



CALCULATION OF STIFFNESS OF RUBBER-METAL ELEMENTS OF ELECTRIC TRAINS AT TORSION TWISTING

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ABSTRACT

The article presents the method of calculation of rubber-metal elements of electric trains in torsion twisting.

A prerequisite for the correct design of rubber-metal elements is knowledge of their specific operating conditions, which allows you to choose rubber with the necessary properties [1,2,3].

Rubber and rubber-metal elements used in units of high-speed electric rolling stock perform different role in structures. By functional purpose, rubber elements are divided into power elements used to transmit dynamic load; shock-absorbing parts and gaskets used for shock, shock and vibration damping, as well as various types of sealing parts [2,3].

Rubber, as a structural material, is especially valuable in that it can, as a rule, perform several functions. Being a good vibration damper, it effectively dampens vibrations and noise, eliminates friction and wear of mating parts. The use of rubber in the assemblies facilitates installation, allowing the use of large tolerances, which reduces the cost of the assembly design; in hinged joints, external friction is replaced by internal friction in rubber, thereby eliminating wear and eliminating the need to lubricate and protect the joint from dust and dirt.

The main reliability requirement for rubber-metal elements for high-speed electric rolling stock is that their durability should ensure the absence of failures during the assigned service life and the assigned service life [4,5].

One of the main mechanical characteristics of rubber-metal elements is their quasi-static and dynamic rigidity [1,2,3].

The calculation of stiffness in torsion twisting is considered especially important for rubber-metal elements of electric trains, while this issue has not been studied enough in foreign literature [1,3].

At operation of cylindrical rubber-metal shock absorbers of electric trains for torsion twisting with torque M around OZ axis (Figure 1) rigidity can be determined by formula [4,5]

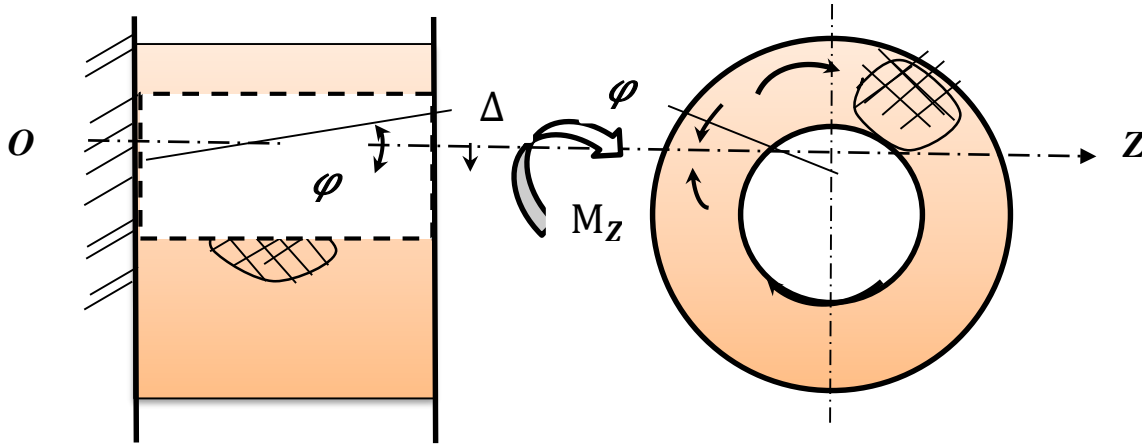


Figure 1. Design diagram of rubber-metal shock absorbers for torsion twisting.

for solid cylindrical shock absorber

$$G_{\varphi} = \frac{M_Z}{\varphi_Z} = \frac{G \cdot I}{h} = \frac{\pi G R^4}{2h}, \quad (1)$$

where I is the polar moment of inertia, which is

$$I = \frac{\pi R^4}{2}. \quad (2)$$

For a shock absorber with a hole, formula (1) will take the form

$$G_{\varphi} = \frac{M_Z}{\varphi_Z} = \frac{G \cdot I}{h} = \frac{\pi G R^4}{2h} = \frac{\pi G (R^4 - r^4)}{2h}, \quad (3)$$

where I is the polar moment of inertia (for a shock absorber with a hole), formula (4) will take the form

$$I = \frac{\pi (R^4 - r^4)}{2}, \quad (4)$$

R, r - respectively, the outer and inner diameters of the rubber-metal shock absorber;
 h - height of shock absorber, cm.

Rubber-metal shock absorbers of axle boxes of electric locomotives and electric trains work for torsion twisting. In these shock absorbers, one of the washers is rigidly connected with the heads of the leash, in the other with a rubberized roller. At vertical oscillations of the bogie frame relative to the wheelset axes, the rubber bushings of the rollers work for coaxial (concentric) twisting, and the end shock absorbers - for torsion twisting.

When the leash turns through the φ angle, the bearing plane of the inner washer rotates relative to the outer one, which increases the angular rigidity of the shock absorber of the leash head (Figure 1). The angular stiffness of two end shock absorbers of one head of the leash can be calculated using the formula

$$G_{\varphi sh} = \frac{2GI}{h} = \frac{2\pi G (D^4 - d^4)}{32h} = \frac{2\pi \cdot 100}{32 \cdot 1,3} \cdot (12^4 - 8,5^4) = 210 \frac{kN \cdot cm}{rad}. \quad (5)$$

In spring suspension of electric locomotives and electric trains, not cylindrical, but conical rubber-metal shock absorbers, which experience torsion twisting during operation, are most often used. These shock absorbers have rectangular sections enclosed between two support surfaces, experience simultaneous deformations of compression and shear (Figure 2). The rigidity of such a shock absorber depends on the angle of inclination of the support plates β .

The stiffness of the conical shock absorber can be determined by the formula [6,7]

$$G_{KA} = \frac{2F \cdot (E_P \sin^2 \beta + G \cdot \cos^2 \beta)}{h} \tag{6}$$

The calculated modulus of elasticity of the E_P depending on the shape factor is determined as well as under uniaxial compression.

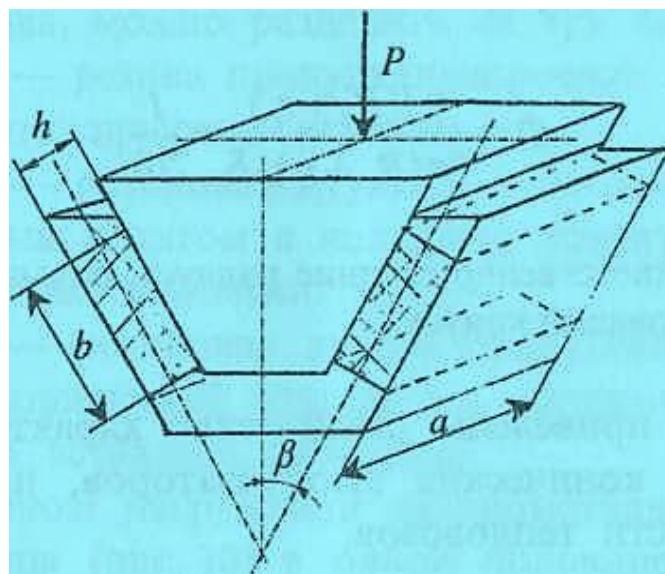


Figure 2. Design diagram for rectangular shock absorbers enclosed between two inclined bearing surfaces.

The form factor in this case is determined by the formule

$$FF = \frac{F_{OP}}{F_B} = \frac{a \cdot b}{2(a+b) \cdot h} \tag{7}$$

The cone-shaped shock absorbers under the action of the force P in the direction of the Z axis also experience joint deformation of shear and compression and can be calculated using formula (7). Here, the shape factor for determining the calculated modulus of elasticity can be approximately found using the formula (8)

$$FF = \frac{\pi l (R_c + r_c)}{2\pi (R_c + r_c) \cdot \delta} \tag{8}$$

where R_c, r_c - are the average radii of the larger and smaller cone bases, respectively.

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