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DYNAMIC ANALYSIS OF THE JOINT MOVEMENT OF DERRICKING MECHANISM AND LIFTING MECHANISM OF A LOAD DURING A STEADY-STATE TURN OF A JIB CRANE

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Increasing the productivity of jib cranes is an urgent problem of improving their operation. Combining the work of separate mechanisms is one of the ways to increase the productivity of jib cranes. The aim of the study is to build a mathematical model and conduct a dynamic analysis of the crane jib system with simultaneous operation of the derricking mechanism and lifting mechanism of the load during a steady-state crane rotation. Methods for constructing discrete dynamic models of a jib crane by using Lagrange equations of the second kind, numerical methods for solving the obtained differential equations, which are presented in the form of a computer program at a steady-state crane rotation, and methods for dynamic analysis of crane mechanisms are used in the conducted research. The task of researching the dynamics of the simultaneous movement of the mechanisms for turning the boom, extending its section and lifting the load during a steady-state crane turn is solved in the presented work. The method of dynamic analysis was developed to study dynamic processes in the hydro-mechanical system of a jib crane during the simultaneous operation of crane mechanisms. The crane boom system is represented by a dynamic model with six degrees of freedom, which takes into account the main movement of the mechanisms and the oscillations of the links and the load on a flexible suspension. The kinematic, dynamic, and energy characteristics of individual links of the crane jib system with simultaneous operation of several mechanisms are determined on the basis of the constructed mathematical model. The high-frequency oscillations of the drive links of the load lifting mechanism and the low-frequency oscillations of the load on a flexible suspension are investigated. It was found that high-frequency vibrations of the links damped within the start-up process, while low-frequency vibrations of the load practically did not damp and continued throughout the entire movement cycle.

Drive modes that ensure smooth movement of the actuators, which leads to reduced loads and increased reliability of the crane are recommended to minimise oscillatory processes of simultaneous movement of the jib system mechanisms.

Keywords: boom turning mechanisms, boom section extension, load lifting, oscillatory processes, dynamic loads.

Introduction. One of the ways to increase the productivity of jib cranes is to combine the simultaneous operation of several mechanisms. Dynamic loads in structural elements and drive mechanisms increase during such operation of jib cranes. These loads become especially dangerous during transient processes (starting, braking, changing speed). Such operation of crane mechanisms affects the reliability of jib cranes. In this regard, determining the actual dynamic loads and studying oscillatory processes in the elements of the jib system during the simultaneous operation of several mechanisms is an important task.Jib cranes often combine the operations of changing the outreach and lifting the load when performing loading, unloading and installation operations. In this regard, there is a necessity to investigate the dynamic processes during the joint operation of the mechanisms for changing the outreach and lifting the load during the steady-state rotation of a jib crane with a retractable jib section. The task of studying the dynamic processes during the joint movement of the mechanisms for turning and extending the boom section and lifting the load during a steady-state crane turn is relevant, as it allows us to identify the real loads during the operation of jib cranes and in the future to make decisions about the possibility of combining the operation of certain mechanisms.

Analysis of publications. The study of dynamic processes in the operation of crane mechanisms and, in particular, their simultaneous use remains an urgent problem. The simultaneous use of crane mechanisms in operation makes it possible to increase the productivity of a crane, but at the same time, the loads in the structural elements increase.

The peculiarities of controlling the electric drive of the mechanism for changing the boom reach during the simultaneous rotation of a jib crane with a suspended load are described in [1]. The dynamic loads during the simultaneous movement of these mechanisms were also studied. The authors of work [2] considered the joint movement of the mechanisms for lifting the boom and the load of a jib crane. The authors modelled the dynamics of the simultaneous movement of the mechanisms and performed a dynamic analysis. It is established that during the simultaneous operation of the mechanisms, the crane operates with overloads. In [3], the loads due to the influence of load sway on the dynamic characteristics of the crane and the accuracy of its positioning are determined on the basis of a multi-mass model of a tower crane.The authors of work [4] modelled the dynamics of a tower crane during the operation of the slewing mechanism. A significant swaying of the load and its significant impact on the crane's performance are noted. Paper [5] modelled the dynamics of the trolley movement mechanism at a steady-state crane rotation. In addition, the minimisation of the driving torque, which has an impact on the swaying of the load, was carried out. The authors of [6] performed a modelling of the dynamics of a tower crane with a load and investigated the dynamic response of the crane caused by pendulum oscillations of the load on a flexible suspension. In article [7], the dynamics of the simultaneous movement of the mechanisms for changing the outreach, lifting the load, and turning the crane is modelled. A dynamic analysis of the simultaneous movement of the three mechanisms was carried out, as a result of which overloading of the mechanisms was revealed. The authors of article [8] investigated the dynamics of the swing oscillations of the load of a double-jib bridge crane and showed the effect of these oscillations on the crane's characteristics. The dynamics of a three-dimensional crane system with a suspended load and the determination of dynamic loads are described in work [9]. Dynamic loads and stresses in the crane structure were also determined. The authors of article [10] carried out modelling of the dynamics and control of a five-stage crane jib system and identified the loads acting on the structural elements. The authors of [11] carried out experimental studies of the dynamics of the simultaneous movement of the mechanisms for turning and moving the crane trolley with the load of a tower crane, which confirmed the adequacy of the constructed mathematical models of the dynamics of a tower crane. In article [12], a dynamic analysis of the movement of a container crane was carried out, taking into account the adhesion between the load and the gripping device. It is shown that dynamic loads have a significant impact during the start-up and braking of a container crane. In work [13], dynamic modelling, dynamic analysis, and control of a five-section retractable jib crane were performed and implemented. Article [14] shows that the dynamics of a jib crane's movement is influenced by the sway of the load on a flexible suspension, the reduction of which leads to an increase in the efficiency of the crane.The authors of work [15] showed that reducing dynamic loads and swaying of the load leads to an increase in the reliability of jib cranes. The researchers of work [16] developed a mathematical model of the dynamics of the joint movement of the mechanism for changing the outreach and rotation of a tower crane, and also conducted a dynamic analysis of the simultaneous movement of these mechanisms. The analysis showed an increase in dynamic loads. A method for controlling and managing the movement of jib crane mechanisms to minimise load oscillations was developed in article [17]. In work [18], the issues of controlling load oscillations on a flexible suspension and the vibration characteristics of the structural elements of a jib crane are considered. The authors of article [19] investigated the dynamics of the joint movement of the load lifting mechanisms and changes in the boom reach and identified the causes of load oscillations on a flexible suspension.

From the literature review, it follows that in order to determine loads and vibrations in the structural elements of a hydraulically operated jib crane, it is necessary to conduct a dynamic analysis during the joint movement of the mechanisms for turning and extending the boom section and lifting the load during a steady-state crane turn.

Recently, research related to modeling, control, analysis of dynamics, and optimization of the movement of cranes or their mechanisms has become widely spread [1-18]. Dynamic analysis and control of a bridge crane with several hoisting mechanisms using sliding mode control is carried out in the study [1]. A dynamic analysis of an offshore boom crane and a study of its nonlinear control were carried out by the authors of the article [2]. The results of the dynamic analysis of a tower crane with a rotating boom using multibody system simulation, as well as using the Kane method, are given in studies [3,4]. The dynamic analysis of the container crane, taking into account the effect of coupling between the cargo and the spreader, is given in the article [5]. A dynamic analysis of the movement of a cable crane with dual winches and the determined loads in the traction body and the structural elements was carried out in the work [6]. The authors of the article [7] developed a mathematical model of the dynamics of the joint movement of the mechanisms of lifting, turning, and changing the outreach of the crane, and also investigated their combined movement. The joint movement of the mechanisms for changing the departure and rotation of the tower crane, where the simulation of the dynamics of the movement was carried out, as well as the optimization of the starting of the mechanisms was considered in the work [8]. It was established that the crane works with overloads when the mechanisms move together. The authors of the article [9] carried out a numerical simulation of the dynamics of the movement of the trolley-cargo-carrying rope system in a rope crane and determined the loads acting on the structural elements of the crane. The load from the influence of the swinging of the load on a flexible suspension on the dynamic characteristics of the crane and the accuracy of the positioning of the load are established in the study [10] based on the constructed multi-mass model of the tower crane. A dynamic analysis of the influence of the design of the new traveling wheels of the bridge crane on the stresses in the structural elements that occur during the movement of the crane was carried out in [11].

The authors of the article [12] developed a model for the formation of reference teams for oscillation control of multi-mode flexible mechanical systems for studying the dynamics of a double-pendulum bridge crane. The study of a reliable observer against rocking of 2D crane systems with lifting and lowering of the cargo was carried out in work [13]. The motion control system of two wired hammer head tower crane, which allows to improve the movement modes of the crane mechanisms, is given and described in the article [14]. The authors of the study [15] developed an anti-sway tracking control of tower cranes with delayed uncertainty using a robust adaptive fuzzy control.

The dynamic characteristics of jib crane mechanisms can be improved by optimization of the parameters and movement modes of crane mechanisms. In the article [16], the optimization of the parameters of the crane mechanisms made it possible to significantly reduce the oscillation of the jib crane and its mechanisms. The authors of the studies [17,18] developed methods for optimizing the modes of movement of crane mechanisms, which allow to minimization of the effect of dynamic loads and vibrations of the structural elements of the crane and cargo. To dynamically assess the movement of crane mechanisms and detect loads and oscillation in the structural elements of a jib crane, there is a need to conduct a dynamic analysis when the boom and load lifting mechanisms are simultaneously started.

Purpose of the paper. The purpose of this study is to build a mathematical model of the dynamics of movement and conduct a dynamic analysis of the crane boom system with an extendable boom section when the mechanisms for turning the boom and extending its section and lifting the load at steady-state rotation of the crane move together.

Research results. The dynamic model shown in fig. 1 is presented to study the dynamics of changes in the outreach and lifting of the load during a steady-state crane rotation of the jib system. In this model, the boom consists of a main 1 and an outrigger 2 section. The extendable section of the boom is driven by a hydraulic cylinder located inside the main section. The boom is rotated by a hydraulic cylinder 6, which is pivotally connected to the crane's turntable by a cylinder and to the boom section by a rod. The load lifting mechanism 4 consists of a hydraulic drive, a gearbox, a drum 5 and a flexible load suspension, which is represented by a single chain hoist 3. The drive with a hydraulic motor and gearbox sets in motion the drive drum 5, on which the rope of the load lifting mechanism is wound. The dynamic model of the boom system consists of absolutely rigid and elastically dissipative links. The rope of the load lifting mechanism has elastic-dissipative properties, while the boom root and extension sections, the load and the drive elements are absolutely rigid links. The flexible load suspension performs pendulum oscillations in the plane of outreach change and has

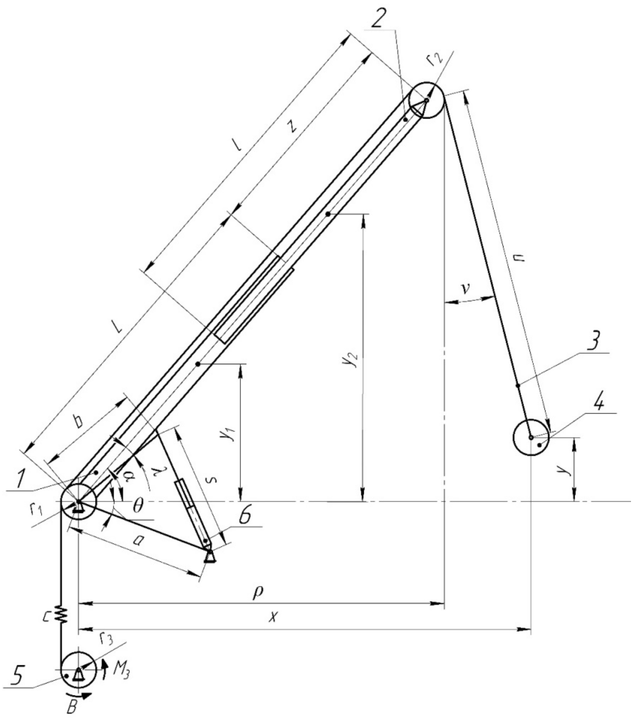

Fig. 1. Dynamic model of the crane jib system with joint operation of the mechanisms for changing the outreach and lifting the load at a steady-state crane rotation

dissipative properties. The drive elements of the load lifting mechanism are reduced to the axis of rotation of the drive drum. In the present study, the outreach change mechanism consists of mechanisms for turning the boom and extending the boom section. At the same time, the study of the movement of the mechanisms for changing the outreach of the boom and lifting the load was carried out at a steady-state crane rotation.

Thus, the dynamic model of the boom system with hydraulic drive mechanisms for simultaneous boom rotation, extension of its section and lifting of the load at a steady-state crane rotation is represented by a hydro-mechanical system with six degrees of freedom (Fig. 1). The generalized coordinates of this system are the angular coordinates of the boom rotation α, the drum of the lifting mechanism β, the crane turntable φ and the deviation from the vertical of the flexible load suspension v, as well as the linear coordinates of the movement of the extendable boom section z and the length of the flexible load suspension u. Since the crane rotation mechanism operates in a steady-state mode, we assume that the angular velocity of the rotating platform is constant, i.e. $\omega=const$, and its angular coordinate is determined by the relationship

$$\varphi=\varphi_0+\omega t, \tag{1}$$

where t is the time coordinate; φ_0 is the initial value of the angular coordinate of the crane's slewing platform; ω is the angular velocity of the crane's slewing platform.

The elements of the boom system are subject to gravitational forces from the weight of the main and extension boom sections, the weight of the load, as well as driving forces in the hydraulic cylinders of the boom slewing mechanism F_1 and the movement of the extension boom section F_2. The force F_1 creates a moment M_1 that acts on the boom and turns it. In addition, the torque of the hydraulic motor of the load lifting mechanism M_3 is applied, which is reduced to the axis of rotation of the drive drum. In the elastic-dissipative rope of the load lifting mechanism, elastic and dissipative forces act, and when the flexible load suspension deviates from the vertical, a dissipative moment acts.

The Lagrange equation of the second kind was used to construct a mathematical model of the dynamics of the joint movement of the mechanisms for changing the outreach and lifting the load during a steady-state crane rotation. These equations have the following form:

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\alpha}}-\frac{\partial T}{\partial \alpha}=M_1-\frac{\partial \Pi}{\partial \alpha};$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\beta}}-\frac{\partial T}{\partial \beta}=M_3-\frac{\partial \Pi}{\partial \beta}-\frac{\partial R}{\partial \dot{\beta}};$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{z}}-\frac{\partial T}{\partial z}=F_2-\frac{\partial \Pi}{\partial \beta}-\frac{\partial R}{\partial \dot{z}};$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{u}}-\frac{\partial T}{\partial u}=-\frac{\partial \Pi}{\partial u}-\frac{\partial R}{\partial \dot{u}};$$
$$\frac{d}{dt}\frac{\partial T}{\partial \dot{v}}-\frac{\partial T}{\partial v}=-\frac{\partial \Pi}{\partial v}-\frac{\partial R}{\partial \dot{v}}, \qquad (2)$$

where T, Π, R are the kinetic and potential energy of the system and the dissipative Rayleigh function, respectively; M_1, M_3 – driving torques of the drives of the boom rotation and lifting mechanisms, reduced according to the rotation of the boom and the drive drum of the lifting mechanism; F_2 is the driving force of the hydraulic cylinder for moving the extendable boom section.

The kinetic energy of the combined movement of the mechanisms for turning the boom, extending its section and lifting the load during a steady rotation of the crane is determined by the following dependence:

$$T=\frac{1}{2}\left[J_1+J_2+m_2(z+L)(z+L-l)\right]\left(\dot{\alpha}^2+\omega^2(\cos\alpha)^2\right)+\frac{1}{2}m_2\dot{z}^2+\frac{1}{2}J_3\dot{\beta}^2+\frac{1}{2}J_4\omega^2+\frac{1}{2}m\left(\dot{x}^2+\dot{y}^2+x^2\omega^2\right), \qquad (3)$$

where J_1 is the moment of inertia of the main boom section relative to the axis of its rotation; J_2 is the moment of inertia of the extendable boom section relative to the axis of its base; J_3 is the moment of inertia of the drive of the load lifting mechanism reduced to the axis of rotation of the drum; J_4 is the moment of inertia of the crane turntable relative to its own axis of rotation; m_2, m are respectively, the masses of the extendable boom section and the load; x, y are horizontal and vertical coordinates of the centre of mass of the load in the plane of change of the load outreach; l is length of the extendable boom section; L is length of the main boom section.

The potential energy of the jib system with the joint movement of the mechanisms for changing the outreach and lifting the load at a steady-state crane rotation is represented by the following dependence

$$\Pi=\frac{1}{2}c\left[\beta r-(z-z_0)(u_0-u)n\right]^2+(m_1y_1+m_2y_2+my)g, \qquad (4)$$

where c is the stiffness coefficient of the rope of the cargo hoisting mechanism, reduced to the axis of rotation of the drive drum; r – the radius of the drive drum of the cargo hoisting mechanism; n – the pulley block ratio of the hoisting mechanism, g – acceleration of gravity; m_1 - mass of the main boom section; y, y_1, y_2 are vertical coordinates of the centres of mass of the main and extendable sections of the boom and the load, respectively; u_0, z_0 - initial coordinate values of the length of the flexible load suspension and the movement of the extendable section of the boom, respectively.

The dissipative function of the Relay system has the following form

$$R=\frac{1}{2}b_1\left(\dot{\beta}r-\dot{z}+n\dot{u}\right)^2+\frac{1}{2}b_2\dot{v}^2 , \qquad (5)$$

where b_1, b_2 are the damping coefficients of the elastic elements, respectively, of the drive of the cargo hoisting mechanism and the deviation from the vertical of the flexible suspension of the cargo.

The coordinates of the load and boom sections are represented by the following expressions:

$$x=(z+L)\cos\alpha+u\sin v;\ \ y=(z+L)\sin u-u\cos v\ ; \qquad (6)$$

$$y_1=\frac{1}{2}L\sin\alpha,\ \ y_1=\left(z+L-\frac{1}{2}\right)\sin\alpha. \qquad (7)$$

From expressions (6), it can be seen that the position of the load in the plane of change of overhang depends on four generalised coordinates: α, v, z and u.

The methodology for constructing a mathematical model of a crane jib system with joint movement of mechanisms is given in [2], so only the final result is presented here. After determining the necessary derivatives in accordance with system (2) from the kinetic energy (3), potential energy (4), and Rayleigh function (5) and substituting them into the same system, the differential equations of the joint motion of the mechanisms for turning the boom, extending its section, and lifting the load during the steady-state rotation of the crane are obtained:

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$$\left[J_1+J_2+m_2(z+L-l)(L+z)\right]\left[\ddot{\alpha}+\omega^2\sin\alpha\cos\alpha\right]+2m_2\dot{\alpha}\dot{z}\left(z+L-\frac{l}{2}\right)+$$
$$+m\left[\left(\ddot{x}-x\omega^2\right)\frac{\partial x}{\partial\alpha}+(g+\ddot{y})\frac{\partial y}{\partial\alpha}\right]=M_1-\left(m_1\frac{\partial y_1}{\partial\alpha}+m_2\frac{\partial y_2}{\partial\alpha}\right)g;$$
$$J_2\ddot{\beta}=M_3-cr\left[\beta+r-(u_0-u)n\right]-b_1r\left(\dot{\beta}r-n\dot{u}\right);$$
$$m_2\ddot{z}-m_2\left(z+L-\frac{l}{2}\right)\left[\dot{\alpha}^2+(\omega\cos\alpha)^2\right]+m\left[\left(\ddot{x}-x\omega^2\right)\frac{\partial x}{\partial z}+(g+\ddot{y})\frac{\partial y}{\partial z}\right]=$$
$$=F_2-c\left[\beta r-z+z_0-(u_0-u)m\right]-m_2g\frac{\partial y_2}{\partial z}-b_1\left(\dot{\beta}r-\dot{z}+n\dot{u}\right);$$
$$m\left[\left(\ddot{x}-x\omega^2\right)\frac{\partial x}{\partial u}+(g+\ddot{y})\frac{\partial y}{\partial u}\right]=-cn\left[\beta r+z+z_0-(u_0-u)n\right]+b_1n\left(\dot{\beta}r+\dot{z}-n\dot{u}\right);$$
$$m\left[\left(\ddot{x}-x\omega^2\right)\frac{\partial x}{\partial v}+(g+\ddot{y})\frac{\partial y}{\partial v}\right]=-b_2\dot{v}. \tag{8}$$

The system of differential equations (8) includes partial and full derivatives of the coordinates of individual links of the boom system. These derivatives were determined for the dynamic model of the crane boom system shown in Fig. 1. Below are the partial derivatives of the coordinates of the boom and load by generalised coordinates:

$$\frac{\partial y_1}{\partial\alpha}=\frac{1}{2}L\cos\alpha;\ \frac{\partial y_2}{\partial\alpha}=\left(z+\frac{l}{2}\right)\cos\alpha;\ \frac{\partial y_2}{\partial z}=\sin\alpha; \tag{9}$$

$$\frac{\partial x}{\partial\alpha}=-(z+L)\sin\alpha;\ \frac{\partial y}{\partial\alpha}=(z+L)\cos\alpha; \tag{10}$$

$$\frac{\partial x}{\partial z}=\cos\alpha;\ \frac{\partial y}{\partial z}=\sin\alpha; \tag{11}$$

$$\frac{\partial x}{\partial u}=\sin v;\ \frac{\partial y}{\partial u}=-\cos v; \tag{12}$$

$$\frac{\partial x}{\partial v}=u\cos v;\ \frac{\partial y}{\partial v}=u\sin v; \tag{13}$$

$$\frac{\partial^2 x}{\partial\alpha^2}=-(z+L)\cos\alpha;\ \frac{\partial^2 y}{\partial\alpha^2}=-(z+L)\sin\alpha; \tag{14}$$

$$\frac{\partial^2 x}{\partial v^2}=-u\sin\alpha;\ \frac{\partial^2 y}{\partial v^2}=u\cos\alpha; \tag{15}$$

$$\frac{\partial^2 x}{\partial\alpha\partial z}=-\sin\alpha;\ \frac{\partial^2 y}{\partial\alpha\partial z}=\cos\alpha; \tag{16}$$

$$\frac{\partial^2 x}{\partial u\partial v}=\cos v;\ \frac{\partial^2 y}{\partial u\partial v}=\sin v; \tag{17}$$

$$\frac{\partial^2 x}{\partial\alpha\partial u}=\frac{\partial^2 y}{\partial\alpha\partial u}=\frac{\partial^2 x}{\partial\alpha\partial v}=\frac{\partial^2 y}{\partial\alpha\partial v}=0. \tag{18}$$

The first full time derivatives of the coordinates of the centre of mass of the cargo from expressions (6) were found:

$$\dot{x}=\dot{\alpha}\frac{\partial x}{\partial\alpha}+\dot{z}\frac{\partial x}{\partial z}+\dot{u}\frac{\partial x}{\partial u}+\dot{v}\frac{\partial x}{\partial v};$$
$$\dot{y}=\dot{\alpha}\frac{\partial y}{\partial\alpha}+\dot{z}\frac{\partial y}{\partial z}+\dot{u}\frac{\partial y}{\partial u}+\dot{v}\frac{\partial y}{\partial v}. \tag{19}$$

Taking into account the first and second partial derivatives of the coordinates given in (10), (18), the second full time derivatives of the coordinates of the centre of mass of the cargo are presented:

$$\ddot{x}=\ddot{\alpha}\frac{\partial x}{\partial \alpha}+\ddot{z}\frac{\partial x}{\partial z}+\ddot{u}\frac{\partial x}{\partial u}+\ddot{v}\frac{\partial x}{\partial v}+2\dot{\alpha}\dot{z}\frac{\partial^2 x}{\partial \alpha \partial z}+2\dot{u}\dot{v}\frac{\partial^2 x}{\partial u \partial v}+\dot{\alpha}^2\frac{\partial^2 x}{\partial \alpha^2}+\dot{v}\frac{\partial^2 x}{\partial v^2}; \quad (20)$$

$$\ddot{y}=\ddot{\alpha}\frac{\partial y}{\partial \alpha}+\ddot{z}\frac{\partial y}{\partial z}+\ddot{u}\frac{\partial y}{\partial u}+\ddot{v}\frac{\partial y}{\partial v}+2\dot{\alpha}\dot{z}\frac{\partial^2 y}{\partial \alpha \partial z}+2\dot{u}\dot{v}\frac{\partial^2 y}{\partial u \partial v}+\dot{\alpha}^2\frac{\partial^2 y}{\partial \alpha^2}+\dot{v}\frac{\partial^2 y}{\partial v^2}. \quad (21)$$

The driving force in the drive hydraulic cylinder 6 of the boom swing mechanism is determined as follows:

$$F_1=p_{n1}A_1\sqrt{1-\frac{A_1\dot{s}}{Q_1}} \quad (22)$$

$$s=\sqrt{a^2+b^2-2ab\cos(\theta-\lambda+\alpha)};\ \dot{s}\doteq\frac{ab}{s}\sin(\theta-\lambda+\alpha). \quad (23)$$

The torque that turns the crane boom is determined through the force in the hydraulic cylinder 6 F_1

$$M_1=F_1h, \quad (24)$$

where

$$h=\frac{\sqrt{4a^2s^2-(a^2-b^2+s^2)^2}}{2s}. \quad (25)$$

Here: h is the flowing action of the force F_1 relative to the axis of the lower boom joint; p_{n1} is working pressure in the cavity of the boom lift hydraulic cylinder; A_1 is cross-sectional area of the boom lift hydraulic cylinder piston; Q_1 is working fluid consumption by the boom swing hydraulic cylinder; a, θ - respectively, the length of the boom slewing cylinder mounting post and the angle of its inclination to the horizon; b, λ - length of the hydraulic cylinder lever on the boom and its angle of inclination to the boom axis; s - length of the hydraulic cylinder in working condition.

The driving force in the hydraulic cylinder for extending the boom section is represented by a dependence similar to (22)

$$hF_2=p_{n2}A_2\sqrt{1-\frac{A_2\dot{s}}{Q_2}}; \quad (26)$$

where p_{n2} - working pressure in the cavity of the boom section extension hydraulic cylinder; A_2 - cross-sectional area of the boom section extension hydraulic cylinder piston; Q_2 - working fluid consumption by the boom section extension hydraulic cylinder; $\dot{z}$ - stroke rate of the boom section extension hydraulic cylinder.

The driving torque on the shaft of the hydraulic motor of the load lifting mechanism is represented by a quadratic dependence on the angular velocity of the shaft and is reduced to the axis of rotation of the drive drum:

$$M=\left(M_p+\left(K\omega_0-\frac{M_p}{\omega_0}\right)i\dot{\beta}-Ki^2\dot{\beta}^2\right)i\eta. \quad (27)$$

Here

$$K=\frac{M_n-M_p\left(1-(\omega_n/\omega_0)\right)}{\omega_n(\omega_0-\omega_n)}; \quad (28)$$

M_p, M_n are respectively, the starting and rated torques of the hydraulic motor of the load lifting mechanism; ω_n, ω_0 – nominal and synchronous angular velocity of the hydraulic motor shaft of the hoisting mechanism; i – transmission ratio of the drive of the hoisting mechanism.; η - efficiency of the drive of the load lifting mechanism.

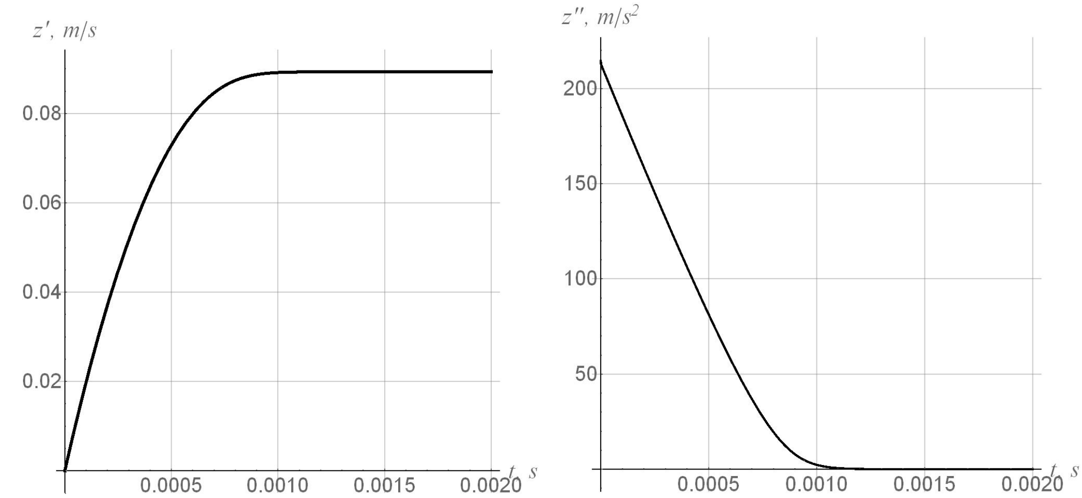
Research results

The system of second-order differential equations (8) together with expressions (9), ..., (29) is nonlinear, so a numerical method was used to solve it. The equations are solved under the following initial conditions of the joint movement of the mechanisms for changing the outreach and lifting the load at a steady-state crane rotation:

$$t=0: \alpha=\alpha_0, \dot{\alpha}=0, \beta=\frac{mg}{cnr}, \dot{\beta}=0, u=u_0, \dot{u}=0, v=0, \dot{v}=0, z=z_0, \dot{z}=0. \quad (29)$$

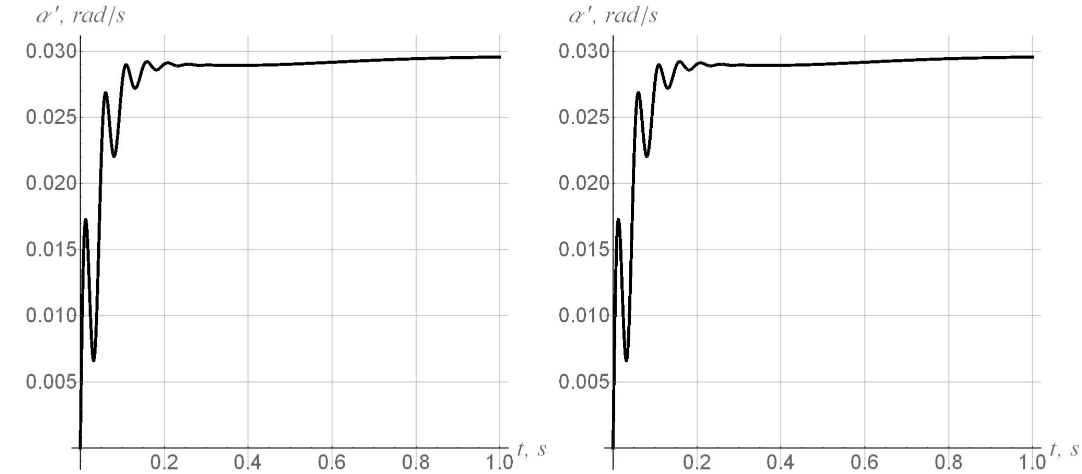
The dynamic analysis of the joint movement of the mechanisms for changing the outreach and lifting the load during a steady-state crane rotation was carried out at the following values of the parameters of the crane jib system: m=4500 kg, m_1=1800 kg, m_2=900 kg, J_1=48600 kg· m^2, J_2=12675 kg· m^2, J_3=1183 kg· m^2, c =1,25· 10^6 Nm/rad, l=6,5 m; L=9,0 m, i=41,34, η=0,85, p_{n1}=14· 10^6 N/m^2, p_{n2}=14· 10^6 N/m^2, A_1=0,0314 m^2, A_2=0,0050 m^2, Q_1=0,00178 m^3/s, Q_2=0,00075 m^3/s, ω_0=157 rad/s, ω_n=119,25 rad/s, u_0=6,0 m, n=4, g=9,81 m/s^2, M_n=133 Nm, M_p=199,5 Nm, a=1,5 m, b=2,1 m, λ=0,0872 rad, θ=0,3189 rad, α_0=0,5857 rad, r=0,208 m, b_1=1,8· 10^3 Nm/(rad/s), b_2=1,1·10^3 Nm/(rad/s), z_0=0, ω=157 rad/s.

As a result of numerically solving the nonlinear system of differential equations (8, taking into account expressions (9), ...,(29) under the initial conditions (29) of the process of joint movement of the mechanisms for changing the outreach and lifting the load during the steady-state rotation of a hydraulically operated jib crane, graphical dependences of kinematic (Figs. 2-8), dynamic (Figs. 9,...,12), and energy (Figs. 13,...,15) characteristics were constructed.



(a) (b)

Fig. 1. The driving of velocity (a) and acceleration (b) of boom section extension

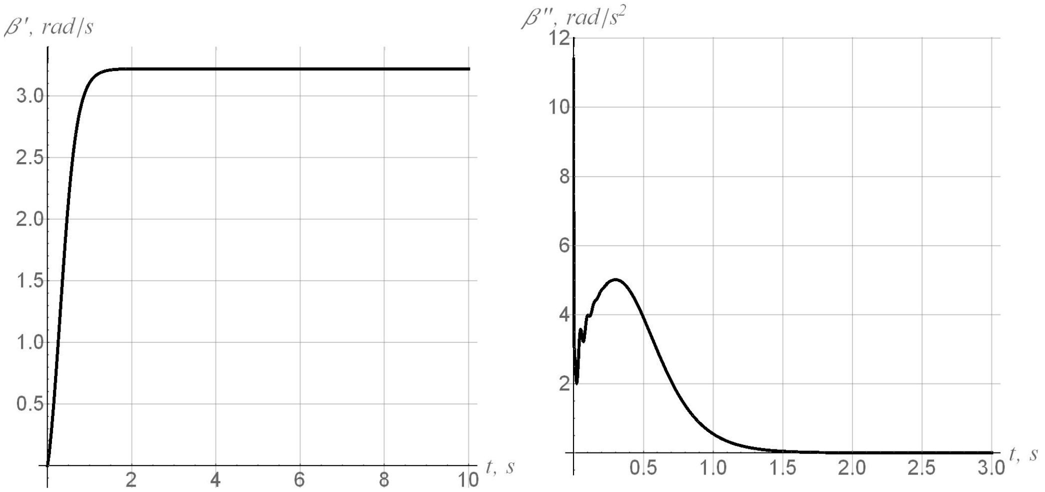


(a) (b)

Fig. 3. The driving of angular velocity (a) and acceleration (b) of boom turning

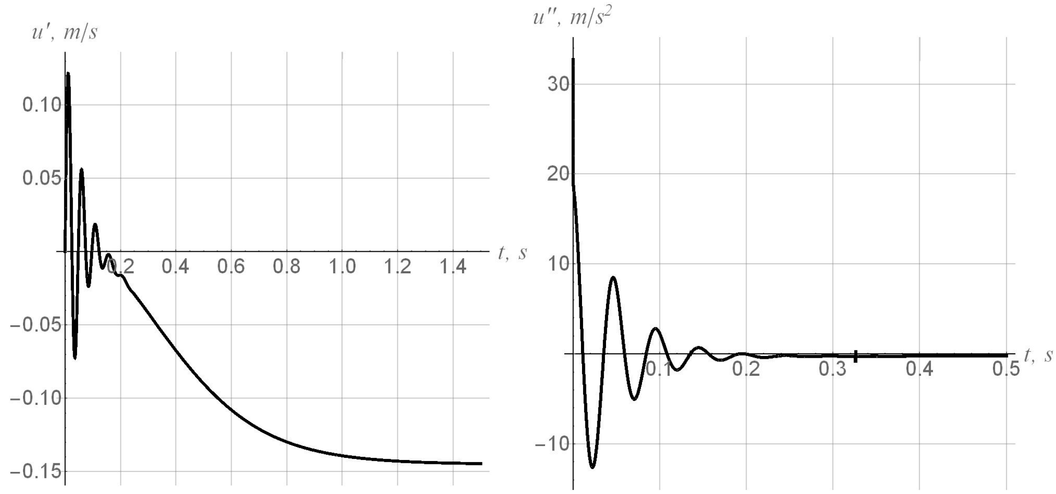
Fig. 2 shows that the speed and acceleration of the extension of the hydraulic cylinder rod during the start-up of the boom section extension mechanism change smoothly without oscillations. The velocity (Fig. 2, a) varies according to a parabolic law, and the acceleration (Fig. 3, b) - according to a law that is close to linear. The start-up process of this mechanism is carried out with a short duration of 0.001 s, which leads to a significant maximum acceleration of 220 m/s^2 and, as a result, large dynamic loads acting on the boom and the extension section.

The process of starting the boom rotation mechanism, which is shown in Fig. 3, is also carried out in a short period of time (0.2 s), which leads to a significant maximum value of the angular acceleration of the boom, which reaches 3.8 rad/s^2. This results in increased dynamic loads and low-frequency oscillatory processes of speed and acceleration, which decay by the end of the start-up process. Subsequently, the boom rotates in a steady-state mode with a constant angular velocity.



(a) (b)

Fig. 4. The driving of angular velocity (a) and acceleration (b) of the drive drum



(a) (b)

Fig. 5. The driving of linear velocity (a) and acceleration (b) of the change in the length of a flexible load suspension

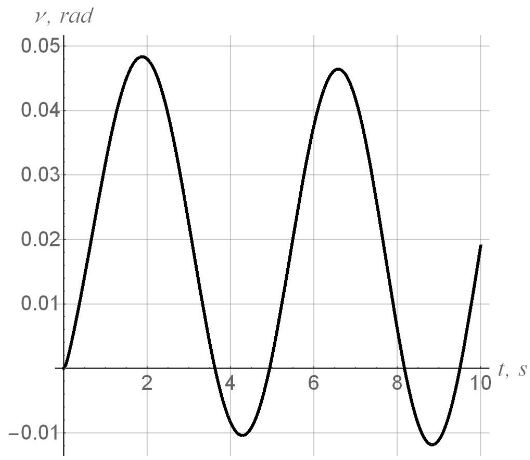
At the beginning of the movement, the velocity and acceleration (Fig. 5) of the change in the length of the flexible load suspension have a clearly expressed oscillatory character, which is caused by a significant value of the maximum acceleration, which is 33 m/s^2. At the same time, the rate of change in the length of the flexible load suspension changed by sign to opposite values. However, the oscillatory processes damped down within 0.2 s and then the speed changed smoothly to the steady-state value of the load lifting mechanism.

Figure 6.a shows a graph of the deviation from the vertical of the flexible suspension of the load, which looks like a sinusoid shifted relative to the abscissa axis. This character of the graph is caused by the fact that the flexible suspension additionally deviated from the action of the centrifugal force on the load, which occurs during the operation of the rotation mechanism. At the beginning of the movement, high-frequency fluctuations in the rate of deviation from the vertical of the flexible suspension are observed, which are the source of low-frequency vibrations of the load on the flexible suspension. After a short period of time (0.2 s), the high-frequency oscillations dampen, and the low- and the low-frequency oscillations practically do not dampen, as can be seen from Fig. 6.

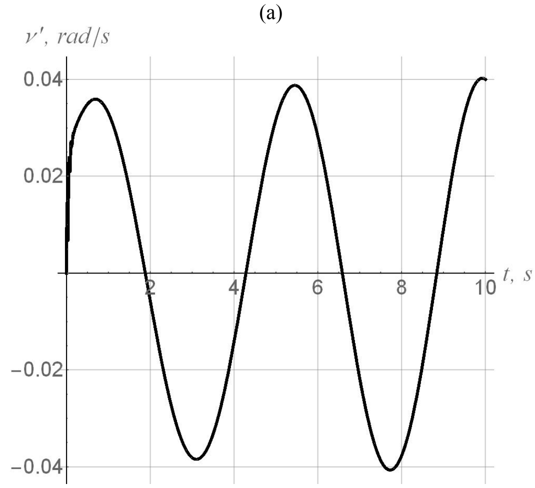
Figure 7.a shows a phase portrait of low-frequency vibrations of a load on a flexible suspension, which shows that these vibrations are practically unattenuated during the operation of jib crane mechanisms. This phase portrait also shows high-frequency oscillations at the beginning of the movement, which decay rather quickly. The phase portrait of the elastic vibrations of the traction rope of the load lifting mechanism (Fig. 7, b) shows that these vibrations dampen within three cycles.

Oscillatory processes are also observed in the graphs of load speed and acceleration (Figures 8 and 9) when the overhang and lifting mechanisms are operating. In the first case, these oscillations are practically undamped, so they negatively affect the operation of the outboard mechanism. In the second case, the oscillations are quickly damped, so they do not have a significant impact on the mechanism.

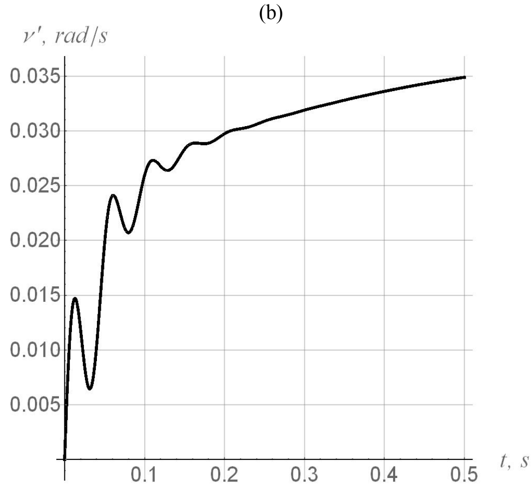
The driving force in the hydraulic cylinder of the boom lifting drive (Fig. 10, a) at the beginning of the movement increases sharply to 430 kN and in the same way decreases to 300 kN, followed by a slight gradual decrease according to a law close to linear. When the driving force decreases at the beginning of the movement, oscillatory processes occur (Fig. 10, b), which decay rather quickly. At the same time, they cause fluctuations in the speed of the hydraulic cylinder rod (Fig. 10, c), which are transmitted to the crane boom.



(a)

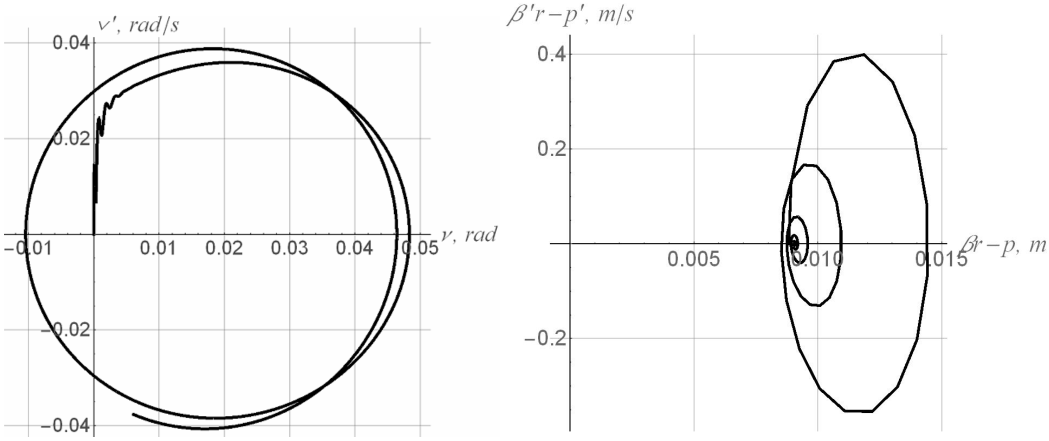


(b)



(c)

Fig. 6.The driving of angular deviation from the vertical (a) and velocity (b, c) of a flexible load suspension



(a) (b)

Fig. 7. The phase portraits of oscillations of a load on a flexible suspension (a) and a traction rope (b) of the lifting mechanism

The driving force of the hydraulic cylinder of the boom section movement drive at the beginning of the movement smoothly changes from a maximum value of 210 kN to a steady-state value of 25 kN according to a law close to linear (Fig. 11, a). In this case, the change in driving force is carried out almost instantaneously in 0.001 s. The driving torque of the load lifting mechanism also changes smoothly (Fig. 11, b). Initially, the driving torque increases smoothly from an initial value of 6.0 kNm to 8.25 kNm, and then gradually decreases to a steady-state value of 2.25 kN. At the same time, there are no oscillatory processes in the drives of the mechanisms for moving the boom section and lifting the load, which is due to the smoothness of the change in the driving forces of the drives.

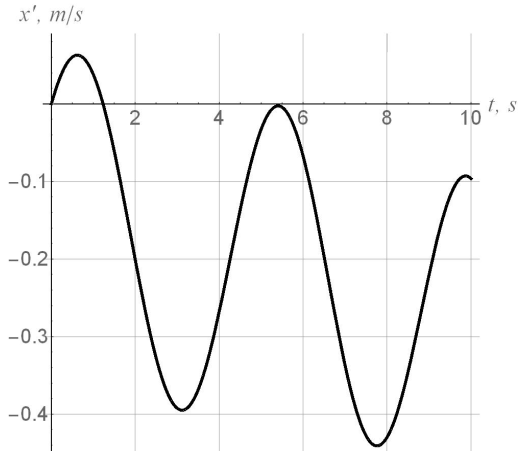
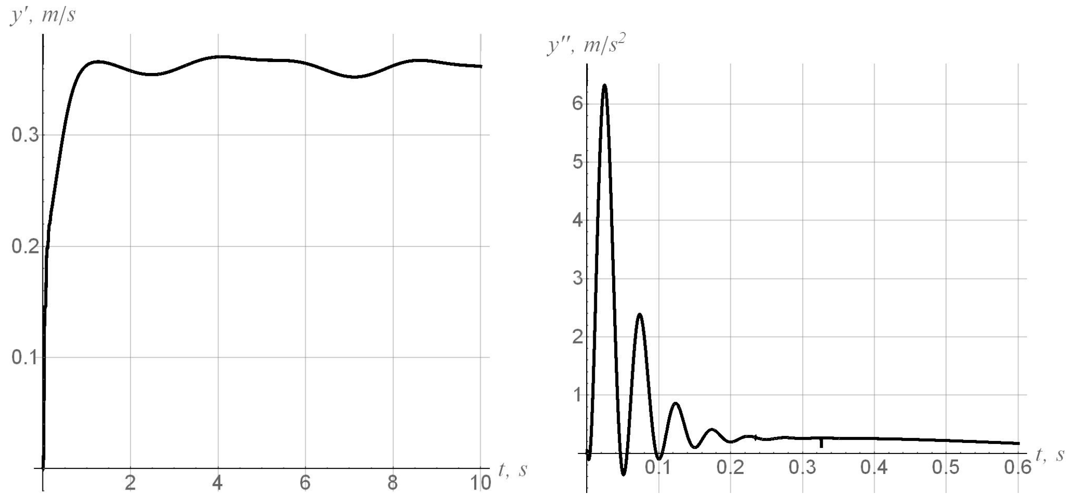


Fig. 8. The graph of the rate of change of cargo departure



(a) (b)

Fig. 9. The graph of change in speed (a) and acceleration (b) of lifting a load

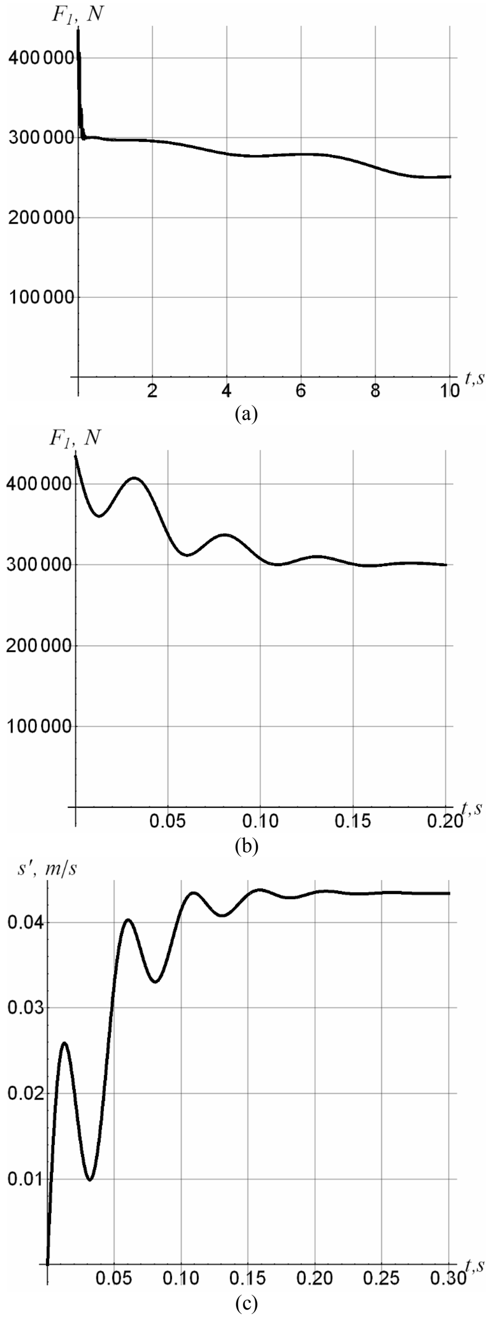


Fig. 10. The graph of change in the driving force (a, b) of the boom lifting cylinder and the speed (c) of the rod extension

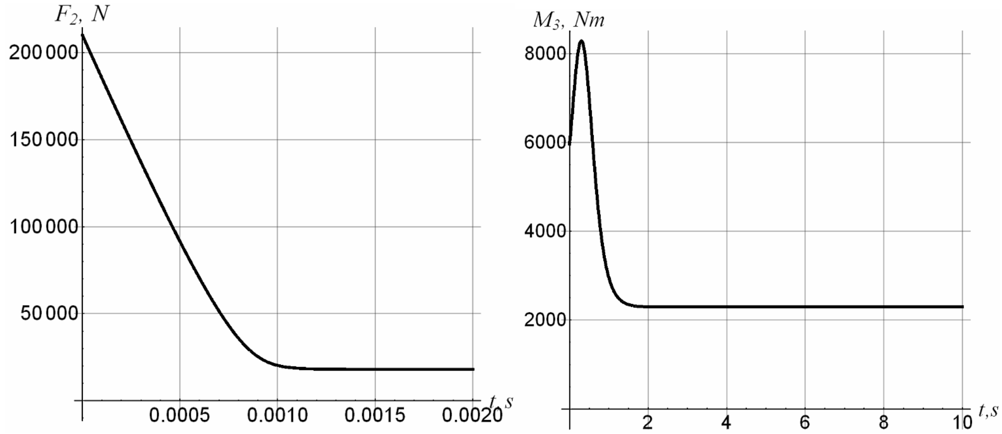
The drive power of the boom lifting mechanism (Fig. 12, a) increases in the oscillatory mode to a steady-state value of 13.0 kW. At the same time, the oscillations of the drive power decay in 0.2 s, i.e. during the start-up process. The drive power of the mechanism for moving the boom section (Fig. 12, b, c) first instantly increases to a maximum value of 7.0 kW, and then also decreases to a minimum value of 1.9 kW (Fig. 12, c). After that, the drive power gradually increases in the oscillatory mode (caused by oscillations of the load on the flexible suspension) to 2.7 kW (Fig. 12,b). The drive power of the load lifting mechanism (Fig. 12, d) first gradually increases to a maximum value of 15.0 kW, and then also gradually decreases to a steady-state value of 8.0 kW.

Conclusions. Based on the results of studies of the dynamics of the joint movement of the mechanisms for turning the boom, extending its section and lifting the load at a steady-state turn of a jib crane, the following conclusions are proposed:

1. A dynamic model of the combined movement of the mechanisms for turning the boom, extending its section and lifting the load during the steady-state turning a hydraulically driven jib crane is proposed. This dynamic model presents the main movement of the drive mechanisms for turning the boom, extending its section and lifting the load during a steady-state turn of the crane. The elastic-dissipative properties of the traction rope of the load lifting mechanism, as well as low-frequency pendulum oscillations of the load on a flexible suspension, are also taken into account. The dynamic model takes into account the characteristics of the driving forces of the hydraulic cylinders for turning the boom and extending its section, as well as the hydraulic drive of the load lifting mechanism. On the basis of the developed dynamic model of the boom system, a mathematical model was constructed, which is described by a system of nonlinear differential equations of the second order. To solve the system of equations, numerical methods applied using the developed computer program.

2. On the basis of solving the mathematical model, a dynamic analysis of the joint start-up of the mechanisms for turning the boom, extending its section and lifting the load at a steady-state turn of the jib crane was carried out. As a result of the dynamic analysis, oscillatory processes and increased dynamic loads in the drives and structure of a jib crane were revealed in the process of simultaneous start-up of the mechanisms for turning the boom, extending its section and lifting the load during a steady-state turn of the crane. The instantaneous change in the driving forces of the drive mechanisms at the beginning of the start-up led to high-frequency oscillations of the kinematic and dynamic

characteristics of the boom system, which quickly dampen. The high-frequency oscillations at the beginning of the start-up of the drive mechanisms are the source of almost unattenuated low-frequency oscillations of the load on the flexible suspension. The process of dampening these vibrations leads to a decrease in productivity and an increase in energy consumption of the crane. To reduce low-frequency oscillations, it is necessary to select the appropriate starting modes for drive mechanisms.



(a) (b)

Fig. 11. The graph of the driving force of the hydraulic cylinder of the boom extension section (a) and the driving torque on the drum of the lifting mechanism (b)

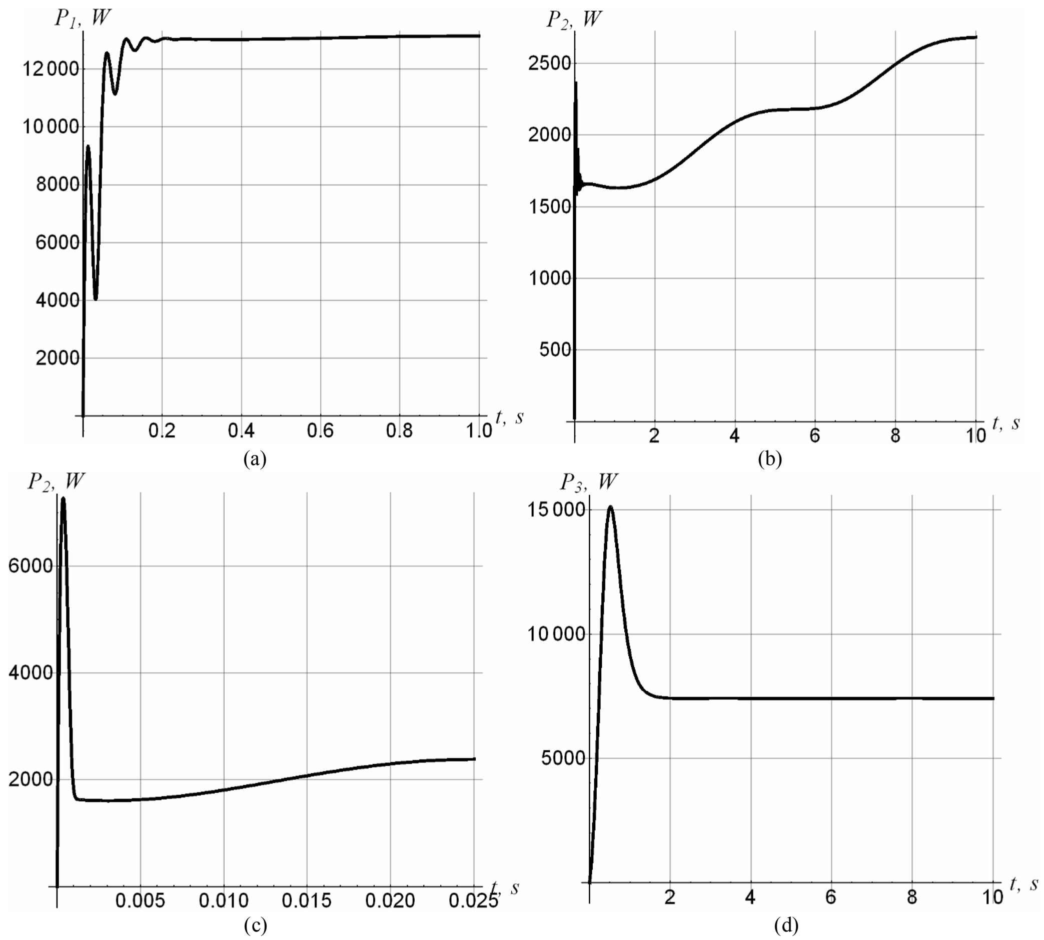


Fig. 12.Power change graphs for boom lifting (a), boom section extension (b, c) and load lifting (d)

3. As a result of the dynamic analysis of the simultaneous start-up of the mechanisms for turning the booms, extending its section and lifting the load during the steady-state rotation of the jib crane, the cause of the oscillatory processes was established, which is a rapid change in the driving forces of the drives. To reduce oscillations in the design of the boom system, it is necessary to increase the smoothness of the change in the driving forces of the drive mechanisms.

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REFERENCES

1. *Herasymiak R.P., Naidenko O.V.* Osoblyvosti keruvannia elektropryvodom mekhanizmu vylotu strily pid chas obertanniakrana z pidvishenym vantazhem (Features of controlling the electric drive of the boom extension mechanism during rotation of the crane with a suspended load). - Elektromashynobuduvannia ta elektroobladnannia. 2007. Vyp. 68. S. 11–15.
2. *Loveikin V., Romasevych Y., Loveikin A., Liashko A., Pochka K.* Dynamic analysis of the simultaneous starting of the boom and load lifting mechanisms hoisting for the jib and the cargo of the jib crane a hydraulic for drive. Journal Strength of Materials and Theory of Structures. 2024. №113. pp. 149-160.DOI: 10.32347/2410-2547.2024.113.149-160.
3. *Doçi I., Shpetim L.* Rotational motion of tower crane – dynamic analysis and regulation using schematic modeling. International Scientific Journal "Mathematical Modeling". 2018. Issue 1. pp. 21-25.
4. *Stölzner M., Kleeberger M., Moll M., Fottner J.* Investigation of the dynamic loads on tower cranes during slewing operations. SIMULTECH 2020 - Proceedings of the 10th International Conference on Simulation and Modeling Methodologies. Technologies and Applications. 2020. pp. 59–67.
5. *Loveikin V., Romasevych Y., Loveikin A., Shymko L., Liashko A.* Minimization of the drive torque of the trolley movement mechanism during tower crane steady slewing. Journal of Theoretical and Applied Mechanics. 2023. Vol. 53. pp. 19–33. DOI: 10.55787/jtams.23.53.1.19.
6. *Ju F., Choo Y.S., Cui F.S.* Dynamic response of tower crane induced by the pendulum motion of the payload. International Journal of Solids and Structures. 2006. Vol. 43(2). pp. 376–389. DOI: 10.1016/j.ijsolstr.2005.03.078.
7. *BalogunWasiu Adebayo, Mohamed Z., Abdullahi A.M., FasihurRehman S.M.* Design and real-time implementation of a distributed-delay input shaper for sway control of a double-pendulum overhead crane. International Journal of Mechatronics and Manufacturing Systems. 2023. Vol. 16. No. 4. pp. 364-380. DOI: 10.1504/IJMMS.2023.137371.
8. *Jaafar H.I., Mohamed Z., Ahmad M.A., Wahab N.A., Ramli L., Shaheed M.H.* Control of an underactuated double-pendulum overhead crane using improved model reference command shaping: design, simulation and experiment. Mechanical Systems and Signal Processing. 2021. Vol. 151. pp. 107358. DOI: 10.1016/j.ymssp.2020.107358.
9. *Oguamanam D.C.D., Hansen J.S., Heppler G.R.* Dynamics of a three-dimensional overhead crane system. Journal of Sound and Vibration. 2001. Vol. 242(3). pp. 411–426. DOI: 10.1006/jsvi.2000.3375.
10. *Bello M.M., Mohamed Z., Efe M.Ö., Ishak H.* Modelling and dynamic characterisation of a double-pendulum overhead crane carrying a distributed-mass payload. Simulation Modelling Practice and Theory. 2024. Vol. 134. Article 102953. DOI: 10.1016/j.simpat.2024.102953.
11. *Grigorov O., Druzhynin E., Strizhak V., Strizhak M., Anishchenko G.* Numerical simulation of the dynamics of the system "trolley - load - carrying rope" in a cable crane. Eastern-European Journal of Enterprise Technologies. 2018. Vol. 3. Issue 7-93. pp. 6–12. DOI: 10.15587/1729-4061.2018.132473.
12. *Wu M., Li L., Li Y.* Dynamic analysis of a container crane considering the coupling effect between spreader and cargo. Journal of Vibroengineering. 2019. Vol. 21. Issue 2. March. pp. 360-373.
13. *Ambrosino M., Berneman M., Carbone G., Dawans A., Garone E.* Modeling and control of a crane with a 5-boom reach. ISARC: Proceedings of the International Symposium on Automation and Robotics in Construction. Kitakyushu. 2020. pp. 25-30.
14. *Rigatos G., Abbaszadeh M., Pomares J.* Nonlinear optimal control for the 4-DOF underactuated robotic tower crane. Autonomous Intelligent Systems. 2022. Vol. 2(1). Article 21. DOI: 10.1007/s43684-022-00040-4.
15. *Buczkowski R., Żyliński B.* Finite element fatigue analysis of unsupported crane. Polish Maritime Research. 2021. Vol. 28(1). pp. 127-135. DOI: 10.2478/pomr-2021-0012.
16. *Yang T., Sun N., Chen H., Fang Y.* Neural network-based adaptive anti-swing control of an underactuated ship-mounted crane with roll motions and input dead zones. IEEE Transactions on Neural Networks and Learning Systems. 2019. DOI: 10.1109/TNNLS.2019.2910580.
17. *Qian Y., Hu D., Chen Y., Fang Y., Hu Y.* Adaptive Neural Network-Based Tracking Control of Underactuated Offshore Ship-to-Ship Crane Systems Subject to Unknown Wave Motion Disturbances. IEEE Transactions on Systems, Man, and Cybernetics: Systems. 2022. Vol. 52. No. 6. pp. 3626-3637. DOI: 10.1109/TSMC.2021.3071546.
18. *Fu Liu, Yang J., Wang J., Liu C.* Swing Characteristics and Vibration Feature of Tower Cranes under Compound Working Condition. Shock and Vibration. 2021. Article 8997396. DOI: 10.1155/2021/8997396.
19. *Ye J., Huang J.* Control of pendulum beam dynamics in a thin boom tower crane transporting a distributed mass load. IEEE Transactions on Industrial Electronics. 2023. Vol. 70(1). pp. 888-897.
20. *Loveikin V.S., RomasevychYu.O., LoveikinA.V., LiashkoA.P., Pochka K.I., Korobko M.M.*Analysis of derrikingand slewing of the tower crane with consideration to driving mechanisms characteristics. Strength of Materials and Theory of Structure: Scientific and technical collected articles. 2020. Vol. 110. pp. 316-327. DOI: 10.32347/2410-2547.2023.110.316-327.
21. *Loveikin V.S., RomasevychYu.O., LoveikinA.V., LiashkoA.P., Pochka K.I., Balaka M.M.*Drive power minimization of outreach change mechanism of tower crane during steady-state slewing mode. Strength of Materials and Theory of Structure: Scientific and technical collected articles. 2020. Vol. 109. pp. 317-330. DOI: 10.32347/2410-2547.2022.109.317-330.

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Ловейкін В.С., Ромасевич Ю.О., Ловейкін А.В., Ляшко А.П., Почка К.І.

ДИНАМІЧНИЙ АНАЛІЗ СУМІСНОГО РУХУ МЕХАНІЗМІВ ЗМІНИ ВИЛЬОТУ ТА ПІДЙОМУ ВАНТАЖУ ПРИ УСТАЛЕНОМУ ПОВОРОТІ СТРІЛОВОГО КРАНА

Актуальною проблемою покращення роботи стрілових кранів є підвищення їхньої продуктивності. Одним із шляхів підвищення продуктивності стрілових кранів є суміщення роботи окремих механізмів. Метою дослідження є побудова математичної моделі та проведення динамічного аналізу стрілової системи крана при одночасній роботі механізмів зміни вильоту та підйому вантажу при усталеному повороті крана. В проведених дослідженнях використано методи побудови дискретних динамічних моделей стрілового крана шляхом використання рівнянь Лагранжа другого роду, чисельні методи розв'язування отриманих диференціальних рівнянь, які при усталеному повороті крана представлені у вигляді комп'ютерної програми та методи динамічного аналізу кранових механізмів. В представленій роботі вирішується задача дослідження динаміки одночасного руху механізмів повороту стріли, висування її секції та підйому вантажу при усталеному повороті крана. Для дослідження динамічних процесів в гідро-механічній системі стрілового крана при одночасній роботі кранових механізмів розроблено методику динамічного аналізу. Стрілова система крана представлена динамічноюмоделлю з шістьома ступенями вільності, де враховано основний рух механізмів і коливання ланок та вантажу на гнучкому підвісі. На основі побудованої математичної моделі визначені кінематичні, динамічні та енергетичні характеристики окремих ланок стрілової системи крана при одночасній роботі декількох механізмів. Досліджено високочастотні коливання ланок приводу механізму підйому вантажу та низькочастотні коливання вантажу на гнучкому підвісі. Встановлено, що високочастотні коливання ланок затухають в межах процесу пуску, а низькочастотні коливання вантажу практично не затухають і тривають протягом всього циклу руху.

Для мінімізації коливальних процесів одночасного руху механізмів стрілової системи рекомендовано обирати режими руху приводів, які забезпечують плавний рух виконавчих елементів, що приводить до зменшення навантажень та підвищення надійності роботи крана.

Ключові слова: механізми повороту стріли, висування секції стріли, підйому вантажу, коливальні процеси, динамічні навантаження.

Loveikin V.S., Romasevych Yu.O., Loveikin A.V., Liashko A.P., Pochka K.I.

DYNAMIC ANALYSIS OF THE JOINT MOVEMENT OF DERRICKING MECHANISM AND LIFTING MECHANISM OF A LOAD DURING A STEADY-STATE TURN OF A JIB CRANE

Increasing the productivity of jib cranes is an urgent problem of improving their operation. Combining the work of separate mechanisms is one of the ways to increase the productivity of jib cranes. The aim of the study is to build a mathematical model and conduct a dynamic analysis of the crane jib system with simultaneous operation of the derricking mechanism and lifting mechanism of the load during a steady-state crane rotation. Methods for constructing discrete dynamic models of a jib crane by using Lagrange equations of the second kind, numerical methods for solving the obtained differential equations, which are presented in the form of a computer program at a steady-state crane rotation, and methods for dynamic analysis of crane mechanisms are used in the conducted research. The task of researching the dynamics of the simultaneous movement of the mechanisms for turning the boom, extending its section and lifting the load during a steady-state crane turn is solved in the presented work. The method of dynamic analysis was developed to study dynamic processes in the hydro-mechanical system of a jib crane during the simultaneous operation of crane mechanisms. The crane boom system is represented by a dynamic model with six degrees of freedom, which takes into account the main movement of the mechanisms and the oscillations of the links and the load on a flexible suspension. The kinematic, dynamic, and energy characteristics of individual links of the crane jib system with simultaneous operation of several mechanisms are determined on the basis of the constructed mathematical model. The high-frequency oscillations of the drive links of the load lifting mechanism and the low-frequency oscillations of the load on a flexible suspension are investigated. It was found that high-frequency vibrations of the links damped within the start-up process, while low-frequency vibrations of the load practically did not damp and continued throughout the entire movement cycle.

Drive modes that ensure smooth movement of the actuators, which leads to reduced loads and increased reliability of the crane are recommended to minimise oscillatory processes of simultaneous movement of the jib system mechanisms.

Keywords: boom turning mechanisms, boom section extension, load lifting, oscillatory processes, dynamic loads.

УДК 621.87

Ловейкін В.С., Ромасевич Ю.О., Ловейкін А.В., Ляшко А.П., Почка К.І. **Динамічний аналіз сумісного руху механізмів зміни вильоту та підйому вантажу при усталеному повороті стрілового крана** // Опірматеріалів і теоріяспоруд: наук.-техн. збірник. – К.: КНУБА, 2025. –Вип. 114. – С. 111-126.

Проблемою підвищення продуктивності стрілових кранів є суміщення роботи їхніх механізмів. Досліджено динаміку одночасного руху механізмів зміни вильоту, підйому вантажу та повороту крана. Розроблено математичну модель із шістьома ступенями вільності та проведено динамічний аналіз. Визначено кінематичні, динамічні та енергетичні характеристики системи. Рекомендовано режими руху приводів для мінімізації коливань і підвищення надійності крана.

Іл. 12.Бібліогр. 19 назв.

UDC 621.87

Loveikin V.S., Romasevych Yu.O., Loveikin A.V., Liashko A.P., Pochka K.I. **Dynamic analysis of the joint movement of derricking mechanism and lifting mechanism of a load during a steady-state turn of a jib crane**// Strength of Materials and Theory of Structure: Scientific and technical collected articles. – K.: KNUCA, 2025. – Issue 114. – P. 111-126.

Increasing jib crane productivity requires combining the work of its mechanisms. This study develops a mathematical model and dynamic analysis of the jib system with simultaneous operation of derricking and lifting mechanisms during steady-state rotation. A six-degree-of-freedom model accounts for mechanism movements and oscillations. High- and low-frequency oscillations were analyzed, revealing persistent low-frequency load vibrations. Smooth drive modes are recommended to reduce loads and improve crane reliability.

Fig. 12. Ref. 19.

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