DIAGNOSTICS OF STABILIZATION MODES OF SHAPE STRAND ROPES

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Abstract

The work is devoted to the diagnostic issues of the possibilities of technological balancing of steel shaped strand ropes of lifting and transport equipment, which is of great scientific and practical importance, since it will improve their performance. A literature review showed that this topic is practically not covered and only the issues of straightening (bending) strands, which are not effective enough, are considered. The research objective and the problem that must be solved to obtain ropes with better quality characteristics (greater durability and strength) have been formulated. A more accurate expression for the technological torque of shaped strands is presented. The ways of reducing the torque as a harmful factor are indicated. Diagnostics of possible options for technological balancing of ropes was carried out. Rational modes of manufacturing shaped strand ropes have been determined.

Keywords: diagnostics, internal force factors of wires, shaped strand ropes, balancing, properties of ropes, manufacturing technology

LIST OF SYMBOLS

Parameter	Indication	Dimension
average lay radius	a	m
coefficients of bending and torque	A_{1B}, A_{1H}, B_K, A_K	
semiaxes of the elastic zone of the ellipse in the wire section	a_e, b_e	m
amplitude value of lay radius	b	m
average lay angle	c	degree
amplitude value of the lay angle	e	degree
number of layers of core wires	i	
technological torque of the triangular strand	M_{tors}^{tech}	Nm
bending moment in wire section	M_b	Nm
torque in wire section	M_t	Nm
bending and torque in the core wires	$M_{b,i}$, $M_{t,i}$	Nm
technological torque of the layer of wires of the shaped strand	M_{tors}^{layer}	Nm
number of layers of wires in a shaped strand	m	
number of core wires	n_i	
curvature amplitude value	p	m^{-1}
curvature average value	q	m^{-1}
wire lay radius	R_{ij}	m

wire lay radius on a shaped strand	$R(\phi)$	m
wire length at the first laying step on a round cylinder	S	m
parametric equations of the screw axis of wires on a shaped cylinder	$x,\ y,\ z$	
boundary value of the ordinate of the elastic zone in the wire section	y_{upp}, y_{low}	m
wire lay angle on a round cylinder	α	degree
wire lay angle	α_j	degree
lay angle of the wires on the shaped cylinder	α_i	degree
angle between curvature radius and lay radius	γ_{ij}	degree
twisting coefficient	η	
total torsion deformation of the wires	θ	m^{-1}
kinematic torsion of wires	θ_{jt}	m^{-1}
kinematic torsion of wires on a shaped cylinder	θ_k	m^{-1}
radii of the upper and lower parts of the contour line of the ellipse of the elastic zone	ρ_{upp}, ρ_{low}	m
conditional yield strength of the wire material	$\sigma_{0,2}$	MPa
curvature of the wires of the shaped strand	χ	m^{-1}
wire curvature	χ_{ij}	m^{-1}

curvature of the wires of the shaped strand	$\chi(\phi)$	m^{-1}
kinematic curvature and torsion of wires on a round cylinder	χ, θ	
boundary angles of the elastic and plastic zones of the wire section	ψ_1, ψ_2	degree

1. INTRODUCTION

Steel ropes and reinforcing strands are widely used in industrial engineering. The first designs of twisted products were wound wires on round cylinders.

The long period of operation of such ropes has led engineers to the need to increase their performance and strength.

The cross section of the round strand and shaped strand ropes is shown in Fig.1.
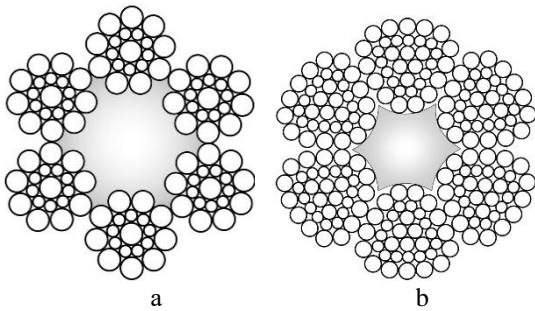

Fig. 1. The design of steel ropes: a - round-strand rope; b - triangular strand rope

It turned out to be practically impossible to increase the operational capabilities of the ropes due to the quality of the wire materials because of the absence of the corresponding metals in nature.

A way out for this problem was found by the engineers (for the first time in Germany) in the creation of ropes with a new geometry. Structures of steel shaped strand ropes appeared, where wires were wound on cylinders of triangular and oval sections.

It turned out that this design allows to reduce the frictional force of individual wires on the pulleys and drums (that is, to increase the lifetime of the ropes by 42%) and to increase the strength due to the high value of the cross-sectional utilization factor by 33% [1, 2].

The manufacture of such ropes has also been mastered in Ukraine at the Khartsyzk Steel Wire Rope Plant.

The shaped cylinder will have variable geometry (lay angle, curvature and torsion) and stress state. Existing works on the geometry and stress state of wires describe their behavior only approximately and with a distortion of the actual state.

For this reason, it is very difficult to diagnose and obtain sufficiently accurate optimal parameters of technological balancing and additional processing of steel shaped strand ropes.

In this regard, the determination of the optimal parameters of the manufacturing technology and additional processing of shaped strand ropes on the basis of more accurate solutions for the geometry and stress state of wires is an urgent scientific and practical problem in the mechanics of steel ropes [3-5].

2. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

The laying stresses, being summed up over the cross section of the wires, are integrally given to both the main vector and the main moment of internal forces.

The zones of elastic and plastic strains in the cross section of the wires of the shaped strand, depending on the laying mode, are shown in Fig. 2.
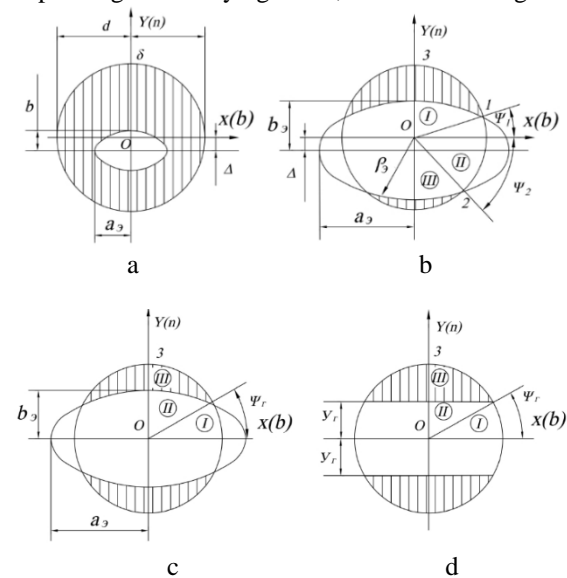

Fig. 2. Zones of elastic plastic strains in the cross section of the rods when winding on a shaped cylinder: a - bending + torsion + tension; b – bending + torsion with twisting + stretching; c – bending + torsion with twisting; d - bending

If we project the main vector and the main moment on the axis of the Frenet trihedron associated with the center of gravity of the wire section, then the normal ΣN_t and transverse forces Q_n, Q_b, bending M_n, M_b and torque M_t moments (t, n, b are the axes of the Frenet trihedron) are obtained.

These forces and moments form the so-called technological torque, which is a harmful factor. It causes twisting of the strands and the structure of the strands in sections may be disrupted [3]. Therefore, it is necessary to either eliminate M_{tors}^{tech} or reduce it in all possible ways. In [4], the value of M_{tors}^{tech} is presented in the form of an approximate expression

$$M_{tors}^{tech} = \sum_{j=1}^{m} \sum_{i=1}^{m} \left[M_{b,ji}(\chi_{ij} \sin \alpha_j + \theta_{ji} R_{ij} \cos_{ji} \cos \alpha_j \cos \gamma_{ji}) + M_{t,ji} \cos \alpha_j (1 - \alpha_{ji} R_{ji} \cos \alpha_j) \right] \tag{1}$$

where:
M_b – bending moment;
M_t – torque;
α_j – wire lay angle;
θ_{ji} – wire kinematic torsion;
R_{ji} – wire lay radius;
γ_{ji} – angle between the curvature radius and the lay radius;
χ_{ji} – wire curvature.

Here it is assumed that the lay angle is a constant value, and the curvature and torsion of the wires change abruptly, which does not correspond to the actual behavior of the wires. It can be seen that the values of M_{tors}^{tech} calculated by (1) will have significant errors [5-6].

One of the effective methods of reduction of M_{tors}^{tech} is the use of technological torsional strain. By adjusting its value, which is quite simple to implement, the value of M_{tors}^{tech} can be changed in very wide aisles. With regard to the winding of wires onto shaped cylinders, the optimal design for the technological torque has not been studied at all and is of scientific and practical interest [7].

This paper discusses the issues of reducing the value of M_{tors}^{tech} during the technological balancing of steel shaped strand ropes. It is noted that these issues do not yet have even an approximate coverage in the technical literature.

3. THE PURPOSE OF THE RESEARCH

In this regard, the purpose of the paper is to obtain the most accurate solutions for diagnostics of technological balancing of steel shaped strand ropes. This goal is achieved by solving the following tasks:
1. Determine the internal bending and torque moments of the wires, taking into account the variable geometric parameters.
2. Construct a more accurate expression for the technological torque.
3. Optimize the technological parameters of balancing shaped strand ropes.
4. Formulate recommendations for improving the properties of shaped strand ropes during manufacturing.

4. MATERIALS OF THE RESEARCH

4.1. Diagnostics and theoretical substantiation of technological balancing of shaped strands

The value of M_{tors}^{tech} of the shaped strand is equal to the sum of the moments of the internal force factors of the wires relative to the axis of the cylinder [6, 7]. According to [8], the technological torque of the shaped strand is determined by the expression:

$$M_{tors}^{tech} = M_{tors} + \sum_{j=1}^{m} M_{tors, j}^{layer}, \quad (2)$$

where:
m – number of wires in the layer;
M_{tors} – technological torque of the strand core, which is determined by the formula [9]:

$$M_{tors}^{tech} = \sum_{i=1}^{l} n_i \left[(1 + cos^2 \alpha_i) \, sin \, \alpha_i \, M_{b,i} + cos^3 \alpha_i \, M_{t,i} \right] \quad (3)$$

where:
n_i – number of wires of the core;
α_i – core wire lay angle;
l – number of layers of core wires;
$M_{b,i}, M_{t,i}$ – bending moment and torque in the core wires;
m – number of layers of shaped strand wires;
M_{tors}^{layer} – technological torque of the layer of shaped strand wires.

The value of M_{tors}^{layer} is determined by the expression [8]:

$$M_{tors,}^{layer} = k(M_b \cdot A_b + M_t \cdot A_t), \quad (4)$$

where:
$k, A_{b,j}, A_{t,j}$ – coefficients determined for triangular strands by the formulas:

k=3 for the triangular strand

$A_{b,j} = sin \, \alpha \, (0°) + 2 sin \, \alpha \, (30°) + sin \, \alpha \, (60°) + \theta_k R(0°) \, cos \, \alpha \, (0°) \cdot cos \, \gamma \, (0°) + $
$+ 2\theta_k R(30°) \, cos \, \alpha \, (30°) \, cos \, \gamma \, (30°) + \theta_k R(60°) \cdot cos \, \alpha \, (60°) \, cos \, \gamma \, (60°)$ (5)

$$A_{t,j} = cos \, \alpha \, (0°) + 2 \, cos \, \alpha \, (30°) + cos \, \alpha \, (60°) -$$
$-R(0°)\alpha(0°) \, cos \, \alpha \, (0°) \, cos \, \gamma \, (0°) -$
$-2R(60°) \, cos \, \alpha \, (60°) \, cos \, \gamma \, (60°)\alpha(30°) -$
$-R(60°)\alpha(60°) \, cos \, \gamma \, (60°)$ (6)

where:
$R(\varphi)$, $\alpha(\varphi)$ – functions of the lay radius and layer wire lay angle;
$\chi(\phi)$, Θ_k – curvature and kinematic torsion of layer wires;
$\gamma(\varphi)$ – angle between the lay radius and the curvature radius of the shaped cylinder contour.

The moments $M_{b,i}, M_{t,i}$ are determined by integration over the plastic and elastic zones of the wire sections [8]:

$$M_b = \pi E a_e b_e \left[(q + p \, cos \, 3 \phi) \frac{b_e^2}{4} \right]$$
$$+ 2\sigma_T \left(1 - \frac{E_1}{E} \right) E \times$$
$$\times (q + p \, cos \, 3 \phi) \cdot (A_{1b} + A_{1H}) + 2E_1(q + p \, cos \, 3)(x_o - x_e) \quad (7)$$

where:
$x_0 = \frac{\pi d^4}{64}$ – axial moment of the wire section inertia;
$x_e = \pi a_e b \cdot \frac{b_e^2}{4}$ – moment of inertia of the elastic zone ellipse.

Semi-axes of the elastic zone ellipse in the wire cross-section:

$$b_e = \sqrt{\frac{\sigma_T^2 [E^2(q + p \, cos \, 3\phi) + 3G^2 \theta_k^2]}{[E^2(q + p \, cos \, 3\phi)^2 + 3G^2 \theta_k^2]^2}} \quad (8)$$

$$a_e = \sqrt{\frac{\sigma_T^2 [E^2(q + p \, cos \, 3\phi) + 3G^2 \theta_k^2]}{3G^2 \theta^2 [E^2(q + p \, cos \, 3\phi)^2 + 3G^2 \theta_k^2]}} \quad (9)$$

where:
χ = q + $p$cos3 – variable curvature of the wires on

the shaped cylinder;

E – elasticity modulus of the 1st row of wire material;

σ_T – conditional yield strength of wire material.

Coefficients:

$$A_{1B} = \int_0^{\frac{\pi}{2}} \int_{\rho_{upp}}^{\frac{d}{2}} \rho^3 \sin^2 \psi \, [B_k(\rho \sin \psi)]^{-\frac{1}{2}} d\rho \psi; \quad (10)$$

$$A_{1H} = \int_0^{\frac{\pi}{2}} \int_{\rho_{ow}}^{\frac{d}{2}} \rho^3 \sin^2 \psi \, [B_k(\rho \sin \psi)^2 +$$

$$A_k \rho^2 \cos^2 \psi]^{-\frac{1}{2}} d\rho d\psi. \quad (11)$$

$$B_k = E^2(q + p \cos 3\phi)^2 + 3G^2 \theta_k^2; \quad (12)$$

$$A_k = 3G^2 \theta_k^2; \quad (13)$$

where:

G – elasticity modulus of the 2nd row of wire material;

θ_k – kinematic torsion of wires on a shaped cylinder.

The moments $M_{b,i}, M_{t,i}$ are determined by integration over the plastic and elastic zones of the wire cross sections [8].

$$M_{b,i} = \pi E a_э b_э \left[(q + p \cos 3\phi) \frac{b_э^2}{4} \right]$$

$$+ 2\sigma_T \left(1 - \frac{E_1}{E} \right) E(q + p \cos 3\phi) \times$$

$$\times (A_{1b} + A_{1H}) + 2E_1(q + p \cos 3)(x_o - x_э);$$

$$M_{t,i} = \frac{\pi d^4 G \theta_k E_1}{32E} + \frac{1}{2} G \theta_k \left(1 - \frac{E_1}{E} \right) [A_2 + 4\sigma_T(A_{2b} + A_{2H})],$$

where

$$A_2 = \int_0^{\frac{\pi}{2}} \rho_{upp}^4 d\psi + \int_0^{\frac{\pi}{2}} \rho_{low}^4 d\psi;$$

$$A_{2b} = \int_0^{\frac{\pi}{2}} \int_{\rho_{upp}}^{\frac{d}{2}} \rho^3 \, [B_k(p \sin \psi)^2 +$$

$$A_k p^2 \cos^2 \psi]^{-\frac{1}{2}} d\rho d\psi; \quad (14)$$

$$A_{2b} = \int_0^{\frac{\pi}{2}} \int_{\rho_{low}}^{\frac{d}{2}} \rho^3 \, [B_k(p \sin \psi)^2 +$$

$$A_k p^2 \cos^2 \psi]^{\frac{1}{2}} d\rho d\psi,$$

ρ_{upp}, ρ_{low} – the radii of the upper and lower parts of the contour line of the ellipse of the elastic zone, which are determined by the formula:

$$\rho = -\frac{a_e^2 \Delta \sin \psi}{b_e^2 \cos^2 \psi + a_e^2 \sin^2 \psi} \pm$$

$$\pm \sqrt{\frac{a_e^4 \Delta^2 \sin^2 \psi}{(b_e^2 \cos^2 \psi + a_e^2 \sin^2)^2} - \frac{a_e^2(\Delta^2 - b_e^2)}{b_e^2 \cos^2 \psi + a_e^2 \sin^2}}$$

«+» – to calculate ρ_{upp}, and «-» – for ρ_{low}.

In this formula, the semi-axes of the elastic ellipse are respectively equal to

$$a_e = \sqrt{\frac{C_k^2 - D_k B_k}{A_k B_k}}; \quad b_e = \sqrt{\frac{C_k^2 - D_k B_k}{B_k^2}},$$

the coefficients in these formulas are

$$A_k = 3G^2 \theta^2; \quad B_k = E^2 \chi^2 + A_k; \quad C_k = E^2 \varepsilon_{str} \cdot \chi;$$

$$D_k = E^2 \varepsilon_{str}^2 - \sigma_{yistr}^2; \quad \Delta = -\frac{C_k}{B_k}.$$

Boundary angles are determined by the formulas ψ_1 and ψ_2 [10, 11]

$$\psi_1 = arctg \frac{y_{upp}}{\sqrt{\frac{d^2}{4} - y_{upp}^2}};$$

$$\psi_2 = arctg \frac{y_{low}}{\sqrt{\frac{d^2}{4} - y_{low}^2}};$$

$$y_{upp} = -\frac{a_e^2 \Delta}{a_e^2 - b_e^2} \pm$$

$$\pm \sqrt{\frac{a_e^4 \Delta^2}{(a_e^2 - b_e^2)^2} - \frac{\frac{d^2}{4} b_e^2 + \Delta^2 a_e^2 - a_e^2 b_e^2}{a_e^2 - b_e^2}},$$

in this formula, "+" is the sign to calculate y_{upp} and «–» is for y_{low}

Calculation of the values of M_{tors}^{tech} according to the formula (2) for some shaped strand ropes gave the following values:

for a rope strand with a diameter of 27.5 mm:
$M_{tors}^{tech} = 32.63$ Nm;

for a rope strand with a diameter of 38.5 mm:
$M_{tors}^{tech} = 42.42$ Nm;

for a rope strand with a diameter of 43.5 mm:
$M_{tors}^{tech} = 48.63$ Nm.

The experimental verification of the values of M_{tors}^{tech} was carried out according to the method of work [10]. Experienced values of strands were:

for a rope strand with a diameter of 38.5 mm:
$M_{tors}^{tech} = 37.8$ Nm;

for a rope strand with a diameter of 43.5 mm:
$M_{tors}^{tech} = 45.63$ Nm;

The theoretical error of M_{tors}^{tech} is:

$$\Delta_{38,5} = \frac{(42.42 - 37.8) \cdot 100\%}{37.8} = 12.2\%;$$

$$\Delta_{43,5} = \frac{(48.72 - 45.5) \cdot 100\%}{45.5} = 7.1\%. \quad (15)$$

It can be seen that the accuracy of the theoretical values of M_{tors}^{tech} is quite high.

Technologically, M_{tors}^{tech} of shaped strands can be reduced in two ways:

1. Apply cross lay of wire layers. In this case, different wire layers are twisted in the opposite direction and M_{tors}^{layer} have different meanings.
2. Apply technological torsional strain in the direction opposite to the kinematic torsional strain.

With cross-lay, M_{tors}^{layer} are subtracted and the technological torque of the shaped strand is significantly reduced. Calculation of M_{tors}^{tech} by formula (2), taking into account different signs of torgues of the layers, gave the following results:

for a rope strand with a diameter of 27.5 mm:
$M_{tors}^{tech} = 9.6$ Nm; i.e. decreased 3.4 times;

for a rope strand with a diameter of 38.5 mm:
$M_{tors}^{tech} = 11.4$ Nm; i.e. decreased 3.7 times;

for a rope strand with a diameter of 43.5 mm:
$M_{tors}^{tech} = 15.54$ Nm; i.e. decreased 3.4 times.

Thus, the cross lay can significantly reduce the internal factor of the operability of shaped strand ropes. However, to achieve the result

$$M_{tors}^{tech} = 0 \quad (16)$$

is not possible for shaped ropes due to a small number of wire layers.

The method of axial twisting of wires has great

possibilities for balancing shaped strands. During winding, the wires are imparted such a torsional strain that condition (16) is satisfied. This is easily done by adjusting the stranding machine. The total torsional strain of the shaped strand wires is determined by the expression [11]:

$$\theta = \theta_k + \eta \frac{2\pi}{\int_0^{2\pi} \sqrt{\dot{R}^2(\phi) + R^2(\phi) + \left(\frac{h}{2\pi}\right)^2} d\phi}. \quad (17)$$

In (17), it is denoted:

η - twisting coefficient;

$R(\varphi)$ – wire lay radius function (radius of the shaped cylinder contour in the normal section);

h – wire lay pitch.

It is obvious that the wire, when wound on a shaped cylinder, will not experience torsion strain if

$$\eta = -\frac{1}{2\pi} \theta_k \int_0^{2\pi} \sqrt{\dot{R}^2(\phi) + R^2(\phi) + \left(\frac{h}{2\pi}\right)^2} d\phi \quad (18)$$

The values of the twisting coefficient for a rope with a diameter of 27.5 mm are presented in Table 1.

Table 1. Values $M_b^*(\eta)$, $M_t^*(\eta)$, $M_{tors}^{l*}(\eta)$, $\varepsilon_i(\eta)$ for wires of a triangular rope strand with a diameter 27.5 mm

A	B	C	D	E	F	G
1		0.00	1.152	1.587	23.95	5.01
2		-0.36	1.546	1.357	24.13	3.47
3	I	-0.76	2.059	0.595	19.20	2.15
4		-1.16	2.059	-0.598	6.48	2.15
5		-1.56	1.544	-1.358	-4.84	3.48
6		-2.00	1.055	-1.616	-10.64	5.38
7		0.00	0.252	0.377	5.61	4.96
8		-0.32	0.331	0.327	5.57	3.59
9	II	-0.72	0.466	0.167	4.70	2.20
10		-1.12	0.468	-0.129	1.55	2.02
11		-1.52	0.358	-0.306	-1.03	2.26
12		-2.00	0.237	-0.385	-2.63	5.30

Note: A – line number; B – number of the layer of wires in the strand; C – value of the twisting coefficient η; D – bending moment $M_b^*(\eta)$; E - torque $M_t^*(\eta)$; F – elastic recoil moment of the wire layer $M_{tors}^{l*}(\eta)$; G – strain intensity in the outer fiber of the wires $\varepsilon_i(\eta)$

Equation (17) can be solved with respect to the twisting coefficient η:

$$\eta = -\frac{1}{2\pi} \left[\theta_k + \frac{A_b}{A_t} \frac{M_b(\eta)}{M_t(\eta)} \right] \times$$
$$\times \int_0^{2\pi} \sqrt{\dot{R}^2(\phi) + R^2(\phi) + \left(\frac{h}{2\pi}\right)^2} d\phi. \quad (19)$$

This equation is transcendental and can be easily solved by the method of successive approximations. The initial value η can be selected from the interval $-1.3 \leq \eta \leq -1.2$, and moments $M_{t,j}$, $M_{b,j}$ can be calculated using expressions (7) and (14).

For wire layers of triangular strands, the numerical solution of equation (20) shows that condition (16) is fulfilled if the twisting coefficient is equal to

$$\eta = -1.34 \quad (20)$$

Of great practical interest is the simultaneous fulfillment of the criteria of durability and aggregate strength of shaped strands. In this case, the internal force factors, the wire layer technological torque and the strain intensity in the outer wire layers can be provided as functions of one argument η and its values can be obtained for various criteria of durability and aggregate strength of shaped strands [12-14].

The various technological modes for the manufacture of shaped strands are considered.

First option. Strands with the highest possible aggregate strength.

This requirement is satisfied when the strain intensity in the outer wire layers:

$$E_i = \frac{\sigma_T}{E} - \frac{\sigma_i - \sigma_T}{E_1}, \quad (21)$$

where $\sigma_i = \sqrt{\sigma_z^2 + 3\tau_k^2}$ – the intensity of stresses in the wire outer fibers has a minimum.

The solution of equation (21) shows that the value η corresponds to this condition.

$$\eta = -0.96, \quad M_{t,j} = 0 \quad (22)$$

The moment of elastic recoil of the strand will change by 15-20%. It is seen that the existing lay mode, when $\eta = -(0.9-0.9167)$, does not correspond to the optimal one. Therefore, by bringing the value of η layers of strand wires to $\eta = -0.96$ (which can be done quite simply by changing the gears of the stranding machine unwinding mechanism), it is possible to increase the strength and balance of three-sided strand ropes.

Second option. Strands with the greatest possible balance and strength equal to the ropes being produced [15, 16].

This requirement is satisfied when $\varepsilon_i(\eta)$ is equal to the strain of the wires of the ropes being produced, i.e. at $\eta = -0.9$.

In this case, the total torsional strain Θ must be opposite in sign of the kinematic torsional strain. The numerical solution of equation (20) shows that this can be achieved when

$$\eta = -1.08. \quad (23)$$

The moment of elastic recoil is reduced by 35-50%.

This mode of stranding strands is the most appropriate for modern rope manufacture. On the one hand, the strength of the ropes will practically not change, but on the other hand, the balance and stability of the geometry of the wires of the strands will improve to a greater extent than in the 1st option [17, 18].

The third option. Strands with the highest possible strength and balance at the same time.

This option corresponds to the minimum of simultaneous changes of M_{tors}^{layer} and ε_i. The solution to equation (20) shows that

$$\eta = -1.18. \quad (24)$$

With such twisting coefficients, the strength of the ropes decreases in relation to those produced, but M_{tors}^{layer} decreases significantly by 4-5 times.

<u>The fourth option</u>. Strands with maximum balance.

The value η (20) corresponds to this case. With such values of η, the wire will experience significant torsional strain.

At the same time, the strength of the ropes decreases [19, 20].

Therefore, this laying mode of strands should be used for cases where balance is prevalent (carrying ropes, cable ropes) [21-24].

The values of the bending moment, the torque of the wire sections, the torque in the layer of wires and the intensity of strains in the outer fiber of the wires are presented in Table 1. Graphs of these values are presented in Fig. 3.
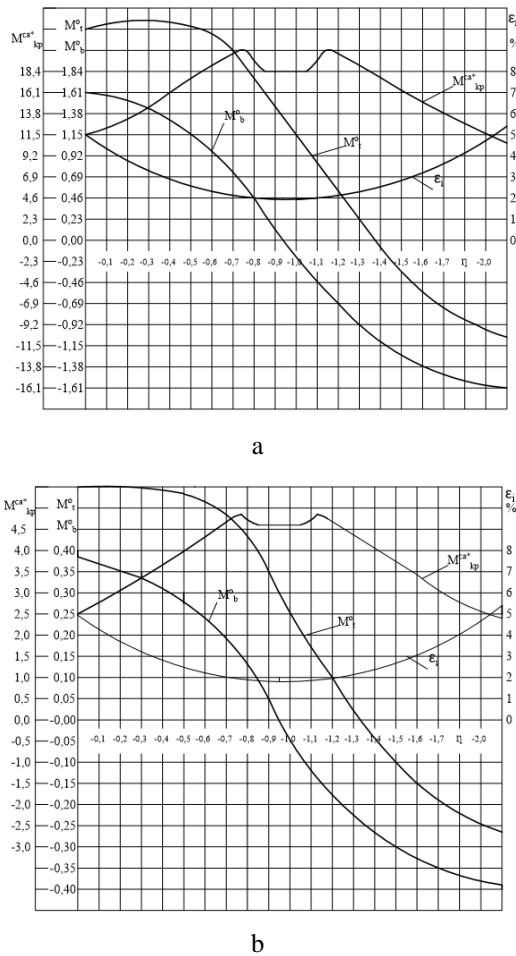
a

b

Fig. 3. Change in values $M_b^*(\eta)$, $M_t^*(\eta)$, $M_{tors}^{l*}(\eta)$, $\varepsilon_i(\eta)$ for wires of a rope strand with a diameter of 27.5 mm: a – the first layer of wires; b – the second layer of wires

4.2. Diagnostics of the geometry and elastoplastic state of wires wound on a shaped cylinder

For a wire wound on a round cylinder, the geometry formulas are known, expressed in terms of the cylinder radius and lay pitch h:

The wire length on the 1st lay pitch is

$$s = \sqrt{4\pi^2 R^2 + h^2}, \qquad 25)$$

the lay angle is

$$\alpha = arccos \frac{h}{\sqrt{4\pi^2 R^2 + h^2}} \qquad (26)$$

The kinematic curvature and torsion are

$$\chi = \frac{4\pi^2 R^2}{4\pi^2 R^2 + h^2}; \quad \theta_k = \frac{2\pi h}{4\pi^2 R^2 + h^2}. \qquad (27)$$

The similar approach in diagnosing the geometry of wires on a shaped cylinder is used. The parametric equations of the wire screw axis will have the form

$$x = R(\phi) \cos \phi; \; y = R(\phi) \cdot \sin \phi; \; z = \frac{h}{2\pi}\phi. \qquad (28)$$

where:

$R(\phi)$ – shaped cylinder contour function (wire lay radius);

h – wire lay pitch;

ϕ – screw axis parameter.

The geometrical parameters of the laying of wires in round and shaped strands are shown in Fig. 4.
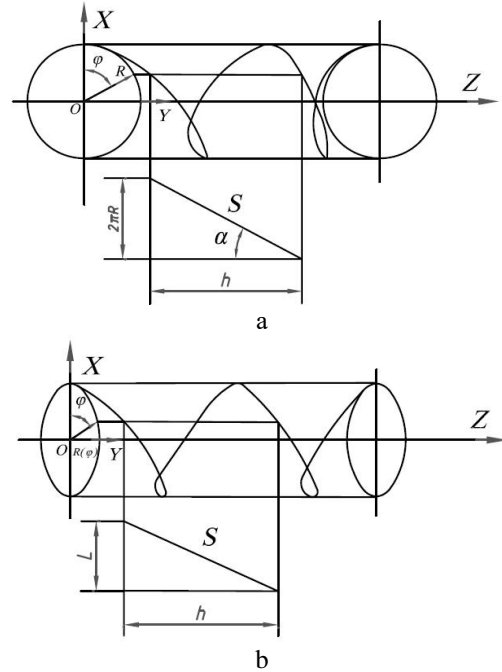
a

b

Fig. 4. Geometrical parameters of of the lay of wires: a – round strand; b – shaped strand

The change in the geometry of the lay of wires in the rope strand with a diameter of 27.5 mm is shown in Fig. 5.

The main difference between formulas (26) - (28) from (29) is the dependence of the lay radius on angle ϕ. As a result, the exact determination of the geometry of the wires on the shaped cylinder becomes much more complicated. If (29) is differentiated on ϕ one, two and three times, then general formulas are obtained for:

the length of the wires on the shaped cylinder

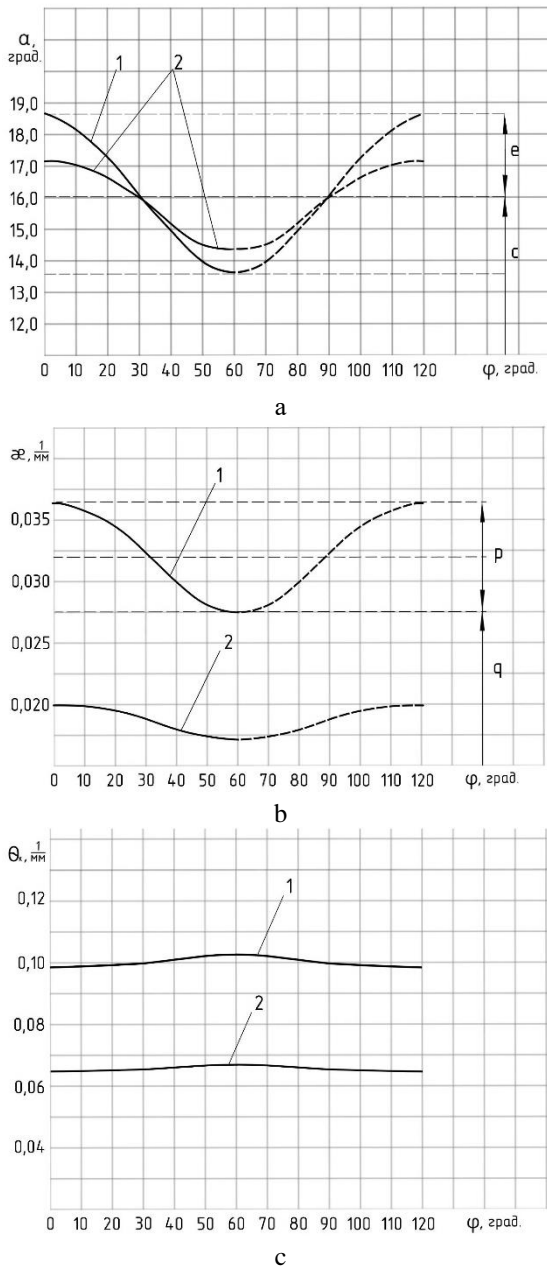$$S = \int_0^{2\pi} \sqrt{R_{(\phi)}^2 + R(\phi) + \left(\frac{h}{2\pi}\right)^2} \, d\phi; \qquad (29)$$

Fig. 5. Changing the geometric parameters of the lay of wires in the strand of a rope with a diameter of 27.5 mm: 1 - for wires of the first layer; 2 - for wires of the second layer; a - change in lay angle; b - change in curvature; c - twist change

the lay angle when $h = const$_is

$$\alpha = arccos \frac{h}{\sqrt{4\pi^2 R^2(\phi) + 4\pi^2 R^2(\phi) + h^2}}; \quad (30)$$

if $h = h(\phi)$, then

$$\alpha = arccos \frac{h(\phi) + \dot{h}(\phi)\cdot\phi}{\sqrt{4\pi^2 \dot{R}^2(\phi) + 4\pi^2 R^2(\phi) + h^2(\phi) + 2h(\phi)\cdot h(\phi)\phi + \dot{h}^2_{(\phi)}\phi}}; \quad (31)$$

the curvature is

$$\chi = \frac{1}{\sqrt{\frac{\frac{h^2}{4\pi^2}\left[\left(\ddot{R}(\phi) - R(\phi)\right)^2 + 4\dot{R}^2_{(\phi)}\right] + \left(2\dot{R}^2_{(\phi)} - \ddot{R}(\phi)\cdot R(\phi) + R^2_{(\phi)}\right)^2}{\left(2\dot{R}^2_{(\phi)} + R^2_{(\phi)} + \frac{h^2}{4\pi^2}\right)^3}}}; \quad (32)$$

the kinematic torsion is

$$\theta_k = \frac{\frac{h}{2\pi}\left(3\ddot{R}^2_{(\phi)} - 4\ddot{R}(\phi)\cdot R(\phi) - 2\ddot{R}(\phi)\dot{R}(\phi) + 6\dot{R}^2_{(\phi)} + R^2_{(\phi)}\right)}{\frac{h^2}{4\pi^2}\left[\left(\ddot{R}(\phi) - R(\phi)\right)^2 + 4\dot{R}^2_{(\phi)}\right] + \left(2\dot{R}^2_\phi - \ddot{R}(\phi)R(\phi) + R^2_{(\phi)}\right)^2}; \quad (33)$$

If the lay radius of wires in a triangular strand is represented by the formula

$$R(\phi) = a + b\,cos\,3\,\phi, \quad (34)$$

where

$a = \dfrac{R_{max} + R_{min}}{2}$ – average lay radius;

$b = \dfrac{R_{max} - R_{min}}{2}$ – amplitude lay radius;

the following dependences of geometric parameters are obtained:
the lay angle is

$$\alpha(\phi) = c + e\,cos\,3\,\phi, \quad (35)$$

where

$c = \dfrac{\alpha_N + \alpha_L}{2}$ –average lay angle;

$e = \dfrac{\alpha_N - \alpha_L}{2}$ –amplitude lay angle;

α_N, α_L – lay angles at the points of the cylinder shaped contour.

The curvature of the wires is well described by the dependencies

$$\chi(\phi) = q + P\,cos\,3\,\phi \quad (36)$$

where

$q = \dfrac{\chi_N + \chi_L}{2}$ – mean curvature;

$p = \dfrac{\chi_N - \chi_L}{2}$ – amplitude curvature;

The kinematic torsion of the wires is almost constant (does not depend on the parameter ϕ) and is determined by the expression

$$\theta_k = \frac{cos\,c\cdot sin\,c}{a} \quad (37)$$

As the studies have shown, expressions (35), (36) and (37) most accurately describe the actual geometry of the wires on the shaped cylinder. In this work, these formulas are used to describe the elastoplastic state of the wire material, which subsequently made it possible to develop the optimal parameters for technological balancing of steel shaped ropes.

5. CONCLUSIONS AND RECOMMENDATIONS FOR DIAGNOSTICS OF TECHNOLOGICAL BALANCING WHEN MANUFACTURING SHAPED STRAND ROPES

As a result of the research, the following conclusions can be drawn.
1. Expressions for internal bending and torque moments are constructed, taking into account the variable geometric parameters of winding shaped strand wires.

2. Based on the formula (2), the most accurate expression for the technological torque (elastic recoil moment) of shaped strands is given.

3. Recommendations for improving the properties of shaped-strand ropes when manufacturing are formulated (see options 1-4).

4. Additionally, it can be noted that condition (16) can be achieved only due to a significant value of the axial twisting of the wires. In this case, the elastic plastic stresses in the wires reach dangerous values and, therefore, this method has significant limitations. Thus, in the manufacture of shaped-strand ropes, it is necessary to use the values of the twisting coefficients of options 1 and 2, and to significantly reduce the moment of elastic recoil and neutralize the lay stresses, subject the ropes and strands to additional processing.

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