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DYNAMIC ANALYSIS OF THE SIMULTANEOUS MOVEMENT OF THE JIB LIFTING AND CRANE TURNING MECHANISMS

V.S. Loveikin[1],
Doctor of Science (Engineering), Professor

Yu.O. Romasevych[1],
Doctor of Science (Engineering), Professor

Yu.V. Loveikin[2],
Candidate of Science (Physics and Mathematics), Associate Professor

A.P. Liashko[1],
Candidate of Science (Engineering), Associate Professor

K.I. Pochka[3],
Doctor of Science (Engineering), Professor

[1]*National University of Life and Environmental Sciences of Ukraine*

[2]*Taras Shevchenko National University of Kyiv*

[3]*Kyiv National University of Construction and Architecture*

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The efficiency of jib cranes largely depends on increasing their productivity when performing loading, unloading, and installation operations. The simultaneous operation of individual mechanisms increases the productivity of jib cranes. The purpose of this study is to construct a mathematical model and perform a dynamic analysis of the crane jib system during the simultaneous operation of the jib lifting and crane turning mechanisms. The presented research is based on methods for constructing discrete dynamic models of a jib crane using second-order Lagrange equations, numerical methods for solving nonlinear ordinary differential equations, which are presented in the form of a computer program, and methods for dynamic analysis of the simultaneous motion of crane mechanisms. The problem of the dynamics of simultaneous motion of the jib lifting and crane turning mechanisms is solved in this article. The crane jib system is represented by a dynamic model with four degrees of freedom, which considers the main motion of the jib lifting and crane turning mechanisms and the oscillation of the cargo on a flexible suspension. As a result of the dynamic analysis, the kinematic, dynamic, and energy characteristics of individual links of the crane jib system are determined during the simultaneous operation of the jib lifting and crane turning mechanisms. The main movement of the drive mechanisms for lifting the jib and crane turning, as well as the low-frequency spatial oscillations of the cargo on a flexible suspension, were investigated. It has been established that the dynamic motion of the mechanisms depends on the character of the change in the driving forces of the drives. Low-frequency oscillations of the cargo on a flexible suspension practically do not dampen and continue throughout the entire movement cycle.

To improve the dynamics of simultaneous motion of mechanisms and minimise oscillatory processes of the jib system links, it is recommended to select modes of smooth change of driving forces of drives in transition processes (starting, braking), which ensure the desired movement of executive mechanisms and lead to a reduction in loads.

Keywords: jib lifting mechanism, crane turning mechanism, oscillatory processes, driving forces, dynamic loads.

Introduction. The efficiency of jib cranes largely depends on increasing their productivity when performing loading, unloading, transport, and installation operations. One way to increase the productivity of jib cranes is to operate several mechanisms simultaneously in pairs, particularly the simultaneous operation of the jib lifting and crane turning mechanisms. When the mechanisms of jib cranes operate simultaneously, the dynamic loads on the structural elements and drives that have the most significant impact during transitional processes (starting, braking, changing speed) increase.

The increase in dynamic loads significantly impacts the reliability of operation and energy consumption of jib cranes. Therefore, there is a need to determine the actual dynamic loads and study the oscillatory processes in the elements of the jib system during the simultaneous operation of several mechanisms. When performing loading and unloading operations with jib cranes, the jib lifting and crane turning mechanisms are often combined. Therefore, studying the dynamic processes during the simultaneous operation of the jib lifting and crane turning mechanisms is necessary. The research of dynamic processes during the simultaneous operation of the jib lifting and crane turning mechanisms is relevant because it allows the real loads during the operation of jib cranes to be identified and, in the future, decisions to be made regarding the possibility of combining the operation of certain jib crane mechanisms.

Analysis of publications. To increase the productivity of jib and tower cranes, several mechanisms are combined to work in pairs simultaneously, but this increases the dynamic loads on structural elements and drives. At the same time, dynamic loads significantly impact vital characteristics of crane operation, such as productivity, reliability, energy consumption, and safe operation [1]. Therefore, the research on the dynamics of crane mechanisms during their simultaneous operation remains relevant. Considerable attention is paid to the research of the dynamics of the simultaneous motion of the mechanisms for changing the reach and rotation of lifting cranes [2,3]. These papers define the mechanisms' kinematic, dynamic, and energy characteristics under consideration and establish directions for their improvement. In addition, paper [3] describes the features of controlling the electric drive of the jib adjustment mechanism during simultaneous crane turning with a suspended cargo. Research on dynamic loads during the simultaneous motion of these mechanisms was also carried out. In [4], the dynamics of the simultaneous motion of the boom extension and rotation mechanisms of a tower crane were modelled, and the start-up mode of these mechanisms was optimised according to an integral dynamic criterion. The optimisation made it possible to significantly reduce the dynamic overload during the simultaneous motion of the outreach and rotation mechanisms [4]. The authors of [5] researched the dynamics of the jib lifting mechanisms' simultaneous motion and the jib crane's cargo. Here, the mechanisms' simultaneous motion dynamics were modelled, and a dynamic analysis was performed. It was established that the jib crane operates with overloads when these mechanisms operate simultaneously. In article [6], the dynamics of the simultaneous motion of the jib lifting mechanisms and the extension of its section during the steady crane turning were modelled, based on which a dynamic analysis of the movement of these mechanisms was carried out. As a result of the study, significant force loads on the structural elements and cargo oscillations on the flexible suspension were identified.

Dynamic loads during transitional processes (starting, braking) significantly impact crane structural elements [7]. Scientific papers [8-10] investigated dynamic processes in tower cranes, while papers [11, 12] investigated bridge cranes, particularly container handlers with motion control elements. Work [13] established that a slight reduction in dynamic loads during transitional processes significantly improves the reliability of lifting cranes. In addition, dynamic loads have a significant impact on the accuracy of cargo positioning [14].

The swaying of the cargo on a flexible suspension significantly impacts the dynamics of jib cranes, which has been the subject of considerable attention by researchers studying the dynamics of lifting cranes. In [15], dynamic modelling was performed, and a self-adjusting control against the swaying of the cargo of a ship crane with a slewing mechanism was proposed. Here, the crane is represented by a mechanical system with seven degrees of freedom. The authors of [16] propose reliable control against cargo swaying while lifting and lowering in a vertical plane. A group of researchers [17] optimised a nonlinear input data shaper to suppress cargo oscillations on a flexible suspension during crane manoeuvring from one position to another. In [18], the command system of a nonlinear bridge crane system was optimised for manoeuvring between specific positions. The authors of [19] solved the control task for a linear model and an S-shaped curve of the speed trajectory of cargo oscillation suppression in bridge cranes with a double pendulum effect. The researchers in [20] constructed a model of the combined movement of the lifting, turning, and extension mechanisms of the jib of the DEK-251 crane. In [21], a transport control system for double pendulum cranes with a nonlinear quasi-PID system was developed, which minimizes cargo oscillations on a flexible suspension. Researchers [22] developed a gantry crane control system using a hybrid input data shaper and a PID controller, eliminating cargo oscillations. The authors of the article [23] on a non-active double pendulum device conducted a dynamic analysis and experimental research on anti-sway devices for ship jib cranes. In [24], a dynamic analysis and experimental research were performed on a device to prevent cargo sway for ship cranes, which minimizes sway. In [25], a group of authors conducted a dynamic analysis and time-optimal control against a double pendulum bridge crane oscillations with distributed bridge beams.

Based on the review, it can be concluded that to identify loads and oscillations in the structural elements of a jib crane, it is necessary to perform a dynamic analysis during simultaneous motion of the jib lifting and jib crane turning mechanisms.

Purpose of the paper. The purpose of this work was to construct a mathematical model of motion dynamics and perform a dynamic analysis of the crane jib system during simultaneous motion of the jib lifting and crane turning mechanisms.

Research results. When constructing a dynamic model of the simultaneous motion of the jib lifting and turning mechanisms of a jib crane, the main kinematic, dynamic, and energy properties of crane mechanisms used in previous research on the dynamics of individual mechanisms or their simultaneous operation were taken into account [4-6].

The dynamic model of the crane jib system is presented in the form of absolutely rigid and dissipative links of the jib lifting and crane turning mechanisms (Fig. 1). In this case, we assume that the flexible suspension rope of the cargo has dissipative properties, while the jib, cargo, and drive elements are absolutely rigid bodies. The flexible cargo suspension in a pulley system performs spatial pendulum oscillations in the plane of crane outreach and turning. The links of the jib lifting mechanism drive are reduced to its axis of rotation, and the links of the turning mechanism drive are reduced to the crane turning axis. The dynamic model of simultaneous motion of the mechanisms considers the main movement of the jib lifting and crane turning drives, and the spatial oscillations of the cargo on a flexible suspension.

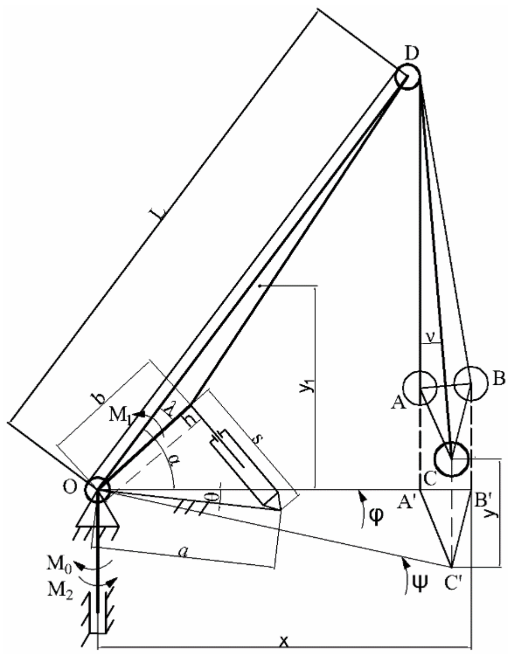


Fig. 1. Dynamic model of a crane jib system with simultaneous motion of the jib lifting and turning mechanisms

Thus, the dynamic model of a jib crane with simultaneous motion of the jib lifting and crane turning mechanisms is represented by a mechanical system with four degrees of freedom (Fig. 1). The following designations are used in this figure: O – position of the lower jib hinge; A, B, C – positions of the centre of gravity of the cargo at rest, when deflected in the plane of reach change and deflected in space, respectively; A^I, B^I, C^I – projections of points A, B, C onto the horizontal plane passing through the axis of the lower jib hinge (point O). The generalised coordinates of the dynamic model shown use the angular coordinates of the jib rotation in the plane of reach change – α, the crane – φ, and the cargo – ψ, as well as the linear horizontal coordinate of the centre of gravity of the cargo (cargo reach) – x. The jib is subjected to resistance forces from the weight of the jib and cargo, and the slewing mechanism is subjected to the moment of static resistance forces – M_0, which is reduced to the crane turning axis. There is also a driving force in the drive hydraulic cylinder of the jib lifting mechanism, which creates a moment M_1 relative to its axis of rotation, and the driving moment of the hydraulic motor of the crane turning mechanism is reduced to the axis of rotation of the crane turning – M_2. In addition, a dissipative moment acts in the flexible cargo suspension when deviating from the vertical.

Lagrange equations of the second kind were used in constructing a mathematical model of the simultaneous motion of the boom lifting and slewing mechanisms of a jib crane (Fig. 1)

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\alpha}} - \frac{\partial T}{\partial \alpha} = M_1 - \frac{\partial \Pi}{\partial \alpha} - \frac{\partial R}{\partial \dot{\alpha}};$$
$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\varphi}} - \frac{\partial T}{\partial \varphi} = M_2 - \frac{\partial \Pi}{\partial \varphi} - \frac{\partial R}{\partial \dot{\varphi}};$$
$$\frac{d}{dt}\frac{\partial T}{\partial \dot{x}} - \frac{\partial T}{\partial x} = -\frac{\partial \Pi}{\partial x} - \frac{\partial R}{\partial \dot{x}};$$
$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\psi}} - \frac{\partial T}{\partial \psi} = -\frac{\partial \Pi}{\partial \psi} - \frac{\partial R}{\partial \dot{\psi}}, \qquad (1)$$

where T, Π, R are kinetic and potential energy, respectively, and the dissipative function of Rayleigh for the simultaneous motion of the jib lifting and crane turning mechanisms; M_1, M_2 are 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.

The kinetic energy of the simultaneous motion of the jib lifting and crane turning mechanisms is represented by the following relationship

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

where J_1, J_2 are accordingly, the moments of inertia of the jib relative to the axis of its lower hinge, and the drive of the crane turning mechanism and the slewing platform are reduced to the crane turning axis; m is the masses of cargo; x, y are horizontal and vertical coordinates of the centre of mass of the cargo in the plane of change in lift; L is the jib length; α − angular coordinate of jib rotation.

The potential energy of the simultaneous motion of the jib lifting and crane turning mechanisms is determined by the following relationship

$$\Pi=(m_1y_1+my)g\ , \tag{3}$$

where g – acceleration of gravity; m_1, y_1 are the mass of the jib and its vertical coordinate of the centre of mass.

The dissipative function of the Relay system has the following form

$$R=\frac{1}{2}b\dot{v}^2\ , \tag{4}$$

where b is the damping coefficient of the flexible suspension of the cargo when deviating from the vertical; v is the angular coordinate of the deviation from the vertical of the flexible suspension of the cargo, which is determined by the following dependence

$$v=\frac{1}{l}\sqrt{(x-L\cos\alpha)^2+x^2(\kappa-\psi)^2-2x(x-L\cos\alpha)(\varphi-\psi)\sin\left[\frac{(\varphi-\psi)}{2}\right]}. \tag{5}$$

Dependence (5) obtained from geometric considerations when considering trianglesΔABC, $\Delta A^IB^IC^I$, ΔOB^IC^I and ΔACD.

The vertical coordinates of the centres of mass of the jib and cargo and their derivatives over time are determined as follows:

$$y_1=kL\sin\alpha,\quad \dot{y}_1=\dot{\alpha}kL\cos\alpha, \tag{6}$$

$$y=L\sin\alpha-l\cos v,\quad y=\dot{\alpha}L\cos\alpha-\dot{v}l\sin v\ . \tag{7}$$

Taking into account expressions (5) – (7), the kinetic and potential energy of the jib system during the simultaneous motion of the jib lifting and crane turning mechanisms can be represented as follows:

$$T=\frac{1}{2}J_1\dot{\alpha}^2+\frac{1}{2}\left[J_1(\cos\alpha)^2+J_2\right]\dot{\varphi}^2+\frac{1}{2}m\left(\dot{x}^2+\dot{y}^2+x^2\dot{\psi}^2\right), \tag{8}$$

$$\Pi=(m_1k+m)gL\sin\alpha-mgl\cos v. \tag{9}$$

Partial derivatives of kinetic energy (9) with respect to the generalised coordinates of the jib system are defined as follows:

$$\frac{\partial T}{\partial\alpha}=\frac{1}{2}J_1\sin 2\alpha\dot{\varphi}^2+m\dot{y}\frac{\partial\dot{y}}{\partial s},\ \frac{\partial T}{\partial x}=m\left(x\dot{\psi}^2+\dot{y}\frac{\partial\dot{y}}{\partial x}\right),\ \frac{\partial T}{\partial\varphi}=m\dot{y}\frac{\partial\dot{y}}{\partial\varphi},\ \frac{\partial T}{\partial\psi}=m\dot{y}\frac{\partial\dot{y}}{\partial\psi}. \tag{10}$$

Partial derivatives of kinetic energy (9) with respect to the generalised velocities of the system are also found:

$$\frac{\partial T}{\partial\dot{\alpha}}=J_1\dot{\alpha}+m\dot{y}\frac{\partial y}{\partial s},\ \frac{\partial T}{\partial\dot{x}}=m\left(\dot{x}+\dot{y}\frac{\partial y}{\partial x}\right),\ \frac{\partial T}{\partial\dot{\varphi}}=\left(J_1(\cos\alpha)^2+J_2\right)\dot{\varphi}+m\dot{y}\frac{\partial y}{\partial v},\ \frac{\partial T}{\partial\dot{\psi}}=m\left(x^2\dot{\psi}+\dot{y}\frac{\partial y}{\partial\psi}\right). \tag{11}$$

Now, let us take the full-time derivatives of expressions (12), which will give us:

$$\frac{d}{dt}\frac{\partial T}{\partial\dot{\alpha}}=J_1\ddot{\alpha}+m\left(\ddot{y}\frac{\partial y}{\partial s}\right)+\dot{y}\frac{\partial\dot{y}}{\partial s}, \tag{12}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\varphi}} = \left(J_1(\cos\alpha)^2 + J_2\right)\ddot{\varphi} - J_1\dot{\alpha}\dot{\varphi}\sin 2\alpha + m\left(\ddot{y}\frac{\partial y}{\partial \varphi} + \dot{y}\frac{\partial \dot{y}}{\partial \varphi}\right), \tag{13}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{x}} = m\left(\ddot{x} + \ddot{y}\frac{\partial y}{\partial x} + \dot{y}\frac{\partial \dot{y}}{\partial x}\right), \tag{14}$$

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\psi}} = m\left(\ddot{y}\frac{\partial y}{\partial \psi} + \dot{y}\frac{\partial \dot{y}}{\partial \psi} + x^2\ddot{\psi} + 2x\dot{x}\dot{\psi}\right). \tag{15}$$

Let us find the partial derivatives of the potential energy (9) with respect to the generalised coordinates:

$$\frac{\partial \Pi}{\partial \alpha} = \left(m_1 k + m\right) gl\cos\alpha + mgl\frac{\partial \nu}{\partial \alpha}\sin\nu, \tag{16}$$

$$\frac{\partial \Pi}{\partial \varphi} = mgl\frac{\partial \nu}{\partial \varphi}\sin\nu; \frac{\partial \Pi}{\partial \psi} = mgl\frac{\partial \nu}{\partial \psi}\sin\nu; \frac{\partial \Pi}{\partial x} = mgl\frac{\partial \nu}{\partial x}\sin\nu. \tag{17}$$

Now, let us determine the partial derivatives of the Rayleigh function (4) with respect to the generalised velocities:

$$\frac{\partial R}{\partial \dot{\alpha}} = b\dot{\nu}\frac{\partial \nu}{\partial \alpha},\ \frac{\partial R}{\partial \dot{x}} = b\dot{\nu}\frac{\partial \nu}{\partial x},\ \frac{\partial R}{\partial \dot{\varphi}} = b\dot{\nu}\frac{\partial \nu}{\partial \varphi},\ \frac{\partial R}{\partial \dot{\psi}} = b\dot{\nu}\frac{\partial \nu}{\partial \psi}. \tag{18}$$

As a result of substituting expressions (12), …, (19) into system (1), we obtain a mathematical model of the dynamics of the simultaneous motion of the jib lifting and crane turning mechanisms.

$$\begin{gathered}
J_1\ddot{\alpha} - \frac{1}{2}J_1\dot{\varphi}^2\sin 2\alpha + m\ddot{y}\frac{\partial y}{\partial \alpha} = M_1 - \left(m_1 k + m\right) gl\cos\alpha - M_\nu\frac{\partial \nu}{\partial \alpha}, \\
\left(J_1(\cos\alpha)^2 + J_2\right)\ddot{\varphi} - J_1\dot{\alpha}\dot{\varphi} + m\ddot{y}\frac{\partial y}{\partial \varphi} = M_2 - M_0 - M_\nu\frac{\partial \nu}{\partial \varphi}, \\
m\left[\ddot{x} + \ddot{y}\frac{\partial y}{\partial x} - 2x\dot{\psi}^2\right] = -M_\nu\frac{\partial \nu}{\partial x},\quad m\left[\ddot{\psi}x^2 + 2x\dot{x}\dot{\psi} + \ddot{y}\frac{\partial y}{\partial \psi}\right] = -M_\nu\frac{\partial \nu}{\partial x}.
\end{gathered} \tag{19}$$

In the system of equations (19), the moment M_ν is determined by the following dependence

$$M_\nu = mgl\nu + b_1\dot{\nu}. \tag{20}$$

The system of differential equations (19) includes partial and total derivatives of the coordinates of individual links and points of the jib system during the simultaneous motion of the jib lifting and crane turning mechanisms. Let us determine these derivatives for the dynamic model of the jib system shown in Fig. 1.

The first complete derivative of the vertical coordinate of the centre of mass of the cargo with respect to time from expressions (7) is found

$$\dot{y} = \dot{\alpha}\frac{\partial y}{\partial \alpha} + \dot{x}\frac{\partial y}{\partial x} + \dot{\varphi}\frac{\partial y}{\partial \varphi} + \dot{\psi}\frac{\partial y}{\partial \psi}. \tag{21}$$

Similarly, the first full derivative with respect to time of the angular coordinate of deviation from the vertical of the flexible suspension of the cargo from expression (8) is determined

$$\dot{\nu} = \dot{\alpha}\frac{\partial \nu}{\partial \alpha} + \dot{x}\frac{\partial \nu}{\partial x} + \dot{\varphi}\frac{\partial \nu}{\partial \varphi} + \dot{\psi}\frac{\partial \nu}{\partial \psi}. \tag{22}$$

For the presented dynamic model of simultaneous motion of the jib lifting and crane turning mechanisms, the partial derivatives of the vertical coordinate of the cargo are determined by the generalised coordinates:

$$\frac{\partial y}{\partial \alpha} = L\cos\alpha + l\nu\frac{\partial \nu}{\partial \alpha};\ \frac{\partial y}{\partial x} = l\nu\frac{\partial \nu}{\partial x};\ \frac{\partial y}{\partial \varphi} = l\nu\frac{\partial \nu}{\partial \varphi};\ \frac{\partial y}{\partial \psi} = l\nu\frac{\partial \nu}{\partial \psi}. \tag{23}$$

Since the deviation from the vertical of the flexible suspension of the cargo does not exceed 120, in formulas (22) it is assumed that sinν≈ν, cosν≈1. Formula (22) includes partial derivatives of the angular coordinate of the deviation from the vertical of the flexible suspension of the cargo, so let us determine them as well:

$$\frac{\partial \nu}{\partial \alpha}=\frac{L\sin\alpha}{l^2\nu}\left\{x\left[1-\frac{(\varphi-\psi)^2}{2}\right]-L\cos\alpha\right\}, \tag{24}$$

$$\frac{\partial \nu}{\partial x}=\frac{1}{l^2\nu}\left\{x-\left[1-\frac{(\varphi-\psi)^2}{2}\right]L\cos\alpha\right\}. \tag{25}$$

As a result of substituting expressions (23), …, (25) into dependence (21), we will have

$$\dot{y}=\dot{\alpha}L\left\{\cos\alpha+\frac{\sin\alpha}{l}\left[x\left(1-\frac{(\varphi-\psi)^2}{2}\right)-L\cos\alpha\right]\right\}+\frac{\dot{x}}{l}\left\{x-\left[1-\frac{(\varphi-\psi)^2}{2}\right]L\cos\alpha\right\}+\frac{L}{l}(\dot{\varphi}-\dot{\psi})(\varphi-\psi)x\cos\alpha. \tag{26}$$

Taking the time derivative of expression (26), we obtain

$$\begin{gathered}\ddot{y}=\ddot{\alpha}L\left\{\cos\alpha+\sin\alpha\left[x\left(1-\frac{(\varphi-\psi)^2}{2}\right)-\frac{L\cos\alpha}{l}\right]\right\}+\dot{\alpha}L\left\{\frac{1}{l}\dot{\alpha}(-\sin\alpha+\cos\alpha)\left[x\left(1-\frac{(\varphi-\psi)^2}{2}\right)-L\cos\alpha\right]+\right.\\ \left.+\frac{1}{l}\sin\alpha\left[\dot{x}\left(1-\frac{(\varphi-\psi)^2}{2}\right)-x(\dot{\varphi}-\dot{\psi})(\varphi-\psi)+\dot{\alpha}L\sin\alpha\right]\right\}+\frac{\ddot{x}}{l}\left[x-\left[1-\frac{(\varphi-\psi)^2}{2}\right]L\cos\alpha\right]+\\ +\frac{\dot{x}}{l}\left[\dot{x}+(\dot{\varphi}-\dot{\psi})(\varphi-\psi)L\cos\alpha+\dot{\alpha}L\sin\alpha\left(1-\frac{(\varphi-\psi)^2}{2}\right)\right]+\\ +\frac{L}{l}\left\{x\cos\alpha\left[(\ddot{\varphi}-\ddot{\psi})(\varphi-\psi)+(\dot{\varphi}-\dot{\psi})^2\right]+(\dot{x}-\dot{\alpha}x)(\dot{\varphi}-\dot{\psi})(\varphi-\psi)\cos\alpha\right\}.\end{gathered} \tag{27}$$

Let us determine the characteristics of the drives for the jib lifting and crane turning mechanisms. The jib lifting drive is located in the plane of reach change and consists of a hydraulic cylinder attached to the rotating platform with a sleeve and a rod to the jib. The crane turning drive is on the rotating platform and consists of a hydraulic motor, a planetary gearbox, and an open cylindrical transmission.

The torque that rotates the crane jib is found through the force in the hydraulic cylinderF_1, the method for determining which is given in [5].

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

There

$$h=\frac{\sqrt{4a^2s^2-\left(a^2-b^2+s^2\right)^2}}{2s},\quad s=\sqrt{a^2+b^2-2ab\cos(\theta-\lambda+\alpha)}. \tag{29}$$

The force dependence in the hydraulic cylinder of the jib lifting mechanism is determined by the formulas given in [5]:

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

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

Here, h is the lever arm of the forceF_1; p_n is the working pressure in the hydraulic cylinder cavity; A is the cross-sectional area of the hydraulic cylinder piston; Q is the working fluid flow rate through the hydraulic cylinder; α, θ are the length of the hydraulic cylinder mounting bracket and its angle of inclination to the horizontal; b, $\lambda$are the length of the hydraulic cylinder lever acting 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 crane turning mechanism is determined by a similar dependence, which is given in [5] for the cargo lifting mechanism, but reduced to the crane turning axis. It is represented by a quadratic dependence on the angular velocity of the crane turning and has the following form:

$$M_2 = M_p + \left(K\omega_0 - \left(M_p/\omega_0\right)\right)\dot{\varphi} - K\dot{\varphi}^2, \qquad (32)$$

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

where M_p, M_n are the starting and rated torques of the crane turning mechanism hydraulic motor, respectively, reduced to the crane turning axis; ω_n, $\omega_0$are the rated and synchronous angular velocities of the crane turning mechanism hydraulic motor shaft, reduced to the crane turning axis.

Research results

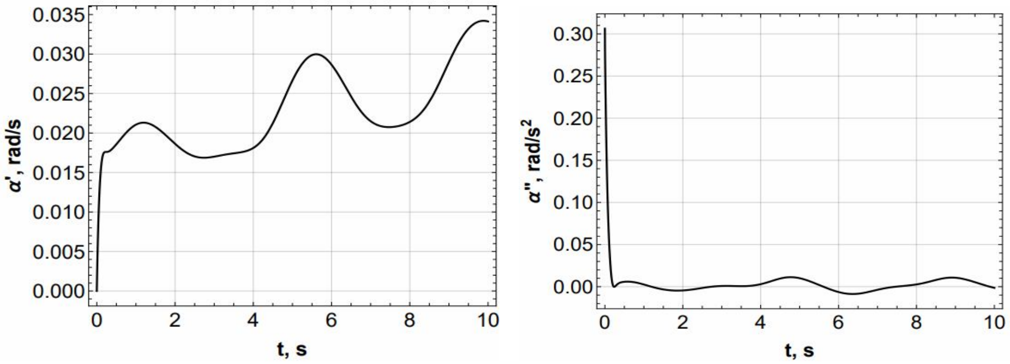
The system of equations (20) for the simultaneous motion of the jib lifting and crane turning mechanisms, together with expressions (21), ..., (34), is a nonlinear system of second-order differential equations, therefore a numerical method is used to solve it. The equations are solved under the following initial conditions of simultaneous motion of the jib lifting and crane turning mechanisms:

$$t = 0: s = s_0, \dot{s} = 0, x = x_0\ \dot{x} = 0, \varphi = \varphi_0, \dot{\varphi} = 0, \psi = \psi_0, \dot{\psi} = 0. \qquad (34)$$

A computer program has been developed to perform a dynamic analysis of the simultaneous motion of the jib boom lifting and slewing mechanisms of a jib crane. The programme consists of the following main blocks: 1) initial parameters of the jib lifting and jib turning mechanisms of the crane required for the calculation; 2) numerical solution of a system of second-order nonlinear differential equations of the simultaneous motion of the jib lifting and jib turning mechanisms of the crane; 3) calculation of the kinematic characteristics of the mechanism links; 4) calculation of the dynamic (force) characteristics of the mechanism links; 5) calculating the energy characteristics of the mechanism links; 6) constructing graphical dependencies of the kinematic characteristics of the mechanism links; 7) construction of graphical dependencies of the dynamic characteristics of the mechanism links; 8) construction of graphical dependencies of the energy characteristics of the mechanism links; 9) determination of the maximum, arithmetic mean and root mean square values of the kinematic, dynamic and energy characteristics of the simultaneous motion of mechanisms.

Dynamic analysis of the simultaneous movement of mechanisms for changing the reach and lifting of cargo during steady crane turning was performed using the following values for the crane jib system parameters:m = 4500 kg, m_l = 2700 kg, J_1=72900 kg· m^2, J_2=38770 kg· m^2, l=4,5 m, L = 9,0 m, η=0,85, p_n=14·10^{6}N/m^2, A=0,025 m^2, Q=0,0022m^3/s, ω_0= 0,175rad/s, ω_n=0,157 rad/s, g = 9,81 m/s^2, M_0=36800 Nm, M_n=38500 Nm, M_p=52100 Nm, a=1,5m, b=2,1 m, λ=0,0872 rad, θ=0,3189 rad, α_0=0,5857 rad, x_0=7,5 m, φ_0=0, ψ_0=0, b_1=36 Nm/(rad/s), k=0,5.

After numerical solution of the nonlinear system of differential equations (19) taking into account expressions (9),…, (32) under the initial conditions (32) of the simultaneous motion of the jib lifting and crane turning mechanisms, graphical dependencies of the kinematic (Fig. 2, …, 5), dynamic (Fig. 6) and energy (Fig. 7) characteristics were constructed.

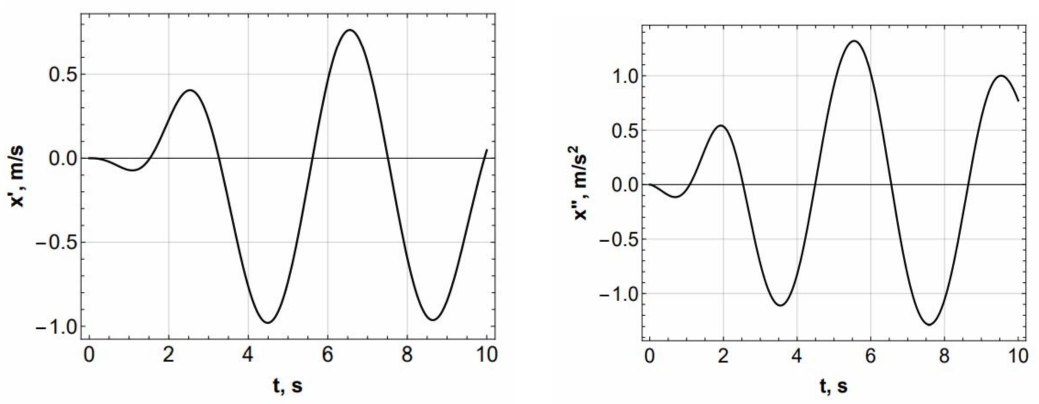


(a) (b)

Fig. 1.Graphs of angular velocity (a) and acceleration (b) of jib rotation

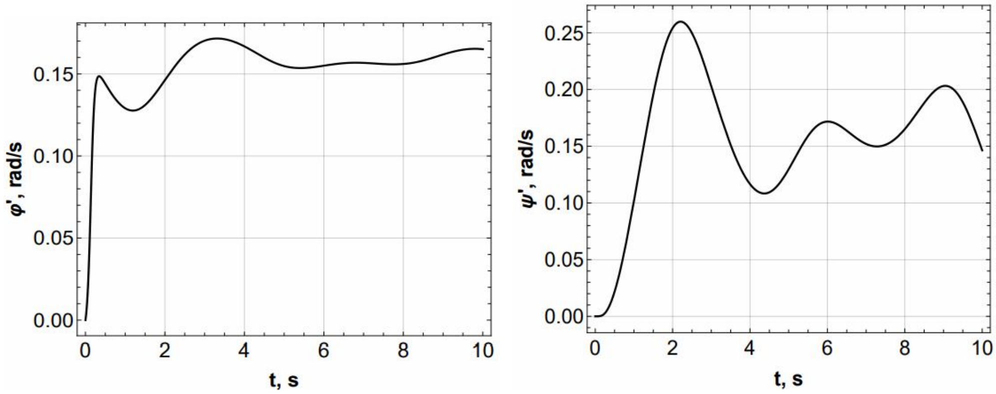
From Fig. 2 (a), it can be seen that the angular velocity of the jib rotation increases to a steady value in oscillatory mode. At the same time, at the beginning of the start-up, this increase is quite rapid for 0.25 seconds. Subsequently, the increase is slower, but with oscillations caused by the character of the change in driving torque (Fig. 6 (a)). The angular acceleration of the jib rotation at the start of the start-up drops sharply from the maximum value (0.31 rad/s^2) to zero with subsequent slight oscillations relative to zero (Fig. 2 (b)).

The process of changing the speed of the cargo, shown in Fig. 3, is also carried out in an oscillatory mode, which leads to the appearance of a significant maximum value of the cargo acceleration, which is 1.3 m/s^2 (Fig. 3 (b)). This results in increased dynamic loads acting on the cargo and transmitted to the crane's metal structures. The oscillations in the speed and acceleration of the cargo when changing the outreach are practically undamped in the considered section of movement.



(a) (b)

Fig. 3. Graphs of velocity (a) and acceleration (b) of the change in the centre of mass of the cargo



(a) (b)

Fig. 4. Graphs of angular rotation speeds of the jib (a) and crane (b)

At the beginning of the crane turning mechanism start-up process, the angular velocity (Fig. 4 (a)) of the rotating platform changes quite rapidly and reaches a value of 0.15 rad/s after 0.5 s, followed by a steady state in oscillatory mode. At the same time, the angular speed of the cargo rotation increases more slowly and after 2.2 s of movement reaches its maximum value of 0.26 rad/s. Subsequently, the angular speed of the cargo changes in an oscillatory mode around a steady state value. In this mode of rotation, the dynamic loads increase significantly, which negatively affects the metal structures of the crane and the drive of the rotation mechanism.

Fig. 5 shows a phase portrait of low-frequency oscillations of a cargo on a flexible suspension during simultaneous motion of the jib lifting and crane turning mechanisms, which shows that these oscillations are practically undamped during the operation of these jib crane mechanisms. The phase portrait shows that complex oscillatory processes occur when the cargo oscillations overlap during changes in the crane's reach and crane turning. At the same time, the maximum deviation from the vertical of the flexible cargo suspension is 0.178 rad (10.20), and the maximum deviation speed is observed at the start of the jib lifting and crane turning mechanisms and is 0.23 rad/s.

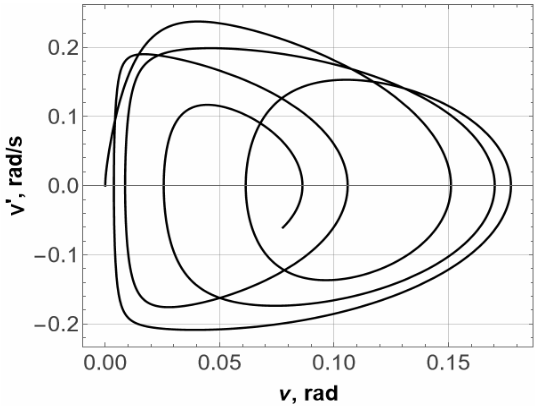
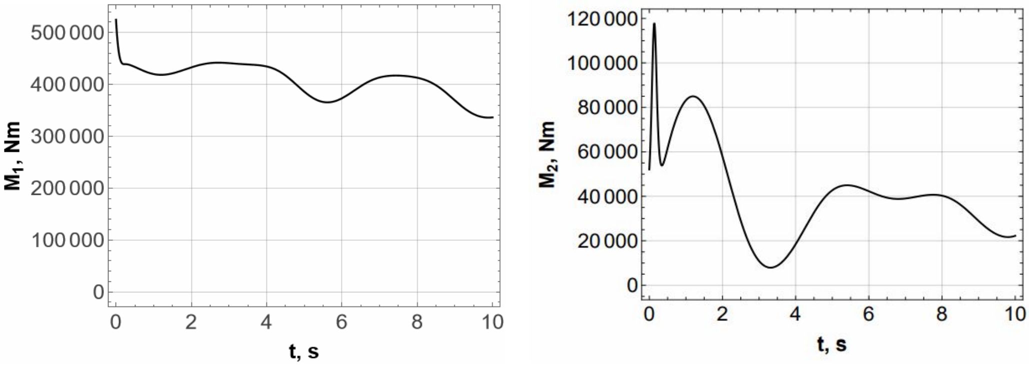


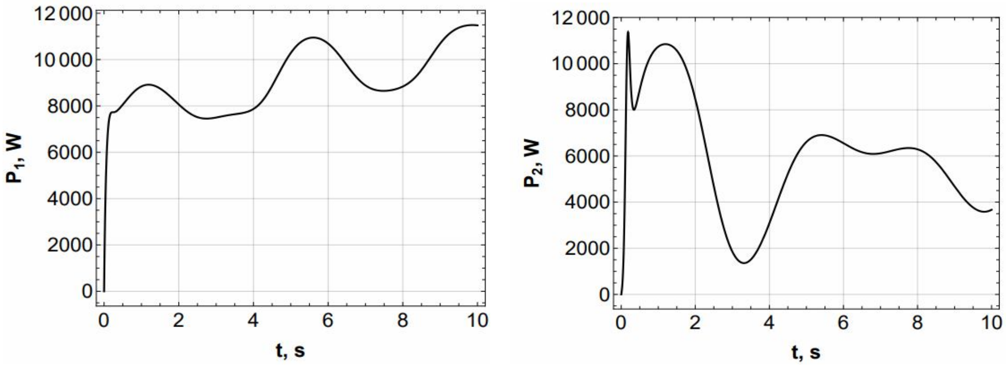
Fig. 5. Phase portrait of cargo oscillations on a flexible suspension

The driving torque of the jib lifting mechanism (Fig. 6 (a)) varies from the maximum (525 kNm) to the minimum (340 kNm) values in oscillatory mode. At the same time, at the start of the start-up, the torque decreases sharply to 440 kNm within 0.25 s. This mode of change in the driving torque of the jib lifting mechanism leads to oscillatory processes and, as a result, significantly increases the dynamic loads on the crane structure and drive mechanism components. The driving torque of the crane turning mechanism drive (Fig. 6 (b)) increases from the start value (52 kNm) to the maximum (118 kNm) within 0.2 s at the start, and then drops sharply to the initial value. Subsequently, the driving torque changes in an oscillatory mode to a steady value. This mode of change in the driving torque of the crane turning mechanism leads to cargo oscillations on the flexible suspension and an increase in dynamic loads on the structural elements.



(a) (b)

Fig. 6. Graphs showing changes in the driving torques of the mechanisms for lifting the jib (a) and crane turning (b)



(a) (b)

Fig. 7. Power change graphs for jib lifting (a) and crane turning (b) drives

The power of the jib lifting mechanism drive (Fig. 7 (a)) increases sharply to 7.8 kW at the start of operation, followed by an increase in oscillatory mode to a steady value of 11.5 kW. At the same time, no damping of the power oscillations of the jib lifting mechanism is observed. The power of the crane turning mechanism drive (Fig. 7 (b)) initially increases instantly to a maximum value of 11.5 kW, and then also drops sharply to 8.0 kW (Fig. 7 (b)). Subsequently, the power of the crane turning mechanism drive gradually decreases in oscillatory mode to 3.75 kW. The oscillatory nature of the power change of the drives of the simultaneous movement of the jib lifting and crane turning mechanisms negatively affects the energy characteristics of the crane.

Conclusions. As a result of research into the dynamics of the simultaneous movement of the jib lifting and turning mechanisms of a jib crane, the following conclusions were made:

1. A dynamic model of the simultaneous movement of the jib lifting and turning mechanisms of a jib crane was developed. The presented dynamic model takes into account the main movement of the drive mechanisms for lifting the jib and turning the crane, as well as low-frequency pendulum spatial oscillations of the load on a flexible suspension. The dynamic model takes into account the characteristics of the driving force of the jib lifting hydraulic cylinder and the hydraulic drive of the crane turning mechanism. The basis for constructing a mathematical model of the simultaneous movement of the jib lifting and crane turning mechanisms was their dynamic model. The mathematical model is described by a system of four nonlinear second-order differential equations, which are solved numerically using a developed computer program.

2. As a result of solving the constructed mathematical model, a dynamic analysis of the simultaneous movement of the jib lifting and crane turning mechanisms was performed. The results of the dynamic analysis revealed significant fluctuations in the kinematic, dynamic, and energy characteristics of the jib system during the simultaneous movement of the jib lifting and crane turning mechanisms. The presence of oscillatory processes led to an increase in dynamic loads in the drives and structure of the jib crane. The instantaneous change in the driving forces of the drive mechanisms caused virtually undamped oscillations in the characteristics of the jib system. To reduce oscillations, it is proposed to control the start modes of the drives of the jib lifting and crane turning mechanisms.

3. Based on the results of a dynamic analysis of the simultaneous movement of the jib lifting and turning mechanisms of a jib crane, the cause of virtually undamped oscillations of the load on a flexible suspension has been identified, which is determined by the nature of the change in the driving forces of the drives. In particular, the almost instantaneous change in the driving forces of the drive mechanisms has a significant impact on the oscillatory processes. To minimize the oscillatory processes in the structural elements of the jib system, it is necessary to optimize the starting and braking modes of the drive mechanisms.

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Ловейкін В. С., Ромасевич Ю. О., Ловейкін Ю. В., Ляшко А. П., Почка К. І.
ДИНАМІЧНИЙ АНАЛІЗ ОДНОЧАСНОГО РУХУ МЕХАНІЗМІВ ПІДЙОМУ СТРІЛИ ТА ПОВОРОТУ КРАНА

Ефективність роботи стрілових кранів в значній мірі залежить від підвищення їхньої продуктивності при виконанні навантажувальних, розвантажувальних та монтажних операцій. Підвищення продуктивності стрілових кранів досягається за рахунок здійснення одночасної роботи окремих механізмів. Метою наведеного дослідження є побудова математичної моделі та здійснення динамічного аналізу стрілової системи крана при одночасній роботі механізмів підйому стріли та повороту крана. Наведені дослідження базуються на методах побудови дискретних динамічних моделей стрілового крана із застосуванням рівнянь Лагранжа другого роду, чисельних методах розв'язування нелінійних звичайних диференціальних рівнянь, які представлені у вигляді комп'ютерної програми та методах динамічного аналізу одночасного руху кранових механізмів. В наведеній статті розв'язується задача динаміки одночасного руху механізмів підйому стріли та повороту крана. Стрілова система крана представлена динамічною моделлю з чотирма ступенями свободи, де враховано основний рух механізмів підйому стріли та повороту крана і коливання вантажу на гнучкому підвісі. В результаті проведеного динамічного аналізу визначені кінематичні, динамічні та енергетичні характеристики окремих ланок стрілової системи крана при одночасній роботі механізмів підйому стріли та повороту крана. Досліджено основний рух приводних механізмів підйому стріли та повороту крана, а також низькочастотні просторові коливання вантажу на гнучкому підвісі. Встановлено, що динаміка руху механізмів залежить від характеру зміни рушійних сил приводів, а низькочастотні коливання вантажу на гнучкому підвісі практично не затухають і тривають протягом всього циклу руху.

Для покращення динаміки одночасного руху механізмів та мінімізації коливальних процесів ланок стрілової системи рекомендовано обирати режими плавної зміни рушійних сил приводів на ділянках перехідних процесів (пуск, гальмування), які забезпечують бажаний рух виконавчих механізмів та приводять до зниження навантажень.

Ключові слова: механізм підйому стріли, механізм повороту крана, коливальні процеси, рушійні сили, динамічні навантаження.

Loveikin V.S., Romasevych Yu.O., Loveikin Yu.V., Liashko A.P., Pochka K.I.
DYNAMIC ANALYSIS OF THE SIMULTANEOUS MOVEMENT OF THE JIB LIFTING AND CRANE TURNING MECHANISMS

The efficiency of jib cranes largely depends on increasing their productivity when performing loading, unloading, and installation operations. The simultaneous operation of individual mechanisms increases the productivity of jib cranes. The purpose of this study is to construct a mathematical model and perform a dynamic analysis of the crane jib system during the simultaneous operation of the jib lifting and crane turning mechanisms. The presented research is based on methods for constructing discrete dynamic models of a jib crane using second-order Lagrange equations, numerical methods for solving nonlinear ordinary differential equations, which are presented in the form of a computer program, and methods for dynamic analysis of the simultaneous motion of crane mechanisms. The problem of the dynamics of simultaneous motion of the jib lifting and crane turning mechanisms is solved in this article. The crane jib system is represented by a dynamic model with four degrees of freedom, which considers the main motion of the jib lifting and crane turning mechanisms and the oscillation of the cargo on a flexible suspension. As a result of the dynamic analysis, the kinematic, dynamic, and energy characteristics of individual links of the crane jib system are determined during the simultaneous operation of the jib lifting and crane turning mechanisms. The main movement of the drive mechanisms for lifting the jib and crane turning, as well as the low-frequency spatial oscillations of the cargo on a flexible suspension, were investigated. It has been established that the dynamic motion of the mechanisms depends on the character of the change in the driving forces of the drives. Low-frequency oscillations of the cargo on a flexible suspension practically do not dampen and continue throughout the entire movement cycle.

To improve the dynamics of simultaneous motion of mechanisms and minimise oscillatory processes of the jib system links, it is recommended to select modes of smooth change of driving forces of drives in transition processes (starting, braking), which ensure the desired movement of executive mechanisms and lead to a reduction in loads.

Keywords: jib lifting mechanism, crane turning mechanism, oscillatory processes, driving forces, dynamic loads.

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Fig. 7. Ref. 25.

Автор (науковий ступінь, вчене звання, посада): доктор технічних наук, професор, завідувач кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України ЛОВЕЙКІН Вячеслав Сергійович
Адреса: 03041, Україна, м. Київ, вул. Героїв Оборони, 12, навчальний корпус № 11, Національний університет біоресурсів і природокористування України, кафедра конструювання машин і обладнання, ЛОВЕЙКІНУ Вячеславу Сергійовичу
Тел.: +38(044) 527-87-34
E-mail: lovvs@ukr.net
ORCID ID: https://orcid.org/0000-0003-4259-3900

Автор (науковий ступінь, вчене звання, посада): доктор технічних наук, професор, професор кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України РОМАСЕВИЧ Юрій Олександрович
Адреса: 03041, Україна, м. Київ, вул. Героїв Оборони, 12, навчальний корпус № 11, Національний університет біоресурсів і природокористування України, кафедра конструювання машин і обладнання, РОМАСЕВИЧУ Юрію Олександровичу
Тел.:+38(044) 527-87-34
E-mail: romasevichyuriy@ukr.net
ORCID ID: https://orcid.org/0000-0001-5069-5929

Автор (науковий ступінь, вчене звання, посада): кандидат фізико-математичних наук, доцент, доцент диференціальних та інтегральних рівнянь Київського національного університету імені Тараса Шевченка ЛОВЕЙКІН Юрій Вячеславович
Адреса: 03022, Україна, м. Київ, проспект академіка Глушкова, 4е, корпус механіко-математичного факультету, Київський національний університет імені Тараса Шевченка, кафедра математичної фізики, ЛОВЕЙКІНУ Юрію Вячеславовичу
E-mail: y.loveikin@gmail.com
ORCID ID: https://orcid.org/0000-0003-4570-563X

Автор (науковий ступінь, вчене звання, посада): кандидат технічних наук, доцент, старший викладач кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України ЛЯШКО Анастасія Петрівна
Адреса: 03041, Україна, м. Київ, вул. Героїв Оборони, 12, навчальний корпус № 11, Національний університет біоресурсів і природокористування України, кафедра конструювання машин і обладнання, ЛЯШКО Анастасії Петрівні
Тел.: +38(044) 527-87-34;
E-mail: liashko@nubip.edu.ua
ORCID ID: https://orcid.org/0000-0002-3774-3348

Автор (науковий ступінь, вчене звання, посада): доктор технічних наук, професор, член-кореспондент НАПН України, завідувач кафедри професійної освіти Київського національного університету будівництва і архітектури ПОЧКА Костянтин Іванович
Адреса: 03037, Україна, м. Київ, проспект Повітряних Сил, 31, кафедра професійної освіти, КНУБА ПОЧЦІ К.І.
E-mail: pochka.ki@knuba.edu.ua
ORCID ID: https://orcid.org/0000-0002-0355-002X