Наведено результати досліджень, присвячених вивченню закономірностей протікання динамічних процесів і виникнення динамічних навантажень при русі візка кабельного крана з підвішеним вантажем. Дані закономірності в подальшому можуть бути враховані при розрахунку реальних кранів для підвищення їх надійності та довговічності, недопущення несприятливих явищ при русі вантажного візка, а також для визначення параметрів кранів нової конструкції. Динаміка кабельного крана розглядається з точки зору взаємодії елементів системи «візок-вантаж-несучий канат». Удосконалено математичну модель системи «візок-вантаж-несучий канат» шляхом введення трьох демпфуючих коефіцієнтів, кожен з яких характеризує розсіювання енергії при різних фізичних процесах – рух візка, вантажу та швидкості вітрового навантаження. Для чисельного моделювання використаний програмний комплекс KiDyM, який на аналітичному рівні дозволяє будувати рівняння руху систем, що описуються сукупністю звичайних диференціальних рівнянь. Отримано закономірності зміни нормальної і дотичної сил інерції, що мають місце при русі візка по криволінійній траєкторії. Оцінені їх характер і величина. Визначено динамічні характеристики системи з урахуванням впливу мас вантажу що розгойдується, візка і кривизни каната. Розглянуто аварійний режим, що виникає при обриві тягового каната і вплив вітрового навантаження на розгойдування вантажу. Визначено причини виникнення реверсу швидкості вантажного візка і шляхи його усунення. Досліджено вплив вітрового навантаження на кут відхилення вантажу від вертикалі

Ключові слова: кабельний кран, візок крана, несучий канат, динаміка крана, чисельне моделювання

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NUMERICAL SIMULATION OF THE DYNAMICS OF THE SYSTEM "TROLLEY – LOAD – CARRYING ROPE" IN A CABLE CRANE

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1. Introduction

The distinctive feature of cable cranes are large spans and high speed of movement of freight trolleys, which ensures their required performance. In addition, to improve reliability and durability, it is necessary to take into consideration additional dynamic loads that emerge in separate nodes of the structure. Moreover, at present there is a strong tendency to a more sophisticated design of cable cranes, hence the need to employ the existing, and to develop new, approximation methods for both analytical and numerical analyses.

2. Literature review and problem statement

Given the above, we shall analyze existing mathematical models of the system “trolley – load on a flexible suspension”, which will allow us to determine the one that most closely matches the requirements of performance, reliability durability, and accounting for the design complexity.

When investigating the dynamics of load-lifting machines, the model “trolley – load on a flexible suspension” is traditionally applied. Paper [1] describes the model “trolley – load on a flexible suspension” by a system of differential equations, in which a rope deviation from the vertical is accepted as the load displacement and a speed direction is taken into consideration when determining a resistance to the trolley displacement. However, the authors assumed the linear motion of a load, so the model does not account for the vertical displacement at swinging. Paper [2] presented a nonlinear model of the crane trolley. The equations are constructed in the independent generalized coordinates: a trolley motion and a load deviation angle. Control parameter is the driving force, which is applied through the rope of a traction winch. A similar model is used in [3], which also employs as the second generalized coordinate a rope deviation angle from the vertical. Authors of paper [4] derived, based on the Lagrangian equations, a mathematical model of the pendulum with a variable length of the load suspension aimed to simulate simultaneous lifting and horizontal displacement of the load.

If the mass of the suspended load is very much distributed for height, the model of a trolley with a double pendulum is considered [5].

There are known models that take into consideration the geometrical features of the suspension point of the load and trajectories of its movement. Worth noting is a model for the transportation of a long load between two bridge cranes, which are at different levels of height and work together [6]. Authors of paper [7] obtained, based on the linearized Lagrangian equations, a model that describes oscillations of a spherical pendulum.

Damping properties of a cable crane elements must be taken into account because they have a significant quantitative and qualitative impact on the character of change in the generalized coordinates and velocities. In addition, a cable crane's trolley model in all cited sources should be improved by integrating the vertical deflection of the rope along which it moves.

Paper [8] examines oscillatory processes in the rope along which a freight trolley moves. The focus is on the task on determining the deflection and tension of the rope. In this case, the rope is described by the equation of a chain line; losses for the internal friction in the rope are also accounted for. However, a trolley in the estimated model is represented in the form of a concentrated mass, which means that the impact of the trolley on a rope is not taken into consideration in full.

Calculation of the shape of the sagging of a rope that has a linear weight and rigidly fixed ends is addressed in study [9]. Determining the geometrical characteristics of the rope excluded the trolley and the load.

Paper [10] analyses the dynamics of a cable system under the modes of load lifting and trolley movement along the carrying rope. In this case, using the Lagrange equations, the authors defined the character of oscillations of the carrying and traction ropes. Dissipative function, however, does not take into consideration the influence of wind and fluctuations of the suspended load.

Work [11] investigated the dynamics of a cable crane with a varying height of supports; the trolley with a suspended load moved under the force of gravity. The principle of crane operation is similar to the design described in [12]. A mathematical model consisted of two ordinary differential equations and made it possible to determine the vertical and horizontal position of the load's point of suspension. The authors took into consideration elastic properties of the rope. They performed numerical simulation of the trolley motion with a suspended load based on the constructed mathematical model and applied parameters of an actual crane. However, a given example also did not take into consideration either friction forces or damping.

In [13], authors built a mathematical model that allows taking into consideration the curvature of a carrying rope, as well as the forces of resistance to motion in the presence of friction and wind. In this case, the authors chose a single coefficient for the equivalent viscous friction that relates to the trolley displacement speed, to the load angular velocity, and to the speed of a wind load. Given different physical nature of the enumerated phenomena, such a choice appears too general.

Thus, the existing mathematical models pay most attention on either the oscillatory processes in carrying and traction ropes, or the characteristics of motion of a trolley and a load along a simplified trajectory of the sagging rope. The interaction between elements of the system "trolley – load – carrying rope" remains insufficiently studied. Researchers do not consider the normal and tangential inertial forces occurring at the motion of a trolley along a curved trajectory, nor the impact of masses of a swinging load, trolley, and a curvature of the rope, on the dynamic characteristics of the system. None of the cited papers investigates an emergency mode related to the break of a traction rope; nor the effect of a wind load on the swinging of a load.

3. The aim and objectives of the study

The aim of present research is to study patterns in the progress of dynamic processes and in the occurrence of dynamic loads when a trolley of the cable crane moves with a suspended load. Such patterns could be subsequently taken into consideration when calculating actual cranes, in order to improve their reliability and durability, to avoid unfavorable events during motion of a freight trolley, as well as to define parameters of cranes of the new design.

To accomplish the aim, the following tasks have been set:

– to improve a mathematical model of the system "trolley – load – carrying rope" through better differentiated accounting for the damping properties of the system;

– to determine the patterns in change in the dynamic parameters of the system, to identify unfavorable events and techniques to address them;

– to estimate the character and magnitude of additional dynamic loads when a trolley moves along a curvilinear rope under standard and emergency operating modes;

– to define patterns in the impact of wind load on the swinging of the load.

4. Mathematical model of the system "trolley – load – carrying rope" in a cable crane

The design scheme, shown in Fig. 1, is described by a system of equations (1) [13].

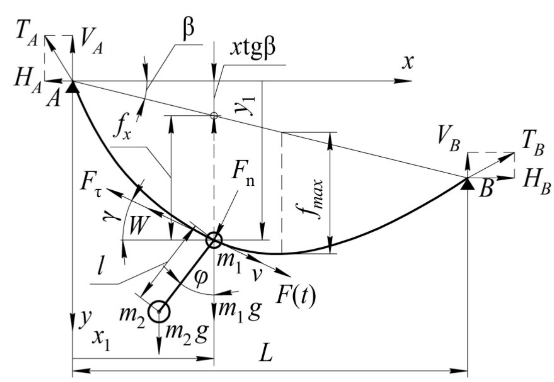

Fig. 1. Design scheme for the motion of a trolley along a carrying rope of the cable crane

Differential equations take the form:

$$\begin{cases}(m_1+m_2)\left(1+\left(\dfrac{dy}{dx}\right)^2\right)\ddot{x}+m_2 l\sqrt{1+\left(\dfrac{dy}{dx}\right)^2}\times\\ \times\left(\ddot{\varphi}\cos(\gamma+\varphi)-\dot{\varphi}^2\sin(\gamma+\varphi)\right)-\\ -(m_1+m_2)g\left(tg\beta+\dfrac{L-2x}{2H}\left(\dfrac{g_k}{\cos\beta}+\dfrac{Q}{L}\right)\right)=\\ =F(t)-Wsign(\dot{x});\\ m_2 l\sqrt{1+\left(\dfrac{dy}{dx}\right)^2}\left(\ddot{x}\cos(\gamma+\varphi)\right)+m_2 l^2\ddot{\varphi}+\\ +\mu_1 l(\dot{x}+l\dot{\varphi}-V)+m_2 gl\sin\varphi=0,\end{cases} \quad (1)$$

where m_1 and m_2 are the masses of a trolley and a load, respectively; x is the horizontal coordinate of the trolley; y is the vertical coordinate of the trolley; μ_1 is the generalized coefficient of resistance to the motion of a trolley and a load; φ is the angle of rope deviation from the vertical; $F(t)$ is a function of the trolley control; l is the length of the load; β is the inclination angle of the straight line connecting supports; W is the force of resistance to the motion of a trolley.

$$W=Q\mu_2\cos\gamma,$$

where Q is the total weight of the trolley and load; μ_2 is the reduced resistance coefficient when a trolley moves along a traction rope; H_x is the horizontal component of the rope tension; g_k is the rope weight per unit length; L is the span of a crane; γ is the inclination angle of the tangent to curve $y(x)$ at the point of trolley position; V is the wind load.

In the reduced system, $y(x)$ is the parabolic trajectory of a trolley whose equation takes the form [14]:

$$y(x)=x\,\mathrm{tg}\beta+\frac{x\,(L-x)}{2H_x}\left(\frac{g_k}{\cos\beta}+\frac{Q}{L}\right).$$

Because the physical processes that determine the damping properties of the system related to the movement of a trolley, a load and the speed of a wind load are of different nature, it is wrong to describe these processes with a single coefficient of equivalent viscous friction μ_1. It is therefore proposed to improve system (1) by introducing three separate coefficients μ_{11}, μ_{12}, μ_{13} to the second equation (2).

$$m_2 l\sqrt{1+\left(\frac{dy}{dx}\right)^2}\left(\ddot{x}\cos(\gamma+\varphi)\right)+m_2 l^2\ddot{\varphi}+$$
$$+l(\mu_{11}\dot{x}+\mu_{12}l\dot{\varphi}-\mu_{13}V)+m_2 gl\sin\varphi=0. \quad (2)$$

A system of equations (1) takes the form:

$$\begin{cases}(m_1+m_2)\left(1+\left(\dfrac{dy}{dx}\right)^2\right)\ddot{x}+m_2 l\sqrt{1+\left(\dfrac{dy}{dx}\right)^2}\times\\ \times\left(\ddot{\varphi}\cos(\gamma+\varphi)-\dot{\varphi}^2\sin(\gamma+\varphi)\right)-\\ -(m_1+m_2)g\left(tg\beta+\dfrac{L-2x}{2H}\left(\dfrac{g_k}{\cos\beta}+\dfrac{Q}{L}\right)\right)=\\ =F(t)-Wsign(\dot{x});\\ m_2 l\sqrt{1+\left(\dfrac{dy}{dx}\right)^2}\left(\ddot{x}\cos(\gamma+\varphi)\right)+m_2 l^2\ddot{\varphi}+\\ +l(\mu_{11}\dot{x}+\mu_{12}l\dot{\varphi}-\mu_{13}V)+m_2 gl\sin\varphi=0.\end{cases} \quad (3)$$

Numerical analysis of the dynamics of a cable crane requires the integration of a system of equations (3). To this end, we use the Runge-Kutta-Merson method, which, along with a system of computer algebra, is built into the software package KiDyM [15]. The system of computer algebra from the package KiDyM, based on the application of an apparatus of structural matrices, at the analytical level enables the construction of differential equations of motion of mechanical systems.

System (3) in the form of Cauchy takes the form:

$$\begin{cases}\dot{x}=\nu_x;\\ \dot{\nu}_x=(a_{22}b_1-a_{12}b_2)/\Delta;\\ \dot{\varphi}=\omega;\\ \dot{\omega}=(-a_{21}b_1+a_{11}b_2)/\Delta,\end{cases} \quad (4)$$

where

$$a_{11}=(m_1+m_2)(1+\alpha^2);$$

$$a_{12}=a_{21}=m_2 l\sqrt{1+\alpha^2}\cdot\cos\varepsilon;$$

$$a_{22}=m_2 l^2;$$

$$b_1=m_2 l\sqrt{1+\alpha^2}\sin\varepsilon\cdot\dot{\varphi}^2+(m_1+m_2)\times$$
$$\times g\left[tg\beta+\frac{L-2x}{2H}\left(\frac{g_k}{\cos\beta}+\frac{Q}{L}\right)\right]+F(t)-Wsign(\dot{x});$$

$$b_2=-\mu_1 l(\dot{x}+l\dot{\varphi}-V)-m_2 gl\sin\varphi;$$

$$\varepsilon=\gamma+\varphi;\quad \alpha=\frac{dy}{dx}.$$

5. Results of numerical simulation

To perform numerical calculations, we accepted parameters of a full-scale crane designed at the Institute "Soyuzprommekhanizaciya", Kharkiv, Ukraine (Fig. 2). This allowed us to compare simulation results with the design calculations of an actual crane and quantify the magnitude of a dynamic addition.

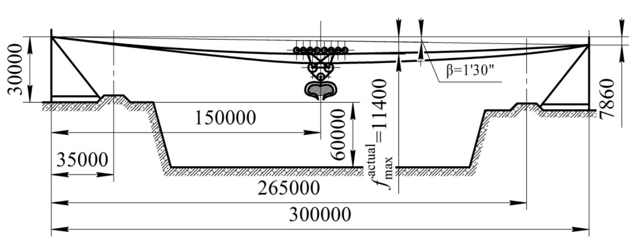

Fig. 2. Dimensional drawing of an actual cable crane used for modeling

We made several assumptions while modeling, which could be disregarded in further research with a more complex problem statement:

1) elastic properties of the carrying, freight, and traction ropes shall not be taken into consideration;

2) inertial properties of the traction and freight ropes shall not be taken into consideration;

3) a distance between outer wheels of the trolley is considered small compared with the magnitude of the span;

4) a freight trolley moves along a parabolic trajectory;

5) wind load is ignored.

Parameters of an actual crane, applied in mathematical model (2), accept the following values:

$$m_1 = 2900 \text{ kg}; \; m_2 = 20000 \text{ kg};$$

$$Q = 224649 \text{ H}; \; L = 300 \text{ m}; \; l = 1 - 5 \text{ m};$$

$$g_k = 277 \text{ N/m}; \; H_x = 1500000 \text{ N};$$

$$\mu_{11} = 0{,}0021 \text{ kg/s}; \; \mu_{12} = 0{,}0005 \text{ kg/s}; \; \mu_{13} = 0{,}0026 \text{ kg/s};$$

$$\mu_2 = 0{,}02; \; \beta = 1{,}5°; \; W = Q \cdot \mu_2 \cdot \cos\gamma.$$

Initial speed of the trolley is $v_0 = 0{,}001$ m/s.

The trolley moves under the action of control influence that changes according to law:

$$F(t) = -0{,}5 \cdot Q \cdot \text{abs}(\sin\gamma),$$

and the force of its natural weight. The form of control function is chosen with respect to the required motion speed limit of the trolley in accordance with actual operational values.

Integration time is taken to equal 48 seconds over which a loaded trolley has passed the lowest trajectory point and stops at the point with coordinate $x = 0{,}8L$. Fig. 3 shows diagrams of change in x and φ over time.

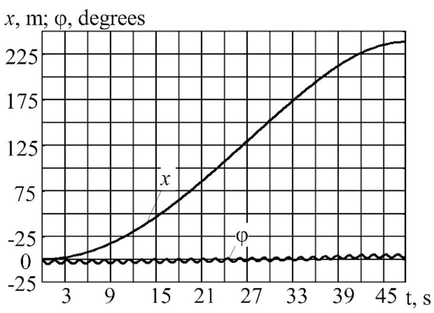

Fig. 3. The law of change in the displacement of trolley x across the span and in angle of the load deviation from the vertical φ over time t

A chart of change in the x coordinate of a trolley takes an S-shaped form, indicating the motion of a trolley along the descending and ascending branches of the trajectory defined by a carrying rope. The mean component of a change in coordinate φ alternates negative and positive signs.

The charts of change in the trolley speed v and in the angular velocity of load ω over time are shown in Fig. 4. The maximum motion speed is about 8 m/s, which is within the range of operating speeds of cable cranes. Because the cable crane's trolley weight is typically less than the weight of a transported load, its oscillations exert a significant impact on the law of change in the trolley speed. Fig. 4 shows that the frequency of a speed change of the trolley coincides with the frequency of load oscillations; these oscillations are in counter phase. The minimum range of the trolley speed fluctuations occurs along the downward section of motion trajectory and is 0.7 m/s. It increases subsequently to 1.1 m/s at a moment when the lowest point of the trajectory is being passed and reaches a maximum value of 1.6 m/s along the ascending section of the trajectory.

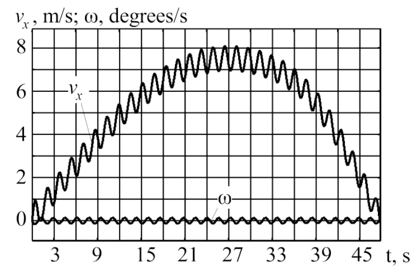

Fig. 4. The law of change in the trolley displacement speed v_x across the span and in the load angular velocity ω over time t

Fig. 5 shows charts of the trolley displacement and the angle of load deviation at the initial time. We clearly observe the counter-phase motion of the trolley and load. The charts demonstrate that at this stage, as a consequence of the load swinging, there is a possibility that the trolley can change its motion direction.

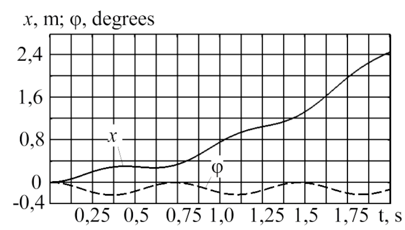

Fig. 5. The law of change in the trolley displacement across the span x and in the angle of load deviation from the vertical φ at the initial stage of motion

Fig. 6 shows that there is a non-zero period of time over which a projection of the trolley speed onto the "x" axis accepts a negative value (that is, a reverse). This is undesirable, both from the point of control tasks, as it is associated with an increase in the time required to travel the required path, and in terms of the emergence of additional dynamic loads. A study was conducted, which revealed that it is impossible to avoid the reverse of the trolley at the initial stage of its motion by changing the length of the rope and damping coefficients μ_{11}, μ_{12}, μ_{13} over the entire permissible range of values.

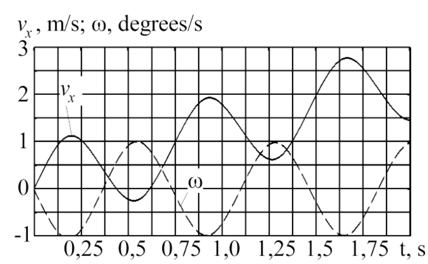

Fig. 6. The law of change in the trolley displacement speed across the span v_x and in the angular velocity of load deviation from the vertical ω at the initial stage of motion

However, at a value of the initial trolley speed equal to 0.2 m/s or at a short release of rope tension to $7.5 \cdot 10^5$ N, the projection of trolley speed onto the x axis remains positive. Results are shown in Fig. 7, 8.

The trajectory of trolley motion shown in Fig. 9 was derived from the integration of equations (1); its shape coincides with the shape of a theoretical parabola. Fig. 7 shows the part of the parabola along which we simulate the motion of a trolley with a resulting coordinate of $x = 0{,}8L = 240$ m. The largest sagging is equal to $f_{\max} = 12{,}2$ m, which is comparable with the value of $f_{\max}^{\text{actual}} = 11{,}4$ m for an actual crane during static estimation. In this case, throughout the entire length of the span $x = L = 300$ m, a vertical displacement of the lower support is $y_{\text{displ}} = 7{,}86$ m.

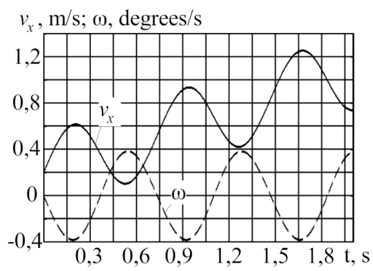

Fig. 7. The law of change in the trolley motion speed projection across the span v_x and in the angular velocity ω at the initial stage of motion with an initial speed of v_x=0.2 m/s

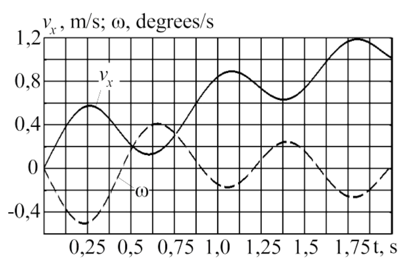

Fig. 8. The law of change in the speed of trolley displacement across the span v_x and in the angular velocity ω at the initial stage of motion at a short release of the carrying rope tension H_x=7.5·10^5 N

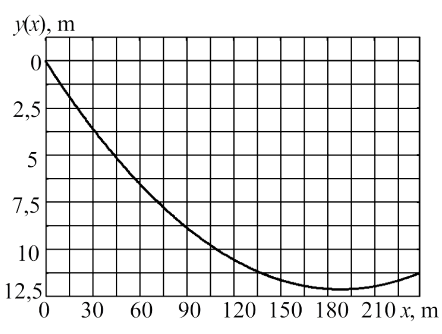

Fig. 9. Calculated trajectory of the trolley motion $y(x)$

When modeling, we also determined additional dynamic loads exerted on the carrying. freight, and traction ropes, caused by the presence of a tangential: $F_\tau^u = m_1 \cdot a_\tau$ ($a_\tau = \dot{v}/\cos\gamma$) and centrifugal forces of inertia for the trolley and load:

$$F_n^u = (m_1 + m_2) \cdot a_n,$$

where

$$a_n = \frac{(v/\cos\gamma)^2}{\rho}; \quad \rho = \frac{\sqrt{\left(1 + y_x'^2\right)^3}}{\left|y'_{xx}\right|};$$

$$y'_x = tg\beta + \frac{L - 2x}{2H}\left(\frac{g_k}{\cos\beta} + \frac{Q}{L}\right); \quad y''_{xx} = -\frac{1}{H}\left(\frac{g_k}{\cos\beta} + \frac{Q}{L}\right).$$

In this case, the formula that defines the radius of a curvature of function $y(x)$ is precise. We constructed charts of change in the tangent and centrifugal forces of inertia over time when a trolley moves along a carrying rope; they are shown in Fig. 10, 11.

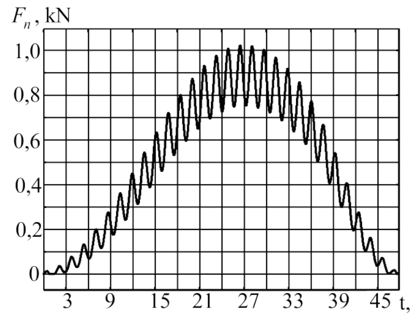

Fig. 10. The law of change in the normal dynamic load

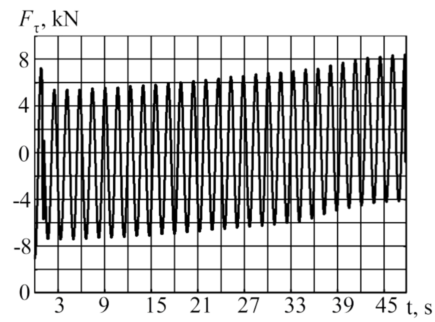

Fig. 11. The law of change in the tangent dynamic load

It should be noted that the additional normal load is mostly accepted by a carrying rope and partially by a freight rope. At the same time, a tangential load mainly falls on the traction rope and partially on the carrying rope.

In addition, we estimated the case when a traction winch rope breaks, which under actual operating conditions is regarded to constitute the emergency mode. The charts of loads corresponding to a given regime are shown in Fig. 12, 13.

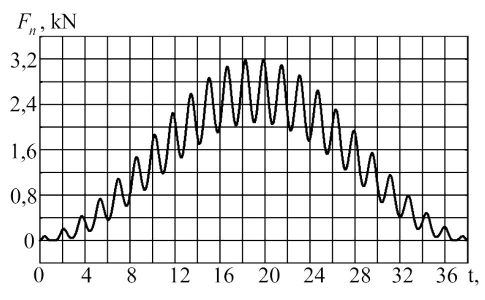

Fig. 12. Change in the normal dynamic loading (break of a traction winch rope)

Comparison of charts shown in Fig. 10–13 reveals that both the tangent and normal loads increased by not more than three times. However, the order of numbers indicates that at an increase in the operating speeds a given dynamic load may exert a significant impact on the behavior of the system "trolley-load".

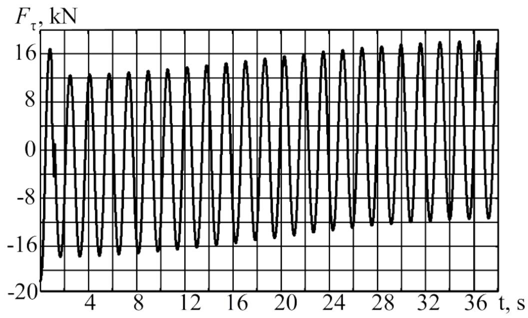

Fig. 13. Change in the tangent dynamic loading (break of a traction winch rope)

The charts for the normal inertia forces shown in Fig. 10, 12, which correspond to the standard and emergency modes, the maximum values of swings are reached at the lowest point of the trolley trajectory and make up 0.3 kN and 1.2 kN, respectively. The percentage ratio of a swing to the mean component is equal to 25–30 % for a standard mode and to 50 % for an emergency mode.

The mean component of the tangential inertial force at the onset of motion is, under a standard mode, −2 kN, and under emergency mode, –4 kN. At the end of the motion: for a standard mode, +2 kN; for emergency mode, +4 kN. Changing a sign of the mean component attests, similar to the above, to the trolley motion along a descending and ascending branch of the trajectory.

We have derived dependences of the angle of load deviation on the speed of a wind load, shown in Fig. 14, 15. The charts demonstrate that at a disturbing frequency of the wind load equal to 0.16 Hz, the swing of load oscillations does not exceed 7 degrees. However, if the frequency of a disturbing wind load turns out to be close to the frequency of free oscillations of the load, a phenomenon of resonance may occur.

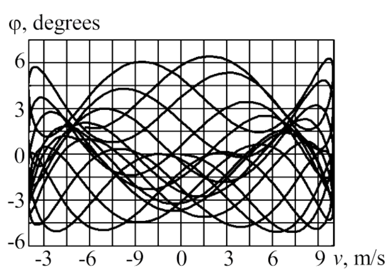

Fig. 14. Dependence of the load deviation angle on the speed of a bilateral wind, which varies from 0 to 10 m/s at frequency 0.16 Hz

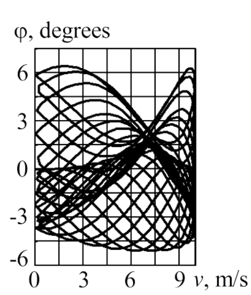

Fig. 15. Dependence of the load deviation angle on speed of an oncoming wind, which changes in the range from 0 to 10 m/s at frequency 0.16 Hz

6. Discussion of results of studying the dynamics of the system "trolley – load – carrying rope" in a cable crane

An *S*-shaped curve of the trolley displacement along a carrying rope is predetermined by different inclination angles of the tangents to it at the initial, middle, and final points of the trajectory. Inclination angles of the tangents at the initial and final points of the trajectory are equal to zero, and at the middle point the tangent's inclination angle is maximum, which corresponds to the maximum speed of the trolley. The model that describes the system "trolley – load – carrying rope" is a dual-mass system. Such a system has one zero frequency, which corresponds to the movement of the system "trolley – load" as an entire whole, and one non-zero frequency at which the oscillations of the trolley and load are in antiphase; in this case, the amplitudes of oscillations are inversely proportional to the masses. Given this, at low speeds of the trolley, with a weight less than that of the load, the reverse phenomenon may occur. It can be avoided by both increasing the initial speed of the trolley and increasing the speed of the trolley through a release in the tension of the carrying rope at moment in time immediately preceding the reverse.

The maximum span of the normal dynamic load at the lowest point of the trajectory is due on the one hand to the minimum radius of a curvature at that point and, on the other hand, to the maximum change in the amplitude of trolley speed oscillations. The maximum amplitude of the tangential dynamic load at the onset of trolley motion is predetermined by the presence of the reverse.

The angle of load deviation from the vertical in both cases for the examined wind load does not exceed 7 degrees due to a low speed of the wind load and the remoteness of its disturbing frequency from the frequency of free oscillations of the system "trolley-load".

The benefit of a given numerical analysis is the application of the programming package KiDyM, which, by using the built-in system of computer algebra and by employing an apparatus of structural matrices, allows at the analytical level the construction of motion equations for the systems that are described by a combination of ordinary differential equations.

In contrast to papers cited in our review of the scientific literature in which the dynamics of a cable crane is considered either in terms of the oscillatory processes in ropes or the motion characteristics of a trolley and a load, this study examines the interaction between elements in the system "trolley – load – carrying rope". We established patterns in a change of normal and tangential inertial forces occurring when a trolley moves along a curvilinear trajectory. We defined dynamic characteristics of the system taking into consideration the influence of the masses of a swinging load, trolley, and the curvature of a rope. We investigated an emergency mode that occur at a break of the traction rope, as well as the influence of a wind loading on the swinging of the load.

The shortcoming of present research includes the assumptions described above, reducing the number of which could be considered a direction for the further research. The difficulties associated with further research imply that the examined system "trolley-load-carrying rope" is essentially non-linear; that almost eliminates the possibility of applying purely analytical analysis methods.

7. Conclusions

We have improved a mathematical model of the system "trolley – load – carrying rope" by introducing three damping coefficients, each of which characterizes energy dissipation under different physical processes – the motion of a trolley, a load, and the speed of a wind load.

We have defined time-dependent patterns of change in the position of the trolley in the span and in the angle of load deviation, as well as their derivatives. During modeling, we determined dynamic loads on the carrying, freight, and traction ropes, caused by the presence of the tangential and centrifugal inertia forces of the trolley and load. Their character and magnitude were quantified. Under a standard mode, a span of the normal load oscillations at the lowest point of the trolley trajectory is 25–30 % of the mean component, and under emergency mode, it is 50 %. As far as the tangential load is concerned, the ratio of span in its dynamic component to the mean value under a normal mode of operation of a cable crane is equal to 10; it increases to 25–30 at a break of the winch rope. That should be taken into consideration when predicting the service life of the carrying rope.

A wind load impact is negligible if its frequency is far from the frequency of free oscillations of the system "trolley-load-carrying rope" and wind speed does not exceed 10 m/s. In this case, the angle of load deviation will not exceed 6.5 degrees.

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