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ANALYSIS OF DERRIKINGAND SLEWING OF THE TOWER CRANE WITH CONSIDERATION TO DRIVING MECHANISMS CHARACTERISTICS

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Based on the constructed dynamic model, the differential equations of motion of the mechanical part of the mechanisms and the dynamic electromagnetic transients of the drive electric motors are composed. As a result of solving the obtained equations for the numerical parameters of the real tower crane, a dynamic and energy analysis of the over lapping work of the derriking and slewing mechanisms was performed, which revealed significant dynamic and energy overloads of crane drive mechanisms and crane metal structure.

Keywords: tower crane, derriking, slewing, dynamic and mathematic models, dynamic and energy loads.

Introduction. Tower cranes with a saddle jib (boom) are widely used in construction operations, in which the load moves in space due to the operation of hoisting, derriking and slewing mechanisms. To increase the productivity of these cranes in many cases, the overlapping operation of two mechanisms is performed. There appeared considerable dynamic loads with oscillatory character in over lapping work of the derriking and slewing mechanisms in the construction elements and driving mechanisms. Especially dangerous are the spatial oscillations of the load on a flexible suspension during the transition processes (start, braking). At the same time, dynamic loads lead to a decrease in the reliability and productivity of the crane, as well as put at risk the work of slingers and crane operators due to the load oscillation. Based on the above, it can be stated that the problem of studying the dynamics of the over lapping movement of the derriking and slewing mechanisms of the crane is relevant. It reflects the real work conditions of tower cranes.

Analysis of publications. A significant amount of scientific work is devoted to the problem of eliminating of the load oscillations on a flexible suspension and reducing dynamic loads in the metal structures and mechanisms of tower cranes. Moreover, considerable attention was paid to determining the causes of load oscillations on the flexible suspension and methods and ways of its reducing, and if possible, complete elimination.

Studies of the motion of the mechanisms of the cranes slewing [1-2] were carried out, which revealed significant oscillations of the load on the flexible suspension, some solutions to its elimination were proposed. In studies [1-8] the problems of the dynamics of the over lapping movement of the derriking and slewing mechanisms were considered, and significant spatial oscillations of the load on the flexible suspension were revealed. In [1], the features of the electric drive control of the derriking mechanism during the slewing of the crane with the suspended load are established, which allows reducing its oscillations. The significant oscillations of the load on the flexible suspension were revealed by the research of the motion of the derriking and slewing mechanisms [5]. Therewith, in the dynamic model of the crane, the elastic and dissipative characteristics of the drive mechanisms have not been taken into account. The static mechanical characteristics of the drive motors were used.

However, it should be noted that a detailed analysis of the motion of the derriking and slewing mechanisms of the tower crane, which takes into account the elastic and dissipative features of the drive mechanisms with dynamic mechanical characteristics of the electric motors in transient processes, has not been investigated. Therefore, such investigations are relevant and require a more profound study.

Purpose of the paper. The purpose of the work is to study the dynamic processes of overlapping movement of the derriking and slewing mechanisms of the tower crane with a saddle jib, taking into account the dynamic mechanical characteristics of drive electric motors and elastic and dissipative features of the drive mechanisms.

Research results. The jib system of the tower crane is represented as a holonomic system (Fig. 1), which includes reduced to the axis of crane slewing absolutely solid masses of the slewing part of the crane with a jib and with a moment of inertia I_2, the slewing mechanism with the moment of inertia I_1, as well as reduced to the axis of rope drum rotation the mass of the drive of the derriking mechanism with the moment of inertia I_3. The masses of the slewing part of the crane and the drive slewing mechanism are interconnected by an elastic element with a stiffness coefficient C_1 and a dissipation coefficient b_1. The reduced mass of the drive mechanism of derriking is connected with the trolley (with mass m_3) by an elastic rope with stiffness coefficient C_3 or C'_3 depending on the direction of movement of the trolley and coefficients of dissipation accordingly b_3 or b'_3. The load of mass m is connected to the center of mass of the trolley by means of a flexible suspension and performs pendulum spatial oscillations in the planes of derriking and slewing of the crane.

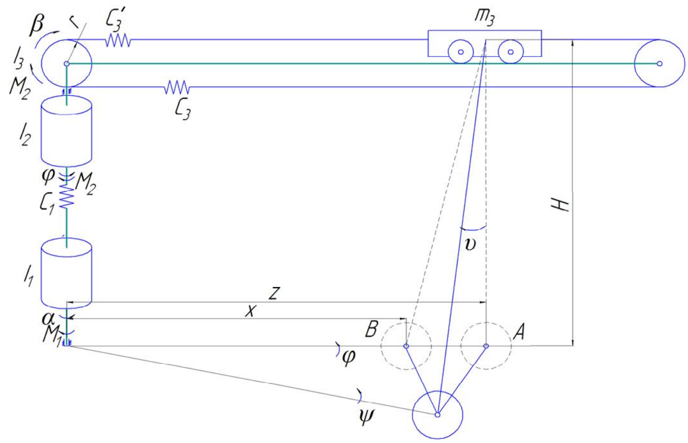

Fig. 1. The dynamic model of the derriking and the slew of the crane

Therefore, the presented dynamic model of the jib system of the tower crane has six degrees of freedom. The linear coordinates of the centers of mass of the trolley *z* and the load *x*, as well as the angular coordinate of the slewing of the reduced mass of the drive of the derriking *β* have been chosen as the generalized coordinates in the plane of the derriking. In the plane of slewing of the crane the angular coordinates of the reduced mass of the drive of the derriking *α*, the slewing part of the crane *φ* and slewing of the load *ψ* have been chosen as the generalized coordinates. The angular deviation of the flexible suspension of the load from the vertical is determined by the angular coordinate ϑ.

The system is under the driving torques of the drive mechanisms of slewing M_1 and derriking M_3, as well as the torque of resistance M_2 in the slewing part of the crane and the force of resistance of the trolley movement *W*. In addition, the dissipative forces act in the elastic elements which connect the drive slewing mechanism with the slewing part of the crane and the drive of derriking mechanism with the trolley. We assume that the length of the flexible suspension *H* is constant *H=const*.

In the plane of derricking mechanism, the centers of mass of the load and the trolley are deflected by the distance *AB*, and when the crane slews, it moves in the horizontal plane by the distance *BC*. Then the angular coordinate of the deviation of the flexible suspension of the load from the vertical was found:

$$\vartheta = \frac{1}{H}\sqrt{(z-x)^2 + x^2\cdot(\varphi-\psi)^2 + 2\cdot(z-x)\cdot x\cdot(\varphi-\psi)\cdot\sin(\frac{\psi-\varphi}{2})}\ . \qquad (1)$$

For a dynamic model of the over lapping motion of the derriking and slewing of the tower crane (Fig. 1) by using Lagrange's equations we obtain a system of differential equations of motion of derriking and slewing of the tower crane, which may be presented in the following form:

$$\begin{cases} I_1\cdot\ddot{\alpha}=M_1\cdot u_1\cdot\eta_1-C_1\cdot(\alpha-\varphi)-b_1\cdot(\dot{\alpha}-\dot{\varphi}); \\ I_3\cdot\ddot{\beta}=M_3\cdot u_3\cdot\eta_3-C_3\cdot r\cdot(\beta\cdot r-z)-b_3\cdot r\cdot\left(\dot{\beta}\cdot r-\dot{z}\right); \\ \left(I_2+m_3\cdot z^2\right)\cdot\ddot{\varphi}+2\cdot m_3\cdot z\cdot\dot{z}\cdot\dot{\varphi}=-M_2-\dfrac{m\cdot g}{H}\cdot x\cdot z\cdot(\varphi-\psi)+C_1\cdot(\alpha-\varphi)+b_1\cdot(\dot{\alpha}-\dot{\varphi}); \\ m\cdot x^2\cdot\ddot{\psi}+2\cdot m\cdot x\cdot\dot{x}\cdot\dot{\psi}=\dfrac{m\cdot g}{H}\cdot x\cdot z\cdot(\varphi-\psi); \\ m\cdot\ddot{x}-m\cdot x\cdot\dot{\psi}^2=-\dfrac{m\cdot g}{H}\cdot\left\{x-z\cdot\left[1-(\varphi-\psi)^2/2\right]\right\}; \\ m_3\cdot\ddot{z}-m_3\cdot z\cdot\dot{\varphi}^2=-W-\dfrac{m\cdot g}{H}\cdot\left\{z-x\cdot\left[1-(\varphi-\psi)^2/2\right]\right\}-C_3\cdot(\beta\cdot r-z)-b_3\cdot\left(\dot{\beta}\cdot r-\dot{z}\right), \end{cases} \quad (2)$$

where C_1 – the coefficient of torsional stiffness of the drive elements of the slewing mechanism, which are reduced to the axis of the crane tower; C_3 – the coefficient of linear stiffness of the rope of the derriking from the point of attachment at the trolley to the point of application on the drum; g – acceleration of free fall; r – the radius of the drum; M_1, M_3 – the driving torques on the shafts of the drive motors of the slewing and derriking mechanisms respectively; u_1, u_3 – the gear ratios of drives of the slewing and derriking mechanisms respectively; η_1, η_2 – the efficiency coefficients of drives of the slewing and derriking mechanisms respectively.

The calculation of the mathematical model should be complete with the following initial conditions:

$$\begin{gathered} t=0,\ \ \alpha=0,\ \ \dot{\alpha}=0,\ \ \varphi=0,\ \ \dot{\varphi}=0,\ \ \psi=0, \dot{\psi}=0, \\ x=x_0;\ \dot{x}=0;\ \ z=x_0,\ \ \dot{z}=0,\ \ \beta=\frac{x_0}{R},\ \dot{\beta}=0. \end{gathered} \quad (3)$$

The movement of the mechanisms of hoisting machines is significantly influenced by the driving torques created by the drive mechanisms [9]. To determine them, we use the differential equations of asynchronous motors.

Differential equations of asynchronous electric motors of the slewing mechanisms are written in a fixed coordinate system. These equations are characterized by the presence of variable (periodic) coefficients due to the change in mutual inductance between the stator and rotor windings of each of the motors. They may be expressed by the following equations:

$$\begin{cases} \dfrac{di_{1\alpha}}{dt}=\dfrac{1}{\delta_1\cdot L_1}\cdot\left(u_{1\alpha}-i_{1\alpha}\cdot R_1+(L_{12}/L_2)\cdot e_{2\alpha}\right); \\ \dfrac{di_{1\beta}}{dt}=\dfrac{1}{\delta_1\cdot L_1}\cdot\left(u_{1\beta}-i_{1\beta}\cdot R_1+(L_{12}/L_2)\cdot e_{2\beta}\right); \\ \dfrac{di_{2\alpha}}{dt}=-\dfrac{1}{\delta_1\cdot L_2}\cdot\left(\left(u_{1\alpha}-i_{1\alpha}\cdot R_1\right)\cdot(L_{12}/L_1)+e_{2\alpha}\right); \\ \dfrac{di_{2\beta}}{dt}=-\dfrac{1}{\delta_1\cdot L_2}\cdot\left(\left(u_{1\beta}-i_{1\beta}\cdot R_1\right)\cdot(L_{12}/L_1)-e_{2\beta}\right); \\ M_1=\dfrac{3}{2}\cdot P_1\cdot L_{12}\cdot\left(i_{1\beta}\cdot i_{2\alpha}-i_{1\alpha}\cdot i_{2\beta}\right). \end{cases} \quad (4)$$

Where $i_{1\alpha}$, $i_{1\beta}$ and $i_{2\alpha}$, $i_{2\beta}$ – the projections of generalized vectors of stator and rotor currents on the fixed orthogonal coordinate axes α and β; L_1, L_2– the inductance of stator and rotor windings of the engine; L_{12} – the engine mutual induction; M_1 – the electromagnetic engine torque; P_1 – the number of pairs of motor poles; $u_{1\alpha}$, $u_{1\beta}$– projections of generalized stator voltage vectors on the coordinate axes α and β. Substitution of the inferior indexes 1 and 2 to the 3 and 4 (1 → 3, 2 → 4) in the above dependencies brings the asynchronous electric motor equations of derricking mechanism.

Combining the system of differential equations of the mechanical part of the slewing and derriking mechanisms of the crane (2) with systems of electromagnetic transients in asynchronous electric motors(4), we obtain a generalized system of differential equations of over lapping movement of slewing and derriking, which takes into account dynamic mechanical characteristics of the drive motors.

Dynamical analysis

For tower crane with parameters: m=5000 kg; m_3=300 kg; I_1=168346 kg·m^2; I_2=5.5·10^6 kg·m^2; I_3=30 kg·m^2; W=5500 N; M_2=20100 Nm; u_1=1429; u_2=17; η_1=0.8; η_2=0.71; H=10 m; r=0,15 m; C_1=3.28·106 Nm/rad; C_3=1,36·10^{5}N/m; b_1=1.25·10^{5}Nm·s/rad; b_3=2.5·10^4 Ns/m and characteristics of drive electric motors MTH 112-6:P=4.5 kW; $n_н$=910 rpm; I=0.0687 kg·m^2; n_0=1000 rpm; $M_{\max}$=120 Nm; $\omega_н$=95.2 rad/s; ω_0=104.7 rad/s; $M_н$=47.3 Nm; λ=2.54 under the initial conditions of motion (3) the systems of equations (2), (4) and (5) were solved. For that, numerous methods were used, which have become widely used to study the dynamics of hoisting cranes [10]. As a result of the calculations, the kinematic and power characteristics of the tower crane with the over lapping of derriking and slewing mechanisms movement are obtained. Energy characteristics have also been studied, as they have a significant impact on the efficient operation of modern machines [11].

Fig. 2 shows the graphical dependencies of the linear velocities of the centers of mass of the trolley (gray line) and the load (black line) in the plane of derriking. One may conclude, that there are fluctuations in speeds of both trolley and load. Moreover, the fluctuations of the trolley speed are insignificant (there is only a small peak of fluctuations at the beginning of the movement caused by high-frequency oscillations of the drive). The load has significant, as lightly-damped, oscillations. A similar pattern is observed during the jib and the load slewing (Fig. 3). The figure shows that the oscillations of the angular velocity of the jib (light curve) are insignificant and only in the initial phase of movement there are high-frequency oscillations caused by electromagnetic dynamic processes in the electric motor of the slewing mechanism. The load in the plane of slewing also has significant oscillations in angular velocity (dark curve in Fig. 3).

It should be noted, that the complex oscillating motion in the dynamical systems is observed not only for hoisting machines [12]. The driving torques of the shafts of the drive motors have an oscillatory nature caused by pendulum oscillations of the load on a flexible suspension. The study of these oscillations will allow to developing the means to their elimination, which, in turn, will increase the productivity of the hoisting machine.

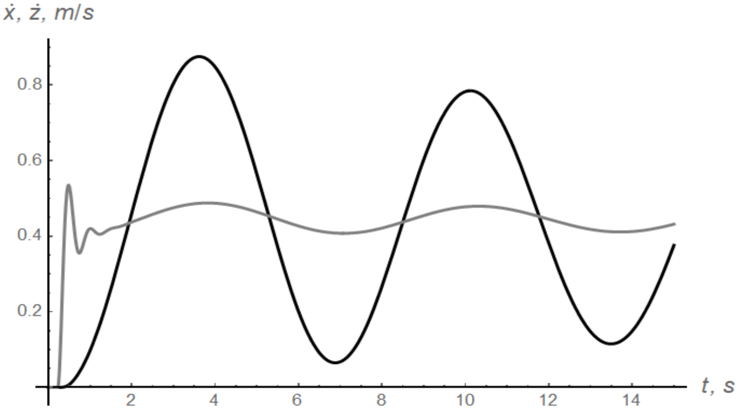

Fig. 1.The speed of the trolley and the load in the plane of derriking

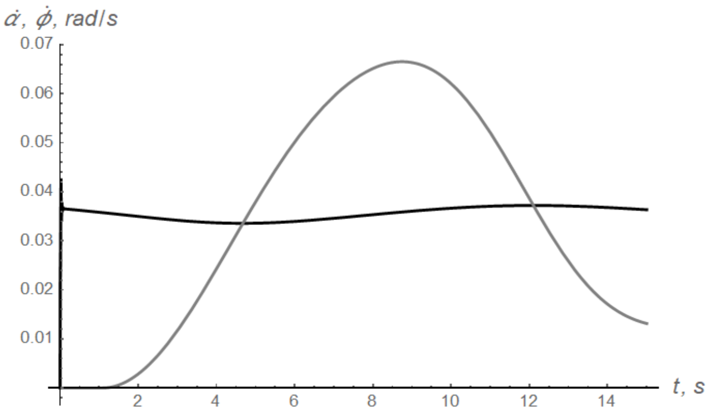

Fig. 3.The speed of the jib and the load in the plane of the crane slewing

The electric motor of the slewing mechanism at some moments goes into the generator mode, which is undesirable for its operation. The power consumption of the drive motors of the derriking and slewing mechanisms (Fig. 4) is also oscillating. In this figure, the black curve shows the power of the slewing mechanism, the black dashed – the derriking mechanism and the gray one – the total power consumed by the two mechanisms.

The table shows some amplitude, maximum and root-mean-square (RMS) values of kinematic, dynamic, and energy characteristics of the over lapping movement of the derriking and slewing mechanisms of the tower crane.

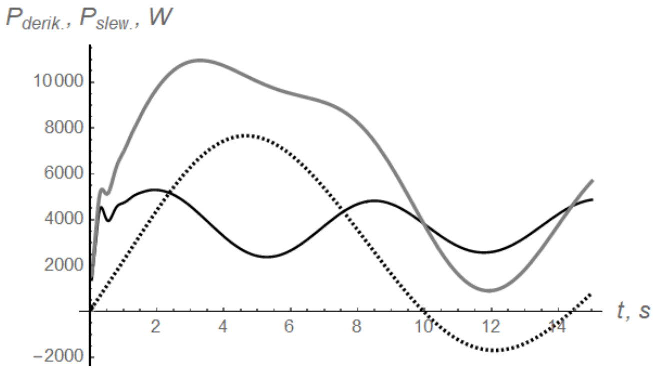

Fig. 4.The power of the drive of the derriking and slewing mechanisms of the crane

Table 1

Kinematic, power and energy characteristics of the mechanisms

Characteristic	Unit	Value
The amplitude of pendulum oscillations of the load along the boom	rad	0.42
The amplitude of the pendulum oscillations of the load perpendicular to the boom	rad	0.0294
The amplitude of the elastic-viscous oscillations of the rope	m	0.168
The amplitude of the elastic-viscous oscillations of the tower	rad	0.1176
The maximum torque of the drive of the derriking mechanism	Nm	150483
The maximum torque of the drive of the slewing mechanism	Nm	1136
The RMS value of the torque of the drive of the derriking mechanism	Nm	85861
The RMS value of the driving torque of the drive of the slewing mechanism	Nm	85859
The maximum elastic-viscous force in the rope of the derriking mechanism	N	7575
The maximum elastic-viscous torque in the tower	Nm	150483
The RMS value of the elastic-viscous force in the rope of the derriking mechanism	N	5615
The RMS value of the elastic-viscous torque in the tower	Nm	85859
The maximum power of the derriking mechanism	W	23889
The maximum power of the slewing mechanism	W	24808

Data in table 1 show the significant deviations of the maximum values of kinematic, dynamic, and energy characteristics from their RMS values, which indicates the dynamic and energy overloads of the derriking and slewing mechanisms during over lapping movement in the transient processes.

Prospects for further research in this direction are connected with the determination of the optimal modes of movement of individual mechanisms [2, 7, 9], as well as the rationalization of individual components and elements of the crane mechanisms [13]. In addition, developed model may be applied to investigation of frequency control of the drives of slewing and derriking mechanisms [14-16].

Conclusions. The article presents the results of studies of dynamic processes in the over lapping movement of the derriking and slewing mechanisms of the tower crane. For this purpose, an elastic-dissipative dynamic model of the over lapping motion of the derriking and slewing of the crane is developed. In such a model, the dynamic mechanical characteristics of drive electric motors are used with regards to the electrodynamic transients of the derriking and slewing mechanisms. Based on the constructed dynamic model, the differential equations of motion of the mechanical part of the mechanisms and the dynamic electromagnetic transients of the drive electric motors are composed. The solution of the obtained equations is carried out by numerical methods on the basis of the developed computer program. As a result of solving the obtained equations for the numerical parameters of the real tower crane, a dynamic and energy analysis of the over lapping work of the derriking and slewing mechanisms was performed, which revealed significant dynamic and energy overloads of crane drive mechanisms and crane metal structure. The presence of such overloads has a significant impact on the reliability and performance of the crane, as well as on the working conditions of the crane operator and maintenance personnel.

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АНАЛІЗ ЗМІНИ ВИЛЬОТУ ТА ПОВОРОТУ БАШТОВОГО КРАНА З УРАХУВАННЯМ ДИНАМІЧНИХ ХАРАКТЕРИСТИК ПРИВОДНИХ МЕХАНІЗМІВ

При роботі баштових кранів з метою підвищення продуктивності здійснюють суміщення роботи механізмів. В статті розглянуто суміщення операцій зміни вильоту та повороту крана. Для дослідження динамічних процесів при спільній роботі механізмів зміни вильоту та повороту баштового крана побудовано динамічну модель, в якій враховано основний рух механізмів зміни вильоту та повороту крана, а також коливальні процеси в передавальних механізмах приводів, а також коливання вантажу на гнучкому підвісі в двох площинах: в площині зміни вильоту та в площині повороту крана. В розробленій динамічній моделі враховані пружно-дисипативні характеристики передавальних механізмів, а також динамічні механічні характеристики приводних електродвигунів, які описують перехідні електромагнітні процеси. На основі розробленої динамічної моделі за допомогою рівнянь Лагранжа другого роду складено шість диференціальних рівнянь другого порядку, які описують механічну частину руху механізмів. Крім того, перехідні динамічні електромагнітні процеси кожного з двигунів описуються чотирма диференціальними рівняннями. В результаті чого отримано систему чотирнадцяти диференціальних рівнянь, які описують спільну динаміку механічної частини механізмів крана та електромагнітних процесів в електродвигунах механізмів зміни вильоту та повороту крана. Отримана система диференціальних рівнянь розв’язана чисельними методами за допомогою розроблених комп’ютерних програм. Для реальної конструкції баштового крана при спільній роботі механізмів зміни вильоту та повороту визначені кінематичні, силові та енергетичні характеристики, які дали можливість оцінити реальні навантаження в елементах конструкції та приводних механізмах. З отриманих результатів досліджень встановлено, що елементи конструкції та приводних механізмів мають значні динамічні та енергетичні перевантаження. Виявлені високочастотні коливання приводних механізмів на початку руху та низькочастотні коливання елементів конструкції та вантажу. Це значно знижує надійність роботи баштових кранів,збільшує їхні енергетичні витрати та впливає на роботу кранівника та обслуговуючого персоналу.

Ключові слова: баштовий кран, механізми повороту та зміни вильоту, динамічна і математична модель, динамічні та енергетичні навантаження.

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ANALYSIS OF DERRIKINGAND SLEWING OF THE TOWER CRANE WITH CONSIDERATION TO DRIVING MECHANISMS CHARACTERISTICS

The variational problem of the movement mode selection for the load outreach change mechanism during a steady-state tower crane slewing was formulated and solved in the paper, that ensures the minimization of the drive motor power. The variational problem is nonlinear, and so we used the modified PSO-Rot-Ring particle swarm met heuristic method for its solution. Low- and high-frequency oscillations of the outreach change mechanism elements during the start-up were detected in the optimization process. These oscillations are eliminated in the section of steady-state movement due to the selection of the motion boundary conditions.

In order to increase the productivity during a tower crane exploitation the overlapping of the mechanisms operation is used. The article considers the overlapping of operations of the derriking and slewing mechanisms of the crane. To study the dynamic processes in the joint work of the derriking and slewing of the tower crane, a dynamic model was built. It takes into account the main movement of the derriking and slewing of the crane, as well as oscillation processes in the drive mechanisms and load oscillation on flexible suspension in two planes: in the plane of the derrikingand in the plane of slewing of the crane. The developed dynamic model takes into account the elastic-dissipative characteristics of the transfer mechanisms, as well as the dynamic mechanical characteristics of the drive motors, which describe the transient electromagnetic processes. On the basis of the developed dynamic model with the help of Lagrange's equations six differential equations of the second order which describe a mechanical part of the mechanisms movement were found. In addition, the transient dynamic electromagnetic processes of each of the engines are described by four differential equations. As a result, a system of fourteen differential equations was obtained. It describes the common dynamics of the mechanical part of the crane mechanisms and electromagnetic processes in the electric engines of the derriking and slewing mechanisms. The obtained system of differential equations was solved by numerical methods with the help of developed computer program. For the actual construction of the tower crane in the overlapping work of the derriking and slewing mechanisms, kinematic, power, and energy characteristics were determined, which made it possible to estimate the actual loads in the elements of the construction and drive mechanisms. From the obtained results it was established that the elements of the construction and drive mechanisms are under significant dynamic and energy overloads.High-frequency oscillations of drive mechanisms at the beginning of the movement and low-frequency oscillations of elements of the construction and load were revealed. This significantly reduces the reliability of tower cranes, increases their energy losses, and affects the work of the crane operator and maintenance staff.

Keywords: tower crane, derriking, slewing, dynamic and mathematic models, dynamic and energy loads.

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На основі побудованої динамічної моделі складені диференціальні рівняння руху механічної частини механізмів та динамічних електромагнітних перехідних процесів приводних електродвигунів. В результаті розв'зку отриманих рівнянь для числових параметрів реального баштового крана проведено динамічний та енергетичний аналіз спільної роботи механізмів зміни вильоту та повороту крана, з якого виявлені значні динамічні та енергетичні перевантаження приводних механізмів та конструкцій крана.

Табл. 1. Іл. 4. Бібліогр. 16.

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Based on the constructed dynamic model, the differential equations of motion of the mechanical part of the mechanisms and the dynamic electromagnetic transients of the drive electric motors are composed. As a result of solving the obtained equations for the numerical parameters of the real tower crane, a dynamic and energy analysis of the over lapping work of the derriking and slewing mechanisms was performed, which revealed significant dynamic and energy overloads of crane drive mechanisms and crane metal structure.
Table 1. Fig. 7. Ref. 16.

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