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DRIVE POWER MINIMIZATION OF OUTREACH CHANGE MECHANISM OF TOWER CRANE DURING STEADY-STATE SLEWING MODE

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

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

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

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

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

M.M. Balaka[3],
Candidate of Science (Engineering), Associate 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 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 metaheuristic 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.

Keywords: tower crane, outreach change mechanism, power, nonlinear variational problem, optimization criterion.

Introduction. The movement of individual mechanisms is combined in order to increase the productivity of tower cranes during their operation. To combine the operation of outreach change and slewing mechanisms of tower cranes is quite common. At that, increased dynamic loads take place in elements of the drive mechanisms and crane construction, which affect the power inputs and crane service reliability. The maximum power inputs are observed during transient processes (start-up, braking) with such operation of crane mechanisms. This is caused by the fact that the dynamic component of drives power is dominant in comparison with the static one during an operation of the load outreach change and crane slewing mechanisms. It is possible to considerably reduce the dynamic power component of the tower crane drive mechanisms due to the movement modes selection during transient processes.

Also the reduction of power inputs leads to the installation of the lower power electric motors and increase their durability, as the increased power inputs go to overheat of the windings insulation.

Thus the reduction of power inputs and drive power minimization during the operation of the outreach change mechanism and steady-state crane slewing is actual problem, which directed to the thrifty use of tower cranes resources in the conditions of modern production.

Analysis of publications. The significant number of research papers [1–10] have been devoted to the study of dynamic loads and load oscillations on the flexible suspension of lifting cranes and, in particular, tower cranes.

The dynamics of the outreach change and load lifting for various types of cranes were studied and the causes of load oscillations on the flexible suspension were identified in [4–8]. The combined movement of the jib outreach change and crane slewing mechanisms were considered in [9, 10]. The drive electric motor of the outreach change mechanism was controlled here during the crane slewing with a hanging load on the flexible suspension in order to reduce its oscillations.

The joint movement dynamics of the outreach change and crane slewing mechanisms is considered in the research paper [11], and the joint movement mode of the outreach change and load lifting mechanisms was determined in [12, 13]. The kinematic, force and energy characteristics of the outreach change, load lifting and crane slewing mechanisms were determined on the basis of these studies. At that, significant overload of the mechanisms and spatial oscillations of the load on the flexible suspension were established.

The number of optimization problems were solved during the operation of individual crane mechanisms to reduce the load oscillations on the flexible suspension [14–18]. Thus the optimization problem of load oscillations reduction on the flexible suspension during the operation of the crane slewing mechanism was solved by use the complex dynamic integral criterion in [14].

The transition process optimization of the outreach change mechanism start-up with the articulated jib system for load horizontal movement and crane slewing by moment control of the drive is proposed in [15] to ensure minimum loads and to reduce load oscillations on the flexible suspension.

The selection of motion estimation criteria for lifting machines is an important problem of the movement modes optimization for lifting cranes [16, 17]. Integral dynamic criteria are common, which are presented in the form of integral functionals with subintegral functions in the form of root-mean-square values for force and energy characteristics. The solution of the optimization problem according to this criterion for the simultaneous movement of outreach change and crane slewing mechanisms is given in [18]. The defined modes of movement cause additional loads on the elements of the drive mechanisms and the crane as whole. Therefore, the criteria selection of movement modes optimization with the combined operation of mechanisms is the actual problem and requires the deeper study.

Purpose of the paper. To minimize the drive power of the outreach change mechanism for the tower crane with the steady-state slewing mode by the start-up mode optimization.

Research results. The dynamic model of the tower crane jib system was developed (Fig. 1) in order to minimize the drive power of the load outreach change mechanism during the steady-state slewing [19].

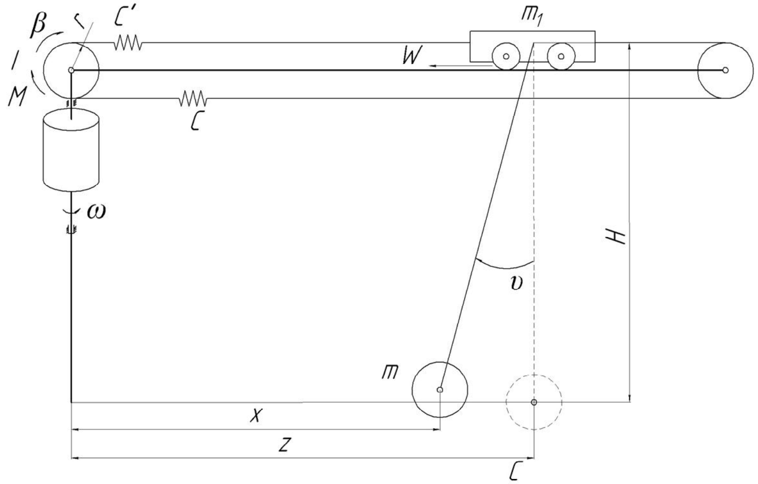

Fig. 1. Dynamic model of the outreach change mechanism during the steady-state crane slewing

The jib system of the tower crane is represented by a mechanical system with perfectly rigid links, except the traction rope for the trolley movement. It is presented by an elastic element and flexible suspension of the load, that carries out pendulum oscillations in the plane of the outreach change.

The inertial mass I and the driving moment M of the mechanism drive for the load outreach change are reduced to the drum rotation axis. The drive drum of the load outreach change mechanism is connected to the trolley by a mass m_1 with an elastic rope from the stiffness coefficient C or C' depending on the trolley moving direction. When the trolley movement, there is a resistance force W, which is always directed opposite to its movement. The flexible suspension length is the constant value and equal to H, and the slewing mechanism rotates with the constant angular velocity ω.

The presented dynamic model of the tower crane jib system has three degrees of freedom. The linear coordinates of the mass center for the trolley z and the load x, and the angular coordinate of the drive drum slewing of the outreach change mechanism β, are taken as generalized coordinates.

The differential equations of mechanisms joint motion of the outreach change and steady-state crane slewing were formulated for such a dynamic model with use Lagrange equations of the second kind:

$$\begin{cases} I\ddot{\beta} = M - Cr(\beta \cdot r - z); \\ m_1\ddot{z} - m_1\omega^2 z = C(\beta \cdot r - z) - (mg/H)(z-x) - W; \\ m\ddot{x} - m\omega^2 x = (mg/H)(z-x), \end{cases} \tag{1}$$

where r – drum radius; g – free fall acceleration; m – load weight.

From the last equation of the system (1) we will express the coordinate z in terms of x and take the first and second time derivatives from it, as a result of which we will have:

$$z = \left(1-(H/g)\omega^2\right)x + (H/g)\ddot{x};$$

$$\dot{z} = \left(1-(H/g)\omega^2\right)\dot{x} + (H/g)\dddot{x};\ \ddot{z} = \left(1-(H/g)\omega^2\right)\ddot{x} + (H/g)\overset{IV}{x}. \tag{2}$$

We find the expression of the drive drum angular coordinate of the load outreach change mechanism from the second equation of the system (1), taking into account the dependencies (2)

$$\beta = \frac{1}{Cr}\left\{\begin{aligned} &\left[\left(C-m_1\omega^2\right)\left(1-(H/g)\omega^2\right)-m\omega^2\right]x + \\ &+\left[\left(C-2m_1\omega^2\right)(H/g)+m_1+m\right]\ddot{x}+m_1(H/g)\overset{IV}{x}+W \end{aligned}\right\}. \tag{3}$$

Taking the first and second time derivatives of expression (3), we find the angular velocity and acceleration of the drive drum of the load outreach change mechanism:

$$\dot{\beta} = \frac{1}{Cr}\left\{\begin{aligned} &\left[\left(C-m_1\omega^2\right)\left(1-(H/g)\omega^2\right)-m\omega^2\right]\dot{x} + \\ &+\left[\left(C-2m_1\omega^2\right)(H/g)+m_1+m\right]\dddot{x}+m_1(H/g)\overset{V}{x} \end{aligned}\right\}; \tag{4}$$

$$\ddot{\beta} = \frac{1}{Cr}\left\{\begin{aligned} &\left[\left(C-m_1\omega^2\right)\left(1-(H/g)\omega^2\right)-m\omega^2\right]\ddot{x} + \\ &+\left[\left(C-2m_1\omega^2\right)(H/g)+m_1+m\right]\overset{IV}{x}+m_1(H/g)\overset{VI}{x} \end{aligned}\right\}. \tag{5}$$

We find the expression of the moment drive of the load outreach change mechanism from the first equation of the system (1), taking into account expressions (2), (3) and (5)

$$M = a_0 + a_1 x + a_2\ddot{x} + a_3\overset{IV}{x} + a_4\overset{VI}{x}, \tag{6}$$

$$a_0 = Wr;\ \ a_1 = -\left[m + m_1\left(1-(H/g)\omega^2\right)\right]\omega^2 r;$$

$$a_2 = \frac{I}{Cr}\left[\left(C-m_1\omega^2\right)\left(1-(H/g)\omega^2\right)-m\omega^2\right]+\left[m+m_1\left(1-2(H/g)\omega^2\right)\right]r;$$

$$a_3 = \frac{I}{Cr}\left[\left(C-m_1\omega^2\right)(H/g)+m_1\left(1-(H/g)\omega^2\right)+m\right]+m_1(H/g)r;$$

$$a_4 = m_1 IH/Crg;\ \ a_{0,1,2,3,4} = \text{const}. \tag{7}$$

We will select the root-mean-square value of the drive mechanism power as the optimization criterion of the start-up mode of the load outreach change mechanism during the steady-state crane slewing. The problem of the drive power minimization of the outreach change mechanism during the steady-state crane slewing is set according to purpose of the study, therefore its root-mean-square value during the start-up time selected as the optimization criterion.

At the same time, a variational problem is set, which is aimed at the root-mean-square value minimization of the drive mechanism power

$$P_{rms} = \left[\frac{1}{t_1} \int_0^{t_1} P^2 dt \right]^{1/2} \to \min , \tag{8}$$

where t – time; t_1 – start-up process duration of the load outreach change mechanism; P – drive power of the load outreach change mechanism.

The solution of the given variational problem (8) $x = x(t)$, $0 \le t \le t_1$, must satisfy the next boundary conditions:

$$t = 0: \ x = x_0; \ \dot{x} = 0; \ \ddot{x} = x_0 \omega^2; \ \dddot{x} = 0; \ \overset{IV}{x} = x_0 \omega^4; \ \overset{V}{x} = 0 ;$$

$$t = t_1: \ x = x_0 + \frac{V t_1}{2}; \ \dot{x} = V; \ \ddot{x} = \left(x_0 + (V t_1)/2 \right) \omega^2; \ \dddot{x} = V \omega^2;$$

$$\overset{IV}{x} = \left(x_0 + (V t_1)/2 \right) \omega^4; \ \overset{V}{x} = V \omega^4, \tag{9}$$

where x_0 – initial position of the mass center coordinates of the trolley and load; V – steady-state motion velocity of the trolley and load.

The drive power of the load outreach change mechanism during the steady-state crane slewing is determined by the next dependence

$$P = M \dot{\beta} , \tag{10}$$

where $\dot{\beta}$ is determined by expression (4), and M – by dependence (6) taking into account expressions (7).

The variational problem (8) and (9) is nonlinear based on the dependence (10) and the expressions of its component elements (4), (6) and (7). So it is necessary to use approximate numerical methods for its solution. It is desirable to have the first approximate initial solution of the nonlinear variational problem in these methods, which will allow to significantly reduce calculated resources in the future.

We will use the analytical solution of the next linear variational problem as such an initial approximate solution of the nonlinear variational problem, where the root-mean-square value of the drive moment is select as the movement mode optimization criterion of the outreach change mechanism during the steady-state crane slewing

$$M_{rms} = \left[\frac{1}{t_1} \int_0^{t_1} M^2 dt \right]^{1/2} \to \min . \tag{11}$$

The solution of the given linear optimization problem $x = x(t)$, $0 \le t \le t_1$, must satisfy the boundary conditions of the nonlinear problem (9). The variational problem (11) and (9) with expressions (6) and (7) is linear, the solution of which is presented in the next form

$$x(t) = y(t) - \frac{a_0}{a_1} = (C_1 + C_2 t)\cos(\alpha_1 t) + (C_3 + C_4 t)\sin(\alpha_1 t) + (C_5 + C_6 t)\cos(\alpha_2 t) +$$

$$+(C_7 + C_8 t)\sin(\alpha_2 t) + (C_9 + C_{10} t)e^{\alpha_3 t} + (C_{11} + C_{12} t)e^{-\alpha_3 t} - \frac{a_0}{a_1}, \ 0 \le t \le t_1, \quad (12)$$

where C_1, C_2, …, C_{12} – constants determined from the motion boundary conditions (9); $\pm a_1 i$, $\pm a_2 i$, $\pm a_3 i$ – roots of the characteristic equation; $i = \sqrt{-1}$ – imaginary unit.

For the tower crane jib system with parameters: $m = 5000$ kg, $m_1 = 300$ kg, $I = 30$ kg·m^2, $H = 10$ m, $g = 9{,}81$ kg·m/s^2, $\omega = 0{,}075$ rad/s, $r = 0{,}15$ m, $c = 1{,}65 \cdot 10^5$ N/m, $V = 0{,}85$ m/s, $x_0 = 7$ m, $t_1 = 5$ s, $W = 5500$ N, the roots of characteristic equation: $\pm a_1 i = \pm 26{,}21 i$, $\pm a_2 i = \pm 1{,}918 i$, $\pm a_3 i = \pm 0{,}06705 i$ and constants: $C_1 \approx -1{,}3540 \cdot 10^{-6}$, $C_2 \approx 1{,}0536 \cdot 10^{-7}$, $C_3 \approx 2{,}09208 \cdot 10^{-8}$, $C_4 \approx 1{,}8714 \cdot 10^{-7}$, $C_5 \approx 0{,}036064$, $C_6 \approx -0{,}014894$, $C_7 \approx 0{,}020835$, $C_8 \approx -0{,}003177$, $C_9 \approx -78{,}196$, $C_{10} \approx 2{,}9682$, $C_{11} = -99{,}386$, $C_{12} \approx -4{,}41522$ were found.

We will obtain the final solution of the linear variation problem (11) and (9) by substituting the found constants $C_{1,\ldots,12}$ in (12). This will ensure the root-mean-square value minimization of the drive moment of the load outreach change mechanism during the steady-state crane slewing.

We will use the analytical solution of the linear variational problem (11) and (9), as the initial approximate solution of the nonlinear variational problem (8) and (9). In order to obtain approximate solution of the variational problem (8) and (9), we add the next polynomial to the known solution (12):

$$x_{dod}(t) = t^6 (t_1 - t)^6 \sum_{i=0}^{I} b_i t^i \ , \quad (13)$$

where I – polynomial order, whose coefficients must be found; b_i – i-th polynomial coefficient. The first and second polynomial factors (13) ensure compliance with the boundary conditions (9), since the polynomial derivatives (13) from first to fifth inclusive at the times $t = 0$ and $t = t_1$ are equal to zero.

Thus the polynomial order used to find the approximate solution of the nonlinear variational problem is equal to $6 + I$.

So the function $x_{nabl}(t)$, which will be used to find the approximate solution of the variational problem, is presented by the superposition of the motion laws (12) and (13):

$$x_{nabl}(t) = x(t) + x_{dod}(t) . \quad (14)$$

The selection of the coefficients number I , that must be found, must satisfy two contradictory requirements: first, it must ensure a sufficiently accurate approximation of the function (14) to the sought problem extremal (8) and (9) – and this requires setting a large number I ; secondly, the number I should be as small, as this reduces the number and scope of necessary calculations. It is possible to meet these demands only on a compromise basis. We will take $I = 5$ in this study. This makes it possible to write functional (8) as a function of unknown coefficients

$$Cr_6 = P_{rms}(b_0, b_1, b_2, b_3, b_4, b_5) . \quad (15)$$

Note that this function is nonlinear and the determination of its minimum is the rather difficult problem. So the PSO-Rot-Ring metaheuristic method was used in this study [20]. This method is the modification of the PSO particle swarm method with a ring topology of connections between particles, and this topology is dynamic: the «ring» of particle's connections returns with each iteration, that causes change in the particles with which this particle exchanges information. This modification of the PSO method makes it possible to better explore the function and find promising areas, in which the minima can be found (in the ideal case – the function global minimum).

The global best value corresponded to parameters b_0 , b_1 , b_2 , b_3 , b_4 , b_5 , which are equal to zero when use the PSO-Rot-Ring method during the first iteration. This value was changed in the future according with the found minima. 1000 iterations were set, and the swarm size is 30.

The PSO-Rot-Ring method application made it possible to obtain the next coefficient values: $b_0 = 8,429 \cdot 10^{-8}$, $b_1 = -1,008 \cdot 10^{-6}$, $b_2 = 4,600 \cdot 10^{-7}$, $b_3 = -6,146 \cdot 10^{-8}$, $b_4 = -9,493 \cdot 10^{-9}$, $b_5 = 2,159 \cdot 10^{-9}$.

In addition, the criterion was minimized, in which function (13) contained only two unknown coefficients

$$Cr_2 = P_{rms}\left(b_0, b_1\right) . \quad (16)$$

Minimization of both criteria variants (15) and (16) led to almost identical (in the sense of the criterion value (8)) results (Fig. 2).

The PSO-Rot-Ring method converges somewhat slower for the criterion variant Cr_6 , as can be seen from Fig. 2, which is caused by a larger number of searched arguments. However, the difference between the found values of Cr_2 and Cr_6 differs only by tenths fraction. So we will consider only the option Cr_2 in the future, the arguments that ensure its minimum are as next: $b_0 = -6,458 \cdot 10^{-7}$, $b_1 = -1,201 \cdot 10^{-8}$.

The obtained result will be presented in the graphical dependencies form (Fig. 3–8) of the kinematic, force and energy motion characteristics of the load outreach change mechanism during the steady-state movement mode of the crane slewing mechanism. Diagrams in black correspond to the case of the initial iteration, i.e. function (12), and in gray – the found approximate solution to problem (8) and (9).

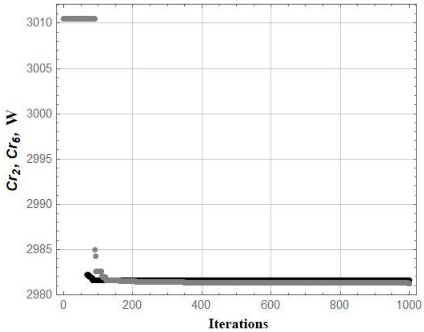

Fig. 2. Diagram of the criterion value (8) reduction when use PSO-Rot-Ring method (gray points – Cr_6 variant, black – Cr_2 variant)

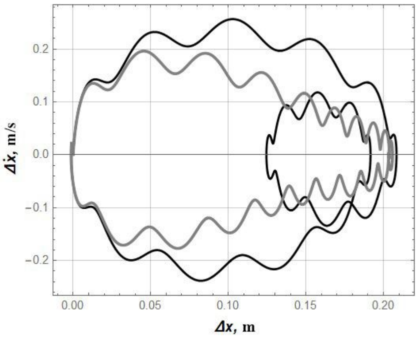

Fig. 3. Phase portrait of the load pendulum oscillations on the flexible suspension

The next notations are accepted in Fig. 3: $\Delta x = x - z$; $\Delta\dot{x} = \dot{x} - \dot{z}$. Based on the phase portraits results of the load oscillations relative to the trolley (Fig. 3), constructed as the solution result of the given variation problems, we can conclude that smaller deviations of the oscillations velocity are observed (gray curve) in comparison with the root-mean-square value criterion of the driving moment (black curve) when the problem solution according to the root-mean-square value criterion of the power. At the same time, the maximum displacement deviations are almost the same.

It can be seen from the graphic dependences of the trolley velocity (Fig. 4) that high-frequency oscillations are observed in both movement modes, which are caused by the change nature of the driving moment of the outreach change mechanism drive. At the same time, there are also low-frequency oscillations caused by the load oscillations on the flexible suspension when the movement mode optimization according to the root-mean-square value of the driving moment, which is practically not observed in the optimal mode according to the root-mean-square power criterion.

High-frequency oscillations of the trolley traction force are observed in both optimal movement modes (Fig. 5), and in the mode determined by the root-mean-square value criterion of the driving moment, there is also the low-frequency oscillations component. However, the maximum value of the trolley traction force is slightly lower with this movement mode.

The driving moment of the outreach change mechanism has the similar changes nature (Fig. 6). Both low-frequency and high-frequency oscillations are also observed in both optimal movement modes, but the maximum moment value is slightly smaller, which are determined by the root-mean-square value criterion of the drive power for the outreach change mechanism.

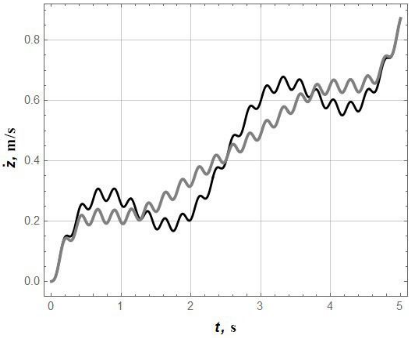

Fig. 4. Trolley velocity diagram

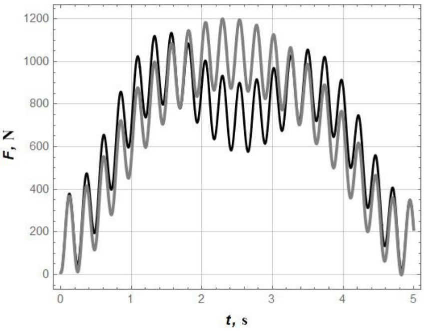

Fig. 5. Diagram of the force change in the trolley traction element

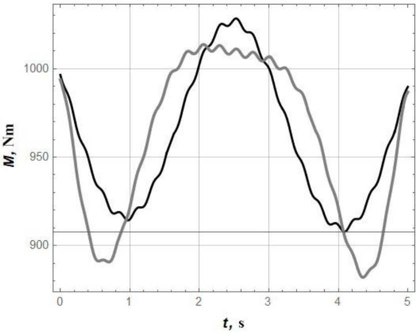

Fig. 6. Diagram of the driving moment change of the outreach change mechanism

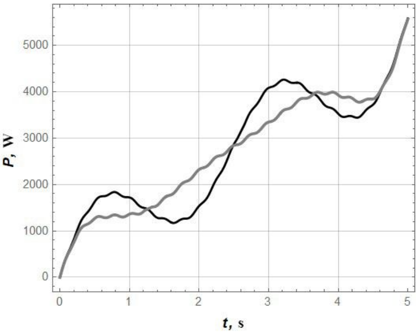

Fig. 7. Diagram of the drive power change of the outreach change mechanism

The power change diagram (Fig. 7), determined by both criteria, show the presence of high-frequency oscillations. However, low-frequency oscillations are also observed in the movement mode, determined by the root-mean-square value criterion of the drive moment.

In addition, the main parameters of the movement characteristics were calculated, which correspond to the found approximate solution of the problem (8) and (9) according to the root-mean-square value criterion of the power and the known solution (12) according to the root-mean-square value criterion of the driving moment (Table 1).

From the maximum and root-mean-square values of the physical parameters (Table 1) of the outreach change mechanism during the steady-state crane slewing, which were determined by the root-mean-square values criteria of the driving moment and drive power, it is shown that the obtained movement modes are quite close in their characteristics. At the same time, each of these movement modes has its own characteristics, which were reflected in the graphic dependencies analysis of kinematic, force and energy characteristics of the outreach change mechanism during the steady-state crane slewing.

Table 1

Parameters of movement characteristics

Physical content of parameter	Maximum values		Root-mean-square values	
	function (12)	function (14)	function (12)	function (14)
Driving moment of outreach change mechanism drive, N·m	1028,4	1013,7	960,6	961,0
Force in traction element, N	1134,6	1200,7	727,7	750,0
Rope deviation with load from vertical, m	0,2084	0,2058	0,1437	0,1457

Conclusions. The dynamic optimization of the load outreach change mechanism during the steady-state crane slewing is given in presented research paper. The variational problem is formulate to carry out such optimization, where the root-mean-square value of the mechanism drive power is select as the optimization criterion of the load outreach change mode. The variational problem was considered to solve this problem, where the root-mean-square value of the driving moment for the mechanism drive was used as the optimization criterion. The first of these problems is nonlinear, and the second is linear, which has an analytical solution, therefore it was used as the solution component of the nonlinear variational problem. The nonlinear variational problem uses an additional component in the form of a polynomial with unknown coefficients. The PSO-Rot-Ring metaheuristic method was used to determine these coefficients, which is a modification of the PSO particle swarm method with the ring topology. The minimum condition of the root-mean-square value for the drive power was determined and the optimal mode of the load outreach change was found. The optimal movement modes of other links of the outreach change mechanism during the steady-state crane slewing were found due to the optimal movement mode of the load. Low- and high-frequency oscillations of kinematic, force and energy characteristics are observed in the optimal start-up mode of the load outreach change mechanism, which are caused by the load rocking on the flexible suspension and the nature of the driving moment change. These oscillations are eliminated by to select the appropriate motion boundary conditions in the steady-state movement section of the outreach change mechanism and crane slewing. The comparison of optimal movement mode obtained, when to solve the nonlinear variational problem with optimal mode, found from the solution of the linear problem was carried out. The solved variational problems give close results for the example of the outreach change mechanism. So nonlinear problems can be replaced by linear ones in some cases.

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

МІНІМІЗАЦІЯ ПОТУЖНОСТІ ПРИВОДУ МЕХАНІЗМУ ЗМІНИ ВИЛЬОТУ БАШТОВОГО КРАНА ЗА УСТАЛЕНОГО РЕЖИМУ ПОВОРОТУ

У статті поставлена та розв’язана варіаційна задача вибору режиму руху механізму зміни вильоту вантажу за усталеного повороту баштового крана, що забезпечує мінімізацію потужності приводного двигуна. Для проведення досліджень використана динамічна модель механізму зміни вильоту, представлена механічною системою з трьома ступенями вільності. Поставлена варіаційна задача є нелінійною, тому для її розв’язку використано модифікований метаевристичний метод рою часточок PSO-Rot-Ring. Для збереження розрахункових ресурсів в якості початкового наближення розв’язку нелінійної варіаційної задачі використане аналітичне розв’язування іншої варіаційної задачі для тієї ж моделі кранового механізму і близького за фізичним змістом оптимізаційного критерію. При розв’язуванні нелінійної варіаційної задачі визначено режим пуску приводу механізму зміни вильоту вантажу, який забезпечує мінімізацію середньоквадратичного значення потужності приводного двигуна. В процесі оптимізації виявлені низько- та високочастотні коливання елементів механізму зміни вильоту під час пуску. Перші коливання викликані розгойдуванням вантажу на гнучкому підвісі під час пуску, а другі – характером зміни рушійного моменту та потужності приводу. Ці коливання усуваються на ділянці усталеного руху за рахунок вибору крайових умов руху, що враховуються в процесі розв’язування варіаційної задачі.

Ключові слова: баштовий кран, механізм зміни вильоту, потужність, нелінійна варіаційна задача, критерій оптимізації.

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

DRIVE POWER MINIMIZATION OF OUTREACH CHANGE MECHANISM OF TOWER CRANE DURING STEADY-STATE SLEWING MODE

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. We used a dynamic model of the outreach change mechanism for research work, which presented the mechanical system with three degrees of freedom. The formulated variational problem is nonlinear, and so we used the modified PSO-Rot-Ring particle swarm metaheuristic method for its solution. The analytical solution of another variational problem for the same model of the crane mechanism and an optimization criterion close in physical content was used as the solution initial approximation of the nonlinear variational problem to save calculated resources. The starting mode of the mechanism drive for the load outreach change was determined during the solution of the nonlinear variational problem, which ensures the root-mean-square value minimization of the drive motor power. Low- and high-frequency oscillations of the outreach change mechanism elements during the start-up were detected in the optimization process. The first oscillations are caused by the load rocking on the flexible suspension during the start-up, and the second – by the nature of the change in the driving moment and drive power. These oscillations are eliminated in the section of steady-state movement due to the selection of the motion boundary conditions, which are taken into account in the solution process of the variational problem.

Keywords: tower crane, outreach change mechanism, power, nonlinear variational problem, optimization criterion.

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При розв'язуванні нелінійної варіаційної задачі визначено режим пуску приводу механізму зміни вильоту вантажу за усталеного повороту баштового крана, що забезпечує мінімізацію потужності приводного двигуна. Коливання елементів механізму зміни вильоту під час пуску усуваються за рахунок вибору крайових умов руху.
Табл. 1. Іл. 7. Бібліогр. 20.

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The starting mode of the mechanism drive for the load outreach change during the steady-state tower crane slewing was determined under the solution of the nonlinear variational problem, that ensures the minimization of the drive motor power. The oscillations of the outreach change mechanism elements during the start-up are eliminated due to the selection of motion boundary conditions.
Table 1. Fig. 7. Ref. 20.

Автор (вчена ступень, вчене звання, посада): доктор технічних наук, професор, завідувач кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України ЛОВЕЙКІН Вячеслав Сергійович
Адреса робоча: 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
Робочий тел.: +38(044) 527-87-34
E-mail: lovvs@ukr.net
ORCID ID: https://orcid.org/0000-0003-4259-3900

Автор (вчена ступень, вчене звання, посада): доктор технічних наук, професор, професор кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України РОМАСЕВИЧ Юрій Олександрович
Адреса робоча: 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
Робочий 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 кафедри математичної фізики Київського національного університету імені Тараса Шевченка ЛОВЕЙКІН Андрій Вячеславович
Адреса робоча: 03022, Україна, м. Київ, проспект академіка Глушкова, 4е, корпус механіко-математичного факультету, Київський національний університет імені Тараса Шевченка, кафедра математичної фізики, ЛОВЕЙКІНУ Андрію Вячеславовичу
Мобільний тел.: +38(097) 350-91-23
E-mail: anlov74@gmail.com
ORCID ID: https://orcid.org/0000-0002-7988-8350

Автор (вчена ступень, вчене звання, посада): кандидат технічних наук, старший викладач кафедри конструювання машин і обладнання Національного університету біоресурсів і природокористування України ЛЯШКО Анастасія Петрівна
Адреса робоча: 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
Робочий тел.: +38(044) 527-87-34
E-mail: laskoanastasia1989@gmail.com
ORCID ID: https://orcid.org/0000-0002-3774-3348

Автор (вчена ступень, вчене звання, посада): доктор технічних наук, професор, завідувач кафедри професійної освіти КНУБА ПОЧКА Костянтин Іванович
Адреса робоча: 03037, Україна, м. Київ, Повітрофлотський проспект 31, Київський національний університет будівництва і архітектури, кафедра професійної освіти, ПОЧЦІ Костянтину Івановичу
Мобільний тел.: +38(097) 212-86-29
E-mail: shanovniy@ukr.net
ORCID ID: https://orcid.org/0000-0002-0355-002X

Автор (вчена ступень, вчене звання, посада): кандидат технічних наук, доцент, доцент кафедри будівельних машин КНУБА БАЛАКА Максим Миколайович
Адреса робоча: 03037, Україна, м. Київ, Повітрофлотський проспект 31, Київський національний університет будівництва і архітектури, кафедра будівельних машин, БАЛАЦІ Максиму Миколайовичу
Робочий тел.: +38(044) 241-55-52
E-mail: balaka.mm@knuba.edu.ua
ORCID ID: https://orcid.org/0000-0003-4142-9703