Machinery & Energetics
Vol. 13, No. 3. 2022

Journal homepage:
https://technicalscience.com.ua/en

UDC 621.87
DOI: 10.31548/machenergy.13(3).2022.43-52

Viatcheslav Loveikin[1*], Yuriy Romasevych[1], Lubov Shymko[1], Yuriy Loveikin[2]

[1]National University of Life and Environmental Sciences of Ukraine
03041, 15 Heroiv Oborony, Kyiv, Ukraine

[2]Taras Shevchenko National University of Kyiv
01033, 60 Volodymyrska Str., Kyiv, Ukraine

Minimisation of the Driving Torque of the Derricking Mechanism of a Tower Crane during Steady Load Hoisting

Abstract. Efficient use of a tower crane often requires combining various operations, such as hoisting load and derricking. In the case when the load is hoisted at a steady speed, the problem of optimal control of the trolley movement mechanism arises, which goes beyond engineering calculations and is a scientific and applied problem. Its relevance is related to improving the controllability of crane mechanisms, increasing the capacity and reliability of the crane, and improving the energy efficiency of its drive mechanisms. These indicators are related to the choice of optimisation criteria. Thus, the purpose of the study is to optimise the starting mode of the derricking mechanism according to the criterion of the RMS value of the driving moment during a steady load hoisting. To achieve this goal, the following methods were applied: dynamics of machines and mechanisms, mathematical modelling, integral and differential calculus, and the ME-D-PSO method. For the boundary conditions, parameters are selected that eliminate load oscillations on the flexible suspension when the derricking mechanism slews to the steady-state driving mode. Based on the results of optimisation of the joint movement of mechanisms for derricking and load hoisting, graphical dependences of kinematic, dynamic, and energy characteristics of the start-up transition process are constructed and their analysis is carried out. The obtained dependences reveal the conditions for eliminating load oscillations on a flexible suspension during steady-state movement and reducing dynamic loads and energy losses during the start-up of the derricking mechanism. To implement the optimal start mode of the derricking mechanism during steady load hoisting, it is recommended to use optimal control of the drive mechanisms. The results obtained should be applied to the development of new and modernisation of existing motion control systems for tower crane mechanisms

Keywords: optimal control, optimisation criteria, load oscillations, joined operations

Article's History: Received:28.03.2022; Revised: 08.07.2022; Accepted: 11.08.2022

INTRODUCTION

When operating tower cranes, it is quite common to combine the operations of movement of mechanisms. There are cases when one of the mechanisms performs a steady movement, and at this time another mechanism is switched on. The operation of mechanisms in this mode leads to an increase in dynamic loads caused by high-frequency oscillations of the links of drive mechanisms and the metal structure of the crane, and low-frequency oscillations of the load on a flexible suspension.

The most common case is the operation of crane mechanisms, when the load hoisting mechanism performs a steady movement with the process of starting or braking the derricking mechanism. Due to the significant dynamic loads in the mechanisms during such movements, there is a need to reduce them. Therefore, there is a need to choose such a mode of movement of the derricking mechanism, in which its driving torque would be minimal.

The review of scientific studies on this topic indicated the relevance and significant interest of the world scientific community in research aimed at improving the reliability and efficiency of loading and unloading processes, including by optimising the operating modes of modern

Suggested Citation:
Loveikin, V., Romasevych, Yu., Shymko, L., & Loveikin, Yu. (2022). Minimisation of the driving torque of the derricking mechanism of a tower crane during steady load hoisting. *Machinery & Energetics,* 13(3), 43-52.

*Corresponding author

crane mechanisms, which complies with safe conditions and provides trouble-free operation of hoisting and transport machines.

During the operation of tower cranes (and not only tower cranes), the payload performs a complex movement, in which a number of mechanisms work simultaneously, the operation of which has both mutual and general effects on the crane system [1; 2]. This feature requires the operator to be precise in choosing the operating modes of crane mechanisms, since the error can have dramatic consequences. Therefore, scientific investigation is often aimed precisely at improving the efficiency of controlling hoisting processes. Thus, in the study [3], a comprehensive review of crane control strategies is performed, where, along with a thorough analysis of control schemes for various types of crane systems that have been performed since 2000, an overview of modelling single - and double-lever crane systems is provided. A valuable contribution to the development of the theory of control of crane systems is the classification of strategies for the development and improvement of means of controlling the working processes of crane mechanisms, which systematises the scientific search that aims to optimise operating modes.

A typical line of scientific research is an attempt to improve the reliability and trouble-free operation of all mechanisms of crane systems. With the development of automation tools, first analogue and then digital, researchers focused their efforts on the development of theoretical foundations and practical research of the operating modes of a very wide arsenal of crane mechanisms: from conventional controllers [4] to neural networks, technical vision systems, and artificial intelligence [5-7]. This thesis also applies to specific crane manipulators [8]. The increase in requirements for the quality of loading and unloading processes has caused the expansion of the methodological tools of researchers in the construction of mathematical models of crane systems. At the same time, as noted in the study [9], the large-scale, fast, and modular development of tower cranes has brought significant problems of construction safety in difficult working conditions. The study simulates changes in the rotation angle. The obtained data should be used to improve control systems.

The purpose of [10] is to investigate the dynamic indicators of crane system mechanisms during crane operation and develop computer training methods to improve the quality of crane operator training. The researchers implemented a powerful visualisation interface for displaying the dynamic behaviour of the crane. The study results allow developing an effective programme for training operators of construction tower cranes.

The method of reducing oscillations for fast crane operation is substantiated in [11]. Researchers from the Faculty of Civil Engineering of the National Taiwan University conducted analytical studies to test the three-stage control method proposed by the authors, which aims to reduce the angles of oscillation during the operation of a tower crane at high speed. The method is based on the physical laws of motion of a double pendulum and uses fixed values of acceleration and deceleration to reduce the oscillation angle and frequency. Checking the mathematical model showed that the working cycle time is proportional to the square root of the working distance at a constant oscillation angle and no oscillation.

The evolution of information technologies and computer graphics has contributed to the development of automated methods for modelling loading and unloading processes. Virtual models allow investigating and predicting possible dangerous or inefficient actions of crane mechanisms. In the study [12], the authors focused on developing a mathematical model to support modelling and visualisation of cranes as the most important equipment in terms of project control. The researchers proposed a mathematical model that can more accurately represent the detailed activity of the crane. The computer programme developed by the researchers simulates and visualises the operation of crane mechanisms in real time. The created animation provides interactive functions.

The development of mathematical models is a special aspect of experimental research in the field of engineering. Thus, in studies [13] and [14], mathematical models are constructed, and a system of second-order linear differential equations describing the dynamics of transverse displacement of the centre of mass is obtained.

Slovak researchers conducted experiments using a three-dimensional laboratory model suitable for research in the field of mechanism control. Their study [15] is devoted to the identification of a dynamic model and the development of a trolley position controller along the boom of a tower crane, and damping load oscillations.

In [16], a mathematical model of a jib crane is proposed, which considers the degrees of freedom that characterise the selected type of system. The researchers developed a nonlinear control law that uses all states of the model. Verification of simulation results with real physical parameters shows the effectiveness of the proposed control scheme even in the presence of wind disturbances.

In the study [17], the author formulated optimisation criteria in the form of quadratic functionals for each tower crane mechanism. Minimisation of functionals allowed the researcher to obtain a set of optimal parameters of crane mechanisms, in which their interaction should be effective. An innovative feature of the presented solution is an integrated approach to optimising multi-drive systems, aimed at reducing the oscillations of a load suspended on a flexible suspension and minimising the forces acting on the rope. This approach, combined with the method of selective analysis of mechanism structures, can be developed as a method for synthesising globally optimal configurations of interacting mechanisms [17].

Based on the materials of the review of research on this topic, it was concluded that the scientific community focuses a lot on the problems of optimising the parameters of crane mechanisms and ensuring the reliability and safety of loading and unloading processes. At the same time, various programmes and methods of experimental research are used. However, showing special interest, this study will focus on the provisions and experience of using the metaheuristic optimisation method in these problems – the Particle Swarm Optimisation method (PSO) [18].

First of all, PSO is identified with an evolutionary algorithm, which is applied with considerable success to solving various engineering and technological problems [19; 20]. Since its first publication in 1995 [21], the algorithm has been constantly modified. It seems that at present there is not a single scientific field where various variants of PSO did not have their application for solving a wide range of optimisation problems. This is evidenced by the study by F. Marini and B. Walczak [22], where the researchers emphasise the importance of choosing the right PSO metaparameters. E. Hussain et al. [23] note that due to their flexibility, these algorithms can solve various optimisation problems in various fields, especially in engineering and computer science. The researchers summarised and analysed some of the most influential and diverse developments in various aspects of the PSO algorithm since its initial formulation. A comprehensive study [24], which was conducted by a team of researchers who are representatives of various scientific fields, revealed positive aspects and weaknesses in PSO algorithms. However, the researchers agree that PSO is a promising candidate for optimising a wide range of real-world optimisation problems. Eight potential research areas were identified to further improve the efficiency of PSO optimisation. One example of improving the effectiveness of PSO can be the study [25]. The problem of limited optimisation was considered to study PSO. The optimal set of parameters determined by sensitivity analysis was verified by comparison with other metaheuristic methods. In [26], a modified metaeuristic ME-PSO particle swarm method is also used to solve the nonlinear optimisation problem.

The purpose of this study is to solve the problem of minimising the driving torque of the drive mechanism for derricking when working together with the load hoisting mechanism by selecting the driving mode of the drive mechanisms.

The originality of the study consists in solving the scientific and applied problem of optimising the movement control of the derricking mechanism of a tower crane, provided that the load is hoisted at a steady speed. A special feature of this problem is the methodology for solving it, which consists in reducing it to the problem of unconstrained minimisation of a complex function, the minimum of which is found by the modified particle swarm method.

MATERIALS AND METHODS

To achieve this goal, the following methods were applied: dynamics of machines and mechanisms, mathematical modelling, integral and differential calculus, and ME-D-PSO.

The jib system of a tower crane (Fig. 1) is presented as a holonomic mechanical system consisting of absolutely solid bodies, except for the traction rope of the trolley movement, which has elastic properties.

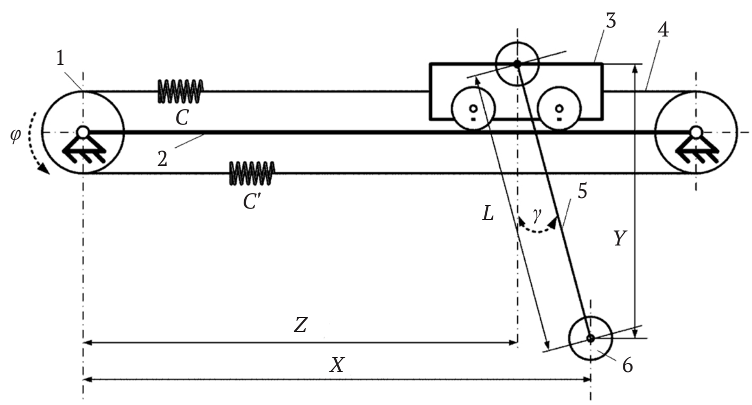

Figure 1. Dynamic model of joint movement of mechanisms for derricking and load hoisting: 1 – drive drum of the trolley; 2 – saddle jib; 3 – trolley; 4 – traction rope; 5 – flexible load suspension; 6 – load

The elastic properties of the rope are represented by the stiffness coefficient C or C' depending on the direction of movement of the trolley, and the flexible suspension of the load, which oscillates in the plane of trolley movement mechanism.

In the accepted dynamic model, the linear coordinates of the trolley's centre of mass are used as generalised coordinates z, and load x, and the angular coordinates of rotation of the trolley drive drum φ. Deviation of the load rope from the vertical is carried out in the plane of trolley movement mechanism by an angle v. At the same time, the length of the flexible load suspension l changes at a constant rate V and is expressed by the following dependence:

$$l=l_0+Vt, \quad (1)$$

where t – time; l_0 – initial length of load suspension.

To create equations of motion of the accepted dynamic model of the boom system (Fig. 1) the second order Lagrange equations are used:

$$\begin{cases} \frac{d}{dt}\frac{\partial T}{\partial \dot{\varphi}} - \frac{\partial T}{\partial \varphi} = M - \frac{\partial \Pi}{\partial \varphi}; \\ \frac{d}{dt}\frac{\partial T}{\partial \dot{z}} - \frac{\partial T}{\partial z} = -W - \frac{\partial \Pi}{\partial z}; \\ \frac{d}{dt}\frac{\partial T}{\partial \dot{x}} - \frac{\partial T}{\partial x} = -\frac{\partial \Pi}{\partial x}, \end{cases} \quad (2)$$

where T, Π – kinematic and potential energy of the system, respectively; M – driving torque on the shaft of the electric motor of the derricking mechanism, reduced to the axis of rotation of the drive drum; W – resistance force to trolley movement.

The vertical component of the movement of the centre of mass of the load is determined by the dependence:

$$y = (l_0 + Vt)\cos\nu = (l_0 + Vt)\cos\left(\frac{x-z}{l_0 - Vt}\right). \quad (3)$$

Taking the time derivative of equation (3), the vertical component of the velocity of the centre of mass of the load is found:

$$\dot{y} = V\cos\left(\frac{x-z}{l_0+Vt}\right) - (l_0+Vt)\sin\left(\frac{x-z}{l_0+Vt}\right)$$
$$\left(\frac{(\dot{x}-\dot{z})(l_0+Vt) - V(x-z)}{(l_0+Vt)^2}\right). \quad (4)$$

Since the first component of equation (4) is more than an order of magnitude larger than the second component, so in further calculations only the first component will be used, i.e.:

$$\dot{y} = V\cos\left(\frac{x-z}{l_0+Vt}\right). \quad (5)$$

Then the kinetic energy of the mechanical system of joint movement of mechanisms for derricking and load hoisting is presented in the following form:

$$T = \tfrac{1}{2}I_1\dot{\varphi}^2 + \tfrac{1}{2}m_3\dot{z}^2 + \tfrac{1}{2}m[\dot{x}^2 + \dot{y}^2] =$$
$$= \tfrac{1}{2}I_1\dot{\varphi}^2 + \tfrac{1}{2}m_3\dot{z}^2 + \tfrac{1}{2}m\left[\dot{x}^2 + +V^2\cos^2\left(\tfrac{x-z}{l_0+Vt}\right)\right]. \quad (6)$$

In this case, the potential energy of the same system, considering the equation (3), is determined by the dependence:

$$\Pi = \frac{1}{2}C(\varphi r - z)^2 - mg(l_0+Vt)\cos\left(\frac{x-z}{l_0+Vt}\right). \quad (7)$$

where I_1 – the moment of inertia of the drive of the derricking mechanism is reduced to the axis of the drive drum; m_3, m – according to the weight of the trolley and load; r – radius of the drive drum of the derricking mechanism; g – gravity acceleration.

Next, the study finds derivatives of equations (6) and (7), which are necessary for system (2):

$$\begin{cases} \dfrac{\partial T}{\partial \varphi} = 0;\ \dfrac{\partial T}{\partial x} = -mV^2\cos\left(\dfrac{x-z}{l_0+Vt}\right)\cdot\sin\left(\dfrac{x-z}{l_0+Vt}\right)\cdot\dfrac{1}{l_0+Vt}; \\ \dfrac{\partial T}{\partial z} = mV^2\cos\left(\dfrac{x-z}{l_0+Vt}\right)\cdot\sin\left(\dfrac{x-z}{l_0+Vt}\right)\cdot\dfrac{1}{l_0+Vt}; \end{cases} \quad (8)$$

$$\frac{\partial T}{\partial \dot{\varphi}} = I_1\dot{\varphi};\ \frac{\partial T}{\partial \dot{z}} = m_3\dot{z};\ \frac{\partial T}{\partial \dot{x}} = m\dot{x};$$
$$\frac{d}{dt}\frac{\partial T}{\partial \dot{\varphi}} = I_1\ddot{\varphi};\ \frac{d}{dt}\frac{\partial T}{\partial \dot{z}} = m_3\ddot{z};\ \frac{d}{dt}\frac{\partial T}{\partial \dot{x}} = m\ddot{x}; \quad (9)$$

$$\frac{\partial \Pi}{\partial \varphi} = cr(\varphi r - z); \quad (10)$$

$$\begin{cases} \dfrac{\partial \Pi}{\partial z} = -c(\varphi r - z) + \mathrm{mg}\sin\left(\dfrac{x-z}{l_0+Vt}\right); \\ \dfrac{\partial \Pi}{\partial x} = -\mathrm{mg}\sin\left(\dfrac{x-z}{l_0+Vt}\right). \end{cases} \quad (11)$$

Given that the angular coordinate $\nu = \dfrac{(x-z)}{l_0+Vt}$ does not exceed 12°, then it is taken in equations (8) and (11) $\sin\nu \approx \nu$, and $\cos\nu \approx 1$; dependencies (8) and (11) will have the form:

$$\frac{\partial T}{\partial \varphi} = 0;\ \frac{\partial T}{\partial z} = mV^2\frac{x-z}{(l_0+Vt)^2};\ \frac{\partial T}{\partial x} = -mV^2\frac{x-z}{(l_0+Vt)^2}; \quad (12)$$

$$\frac{\partial \Pi}{\partial z} = -C(\varphi r - z) - \mathrm{mg}\sin\left(\frac{x-z}{l_0+Vt}\right);\ \frac{\partial \Pi}{\partial x} = \mathrm{mg}\frac{x-z}{l_0+Vt}. \quad (13)$$

After substituting equations (9), (10), (12), and (13) in (2) a system of differential equations of the mechanical system of joint movement of mechanisms for derricking and load hoisting is obtained:

$$\begin{cases} I_1\ddot{\varphi} = M - Cr(\varphi r - z); \\ m_3\ddot{z} - mV^2\dfrac{x-z}{(l_0+Vt)^2} = -W + C(\varphi r - z) + mg\dfrac{x-z}{l_0+Vt} \\ \ddot{x} + V^2\dfrac{x-z}{(l_0+Vt)^2} = -g\dfrac{x-z}{l_0+Vt}. \end{cases} \quad (14)$$

From the last equation of system (14), the coordinate of the trolley's centre of mass is expressed:

$$x - z = -\frac{l_0+Vt}{g + \dfrac{V^2}{(l_0+Vt)}}\ddot{x}. \quad (15)$$

From equation (15):

$$z = x + \ddot{x}P \quad (16)$$

where P – transfer function determined by the dependency:

$$P = \frac{l_0+Vt}{g + \dfrac{V^2}{(l_0+Vt)}}. \quad (17)$$

Now, the time derivatives of equation (16) are taken and the speed and acceleration of the trolley are determined, including the third and fourth derivatives, which will be used in further studies:

$$\dot{z} = \dot{x} + \dddot{x}\dot{P} + \ddot{x}P; \quad (18)$$

$$\ddot{z} = \ddot{x}(\ddot{P} + 1) + 2\dddot{x}\dot{P} + x^{IV}P; \quad (19)$$

$$\dddot{z} = \ddot{x}\dddot{P} + \dddot{x}(3\ddot{P} + 1) + 3x^{IV}\dot{P} + x^{V}P; \quad (20)$$

$$z^{IV} = \ddot{x}P^{IV} + 4\dddot{x}\dddot{P} + x^{IV}(6\ddot{P} + 1) + 4x^{V}\dot{p} + x^{IV}P. \quad (21)$$

The time derivatives of the transfer function (17) used in equation (18)-(21):

$$\dot{P} = V\frac{g + \dfrac{2V^2}{(l_0+Vt)}}{\left[g + \dfrac{V^2}{(l_0+Vt)}\right]^2}; \quad (22)$$

$$\dddot{P} = \frac{2V^6}{(l_0 + Vt)^3 \left[g + \frac{V^2}{(l_0 + Vt)} \right]^3}; \tag{23}$$

$$\ddddot{P} = -\frac{6V^7 g}{(l_0 + Vt)^4 \left[g + \frac{V^2}{(l_0 + Vt)} \right]^4}; \tag{24}$$

$$P^{IV} = \frac{24V^8 g^2}{(l_0 + Vt)^5 \left[g + \frac{V^2}{(l_0 + Vt)} \right]^5}. \tag{25}$$

From the second equation of system (14), the equation of the angular coordinate of the drum of the trolley movement mechanism is found using the relation (15). As a result, the following is obtained:

$$\varphi = \frac{1}{r}\left[z + \frac{1}{C}(m_3\ddot{z} + m\ddot{x} + W) \right]; \tag{26}$$

$$\dot{\varphi} = \frac{1}{r}\left[\dot{z} + \frac{1}{C}(m_3\dddot{z} + m\dddot{x}) \right]; \tag{27}$$

$$\ddot{\varphi} = \frac{1}{r}\left[\ddot{z} + \frac{1}{C}(m_3 z^{IV} + m x^{IV}) \right]. \tag{28}$$

After substituting (26) and (28) into the first equation of system (14), the dependence of the driving moment of the drive of the derricking mechanism on the horizontal coordinate of the centre of mass of the load and its time derivatives are obtained:

$$M = \frac{I_1}{r}\left[\begin{array}{l} \ddot{z} + \frac{1}{c}(m x^{IV} + m_3 z^{IV}) + \\ +(m_3\ddot{z} + m\ddot{x} + W)r \end{array} \right]. \tag{29}$$

Substituting expressions (19 and 21) in dependence (29):

$$\begin{aligned} M &= \left(\frac{I_1}{r} + m_3 r \right)\ddot{z} + \\ &+ \frac{I_1}{cr}(m_3 z^{IV} + m x^{IV}) + (m\ddot{x} + W)r. \end{aligned} \tag{30}$$

To reduce dynamic loads in the jib crane during the joint movement of the mechanisms for derricking and load hoisting, the movement mode of the derricking mechanism will be optimised. Since the purpose of the study is to minimise the driving moment of the derricking mechanism, therefore, its RMS value is taken as a criterion for optimising the movement mode of the jig system (M_{RMS}) at start:

$$M_{RMS} = \left[\frac{1}{t_1}\int_0^{t_1} M^2 dt \right]^{1/2} \to min, \tag{31}$$

where t – time; t_1 – duration of starting the derricking mechanism.

The selected optimisation criterion reflects the effect of loads on the drive elements and structures of the boom system, so it must be minimised. During the movement of the derricking mechanism on the start-up section with a steady load hoisting, it is necessary to provide boundary conditions that eliminate oscillations in the load on the flexible suspension when switching to a steady mode of movement of both mechanisms:

$$\left[\begin{array}{l} t = 0:\ x = z_0, \dot{x} = 0,\ z = z_0,\ \ \dot{z} = 0,\ \varphi = z_0/r\,,\ \dot{\varphi} = 0; \\ t = t_1:\ x = z_0 + \frac{V_y t_1}{2},\ \dot{x} = V_y, z = z_1,\ \ \dot{z} = V_y,\ \varphi = z_1/r\,, \dot{\varphi} = V_y/r\,. \end{array} \right. \tag{32}$$

The boundary conditions (32) are presented as dependences of the horizontal coordinate of the centre of mass of the load and its time derivatives up to and including the fifth order:

$$\left[\begin{array}{l} t = 0:\ x = z_0, \dot{x} = 0,\ \ddot{x} = 0,\ \dddot{x} = 0,\ x^{IV} = -\frac{W\left(g + \frac{V^2}{l_0}\right)}{m_3 l_0};\ x^V = 3\frac{W \cdot V\left(g + \frac{2V^2}{l_0}\right)}{m_3 l_0^2}; \\ t = t_1:\ x = z_1 = z_0 + \frac{V_y t_1}{2},\ \dot{x} = V_y, \ddot{x} = 0,\ \dddot{x} = 0,\ x^{IV} = -\frac{W}{m_3}.\frac{g + \frac{V^2}{(l_0 + Vt_1)}}{l_0 + Vt_1}, \\ x^V = 3\frac{W \cdot V}{m_3} \cdot \frac{g + 2V^2/(l_0 + Vt_1)}{(l_0 + Vt_1)^2}. \end{array} \right. \tag{33}$$

The proposed optimisation problem (31), (33), considering the expression of the driving moment (30) for the dependences of the transfer function (17) and its time derivatives (22),..., (25), cannot be solved by analytical methods. Therefore, an approximate solution to this optimisation problem is used. The essence of the approximate method for solving problems (31), (33) is reduced to the problem of a class of multiparametric functions that satisfy the boundary conditions (33) and provide a minimum of the integral function (31). The class of multiparametric functions on which the solution of the optimisation problem is located is defined as the solution of the following boundary value problem:

$$\begin{cases} L(x) = 0; \\ \begin{cases} t = 0\!: x = z_0, \dot{x} = 0,\ \ddot{x} = 0,\ \dddot{x} = 0,\ x^{IV} = -\dfrac{W(g + V^2/l_0)}{m_3 l_0};\ x^{V} = 3\dfrac{W \cdot V(g + 2V^2/l_0)}{m_3 l_0^2}; \\ t = \dfrac{t_1}{2}\!: x = x_{t_1/2}, \dot{x} = \dot{x}_{t_1/2}, \\ t = t_1\!: x = z_0 + \dfrac{V_y t_1}{2},\ \dot{x} = V_y, \ddot{x} = 0,\ \dddot{x} = 0,\ x^{IV} = -\dfrac{W}{m_3} \cdot \dfrac{g + V^2/(l_0 + Vt_1)}{l_0 + Vt_1}, \\ x^{V} = 3\dfrac{W \cdot V}{m_3} \cdot \dfrac{[g + 2V^2/(l_0 + Vt_1)]}{(l_0 + Vt_1)^2}. \end{cases} \end{cases} \quad (34)$$

where $L(x)$ – an operator that depends on the function $x(t)$ and its time derivatives.

The solution of the boundary value problem (34) has two unknown parameters $x_{t_1/2}$ and $\dot{x}_{t_1/2}$. Operator $L(x)$ for the set optimisation problem is taken as a fourteenth-order differential equation, i.e.: $L(x)=x^{XIV}(t)$. As a result of solving the boundary value problem (34), the expression of the functional (31) is formed:

$$C_r = C_r(x_{t_1/2}, \dot{x}_{t_1/2}, t_1), \quad (35)$$

Which is a nonlinear function of unknown arguments $x_{t_1/2}, \dot{x}_{t_1/2}$ and t_1. To find the following values $x_{t_1/2}$ and $\dot{x}_{t_1/2}$ in which the functional (31) acquires a minimum value, a ME-D-PSO was used [27]. Its use does not require continuity and differentiation of criterion (35) and does not impose strict requirements on the optimisation problem. After constructing the functional dependency (35), it is possible to apply the ME-D-PSO method and find the optimal values of unknown parameters $x_{t_1/2}$ and $\dot{x}_{t_1/2}$. When solving the optimisation problem, the following parameters of the method are used: the rate of reduction of the criterion AR = 0.005; the number of particles (swarm population) is 50; the number of iterations is 40. The given parameters of the method allow effectively using computational resources when solving the optimisation problem of determining the desired driving mode of the drive mechanism.

For calculations, the jig system of a tower crane with a jig saddle with the following parameters was used: m=5000 kg; m_3=300 kg; I_1=50,16 kg·m²; r=0,15 m; z_0=10,0 m; l_0=16,0 m; C=7,24·10⁵ N·m; g=9,81 m/s; W=5500 N; V=0,85 m/s; t_1=5,0 s.

RESULTS AND DISCUSSION

As a result of solving the optimisation problem, graphical dependencies of kinematic parameters (Figs. 2 and 3), power (Fig. 4), and energy characteristics (Figs. 8 and 9) of the derricking mechanism during the start-up process are constructed.

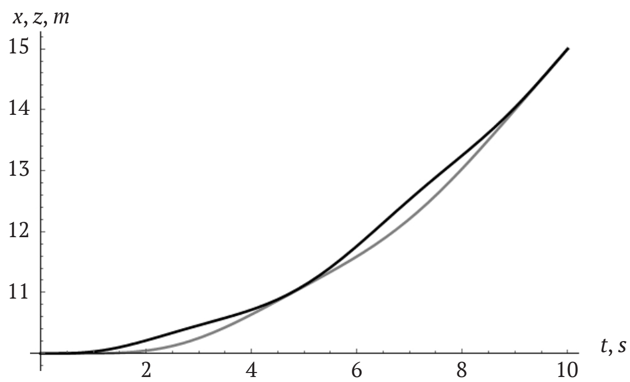

Figure 2. Graphs of horizontal movement of trolley and load

In Figures 2-4, the dark curve shows the characteristics of the trolley, and the light curve shows the characteristics of the load. Fig. 2 shows that at the beginning and end of the start, the movements of the trolley and load coincide. At the same time, during the start-up process, there are deviations in the movements of the trolley and load, which indicates the presence of oscillations of the load on the flexible suspension relative to the trolley. The change in the movement of the load is smoother compared to the movement of the trolley.

Figure 2 shows graphical dependences of changes in the speed of the trolley and load during the start-up process of the derricking mechanism, which indicate that oscillations in the speed of both the trolley and the load are observed during movement. These oscillations are caused by the swinging of the load on the flexible suspension and the elastic properties of the traction body of the trolley.

At the end of the start-up process, the speeds of the trolley and load coincide, and the movements of the trolley and load coincide at this point in time (Fig. 2). This indicates that after the trolley movement mechanism is released, there will be no oscillations in the load on the flexible suspension.

Figure 4 shows that during the start-up of the mechanism, changes in the trolley movement of the acceleration of the trolley and load have an oscillatory character with a significant amplitude of oscillations.

With an average trolley and load acceleration of 0.11 m/s² the maximum acceleration value of the trolley is 0.27 m/s², and the trolley – 0.21 m/s². At the beginning and end of the start-up process, the acceleration of the trolley and load takes zero values, which contributes to the smooth movement of the trolley and load in the area of stable movement of the derricking mechanism.

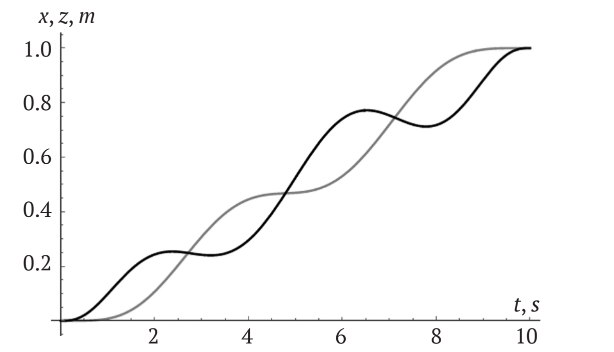

Figure 3. Graphs of horizontal speeds of trolley and load

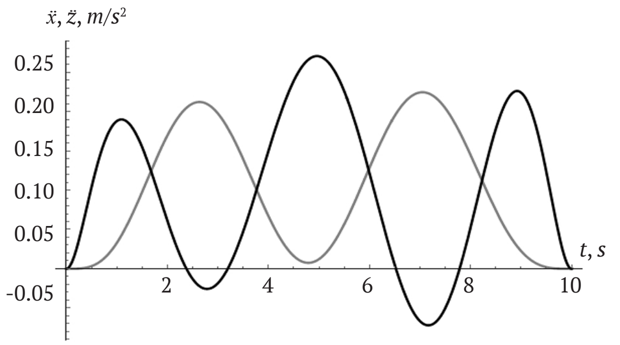

Figure 4. Graphs of horizontal accelerations of trolley and load

From a flat phase trajectory of load oscillations relative to the trolley (Fig. 5) it can be seen that during the start-up of the mechanism, changes in the trolley movement of oscillations fade out during two periods of oscillations. At the same time, the maximum deviation of the trolley and load movements is 0.32 m, and the maximum deviation of the trolley and load speeds reaches 0.21 m/s.

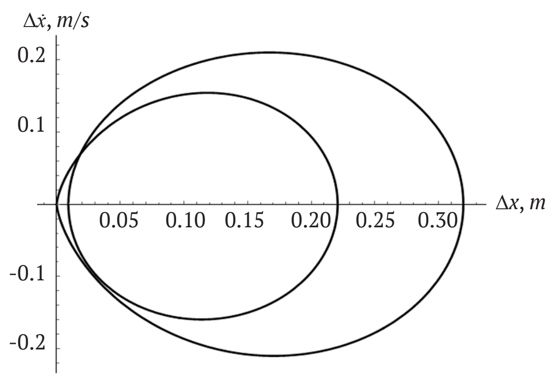

Figure 5. Phase trajectory of load oscillations relative to the trolley

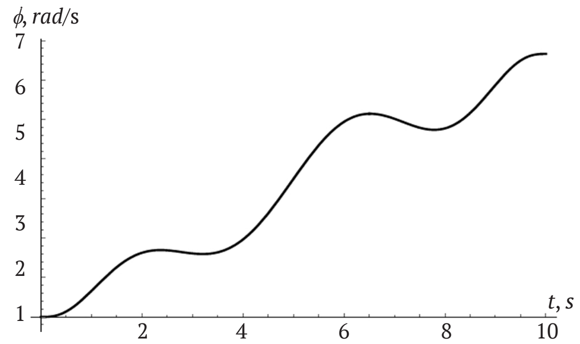

Figure 6. Graph of the angular velocity of the trolley drive drum

Angular velocity of the trolley drive drum (Fig. 6) changes from zero to a steady-state value if there are oscillations during the start-up process. The presence of oscillations of the drive drum is caused by oscillations of the trolley and load on the flexible suspension.

Figure 7 shows that the acceleration of the drive drum also has an oscillatory character. At the same time, in the middle of the start-up, it takes the maximum value, which is 1.8 rad/s², and at the end of the start-up reaches zero. This contributes to the normal operation of the drive drum during the steady movement of the derricking mechanism.

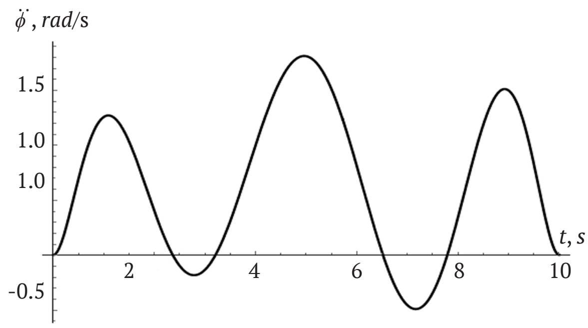

Figure 7. Graph of angular acceleration of the trolley drive drum

Figure 8 shows that at the beginning of start-up, the driving moment of the drive mechanism for derricking from the moment of static resistance forces to the movement of the trolley changes smoothly to a stable value, where moment oscillations are observed caused by the oscillations of the load on the flexible suspension. At the end of the start-up process, the driving moment gradually decreases to the value of the moment of static resistance forces. In the area of steady-state movement, in the absence of oscillations of the trolley and the load on the flexible suspension, the driving torque of the drive takes a steady-state value corresponding to the moment of static resistance forces to the movement of the trolley. Drive power of the derricking mechanism (Fig. 9) during the start-up process, it changes from zero to a steady-state value in the presence of oscillations. These oscillations in the nature of the change are close to oscillations in the angular velocity of the drive drum of the derricking mechanism (Fig. 6).

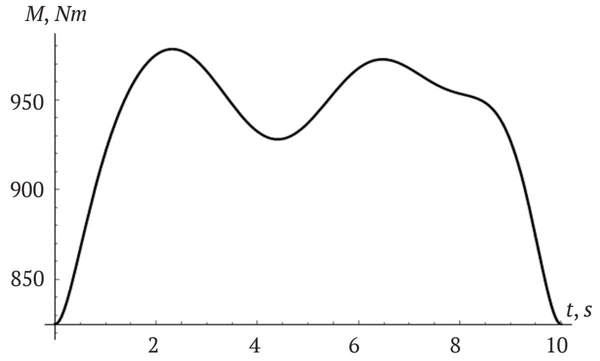

Figure 8. Graph of the driving torque of the derricking mechanism drive

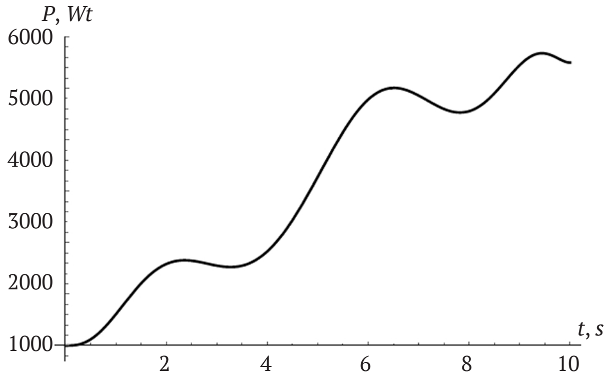

Figure 9. Power graph for the derricking actuator

Comparing the results obtained with well-known studies, it is worth noting a significant variety of methods for solving problems similar to the one that was solved in this study. For example, in [5], the apparatus of artificial neural networks was used. As a result, it was possible to almost completely eliminate the pendulum oscillations of the load on the flexible suspension. However, such control actions of the system in a strict sense are not optimal. In [11], the authors obtained an optimal method for controlling the movement of the crane movement mechanism in terms of speed. At the same time, it was assumed that the length of the flexible suspension of the load is constant, which distinguishes [11] from this study. In [16], the problem of eliminating the pendulum oscillations of the load is set quite broadly – five degrees of freedom of the system are considered, two of which correspond to the pendulum oscillations of the load. The results of the above study also allowed the authors to obtain a control corresponding to the absence of load oscillations in the tangential and radial directions at the end of acceleration. However, such control is also not optimal. The type of control is also different – it is a function of the phase coordinates of the system (the control obtained in this study is a function of time). In [17], the optimisation problem is presented in a variational formulation, similar to this study. The range of issues considered is quite wide, which allowed the author to improve the dynamic and energy performance of the crane and minimise the pendulum oscillations of the load. This makes the above study similar to this one. All calculations are given for a crane whose reach is changed by hoisting/lowering the beam, although the method is quite general and therefore, can obviously be used for other types of tower cranes.

CONCLUSIONS

In the conducted research, the problem of minimising the driving torque of the drive mechanism for derricking during a steady load hoisting is set and solved by selecting the driving mode of the drive.

A mathematical model of the dynamics of the movement of the derricking mechanism of a tower crane with a steady movement of the load hoisting mechanism is constructed, which considers the main movement of the drive mechanisms and oscillations of links with elastic properties. The model is a system of second-order nonlinear differential equations that is reduced to a single sixth-order differential equation.

To optimise the starting mode of the derricking mechanism during the steady-state movement of the hoisting mechanism, an integral dynamic criterion was reasonably chosen, which is presented in the form of the RMS value of the driving torque of the drive during the start-up process. To solve the nonlinear optimisation problem of selecting the mode of movement of the drive of the derricking mechanism during steady-state movement of the load hoisting mechanism, a modified metaheuristic method of a swarm of particles is used.

The results of optimisation of the driving mechanism mode showed that minimising the value of the driving torque led to a reduction in dynamic loads acting on the elements of crane mechanisms, and the elimination of oscillations of the trolley and load during the transition from the start-up process to the steady-state movement of the derricking mechanism. At the same time, oscillations of the trolley and load on a flexible suspension are observed at the start-up site.

Prospects for further study in this area are to consider other factors of the process of joint movement of mechanisms (for example, wind effects on load and trolley, the action of centrifugal force when the crane slews, etc.). The reasonable choice of a multiparameter function, on which an approximate solution of the problem is sought, also remains an open question. In addition, in subsequent studies, a transition will be made from a single (similar to (31)) to complex optimisation criteria that could reflect several undesirable indicators, which would allow the authors to more fully consider the requirements for the mode of movement of mechanisms.

REFERENCES

[1] Abdel-Rahman, E.M., Nayfeh, A.H., & Masoud, Z.N. (2003). Dynamics and control of cranes: A review. *Journal of Vibration and Control*, 9.7, 863-908.

[2] Romasevych, Y., Shymko, L., Ohiienko, M., Duczmal, W., Potwora, W., Titova, L., & Rogovskii, I. (2020). *Agrotronics and optimal control of cranes and hoisting machines*. Opole: The Academy of Management and Administration in Opole.

[3] Ramli, L., Mohamed, Z., Abdullahi, A.M., Jaafar, H.I., & Lazim, I.M. (2017). Control strategies for crane systems: A comprehensive review. *Mechanical Systems and Signal Processing*, 95, 1-23. doi: 10.1016/j.ymssp.2017.03.015.

[4] Singer, N., Singhose, W., & Kriikku, E. (1997). An input shaping controller enabling cranes to move without sway. In *American nuclear society topical meeting on robotics and remote systems* (pp. 12-14). Augusta: Paine College. doi: 10.2172/491559.

[5] Fasih, S.M., Mohamed, Z., Husain, A.R., Ramli, L., Abdullahi, A.M., Anjum, W. (2020). Payload swing control of a tower crane using a neural network-based input shaper. *Measurement & Control*, 53(7-8), 1171-1182. doi: 10.1177%2F0020294020920895.

[6] Qian, Y., Hu, D., Chen, Y., Fang, Y., & Hu, Y. (2022). Adaptive neural network-based tracking control of underactuated offshore ship-to-ship crane systems subject to unknown wave motions disturbances. *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, 52(6), 3626-3637. doi: 10.1109/TSMC.2021.3071546.

[7] Jordan, M.I. (2019). Artificial intelligence – the revolution hasn't happened yet. *Harvard Data Science Review,* 1(1), 1-8. doi: 10.1162/99608f92.f06c6e61.

[8] Stepanov, M.A., & Ilukhin, P.A. (2018). Features and perspectives of automatized construction crane-manipulators. *IOP Conference Series: Materials Science and Engineering*, 317(1), article number 012039. doi: 10.1088/1757-899X/317/1/012039.

[9] Liu, F., Yang, J., Wang, J., & Liu, C. (2021). Swing characteristics and vibration feature of tower cranes under compound working condition. *Shock and Vibration*, 2021, article number 8997396. doi: 10.1155/2021/8997396.

[10] Kang, S.C., & Miranda, E. (2004). Physics based model for simulating the dynamics of tower cranes. In *Proceeding of Xth international conference on computing in civil and building engineering* (pp. 1-8). Weimar: Bauhaus-Universität Weimar. doi: 10.25643/bauhaus-universitaet.240.

[11] Kuo, T., Chiang, Y.-C., Cheng, S.-Y., & Jessy Kang, S.-C. (2015). Oscillation reduction method for fast crane operation. *Modular and Offsite Construction Summit Proceedings*, 1, 388-395. doi: 10.29173/mocs159.

[12] Shih-Chung, K., & Miranda, E. (2009). Numerical methods to simulate and visualize detailed crane activities. *Computer-Aided Civil and Infrastructure Engineering*, 24.3, 169-185. doi: 10.1111/j.1467-8667.2008.00579.x.

[13] Olt, J., Bulgakov, V., Beloev, H., Nadykto, V., Ihnatiev, Ye., Dubrovina, O., Arak, M., Bondar, M., & Kutsenko, A. (2022). A mathematical model of the rear-trailed top harvester and an evaluation of its motion stability. *Agronomy Research*, 20(2), 371-388 doi: 10.15159/ar.21.162.

[14] Mykhailov, Y., Zadosna, N., Postnikova, M., Pedchenko, G., Khmelovskyi, V., Bondar, M., Ionichev, A., Kozdęba, M., & Tomaszewska-Górecka, W. (2021). Energy assessment of the pneumatic sieve separator for agricultural crops. *Agricultural Engineering*, 25, 147-156. doi: 10.2478/agriceng-2021-0012.

[15] Čápková, R., & Kozáková, A. (2019). Experimental modelling and control of a tower crane. In *22nd international conference on process control* (pp. 97-101). Strbske Pleso: Hotel Toliar. doi: 10.1109/PC.2019.8815327.

[16] Ambrosino, M., Berneman, M., Carbone, G., Crépin, R., Dawans, A., & Garone, E. (2021). Modelling and control of 5-dof boom crane. *International Symposium on Automation and Robotics in Construction*, 2, 1-8. doi: 10.48550/arXiv.2103.02454.

[17] Chwastek, S. (2020). Optimization of crane mechanisms to reduce vibration. *Automation in Construction*, 119, article number 103335. doi: 10.1016/j.autcon.2020.103335.

[18] Romasevych, Yu., Loveikin, V., Ohiienko, M., Shymko, L., & Łukawiecki, K. (2021). *Innovation management in agriculture. Agrotronics and design of optimal controllers based on new modifications of particle swarm optimization*. Opole: Publishing House WSZiA

[19] Chopard, B., & Tomassini, M. (2018). Particle swarm optimization. In *An introduction to metaheuristics for optimization* (pp. 97-102). Cham: Springer. doi: 10.1007/978-3-319-93073-2_6.

[20] Piotrowski, A.P., Napiorkowski, J.J., & Piotrowska, A.E. (2020). Population size in particle swarm optimization. *Swarm and Evolutionary Computation*, 58, article number 100718. doi: 10.1016/j.swevo.2020.100718.

[21] Kennedy, J., & Eberhart, R. (1995). Particle swarm optimization. In *Proceedings of International Conference on Neural Networks*, 4, 1942-1948. doi: 10.1109/ICNN.1995.488968.

[22] Marini, F., & Walczak, B. (2015). Particle swarm optimization (PSO). A tutorial. *Chemometrics and Intelligent Laboratory Systems*, 149(B), 153-165. doi: 10.1016/j.chemolab.2015.08.020.

[23] Houssein, E.H., Gad, A.G., Hussain, K., & Suganthan, P.N. (2021). Major advances in particle swarm optimization: Theory, analysis, and application. *Swarm and Evolutionary Computation*, 63, article number 100868. doi: 10.1016/j.swevo.2021.100868.

[24] Shami, T.M. (2022). Particle swarm optimization: A comprehensive survey. *IEEE Access*, 10, 10031-10061. doi: 10.1109/ACCESS.2022.3142859.
[25] Isiet, M., & Gadala, M. (2020). Sensitivity analysis of control parameters in particle swarm optimization. *Journal of Computational Science*, 41, article number 101086. doi: 10.1016/j.jocs.2020.101086.
[26] Loveikin, V., Romasevich, Y., Shymko, L., Loveikin, Y., & Pochka, K. (2022). The dynamic analysis of the joint trolley movement and hoisting mechanism in the tower crane. *Strength of Materials and Theory of Structures*, 108, 267-282.
[27] Romasevych, Yu., Loveikin, V., & Makarets, V. (2020). Optimal constrained tuning of pi-controllers via a new pso-based technique. *International Journal of Swarm Intelligence Research*, 11(4), 87-105. doi: 10.4018/IJSIR.2020100104.

Вячеслав Сергійович Ловейкін[1], Юрій Олександрович Ромасевич[1], Любов Сергіївна Шимко[1], Юрій Вячеславович Ловейкін[2]

[1]Національний університет біоресурсів і природокористування України
03041, вул. Героїв Оборони, 15, м. Київ, Україна

[2]Київський національний університет імені Тараса Шевченка
01033, вул. Володимирська, 60, Київ, Україна

Мінімізація рушійного моменту приводу механізму зміни вильоту баштового крана при усталеному підйомі вантажу

Анотація. Ефективне використання баштового крана досить часто вимагає суміщення різних операцій, наприклад, підйому вантажу та зміни вильоту. У випадку коли вантаж піднімається при усталеній швидкості виникає задача оптимального керування механізмом переміщення візка, яка виходить за рамки інженерних розрахунків і представляє собою науково-прикладну задачу. Її актуальність пов'язана із покращенням керованості кранових механізмів, підвищення продуктивності та надійності роботи крана, покращенні енергоефективності його приводних механізмів. Вказані показники пов'язані з вибором критерію оптимізації. Отже, метою статті є оптимізація режиму пуску механізму зміни вильоту за критерієм середньоквадратичного значення рушійного моменту при усталеному підйомі вантажу. Для досягнення цієї мети були використані методи: динаміки машин і механізмів, математичного моделювання, інтегрального і диференціального числення, модифікований метод рою часточок ME-D-PSO. За крайові умови руху обрано параметри, які усувають коливання вантажу на гнучкому підвісі при виході механізму зміни вильоту на усталений режим руху. За результатами оптимізації сумісного руху механізмів зміни вильоту та підйому вантажу побудовані графічні залежності кінематичних, динамічних та енергетичних характеристик перехідного процесу пуску і проведено їхній аналіз. З отриманих залежностей виявлено умови усунення коливань вантажу на гнучкому підвісі при усталеному русі та зменшення динамічних навантажень та енергетичних витрат в процесі пуску механізму зміни вильоту. Для реалізації оптимального режиму пуску механізму зміни вильоту при усталеному підйомі вантажу рекомендовано використовувати оптимальне керування приводними механізмами. Отримані результати доцільно застосовувати для розробки нових і модернізації існуючих систем керування рухом механізмами баштових кранів

Ключові слова: оптимальне керування, критерій оптимізації, коливання вантажу, суміщення операцій