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            | \(A_{1b}, A_{1h}, B_{x}, A_{k}\) |          |
| semiaxes of the elastic zone of the ellipse   | \(a_c, b_e\) | m         |
| amplitude value of lay radius                 | \(b\)      | m         |
| average lay angle                             | \(c\)      | degree    |
| amplitude value of lay angle                  | \(e\)      | degree    |
| number of layers of core wires                | \(i\)      |           |
| technological torque of the triangular strand | \(M_{\text{tors tech}}\) | 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 | \(M_{\text{tors layer}}\) | 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 | \(y_{\text{upp}, y_{\text{low}}}\) | m |
| wire lay angle on a round cylinder            | \(\alpha\) | degree    |
| wire lay angle                                | \(\alpha_j\) | degree    |
| lay angle of wires on the shaped cylinder     | \(\alpha_i\) | degree    |
| angle between curvature radius and lay radius | \(\gamma_{ij}\) | degree |
| twisting coefficient                          | \(\eta\)   |           |
| total torsion deformation of the wires       | \(\theta\) | m\(^{-1}\) |
| kinematic torsion of wires                   | \(\theta_{iz}\) | m\(^{-1}\) |
| kinematic torsion of wires on a shaped cylinder | \(\theta_{k}\) | m\(^{-1}\) |
| radii of the upper and lower parts of the contour line of the ellipse of the elastic zone | \(\rho_{\text{upp}}, \rho_{\text{low}}\) | m |
| conditional yield strength of the wire material | \(\sigma_{0.2}\) | MPa |
| curvature of the wires of the shaped strand  | \(\chi\)   | m\(^{-1}\) |
| wire curvature                                | \(\chi_{ij}\) | m\(^{-1}\) |
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](image)

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](image)

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 \( \Sigma N \), and transverse forces \( Q_n, Q_o \), 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 \( M_{tors} \) or reduce it in all possible ways. In [4], the value of \( M_{tech}^{tors} \) is presented in the form of an approximate expression

\[
M_{tech}^{tors} = \sum_{i=1}^{m} \sum_{j=1}^{n} \left[ M_{b,ji} \left( x_{ji} \sin \alpha_j \right) + \theta_{ji} R_{ji} \cos \gamma_{ji} \cos \sigma_{ji} \right] + M_{l,ji} \cos \sigma_j \left( 1 - \alpha_{ji} R_{ji} \cos \sigma_j \right) \]

(1)
where:

- $M_b$ – bending moment;
- $M_t$ – torque;
- $\alpha_i$ – wire lay angle;
- $\theta_i$ – wire kinematic torsion;
- $R_b$ – wire lay radius;
- $\gamma_i$ – angle between the curvature radius and the lay radius;
- $\chi_i$ – 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 $M_{\text{tech}}$ calculated by (1) will have significant errors [5-6].

One of the effective methods of reduction of $M_{\text{tech}}$ is the use of technological torsional strain. By adjusting its value, which is quite simple to implement, the value of $M_{\text{tech}}$ 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 $M_{\text{tech}}$ 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 $M_{\text{tech}}$ 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:

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

where:

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

$$M_{\text{layer}} = \sum_{i=1}^{n_l} \left[ (1 + \cos^2 \alpha_i) \sin \alpha_i M_{b,i} + \cos^3 \alpha_i M_{t,i} \right] \quad (3)$$

where:

- $n_l$ – number of wires of the core;
- $\alpha_i$ – 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;
- $M_{\text{layer}}$ – technological torque of the layer of shaped strand wires.

The value of $M_{\text{layer}}$ is determined by the expression [8]:

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

where:

$k, A_b, A_t, j$ – coefficients determined for triangular strands by the formulas:

- $k=3$ for the triangular strand
- $A_{b,i} = \sin \alpha (0^\circ) + 2 \sin \alpha (30^\circ) + \sin \alpha (60^\circ)$
- $\theta_iR(0^\circ) \cos \alpha (0^\circ) \cdot \cos \gamma (0^\circ) + \theta_iR(30^\circ) \cos \alpha (30^\circ) \cos \gamma (30^\circ) + \theta_iR(60^\circ) \cos \alpha (60^\circ) \cos \gamma (60^\circ)$

where:

- $R(\phi), \alpha(\phi)$ – functions of the lay radius and layer wire lay angle;
- $\chi(\phi), \theta_i$ – curvature and kinematic torsion of layer wires;
- $\gamma(\phi)$ – 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]:

$$M_b = \pi E_a e_i b_i \left[ (q + p \cos 3 \phi) \frac{b_i^2}{4} \right] + 2\sigma_f \left(1 - \frac{E_v}{E}\right) E \times \left( q + p \cos 3 \phi \right) \cdot (A_{b,i} + A_{t,i}) + 2E_v(q + p \cos 3 \chi(x_0 - x_e)) \quad (7)$$

where:

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

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

$$b_i = \frac{\sigma_f^2 [q^2 + (q + p \cos 3 \phi)^2 + 3q^2 a_i^2]}{[q^2 + (q + p \cos 3 \phi)^2 + 3q^2 a_i^2]^2} \quad (8)$$

$$a_i = \frac{\sigma_f^2 [q^2 (q + p \cos 3 \phi)^2 + 3q^2 a_i^2]}{3q^2 a_i^2 [q^2 + (q + p \cos 3 \phi)^2 + 3q^2 a_i^2]^2} \quad (9)$$

where:

- $\chi = q + p \cos 3 \phi$ – variable curvature of the wires on...
the shaped cylinder;

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

$\sigma_0$ – conditional yield strength of wire material.

Coefficients:

$$A_{iB} = \int_0^{\frac{\pi}{2}} \int_{\rho_{\text{up}}}^{\rho_{\text{low}}} \rho^3 \sin^2 \psi \left[ B_k(\rho \sin \psi) \right] \frac{1}{2} \rho d\rho d\psi, \quad (10)$$

$$A_{iH} = \int_0^{\frac{\pi}{2}} \int_{\rho_{\text{up}}}^{\rho_{\text{low}}} \rho^3 \sin^2 \psi \left[ B_k(\rho \sin \psi) \right]^2 + A_k \rho^2 \cos^2 \psi \left[ B_k(\rho \sin \psi) \right]^2 + A_k \rho^2 \cos^2 \psi \left[ B_k(\rho \sin \psi) \right]^2 + A_k \rho^2 \cos^2 \psi \left[ B_k(\rho \sin \psi) \right]^2 + \psi_2 = \arctg \frac{\psi_{\text{up}}}{\sqrt{\frac{d^2}{2} + \frac{d^2}{2} - \frac{a_x^2}{2} + \frac{a_y^2}{2}}}$$

$$\pm \frac{A_k \Delta^2}{a_x^2 - a_y^2} \pm \frac{A_k \Delta^2}{a_x^2 - a_y^2}$$

in this formula, "+" is the sign to calculate $\psi_{\text{up}}$ and "+" for $\psi_{\text{low}}$.

Calculation of the values of $M_{\text{tech}}$ 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: $M_{\text{tech}} = 32.63 \text{Nm}$;

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

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

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

1. Experienced values of strands were:

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

   for a rope strand with a diameter of 43.5 mm: $M_{\text{tech}} = 45.63 \text{Nm}$.

   The theoretical error of $M_{\text{tech}}$ is:

   $$\Delta_{38.5} = \frac{(42.42 - 37.8) \cdot 100\%}{37.8} = 12.2\%;$$

   $$\Delta_{43.5} = \frac{(48.72 - 45.5) \cdot 100\%}{45.5} = 7.1\%.$$

   It can be seen that the accuracy of the theoretical values of $M_{\text{tech}}$ is quite high.

Technologically, $M_{\text{tech}}$ 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 $M_{\text{tor}}$ have different meanings.

2. Apply technological torsional strain in the direction opposite to the kinematic torsional strain.

With cross-lay, $M_{\text{tor}}$ are subtracted and the technological torque of the shaped strand is significantly reduced. Calculation of $M_{\text{tor}}$ by formula (2), taking into account different signs of torques of the layers, gave the following results:

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

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

for a rope strand with a diameter of 43.5 mm: $M_{\text{tor}} = 15.54 \text{Nm}$; 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 $M_{\text{tech}} = 0$ (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]:

$$\theta = \theta_k + \eta \int_0^{2\pi} \frac{1}{\sqrt{R^2(\phi) + R^2(\phi) + \left(\frac{h}{2\pi}\right)^2}} \, d\phi$$

In (17), it is denoted:

- \( \eta \) – twisting coefficient;
- \( R(\phi) \) – 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

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

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

| 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 | -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 | -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 \( \eta \); D – bending moment \( M_b(\eta) \); E – torque \( M_t(\eta) \); F – elastic recoil moment of the wire layer \( M_{re}(\eta) \); G – strain intensity in the outer fiber of the wires \( \varepsilon(\eta) \).

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

$$\eta = \frac{1}{2\pi} \left[ \theta_k + \frac{A_b}{A_t} \frac{M_b(0)}{M_t(\eta)} \right] \times \int_0^{2\pi} \sqrt{R^2(\phi) + R^2(\phi) + \left(\frac{h}{2\pi}\right)^2} \, d\phi$$

This equation is transcendental and can be easily solved by the method of successive approximations. The initial value \( \eta \) can be selected from the interval \(-1.3 \leq \eta \leq -1.2\), and moments \( M_{i,j}, M_{b,j} \) 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 \( \eta = -1.34 \). (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 \( \eta \) 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:

$$\frac{E_t}{E} = \frac{\sigma_t - \sigma_l}{\sigma_l}$$

where \( \sigma_t = \sqrt{\sigma^2 + 3\sigma_k^2} \) – the intensity of stresses in the wire outer fibers has a minimum.

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

$$\eta = 0.96, \ M_{i,j} = 0$$

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 \( \eta \) 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 if \( \varepsilon(\eta) \) 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 \( \Theta \) must be opposite in sign of the kinematic torsional strain. The numerical solution of equation (20) shows that this can be achieved when

$$\eta = 1.08.$$ (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 \( M_{tor\ layer} \) and \( \varepsilon_l \).

The solution to equation (20) shows that

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

With such twisting coefficients, the strength of the ropes decreases in relation to those produced, but \( M_{tor\ layer} \) decreases significantly by 4-5 times.
The fourth option. Strands with maximum balance.

The value \( \eta \) (20) corresponds to this case. With such values of \( \eta \), 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) \[21\text{-}24\].

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.

\[\text{Fig. 3. Change in values } M_{b}^{*}(\eta), M_{t}^{*}(\eta), M_{t}^{*}\text{tor}(\eta), \varepsilon(\eta) \text{ 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

\[ s = \sqrt{4\pi^2 R^2 + h^2}, \quad 25 \]

the lay angle is

\[ \alpha = \arccos \frac{h}{\sqrt{4\pi^2 R^2 + h^2}} \quad (26) \]

The kinematic curvature and torsion are

\[ \chi = \frac{4\pi^2 \eta^2}{4\pi^2 R^2 + h^2}; \quad \theta_\phi = \frac{2\pi h}{4\pi^2 R^2 + h^2} \quad (27) \]

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

\[ x = R(\phi) \cos \phi; \quad \cos \phi = R(\phi) \sin \phi; \quad z = \frac{h}{2\pi} \phi. \quad (28) \]

where:

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

\( h \) – wire lay pitch;

\( \phi \) – screw axis parameter.

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

\[\text{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 \( \phi \). 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 \( \phi \) one, two and three times, then general formulas are obtained for:

the length of the wires on the shaped cylinder

\[ S = \int_0^{2\pi} \sqrt{R^2(\phi) + R(\phi) + \left(\frac{h}{2\pi}\right)^2} \, d\phi; \quad (29) \]
the curvature is determined by the expression
\[
\alpha(\phi) = c + e \cos 3\phi,
\]
where
\[
a = R_{\text{max}} + R_{\text{min}} - \text{average lay radius};
\]
\[
b = R_{\text{max}} - R_{\text{min}} - \text{amplitude lay radius};
\]
the following dependences of geometric parameters are obtained:

the lay angle is
\[
\theta_k = \frac{1}{2}\left(\frac{1}{2} R_{\text{max}}(\phi) - R_{\text{min}}(\phi)\right) - \frac{b}{2} \left(R_{\text{max}}(\phi) + R_{\text{min}}(\phi)\right)\frac{\partial \alpha}{\partial \phi};
\]
If the lay radius of wires in a triangular strand is represented by the formula
\[
R(\phi) = a + b \cos 3\phi,
\]
where
\[
a = R_{\text{max}} + R_{\text{min}} - \text{average lay radius};
\]
\[
b = R_{\text{max}} - R_{\text{min}} - \text{amplitude lay radius};
\]
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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REFERENCES

1. Kholifin MN. Study of the work of trihexagonal strand ropes and methods for improving their resistance. In: Abstract of the cand. thesis. Sciences: Novocherkask, NPI; 1972.

2. Kholifin MN. Moment of elastic recoil of a shaped strand. Mine lifting and transport installations. In: Proceedings of the Novocherkask Polytechnic University. Novocherkask: NPI; 1973.

3. Breslavtsheva IV. Stress-strain state of the rope wires during laying and a method for calculating the parameters of preformers. In: Abstract of the cand. thesis. Sciences: Novocherkask, NPI; 2007.

4. Shevtsov VV. Method for assessing the quality of crane lifting ropes. In: Abstract of the cand. thesis. Sciences: Novocherkask, NPI; 2006.

5. Malinovsky VA. Steel ropes. In: Some issues of calculation and design technology. Odessa: Astroprint; 2001.

6. Ivanov BF. Research and improvement of lifting tunneling ropes of a closed structure. In: Abstract of the cand. thesis. Sciences: Novocherkask, NPI; 1984.

7. Kozlov VT, Oroby EF. Influence of elastic recoil of a shaped strand on the change in the pitch of a twist spiral. Theory and practice of hardware production 1979; 8: 110–115.

8. Kozlov VT. On the issue of elastic recoil of steel ropes after laying. Steel ropes 1964; 1:144–151.

9. Nemchuk OO, Krehkovska HV. Fractographic substantiation of the loss of resistance to brittle fracture of steel after operation in the marine gantry crane elements. Metallofizika i Novishche Tehnologii 2019;41(6):825–836.

10. Kolomiets L, Oroby V, Lymarenko A. Method of boundary element in problems of stability of plane bending beams of rectangular cross section structures. Metallurgical and mining industry 2016; 3: 59–65.

11. Chayun IM, Nepomnyaschikh AV. Tension optimization of the conductor-and-support cable elements during stranding process. Works of OPU 2016;3(50).

12. Feyrer K. Wire Ropes – Tension, Endurance, Reliability. 2nd ed. Heidelberg: Springer; 2015. https://doi.org/10.1007/978-3-540-33831-4.

13. Malinovsky VA. Steel ropes. Analytical reference book. Odessa: Astroprint Publ.; 2016. http://www.vira.od.ua/researches.html

14. Noskov S, Kharitonov V, Ivantsov A. Development of adjustment modes for straightening and preforming devices. Processing of solid and layered materials. 2012;38.3-12. https://cyberleninka.ru/article/n/razrabotka-rezhimov-nastroyki-rihuyushchih-i-prezurnuyushchih-nastroyv

15. Guide for Certification of Lifting Appliances. TX: City Plaza Drive Spring. 2021. https://ww2.eagle.com/content/dam/eagle/rules-and-guides/current/equipment_and_component_certificati on/152_lifting_appliances_2021/lifting_appliances_g uide_dec21.pdf

16. Musikhin VA. Determination of a Real Strain-stress State of the Steel-wire Rope Elements. Procedia Engineering 2016; 150: 1848–1852.

17. ISO 3108:2017. Steel wire ropes – test method. 2017. https://www.iso.org/standard/66725.html

18. Ivankevich A, Pierska V, Shakhov A, Shakhov V, Yanovenko V. A Proactive Strategy of Ship Maintenance Operations. 2019 IEEE 14th International Conference on Computer Sciences and Information Technologies (CSIT) 2019: 126-129. http://dx.doi.org/10.1109/STC-CSIT.2019.8929741.

19. Nemchuk OO. Specific features of the diagnostics of technical state of steels of the port reloading equipment. Materials Science 2018; 53(6): 875-878. http://dx.doi.org/10.1007/s11223-018-0148-5.

20. Chernov S, Titov S, Chernova L, Kunanets N, Pierska V, Chernova L, Shcherbyna V, Petryshyn L. Efficient algorithms of linear optimization problems solution. CEUR workshop proceedings 2021; 2851: 116-131.

21. Nemchuk OO, Nesterov OA. In-service brittle fracture resistance degradation of steel in a ship-to-shore portal crane. Strength of Materials 2020; 52(2): 275–280. https://dx.doi.org/10.1007/s11223-020-00175-w.

22. Chernov S, Titov S, Chernova L, Pierska V, Chernova L, Kunanets N. Three-index optimization transportation model. 2021 IEEE 16th international conference on computer sciences and information technologies. (CSIT) 2021:315-318. http://dx.doi.org/10.1007/978-3-030-83831-9_25.

23. Nemchuk O, Hredil M, Pustovoy V, Nesterov O. Role of in-service conditions in operational degradation of mechanical properties of portal cranes steel. Procedia Structural Integrity. 2019;16:245-251. http://dx.doi.org/10.1016/j.prostr.2019.07.048.

24. Nemchuk OO. Influence of the working loads on the corrosion resistance of steel of a marine harbor crane. Materials Science. 2019;54(5):743-747. https://doi.org/10.1007/S11003-019-00241-Y.
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