SUBSTANTIATION OF PARAMETERS OF CARGO MOVEMENT BY CAR ROPE SYSTEMS

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Resume

The automobile cable systems for loading and unloading operations between the vehicle-support and vehicle-vehicle points are presented in this paper. The scheme of the automobile mobile cable system is based on a two-cable cableway. Based on this scheme, the parameters of the movement of goods by automobile cable systems are substantiated at variable parameters: mass m artificial loads together with the mass of the frame structure: 35 ≤ m ≤ 95 (kg); the angle of inclination α pairs of parallel ropes to the horizon: 10 ≤ α ≤ 15 (°); tension force ropes 2400 ≤ T_D ≤ 5000 (N). It has been established that the dominant effect on the minimum time T has angle α, then mass m and tension forces T_D ropes. At the same time, it is worth noting that the range of discrepancy of the quantities T values, when the above factors change, is insignificant and is close to 5,7 % for (m/α) and 6,6 % for (T_D/α).

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1 Introduction

The study of the development of transport and logistics technologies in the system of freight transportation using the rope systems, which has become widely used in various fields, is outlined in this article. The research analysis of the processes of loading artificial cargo into warehouses by various types of transport and technological machines has been provided in [1-2]. However, traditional types of transport mechanisms are structurally complex and material-intensive, and their use is economically feasible during the transportation of goods over long distances [3]. The well-known cable transport mechanisms for moving various artificial loads, which can be prototyped in the development of new transport and technological machines, are presented in [4-6].

The use of rope transport mechanisms on the self-propelled wheeled mobile transport and handling complexes allows for the autonomous delivery of the necessary technological equipment to the workplace and its quick deployment for transport and handling operations [7-8]. In articles [9-10] the mobile lifting and rope mechanisms of transport equipment for the sustainable development of heavy transport are formed by autonomous self-propelled units, connected by a single rope system based on the wheeled chassis of a high load capacity. Regularities of change of optimal parameters of intermediate supports and load-carrying and traction ropes at the change of design capacity

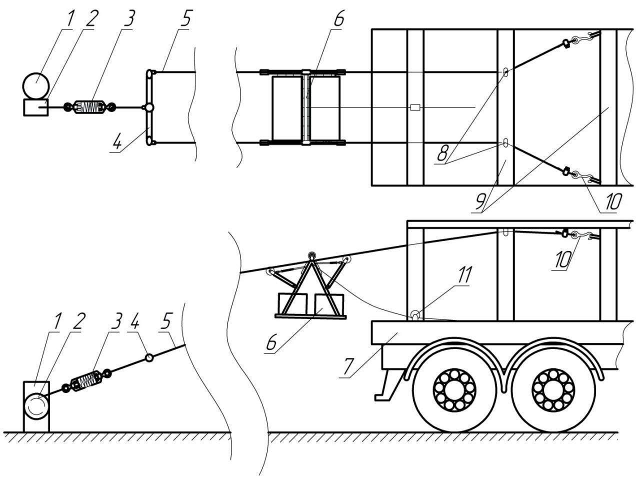

Figure 1 *The schematics of a car mobile rope system for loading and unloading between the points of the car support: 1 - support; 2 - winch; 3 - spring; 4 - rocker arm; 5 - two-cable cable car; 6 - suspended cargo platform for moving cargo; 7 - car; 8 - limiters; 9 - hanging frame: 10 - hooks; 11- tow-line*

of a cableway were revealed based on an analysis of performed calculations [11].

The developed experimental devices and research methods, together with the results of preliminary studies of the proposed rope transport and technological mechanism, are given in the articles [12-13].

It is important to use those devices in automobile construction to facilitate the movement of people and goods through small rivers, various obstacles, in the mountains, transportation of injured or wounded, ammunition, timber, etc. This type of modern multi-purpose cable transport mechanism is perspective and effective.

The high novelty of this transport technology and the impossibility of fully using the existing design and calculation methods, which were developed for stationary cargo mobile transport and transshipment complexes, require the creation of scientifically based theoretical foundation for design, calculation, and modelling of work processes at all stages of the life cycle.

Improving the performance of rope systems for transport and handling operations, which will ensure the creation of a promising design of a multi-purpose rope transport system with the best characteristics, maximum functionality and high-quality indicators, is a key step in solving this technical problem.

2 Materials and methods

Nowadays, the usage of the rope systems has become widely used in various fields. To this end, two structural diagrams of car mobile rope systems for loading and unloading between the points of car-support (Figure 1), and car-car (Figure 2), have been developed [14].

In the first scheme, the car rope system, which consists of a rope (two-rope cableway) 5 and a suspended load platform for moving loads 6, is fastened by hooks 10 through the limiters 8 to the hinged frame 9 of the car 7, and on the other hand through the rocker arm 4 spring 3 and ratchet mechanism (winch) 2 to the support (a screw quick-mounting support can be used) 1. The tension of the rope system is carried out by the ratchet mechanism 2, and the compensation of dynamic loads is carried out by the lanyard spring 3. The speed of the suspended load platform 6 is regulated by the line 11. Such a car rope system can be widely used for loading and unloading, moving people and goods, setting up crossings over small rivers and various obstacles, etc.

Figure 2 shows a car mobile rope system, which provides loading and unloading operations between two cars, consisting of a rope (two-cable car) 1 and a suspended cargo platform for moving goods 2. The rope 1 through limiters 3 is attached by a hook 9 and through the winch 5 to the frame 4 of the car 6, and on the other hand through the tension rollers 8 to the frame of the car 7.

The tension of the rope system is carried out by the ratchet mechanism 5, and the speed of the suspended loading platform 6 is regulated by line 10. This system can be widely used for transshipment of goods in the implementation of road freight transport.

Substantiating the parameters of cargo movement by the car rope systems, it has been stated that the loading processes of two cable cars are identical (by adjusting their equal tension through the rocker arm 4 (Figure 1) or tension rollers 8 (Figure 2). Therefore, a flat problem with conditional movement of a load of half mass $(m = m_{\Sigma}/2)$ on only one rope has been considered where m_{Σ} is the total weight of frame construction and cargo.

It has been assumed that the length of the route (projection of the length of the ropes on the horizontal plane) is equal L, the height of the rope suspension in the loading area (point A) - H, and the unloading area (point B) - h. Then the difference in height of the rope suspension ΔH at the points of loading and unloading will be $\Delta H = H - h$, Figure 3 [15]. Since the rope has

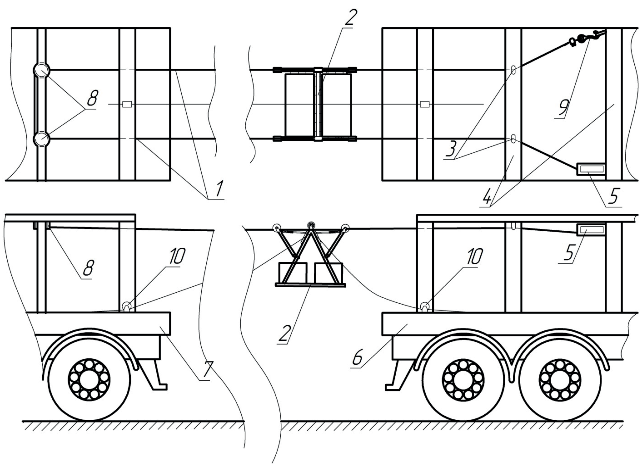

Figure 2 *The schematics of a car mobile rope system for loading and unloading between the points of car-car: 1 - two-cable cableway; 2 - suspended cargo platform for moving cargo; 3 - limiters; 4 - frame; 5 - winch; 6 - car; 7 - car frame; 8 - tension rollers; 9 - hook; 10 - tow-line*

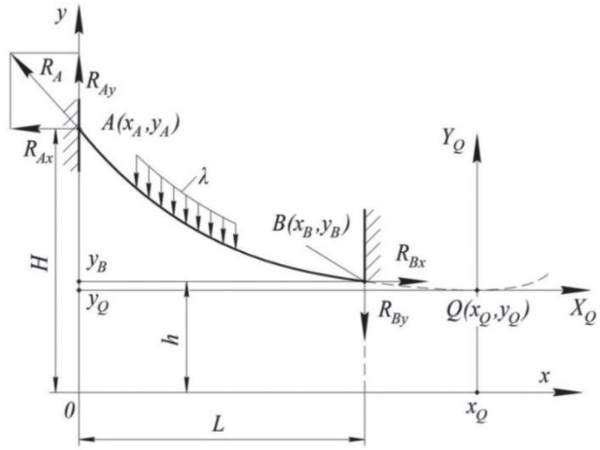

Figure 3 *Calculation scheme of placement and fastening of the ropeway*

its own weight, which is evenly distributed along the route, the unloaded rope (without a suspended platform with a load) is modeled with a flexible inextensible heavy thread with a uniform linear distribution λ of the entire mass of the rope m_k in length l_k [16-19]:

$$\lambda = m_k / l_k = \rho_k S_k \,, \tag{1}$$

where ρ_k - specific weight of the rope; S_k - rope cross-sectional area.

The sagging of an unloaded rope by mass m_k has been considered. According to the model of sagging heavy thread, it is stretched under the action of weight $G_k = m_k g$, and in the rope there are tension forces $T = T(x)$, the axial component of which is constant $T_x = T_0 = const$ and numerically equal to the horizontal component of the reaction force of the supports R_{Ax} and R_{Bx} respectively at the points of suspension A and B. Place the beginning of the coordinate system *xQy* at the beginning of the loading zone, where the axis *Ox* is directed along the movement of the weight horizontally at the floor level of the warehouse. The coordinates of the points of the rope suspension in the loading area are $A(0;H)$, in the unloading area $B(L;h)$.

The equation of the chain line in the coordinate system *xOy* will be [16, 19]:

$$y = a_k \left[ch\left(\frac{x - x_Q}{a_k} \right) - 1 \right] + y_Q \,, \tag{2}$$

where a_k- chain line parameter, $a_k = T_0 / (\lambda g)$; x_Q and y_Q coordinates of the vertex of the parabola Q in the coordinate system *xOy*, that is, the points of the global minimum of the deflection line passing through the suspension points $A(x_A; y_A)$ and $B(x_B; y_B)$.

For roads with previous working mounting tension T_0, which provides a small deflection, the equation of the sag line will be:

$$y_M = \frac{(x - x_Q)^2}{2a_k} + y_Q \,. \tag{3}$$

Accordingly, in the system *xOy* (Figure 3) Equation, which determines the running length of the cable as the load passes, has the form:

$$l_{kM} = x + \frac{x_Q^3 - (x_Q - x)^3}{6a_k^2} \tag{4}$$

Equations (3)-(4) are obtained after expanding the

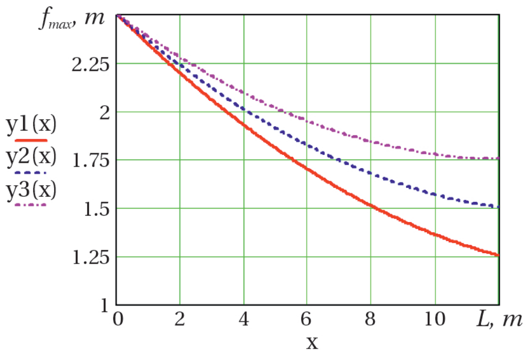

Figure 4 *The shape of the sagging ropes line on the route length L =12 m with the same mounting tension depending on the difference in the levels of attachment of the rope: y1(x) - ΔH = 1.25 m; y2(x) - ΔH = 1.0 m; y3(x) - ΔH = 0.75 m*

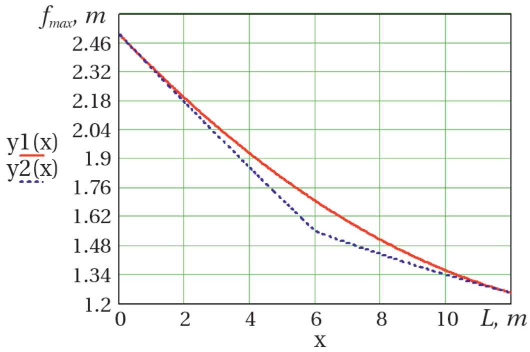

Figure 5 *The shape of the sagging ropes line without cargo and with suspended cargo, located in the middle of the route*

rope deflection Equation (2), and the similar rope length equation, into the Maclaurin series with discarding insignificant components. Thus, for the track length L= 20 m with deflection 1 m (a_k=50 m^{-1}), the difference in deflections, calculated according to dependencies Equations (2) and (3) will be only 3 mm, and the difference in the calculated lengths will be less than 0.02 mm, which is significantly less than the measurement errors of these values.

For a ropeway designed to move cargo under the action of gravity, the height of the suspension in the loading area H (point A) is greater than the height of the suspension in the unloading area h (point B). The equation of a straight line passing through the points of suspension point A and point B, in the xOy system is:

$$y_s = y_A - x\mathrm{tg}\beta = H - x\mathrm{tg}\beta\,, \tag{5}$$

where β - the angle of the straight line connecting the suspension points (the angle of the straight line), $\mathrm{tg}\beta = (H-h)/L = \Delta H/L$.

Running sagging ropes will be [15-17, 19-20]:

$$f(x) = y_s - y_M \tag{6}$$

Taking into account the values of the suspension points $A(0;H)$ and $B(L;h)$ coordinates in the system xOy:

$$x_Q = \frac{L}{2} + \frac{a_k(H-h)}{L} = 0.5L + a_k\mathrm{tg}\beta\,, \tag{7}$$

$$y_Q = H - \frac{x_Q^2}{2a_k}\,. \tag{8}$$

It follows from Equations (3) and (7) that for a ropeway of length L with a height difference $\Delta H = H - h$ at the initial installation value $a_k = 0.5L/\mathrm{tg}\beta$, the zone of maximum sagging coincides with the zone of unloading ($x_Q = x_B$); at $a_k < 0.5L/\mathrm{tg}\beta$ is located on the route between the loading and unloading area ($0 < x_Q < x_B$); at $a_k > 0.5L/\mathrm{tg}\beta$ - off the track ($x_Q > x_B$); or according to Equation (4) on the approximate dependence:

$$l_{AK} = s = x + \frac{3x_Q x(x_Q - x) + x^3}{6a_k^2}\,, \tag{9}$$

where s - the linear path parameter corresponding to the running length of the rope.

The length of the rope, suspended between the supports $A(x_A; y_A)$ and $B(x_B; y_B)$ with tension T_0; will be $l_{ADB} = l_k = L(1+\mathrm{tg}^2\beta) + L^3/(64a_k^2)$. The initial length of the rope, without taking into account the increase in tension, will be $l_0 = l_k/(1+\varepsilon_0)$, where $\varepsilon_0 = T_0/(E_k S_k) = a_k\rho_k g/E_k$ - relative elongation of the rope from the mounting tension.

With the same mounting tension (a = *const*), depending on the height of the suspension h in the unloading zone, the route will have a different curvature (Figure 4).

Horizontal reactions of the supports (suspension points), stretching the rope without load are [19, 21-24]:

$$R_{Bx} = -R_{Ax} = T_0\,. \tag{10}$$

Taking into account Equation (3), the vertical reactions of the supports are:

$$R_{Ay} = \dot{y}_x(0)\cdot T_0 = x_Q T_0/a_k\,, \tag{11}$$

$$R_{By} = \dot{y}_x(L)\cdot T_0 = (x_B - x_Q)\cdot T_0/a_k\,. \tag{12}$$

The case of loading the rope at point D, the suspended frame structure has been considered, with a load whose total weight $G = mg$ significantly exceeds the weight of the ropes (an order of magnitude or more). It is obvious that under the action of the load the rope will be stretched, and the tensile forces will significantly exceed their mounting tension and, accordingly, the curvature of the branches of the route will be negligible.

Figure 5 shows the chain lines of sagging of the rope with the parameter a_k = 100 m without load (solid line) and with suspended load (dashed line) with load parameter f a_k = 350 m.

As follows from Figure 5, the branches of the loaded

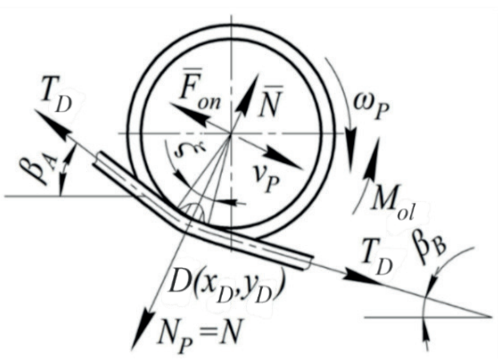

Figure 6 Calculation scheme of interaction of the rope with the suspension roller

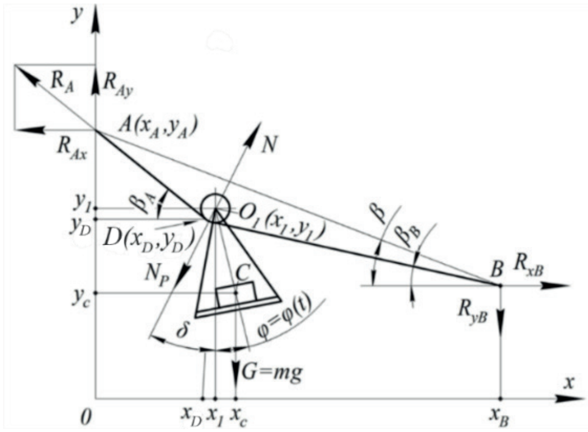

Figure 7 Calculation scheme of the suspension of the frame system with the load on the ropeway

ropeway are practically straightened. In the load zone (point D) the rope covers the suspension roller, and in the contact zone between the rope and the roller there is a normal force N, Figure 6.

Let the angle of coverage of the roller rope is ξ, and the axial tensile strength of the rope in the area of the roller is T_D. If the forces of sliding friction were neglected [25], the tension of the rope point D on both sides of the supports would be equal $T_{DA} = T_{DB} = T_D$.

Then, the normal reaction force of the rope acting on the roller will be [12, 14]:

$$N = 2T_D \sin(\xi/2). \tag{13}$$

For this case, with a small length of the transport route, the weight of the suspended frame structure with the load is significantly greater (an order of magnitude or more) than the weight of the rope (ropes), and, therefore, it is assumed that the rope branches from the supports to the load area are straight, (Figure 3, 7). Let's assume that the angle of inclination of the line is positive when the point of suspension of the rope is above the point of suspension of the load (the angle of inclination of the line towards the loading point A) and negative when the point of suspension of the rope is below the point of suspension of the load (angle β_B, according to Figures 6, 7). In this case, the points of suspension of the rope (point A and point B) with the point of action of the load suspension (point D) form a triangle in which the side AB is placed at an angle β with respect to the axis Ox, the side AD at an angle β_A, and the side BD - at an angle β_B, Figure 7. Accordingly, the angles of the triangle will be equal: $\angle A = \beta_A - \beta; \angle B = \beta + \beta_B$; $\angle D = \beta_D = 180 - -\xi_þ = 180 - \beta_A - \beta_B$, where the angle of the suspension roller coverage $\xi_þ$ is:

$$\xi_þ = \beta_A + \beta_B. \tag{14}$$

The reaction force vector N divides the angle $\angle D$ in half and is directed at an angle $\beta_D/2 = 90^\circ - (\beta_A + \beta_B)/2$ with respect to the sides AD and BD. Therefore, the vector N is placed at an angle concerning the vector of the force of gravity G (or to the axis Oy).

It is assumed that the forces of inertia are insignificant. Then, the normal reaction of the rope, when interacting with the roller, will be:

$$N = G\cos\delta = G\cos\left(\frac{\beta_A - \beta_B}{2}\right). \tag{15}$$

The tension of the ropes due to the action of the load in the area of its suspension:

$$T_D == \frac{G\cos[(\beta_A - \beta_B)/2]}{2\sin[(\beta_A + \beta_B)/2]}. \tag{16}$$

Accordingly, the horizontal and vertical reactions at the points of suspension:

$$R_{Ax} = -T_D\cos\beta_A; \quad R_{Ay} = T_D\sin\beta_A, \tag{17}$$

$$R_{Bx} = T_D\cos\beta_B; \quad R_{By} = T_D\sin\beta_B. \tag{18}$$

Taking into account Equations (15)-(18), the angles of inclination of the branches β_A and β_B interconnected functionally, resulting from the equality of the projections of all the forces on the axis. From the analysis of dependences Equation (16)-(18) it follows that the projection of tension forces on the axis Ox of the rope, from the side of the highest suspension point (point A), is smaller than the tension force from the lower suspension point (point B) by magnitude:

$$T_{Bx} - T_{Ax} = N\sin\delta = G\sin\delta\cos\delta = \\ = \frac{G\sin(\beta_A - \beta_B)}{2}. \tag{19}$$

Since the suspension platform moves on the rope under the action of gravity, the reaction force of the rope on the roller is directed along the axis of the roller and has no tangential component, so the tension force of the rope T_D, in the case where the weight of the rope,

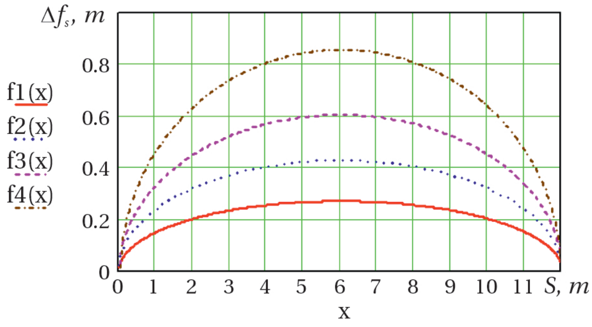

Figure 8 *Dependence of deflection* Δf_s *route on the road s, passed by a roller on a rope at* ε_Σ *: 1 –* ε_Σ *= 0.001; 2 –* ε_Σ *= 0.0025; 3 –* ε_Σ *= 0.005; 4 –* ε_Σ *= 0.01*

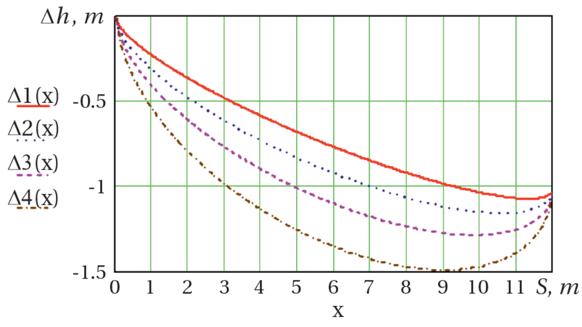

Figure 9 *Dependence of the running height difference* Δh *suspension system with cargo on the road s, passed by a roller on a rope at: 1 –* ε_Σ *= 0.001; 2 –* ε_Σ *= 0.0025; 3 –* ε_Σ *= 0.005; 4 –* ε_Σ *= 0.01*

compared to the weight of the load is insignificant and can be neglected, are virtually unchanged throughout the area between the points of suspension of the rope and point *D*. This is confirmed by the fact that the curvature of the loaded rope is insignificant, and its route is almost rectilinear, which is the basis for the assumption. Let the distance *AB*, Figure 7, between the rope suspensions (point *A* and point *B*) be marked as l_{AB} = 2*c*. With constant rope length l_k, $l_k > l_{AB}$, when moving the load on the rope, the point of its suspension *D* will move along a trajectory that corresponds to an ellipse with foci located at the points of suspension of the rope $A(x_A; y_A)$ and $B(x_B; y_B)$, and the center of which is located in the middle of the segment *AB*.

The graph of change of a deflection of a route *s*, on the way passed by a roller on a rope for various loading which is accepted invariable, that is for various values ε_Σ is shown in Figure 8.

Resistance to movement is determined by the running path *s* that the roller passes along the length of the rope. The length of the rope l_k exceeds the distance l_{AB} between the points of suspension point *A* and point *B*, $l_k = l_{AB}(1+\varepsilon_\Sigma)$. Here ε_Σ is the relative excess length of the rope distance between the points of suspension of the rope in a straight line $l_{AB} = 2c = L/\cos\beta$, taking into account its elongation from the pre-tension and the tension from the load [26].

For determination of the running deflection $\Delta f_s = \Delta f_s(s)$ at the point of suspension depending on the path of movement of the suspension roller on the rope *s* determines the energy parameters of transportation, so it is taken as an independent parameter in the model of transportation of goods by cable car. It is denoted as $s/l_k = k_s$. Then:

$$\Delta f_s = \frac{L}{2\cos^2\beta}\sqrt{\begin{array}{l}(2\varepsilon_\Sigma + \varepsilon_\Sigma^2)\cdot \\ \cdot[1+(1-2k_s)^2(1+\varepsilon_\Sigma)^2]\end{array}}. \quad (20)$$

The graph (Figure 9) shows the running height difference $\Delta h(x) = -(x_s \mathrm{tg}\beta - \Delta f_s)$ for different mounting tension of the cables, providing a given excess ε_Σ of the length of the stretched cable over the distance between the points of suspension of the rope.

The angles of inclination of the branches β_A and β_B (Figure 10) are determined from the triangle Δ*ABD*:

$$\beta_A(s) = \arccos\left(\frac{a}{s}\right) + \beta, \quad (21)$$

$$\beta_B(s) = ar\cos\left(\frac{l_{AB} - a}{l_k - s}\right) - \beta. \quad (22)$$

where *a* - projection of a stretched branch *AD* (running route *s*, which passes the roller suspension on the rope) per segment *AB* (distance between the points of suspension of the rope):

$$a = a(s) = s(1+\varepsilon_\Sigma) - l_{AB}(\varepsilon_\Sigma + 0.5\varepsilon_\Sigma^2). \quad (23)$$

The changing of the angles β_A and β_B in terms of *s*, for different mounting tension of the rope and according to the parameter ε_Σ, is shown in Figure 10.

For gravitational transport of goods, the direction of reaction from the ropes is set by the angle δ, determined by the directions of the branches of the ropes, that is the angles β_A and β_B, $\delta = (\beta_A + \beta_B)/2$. Changing the running value of the angle δ for different values ε_Σ, in terms of the route *s*, passed by the roller suspension is shown in Figure 11.

Given the known placement of the load at the current time and, accordingly, the known laws of change of angles β_A, β_B and δ at a given tension, which determines the required excess length of the cable ε_Σ, it can be stated that the axial tension of the cable mechanism and the reaction determined by Equation (16)-(18).

Figure 12 shows the change in axial tension force T_D loaded rope at different parameters ε_Σ the calculated lengths of the rope are set, and Figure 13 shows the dependence of the reactions of the rope suspension when moving the load along the route, on the parameter *s*.

These dependencies confirm that the main load due to the action of the suspended system for routes with the recommended working sagging of ropes, falls on the horizontal components of the reactions of the supports, R_{Ax} and R_{Bx}, at the point of suspension, which,

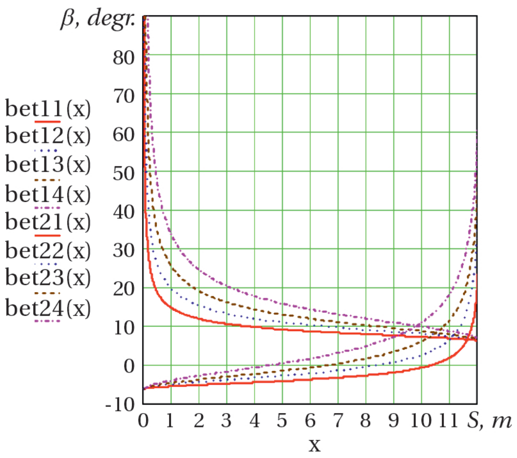

Figure 10 *Changing the angles of the rope branches* β_A *(1) and* β_B *(2) in terms of s, for: 1 –* ε_Σ *= 0.001; 2 –* ε_Σ *= 0.0025; 3 –* ε_Σ *=0.005; 4 –* ε_Σ *= 0.01*

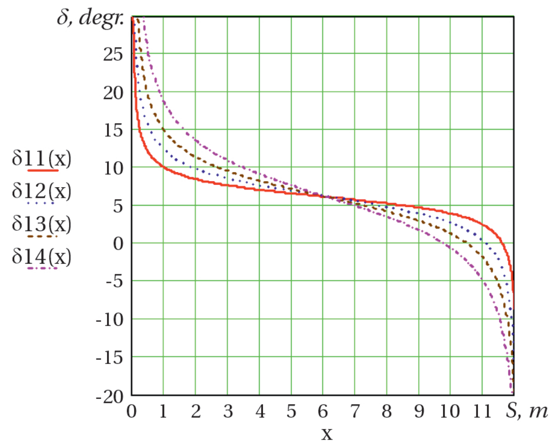

Figure 11 Changing the angle of the rope N on the roller suspension of cargo in terms of s for: 1 – ε_Σ *= 0.001; 2 –* ε_Σ *= 0.0025; 3 –* ε_Σ *= 0.005; 4 –* ε_Σ *= 0.01*

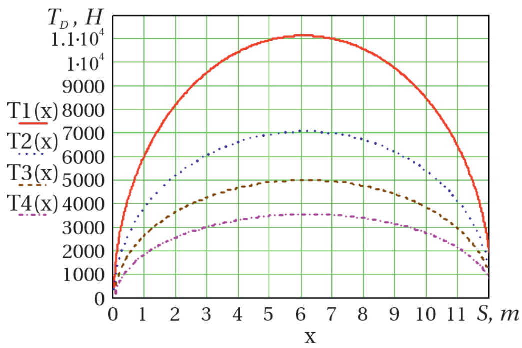

Figure 12 *Changing the force of axial tension* T_D*, H rope in terms of the parameter of the running length of the rope s to the load point D for: 1 –* ε_Σ *= 0.001; 2 –* ε_Σ *= 0.0025; 3 –* ε_Σ *= 0.005; 4 –* ε_Σ *= 0.01*

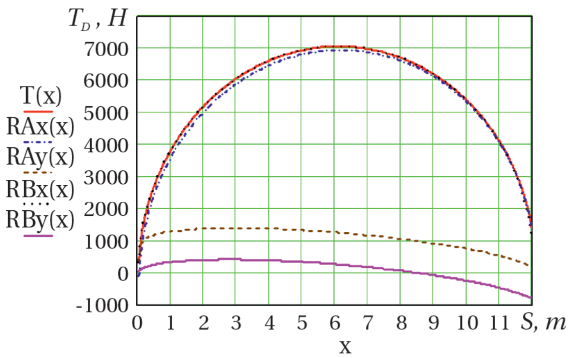

Figure 13 *Dependences of changes in the force of axial tension* T_D *and reactions,* R_{Ax}*,* R_{Ay} *and* R_{Bx}*,* R_{By} *on the parameter of the running length of the rope s at* ε_Σ *= 0.0025*

in practice, can take the forces of axial tension T_D of the rope.

Figures 7-13 show the change of system parameters depending on the path s traversed by the roller on the rope. In the coordinate system xOy the projection of the load on the axis Ox will be $x = s\cos\beta_A(s)$. At insignificant slopes of the loaded route values x and s practically coincide.

In this case, it is possible to assume that the load moves on the ropes without hesitation. The potential energy of the cargo at the loading and unloading points will be $W_{Pdow} = mgh_3$ and $W_{Punl} = mgh_p$, respectively. The potential energy difference will be [16-17]:

$$\Delta W_{Pdow} = mg(y_{dow} - y_{unl}) = mg(x_{unl} - x_{dow}) \times \times \mathrm{tg}\beta + \Delta f(x_{dow}) - \Delta f(x_{unl}). \tag{24}$$

The increase in kinetic energy is:

$$\Delta W_{Punl} = m(v_{Cdow}^2 - v_{Cunl}^2)/2\,, \tag{25}$$

where v_{Cdow} and v_{Cunl} - the initial speed that can be obtained by the system in the loading area and the suspension speed in the unloading area, respectively.

Loss of resistance to displacement due to the rolling friction forces $F_\mu = \delta_N N / r_p = \lambda_N N$ is:

$$\Delta A_{Punl} = \lambda_N N l_{kx} = \lambda_N l_{kx} mg \cdot \cos(\delta + \varphi)\,. \tag{26}$$

Substituting Equation (24)-(26) in the law of conservation of energy $\Delta W_{Pdow} = \Delta W_{Punl} - \Delta A_\lambda$ one obtains:

$$m(v_C^2 - v_{C0}^2)/2 = mg(x_{unl} - x_{dow})\mathrm{tg}\beta + + \Delta f(x_{dow}) - \Delta f(x_{unl}) - \lambda_N l_{kx} mg \cdot \cos(\delta + \varphi), \tag{27}$$

where the speed of a cargo in the unloading area is determined as:

$$v_{Cunl} = \sqrt{\begin{matrix} 2g(x_{unl} - x_{dow})\mathrm{tg}\beta + 2\Delta f(x_{dow}) - \\ - 2\Delta f(x_{unl}) - 2\lambda_N l_{kx} mg \cdot \\ \cdot \cos(\delta + \varphi) + v_{Cdow}^2 \end{matrix}}\,. \tag{28}$$

As a rule, such a speed at sharp stop of cargo and, accordingly, containers, will cause a shock that can lead to injury of a cargo. Springing of a suspended frame design by a clamping roller allows for a gradually reduce the speed completely, or to an acceptable value for a condition of not causing damage to the freight.

When the system is forcibly stopped in the suspension zone, its kinetic energy of gradual motion $\Delta W_{Pdow} = mv_{Cunl}^2/2$ will be converted into the kinetic energy of oscillating motion $\Delta W_{Pdow}^I = I_D\dot{\varphi}_D^2/2$:

$$mv_{Cunl}^2 = I_D\dot{\varphi}_D^2. \quad (29)$$

The moment of inertia I_D of the oscillating suspension platform with the load can be determined both theoretically and experimentally. Since the loads of different weights and configurations can be transported, the moment of inertia of the system is easiest to determine experimentally by period T_ω (frequency ω_I) of oscillations of the suspension system as a physical pendulum.

It is known that the period of oscillation as a physical pendulum [18] is:

$$T_\omega = \frac{2\pi}{\omega_I} = 2\pi\sqrt{\frac{I_D}{d_C m g}}, \quad (30)$$

where m - the total weight of the frame and load; d_C - the distance from the point of suspension to the center of mass.

Hence, the moment of inertia of the suspension system is $I_D = \frac{d_c mg T_\omega^2}{4\pi^2}$. Taking into account Equation (29), the initial angular velocity of the system, at its stop at the point of suspension will be:

$$\dot{\varphi}_{D3} = \sqrt{\frac{mv_{Cunl}^2}{I_D}}. \quad (31)$$

The equation of circular motion at a stop will be:

$$I_D\frac{d\dot{\varphi}}{dt} = T_I, \quad (32)$$

where T_I - moment creating springs and rope.

Since the spring system is pre-loaded, the average value of the torque, as the calculated value T_{IC}, has been taken. Then, the time before stopping the rotation of the system with the load, from the solution of the differential Equation (32) will be:

$$t_\Delta = I_D\frac{\dot{\varphi}_{D3}}{T_{IC}} = \frac{\sqrt{I_D m v_{Cp}^2}}{T_{IC}}. \quad (33)$$

The analysis of the obtained results allowed to establish that the increase in speed and, accordingly, the decrease in transport time occurs with a decrease in the rolling resistance, which is achieved by increasing the mounting tension, which leads to a decrease in the angle of the roller rope coverage ξ and, accordingly, reducing the coefficient of friction λ_N. In addition, transporting the heavier loads leads to an increase in sag difference $\Delta f(x_{dow}) - \Delta f(x_{unl})$, which also reduces transportation time. Structurally, the rotation of the system is limited by locking devices in the unloading zone, which dampen the excess speed in cases where it is not damped by the spring-loaded rollers. The residual speed has been observed for heavy loads (more than 50 kg) and was not more than 10% (less than 0.2 m / s) of the suspension speed at the end of the route.

Theoretical studies have shown that the spring rollers significantly reduce the oscillation amplitude of the system and clamping the rope also act as a damper, as the oscillating process leads to bending of the rope and internal friction between its threads, which leads to sharp damping of oscillations that can occur when starting and stopping the frame construction with cargo.

3 Results and discussion

To confirm the theoretical developments, experimental studies have been conducted. They were based on the developed method, which is presented in [24]. An experimental setup has been developed to adjust and determine the tension of ropes, the magnitude of their deflection in the vertical direction from the value of the discrete load in different zones, the angle of the pair of ropes to the horizon, and the time of movement on the frame structure of a piece cargo from the loading zone to the unloading zone. The scheme of functioning of the automobile mobile rope system based on a two-cable cable car, on which the suspended cargo platform moves on rollers has been outlined. This platform is made in the form of a rigid frame structure, which is suspended on load-bearing rollers and its movement is limited by the action of spring-loaded pressure rollers [15] Figure 14. At measurement of forces at loading of levers with pressure the rollers of rope system the dynamometer electronic brand DE 5-0.5 was applied [14].

When determining the dependence of the transportation time T of the piece goods in containers by rope mechanism in the warehouse, the range of changes in the values of the factors was as follows: mass m of artificial loads together with the constant mass of the frame structure for their location: $35 \le m \le 95$ (kg); the angle of inclination α of a pair of ropes parallel to the horizon: $10 \le \alpha \le 15$ (°); tension force T_D of ropes $2400 \le T_D \le 5000$ (N).

Based on the carried-out statistical processing of the received results of research on multifactor experiment the regression equation of dependence is constructed $T = f(m; \alpha; T_D)$ [2]:

$$\begin{aligned} T = {} & 11.915 - 0.2\cdot 10^{-3}T_D - 0.543\alpha - 0.0153m + \\ & + 0.71\cdot 10^{-8}T_D^2 + 0.35\cdot 10^{-5}T_D\alpha + 0.0121\alpha^2 + \\ & + 0.12\cdot 10^{-3}\alpha m + 0.51\cdot 10^{-4}m^2. \end{aligned} \quad (34)$$

The analysis of the regression Equation (34) showed that the average values of the factors from the specified ranges of the transportation time t of the piece goods in the warehouse is approximately $T \approx 5.92$ s. The dominant effect on the minimum transportation time t of the piece

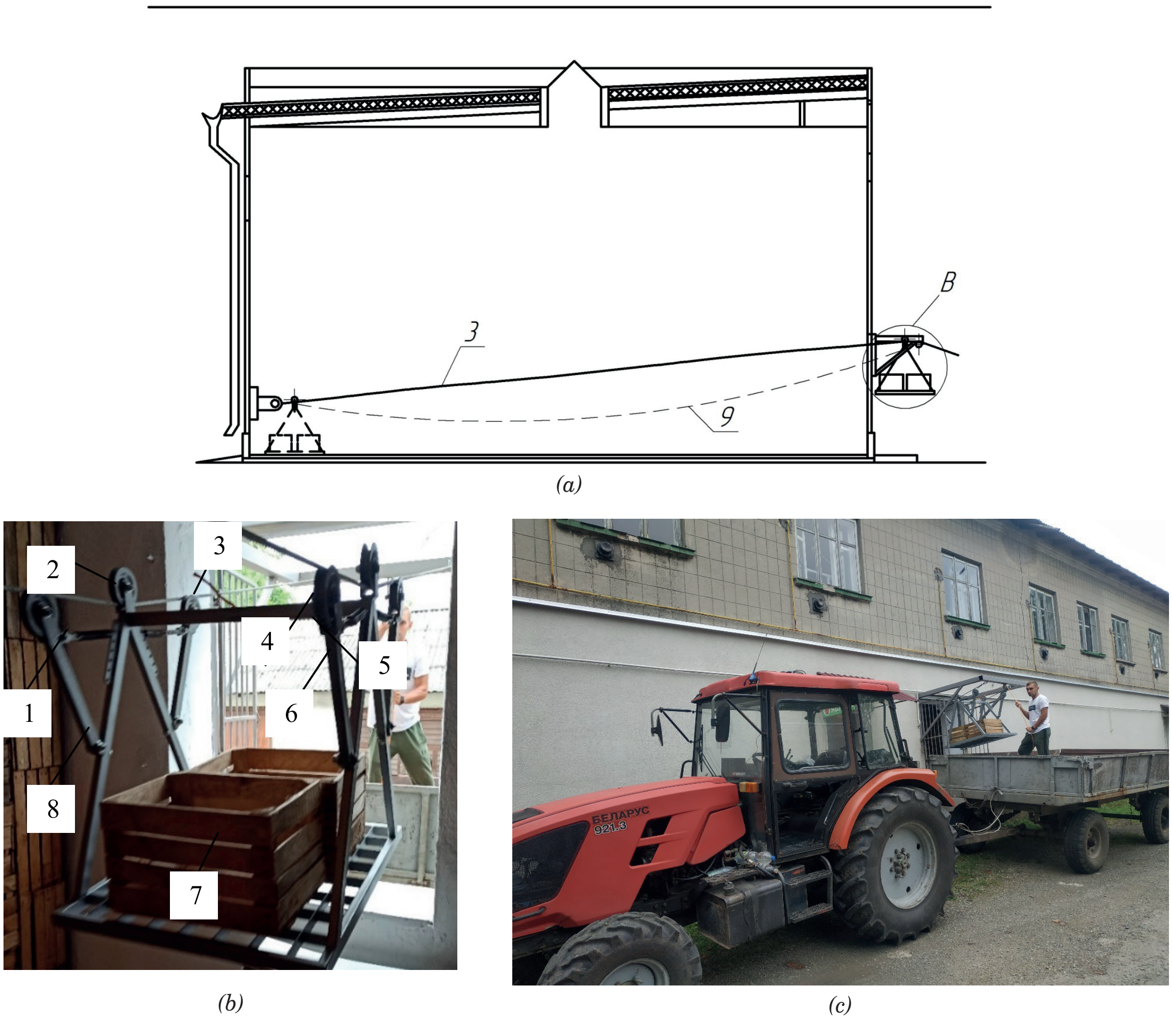

Figure 14 *General view of experimental installation of the rope system: a) structural diagram; b), c) general view of the artificial cargo loading area: 1 - hinged levers; 2 - rollers; 3 - two-cable cable car; 4 - pressure rollers; 5 - tension springs; 6 - frame structure; 7 - artificial loads; 8 - brackets; 9 - helper rope*

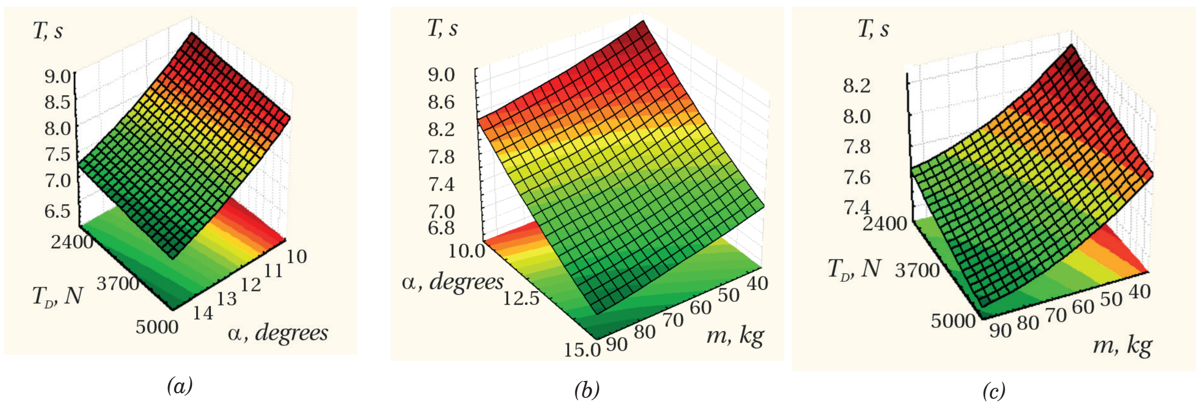

Figure 15 *The Response surface transportation of piece goods in the warehouse due of time change in terms of the force of tension, the angle of their inclination to the horizon and the mass artificial cargo: a -* $T = f(T_D, \alpha)$*; b -* $T = f(\alpha, m)$*; c) -* $T = f(T_D, m)$

goods in the warehouse has the angle of inclination α of a pair of parallel ropes to the horizon ($T \approx 5.47$ s), then the mass m of artificial loads together with the constant mass of the frame structure for their location ($T \approx 5.78$ s) and the tension force T of ropes ($T \approx 5.83$ s). It should be noted that the range of differences in values T_D when changing the above factors is negligible and is close to 5.7% for (m / α) and 6.6 % for (T_D / α).

Based on the performed calculations, which were carried out with the help of a package of statistical applications for processing and analysis of the obtained results of experimental research for PC, the threedimensional spatial dependences of time response surfaces T of transportation of the piece goods in the warehouse and their two-dimensional cross-sections for a visual representation of laboratory tests have been constructed.

The conducted multifactor experiment had the following limits of variable parameters: mass m artificial cargo together with the mass of the frame structure: $35 \le m \le 95$ (kg); the angle of inclination α pairs of parallel ropes to the horizon: $10 \le \alpha \le 15$ (°); the force of tension T_D of ropes $2400 \le T_D \le 5000$ (N) (Figure 15).

In the study of the influence of two variable parameters on the value of the transportation time T of the piece goods to the third party was given a fixed average value with the appropriate values: $m = 65$ (kg); $\alpha = 15$ (°); $T_D = 5000$ (N). It is established that the dominant influence on the minimum time T has angle α, then mass m and the force of tension T_D of ropes. However, it should be noted that the range of differences in the values t when changing the above factors is insignificant and is close to 5.7% for (m / α) and 6.6 % for (T_D / α).

4 Conclusion

The developed automobile cable systems for loading and unloading operations between the vehicle-support and vehicle-vehicle points are presented in this paper. In the developed cable system, the rotation of the system is limited by locking devices in the unloading area, which eliminates the excess speed in cases where it is not eliminated by spring-loaded rollers. It has been established that the residual speed was observed for heavy loads (more than 50 kg) and was no more than 10% (less than 0.2 m/s) of the suspension speed at the end of the track. The analysis of the obtained theoretical results made it possible to establish that an increase in speed and, accordingly, a decrease in transportation time, occurs with a decrease in the rolling resistance, which is achieved by an increase in mounting tension, which leads to a reduction in the roller coverage angle ξ with a rope and, accordingly, reducing the coefficient of friction λ_N. In addition, transporting a load with a greater weight leads to an increase in the difference in sags $\Delta f(x_{dow}) - \Delta f(x_{unl})$, which also reduces the transportation time. It has been established that for a change of time T in terms of the force T_D within the limits $2400 \le T_D \le 5000$ (N), and angle α within the limits $10 \le \alpha \le 15$ (°) at mass: $m = 65$ (kg), the dominant factor is the influence of the angle α. Thus, for $T_D = 2400$ (N) the increase of angle α from 10° to 15° leads to a decrease in value T from 6.69 to 5.58 s. For $T_D = 5000$ (N) the increase of angle α from 10° to 15° leads to a decrease in value T from 6.46 to 5.39 s. Accordingly, the shorter the transportation time T, the higher the performance of the transportation system. It was established that the change in time T due to the angle of inclination α within the limits $10 \le \alpha \le 15$ (°) and mass $35 \le m \le 95$ (kg); with a stable tension force T_D ropes $T_D = 3700$ (N), the dominant factor is the influence of the angle α. Thus, for $T_D = 2400$ (N) the increase of angle α from 10° to 15° leads to a decrease in value T from 6.69 to 5.58 s. For $T_D = 5000$ (N) the increase of angle α from 10° to 15° leads to a decrease in value t from 6.46 to 5.39 s. It has been established that for the change of time T in terms of the tension force T_D of ropes within the limits $2400 \le T_D \le 5000$ (N), and mass m within the limits $35 \le m \le 95$ (kg), at a stable angle of inclination α to the horizon $\alpha = 12.5$ (°) the dominant factor influencing the change in T value is the influence of the mass of artificial loads m. Thus, for $T_D = 2400$ (N) an increase in mass m from 35 to 95 kg leads to a decrease in the value T from 6.32 to 5.89 s. For $T_D = 5000$ (N) an increase in mass m from 35 to 95 kg leads to a decrease in the value T from 6,11 to 5.68 s.

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Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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