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Dynamic analysis of the joint movement of the hoisting and slewing mechanisms of a boom crane

Abstract. To increase the productivity of boom cranes, the operation of individual mechanisms is combined. At the same time, dynamic loads on structural elements, drive mechanisms and loads on a flexible suspension increase, which reduces the reliability of crane operation and increases energy losses. Therefore, the research aims to consider the problem of the dynamics of the joint movement of the mechanisms for load slewing and hoisting of a boom crane. To study the dynamics of the joint movement of the mechanisms, the boom system was represented by a mechanical system with 6DOF, where the basic movement of the mechanisms and the oscillatory movement of the structural links with elastic and dissipative properties, as well as the load on a flexible suspension in the plane of crane slewing and hoisting were considered. For such a mechanical system of a crane, the differential equations of the joint motion of the crane slewing and hoisting mechanisms were developed. The obtained equations are a system of the second order nonlinear differential equations, for solving which a numerical method in the form of a computer program was used. Using the developed program, the dynamics of the joint movement of the mechanisms of a jib crane with specific numerical parameters were calculated. Based on the calculations, a dynamic analysis of the joint movement of the mechanisms for slewing and hoisting the load of a jib crane with a hoisting boom was carried out, which revealed high-frequency vibrations of links with elastic and dissipative properties, as well as low-frequency oscillations of the load on a flexible suspension. The greatest impact of oscillations is observed during the start-up of mechanisms, where high-frequency oscillations dampen during the transient process, and low-frequency oscillations dampen over a fairly significant period. To improve the dynamic properties of the mechanisms for turning and hoisting a load during their joint movement, it is proposed to

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optimise the mode of movement in the areas of transient processes (start-up, braking). The research results can be used in the development and operation of cranes in mechanical engineering, construction, and other industries

Keywords: tower; drive; slewing; load; flexible suspension; oscillation; amplitude

INTRODUCTION

Heavy loads are moved by various types of cranes: overhead, container, rotary and tower cranes. They are used in industry and construction. To increase the productivity of boom cranes, the combined operation of several mechanisms is often used. T. Yang *et al.* (2019) found that, in particular, it is quite common to combine the operation of the load hoisting mechanism and the crane's slewing mechanism. At the same time, with such operation of crane mechanisms, dynamic loads increase in their drive elements and structures. These loads have a significant impact on the reliability of the structure and energy consumption.

M. Ambrosino *et al.* (2020) traced the relationship between performance and dynamic loads in a crane system. O. Podolyak *et al.* (2022) investigated the dependence of the dynamic load on the crane on the swing movement of the load. Mathematical methods were used to build a mathematical model of motion dynamics describing the transient processes of the DEK-251 diesel-electric crawler crane (Soviet Union). The dynamic loads are estimated according to the design schemes. V. Lovejkin *et al.* (2020) focused on the dynamic analysis of flexible crane systems with programmable constraints. It is established that during the start-up process, oscillations occur that can affect the capacity and reliability of the crane. It is proposed to optimise start-up modes to reduce oscillations and improve performance. Q. Wu *et al.* (2020) considered the dynamic analysis of a crane with a double pendulum and a distributed mass beam.

Therefore, the need to study the dynamics of boom cranes when the mechanisms for hoisting and slewing the crane are in operation is determined. It is necessary to consider the low-frequency oscillations of the load on a flexible suspension and the high-frequency vibrations of links with elastic and dissipative elements.

To improve the efficiency of a crane system (speed and positioning accuracy, etc.), the angular oscillations of the load must be kept within small limits, which is achieved by using a controller (Zhang, 2019). Research on the control of crane systems is widespread and long-term. However, most control methods have been proposed mainly for overhead or container cranes (Jaafar *et al.* 2019), and the number of control methods for tower cranes is much smaller than for overhead cranes. The strength and wear of the crane and its parts: body, jib, and boom, can be assessed by numerical analysis (Buczkowski & Żyliński, 2021).

A. Mohammed *et al.* (2019) suggested a method for controlling the input signal alongside a logic algorithm strategy for tuning the acceleration of a trolley. The polynomial function is defined as an input to accelerate the trolley to move the load along a certain trajectory that meets the system constraints while achieving the desired end conditions. X. Wang *et al.* (2019) proposed a unified symplectic pseudospectral method for motion planning and control of 3D-guided overhead cranes. I. Umaru *et al.* (2021) showed a hybrid control system for oscillation damping and precise positioning of the trolley in gantry crane systems. The hybrid control system combines the formation of an input signal with zero oscillations and the zero derivative of oscillations to suppress oscillations. A. Mohammed *et al.* (2023) demonstrated an optimised controller using a multi-objective genetic algorithmic optimisation strategy. The purpose of the proposed method is to minimise the residual oscillations of the crane load that occur when the load is moved from one position to another, starting from non-zero initial conditions.

Thus, the problem of determining the actual dynamic loads that arise during the joint operation of the crane hoisting and slewing mechanisms is quite relevant, as it reflects the real conditions of boom cranes during operation. Therefore, the research aims to determine the dynamic loads in the elements of boom cranes during the joint operation of the mechanisms of the hoisting and crane slewing.

MATERIALS AND METHODS

The boom system of the crane was represented by a mechanical system consisting of absolutely rigid links of the crane slewing and hoisting mechanisms. At the same time, the traction rope of the hoisting mechanism has elastic and dissipative properties, and the flexible load suspension oscillates in the crane outreach and slewing plane and has dissipative properties. In addition, the slewing mechanism drive has elastic and dissipative properties. All the elements of the drive of the hoisting mechanism were connected to the axis of the drive drum, and the drive of the slewing mechanism to the crane's rotation part (column). The hoist rope forms a poly-sling system in which the load is suspended.

Based on the aforementioned, Figure 1 shows a dynamic model of the boom system with the combined movement of the slewing and hoisting mechanisms, which has 6 degrees of freedom. The generalised coordinates of this system are the angular coordinates of the rotation drive α, the drive of the hoisting mechanism β, the crane rotation part φ, the load slewing ψ and the linear coordinates of the load centre of mass when moving in the plane, where coordinate x changes and hoisting in the vertical plane u.

The boom system is subject to the driving forces of the crane's slewing drive mechanisms M_1 and load hoisting M_2, as well as the torque of the resistance forces in the swivel part of the crane M_0 and the mass of the load with the gripper and the bock-and-tackle system. In addition, elastic and dissipative forces act in the elastic elements of the drive mechanisms and the flexible load suspension.

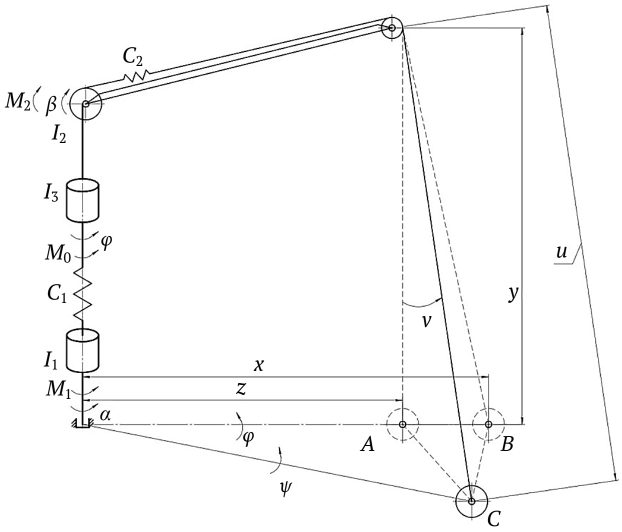

Figure 1. Dynamic model of a boom system with simultaneous movement of the turning and hoisting mechanisms

Note: generalised angular coordinates of the drive of the slewing mechanism α, drive of the load hoisting mechanism β, the rotation part of the crane φ, rotation the load ψ and the linear coordinates of the centre of mass of the load when moving in the plane of starting position change x and hoisting in the vertical plane u; driving forces of the crane slewing drive mechanisms M_1 and load hoisting M_2, moment of resistance forces in the rotation part of the crane M_0; I_1, I_2 – the moments of inertia of the drives of the crane's rotation part and the drive drum of the hoisting mechanism, respectively, are reduced to the axes of the slewing and hoisting mechanisms; I_3 – moment of inertia of the crane slewing part relative to its rotational axis; y – vertical coordinate of the load's centre of mass; z – initial value of the load starting position; v – angular coordinate of deviation from the vertical of the poly-sling system of the load hoisting mechanism; C_1, C_2 – the stiffness coefficients of the drive mechanisms for rotation and hoisting the load are reduced in accordance with the axes of rotation of the crane and the drive drum of the hoisting mechanism; A, B, C - initial, intermediate and final positions of the load during movement

Source: compiled by the authors

To build a mathematical model of the joint movement of the load hoisting and crane slewing mechanisms, represented by the dynamic model in Figure 1, the second order Lagrange equations were applied:

$$\begin{cases} \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{\alpha}} - \frac{\partial T}{\partial \alpha} = M_1 - \frac{\partial P}{\partial \alpha} - \frac{\partial R}{\partial \dot{\alpha}}; \\ \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{\beta}} - \frac{\partial T}{\partial \beta} = M_2 - \frac{\partial P}{\partial \beta} - \frac{\partial R}{\partial \dot{\beta}}; \\ \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{\phi}} - \frac{\partial T}{\partial \phi} = -M_0 - \frac{\partial P}{\partial \phi} - \frac{\partial R}{\partial \phi} \\ \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{\psi}} - \frac{\partial T}{\partial \psi} = -\frac{\partial P}{\partial \psi} - \frac{\partial R}{\partial \dot{\psi}}; \\ \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{x}} - \frac{\partial T}{\partial x} = -\frac{\partial P}{\partial x} - \frac{\partial R}{\partial \dot{x}}; \\ \frac{d}{\mathrm{dt}}\frac{\partial T}{\partial \dot{u}} - \frac{\partial T}{\partial u} = -\frac{\partial P}{\partial u} - \frac{\partial R}{\partial \dot{u}}, \end{cases} \quad (1)$$

where T, P, R – are the kinetic and potential energy of the system, respectively, and the Rayleigh dissipative function; M_1, M_2 – driving torques of the drive mechanisms for crane slewing and hoisting, summed up according to the rotation part of the crane and the drive drum of the hoisting mechanism (torques M_1, M_2 are determined from the mechanical characteristics of electric motors according to the Kloss formula (Lovejkin *et al.*, 2020)); M_0 – the moment of resistance forces in the rotary part of the crane, reduced to the rotation axis.

The kinetic energy of the boom system during the joint movement of the mechanisms for rotation and hoisting the load was expressed through generalised coordinates and is represented by the following dependence:

$$T = \tfrac{1}{2}I_1\dot{\alpha}^2 + \tfrac{1}{2}I_2\dot{\beta}^2 + \tfrac{1}{2}I_3\dot{\phi}^2 + \tfrac{1}{2}m\left(\dot{x}^2 + \dot{y}^2 + \dot{\psi}^2x^2\right), \quad (2)$$

where I_1, I_2 – the moments of inertia of the drives of the crane's rotation part and the drive drum of the hoisting mechanism, respectively, are reduced to the axes of the slewing and hoisting mechanisms; m – load mass; I_3 – moment of inertia of the crane slewing part relative to its rotation axis; y – vertical coordinate of the centre of load mass.

The potential energy function of the joint movement system of the slewing and hoisting mechanisms:

$$P = \tfrac{1}{2}C_1(\phi - \alpha)^2 + \tfrac{1}{2}C_2(\mathrm{nu}\text{-}\beta r)^2 - mgu\cos v, \quad (3)$$

where C_1, C_2 – the stiffness coefficients of the drive mechanisms for rotation and hoisting the load are reduced following the axes of rotation of the crane and the drive drum of the hoisting mechanism; r – radius of the drive drum of the load hoisting mechanism; n – multiplicity of the load hoisting mechanism; g – free fall acceleration; v – angular coordinate of deviation from the vertical of the block-and-tackle system of the load hoisting mechanism.

The Rayleigh dissipative function is expressed by the following relationship:

$$R = \frac{1}{2}b_1(\dot{\phi} - \dot{\alpha})^2 + \frac{1}{2}b_2(\mathrm{n}\dot{\mathrm{u}} - \dot{\beta}r)^2 + \frac{1}{2}b\dot{v}^2, \qquad (4)$$

where b_1, b_2, b – Dissipation coefficients of elastic elements, respectively, of the drives of the mechanisms for slewing and hoisting the load, as well as the flexible load suspension.

To find the coordinate of the deviation from the vertical of the load rope (block-and-tackle system) (Fig. 1), the following geometric relationships are used:

$$\mathrm{AB} = \text{x-z}; \ \mathrm{BC} = x(\phi - \psi); \angle\mathrm{ABC} = \frac{\pi}{2} - \frac{\phi-\psi}{2}; \qquad (5)$$

$$AC = \sqrt{AB^2 + BC^2 - 2AB \cdot BC \cdot cos\angle ABC} =$$
$$= \sqrt{(x-z)^2 + x^2(\phi-\psi)^2 - 2(x-z)x(\phi-\psi) \cdot sin\frac{\phi-\psi}{2}}. \qquad (6)$$

Since during the operation of cranes, the deviation of the load rope from the vertical does not exceed 12°, then it is possible to assume $\sin\frac{\varphi-\psi}{2} = \frac{\varphi-\psi}{2}$ which will result in:

$$AC = \sqrt{(x-z)^2 + x^2(\phi-\psi)^2 - x^2(\phi-\psi)^2 + zx(\phi-\psi)^2} =$$
$$= \sqrt{(x-z)^2 + zx(\phi-\psi)^2}. \qquad (7)$$

Given the latter expression, the angular coordinate of the deviation of the block-and-tackle system from the vertical was determined by the dependence:

$$v = \frac{\mathrm{AC}}{u} = \frac{1}{u}\sqrt{(x-z)^2 + zx(\phi-\psi)^2}, \qquad (8)$$

where z – initial load starting position.

Dependencies were used to determine the vertical coordinate and velocity of the centre of mass of the load during the joint operation of the mechanisms for hoisting the load and the crane slewing:

$$y = u\cos v; \ \dot{y} = \dot{u}\cos v - \dot{v}u\sin v. \qquad (9)$$

The derivatives of the kinetic energy required for the system of equations (1) are found:

$$\frac{\partial T}{\partial \alpha} = \frac{\partial T}{\partial \beta} = \frac{\partial T}{\partial \phi} = \frac{\partial T}{\partial \psi} = 0; \frac{\partial T}{\partial x} = \mathrm{m}x\dot{\psi}^2; \frac{\partial T}{\partial u} = m\dot{y}\frac{\partial \dot{y}}{\partial u}; \qquad (10)$$

$$\frac{\partial T}{\partial \dot{\alpha}} = I_1\dot{\alpha}; \frac{\partial T}{\partial \dot{\beta}} = I_2\dot{\beta}; \frac{\partial T}{\partial \dot{\phi}} = I_3\dot{\phi}; \frac{\partial T}{\partial \dot{\psi}} = \mathrm{m}x^2\dot{\psi}; \qquad (11)$$

$$\frac{\partial T}{\partial \dot{x}} = \mathrm{m}\dot{x}; \ \frac{\partial T}{\partial \dot{u}} = m\dot{y}\frac{\partial \dot{y}}{\partial \dot{u}} = m\dot{y}\frac{\partial y}{\partial u}; \qquad (12)$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\alpha}} = I_1\ddot{\alpha}; \frac{d}{dt}\frac{\partial T}{\partial \dot{\beta}} = I_2\ddot{\beta}; \frac{d}{dt}\frac{\partial T}{\partial \dot{\phi}} = I_3\ddot{\phi}; \qquad (13)$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\psi}} = \mathrm{m}x^2\ddot{\psi} + 2mx\dot{x}\dot{\psi}; \frac{d}{dt}\frac{\partial T}{\partial \dot{x}} = \mathrm{m}\ddot{x}; \qquad (14)$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{u}} = m\ddot{y}\frac{\partial y}{\partial u} + m\dot{y}\frac{\partial \dot{y}}{\partial u}. \qquad (15)$$

Several derivatives included in expressions (9)...(15) were found:

$$\frac{\partial \dot{y}}{\partial u} = -\dot{u}\frac{\partial v}{\partial u}sin\,v - (\dot{v} + u\frac{\partial \dot{v}}{\partial u})\,sin\,v - u\dot{v}\frac{\partial v}{\partial u}cos\,v;$$
$$\frac{\partial y}{\partial u} = \cos v - u\frac{\partial v}{\partial u}sin\,v. \qquad (16)$$

$$\ddot{y} = \ddot{u}\cos v - \dot{u}\dot{v}\sin v - \ddot{v}u\sin v - \dot{u}\dot{v}\sin v - \dot{v}^2u\cos v = \ddot{u}\cos v -$$
$$- 2\dot{u}\dot{v}\sin v - \dot{v}^2u\cos v - \ddot{v}u\sin v. \qquad (17)$$

Partial derivatives of the potential energy of the system were taken, which was expressed by dependence (3):

$$\frac{\partial P}{\partial \alpha} = -C_1(\phi - \alpha); \frac{\partial P}{\partial \beta} = -C_2r(nu - \beta r); \qquad (18)$$

$$\frac{\partial P}{\partial u} = C_2n(nu - \beta r) - mg(cos\,v - u\frac{\partial v}{\partial u}sin\,v); \qquad (19)$$

$$\frac{\partial P}{\partial \phi} = C_1(\phi - \alpha) + mgu\,sin\,v\frac{\partial v}{\partial \phi}; \qquad (20)$$

$$\frac{\partial P}{\partial \psi} = mgu\frac{\partial v}{\partial \psi}sin\,v; \frac{\partial P}{\partial x} = mgu\frac{\partial v}{\partial x}sin\,v. \qquad (21)$$

Expressions (18), and (21) contain partial derivatives of the function of the load rope deviation from the vertical by generalised coordinates. These derivatives were determined:

$$\frac{\partial v}{\partial u} = -\frac{1}{u^2}\sqrt{(x-z)^2 + zx(\phi-\psi)^2}; \qquad (22)$$

$$\frac{\partial v}{\partial \phi} = \frac{1}{u}\frac{zx(\phi-\psi)}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}; \qquad (23)$$

$$\frac{\partial v}{\partial \psi} = -\frac{1}{u}\frac{zx(\phi-\psi)}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}; \qquad (24)$$

$$\frac{\partial v}{\partial x} = \frac{1}{u}\frac{(x-z)+z(\phi-\psi)^2/2}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}. \qquad (25)$$

Partial derivatives of the Rayleigh dissipative function:

$$\frac{\partial R}{\partial \dot{\alpha}} = -b_1(\dot{\phi} - \dot{\alpha}); \frac{\partial R}{\partial \dot{\beta}} = -b_2r(n\dot{u} - \dot{\beta}r); \qquad (26)$$

$$\frac{\partial R}{\partial \dot{u}} = b_2n(n\dot{u} - \dot{\beta}r) + b\dot{v}\frac{\partial \dot{v}}{\partial \dot{u}}; \qquad (27)$$

$$\frac{\partial R}{\partial \dot{\phi}} = b_1(\dot{\phi} - \dot{\alpha}) + b\dot{v}\frac{\partial \dot{v}}{\partial \dot{\phi}}; \qquad (28)$$

$$\frac{\partial R}{\partial \dot{\psi}} = b\dot{v}\frac{\partial \dot{v}}{\partial \dot{\psi}}; \frac{\partial R}{\partial \dot{x}} = b\dot{v}\frac{\partial \dot{v}}{\partial \dot{x}}. \qquad (29)$$

Angular rate of deviation of the load rope from the vertical:

$$\dot{v} = \frac{1}{u^2}\left\{\frac{(x-z)\cdot\dot{x} + z[\dot{x}(\phi-\psi)^2/2 + x(\phi-\psi)(\dot{\phi}-\dot{\psi})]}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}} \cdot \right.$$
$$\left. \cdot\, u - \dot{u}\sqrt{(x-z)^2 + zx(\phi-\psi)^2}\right\}. \qquad (30)$$

The partial derivatives of function (30) are taken, and the result is:

$$\frac{\partial \dot{v}}{\partial \dot{u}} = \frac{\partial v}{\partial u} = -\frac{1}{u^2}\sqrt{(x-z)^2 + zx(\phi-\psi)^2}; \qquad (31)$$

$$\frac{\partial \dot{v}}{\partial \dot{\phi}} = \frac{\partial v}{\partial \phi} = \frac{1}{u}\frac{zx(\phi-\psi)}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}; \qquad (32)$$

$$\frac{\partial \dot{v}}{\partial \dot{\psi}} = \frac{\partial v}{\partial \psi} = -\frac{1}{u}\frac{zx(\phi-\psi)}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}; \qquad (33)$$

$$\frac{\partial \dot{v}}{\partial \dot{x}} = \frac{\partial v}{\partial x} = \frac{1}{u}\frac{x+z[(\phi-\psi)^2/2-1]}{\sqrt{(x-z)^2+zx(\phi-\psi)^2}}. \qquad (34)$$

After substituting expressions (9), (10), (14), (15), (18)..., (21), (26), ... (29) into system (1), a system of differential equations of motion of the joint movement of the mechanisms for slewing and hoisting the load of a boom crane is obtained:

$$\begin{cases} I_1\ddot{\alpha} = M_1 + C_1(\phi - \alpha) + b_1(\dot{\phi} - \dot{\alpha}); \\ I_2\ddot{\beta} = M_2 + C_2 r(nu - \beta r) + b_2(n\dot{u} - \dot{\beta} r); \\ I_3\ddot{\phi} = -M_0 - C_1(\phi - \alpha) - mgu\frac{\partial v}{\partial \phi}\sin v - \\ \qquad -b_1(\dot{\phi} - \dot{\alpha}) - b\dot{v}\frac{\partial v}{\partial \phi}; \\ mx^2\ddot{\psi} + 2mx\dot{x}\dot{\psi} = -mgu\frac{\partial v}{\partial \psi}\sin v - b\dot{v}\frac{\partial v}{\partial \psi}; \\ m\ddot{x} - mx\dot{\psi}^2 = -mgu\frac{\partial v}{\partial x}\sin v - b\dot{v}\frac{\partial v}{\partial x}; \\ m(\ddot{u}\cos v - 2\dot{u}\dot{v}\sin v - \dot{v}^2 u\cos v - \ddot{v}u\sin v)\cos v = \\ \qquad = -C_2 n(nu - \beta r) + mg(\cos v - u\frac{\partial v}{\partial u}\sin v) - \\ -b_2 n(n\dot{u} - \dot{\beta} r) - b\dot{v}\frac{\partial v}{\partial u}. \end{cases} \quad (35)$$

The resulting system of second-order differential equations is nonlinear and cannot be solved by analytical methods, so a numerical method is used to solve it, which is presented in the form of a developed computer program in the Wolfram Mathematica software. The following initial conditions for the joint movement of the mechanisms for slewing and hoisting the load are used:

$$t = 0,\ \alpha = \frac{M_0}{C_1}, \dot{\alpha} = 0, \beta = \frac{u_0 \cdot 4}{r}, \dot{\beta} = 0, \beta = \frac{u_0 \cdot 4}{r}, \\ \dot{u} = 0, x = z = 40m, \dot{x} = 0, \phi = 0, \dot{\phi} = 0, \psi = 0, \dot{\psi} = 0. \quad (36)$$

The calculations of the dynamics of the joint movement of the mechanisms for slewing and hoisting the load of a boom crane were performed for the following initial data: I_1 = 71626.115 kg·m²; I_2 = 1249.93 kg·m²; I_3 = 4920738.85 kg·m²; C_1 = 6626669.045 (N·m)/rad; C_2 = 150796 (N·m)/rad; m = 2000 kg; z = 40 m; u = 20 m; r_b=0.25 m; n=4; M_{cr1}=85N·m; M_{cr2}=487.5N·m; M_{n1}=36.8 N·m; M_{n2} = 195 N·m; u_1 = 1355.2; u_2 = 116.56; η_1 = 0.86; η_2 = 0.85; ω_{o1} = 104.67 rad/c; ω_{o2} = 157 rad/c; ω_{n1} = 95.04 rad/c; ω_{n2} = 153.86 rad/c; λ_1 = 2.8; λ_2 = 2.5; b = 2.1 · 10⁵ $\frac{Nm}{m/s}$; b_1 = 1 · 10⁵ $\frac{Nm}{m/s}$; b_2 = 1.5 · 10⁵ $\frac{Nm}{m/s}$; g = 9.81 m/c².

RESULTS AND DISCUSSION

As a result of numerically solving the system of differential equations (35) under the initial conditions of movement of the mechanisms (36) using the developed computer program for a specific hoisting boom crane, graphical dependences of kinematic, dynamic, and energy characteristics were built. Figure 2 shows the graphical dependences of the angular velocities of the drive mechanisms for crane slewing and hoisting, plotted following the axes of rotation of the crane and the drive drum of the hoisting mechanism. The angular velocity graph of the slewing mechanism drive (Fig. 2a) shows that during the 5-second start-up period, there are alternating speed oscillations with an amplitude of -0.09 to +0.07 rad/s and a frequency of 1.4 s^{-1}, followed by reaching a steady-state value within 15 s. This mode of starting the slewing mechanism leads to significant dynamic loads in the drive elements and the rotary column. Since this study considers lowering the load, the drive mechanism for hoisting the load almost instantly (within 0.1 s) acquires the steady-state angular velocity of the drive drum (Fig. 2b). In this case, significant dynamic loads also act, but they take on an instantaneous value and in the steady-state movement section, the angular velocity of the drive drum takes on a constant value.

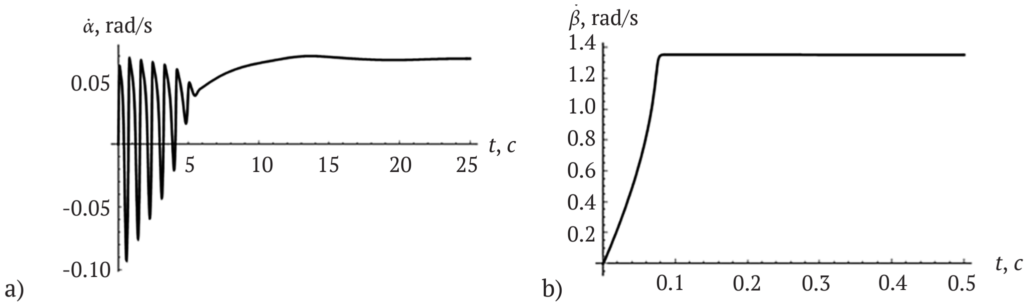

Figure 2. Angular speeds of drive mechanisms

Note: a) slewing; b) hoisting
Source: compiled by the authors

The graphical dependences of the angular accelerations of the drive mechanisms for crane slewing and hoisting (Fig. 3) confirm the results of the analysis of the angular velocities of these mechanisms. From Figure 3a, it can be seen that at the start-up section of the slewing mechanism, the drive elements exhibit alternating oscillations of angular accelerations with a maximum oscillation amplitude of 1.5 rad/s². After the start-up process, the acceleration oscillations of the slewing mechanism drive dampen and are practically zero in the steady-state motion area. When the load hoisting mechanism is started, the angular acceleration of the drive almost instantly increases to a maximum value of 40 rad/s² and then instantly decreases to zero. The instantaneous maximum acceleration of the hoisting actuator is an order of magnitude greater than the instantaneous maximum acceleration of the slewing actuator, but this action is instantaneous without further oscillations, which is not the case with the slewing actuator, which has damped oscillations.

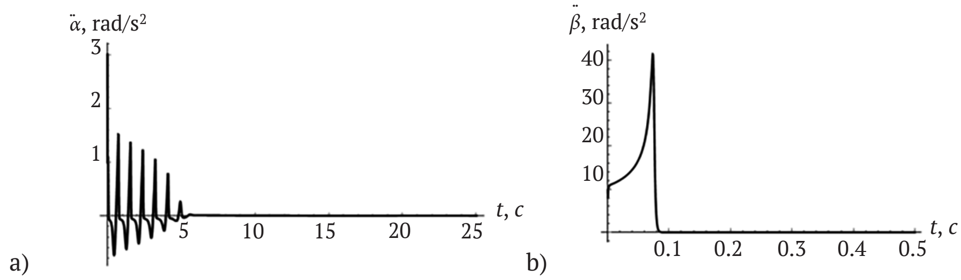

Figure 3. Angular accelerations of drive mechanisms
Note: a) crane slewing; b) load hoisting
Source: compiled by the authors

Figure 4 shows the angular velocity and acceleration of a crane slewing column. Figure 4a shows that the steady-state value of the column's angular velocity is reached after approximately 13 seconds of movement. At the same time, at the beginning of the start-up, within 5 seconds of movement, high-frequency speed oscillations with a small amplitude are observed. Subsequently, during the steady-state movement, there are slight low-frequency fluctuations in the column speed caused by the oscillations of the load on the flexible suspension. At the start-up section, high-frequency alternating oscillations of the column's angular acceleration are observed for 5 seconds of motion (Fig. 4b). Such a change in acceleration leads to the appearance of alternating dynamic loads in the column during the first seconds of start-up. Subsequently, the acceleration of the column decreases with minor low-frequency oscillations and tends to zero by the 25th second of movement.

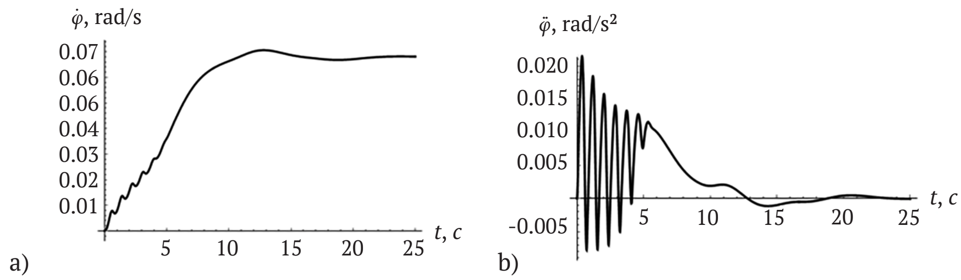

Figure 4. Angular velocity (a) and acceleration (b) of the crane column
Source: compiled by the authors

The linear velocity and acceleration of the axial movement of the load (Fig. 5) have a familiar oscillatory character caused by the centrifugal force when the crane is turning and the action of the horizontal component of the load's gravity when the load rope deviates from the vertical. As can be seen from Figures 5a and 5b, these oscillations require a significant duration of movement to dampen, which is not often enough for cranes to operate. Therefore, at the end of the crane's movement cycle, it is necessary to cease the load oscillations, which takes time and leads to a decrease in the crane's productivity.

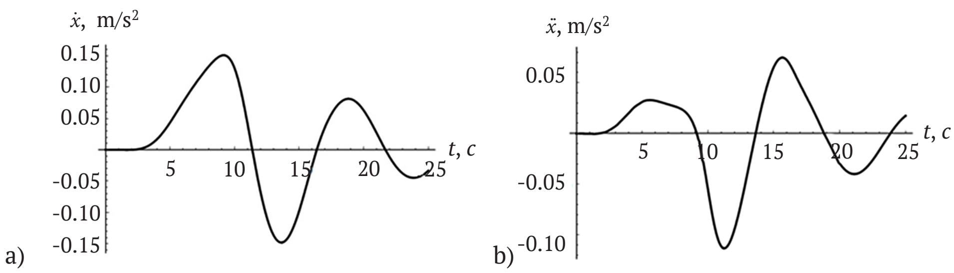

Figure 5. Linear velocity (a) and acceleration (b) of the load in the plane of starting position change
Source: compiled by the authors

Figure 6 shows the graphs of angular displacement, speed, and acceleration of the deviation from the vertical of the block-and-tackle system of the load hoisting mechanism. From Figure 6a, it can be seen that the deviation of the load chain from the vertical initially (during the first five seconds of movement) increases, and then low-frequency oscillations are observed with a decrease in the value of the deviation. At the same time, the deviation of the load hoist from the vertical does not change its sign, which indicates the dominant role of the centrifugal force of the load when the crane is turning. The rate of change of the deviation of the hoist from the vertical (Fig. 6b) consists of low-frequency and high-frequency components of oscillations in a section lasting up to 5 seconds, and then only the low-frequency component of oscillations remains, which fading over time. The angular acceleration of the deviation from the vertical of the load sling (Fig. 6c) also has a familiar high-frequency component of oscillations up to 5 seconds of movement, and after that, insignificant low-frequency oscillations of acceleration are observed, which practically dampen by 25 seconds of movement. Moreover, the amplitude of high-frequency oscillations significantly exceeds the amplitude of low-frequency oscillations of the angular acceleration of the load sling deviation from the vertical.

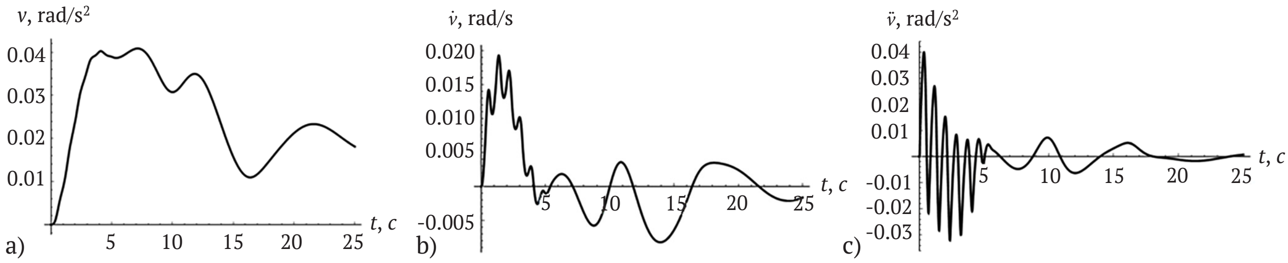

Figure 6. Poly-harness system deviation from the vertical

Note: a) angular changes, b) speed, c) velocity

Source: compiled by the authors

The driving torque of the electric motor (Fig. 7a) and the elastic torque in the transmission mechanism (Fig. 7b) of the rotation drive during the start-up phase change in an oscillatory mode with a high oscillation frequency for up to 6 seconds. Then, these moments decrease and take on steady-state values by the 25th second of movement. The maximum value of the driving torque is 100 kNm, and the elastic torque in the transmission mechanism is 120 kNm. Such a change in the driving torque leads to a significant increase in dynamic loads of an oscillatory nature in the structural elements and the drive, as evidenced by the graph of the elastic moment in the transmission mechanism, where the maximum value of the elastic moment is 20% higher than the maximum value of the driving torque. To reduce the dynamic loads in the elements of the boom system during the joint movement of the mechanisms for turning and hoisting the load, there is a need to eliminate oscillations and increase the smoothness of the change in the driving torques of the drive mechanisms.

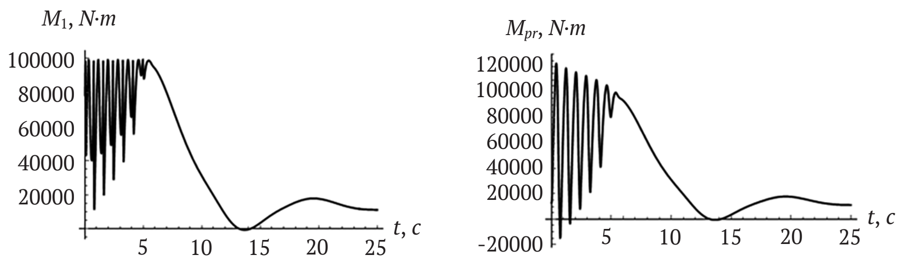

Figure 7. Crane rotational torques

Note: a) driving, b) elastic

Source: compiled by the authors

Figure 8 shows a graph of the power change of the slewing mechanism drive, which shows that at the beginning of the start-up, up to 5 seconds of movement, there is a familiar oscillating character of power change with its maximum value of 4.0 kW. Subsequently, the power changes smoothly and by the 25th second of movement, its steady-state value is established. The reason for the oscillations in the power change of the slewing drive mechanism is the oscillatory nature of the drive torque change, which depends on electromagnetic transients in the electric motor. Therefore, to eliminate oscillatory processes in the boom crane mechanisms, it is necessary to optimally control electromagnetic transients in the drive motors.

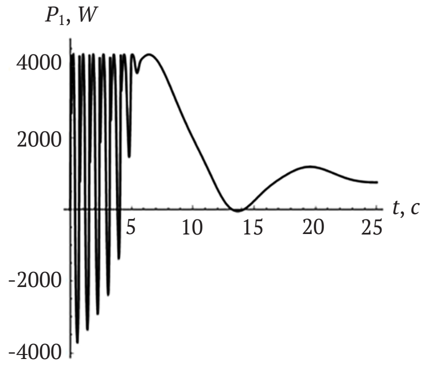

Figure 8. Diagram of power variation of the drive of the turning mechanism
Source: compiled by the authors

The research results obtained were compared with those of other authors. O. Podolyak *et al.* (2022) presented the construction of a mathematical model of the dynamics of the boom crane movement when three mechanisms work jointly: trolley movement, slewing, and load hoisting. Differential equations of motion are presented that describe the joint movement of the mechanisms without considering the elasticity of the links, but no numerical calculations are used in the constructed model. The study considers the elastic-dissipative properties of the drive mechanisms for crane hoisting and turning with the calculation of kinematic, dynamic, and energy numerical characteristics. At the same time, a significant influence of oscillatory processes on the dynamics of movement of the mechanisms for slewing and hoisting loads is shown. V. Lovejkin *et al.* (2020) considered a dynamic model of the motion of two mechanisms for the trolley movement and crane slewing, which considers the oscillations of the load on a flexible suspension, but all other links of the mechanisms are assumed to be rigid bodies. As a result of the calculation, the maximum rotation and acceleration speeds were obtained, which are 0.1 rad/s and 0.03 rad/s^2, respectively, while in the present article, the maximum speed in the slewing mechanism is 0.07 rad/s and the acceleration is 0.022 rad/s^2. N. Sun *et al.* (2019) built a mathematical model of the dynamics of load movement on a flexible suspension. They developed the differential equations of motion and conducted experimental studies, which showed that the maximum deviation of the load from the vertical is 0.11 rad. In the presented study, a similar result is 0.041 rad, which is almost 2.5 times less compared to the result obtained by N. Sun *et al.* (2019). M. Ambrosino *et al.* (2020) developed a mathematical model of the dynamics of movement of crane slewing and boom hoisting mechanisms with load oscillations on a flexible suspension in the plane of the trolley movement and crane slewing. As a result of the calculations, the maximum values of the load rope deviations from the vertical were obtained, which are 0.047 rad, In the proposed article, a similar result is 0.041 rad, which is almost the same as the result obtained by M. Ambrosino *et al.* (2020). R. Miranda-Colorado (2021) presents a new robust controller to dampen the oscillation of the deflection angle of a crane load suspension with a variable cable length. E. Jarzębowska *et al.* (2020) investigated the vibrations of flexible systems with programmable constraints. The automated procedure allows the analysis of both rigid and flexible systems. An example of a crane with a flexible boom and programmable constraints is demonstrated. S. Wang *et al.* (2019) studied an anti-error device with a triple cable mechanism for cranes that eliminates the disadvantages of existing methods of reducing oscillations. J. Wang *et al.* (2022) analysed a three-degree-of-freedom anti-error method for damping crane load oscillations on a ship. I.A. Martin *et al.* (2021) developed an anti-error control system for a crane on a ship with multiple degrees of freedom.

M. Zhang *et al.* (2019) developed a dynamic model of the movement of an overhead crane and simulated the dynamics of the movement of a load on a flexible suspension. The maximum deviation of the load from the vertical was determined to be 0.094 rad, while in the proposed study this value reaches 0.041 rad, which is 2.29 times higher than the result obtained. H. Ouyang *et al.* (2019) presented a dynamic model of an overhead crane, however, which is presented in the form of a double pendulum, and equations of motion describing transient processes (start-up, steady-state motion, and braking) were developed. The motion analysis was performed, and the method proposed by H. Ouyang *et al.* (2019) and the method of continuous sliding mode control (CSMC). The method proposed by H. Ouyang *et al.* (2019) proved to be better than the CSMC method, reducing the maximum values of load oscillations by almost 3.5 times. Comparing the method proposed by H. Ouyang *et al.* (2019) with the results obtained in the proposed study, which are 0.024 rad and 0.041 rad, respectively, it has a 1.7 times lower maximum amplitude of load oscillations, in contrast to the results obtained in the proposed article. However, it should be noted that the method proposed by H. Ouyang *et al.* (2019) is an optimization, while the proposed article was based on the mechanical characteristics of the engine (without optimization).

Considering the results obtained in the research and studies of other authors, it can be stated that the study of the dynamics of movement of crane mechanisms is relevant, and the results obtained in this article are close to those obtained by other authors.

CONCLUSIONS

Based on the results of the study of the dynamics of the joint movement of the mechanisms for boom crane slewing and hoisting, a mathematical model of the dynamics of the joint movement of the mechanisms for slewing and hoisting of the load of a boom crane with a hoisting boom was built, which is a system of six nonlinear second order differential equations. The developed model considers the main motion of the drive mechanisms for the crane slewing and hoisting the load, as well as high-frequency vibrations of the links of these mechanisms with elastic and dissipative properties, as well as low-frequency oscillations of the load on a flexible suspension. The solution of the system of nonlinear differential equations for the joint motion of the crane slewing and hoisting mechanisms was found by the numerical method using a specially developed computer program.

Based on the results of the calculations, a dynamic analysis of the joint movement of the mechanisms for turning and hoisting the load of a boom crane was carried out. The dynamic analysis of the joint movement of the hoisting and slewing mechanisms revealed significant dynamic loads in the elements of the drives and the crane structure during the process of starting the drive mechanisms. In the slewing mechanism elements, which have elastic and dissipative properties, the authors found familiar high-frequency oscillatory processes of kinematic, power, and energy characteristics that dampen rather quickly. At the same time, the low-frequency oscillations of the mechanism links and the load on a flexible suspension decay rather slowly or practically do not decay during the movement cycle. This primarily applies to the load on the flexible suspension, whose oscillations need to be dampened, which takes time and reduces crane productivity. When lowering the load during the start-up process, the hoisting mechanism is subject to instantaneous significant overloads that do not cause oscillations of the mechanism links.

It has been established that one of the reasons for the oscillatory processes in the crane slewing mechanism is the abrupt nature of the change in the drive torque. To reduce oscillatory processes in the crane elements, it is necessary to increase the smoothness of the change in the driving torque of the drive mechanisms. Low-frequency oscillations of crane structural elements, drive mechanisms, and load on a flexible suspension are recommended to be calmed by introducing boundary conditions of movement after the passage of transient processes (start-up, braking). The prospect for further research is to optimise the modes of movement of drive mechanisms to minimise dynamic loads on structural elements and eliminate oscillations of the load on a flexible suspension.

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CONFLICT OF INTEREST

None.

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Динамічний аналіз сумісного руху механізмів підйому та повороту стрілового крана

Анотація. Задля підвищення продуктивності стрілових кранів здійснюють суміщення роботи окремих механізмів. При цьому зростають динамічні навантаження на елементи конструкції, приводних механізмів і вантажу на гнучкому підвісі, які знижують надійність роботи кранів і підвищують енергетичні витрати. Тому метою статті було розглянути задачу динаміки спільного руху механізмів повороту та підйому вантажу стрілового крану з підйомною стрілою. Для дослідження динаміки спільного руху механізмів стрілова система була представлена механічною системою з шістьома ступенями вільності, де враховано основний рух механізмів та коливальний рух ланок конструкції з пружними та дисипативними властивостями, а також вантажу на гнучкому підвісі в площині повороту крана та підйому вантажу. Для такої механічної системи вантажопідйомного крана складено диференціальні рівняння сумісного руху механізмів повороту та підйому вантажу. Отримані рівняння являють собою систему нелінійних диференціальних рівнянь другого порядку, для розв'язування яких використано чисельний метод у вигляді комп'ютерної програми. За допомогою розробленої програми проведені розрахунки динаміки спільного руху механізмів стрілового крана з конкретними числовими параметрами. На основі проведеного розрахунку здійснено динамічний аналіз спільного руху механізмів повороту та підйому вантажу стрілового крана з підйомною стрілою, в результаті якого встановлені високочастотні коливання ланок з пружними та дисипативними властивостями, а також низькочастотні коливання вантажу на гнучкому підвісі. Найбільший вплив коливань спостерігається в процесі пуску механізмів, де високочастотні коливання затухають протягом перехідного процесу, а низькочастотні коливання затухають протягом достатньо значного проміжку часу. Для покращення динамічних властивостей механізмів повороту та підйому вантажу під час їх спільного руху запропоновано здійснювати оптимізацію режиму руху на ділянках перехідних процесів (пуск, гальмування). Результати досліджень можуть бути використані при розробленні та експлуатації вантажопідйомних кранів в машинобудуванні, будівництві та інших галузях виробництва

Ключові слова: башта; привод; обертання; навантаження; гнучкий підвіс; коливання; амплітуда