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THE DYNAMIC ANALYSIS OF THE JOINT TROLLEY MOVEMENT AND HOISTING MECHANISM IN THE TOWER CRANE

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The dynamic model of joint movement of mechanisms has been developed, which takes into account the main movement of drive mechanisms and oscillating movement of the load on a flexible suspension and links of mechanisms with elastic properties. A mathematical model of the motion of mechanisms is constructed on the basis of a dynamic model. For a specific jib system of the tower crane, dynamic calculations were performed. According to the results of the calculations, a dynamic analysis of the joint trolley movement and hoisting mechanism was carried out. The analysis revealed significant dynamic and energy overloads of mechanisms during transients (start, braking) and the presence of high-frequency oscillations in the links with elastic properties and low-frequency oscillations of the load on a flexible suspension.
Keywords: dynamic, analysis, tower crane, joint movement, hoisting mechanism, hook blocks, load, optimization.

Introduction

In real tower cranes operation conditions for the purpose of productivity increase of performance of load operations the combination of simultaneous use of several mechanisms is carried out. So, for example, simultaneous use of trolley and hoisting mechanism, and also other mechanisms can be used. However, at the same time, especially when performing transients (start, braking), the elements of the drive mechanisms and the design of the crane increase the dynamic loads, which reduce the reliability of the crane as a whole and its individual mechanisms.

Therefore, there is a need for research on the dynamics of motion in the joint work of mechanisms. The most common are the combination of the lifting mechanism with other mechanisms of the crane (changes of departure, rotation and movement).

In this regard, we will conduct a dynamic analysis of the joint operation of the trolley and hoisting mechanism, as during the operation of these mechanisms there are significant high-frequency oscillations of the drive elements and low-frequency oscillations of the load on the flexible suspension.

Thus, the task of establishing dynamic loads during the operation of the trolley and hoisting mechanism is relevant, as it reflects the real operating conditions of tower cranes.

Analysis of publications

Most of the authors have investigated the dynamic, kinematic, moving and hoisting mechanisms, energetic, electric processes, durability, and vibration, which take place in the different types of cranes and hoisting machines including tower cranes [1-8].

The scientists [9] created a mathematical movement model of the cart with loaded block pulley in hoisting machines and simulated a dynamic pulley block model using software and concluded that by neglecting the mass variation and the rope rigidity of the block, could simplify the dynamics of the modeling block pulley in hoisting cranes.

The purpose of the paper [10-11] is a dynamic identification of the tower crane`s laboratory model and the development of the controller of the cart position along the jib, along with, the damping payload oscillations in the *x*-line.

In the scientific paper [12] the dynamics of the joint movement of both the crane slewing and cart moving mechanisms are investigated. Research objects overloads of exceeding admissible values are revealed as well as a load of spatial fluctuations.

With the object of cutting the oscillations of the load on the flexible suspension, some optimization tasks have been solved [13-14]. The investigators used a complex dynamic integral criterion for solving a problem of cutting load oscillations during the operation of the tower crane slewing mechanism.

In the paper [14] a particle swarm method for numerically solving optimization problems was used. The authors make a mathematical model that solves by method of time optimal control, and can be integrated analytically.

The scientific article [15] describes the methodology of laboratory research of samples of drive systems with energy-saving permanent-magnet electric motors, for modern construction cranes operation. The elements of the crane drive system were also experimentally checked and their compliance with the current standards was confirmed. And to do this necessary software that simulated the control the dynamometer as well as the load per crane was developed.

The researches [16] have developed a mathematical model to simulate the tower crane operation in real time, which allows intensifying and speeding up engineering construction operations, as well as simplifying project planning.

This model was presented in the form of two subsystems: kinematic and dynamic model.

The author [17-18] emphasizes the need for a comprehensive approach to the optimization of multi-drive crane systems and focus on reducing vibration, minimizing static and dynamic loads, energy consumption.

Purpose of the paper

The aim of the work is to determine and analyse the dynamic loads that occur in the elements of the tower crane during the joint trolley movement and hoisting mechanism in tower crane.

Research results

The jib system of the tower crane (Fig. 1) will be presented as a holonomic mechanical system consisting of absolutely rigid bodies of trolley and hoisting mechanism, except for a traction rope of movement of the cart and a load rope which have elastic properties and a flexible suspension of load in the plane of change of tower crane flight.

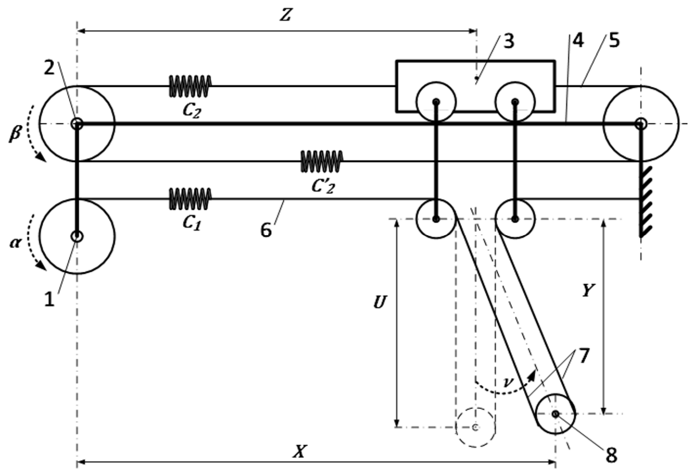

Fig. 1. Dynamic model of trolley and hoisting mechanism

All drive elements of the lifting mechanism are reduced to the axis of the drive drum 1, and the mechanism of change of tower crane flight to the drum 2. The trolley 3 moves on the beam jib 4 by means of a traction rope 5 with stiffness c_2 or c_2' depending on the direction of the truck movement. The load rope 6 with the stiffness coefficient c_1 forms a pulley system 7, to which the load 8 is suspended.

Thus, the presented dynamic model of the jib system with the joint movement of the mechanisms of change of tower crane flight and load lifting has 5 degrees of freedom. The generalized coordinates of this model are the angular coordinates of the drive drums according to the lifting mechanism α and change of of tower crane flight β, as well as the deviation of the load hoist from the vertical ν and the linear coordinates of the centers of mass of the truck z and load u.

The system is driven by the driving moments of the drive mechanisms for lifting the load M_1 and change of tower crane flight M_2, as well as the force of resistance of the trolley W.

To compile the differential equations of motion of the presented dynamic model of the joint motion of the trolley and hoisting mechanism, we use the Lagrange equations of the second kind, which have the form:

$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial T}{\partial \dot{\alpha}}-\frac{\partial T}{\partial \alpha}=M_1-\frac{\partial \Pi}{\partial \alpha}-\frac{\partial R}{\partial \dot{\alpha}},$$
$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial T}{\partial \dot{\beta}}-\frac{\partial T}{\partial \beta}=M_2-\frac{\partial \Pi}{\partial \beta}-\frac{\partial R}{\partial \dot{\beta}},$$
$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial T}{\partial \dot{z}}-\frac{\partial T}{\partial z}=-W-\frac{\partial \Pi}{\partial z}-\frac{\partial R}{\partial \dot{z}},$$
$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial T}{\partial \dot{u}}-\frac{\partial T}{\partial u}=-\frac{\partial \Pi}{\partial u}-\frac{\partial R}{\partial \dot{u}},$$
$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial T}{\partial \dot{v}}-\frac{\partial T}{\partial v}=-\frac{\partial \Pi}{\partial v}-\frac{\partial R}{\partial \dot{v}}, \tag{1}$$

where T, Π – respectively the kinetic and potential energy of the system; M_1, M_2 – the driving moments are reduced to the drive drums according to the mechanisms of load lifting and change of tower crane flight; t – time change coordinate; R – dissipative Rayleigh function; W – resistive force of the movement of the cart.

Determine the kinetic energy of the system

$$T=\frac{1}{2}I_1\dot{\alpha}^2+\frac{1}{2}I_2\dot{\beta}^2+\frac{1}{2}m_3\dot{z}^2+\frac{1}{2}m(\dot{x}^2+\dot{y}^2), \tag{2}$$

where I_1, I_2 – reduced to the axes of the drive drums moments of inertia of the drives, respectively, the mechanisms of lifting and change of tower crane flight; m_3, m – the mass of the cart and the load, respectively; x, y – coordinates of the center of mass of the load.

Find the function of the potential energy of the system

$$\Pi=\frac{1}{2}C_1(\alpha r_1-nu)^2+\frac{1}{2}C_2(\beta r_2-z)^2-mgu\cos v, \tag{3}$$

where r_1, r_2 – radii of drive drums according to mechanisms of lifting of load and change of departure; n – the multiplicity of the hoist of the lifting mechanism; g – free fall acceleration.

The dissipative function of Rayleigh is determined by the following dependence

$$R=\frac{1}{2}b_1(\dot{\alpha}r_1-n\dot{u})^2+\frac{1}{2}b_2(\dot{\beta}r_2-\dot{z})^2+\frac{1}{2}b\dot{v}^2, \tag{4}$$

where b_1, b_2, b – dissipation coefficients of elastic elements of load and traction ropes, respectively, as well as flexible load suspension.

Find the coordinates of the centers of mass of the load and their time derivatives:

$$x = z + u\sin v, \;\; y = u\cos v, \tag{5}$$

$$\dot{x} = \dot{z} + \dot{u}\sin v + \dot{v}u\cos v, \;\; \dot{y} = \dot{u}\cos v - \dot{v}u\sin v. \tag{6}$$

Find the derivatives of the expressions (2), …, (4), necessary for the system of equations (1):

$$\frac{\partial T}{\partial \alpha} = \frac{\partial T}{\partial \beta} = \frac{\partial T}{\partial z} = 0, \;\; \frac{\partial T}{\partial u} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial u} + \dot{y}\frac{\partial \dot{y}}{\partial u}\right), \;\; \frac{\partial T}{\partial v} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial v} + \dot{y}\frac{\partial \dot{y}}{\partial v}\right), \tag{7}$$

$$\frac{\partial T}{\partial \dot{\alpha}} = I_1\dot{\alpha}, \;\; \frac{\partial T}{\partial \dot{\beta}} = I_2\dot{\beta}, \;\; \frac{\partial T}{\partial \dot{z}} = m_3\dot{z} + mx\frac{\partial \dot{x}}{\partial \dot{z}} = m_3\dot{z} + m\dot{x},$$

$$\frac{\partial T}{\partial \dot{u}} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial \dot{u}} + \dot{y}\frac{\partial \dot{y}}{\partial \dot{u}}\right) = m\left(\dot{x}\frac{\partial x}{\partial u} + \dot{y}\frac{\partial y}{\partial u}\right),$$

$$\frac{\partial T}{\partial \dot{v}} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial \dot{v}} + \dot{y}\frac{\partial \dot{y}}{\partial \dot{v}}\right) = m\left(\dot{x}\frac{\partial x}{\partial v} + \dot{y}\frac{\partial y}{\partial v}\right),$$

$$\begin{cases} \dfrac{\mathrm{d}}{\mathrm{d}t}\dfrac{\partial T}{\partial \dot{\alpha}} = I_1\ddot{\alpha}; \;\; \dfrac{\mathrm{d}}{\mathrm{d}t}\dfrac{\partial T}{\partial \dot{\beta}} = I_2\ddot{\beta}; \;\; \dfrac{\mathrm{d}}{\mathrm{d}t}\dfrac{\partial T}{\partial \dot{z}} = m_3\ddot{z} + m\ddot{x}, \\ \dfrac{\mathrm{d}}{\mathrm{d}t}\dfrac{\partial T}{\partial \dot{u}} = m\left(\ddot{x}\dfrac{\partial x}{\partial u} + \dot{x}\dfrac{\partial \dot{x}}{\partial u} + \ddot{y}\dfrac{\partial y}{\partial u} + \dot{y}\dfrac{\partial \dot{y}}{\partial u}\right), \\ \dfrac{\mathrm{d}}{\mathrm{d}t}\dfrac{\partial T}{\partial \dot{v}} = m\left(\ddot{x}\dfrac{\partial x}{\partial v} + \dot{x}\dfrac{\partial \dot{x}}{\partial v} + \ddot{y}\dfrac{\partial y}{\partial v} + \dot{y}\dfrac{\partial \dot{y}}{\partial v}\right), \end{cases} \tag{8}$$

$$\begin{cases} \dfrac{\partial \Pi}{\partial \alpha} = C_1 r_1(\alpha r_1 - nu); \; \dfrac{\partial \Pi}{\partial \beta} = C_2 r_2(\alpha r_2 - z); \; \dfrac{\partial \Pi}{\partial \beta} = -C_1 n(\alpha r_1 - nu) - mg\cos v, \\ \dfrac{\partial \Pi}{\partial \beta} = -C_2(\beta r_2 - z); \; \dfrac{\partial \Pi}{\partial v} = -mgu\sin v, \end{cases} \tag{9}$$

$$\begin{cases} \dfrac{\partial R}{\partial \dot{\alpha}} = b_1 r_1(\dot{\alpha} r - n\dot{u}); \; \dfrac{\partial R}{\partial \dot{\beta}} = b_2 r_2(\dot{\beta} r_2 - \dot{z}); \; \dfrac{\partial R}{\partial \dot{z}} = -b_2(\dot{\beta} r_2 - \dot{z}), \\ \dfrac{\partial R}{\partial \dot{u}} = -bn(\dot{\alpha} r_1 - n\dot{u}); \; \dfrac{\partial R}{\partial \dot{v}} = b\dot{v}. \end{cases} \tag{10}$$

After substituting expressions (7),…, (10) in the system (1) we obtain a system of differential equations of the joint motion of the mechanisms of change of departure on the lifting of the load of the tower crane with a beam jib:

$$\begin{cases} I_1\ddot{\alpha} = M_1 - C_1 r_1(\alpha r_1 - nu) - b_1 r_1(\dot{\alpha} r_1 - nu), \\ I_2\ddot{\beta} = M_2 - C_2 r_2(\beta r_2 - z) - b_2 r_2(\dot{\beta} r_2 - \dot{z}), \\ m_3\ddot{z} + m\ddot{x} = C_2(\beta r_2 - z) - b_2 r_2(\dot{\beta} r_2 - \dot{z}) - W, \\ m\left(\ddot{x}\dfrac{\partial x}{\partial u} + \ddot{y}\dfrac{\partial y}{\partial u}\right) = C_1 n(\alpha r_1 - nu) + mg\cos v + bn(\dot{\alpha} r_1 - n\dot{u}), \\ m\left(\ddot{x}\dfrac{\partial x}{\partial v} + \ddot{y}\dfrac{\partial y}{\partial v}\right) = -mgu\sin v + b\dot{v}. \end{cases} \tag{11}$$

The driving moments on the shafts of the electric motors of the mechanisms of lifting and changing the departure of the load at a power of more than 5 kW can be determined by the following Kloss formulas, taking into account their reduction to the drive drums we have:

$$M_1 = \frac{2M_{cv_1} \cdot u_1 \cdot \eta_1}{\dfrac{1-(\alpha u_1/\omega_{10})}{S_{cs_1}} + \dfrac{S_{cs_1}}{1-(\alpha u_1/\omega_{10})}}, \tag{12}$$

$$M_2 = \frac{2M_{cv_2} \cdot u_2 \cdot \eta_2}{\dfrac{1-(\beta u_2/\omega_{20})}{S_{cs_2}} + \dfrac{S_{cs_2}}{1-(\alpha u_2/\omega_{20})}}. \tag{13}$$

Here M_{cv_1}, M_{cv_2} – critical values of moments of electric motors of mechanisms of rise and change of tower crane flight accordingly; u_1, u_2 – gear ratios of drives according to the mechanisms of lifting and change of tower crane flight; η_1, η_2 – k.k.d. drives of mechanisms of rise and change of tower crane flight accordingly; ω_{10}, ω_{20} – synchronous angular velocities of the rotors of electric motors of lifting and changing mechanisms; S_{cs_1}, S_{cs_2} – critical sliding of electric motors of lifting mechanisms and change of tower crane flight which are defined by the following dependences:

$$S_{cs_1} = \left(1-\frac{\omega_{1n}}{\omega_{10}}\right)\left(\lambda_1+\sqrt{\lambda_1^2-1}\right),\ S_{cs_2} = \left(1-\frac{\omega_{2n}}{\omega_{20}}\right)\left(\lambda_2+\sqrt{\lambda_2^2-1}\right), \tag{14}$$

where ω_{1n}, ω_{2n} – nominal angular speeds of rotors of electric motors of mechanisms of rise on changes of tower crane flight; λ_1, λ_2 – coefficients of overload of electric motors of mechanisms of lifting and changes of toweR crane flight.

Find the derivatives of the coordinates of the center of mass of the load present in the system of equations (11)

$$\begin{cases} \dfrac{\partial x}{\partial u} = \sin v, & \dfrac{\partial y}{\partial u} = \cos v, \\ \dfrac{\partial x}{\partial v} = u\cos v, & \dfrac{\partial y}{\partial v} = -u\sin v, \end{cases} \tag{15}$$

$$\begin{cases} \ddot{x} = \ddot{z} + (\ddot{u} - \dot{v}^2 u)\sin v + (\ddot{v}u + 2\dot{u}\dot{v})\cos v, \\ \ddot{y} = (\ddot{u} - \dot{v}^2 u)\cos v + (\ddot{v}u + 2\dot{u}\dot{v})\sin v. \end{cases} \tag{16}$$

The solution of the system of equations (11) taking into account expressions (12),…, (16) is carried out under the following initial conditions of motion:

$$t=0,\ \alpha=\frac{u_0 n}{r_1},\ \dot{\alpha}=0,\ \beta=\frac{z_0}{r_2},\ \dot{\beta}=0,\ z=z_0,\ \dot{z}=0,\ u=u_0,\ \dot{u}=0,\ v=0,\ \dot{v}=0, \tag{17}$$

where z_0 – the initial coordinates of the position of the cart; u_0 – the initial coordinate of the vertical position of the load.

For the tower crane jib system with parameters: $m = 5000\,\text{kg}$; $m_3 = 300\,\text{kg}$; $I_1 = 50.16\,\text{kg}\cdot\text{m}^2$; $I_2 = 87.48\,\text{kg}\cdot\text{m}^2$; $M_{cv_1} = 2881.0\,\text{N}\cdot\text{m}$; $M_{cv_2} = 145.3\,\text{N}\cdot\text{m}$; $\lambda_1 = 2.9$; $\lambda_2 = 2.4$; $u_1 = 3.02$; $u_2 = 18$; $\eta_1 = 0.8$; $\eta_2 = 0.71$; $r_1 = 0.20\,\text{m}$; $r_2 = 0.15\,\text{m}$; $S_{cs_1} = 0.206$; $S_{cs_2} = 0.389$; $\omega_{1n} = 60.5\,\text{rad/s}$; $\omega_{2n} = 95.8\,\text{rad/s}$; $\omega_{10} = 62.8\,\text{rad/s}$; $\omega_{20} = 104.7\,\text{rad/s}$; $z_0 = 10.0\,\text{m}$; $u_0 = 16.0\,\text{m}$; $n = 2$; $C_1 = 7.24\cdot10^5\,\text{N/m}$; $C_2 = 2.15\cdot10^5\,\text{N/m}$; $b_1 = 6.37\cdot10^4\,\text{N}\cdot\text{s/m}$; $b_2 = 2.8\cdot10^4\,\text{N}\cdot\text{s/m}$; $b = 1.05\cdot10^5\,\text{N}\cdot\text{m}\cdot\text{s/rad}$; $g = 9.81\,\text{m/s}$ the calculation of the developed mathematical model of joint movement of mechanisms of change of tower crane flight and load lifting is carried out.

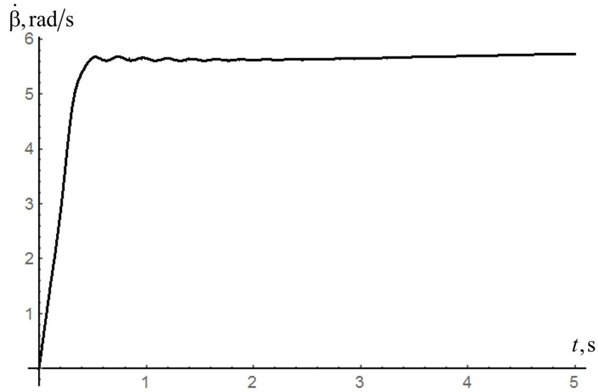

Fig. 2. Graph of the angular velocity of the trolley drive drum

As a result of calculations of the dynamic analysis of the joint movement of mechanisms of change of tower crane flight and load lifting of the tower crane graphic dependences of kinematic (Fig. 2,…, Fig. 5), dynamic (Fig. 6,…, Fig. 9) and power are constructed (Fig. 10 and Fig. 11) characteristics of the jib system.

Figure 2 shows a graph of the angular velocity of the drive drum of the cart, which shows that the drum acquires a steady speed of 5.6 rad/s for 0.5 s. At the same time, small high-frequency oscillations of the angular velocity of the drum are observed in the area of steady motion, which attenuate rather quickly (within 3 s).

However, the cart in the area of steady motion has significant high-frequency oscillations of the speed (Fig. 3), with a maximum value of the amplitude of oscillations of 0.1 m/s, which is 12% of the steady speed. But, these fluctuations also attenuate fairly quickly.

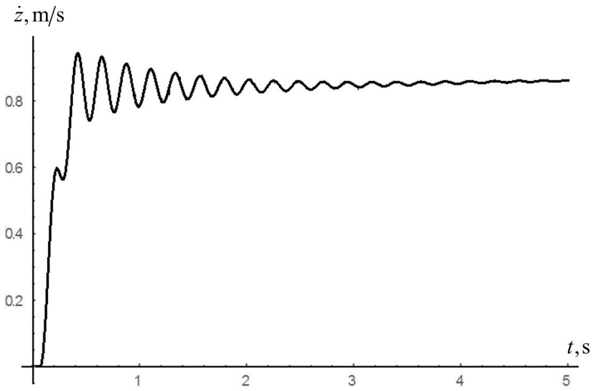

Fig. 3. Graph of the speed of the cart

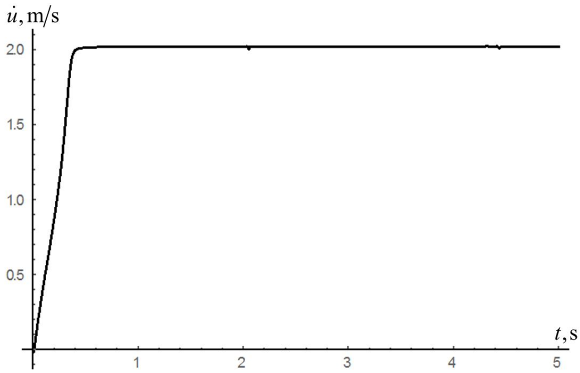

Fig. 4. Chart of speed of load lifting

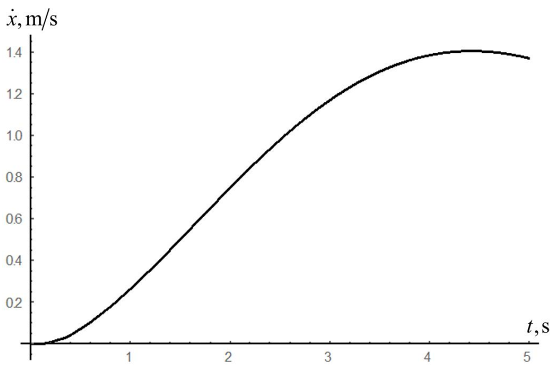

Fig. 5. Graph of the speed of horizontal movement of the load

In the horizontal component of the load speed (Fig. 4) there are low-frequency oscillations, which are caused by pendulum oscillations of the load on a flexible suspension. The maximum value of the horizontal component of the load speed reaches 1.4 m/s, which is 1.7 times higher than the steady speed of the cart. Low-frequency oscillations of the load on the flexible suspension attenuate quite slowly, which must be damped by special means of controlling the movement of crane mechanisms.

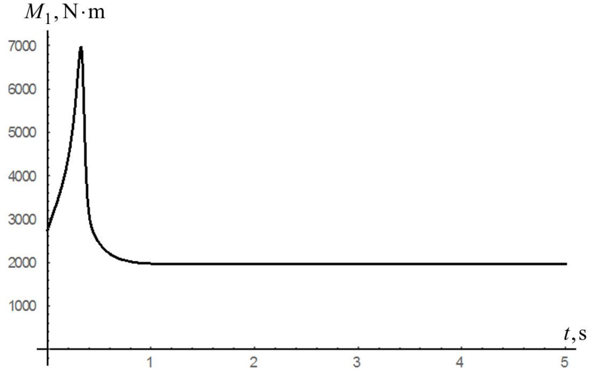

Fig. 6. Graph of the driving moment of the drive of the mechanism of load lifting

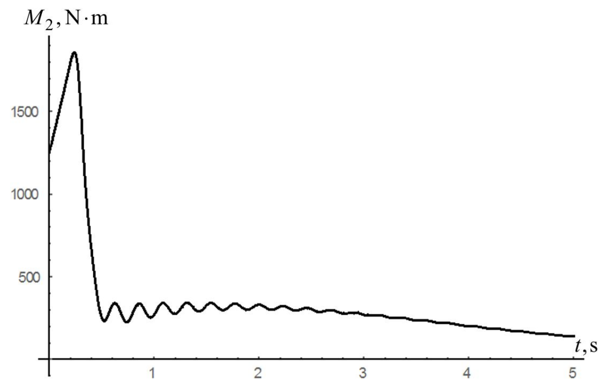

Fig. 7. Graph of the driving moment of the drive of the cart movement

The vertical component of the speed of the load (Fig. 5) is quite fast (for 0.4 s) becomes stable without any fluctuations in the speed of the lifting mechanism.

The driving moment of the load lifting mechanism at the launch site (Fig. 6) reaches 7000 Nm, which is 3.5 times higher than its established value. This indicates a significant overload of the lifting mechanism during the transition processes (start-up, braking).

The maximum value of the driving moment of the drive of movement of the cart during start-up makes 1850 N·m (Fig. 7) that is almost 9 times higher than its minimum and 6 times the maximum value on a site of the established movement. In addition, high-frequency oscillations of the driving moment of the drive of the trolley movement are observed in the area of steady traffic, which attenuate within three seconds. From the above it can be concluded that the mechanism of movement of the trolley at the launch site works with significant overloads.

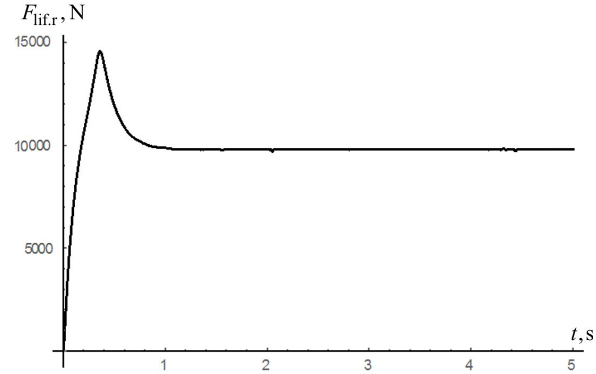

Fig. 8. Graph of effort change in a load lifting rope

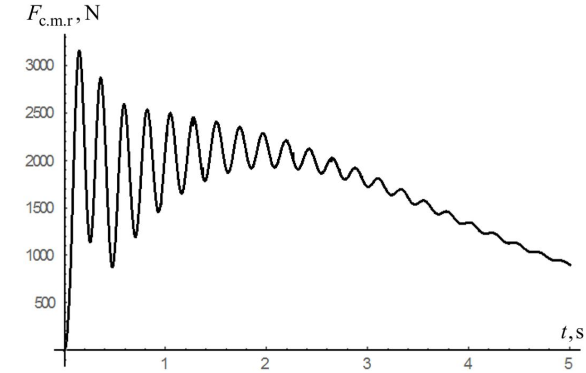

Fig. 9. Graph of effort change in a rope of the cart movement

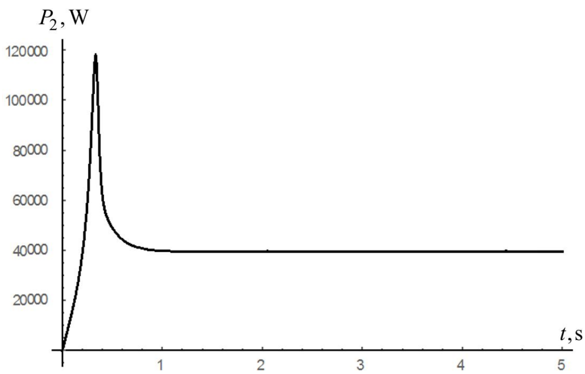

Fig. 10. Graph of power change of the drive of the load lifting mechanism

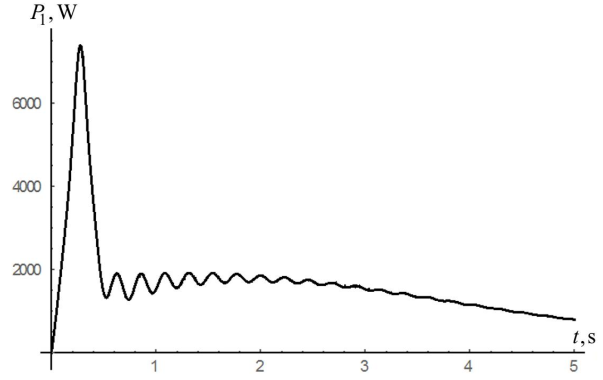

Fig. 11. Graph of power change of the drive of the cart movement mechanism

The maximum value of the traction force in the rope during the start of the lifting mechanism is 15 kN (Fig. 8), which is 1.5 times higher than its established value. These results also indicate the overload of the traction rope of the lifting mechanism during the transient start-up process.

Significant high-frequency oscillations take place in the traction rope of the cart movement mechanism (Fig. 9), which attenuate within five seconds of movement. The maximum value of the traction force is observed at the beginning of the start and is 3.2 kN, which is more than three times its minimum value in the area of steady motion. All this indicates a significant dynamic overload of the mechanism for moving the cart with the load.

In the process of starting the load lifting mechanism, the power of the drive motor increases sharply to a maximum value of 120 kW (Fig. 10), and then also drops sharply to a steady state value of 40 kW. This mode of starting the lifting mechanism leads to three times the overload of the engine in power compared to the steady state mode.

The power of the drive motor of the mechanism of movement of the cart (Fig. 11) also sharply increases in the course of start-up to the maximum size of 7,5 kW then decreases sharply decreases to the steady value of 2,0 kW. There are slight high-frequency power fluctuations in the steady-state area, which attenuate for five seconds.

Conclusions

According to the results of research, the following conclusions can be drawn:

1. The mathematical model of dynamics of joint movement of change of tower crane flight and load lifting of the tower crane with a beam jib is constructed. The developed model takes into account the main movement of the drive mechanisms, as well as high-frequency oscillations of the links with elastic properties and low-frequency oscillations of the load on the flexible suspension.

2. As a result of the calculations, a dynamic analysis of the joint movement of the trolley and hoisting mechanism was carried out. There are high-

frequency oscillations of the links of the mechanisms possessing elastic properties which attenuate quickly enough, and also low-frequency oscillations of freight on a flexible suspension which practically do not attenuate.

3. To reduce the overload of the trolley and hoisting mechanism during start-up and braking, as well as elimination of oscillations in areas of steady traffic, it is recommended to optimize the modes of joint movement of mechanisms and develop drive control systems.

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Ловейкін В.С., Ромасевич Ю.О., Шимко Л.С., Ловейкін Ю.В., Почка К.І.
ДИНАМІЧНИЙ АНАЛІЗ СУМІСНОГО РУХУ МЕХАНІЗМІВ ЗМІНИ ВИЛЬОТУ ТА ПІДЙОМУ ВАНТАЖУ БАШТОВОГО КРАНУ

В представленій роботі поставлена задача дослідження сумісної роботи механізмів зміни вильоту та підйому вантажу баштового крану з балочною стрілою. Для проведення досліджень розроблено динамічну модель сумісного руху механізмів, яка враховує основний рух приводних механізмів та коливальний рух вантажу на гнучкому підвісі і ланок механізмів, що володіють пружними властивостями. На базі динамічної моделі за допомогою рівнянь Лагранжа другого роду побудовано математичну модель руху механізмів. Для конкретної стрілової системи баштового крана здійснено динамічні розрахунки за допомогою розробленої математичної моделі. За результатами розрахунків проведено динамічний аналіз сумісного руху механізмів зміни вильоту та підйому вантажу. В процесі проведеного аналізу виявлені значні динамічні та енергетичні перевантаження механізмів під час проходження перехідних процесів (пуск, гальмування) та наявність високочастотних коливань в ланках, що володіють пружними властивостями та низькочастотних коливань вантажу на гнучкому підвісі.

Для зменшення перевантажень кранових механізмів на ділянках перехідних процесів і усунення коливань під час усталеного руху рекомендовано здійснювати оптимізацію режимів руху і розробляти системи керування приводами, щоб реалізувати бажані режими руху.

Ключові слова: динаміка, аналіз, баштовий кран, сумісний рух, підйомний механізм, візок, вантаж, оптимізація

Loveikin V.S., Romasevich Yu.A., Shymko L.S., Loveikin Yu.V., Pochka K.I.
THE DYNAMIC ANALYSIS OF THE JOINT TROLLEY MOVEMENT AND HOISTING MECHANISM IN THE TOWER CRANE

The task of this paper is to research the simultaneous use of trolley and hoisting mechanism by the tower crane with a beam jib. To conduct research, a dynamic model of joint movement of mechanisms has been developed, which takes into account the main movement of drive mechanisms and oscillating movement of the load on a flexible suspension and links of mechanisms with elastic properties. A mathematical model of the motion of mechanisms is constructed on the basis of a dynamic model with the help of Lagrange equations of the second kind. For a specific jib system of the tower crane, dynamic calculations were performed using the developed mathematical model. According to the results of the calculations, a dynamic analysis of the joint movement of trolley and hoisting mechanism was carried out. The analysis revealed significant dynamic and energy overloads of mechanisms during transients (start, braking) and the presence of high-frequency oscillations in the links with elastic properties and low-frequency oscillations of the load on a flexible suspension.

To reduce the overload of crane mechanisms in the areas of transients and eliminate oscillations during steady traffic, it is recommended to optimize traffic modes and develop drive control systems to implement the desired traffic modes.

Keywords: dynamic, analysis, tower crane, joint movement, hoisting mechanism, hook blocks, load, optimization.

Ловейкин В.С., Ромасевич Ю.А., Шимко Л.С., Ловейкин Ю.В., Почка К.И.

ДИНАМИЧЕСКИЙ АНАЛИЗ СОВМЕСТНОГО ДВИЖЕНИЯ МЕХАНИЗМОВ ИЗМЕНЕНИЯ ВЫЛЕТА И ПОДЪЕМА ГРУЗА БАШЕННОГО КРАНА

В представленной работе поставлена задача исследования совместной работы механизмов изменения вылета и подъема груза башенного крана с балочной стрелой. Для проведения исследований разработана динамическая модель совместного движения механизмов, которая учитывает основное движение приводных механизмов и колебательное движение груза на гибком подвесе и звеньев механизмов, обладающих упругими свойствами. На базе динамической модели с помощью уравнений Лагранжа второго рода построена математическая модель движения механизмов. Для конкретной стреловой системы башенного крана осуществлен динамические расчеты с помощью разработанной математической модели. По результатам расчетов проведено динамический анализ совместного движения механизмов изменения вылета и подъема груза. В процессе проведенного анализа выявлены значительные динамические и энергетические перегрузки механизмов при прохождении переходных процессов (пуск, торможение) и наличие высокочастотных колебаний в звеньях, обладающих упругими свойствами и низкочастотных колебаний груза на гибком подвесе.

Для уменьшения перегрузок крановых механизмов на участках переходных процессов и устранения колебаний во время установившегося движения рекомендуется осуществлять оптимизацию режимов движения и разрабатывать системы управления приводами, чтобы реализовать желаемые режимы движения.

Ключевые слова: динамика, анализ, башенный кран, совместное движение, подъемный механизм, тележка, груз, оптимизация.

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Ловейкін В.С., Ромасевич Ю.О., Шимко Л.С., Ловейкін Ю.В., Почка К.І. **Динамічний аналіз сумісного руху механізмів зміни вильоту та підйому вантажу баштового крану** // Опір матеріалів і теорія споруд: наук.- тех. збірн. – К.: КНУБА. 2022. – Вип. 108. – С. 267-282.

Розроблено динамічну модель сумісного руху механізмів, яка враховує основний рух приводних механізмів та коливальний рух вантажу на гнучкому підвісі і ланок механізмів, що володіють пружними властивостями. Побудовано математичну модель руху механізмів. Для конкретної стрілової системи баштового крана здійснено динамічні розрахунки та проведено динамічний аналіз сумісного руху механізмів зміни вильоту та підйому вантажу. Виявлені значні динамічні та енергетичні перевантаження механізмів під час проходження перехідних процесів (пуск, гальмування) та наявність високочастотних коливань в ланках, що володіють пружними властивостями та низькочастотних коливань вантажу на гнучкому підвісі.

Рис. 11. Бібліогр. 20 назв.

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Loveikin V. S., Romasevich Yu.O, Shymko L.S., Loveikin Yu.V., Pochka K.I. **The dynamic analysis of the joint trolley movement and hoisting mechanism in the tower crane** // Strength of Materials and Theory of Structures: Scientific-and-technical collected articles. – K.: KNUBA. 2022. – Issue 108. – P. 267-282.

The dynamic model of joint movement of mechanisms has been developed, which takes into account the main movement of drive mechanisms and oscillating movement of the load on a flexible suspension and links of mechanisms with elastic properties. A mathematical model of the motion of mechanisms is constructed on the basis of a dynamic model. For a specific jib system of the tower crane, dynamic calculations were performed. According to the results of the calculations, a dynamic analysis of the joint movement of trolley and hoisting mechanism was carried out. The analysis revealed significant dynamic and energy overloads of mechanisms during transients (start, braking) and the presence of high-frequency oscillations in the links with elastic properties and low-frequency oscillations of the load on a flexible suspension.

Fig. 11. Ref. 20.

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Ловейкин В.С., Ромасевич Ю.А., Шимко Л.С., Ловейкин Ю.В., Почка К.И. **Динамический анализ совместного движения механизмов изменения вылета и подъема груза башенного крана** // Сопротивление материалов и теория сооружений. – 2021. – Вып. 108. – С. 267-282.

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Ил. 11. Библиогр. 20.

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