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# DYNAMIC ANALYSIS OF THE SIMULTANEOUS STARTING OF THE BOOM AND LOAD LIFTING MECHANISMS HOISTING FOR THE JIB AND THE CARGO OF THE JIB CRANE WITH A HYDRAULIC DRIVE

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Combining the work of individual mechanisms is carried out to improve the productivity of jib cranes.In particular, the task of joint starting of hoisting mechanisms of jib and cargo is considered in this article. Dynamic loads on crane structural elements, drives, and cargo on a flexible suspension increase with such movement of mechanisms. Increased loads lead to a decrease in the reliability of the crane and an increase in energy costs. When researching the dynamics of the joint starting of hoisting mechanisms of jib and cargo, the jibs system is represented by a mechanical system with four degrees of freedom, which takes into account the main movement of the mechanisms and the oscillatory movement of links with elastic and dissipative properties, as well as the cargo on a flexible suspension in the plane of departure change. A mathematical model of the joint movement of hoisting mechanisms of the jib and cargo was built for such a dynamic model of the boom system of the crane. The obtained model is a system of nonlinear differential equations of the second order, the solution of which was carried out by a numerical method in the form of a computer program. The dynamic of the joint starting of the hoisting mechanisms of the jib and cargo in the jib crane with specific numerical parameters were calculated and studied based on the developed program. The performed calculation made it possible to conduct a dynamic analysis of the joint starting of the hoisting mechanisms. High-frequency oscillations of links with elastic and dissipative properties in the drive of the hoisting mechanism were found as a result of the analysis. Also, low-frequency vibrations of the load on the flexible suspension were found. In the process of starting the mechanisms, high-frequency oscillations are attenuated during the transition process, and low-frequency oscillations of the load are sufficiently long and are attenuated during a significant period.

It is recommended to optimize the starting and braking modes to improve the dynamic properties of the hoisting mechanisms of the jib and cargo during their joint movement.

**Keywords:** jib crane, drives, hoisting mechanisms of the jib and cargo, dynamic loads, flexible load suspension.

**Introduction.** The need to increase the productivity of carrying out cargo operations arises when using mobile jib cranes. Combining the simultaneous operation of several mechanisms is one of the ways to increase the productivity of jib cranes. Quite often, the operation of the hoisting mechanisms of the jib and cargo are combined during the operation of jib cranes. In this case, the dynamic loads that reduce the reliability of the crane as a whole and its mechanisms and lead to higher energy costs increase in the elements of the crane's drive mechanisms and construction. Dynamic loads during the combination of transient processes (starting, braking) of individual mechanisms, in particular hoisting the boom and cargo, are particularly dangerous.

In connection with this, there is a need to conduct studies of the dynamics of the simultaneous starting of the cargo and jib hoisting mechanisms. High-frequency oscillations of drive elements and low-frequency oscillations of the load on a flexible suspension have a special effect on the dynamics of starting mechanisms.

Therefore, the task of researching dynamic loads at the simultaneous start of cargo and jib hoisting mechanisms is relevant, as it will provide an opportunity to establish real loads when using jib cranes in intensive work conditions.

**Analysis of publications.** 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 the study is to detect dynamic loads and oscillations in the structural elements of the jib crane by conducting a dynamic analysis of the joint movement of the mechanisms hoisting for the jib and the cargo.

**Research results.** The dynamic model of the boom system of the crane is presented in the form of absolutely solid and elastic-dissipative links of the cargo and jib hoisting mechanisms. We assume that the rope of the cargo hoisting mechanism has elastic-dissipative properties, and the jib, cargo and drive elements are absolutely solid bodies. The flexible cargo suspension in the form of a polyspast system carries out pendulum oscillations in the plane of change of departure and has dissipative properties. The drive links of the cargo hoisting mechanism are brought to the axis of the drum on which the cargo rope is wound. The drum is driven by a hydraulic motor, and the jib is lifted by a hydraulic cylinder.

So, the dynamic model of the jib system with the joint movement of the hoisting mechanisms of the cargo and jib is represented by a mechanical system with four degrees of freedom (Fig. 1). The angular coordinates of the rotation of the jib $\alpha$, the drive drum of the cargo hoisting mechanism $\beta$, and the deviation from the vertical of the flexible suspension of the load ν, as well as the linear coordinate of its length $u$ are used for the generalized coordinates of the given model. The resistance forces from the weight of the jib and the cargo, as well as the driving force in the drive hydraulic cylinder of the jib hoisting mechanism, which creates the moment $M_1$ and the moment of the hydraulic motor of the cargo hoisting mechanism, reduced to the axis of the drive drum $M_2$ act on the boom system. In addition, elastic and dissipative forces act on the elastic rope of the cargo hoisting mechanism, and the dissipative moment acts when the flexible suspension of the cargo deviates from the vertical.

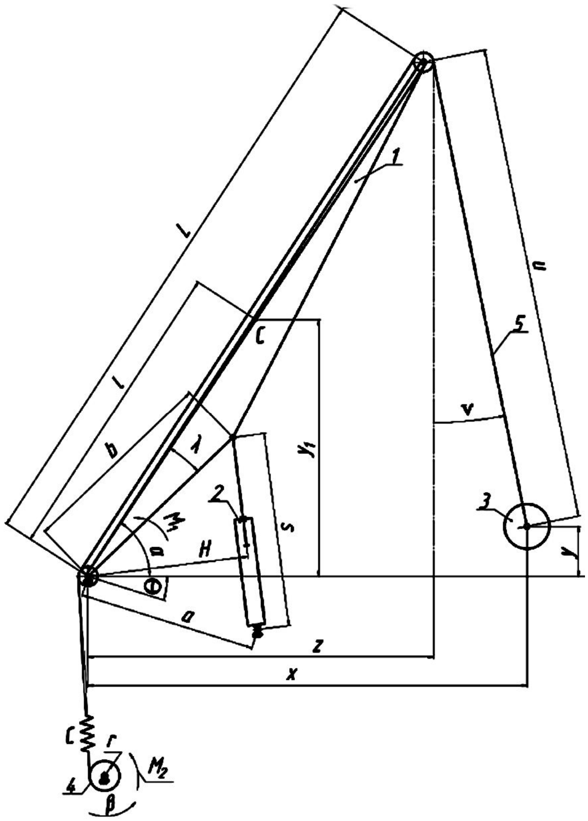


Fig. 1. The dynamic model of the jib system of the crane with the joint movement of the hoisting mechanisms of the cargo and jib

When compiling the equations of the joint movement of the mechanisms for hoisting the jib and the cargo (Fig. 1), we will use the Lagrange equations of the second kind:

$$\begin{aligned}
&\frac{d}{dt}\frac{\partial T}{\partial \dot{\alpha}}-\frac{\partial T}{\partial \alpha}=M_1-\frac{\partial \Pi}{\partial \alpha};\\
&\frac{d}{dt}\frac{\partial T}{\partial \dot{\beta}}-\frac{\partial T}{\partial \beta}=M_2-\frac{\partial \Pi}{\partial \beta}-\frac{\partial R}{\partial \dot{\beta}};\\
&\frac{d}{dt}\frac{\partial T}{\partial \dot{u}}-\frac{\partial T}{\partial u}=-\frac{\partial \Pi}{\partial u}-\frac{\partial R}{\partial \dot{u}};\\
&\frac{d}{dt}\frac{\partial T}{\partial \dot{v}}-\frac{\partial T}{\partial v}=-\frac{\partial \Pi}{\partial v}-\frac{\partial R}{\partial \dot{v}},
\end{aligned} \tag{1}$$

where $T$, $\Pi$, $R$ are the kinetic and potential energy of the system and the dissipative Rayleigh function, respectively; $M_1$, $M_2$ - driving moment of the drives of mechanisms hoisting for jib and cargo, reduced to the pivoting jib and the drive drum of the cargo hoisting mechanism.

The kinetic energy of the joint motion of the mechanisms hoisting for jib and cargo is expressed as follows

$$T=\frac{1}{2}J_1\dot{\alpha}^2+\frac{1}{2}J_2\dot{\beta}^2+\frac{1}{2}m\left(\dot{x}^2+\dot{y}^2\right), \tag{2}$$

where $J_1$, $J_2$ are, respectively, moments of inertia of the jib and drive of the cargo hoisting mechanism reduced to the axis of rotation of the drive drum; $m$ – the weight of cargo; $x$, $y$ - horizontal and vertical coordinates of the center of mass of the cargo in the plane of derricking change.

The potential energy of the joint movement of the mechanism hoisting for jib and cargo is determined by this dependence

$$\Pi=\frac{1}{2}c\left[\beta r-\left(u_0-u\right)n\right]^2+\left(m_1y_1+my\right)g\ , \tag{3}$$

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; $m$, $y_1$ – mass of the arrow and its vertical coordinate of the center of mass; $u_0$ is the initial length of the flexible cargo suspension.

The dissipative function of the Relay system has the following form

$$R = \frac{1}{2}b_1\left(\dot{\beta}r - n\dot{u}\right)^2 + \frac{1}{2}b_2\dot{v}^2 \ , \tag{4}$$

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.

Let's find the coordinates of the centers of mass cargo and jib:

$$x = L\cos\alpha + u\sin v; \quad y = L\sin u - u\cos v\,; \tag{5}$$

$$y_1 = \frac{1}{2}L\sin\alpha, \tag{6}$$

where $L$ is the length of the jib.

Expressions (5) show that the position of the cargo depends on three generalized coordinates: $\alpha$, $v$ and $u$.

Let's take the partial derivatives of the kinetic energy (2) in terms of the joint coordinates of the jib system:

$$\begin{aligned} &\frac{\partial T}{\partial \alpha} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial \alpha} + \dot{y}\frac{\partial \dot{y}}{\partial \alpha}\right); \quad \frac{\partial T}{\partial u} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial u} + \dot{y}\frac{\partial \dot{y}}{\partial u}\right); \\ &\frac{\partial T}{\partial v} = m\left(\dot{x}\frac{\partial \dot{x}}{\partial v} + \dot{y}\frac{\partial \dot{y}}{\partial \alpha}\right); \quad \frac{\partial T}{\partial \beta} = 0. \end{aligned} \tag{7}$$

Let us also take the partial derivatives of the kinetic energy (2) in terms of the joint velocities of the system:

$$\begin{gathered} \frac{\partial T}{\partial \dot{\alpha}} = J_1\dot{\alpha} + m\left(\dot{x}\frac{\partial x}{\partial \alpha} + \dot{y}\frac{\partial x}{\partial \alpha}\right); \quad \frac{\partial T}{\partial \dot{u}} = m\left(\dot{x}\frac{\partial x}{\partial u} + \dot{y}\frac{\partial x}{\partial u}\right); \\ \frac{\partial T}{\partial \dot{v}} = m\left(\dot{x}\frac{\partial x}{\partial v} + \dot{y}\frac{\partial x}{\partial v}\right); \quad \frac{\partial T}{\partial \dot{\beta}} = J_2\dot{\beta}. \end{gathered} \tag{8}$$

Now let's take the complete time derivatives of expressions (8):

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\alpha}} = J_1\ddot{\alpha} + m\left(\ddot{x}\frac{\partial x}{\partial \alpha} + \ddot{y}\frac{\partial y}{\partial \alpha} + \dot{x}\frac{\partial \dot{x}}{\partial \alpha} + \dot{y}\frac{\partial \dot{y}}{\partial \alpha}\right); \tag{9}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{u}} = m\left(\ddot{x}\frac{\partial x}{\partial u} + \ddot{y}\frac{\partial y}{\partial u} + \dot{x}\frac{\partial \dot{x}}{\partial u} + \dot{y}\frac{\partial \dot{y}}{\partial u}\right); \tag{10}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{v}} = m\left(\ddot{x}\frac{\partial x}{\partial v} + \ddot{y}\frac{\partial y}{\partial v} + \dot{x}\frac{\partial \dot{x}}{\partial v} + \dot{y}\frac{\partial \dot{y}}{\partial v}\right); \tag{11}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\beta}} = J_2\ddot{\beta}\,. \tag{12}$$

Let's find the partial derivatives of the potential energy (3) in joint coordinates

$$\frac{\partial \Pi}{\partial \alpha} = \left(m_1\frac{\partial y_1}{\partial \alpha} + m\frac{\partial y_1}{\partial \alpha}\right)g; \quad \frac{\partial \Pi}{\partial \beta} = cr\left[\beta r - (u_0 - u)n\right]; \tag{13}$$

$$\frac{\partial \Pi}{\partial u} = cr\left[\beta r - (u_0 - u)n\right] + mg\tfrac{\partial y}{\partial u}; \quad \frac{\partial \Pi}{\partial v} = mg\tfrac{\partial y}{\partial v}. \tag{14}$$

Let us also take the partial derivatives of the Rayleigh function (4) in terms of the joint velocities:

$$\frac{\partial R}{\partial \dot{\beta}} = b_1 r\left(\dot{\beta}r - n\dot{u}\right); \quad \frac{\partial R}{\partial \dot{u}} = b_1 n\left(\dot{\beta}r - n\dot{u}\right); \quad \frac{\partial R}{\partial \dot{v}} = b_2\dot{v}. \tag{15}$$

As a result of substituting expressions (7), (9),...,(15) into the system (1), we obtain the differential equations of the simultaneous movement of the mechanisms hoisting for jib and cargo:

$$J_1\ddot{\alpha}+m\left[\ddot{x}\frac{\partial x}{\partial\alpha}+(g+\ddot{y})\frac{\partial y}{\partial\alpha}\right]=M_1-m_1g\frac{\partial y_1}{\partial\alpha};\; J_2\ddot{\beta}=M_2-cr\left[\beta r-(u_0-u)n\right]-b_1r\left(\dot{\beta}r-n\dot{u}\right);$$
$$m\left[\ddot{x}\frac{\partial x}{\partial u}+(g+\ddot{y})\frac{\partial y}{\partial u}\right]=-cn\left[\beta r-(u_0-u)n\right]-b_1n\left(\dot{\beta}r-n\dot{u}\right);\; m\left[\ddot{x}\frac{\partial x}{\partial v}+(g+\ddot{y})\frac{\partial y}{\partial u}\right]=-b_2\dot{v}. \tag{16}$$

Partial and complete derivatives of the coordinates of individual links of the jib system are included in the system of equations (16). Let's determine these derivatives for the dynamic model of the jib system presented in Fig. 1. Let's find the partial derivatives of the jib and cargo coordinates:

$$\frac{\partial y_1}{\partial\alpha}=\frac{1}{2}L\cos\alpha; \tag{17}$$

$$\frac{\partial x}{\partial\alpha}=-L\sin\alpha;\quad \frac{\partial y}{\partial\alpha}=L\cos\alpha; \tag{18}$$

$$\frac{\partial x}{\partial u}=\sin v;\quad \frac{\partial y}{\partial u}=-\cos v; \tag{19}$$

$$\frac{\partial x}{\partial v}=u\cos v;\quad \frac{\partial y}{\partial v}=u\sin v; \tag{20}$$

$$\frac{\partial^2 x}{\partial\alpha^2}=-L\cos\alpha;\quad \frac{\partial^2 y}{\partial\alpha^2}=-L\sin\alpha; \tag{21}$$

$$\frac{\partial^2 x}{\partial v^2}=-u\sin\alpha;\quad \frac{\partial^2 y}{\partial v^2}=u\cos\alpha; \tag{22}$$

$$\frac{\partial^2 x}{\partial u\partial v}=\cos v;\quad \frac{\partial^2 y}{\partial u\partial v}=\sin v; \tag{23}$$

$$\frac{\partial^2 x}{\partial\alpha\partial u}=\frac{\partial^2 y}{\partial\alpha\partial u}=\frac{\partial^2 x}{\partial\alpha\partial v}=\frac{\partial^2 y}{\partial\alpha\partial v}=0. \tag{24}$$

Let's find the first complete time derivatives of the cargo coordinates from expressions (5):

$$\dot{x}=\dot{\alpha}\frac{\partial x}{\partial\alpha}+\dot{u}\frac{\partial x}{\partial u}+\dot{v}\frac{\partial x}{\partial v};\quad \dot{y}=\dot{\alpha}\frac{\partial y}{\partial\alpha}+\dot{u}\frac{\partial y}{\partial u}+\dot{v}\frac{\partial y}{\partial v}. \tag{25}$$

Taking into account the second partial derivatives of the coordinates given in (21),...,(24), we present the second-time derivatives of the cargo coordinates:

$$\ddot{x}=\ddot{\alpha}\frac{\partial x}{\partial\alpha}+\ddot{u}\frac{\partial x}{\partial u}+\ddot{v}\frac{\partial x}{\partial v}+2\dot{u}\dot{v}\frac{\partial^2 x}{\partial u\partial v}+\dot{\alpha}^2\frac{\partial^2 x}{\partial\alpha^2}+\dot{v}\frac{\partial^2 x}{\partial v^2}; \tag{26}$$

$$\ddot{y}=\ddot{\alpha}\frac{\partial y}{\partial\alpha}+\ddot{u}\frac{\partial y}{\partial u}+\ddot{v}\frac{\partial y}{\partial v}+2\dot{u}\dot{v}\frac{\partial^2 y}{\partial u\partial v}+\dot{\alpha}^2\frac{\partial^2 y}{\partial\alpha^2}+\dot{v}\frac{\partial^2 y}{\partial v^2}. \tag{27}$$

Through the force in the hydraulic cylinder $F_1$, we determine the torque that turns the crane jib:

$$M_1=F_1h. \tag{28}$$

There

$$h=\frac{\sqrt{4a^2s^2-\left(a^2-b^2+s^2\right)^2}}{2s};\; s=\sqrt{a^2+b^2-2ab\cos(\theta-\lambda+\alpha)}.$$

The force in the drive hydraulic cylinder of the jib hoisting mechanism is determined as follows

$$F_1=p_nA\sqrt{1-\frac{A\dot{s}}{Q}}; \tag{30}$$

$$\dot{s}\doteq\frac{ab}{s}\sin(\theta-\lambda+\alpha). \tag{31}$$

Here $h$ is the shoulder of force $F_1$; $p_n$ is the working pressure in the hydraulic cylinder cavity; $A$ - cross-sectional area of the hydraulic cylinder piston; $Q$ - consumption of the working fluid by the hydraulic cylinder; $a$, $\theta$ - the length of the support of the hydraulic cylinder and the angle of its inclination to the horizon; $b$, $\lambda$ - the length of the hydraulic cylinder action lever on the jib and its angle of inclination to the jib axis; $s$ is the length of the hydraulic cylinder in working condition.

The driving moment on the shaft of the hydraulic motor of the cargo hoisting mechanism is represented by a quadratic dependence on the angular velocity of the shaft and is expressed by the following dependence:

$$M = M_p + \left( K\omega_0 - \frac{M_p}{\omega_0} \right) i\dot{\beta} - Ki^2\dot{\beta}^2 ; \tag{32}$$

$$K = \frac{M_n - M_p \left( 1 - \frac{\omega_n}{\omega_0} \right)}{\omega_n \left( \omega_0 - \omega_n \right)} , \tag{33}$$

where $M_p$, $M_n$ – starting and nominal torques of the hydraulic motor of the cargo hoisting mechanism, respectively; $\omega_n$, $\omega_0$ – nominal and synchronous angular velocity of the hydraulic motor shaft of the hoisting mechanism; $i$ – transmission ratio of the drive of the hoisting mechanism.

Then the drive moment of the cargo hoisting mechanism, reduced to the axis of the drive drum, is determined by the following relationship

$$M_2 = Mi\eta . \tag{34}$$

**Research results**

The system of second-order differential equations (16) together with expressions (17),...,(34) is nonlinear, so a numerical method was used to solve it. Solving the equations is carried out under the following initial conditions of the joint movement of the mechanisms hoisting for jib and cargo:

$$t = 0 : \alpha = \alpha_0 , \dot{\alpha} = 0, \beta = \frac{mg}{cnr} , \dot{\beta} = 0, u = u_0 , \dot{u} = 0, v = 0, \dot{v} = 0. \tag{35}$$

Research of the dynamics of the joint movement of the mechanisms hoisting for jib and cargo was carried out at the following values of the parameters of the jib system of the crane: $m$=4500 kg, $m_1$=2700 kg, $J_1$=72900 kg m$^2$, $J_2$=1183 kg m$^2$, $c$ =6,25· 10$^6$ Nm/rad, L=9,0 m, $i$=41,34, $\eta$=0,85, $p_n$=14· 10$^6$ N/m$^2$, $A$=0,0314 m$^2$, $Q$=0,00178 m$^3$/s, $\omega_0$=157 rad/s, $\omega_n$=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, $\lambda$=0,0872 rad, $\theta$=0,3189 rad, $\alpha_0$=0,5857 rad, $r$=0,208 m, $b_1$=1,2· $10^6 \frac{\text{Nm}}{\text{rad/s}}$.

As a result of the numerical solution of the nonlinear system of differential equations (16) under the initial conditions (35) of the process of the joint movement of the mechanisms hoisting for jib and cargo of the jib crane, graphical dependences of kinematic (Figs. 2–8), dynamic (Figs. 9, 10) and energy (Fig. 11, 12) characteristics.

It can be seen from Fig. 2 that the angular velocity of the jib increases rapidly from 0 to 0,036 rad/s, after which it increases slightly as the boom rises. In the section from the start of the launch up to 0,023 rad/s, high-frequency fluctuations in the jib rotation velocity are observed. High-frequency fluctuations in the velocity of rotation of the jib are caused by a sharp increase in the driving force of the drive of the jib hoisting mechanism in this section of the movement.

In contrast to the angular velocity of the jib, the angular velocity of the drive drum of the cargo hoisting mechanism (Fig. 3) smoothly increases from 0 to a steady value within 1,8 seconds. This is clearly shown by the graph of the change in the angular acceleration of the drive drum (Fig. 4). It should be noted that high-frequency acceleration fluctuations are observed in the starting area of the drive drum from 0 to 3,2 rad/s$^2$. Fluctuations in the acceleration of the drive drum are caused by a sharp increase in the drive torque of the cargo hoisting mechanism in this section of the movement.

It can be seen from Fig. 5 that at the beginning of the movement of the change in the length of the flexible suspension of the cargo, high-frequency fluctuations of the velocity take place, which quickly subside due to the damping properties of the rope of the cargo hoisting mechanism. Subsequently, the speed of lifting the load increases quite smoothly to a steady value within 1,8 s.

In the process of joint starting of the mechanisms hoisting for jib and cargo, pendulum oscillations of the cargo on the flexible suspension are observed in the plane of derricking change (Fig. 6). From the figure shown, it can be seen that in the initial phase of oscillations, the maximum deviation from the vertical of the flexible suspension of the load is 1.5 degrees. During the considered movement of the jib system of the crane, the pendulum oscillations of the cargo on the flexible suspension do not subside, as evidenced by the phase portrait of the oscillations shown in Fig. 7.

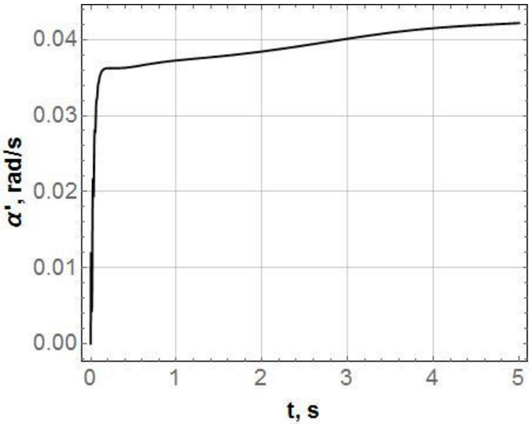


Fig. 1.The angular velocity of the jib

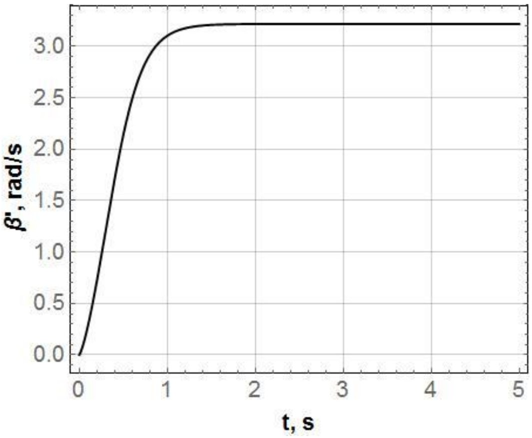


Fig.3. The driving drum angular velocity of the cargo hoisting mechanism

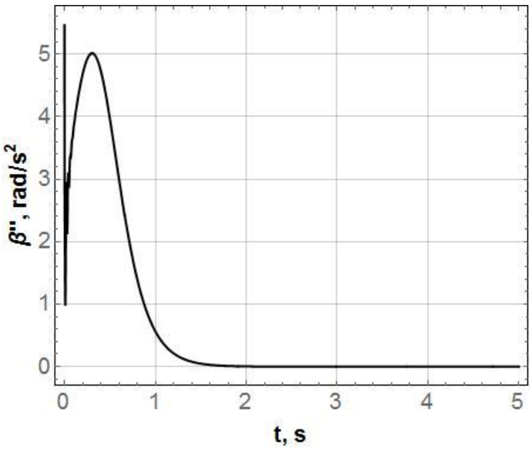


Fig. 4. The driving drum angular acceleration of the cargo hoisting mechanism

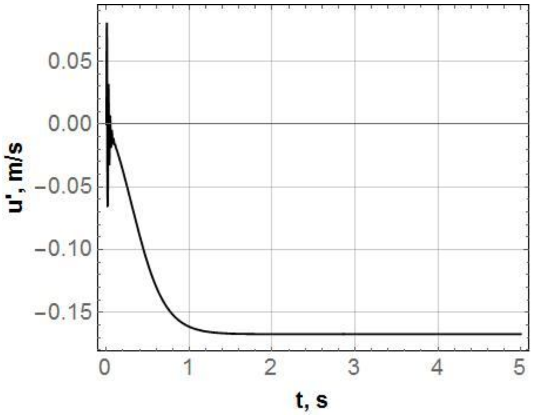


Fig. 5.The rate of change in the length of the flexible cargo suspension

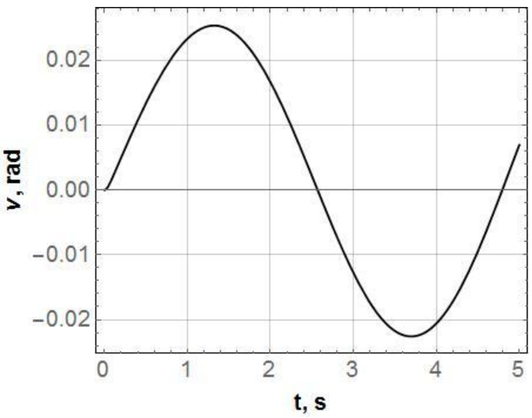


Fig. 6. The deviation from the vertical of the flexible cargo suspension

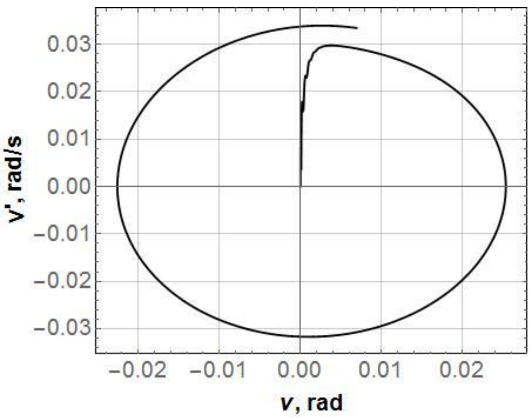


Fig. 7. The phase portrait of cargo oscillation on a flexible suspension

It can be seen from the graph of the change in the velocity of the cargo in the process of changing the departure (Fig. 8) that the departure of the cargo is carried out in an oscillating mode, which leads to a decrease in the performance of transport and installation operations with jib cranes. The reason for this movement of the cargo during the departure change is the pendulum oscillations of the cargo on the flexible suspension, which decay quite slowly.

The graph of the change in driving force in the hydraulic cylinder of the jib hoisting mechanism is shown in Fig. 9. It can be seen from the given graph that the driving force instantly increases to a maximum value of 480 kN, and then decreases to 300 kN at a sufficiently high velocity. In this section, high-frequency fluctuations of the driving force are observed. Later, when the jib is raised, the driving force in the hydraulic cylinder decreases monotonically.

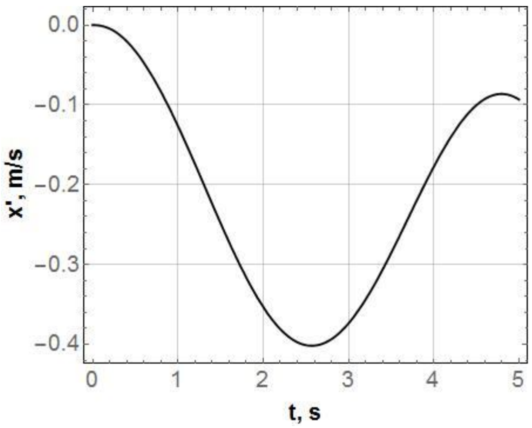

Fig. 8. The velocity of change of cargo departure

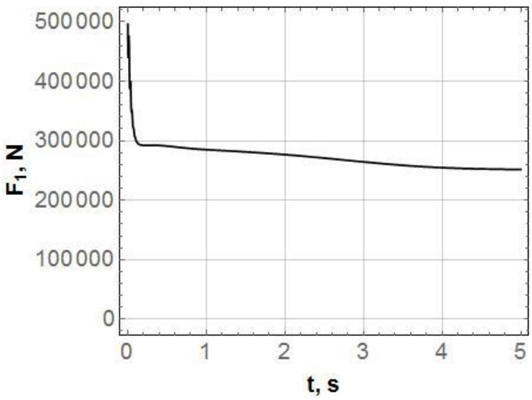

Fig. 9. The change in force in the hydraulic cylinder of the jib hoisting mechanism

The driving moment of the cargo hoisting mechanism drive (Fig. 10) from the starting value of 6000 Nm increases monotonically to the maximum value of 8125 Nm, after which it smoothly decreases to the established value of 2200 Nm. This mode of changing the driving moment ensures smooth movement of the cargo hoisting mechanism.

The drive power of the cargo hoisting mechanism (Fig. 11) increases almost instantly to the maximum value (16 kW), after which it monotonically decreases slightly according to a linear law as the jib rises.

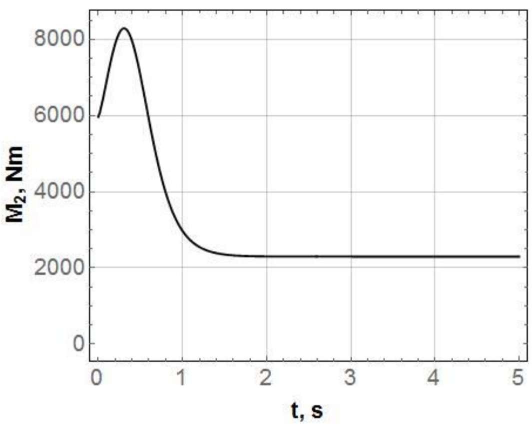

Fig. 10. The change in driving moment on the driving drum of the cargo hoisting mechanism

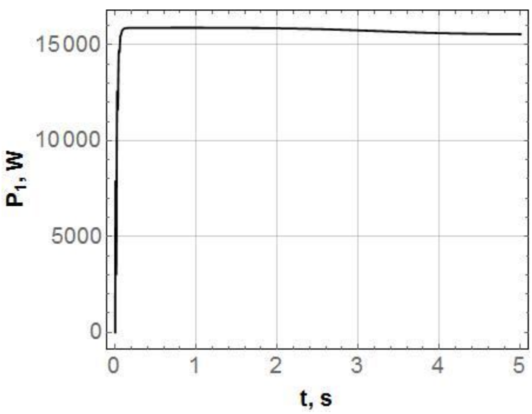

Fig. 11. The drive power of the jib hoisting mechanism

From fig. 12, it can be seen that the drive power of the cargo hoisting mechanism smoothly increases from zero to the maximum value (15 kW), and then also smoothly decreases to a steady value (7,5 kW). This nature of the change in the drive power ensures a smooth start of the cargo hoisting mechanism.

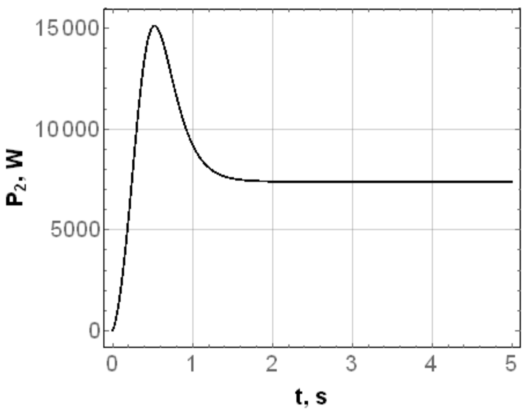

Fig. 12. The drive power of the cargo hoisting mechanism

**Conclusions.** The article presents the results of studies of dynamic processes in the joint starting of the mechanism hoisting for jib and cargo in the jib crane. As a result of the studies the following conclusions were made:

1. The dynamic model of the joint starting of the mechanisms hoisting for jib and cargo of the jib crane with a hydraulic drive is substantiated. The proposed model takes into account the main movement of the drive mechanisms for hoisting the jib and the cargo, as well as high-frequency oscillations of the rope of the cargo hoisting mechanism, which has elastic-dissipative properties. In addition, low-frequency pendulum oscillations of the cargo on a flexible suspension are taken into account. Models of driving forces of hydraulic drives with a hydraulic cylinder and a hydraulic motor have been built. Based on such a dynamic model, a mathematical model has been developed using second-order Lagrange equations, which is described by a system of nonlinear differential equations of the second order. These equations were solved numerically using a specially developed computer program.

2. On the basis of the calculations, a dynamic analysis of the joint movement of the mechanisms hoisting for jib and the cargo of the jib crane was carried out. As a result of the analysis, significant dynamic loads were detected in the drive elements and the jib crane structure in the area of joint starting of the drives of the mechanisms hoisting for the jib and the cargo. Significant peak values of the driving force of the jib hoisting drive and the driving moment of the cargo hoisting drive were established. A sharp change in driving forces led to the emergence of short-term high-frequency fluctuations in the kinematic, dynamic and energy characteristics of the crane's structural elements. At the same time, low-frequency oscilations of the cargo on the flexible suspension decay quite slowly and practically do not decay during the movement of the crane. Therefore, such fluctuations of the cargo must be calmed down, which wastes time and reduces the crane's performance.

3. Studies of the dynamic analysis of the joint movement of the mechanisms hoisting for the jib and the cargo revealed that the main reason for the occurrence of oscillating processes in the crane mechanisms is a sudden change in the driving moment and force-driving mechanisms. In order to minimize fluctuating loads in the structural elements of the crane, it is necessary to increase the smoothness of the change of driving forces of mechanisms. It is expedient to eliminate low-frequency oscillations of crane structural elements and cargo on a flexible suspension by applying the boundary conditions of movement after passing the start-up and braking processes.

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**ДИНАМІЧНИЙ АНАЛІЗ СУМІСНОГО ПУСКУ МЕХАНІЗМІВ ПІДЙОМУ СТРІЛИ ТА ВАНТАЖУ СТРІЛОВОГО КРАНА З ГІДРОПРИВОДОМ**

Для підвищення продуктивності стрілових кранів здійснюють суміщення роботи окремих механізмів. Зокрема, в даній роботі розглянута задача спільного пуску механізмів підйому стріли та вантажу При такому русі механізмів зростають динамічні навантаження на елементи конструкції крана, приводи і вантаж на гнучкому підвісі. Підвищені навантаження приводять до зниження надійності роботи крана та зростання енергетичних витрат. При дослідженні динаміки спільного пуску механізмів підйому стріли та вантажу стрілова система представлена механічною системою з чотирма ступенями вільності, де враховано основний рух механізмів та коливальний рух ланок з пружними та дисипативними властивостями, а також вантажу на гнучкому підвісі в площині зміни вильоту. Для такої динамічної моделі стрілової системи крана побудовано математичну модель сумісного руху механізмів підйому стріли та вантажу. Отримана модель являє собою систему нелінійних диференціальних рівнянь другого порядку, розв'язування яких здійснено чисельним методом у вигляді комп'ютерної програми. На основі розробленої програми розраховано та досліджено динаміку сумісного пуску механізмів підйому стріли та вантажу стрілового крана з конкретними числовими параметрами. Проведений розрахунок дозволив провести динамічний аналіз сумісного пуску механізмів підйому стріли та вантажу. В результаті проведеного аналізу виявлені високочастотні коливання ланок з пружними та дисипативними властивостями в приводі механізму підйому вантажу , а також низькочастотні коливання вантажу на гнучкому підвісі. В процесі пуску механізмів високочастотні коливання затухають протягом перехідного процесу, а низькочастотні коливання вантажу є достатньо тривалими і затухають протягом значного проміжку часу.

Для покращення динамічних властивостей механізмів підйому стріли та вантажу під час їхнього спільного руху рекомендовано здійснювати оптимізацію режимів пуску та гальмування.

**Ключові слова:** стріловий кран, приводи, механізми підйому стріли та вантажу, динамічні навантаження, гнучкий підвіс вантажу.

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**DYNAMIC ANALYSIS OF THE SIMULTANEOUS STARTING OF THE BOOM AND LOAD LIFTING MECHANISMS HOISTING FOR THE JIB AND THE CARGO OF THE JIB CRANE WITH A HYDRAULIC DRIVE**

Combining the work of individual mechanisms is carried out to improve the productivity of jib cranes. In particular, the task of joint starting of hoisting mechanisms of jib and cargo is considered in this article. Dynamic loads on crane structural elements, drives, and cargo on a flexible suspension increase with such movement of mechanisms. Increased loads lead to a decrease in the reliability of the crane and an increase in energy costs. When researching the dynamics of the joint starting of hoisting mechanisms of jib and cargo, the jibs system is represented by a mechanical system with four degrees of freedom, which takes into account the main movement of the mechanisms and the oscillatory movement of links with elastic and dissipative properties, as well as the cargo on a flexible suspension in the plane of departure change. A mathematical model of the joint movement of hoisting mechanisms of the jib and cargo was built for such a dynamic model of the boom system of the crane. The obtained model is a system of nonlinear differential equations of the second order, the solution of which was carried out by a numerical method in the form of a computer program. The dynamic of the joint starting of the hoisting mechanisms of the jib and cargo in the jib crane with specific numerical parameters were calculated and studied based on the developed program. The performed calculation made it possible to conduct a dynamic analysis of the joint starting of the hoisting mechanisms. High-frequency oscillations of links with elastic and dissipative properties in the drive of the hoisting mechanism were found as a result of the analysis. Also, low-frequency vibrations of the load on the flexible suspension were found. In the process of starting the mechanisms, high-frequency oscillations are attenuated during the transition process, and low-frequency oscillations of the load are sufficiently long and are attenuated during a significant period.

It is recommended to optimize the starting and braking modes to improve the dynamic properties of the hoisting mechanisms of the jib and cargo during their joint movement.

**Keywords:** jib crane, drives, hoisting mechanisms of the jib and cargo, dynamic loads, flexible load suspension.

УДК 621.87

*Ловейкін В.С., Ромасевич Ю.О., Ловейкін А.В., Ляшко А.П., Почка К.І.* **Динамічний аналіз сумісного пуску механізмів підйому стріли та вантажу стрілового крана з гідроприводом** // Опір матеріалів і теоріяспоруд: наук.-техн. збірник. – Київ: КНУБА, 2024. –Вип. 113. – С. 149-160.

*На основі розробленої програми розраховано та досліджено динаміку сумісного пуску механізмів підйому стріли та вантажу стрілового крана з конкретними числовими параметрами. Проведений розрахунок дозволив провести динамічний аналіз сумісного пуску механізмів підйому стріли та вантажу. В результаті проведеного аналізу виявлені високочастотні коливання ланок з пружними та дисипативними властивостями в приводі механізму підйому вантажу, а також низькочастотні коливання вантажу на гнучкому підвісі.*

Іл. 12. Бібліогр. 18.

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*Loveikin V.S., Romasevych Yu.O., Loveikin A.V., Liashko A.P., Pochka K.I.* **Dynamic analysis of the simultaneous starting of the boom and load lifting mechanisms hoisting for the jib and the cargo of the jib crane with a hydraulic drive** // Strength of Materials and Theory of Structure: Scientific and technical collected articles. – Kyiv: KNUCA, 2024. – Issue 113. – P. 149-160.

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Fig. 12. Ref. 18.

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