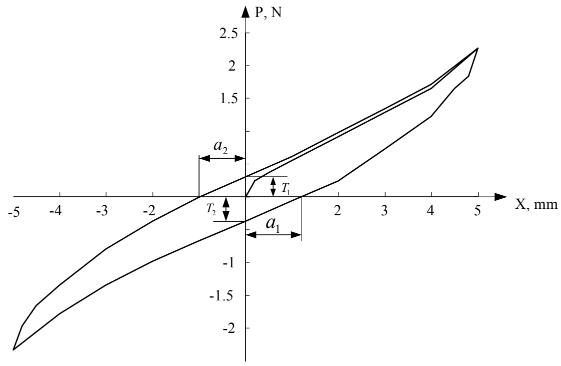


Fig. 3. Obtaining of T_1, T_2, a_1 and a_2 from hysteretic loop

Dependency of Young modulus on $\overline{\rho}$ and d_w is

$$E \approx 10.5 d_w (54.1\overline{\rho} - 1.38) \quad \text{(MPa)} \tag{14}$$

3.3 Shear Modulus and Friction Stress for Shear

In a case of shear the hysteretic loop (if to exclude its ends) is more near to linear (Fig. 4), thus it is possible to use a linear dependency for its description.

$$\tau = G\gamma \pm \tau_H \tag{15}$$

here τ_H is friction stress for shear.

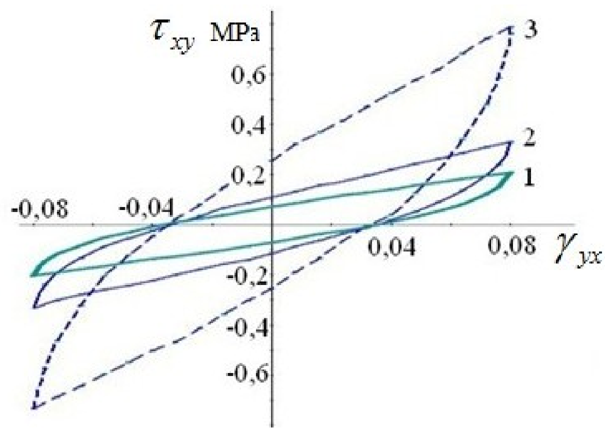


Fig. 4. Hysteretic loop of researched elastic-damping element for shear ($\overline{\rho}$ =0.18, d_w=0.1 mm) and its dependency of shear in XY axis direction on strain in X axis direction. 1 - ε_x =0; 2 - ε_x =0.08; ε_x =0.2

Dependencies on relative density (for d_w= 0.1 mm and the range of deformation $\gamma \in [-0,12; 0,12]$, which excludes ends of the loop) for shear direction XY are

$$G_{yx} \approx 6.65\overline{\rho}^{1.7} \text{ MPa}, \quad \tau_{Hxy} \approx 0,151\overline{\rho}^{1,3} \text{ MPa}. \tag{16}$$

For shear direction YZ

$$G_{yz} \approx 18,3\overline{\rho}^{1,7}, \text{ MPa}, \quad \tau_{Hyz} \approx 0,301\overline{\rho}^{1,3}, \text{ MPa}. \tag{17}$$

Shear modulus and shear friction stress are independent on wire diameter. It is shown in (Yan et al, 2010b) that it is possible if during shear most of wires in MR material work on shear but not on bending.

3.4 Dependency of Constants on Load in Other Direction

Pressure in one axis direction increases force in each wire contact. Of this reason a stiffness and friction should increases in other axis direction. Dependency of shear in XY axis direction on strain in X axis direction is presented on Fig. 4.

In this case if strain ε_x >0, it is necessary to multiply shear modulus G_{yx} and shear friction stress τ_{Hyx} on $(1+22\varepsilon_x^{1.4})$.

Influence of pressure in X axis direction on elastic and friction properties in Y direction is researched for analogous wire material in (Choudhry, 2004).

Because shear changes force in wire contacts insignificantly, there is no influence of shear deformation in Y direction on friction stress in X direction. An elastic stress in this case should be multiplied on

$$(1+1.25\gamma_{yx}).$$

3.5 Poisson Ratio

For load in X axis direction the Poisson ratio is $\mu_{yx}=\mu_{zx}\le 0.03$.

Usually it isn't necessary to take into account so little value, thus it is possible to assume that element made of MR material during its deformation in manufacturing pressing direction has no any pressure on its sides.

If a deformation is in Y direction, one should differ two cases: initial load and multi-time deformation. For initial load there are hysteretic loops in X and Z directions too (Fig. 5).

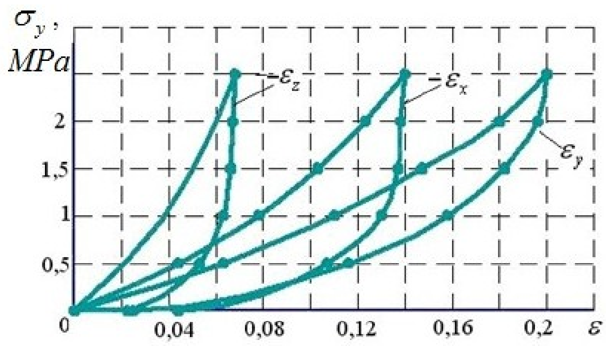


Fig. 5. Hysteretic loops in X and Z axis direction for initial load in Y axis direction

It is possible to recalculate these loops in coordinates $\varepsilon_y-\varepsilon_x$ (or $\varepsilon_y-\varepsilon_z$ respectively). Recalculated loops are described by equations

$$\varepsilon_x\approx-\mu_{x1}\varepsilon_y+\varepsilon'_{x1};\ \ \varepsilon_z\approx-\mu_{z1}\varepsilon_y+\varepsilon'_{z1}, \tag{18}$$

(here part with "+" is taken into account only for unloading process). In these equations coefficients μ are analogous to Poisson ratio, coefficients ε' are residual deformation.

For free surfaces of specimens equations (18) transforms into

$$\varepsilon_x\approx-0,67\varepsilon_y+(0,32-0,83\bar{\rho})\varepsilon_{y\max};$$
$$\varepsilon_z\approx-0,33\varepsilon_y+(0,16-0,41\bar{\rho})\varepsilon_{y\max} \tag{19}$$

If MR is glued to the load surfaces, coefficients of equations (15) are a little bit different:

$$\varepsilon_x\approx-0,67\varepsilon_y+(0,28-0,72\bar{\rho})\varepsilon_{y\max};$$
$$\varepsilon_z\approx-0,33\varepsilon_y+(0,14-0,36\bar{\rho})\varepsilon_{y\max}. \tag{20}$$

For multi-time compression in Y direction the structure of MR material becomes stabile. After 80...100 deformation cycles residual deformation stabilize near values

$$\varepsilon'_{x\max}\approx(0,92-2,3\bar{\rho})\varepsilon_{y\max};$$
$$\varepsilon'_{z\max}\approx(0,45-1,17\bar{\rho})\varepsilon_{y\max}. \tag{21}$$

Hysteretic loops for X and Z direction in this case are absent, dependencies (15) transform to

$$\varepsilon_x\approx-0,49\varepsilon_y;\qquad \varepsilon_z\approx-0,31\varepsilon_y \tag{22}$$

Thus Poisson ratio for multi-time compression are $\mu_{xy}=0,49$ and $\mu_{yz}=0,31$.

Poisson ratio for tension in a range of possible strain $-0.05<\varepsilon<0$ (if tension is more, structure of MR material will be destructed) is equal to 0.

3.6 Friction Coefficient

Beside elastic-damping elements made of MR material, vibration insulator has many other steel details (cups for elastic-damping elements, unload springs, bolts for connection etc). Some energy of vibration dissipates on friction between these details and elastic-damping element. This process depends on friction coefficient *f*.

This coefficient was obtained as a ratio between a force which moves an elastic-damping element along plane steel surface and a force which presses the element to this surface. The coefficient is independent on density and force direction (X axis or Y axis). If pressing stress is less than 0.25 MPa) the value of coefficient is 0.1, its maximal value is 0.125 for pressing stress 0.75 MPa. If pressing stress is higher, the friction coefficient continue to be constant.

4. Discussion

To explain obtained results one need to consider a structure of wire material (Zhu et al, 2011). A reason of increasing of Young modulus and friction stress for pressure is increasing of number of wire contacts during pressing.

There are two processes which form the dependency of Young modulus and friction stress for pressure on wire diameter. When the wire diameter increases, inertia moment of each wire increases too, stiffness of each wire and sum of elastic force increases of this reason. However from another side, if the wire diameter increases for the same density of MR, number of wire becomes less in a unit of volume. It means that number of wire contacts decreases too. Length of parts of wire (curve beams) between contacts becomes more and stiffness of these curve beams decreases. Two these processes together provides maximum for f_L coefficient.

Two different processes exist for friction force too. From one side, when the number of contacts decreases, sum of friction force in all contacts decreases too. From another side, when the number of contacts decreases, contact force in each contact increases, and friction force in each contact increases too. Two these processes together provides maximum for f_H coefficient.

For MR material the Young modulus for bending is more than for pressure or tension, and friction stress is significantly less. During the bending a detail has neutral axis near which a strain is near to zero. In these conditions the wires in MR material almost have no sleep. It makes stiffness of elastic-damping element more and energy dissipation in it less.

For example, for the same $\bar{\rho} = 0.18$ and d_w = 0.1 mm and deformation in Y axis direction (it takes place for ring-shape elastic-damping element, because during its manufacturing the ring is pressed in the direction of the ring axis) initial value of elastic stress near ε_y =0 for pressure is 2.69 MPa, for bending it is 8.7 MPa, friction stress for pressure is 0.06 MPa, for bending is 0.005 MPa (Yan et al. 2010a).

Equation for friction stress for bending (in contrast to Young modulus) isn't obtained in the present research. This stress depends not only on density and wire diameter but on thickness of MR element and amplitude of bend. Perhaps it is possible to solve this problem theoretically: to consider the bending element as a sum of many layers with compression or tension in each of them with different amplitude. Thus this problem needs future research.

Equations (5) – (12) and (16) – (17) allows approximate calculation of area of hysteretic loop (any mistake will be in the ends of loop, because many wire contacts still didn't sleep there) and area under a line of elastic force. A first value is an energy dissipated during one cycle of loading, second one is potential energy of deformation, their ratio is energy dissipation coefficient ψ. Thus it is possible to compare this coefficient for different load directions. If to take only first parts in equations (5) – (8) (let we suppose that $\bar{\rho}$ and d_w are the same and strain is little), ratio of elastic stress for Y and X axis directions is about 49.6:11=4.5, ratio of friction forces is about 0.56:0.167=3.35, therefore ratio $\Psi_x : \Psi_y \approx 1/(3.35:4.5) \approx 1.34$.

Thus the energy dissipation coefficient is more in direction of X axis.

Because the hysteretic loop for shear is near to linear, its area (with any mistake for the ends of loop) is about $2\tau_H \gamma$, and potential energy of deformation is $G\gamma^2/2$. Thus its ratio is $\tau_H / G\gamma$.

For the same shear angle γ from equations (16) and (17)

$$\psi_{yx} : \psi_{yz} \approx \frac{0.151\bar{\rho}^{1.3} 18.3\bar{\rho}^{1.7}}{6.65\bar{\rho}^{1.7} 0.301\bar{\rho}^{1.3}} \approx 1.38 \cdot$$

It is follow from equations (16) and (17) that ψ for shear is proportional to $(\bar{\rho})^{-0.4}$.

Values of ψ_x and ψ_{yx} are comparable.

For $\varepsilon = \gamma = 0.1$ ratio $\psi_{yx} : \psi_x \approx 1.12$.

For $\varepsilon = \gamma = 0.15$ ratio $\psi_{yx} : \psi_x \approx 0.82$.

Unfortunately till this time that few researchers who work on wire damping materials (Andrés and Chirathadam, 2011; Li et al, 2010; Tian et al, 2008) obtained for researched materials the linear stiffness and energy dissipation coefficient only, and for one-dimensional load process. It isn't enough to describe a behavior of material in contemporary vibration protection systems.

5. Conclusions

Results of the present research give constants for calculation of vibration isolators made of MR material by finite element software. It gives a method for research of other pressing wire damping materials (such as "metal-flex", "spring cushion", "wire mesh damper" etc).

All these results are presented for MR material made of stainless steel wire. However sometimes another materials are used in MR (for example, copper wire to increase heat conductivity). Thus the constants will depend on properties of wire material and on part of materials in a wire body. This problem needs future research.

If MR material is used in squeeze film damper, it is necessary to research how its constants will change in other media (such as oil).

If MR material is too thin, during its deformation some wires don't meet other wire, and increasing of number of contacts doesn't take place. All results above are correct if H/d_w ratio is more than 100 (here H if thickness of element in load direction). Elastic-damping elements for pipeline supports have H/d_w ratio about 10…30, it is necessary an additional research of its properties.

All results above are for new MR material. An important problem is a life-time of vibration insulators made of MR material. Constants of material will change during its wearing. It depends (Ao et al, 2006) on number of load cycles, vibration stress, static preload stress, density of material, wire diameter, working temperature of vibration insulator. The present paper gives a method for future researches of these influences.

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