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THE EFFECT OF ELASTIC RECOIL OF CLOSED-TYPE LIFTING ROPES AFTER THEIR MANUFACTURE AND DRAWING

Abstract. *A study was made of stresses and factors of internal forces in the elements of ropes of a closed structure during their manufacture. It is established that when twisting a rope of closed construction, its shaped wires suffer from bending deformation, twisting and stretching. In this case, the shaped wires are subjected to a complex load with the concomitant rotation of the axes of the stress tensor. The stresses in the elastic cross-sectional area of the rope wire are considered and its limit is determined for shaped cross-sections and for asymmetric shaped cross-sections. Formulas for approximate determination of tangential and normal stresses in the elements of a closed rope of non-circular profile are obtained: (wedge-shaped, zeta-shaped and x-shaped).*
Keywords: *ropes of closed construction; stress; internal force factors; cross section; shaped wires; elastic region; normal stress; elastic-plastic deformation.*

Experience in the manufacture and operation of ropes of closed design shows that immediately after twisting the closed rope, which is an elastic-plastic system, at the first load acquires significant elongations, and its stress-strain state changes significantly. As a result, a number of serious structural defects (bundles, "waves", wire breaks, etc.) often appear during the first cycles of rope operation, which is the reason for the failure of a new rope [1, 2].

The influence of the presence of gap between the wires in the outer layers and its value on the compatibility of operation of the layers in the radial direction and the preservation of the structural integrity of the rope during operation is revealed [3]. Analysis of the stress-strain state of closed rope elements under axial tension and torsion [4]. The closed rope consists of an outer layer of Z-profile wires, a subsurface layer of alternating round and H-profile wires [5, 6, 7]. Evaluation of extended structural elements using non-contact mobile systems using body waves and directional waves (piezoelectric, electromagnetic-acoustic transducers) [8]. The axial forces and torques in the cross sections of the layers are found to be redistributed when a rope turns under an external torque, which leads to a decrease in the safety factor of the rope, a violation of the compatibility of the axial and radial displacements of the layers, and a violation of the structural integrity of the rope in the form of breaks in the outer layer wires [9, 10]. The influence of wire cracks on the amplitude distribution of the generated field is specified for two steel rope kinds assuming surface and inner defects [11]. The behavior of short and very short fatigue cracks emanating from so-called "smooth" specimens with stress concentration is described [12].

To assess the reliability of the rope and prevent the occurrence of these defects, as well as to assess its strength and durability, it is necessary to know the stress-strain state of its constituent elements of wires, both during its manufacture and after the first axial load. It is established that when twisting a rope of closed construction shaped wires suffer together with bending and torsional deformation, and then during extraction during operation - tensile deformation. In this case, the shaped wires are subjected to a complex load with the concomitant rotation of the axes of the stress tensor. It should be noted that the bending of shaped wires of some profiles is oblique [3, 4]. For example, when twisting z-shaped and 8-shaped wires, the plane of the bending moment, which contains the normal of the helix, does not coincide with the main axis of the wire cross section y_0 or z_0 (Fig. 1).

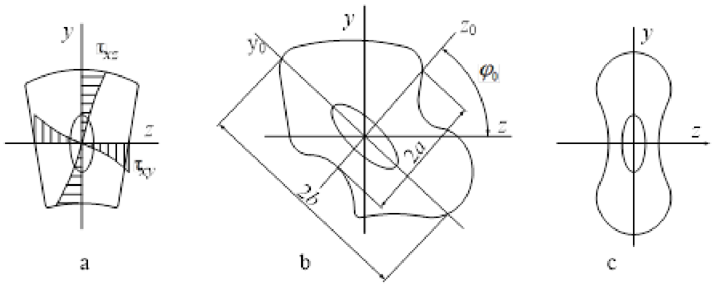

Figure 1 – Sections of shaped wires of closed ropes: wedge-shaped symmetrical (a); z-shaped asymmetric (b); 8-shaped symmetrical (c)

These wires experience oblique bending, and wires of round, x-shaped and wedge-shaped profiles undergo flat bending. Thus, the material of the wires is subjected to a complex load, the analysis of which the existing theories of plasticity do not solve. There are known developments of some methods for solving this problem, but these methods can be obtained only local solutions that meet the rigid arameters [1, 2, 5].

Thus, the study of technological stresses and internal power factors in the shaped wires of closed ropes in general form is an urgent scientific and applied task.

Given the complexity of the task of analysis of technological stresses and internal force factors in the shaped wires of closed ropes, the study is proposed to approximate its solution under the general assumption that the wire material is ideal – elastic-plastic, and the entire cross section of wires covered by elastic-plastic deformation.

Consider the stress in the elastic cross-sectional area of the rope wire and determine its limit. Normal bending stress is determined by:

– for shaped cross-sections symmetrical with respect to the bending plane *xy* (wedge-shaped, 8-shaped) wire ropes (see Fig. 1, a and c) by expression

$$\sigma_{x,y}^{be} = E\chi y, \tag{1}$$

where $\sigma_{x,y}^{be}$ – normal bending stress; $\chi = \sin^2\alpha / R$ - curvature of the centerline of the wire; R – wire twisting radius; E – Jung's module.
– (z-shaped) wire ropes (see Fig. 1, b) by expression for asymmetric shaped cross-sections

$$\sigma_{x,y}^{be} = E\chi\left(z_0 \sin\psi + y_0 \cos\psi\right), \tag{2}$$

where z_0 and y_0 – the main central axes of the wire.

In Figure 1, b φ_0 is the polar angle, which is calculated from the main central axis z_0.

Analyzing the plots of tangential stresses during torsion of non-round rods, for which exact solutions are obtained, it can be noted that they have slight nonlinearity (Fig. 1, a). This nonlinearity is smaller the greater the sloping delineation of the wire profile, and for elliptical and round cross-sections of wires the nature of the change in tangential stresses across the section is linear.

This fact gives the right to make a few additional assumptions. First, for smoothly delineated profiles of sections of shaped wires of a closed rope, the displacements γ_{xy} and γ_{xz} with a sufficiently probable approximation can be calculated from the linear functions of the *z* and *y* coordinates (Fig. 2).

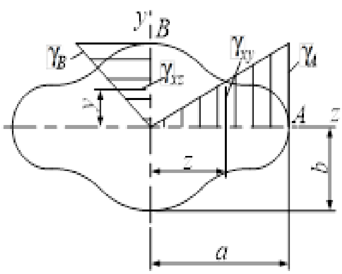

Figure 2 – Smoothly outlined cross-sectional profile of the shaped wire

Coordinate functions for this case

$$\begin{cases} \gamma_{xy} = \dfrac{\gamma_A}{a} z, & 0 \le z \le a; \\ \gamma_{xz} = \dfrac{\gamma_B}{b} y, & 0 \le y \le b; \end{cases} \tag{3}$$

$$\frac{\gamma_A}{\gamma_B} = -\frac{b}{a}, \tag{4}$$

where *a* and *b* – segments on the main axes of the cross section of the wire.

Secondly, the torque in the cross section of the wire with a smoothly delineated profile is determined by the expression

$$M_x = \int_F \left(\tau_{xy} z - \tau_{xz} y\right) \cdot dF = G \int_F \left(\gamma_{xy} z - \gamma_{xz} y\right) \cdot dF \ , \tag{5}$$

where τ_{xy} and τ_{xz} – projections of full stress on the corresponding planes; G – shear strength modulus.

Substitute formula (3) into expression (5) and, given equality (4), we obtain

$$\begin{cases} \tau_{xy} = \dfrac{b^2 M_x}{a^2 I_z + b^2 I_y} z \\ \tau_{xz} = -\dfrac{a^2 M_x}{a^2 I_z + b^2 I_y} y \end{cases} . \tag{6}$$

Total tangential stress:

$$\tau_{tors} = \sqrt{\tau_{xy}^2 + \tau_{xz}^2} = \frac{M_x}{a^2 I_x + b^2 I_y} \sqrt{b^4 z^2 + a^4 y^2} \ , \tag{7}$$

The twisting angle of the shaped wires is determined by the formula

$$\varphi = \frac{M_x l}{G I_к}, \tag{8}$$

where I_{tors} – the moment of inertia at pure torsion with a sufficient degree of accuracy can be determined by the formula of Saint-Venan [6].

$$I_{tors} = \frac{F^4}{4\pi^2 I_p} \ , \tag{9}$$

where F and I_p – plane and polar moment of inertia of the wire cross section, respectively.

From formula (8) we obtain:

$$M_x = G \cdot I_{tors} \cdot \frac{\varphi}{l}, \tag{10}$$

Thirdly, the moment of elastic recoil of each layer of wires of the closed rope, taking into account expression (1) is determined [1]:

$$M_{tors}^{techn} = n \cdot [(1 + \cos^2 \alpha) \sin \alpha M_z + \cos^3 \alpha M_x],$$

where n – the number of wires in each layer; α – the twist angle of the wires; M_z – the moment of bending relative to the z-axis of the cross section of each wire; M_x – torque in the cross section of each wire.

Based on the accepted assumptions and mathematical studies of stresses arising in the process of making ropes, simplified formulas for determining the bending and torques in the cross sections of closed rope wires, taking into account the coefficients A_1, B_1, N_1, N_2, N_3. Formulas for their determination in the polar coordinate system starting at the center of gravity of the wire cross section were obtained in [3].

With an approximate solution of the problem obtained:

– for shaped cross-sections of wires symmetrical about the plane of bending *xy* (x-shaped):

$$M_z = \sigma_{fl} \cdot E \cdot \chi \cdot A_1;$$

$$C_1 = \frac{GI_{tors}}{a^2 I_z + b^2 I_y}; \quad (11)$$

$$M_x = \sigma_{fl} \cdot \Theta C_1 \left(a^2 A_1 + b^2 B_1\right),$$

where σ_{fl} – conditional estimated yield strength; C_1 – coefficient that determines the dependence of the modulus of elasticity of the second kind on the axial moments of inertia for symmetrical shaped cross sections of wires; Θ – twisting of the axial line of the wire;

– for asymmetrical shaped cross-sections of wires (z-shaped):

$$\begin{cases} M_{z0} = \sigma_{fl} E\chi\left(N_2 \cdot \sin\psi + N_3 \cos\psi\right) \\ M_{y0} = \sigma_{fl} E\chi\left(N_2 \cdot \cos\psi + N_1 \sin\psi\right) \end{cases},$$

$$C_2 = \frac{GI_{tors}}{a^2 I_{z0} + b^2 I_{y0}},$$

$$M_x = \sigma_{fl} \cdot \Theta C_2 \left(b^2 N_1 + a^2 N_3\right),$$

where ψ – the angle that determines the position of the main central axes of inertia of the section relative to the axis perpendicular to the plane of bending of the wire; C_1 – the coefficient that determines the dependence of the modulus of elasticity of the second kind on the axial moments of inertia for asymmetric shaped cross-sections of wires; I_{z0} , I_{y0}, I_z , I_y – the main central moments of inertia of the cross section of the wire.

$$\begin{cases} A_1 = \dfrac{2}{3\sqrt{b_2}} \displaystyle\int_{\varphi_{i-1}}^{\varphi_i} \left(\dfrac{\cos^2 \varphi \rho_i^3(\varphi)}{\sqrt{1 - l^2 \sin^2 \varphi}} \right) d\varphi; \\ B_1 = \dfrac{2}{3\sqrt{b_2}} \displaystyle\int_{\varphi_{i-1}}^{\varphi_i} \left(\dfrac{\sin^2 \varphi \rho_i^3(\varphi)}{\sqrt{1 - l^2 \sin^2 \varphi}} \right) d\varphi; \end{cases}; \tag{12}$$

$$\begin{cases} N_1 = \dfrac{1}{3} \displaystyle\int_{\varphi_{i-1}}^{\varphi_i} \left(\dfrac{\cos^2 \varphi}{\Delta(\varphi)} \cdot \rho_i^3(\varphi) \right) \cdot d\varphi; \\ N_2 = \dfrac{1}{6} \displaystyle\int_{\varphi_{i-1}}^{\varphi_i} \left(\dfrac{\sin^2 \varphi}{\Delta(\varphi)} \cdot \rho_i^3(\varphi) \right) \cdot d\varphi; \\ N_3 = \dfrac{1}{3} \displaystyle\int_{\varphi_{i-1}}^{\varphi_i} \left(\dfrac{\sin^2 \varphi}{\Delta(\varphi)} \cdot \rho_i^3(\varphi) \right) \cdot d\varphi, \end{cases} \tag{13}$$

where

$$\begin{cases} b_2 = E^2 \chi^2 + 3C^2 a^4 \Theta^2; \\ l^2 = \dfrac{d_2}{b_2} - 1; \\ d_2 = 3C^2 b^4 \Theta^2; \end{cases} \tag{14}$$

The obtained results can be compared with the coefficients A_0, B_0 and C_0, which characterize the degree of plastic deformation under uniaxial loading (stretching) and have the same intensity as under complex loading.

$$\begin{cases} \Delta(\varphi) = \sqrt{A_0 \cos^2 \varphi + 0{,}5 B_0 \sin 2\varphi + C_0 \sin^2 \varphi}; \\ A_0 = (E\chi \sin \psi)^2 + 3C_2^2 b^4 \Theta^2; B_0 = E^2 \chi^2 \sin 2\psi; \\ C_0 = (E\chi \cos \psi)^2 + 3C_2^2 a^4 \Theta^2; \end{cases} \tag{15}$$

where φ and $\rho_i(\varphi)$ – current polar angle and radius-vector of points of the contour line of section; $\Delta(\varphi)$ – increase in the current polar angle; A_0, C_0 – characterize the degree of plastic deformation under uniaxial loading (stretching).

Conclusions.

1. Further study of the elastic recoil after pulling the closed rope using expressions (3) and (4) shows that the stress-strain state of the components of the closed ropes changes, while the redistribution of stresses in the cross sections of wires, and the moment of elastic recoil of the rope as a whole decreases. This significantly improves its operating conditions and increases service life.

2. The main criterion that determines the change in the moment of elastic recoil is the symmetry or asymmetry of the shaped cross section of the wire. In order to evaluate the effect of stretching on the magnitude of the moment of elastic recoil of the rope, a comparative analysis of the obtained experimental results was carried out, as a result of which a significant difference was observed in the distribution of forces over the layers of an unstretched and pre-stretched rope with a subsequent nominal load.

3. Calculations of technological internal force factors in the cross sections of wires and moments of elastic recoil on the layers of the rope make it possible not only to assess the degree of technological imbalance of the rope, but also rationally choose the direction of twisting in the layers. As a result, reduce the moment of elastic recoil and ensure the reliability and durability of the rope structure during its operation.

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ЕФЕКТ ПРУЖНОЇ ВІДДАЧІ ВАНТАЖОПІДІЙМАЛЬНИХ КАНАТІВ ЗАКРИТОГО ТИПУ ПІСЛЯ ЇХ ВИГОТОВЛЕННЯ ТА ВИТЯЖКИ

Анотація. *Відмінність закритих та напівзакритих канатів від інших їх видів полягає у застосуванні при їх звиванні дроту не тільки круглого, а і не круглого (клиноподібного, z-подібного, 8-подібного та х-подібного) перерізу, що дозволяє відтворити так званий ефект "замка" у вигляді щільного прилягання звитих дротин не круглого перерізу однієї до одної. Ця відмінність обумовлює формування між двома або трьома зовнішніми шарами каната замкнутих порожнин по усій його довжині, наявність яких забезпечує надійний захист каната від проникнення в його середину вологи, агресивних розчинів і виходу мастила з каната назовні. Завдяки такій захищеності канати даної конструкції знаходять широке застосування у підйомнотранспортному обладнанні, що використовується в будуванні технічних споруд, суднобудуванні, ливарному і хімічному виробництвах, в гірничодобувному обладнанні, в конструкціях висотних споруд, радіо і телевеж, а також мостів. В роботі проведено дослідження напружень та внутрішніх силових чинників в елементах канатів закритої конструкції при їх виготовленні. Встановлено, що при звиванні каната закритої конструкції, його фасонні дроти потерпають від деформації вигину, кручення і розтягування. При цьому фасонні дротини зазнають складного навантажування з супутнім поворотом осей тензора напружень. Розглянуто напруження в пружній області поперечного перерізу дроту каната і визначено його межу для фасонних поперечних перерізів і для несиметричних фасонних перерізів. Отримано формули для приблизного визначення дотичних та нормальних напружень в елементах закритого канату не круглого профілю: (клиноподібних, z-подібних та х-подібних). Отримані результати дають можливість оцінити ступінь технологічної неврівноваженості каната, раціонально вибрати направлення звивання по шарах, зменшити момент пружної віддачі і забезпечити надійність та довговічність структури каната в процесі його роботи.*

Ключові слова: *канати закритої конструкції; напруження; внутрішні силові чинники; поперечний переріз; фасонні дроти; пружна область; нормальне напруження; пружно-пластична деформація.*