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CROSS WAY OF STEEL ROPE FASTENING TO A SINGLE-DRUM MINE HOISTING PLANT WITH THE LOCATION OF PULLEYS ON THE SAME AXIS

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ПЕРЕХРЕСНИЙ СПОСІБ КРІПЛЕННЯ КАНАТІВ НА ОДНОБАРАБАННИХ ШАХТНИХ ПІДІЙМАЛЬНИХ УСТАНОВКАХ З РОЗТАШУВАННЯМ ШКИВІВ НА ОДНІЙ ОСІ

Purpose. To study the impact of the methods of steel rope fastening to a single-drum mine hoisting plant with the pulley location on the same axis on the minimum distance between the rope strings in static and dynamic regimes.

Methodology. A mathematical model for determining the distance between any points of the steel rope strings in statics has been developed. To study the transverse-longitudinal oscillations of a steel rope string the finite element model of a hoisting plant has been made.

Findings. It was shown that for the cross-method of rope fastening in case when internal deviation angle is smaller than external one the friction between the winding rope and the adjacent rope wrap is excluded. The impact of rope oscillations on the minimum distance between them for the cross-method of fastening ropes was assessed.

Originality. Larger distance between pulleys or the number of empty flutes between the winding up and uncoiling branches on a drum increases the distance between the strings of cross-disposed ropes. A criterion of contact absence between the rope strings for cross fastening was introduced.

Practical value. The cross way of steel rope fastening to a single-drum mine hoisting plant with location of pulleys on the same axis reduces the pitch of winding rope, thereby the rope capacity of a hosting plant and the lift depth are increased

Keywords: *mine hoisting plant, steel rope, transverse-longitudinal oscillations of the rope string, deviation angle*

Actuality of work. On a plenty of mines further mining is going by the way of deeper horizon development. Increasing the lifting depth demands increasing the rope capacity of a hoisting plant. If a hoisting plant works at its depth ability limit, increasing the rope capacity requires to change the design of machine, that is associated with high capital costs. Therefore, in some cases, more effective technical solution will be increasing the rope capacity of already existing hoist drum by means of rational use of winding surface [1].

In this paper, a method of increasing the rope capacity of a single drum hoisting plant without changing the drum design is being considered.

There are following increasing ways of the rope capacity of a single drum hoisting plants: to reduce the number of friction coils; transition of ropes through a cut on machines with a split drum; the use of multilayer rope winding (however, there are use restrictions on this solution for cargo and human lifting [2]); the use of drum with conical insert on area with a maximum value of the internal deviation angle.

New technical solution is proposed in the patent [3]: to increase the rope capacity of a single-drum hoisting plant by the cross fastening ropes.

Statement of the problem. The drum rope capacity is determined by its diameter *Db*, width *B* and screw fluting step *t*. Parameters *Db* and *B* determine the winding surface size, which is taken as the main constructing criterion for the standard series of hoisting plants. The screw fluting pitch *t* determines the rope capacity of this drum surface. The thread size depends on the maximum value of an internal deviation angle of the hoisting plant. If the internal deviation angle increases, winding rope can rub against the adjacent rope turn [1]. To avoid such situation winding rope is usually pushed back from the adjacent rope turn by increasing the pitch on the drum surface, which reduces the drum rope capacity.

The layout of a single-drum hoisting plant with the pulley location on the same geometrical axis, that uses cross fastening ropes [3] (see fig.1, *a*), allows to change the ratio between internal and external deviation angles, that is to reduce the value of dangerous internal angle via increasing the value of external angle. This prevents friction of winding rope on the adjacent rope turn, which makes it possible to reduce the winding pitch and to increase the drum rope capacity. However, the proposed scheme has the dangerous factor, which consists in the possibility of the rope string contacts in ascending and descending branches. This task needs to be explored.

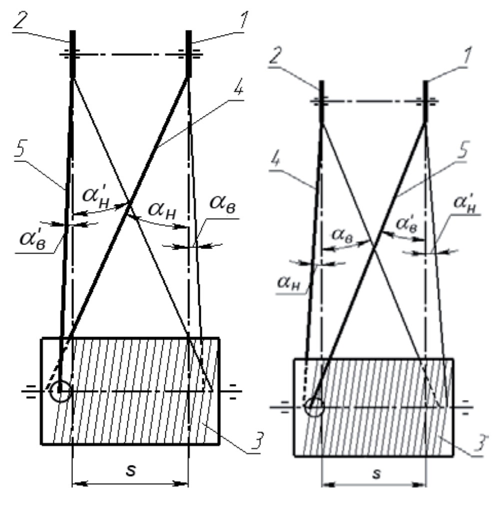
The purpose of work is to study the method impact of the rope-fastening to a single-drum hoisting plant with the pulley locations on the same axis on the minimum distance between steel-rope strings of ascending and descending branches during static and dynamic regimes.

The main part. Fig.1, *a* shows the scheme of cross-fastening of ropes on a single-drum mine hoisting plant. The hoist consists of right 1 and left 2 pulleys with distance *s* between them, single-drum hoisting plant 3 and coiling 4 and uncoiling 5 ropes. Unlike the standard fastening (fig.1, *b*), where rope 4 rounds over the hoisting pulley 2 and rope 5 – over the hoisting pulley 1, for the cross-fastening scheme rope 4 rounds over the hoisting pulley 1 and rope 5 – over the hoisting pulley 2. The thick lines 4 and 5 correspond to the extreme string rope position with fully coiled rope 5 and uncoiled rope 4, while the thin lines 4 and 5 correspond to the opposite extreme string rope position. There are notations on fig. 1: $\alpha_{\text{н}}$ – an external deviation angle of the rope 4; $\alpha_{\text{в}}$ – an internal deviation angle of the rope 4; $\alpha`_{\text{н}}$ – an external deviation angle of the rope 5; $\alpha`_{\text{в}}$ – an internal deviation angle of the rope 5.

A mine single-drum hoisting plant operates as follows. By rotating the drum 3 of a hoisting plant, the rope 4 is winding up on the drum surface, while the rope 5 is uncoiling from a drum. The gap between the ropes 4 and 5 has a few empty turns. Starting from the rope attachment point 4 towards the pulley axis 1 the rope 4 is winding with external deviation angle $\alpha_{\text{н}}$ and at this site during the operation there is no mutual rubbing of the rope 4 on nearby rope, laying in the flute. Right to the pulley axis 1 the rope is winding with internal deviation angle $\alpha_{\text{в}}$. When the drum is rotating at the opposite direction, the rope 4 is uncoiling, while the rope 5 is winding on the hoist drum. Starting from the rope attachment point 5 (the thin line on fig.1, *a*) towards the pulley axis 2 the rope 5 is winding with external deviation angle $\alpha`_{\text{н}}$ without mutual rope friction. Left to the pulley axis 2 the rope 5 is winding with internal deviation angle $\alpha`_{\text{в}}$. In this case $\alpha_{\text{н}} > \alpha_{\text{в}}$ and $\alpha`_{\text{н}} > \alpha`_{\text{в}}$, that had to be achieved to eliminate friction of winding rope and rope on the adjacent turn.

Therefore, the changing rope attachment position provides increasing values of external angles $\alpha_{\text{н}}$ and $\alpha`_{\text{н}}$ via decreasing values of internal angles $\alpha_{\text{в}}$ and $\alpha`_{\text{в}}$ and makes better conditions for winding with maximum rope sting internal deviation angles $\alpha_{\text{в}}$ and $\alpha`_{\text{в}}$.

The possible contact zone of rope with an rope adjacent coil is allocated with a circle on the fig.1. For the standard scheme (fig.1, *b*) contact is possible with a large internal deviation angle $\alpha_{\text{в}}$. Another rope branch is deflected only by the groove scallop with an external deviation angle $\alpha_{\text{н}}$. For the cross fastening (fig.1, *a*) the possible rope contact zone responds to smaller value of internal angle, so there is no need to increase the screw flute pinch in order to avoid winding rope rubbing on the adjacent rope coil. In such case, due to improved exploitation conditions, the value of maximum allowable deviation angle can be increased. For example for bicylinder-conical machines it is allowed to increase the deviation angle up to 2º on the small cylinder. However, for single-drum cylindrical mine hoists the deviation angle is limited up to 1°30' by Safety Regulations [2].



a *b*

Fig. 1. The schemes of fastening ropes: a – standard; b – cross way; 1,2 – pulleys; 3 – drum; 4,5 – strings of ropes; $\alpha_{\text{н}}$ $\alpha`_{\text{н}}$ *– external deviation angles*; $\alpha_{\text{в}}$ $\alpha`_{\text{в}}$ *– internal deviation angles*

Next we'll be analyzing the change of distance between the rope strings for different attachment methods. All numerical calculations were performed for the cargo hoisting plant of the mine "Tchentralnaya" of Open Joint Stock Company "Sukha Balka" (Kriviy Rig), equipped with monorope lifting machine type TsR5x4.66 /0.5. Nowadays the hoist has been working as a single-drum machine with the parameters: the drum radius Rb = 2.54 m; the rope string length $L_{стр}$ = 106.6 m (with support rollers); drum width B=4.44 m; distance between pullyes s = 1.98 m; rope diameter dk = 39 mm; the number of empty flutes k = 6. The fettle design is following: from the flange of a drum jammed part there are 8 flutes with the pitch t = 44 mm, then 12 flutes with t = 48 mm, 52 flutes with t = 46 mm, 13 flutes with t = 48 mm, 7 flutes with t = 44 mm.

The deviation angles for this machine are: for rope standard attachment $\alpha_{\text{н}}$ = 0°26' and $\alpha_{\text{в}}$ = 1°21'; for cross attachment – $\alpha_{\text{н}}$ = 1°30' and $\alpha_{\text{в}}$ = 0°17'. As we see the internal angle is smaller, this will considerably improve working conditions of ropes on the drum. The deviation angles does not exceed allowable by Safety Regulations [2] value of 1°30 '.

For defining the minimum distance between left and right hoist branches the coordinate system XYZ (fig. 2) is introduced, the origin of which coincides with the rope contact point with the right pulley, axis X is directed perpendicular to the pulley axis, axis Z – parallel towards the pulley axis, axis Y – perpendicular to the plane XOZ, creating the right coordinate system.

Next we'll use such definitions: subscript $i=1$ indicates the rope standard attachment, and i = 2 – cross way. For numerical calculations we'll take t = 46 mm.

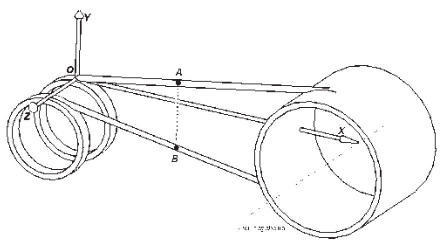


Fig. 2. The coordinate system

The distance between two arbitrary points A and B, which positioned on different rope branches and have the same coordinate x, is

$$S_i(x) = \sqrt{Sy_i(x)^2 + Sz_i(x)^2}\ , \tag{1}$$

where $Sy_i(x)$, $Sz_i(x)$ – projections S_i on the y-axis and z- axis respectively. Considering the triangle OAB on the plan XOY we can write

$$Sy_i(x) = \frac{2x \cdot Rb}{L_{str}} \quad \text{for} \quad x \in [0, L_{str}]\ . \tag{2}$$

Because of the smallness, we will neglect the differences in the x coordinates for the meeting points ropes with pulleys and ropes with drum. We will not take into account the impact of string oscillatory motion and string sagging. Examining the triangle OAB on the plane XOZ we can write

$$\left.\begin{aligned} Sz_1(x) &= s + \frac{x \cdot (k \cdot t - s)}{L_{str}} \\ Sz_2(x) &= s - \frac{x \cdot (s + k \cdot t)}{L_{str}} \end{aligned}\right\} for\ x \in [0, L_{str}]\ . \tag{3}$$

After making substitutes from formulas (2) and (3) to formula (1), collecting and simplifying the resulting expression we can define the minimum value of the distance between branches as

$$\begin{aligned} S\min_1 &= \frac{2s \cdot Rb}{\sqrt{(s - k \cdot t)^2 + 4Rb^2}}; \\ S\min_2 &= \frac{2s \cdot Rb}{\sqrt{(s + k \cdot t)^2 + 4Rb^2}}, \end{aligned} \tag{4}$$

which is reached when $x = x\min_i$

$$\begin{aligned} x\min_1 &= \frac{L_{str} \cdot s \cdot (s - k \cdot t)}{\sqrt{(k \cdot t - s)^2 + 4Rb^2}}; \\ x\min_2 &= \frac{L_{str} \cdot s \cdot (s + k \cdot t)}{\sqrt{(k \cdot t + s)^2 + 4Rb^2}}. \end{aligned} \tag{5}$$

We can make a conclusion, that the minimum distance between strings does not depend on the string length, but is determined by the ratio between diameter, distance between pulleys and the number of empty flutes between branches.

In the string crossing point x_z the distance along the Z axis between the branches is equal to zero

$$Sz_2(x_z) = s - \frac{x \cdot (s + k \cdot t)}{L_{str}} = 0,$$

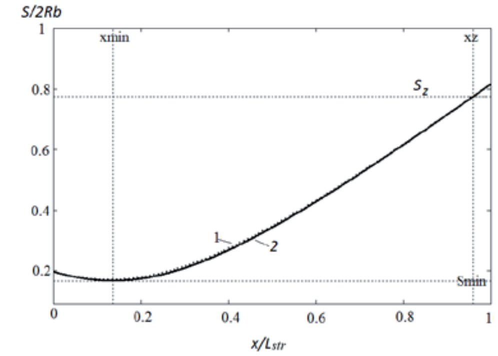
so, the crossing point coordinate is

$$x_z = \frac{s \cdot L_{str}}{s + k \cdot t}\ . \tag{6}$$

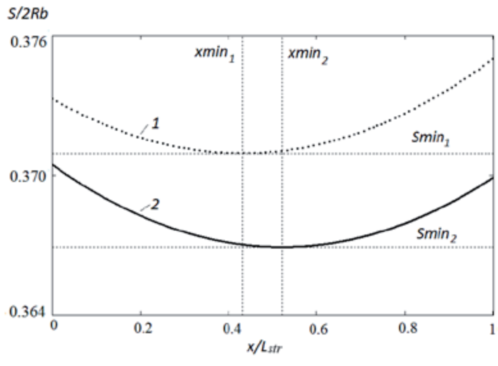
Then the distance S_2 between strings in the crossing point can be calculated as

$$S_2(x_z) = \frac{2s \cdot Rb}{s + k \cdot t}\ . \tag{7}$$

The plots on fig. 3,a demonstrate the changing of distance between rope strings in the range of $x \in [0, L_{стр}]$, where curve 1 responds to distance S_1 for a standard fastening, and curve 2 – to S_2 for cross-fastening. The area of minimum values is shown in a bigger zoom on fig. 3, b. The minimum distance between branches for both ways of fastening is about 2 m ($S\min_1 = 1.88$ м, $S\min_2 = 1.81$ м), the coordinates where the strings are situated nearest correspond to $x\min_1 = 10.25$ m, $x\min_2 = 10.54$ m. For the cross-fastening the distance between branches reduces only by 70 mm, which is only 3.6% of S_1. The horizontal component of Sz changes approximately to 1.8 – 2 m, while the vertical Sy – only to 0.5 m.



a



b

Fig. 3. The distance between branches: a – from pulley to drum; b – in the minimum value range; 1,2 – standard and cross ways of fastening

Besides studying, the minimum distance between branches it is necessary to analyze the vertical component change in the point where ropes are located one above another. From the expression (6) and (2) we can define the values x_z= 93.5m and $Sy(x_z)$ = 4.46 m. Points x_z and $x\min_2$ are sufficiently distanced from each other. Next it'll be analyzed what parameters have the greatest impact on the change of the distance between the cross-fastening ropes. For that we introduce the dimensionless parameters with subscript p, related to the fluting pitch t

$$Rb_p = \frac{Rb}{t}; \quad Db_p = 2 \cdot Rb_p; \quad s_p = \frac{s}{t};$$
$$S\min_p = \frac{s_p \cdot Db_p}{\sqrt{(s_p + k)^2 + Db_p^{\ 2}}}.$$

From analyzing the data on fig. 4 it may be concluded that the drum diameter has virtually no effect on the distance $Smin_p$, while the distance between pulleys and the number of empty grooves between ropes has the most influence. On the vertical plane, from the expression (2), the most important factor is the drum diameter.

Fig. 5 shows in percent the change of minimum distance between ropes ($100 \cdot \Delta S/S\min_1$, where $\Delta S = |S\min_1 - S\min_2|$) in case of cross-fastening, depending on the number of empty grooves between left and right branches k and the distance between pulleys s relatively to the drum width. When parameters k or s are increasing the value ΔS is increasing too. For example, when k = 25 and s = B the difference ΔS is 20% relatively to S_1. The maximum value ΔS = = 113 mm is achieved when x = 3.7 m.

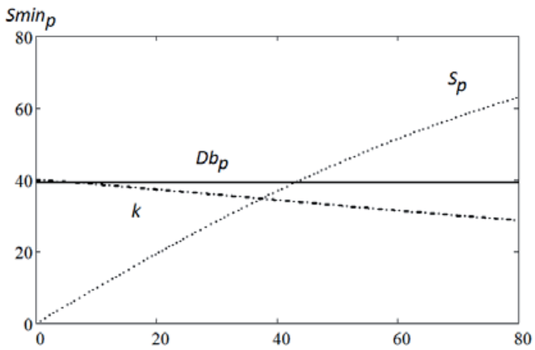


Fig. 4. Dependences of the reduced distance between strings on the dimensionless hoist parameters: 1 – drum diameter Db_p; 2 – distance between pulleys s_p; 3 – number of empty grooves k

Thus, in case of cross-fastening of ropes when $\alpha_в < \alpha_н$ winding rope friction on the rope adjacent turn is absent, this improves ropes exploitation conditions. Using the recommendations on choosing the fluting pitch [1] (using $\alpha_н$ instead of $\alpha_в$) it is possible to reduce the screw flute pitch and to increase the drum rope capacity, correspondingly. While designing a new hoisting plant the string cross-fastening allows to modify the drum width or the pulley location.

For the hoist TSR 5×4.66/0.5 of the mine "Tchentralnaya" rope cross-fastening makes possible to reduce the fluting pitch from 48 mm up to 41–42 mm for the rope diameter dk = 39 mm. The rope capacity increases (excluding friction coils, rope spare length and so on) from 1468 m up to 1728–1687 m, which is for 15–18 %.

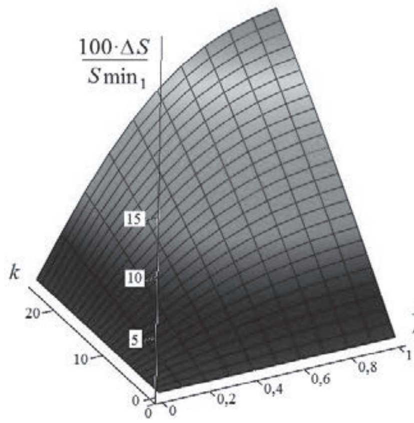


Fig.5. The change of minimum distance between strings in case of cross-fastening ropes in percent

According to the above calculations for mine hoisting plant, it is possible to make the conclusion that changing the ropes fixing very slightly affects on the distance between the rope strings. However, it is necessary to check out how the rope branch position will change if we consider static sag and longitudinal-transverse vibrations of strings.

Impact of the string oscillatory motion of a hoisting plant on the minimum distance between the rope strings in case of cross-fastening ropes. For safe use of this technical solution, it is necessary to analyze the vibration effect of the rope strings on the minimum distance between them in the most dangerous operation mode of a hoisting plant – safety-braking mode, which provokes the maximum possible oscillations in strings.

We adopted the following hypotheses: the rope length during transient process remains constant, since braking distance is a small part of the rope length; energy dissipation in the ropes during vibrations is described by the model of Foigt; the effect of torsional vibrations on the longitudinal-transverse vibrations is not considered; the transverse vibrations are considered only on the machine of the rope static sag.

For the definition of the string shape the finite element mode of a hoisting plant (see fig. 6) has been developed.

The drum and pulleys are represented as mass points M_b, M_{shl}, M_{shr} with corresponding reduced mass. The rope plummet with end load is replaced by a dynamic analogue consisting of parallel-connected visco-elastic oscillators and rigid mass [1]. For the right branch we use the notations: M_{ir} – mass, C_{ir} – rigidity, $\alpha_{ir} = \mu \cdot C_{ir}$ – damping factor of i-th oscillator, M_{0r} – rigid mass. For all tones of vibrations the viscosity parameter is adopted μ = 0.01 c [1]. Similar notations are taken for the left branch: M_{il}, C_{il}, α_{il}.

The rope string is represented as a sequence of nonlinear visco-elastic beam finite elements with bending and longitudinal rigidity. It is accepted hinge-support conditions for the rope points M_b, M_{shl}, M_{shr}. The rope string is loaded by distributed linear load gk, which is equal to the distributed rope weight and directed at the angles θ_l and θ_r, that are equal to the horizontal string gradient. Oscillators are operated by the gravities G_{ir} and G_{il}, the drum – by the braking force F. In the initial time, all points of a hoist are moving with constant velocity V_0.

Constructed finite element hoisting plant model has been tested on examples with well-known analytical solution for such hoist parts as string and plummet. The modeling error of the rope string by beams for longitudinal-transverse bending did not exceed 5% in statics and dynamics. The replacing error of the rope plummet by its dynamic analog did not exceed 10%. The results of the transverse vibrations of string derived by the finite element model are corresponded well with the experimental data in works of A. Obukhov.

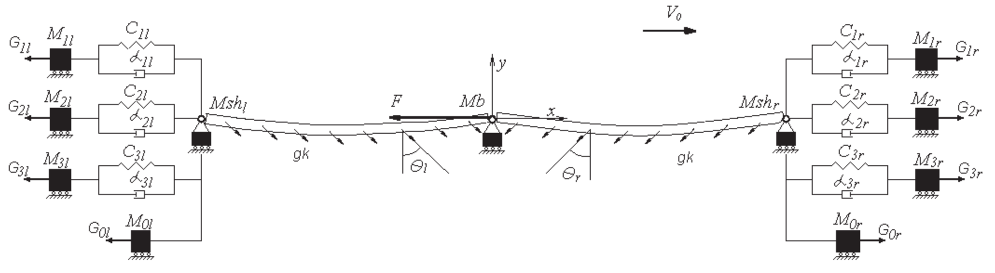


Fig. 6. Finite element model of a hoisting plant: M_{il}, C_{il}, α_{il} – parameters of i-th oscillator in the left branch;, M_{ir}, C_{ir}, α_{ir} – parameters of i-th oscillator in the right;Mb, Msh_l, Msh_r – a drum mass, left and right pulleys; F – braking force; G_{il}, G_{ir} – gravities in left and right branches

The analysis of the vibration effect of rope string on the minimum distance between strings during the mode of safety braking is given bellow for the hoisting plant TsR-5x4.66/0.5 of mine "Tchentralnaya"

While vessel is moving along the shaft, the rope plummet length is constantly changing and consequently the frequencies of the longitudinal and transverse rope vibrations are changing. Therefore, it is necessary to make the calculations for different positions of the empty and loaded vessels in the shaft.

Using the finite element model of a hoisting plant the maximum amplitude dependence of string mid-point transverse oscillations was determined for descent and ascent of the empty and filled vessels. The plots of these dependences are shown on fig.7, where along the abscissa there are ratio values of the plummet length lo to the shaft depth H, along the vertical axis there are maximum amplitudes.

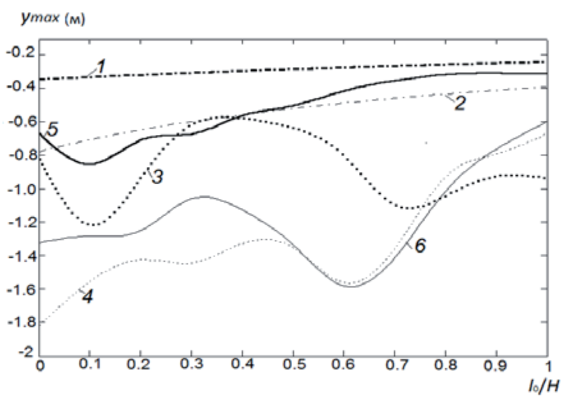


Fig. 7. The maximum amplitude dependences of the string mid-point during transverse oscillations: 1, 2 – static string sagging in the branches with loaded and empty vessels; 3, 4 – maximum amplitude of oscillations in the branches with loaded and empty vessels during lifting; 5, 6 – maximum amplitude of oscillations in the branches with loaded and empty vessels during descending

As it is seen from the plots, the maximum amplitude of rope string vibrations is observed at the position of empty vessel at the receiving area top. The maximum dynamic amplitude factor of string longitudinal-trans-verse vibrations for the studied hoist does not exceed 3.3.

Let us see the worst case of rope string mutual disposition, which corresponds to the maximum amplitude of transverse string vibrations and to the movement of strings towards each other (fig. 8).

The dashed lines 1, 2 show the static sags of upper and lower rope branches, the line 3 corresponds to the lowest possible upper branch position, oscillating with maximum amplitude, and the line 4 – to the upmost possible lower branch position. The minimum distance between branches S_{min} is reached at the point x_{min}, where the ropes are separated from each other along the cylinder axis Z for 2 m, therefore, transverse vibrations cannot cause the contact. Let us consider the neighborhood of the point x_z, where left and right hoist branches are one above the other. At the string crossing point x_z the minimum distance between the strings is denoted as ΔY.

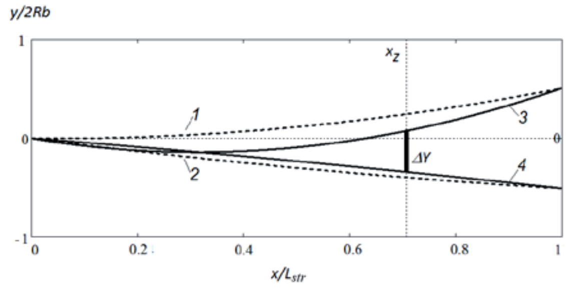


Fig. 8. Location of the string points corresponded to the maximum amplitude: 1, 2 – static sages of the upper and lower branches; 3 – the lowest coordinates of upper branch; 4 – the upmost coordinates of lower branch

The maximum amplitude of transverse rope vibrations can be written as

$$U(x,t) \le \beta \cdot U_{st}(x),$$

where β is the maximum dynamic factor; U_{st} – the static sag. Then the minimum distance between vibrating branches is

$$\Delta Y = S(x_z) - (1+\beta)\cdot\left|U_{st}^{emp}(x_z)\right| - (1-\beta)\cdot\left|U_{st}^{load}(x_z)\right|,$$

where U_{st}^{emp}, U_{st}^{load} are the static sags of empty and loaded branches. Considering the bigger absolute sag value, the minimum distance can be estimated as

$$\Delta Y \approx S(x_z) - 2\beta\cdot\left|U_{st}^{\max}(x_z)\right|.$$

The criteria of non contact of ropes is

$$S(x_z) > 2\beta\cdot U_{st}^{\max}(x_z), \tag{8}$$

after expanding can be representing as

$$\frac{s\cdot Rb}{s+k\cdot t} > \beta\left|\frac{L_{\mathrm{str}}}{tg(\theta)}\cdot\left(\frac{\ln\left(1-\dfrac{gk\cdot\sin(\theta)\cdot L_{\mathrm{str}}\cdot k\cdot t}{(s+k\cdot t)\cdot(M_{\mathrm{emp}}\cdot g+gk\cdot l_0)}\right)}{\ln\left(1-\dfrac{gk\cdot\sin(\theta)\cdot L_{\mathrm{str}}}{(M_{\mathrm{emp}}\cdot g+gk\cdot l_0)}\right)}-\frac{k\cdot t}{s+k\cdot t}\right)\right|>0, \tag{9}$$

where M_{emp} is the mass of empty lifting vessel, l_0 – the rope length from the pulley to the vessel.

For the exploring hoisting plant we received the following results: the maximum dynamic factor is $\beta = 3.3$; the minimum distance between the branches is not less than 2.7 m at the crossing point of the strings; the criterion (9) is satisfied: 2.2>0.9. Thus, the strings do not contact each other.

Conclusions.

1. The cross rope fastening on a single-drum hoisting plant with the pulley location on the same axis let increase external and decrease internal deviation angles, eliminate winding rope friction from adjacent rope coil and reduce the fluting pitch and increase the drum rope capacity up to 15%.
2. The minimum distance between rope strings does not depend on the string length and is determined by the drum diameter ratio, distance between pulleys and the empty groove number between the rope branches.
3. Use of cross rope fastening leads to study the oscillation impact on the minimum distance between the ropes strings at the string crossing point.
4. The maximum amplitude of the rope string oscillations occurs in the branch with empty lifting vessel near upper receiving area.
5. The criterion of non-contact of the rope strings is developed for the case of rope cross-fastening.
6. For the hoisting plant of "Tchentralnayay" mine the minimum distance between the rope strings during the longitudinal-transverse vibrations in the mode of safety braking is less than 2.7 m, the criterion of non-contact of ropes is satisfied. The use of cross-fastening of ropes with the fluting pitch 41 mm allows to increase the drum rope capacity of the hoisting plant up to 18%.

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Мета. Дослідити вплив способів кріплення канатів до барабану однобарабанної підіймальної машини з розташуванням копрових шківів на одній вісі на мінімальну відстань між струнами канатів підіймальної установки у статичному та динамічному режимах.

Методика. Розроблена математична модель для визначення відстані між довільними точками струн канатів у просторі у статиці. Для дослідження поздовжньо-поперечних коливань струни каната складена кінцево-елементна модель підіймальної установки.

Результати. Показано, що при перехресному способі кріплення канатів у разі, якщо внутрішній кут девіації менше зовнішнього, виключається тертя каната, що навивається, об сусідній виток. Проведений аналіз впливу коливань канатів на мінімальну відстань між ними при перехресному кріпленні канатів.

Наукова новизна. При перехресному способі розташування канатів збільшення відстані між копрових шківів або кількості витків на барабані між канатами, що навивається та звивається, призводить до збільшення відстані між струнами канатів. Уведено критерій відсутності торкання струн канатів при перехресному кріпленні.

Практична значимість. Перехресний спосіб кріплення канатів до барабана однобарабанної шахтної підіймальної машини з розташуванням копрових шківів

на одній вісі дозволить зменшити крок навивання каната, збільшити канатоємність підіймальної машини.

Ключові слова: *шахтна підіймальна установка, канат, поздовжньо-поперечні коливання струни каната, кут девіації*

Цель. Исследовать влияние способов крепления канатов к барабану однобарабанной шахтной подъемной машины с расположением копровых шкивов на одной оси на минимальное расстояние между струнами каната подъемной установки в статическом и динамическом режимах.

Методика. Разработана математическая модель для определения расстояния между произвольными точками струн канатов в пространстве в статике. Для исследования продольно-поперечных колебаний струны каната составлена конечно-элементная модель подъемной установки.

Результаты. Показано, что при перекрестном способе крепления канатов в случае, если внутренний угол девиации меньше наружного, исключается трение навиваемого каната о соседний виток. Проведен анализ влияния колебаний канатов на минимальное расстояние между ними при перекрестном креплении канатов.

Научная новизна. При перекрестном способе расположения канатов увеличение расстояния между копровыми шкивами или количества витков между навивающейся и свивающейся ветвями на барабане приводит к увеличению расстояния между струнами канатов. Введен критерий отсутствия касания струн канатов при перекрестном креплении.

Практическая значимость. Перекрестный способ крепления канатов к барабану однобарабанной шахтной подъемной машины с расположением копровых шкивов на одной оси позволит уменьшить шаг навивки каната, увеличить канатоемкость подъемной машины.

Ключевые слова: *шахтная подъемная установка, канат, продольно-поперечные колебания струны каната, угол девиации*

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RESONANCE PHENOMENA IN QUASI-ZERO STIFFNESS VIBRATION ISOLATION SYSTEMS

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РЕЗОНАНСНІ ЯВИЩА У ВІБРОЗАХИСНИХ СИСТЕМАХ КВАЗІНУЛЬОВОЇ ЖОРСТКОСТІ

Purpose. To study dynamic processes in nonlinear oscillatory systems with quasi-zero stiffness and with one or many degrees of freedom, which are widely used in industry for cargo and personnel vibration isolation during transportation. The previous studies of such systems were based only on the numerical approaches. In this paper, we propose to investigate thoroughly the dynamics of the above mentioned systems and the conditions for the occurrence of resonance phenomena in them using the asymptotic methods of nonlinear mechanics and applying the apparatus of special periodic functions.

Methodology. The methods of studying resonance oscillations of vibration isolation equipment are based on the asymptotic methods of nonlinear mechanics, wave theory of motion and the use of special Ateb-functions.

Findings. In this work, for the nonlinear quasi-zero stiffness vibration isolation systems with one and two degrees of freedom, we analytically obtained the conditions of resonance oscillations, threshold values of resonance amplitudes depending on the system parameters.

Originality. For the first time, the dynamic processes in systems with concentrated masses and quasi-zero stiffness were analyzed based on analytical approaches. In contrast to numerical approaches, the analytical approaches allow investigating the features of the dynamics of such systems more precisely.

Practical value. The proposed method may solve the problems of analysis, and the problems oscillatory systems synthesis at the design stage, as they allow us to choose such elastic properties of dynamical systems that prevent resonance phenomena. These modes of equipment operation may assure efficient and safe transportation.

Keywords: *mathematical model, nonlinear oscillations, quasi-zero stiffness, vibration isolation system, resonance, special functions*

Introduction. Background and literature review. Further increase in machine productivity, intensification of technological processes and the application of new technologies based on the theory of oscillations are closely connected with the condition that the modern machine devices should reliably operate in a wide range of loadings, amplitudes and forced oscillations. In particular, rotary power tools are widely used in industrial production. How-