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IMPROVEMENT OF THE GENERALIZED FORCE CRITERION OPTIMIZATION OF OVERHEAD CRANES MOVEMENT MODES

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Abstract. During the operation of overhead cranes, pendulum oscillations of the payload are often observed, which causes uneven movement of these cranes, their trolleys, loads on the ropes and power elements of the cranes, which, in turn, create various inconveniences during their operation, reduce the reliability of the functioning of both the crane as a whole and its individual elements. It is clear that all of these factors must be taken into account in the refined calculations of cranes (especially in the modes of their optimal (with the minimum driving force required for this) start/braking).

The paper uses a standard methodology and scheme for calculating pendulum oscillations of a payload on the cables of an overhead crane, which are usually carried out within a two-mass model of a crane system. Further refinements and improvements to the above methodology have been made on the basis of a well-founded generalized force criterion for the quality of crane movement. The dependencies describing the law of motion of the crane system and the law of change in time of the applied driving force during the startup/braking stages were obtained, which satisfy the above-mentioned power criterion and ensure high-quality (smooth) movement of the system during its startup or braking.

The law of motion of the crane rotation mechanism (crane drive) at its stopping, as well as the law of motion of the cargo at its lifting by the corresponding crane mechanism and sharp braking, at which the dynamic loads in the drive and in the crane rope, respectively, are minimized, is established. The results obtained in the work can be further used to clarify and improve the existing engineering methods for calculating the start-up modes of overhead cranes both at the stages of their design and in the modes of real operation.

Keywords: improvement, generalized force criterion, optimization, motion modes, starting, braking, overhead cranes, two-axle model.

Introduction. During the operation of overhead travelling cranes the oscillations of load are observed which cause unequal motion of overhead travelling cranes or their trolleys, additional loads on the power elements of these cranes, create inconveniences in their operation, which, of course, must be carefully considered in refined calculations of both cranes themselves and (mechatronic) control systems by them.

The existing methods of the analysis of forced (including pendulum) oscillations of cargo on the ropes according to the classical scheme of the mathematical analysis and the simplest model of the crane system (two-mass) for different laws of change in time (t) of the driving/forced force (F(t)) require, in the opinion of the authors of this investigation, further specification and perfection with the purpose of optimization (minimization of force and kinematic characteristics of motion) as the control systems of cranes at their start/braking, as well as the search of new economically proved methods.

Analysis of the latest research sources and publications. The calculation of load oscillations is usually carried out according to the simplest scheme of two-mass system [1-14], at that, considering that angle of ropes deflection from vertical does not exceed 10° ... 12° (so called small oscillations). It is also considered that the period of pendulum oscillations of load on flexible ropes is more or the same order as the period of acceleration (braking) of the crane, and the driving force of the driving motor of the travel mechanism is constant and equal to the average starting (braking) value [8, 9].

To substantiate the force (generalized) criterion for quality of motion of an overhead crane in start-up/braking modes, the approach of the authors [1, 14] has been applied. In addition, the results of research of the authors [15] have been used in the paper.

Aim and objectives. The aim of this paper consists in substantiation of the kinematic-force criterion (generalized) for optimization of modes and quality of bridge crane movements under their start/braking which provides high quality of such movements (minimization of kinematic and force characteristics of movements) as well as in definition of values of the specified parameters and duration of transients for these (stated by this investigation) optimal modes of such crane systems functioning.

Research methodology. We have used the apparatus of mathematical physics, methods of the classical calculus of variations and methods of the solution of the ordinary linear differential equations.

Statement of basic contents of research. In [8, 9] it has been established that within the framework of two-mass model of the bridge crane ("cargo" – "load carriage", connected by a rope), the swinging of cargo at start-up/braking is described by the following equation:

$$\ddot{x} + \omega^2 \cdot x = (P - W)/m_1, \qquad (1)$$

where: X is the horizontal movement of the load relative to the moving point of the load suspended on the rope to the load carriage; $\omega = \sqrt{\frac{(m_1 + m_2)\cdot g}{m_1 \cdot H}}$ – natural frequency of pendulum oscillations of the crane's load during the acceleration period; g=9.81 m/s^2 – acceleration of free fall; m_1 – mass of the crane or trolley, given to translational motion of the crane or trolley; m_2 – mass of the load; H – length of the ropes; P – total traction/braking force of the crane or trolley drive wheels; W – resistance force to movement of the crane or trolley; $x = x_1 - x_2$, where are horizontal displacements of masses m1 and m_2, respectively. It should be noted that in equation (1) the specified variant of motion of the load carriage and the load on the rope, when $(\dot{x}_1 - \dot{x}_2) = \dot{x} > 0$ otherwise the sign in front of the force W is reversed, i.e. at $\dot{x} < 0$ $(\dot{x}_2 > \dot{x}_1)$. Assuming that the drag force against the movement of the crane or trolley is the Coulomb (dry) friction force, it can be added in general as follows:

$$F_{on.} = -W \cdot sign(\dot{x}_1 - \dot{x}_2) = -W \cdot sign(\dot{x}), \ W > 0. \qquad (2)$$

(It is clear that in notation (2) $F_{on.}$. multiplier W denotes the amplitude of the dry friction force). At zero initial conditions:

$$x\big|_{t=0} = \dot{x}\big|_{t=0} = 0. \qquad (3)$$

solution (1) has the form [8, 9]:

$$x = A \cdot (1 - \cos\omega t), \ A = (P - W)\cdot H / [g \cdot (m_1 + m_2)]. \qquad (4)$$

After acceleration to a steady-state movement of the trolley with constant speed V $(\dot{x}_1\big|_{t=t_П} = V)$ the load must have zero velocity (the vibrations disappear $\dot{x}_2\big|_{t=t_П} = 0)$, where $t_П$ – is the duration of the start-up process, so we have:

$$t = t_П \Rightarrow (\dot{x}_1 - \dot{x}_2)\big|_{t=t_П} = V - 0 = V. \qquad (5)$$

This condition makes it easy to determine $t_П$:

$$\dot{x} = (\dot{x}_1 - \dot{x}_2), \ \dot{x}\big|_{t=t_П} = V. \qquad (6)$$

Given (4), we have:

$$\dot{x} = \omega \cdot A \cdot \sin \omega t \Rightarrow V = \omega \cdot A \cdot \sin(\omega t_{\Pi}). \tag{7}$$

Hence:

$$t_{\Pi} = (1/\omega) \cdot \arcsin(V/(A \cdot \omega)). \tag{8}$$

The path the trolley will take in time t_{Π} can be found from the relation [8, 9]:

$$m_1 \cdot \ddot{x}_1 + c \cdot (x_1 - x_2) = P - W, \;\; c = m_2 \cdot g / H. \tag{9}$$

Then for $\ddot{x}_1$ (9) we have:

$$\ddot{x}_1 = (P - W) - c \cdot x = (P - W) - c \cdot A \cdot (1 - \cos \omega t). \tag{10}$$

Then at zero initial conditions for x_1:

$$x_1\big|_{t=0} = \dot{x}_1\big|_{t=0} = 0, \tag{11}$$

from (10) it is easy to obtain the distance L that the crane trolley travels to reach a steady state (with speed V=const.) at the end of the acceleration period ($t = t_{\Pi}$):

$$x_1\big|_{t=t_{\Pi}} = \frac{(P - W - c \cdot A) \cdot t_{\Pi}^2}{2} + \frac{c \cdot A}{\omega^2} \cdot \left\{ \sqrt{1 - \left(\frac{V}{A \cdot \omega} \right)^2} - 1 \right\}. \tag{12}$$

In order to find the minimum effective motive force $F_{mf}^{(eff)}$ to accelerate the system and reach a steady speed V, the following criterion for the quality of motion of the crane trolley during the acceleration period must be fulfilled:

$$F_{mf}^{(eff)} = \frac{(P - W)}{m_1} \Rightarrow \min, \tag{13}$$

which is equivalent to the condition:

$$\int_0^{t_{\Pi}} \left\{ F_{mf}^{(eff)} \right\}^2 dt \Rightarrow \min, \tag{14}$$

or, using equation (1), criterion (14) can be represented as:

$$\int_0^{t_{\Pi}} \left(\ddot{x} + \omega^2 \cdot x \right)^2 dt \Rightarrow \min. \tag{15}$$

Criterion (15) can be realized if the law of motion satisfies the Euler-Poisson equation:

$$x^{(IV)} + 2\omega^2 \cdot \ddot{x} + \omega^4 \cdot x = 0. \tag{16}$$

Solve (16) under the following initial and final conditions:

$$x\big|_{t=0} = \dot{x}\big|_{t=0} = 0; \;\; \ddot{x}\big|_{t=t_{\Pi}} = 0; \;\; \dot{x}\big|_{t=t_{\Pi}} = V, \tag{17}$$

as follows:

$$x(t) = C_1 \cdot \sin \omega t + C_2 \cdot \cos \omega t + C_3 \cdot t \cdot \sin \omega t + C_4 \cdot t \cdot \cos \omega t. \tag{18}$$

Undefined constants C_i, $i = \overline{(1.4)}$, we find from conditions (17), then we have:

$$\begin{cases} C_2 = 0; \;\; \omega \cdot C_1 + C_4 = 0; \\ \omega \cdot C_1 \cdot \cos \omega t_{\Pi} + C_3 \cdot (\sin \omega t_{\Pi} + \omega t_{\Pi} \cdot \cos \omega t_{\Pi}) + C_4 \cdot (\cos \omega t_{\Pi} - \omega t_{\Pi} \cdot \sin \omega t_{\Pi}) = V; \\ -\omega^2 \cdot C_1 \cdot \sin \omega t_{\Pi} + C_3 \cdot \left\{ 2\omega \cdot \cos \omega t_{\Pi} - \omega^2 \cdot t_{\Pi} \cdot \sin \omega t_{\Pi} \right\} + C_4 \cdot \left\{ -2\omega \cdot \sin \omega t_{\Pi} - \omega^2 \cdot t_{\Pi} \cdot \cos \omega t_{\Pi} \right\} = 0. \end{cases} \tag{19}$$

As a result of solving the system of equations (19) with respect to C_i, $i = \overline{(1.4)}$, by Cramer's rule we have:

$$\begin{cases} C_2 = 0; \;\; C_1 = -C_4/\omega; \;\; C_3 = \dfrac{V \cdot (\sin \omega t_{\Pi} + \omega t_{\Pi} \cdot \cos \omega t_{\Pi})}{(\sin^2 \omega t_{\Pi} + \omega^2 \cdot t_{\Pi}^2)}; \\ C_4 = \dfrac{V \cdot (2\cos \omega t_{\Pi} - \omega t_{\Pi} \cdot \sin \omega t_{\Pi})}{(\sin^2 \omega t_{\Pi} + \omega^2 \cdot t_{\Pi}^2)}. \end{cases} \tag{20}$$

The law of optimum (in the sense of the motion quality criterion (13)-(15)) motion $x(t)$ becomes:

$$x(t) = C_1 \cdot \sin \omega t + C_3 \cdot t \cdot \sin \omega t + C_4 \cdot t \cdot \cos \omega t. \tag{21}$$

The law of optimum load carriage movement takes on the following form:

$$\dot{x}_1 = (P - W)t - c\left\{C_1 \omega^{-1} \cdot (1 - \cos \omega t) + C_3\left[-t \cdot \omega^{-1} \cdot \cos \omega t + \omega^{-2} \cdot \sin \omega t\right] + C_4\left[\frac{t}{\omega} \sin \omega t + \frac{\cos \omega t}{\omega^2} - \omega^{-2}\right]\right\} + \overline{C_1}. \tag{22}$$

Given the initial conditions (11) for $\overline{C_1}$ we have a ratio:

$$\overline{C_1} = 0. \tag{23}$$

The final optimum time variation of the load carriage speed $V_{bogie} = x_1(t)$ over the run-up period $(0 < t \le t_n)$ takes the following form:

$$\dot{x}_{1opt}(t) = (P - W) \cdot t - \frac{c}{\omega} \cdot \left\{C_1(1 - \cos \omega t) + C_3(-t \cos \omega t + \frac{\sin \omega t}{\omega}) + C_4\left(t \sin \omega t + \frac{\cos \omega t}{\omega} - \omega^{-1}\right)\right\}. \tag{24}$$

Optimum law of motion of the trolley in time t $x_1(t) \equiv x_{1opt}(t)$ assumes the following form:

$$x_{1opt}(t) = \frac{(P - W) \cdot t^2}{2} - c \cdot \left\{\frac{C_1}{\omega}\left(t - \frac{\sin \omega t}{\omega}\right) + \frac{C_3}{\omega}\left(-\frac{2\cos \omega t}{\omega^2} + \frac{2}{\omega^2} - \frac{t}{\omega} \cdot \sin \omega t\right) + \frac{C_4}{\omega}\left(-\frac{t}{\omega} + \frac{2\sin \omega t}{\omega^2} - \frac{t}{\omega}\cos \omega t\right)\right\} + \overline{\overline{C_2}}. \tag{25}$$

Using the initial conditions (11) for $\overline{\overline{C_2}}$ we have the following relationship:

$$\overline{\overline{C_2}} = 0. \tag{26}$$

Therefore, finally, the law of motion of the trolley in time t when the motion quality criteria (13) -(15) are fulfilled is as follows:

$$x_{1opt}(t) = \frac{(P - W)t^2}{2} - c \cdot \left\{\begin{array}{l} \frac{C_1}{\omega}\left(t - \frac{\sin \omega t}{\omega}\right) + \frac{C_3}{\omega}\left(-\frac{2\cos \omega t}{\omega^2} + \frac{2}{\omega^2} - \frac{t}{\omega}\sin \omega t\right) + \\ + \frac{C_4}{\omega}\left(-\frac{t}{\omega} + \frac{2\sin \omega t}{\omega^2} - \frac{t}{\omega}\cos \omega t\right) \end{array}\right\}. \tag{27}$$

The equivalent scheme for calculation of dynamic loads arising at stopping of crane rotation mechanisms can be presented in the form of a single-mass system with a walled elastic link [3]. Applying the D'Alamber principle it is easy to see that the motion of such a system is described by a differential equation:

$$J \cdot \ddot{\varphi} + c' \cdot \varphi = M_p, \;\; \ddot{\varphi} = d^2\varphi / dt^2, \tag{28}$$

where: $J-$ the reduced moment of inertia of the crane rotating mechanism (e.g. moment of inertia of the mechanism drive or moment of inertia of the machine/crane rotating part), $\varphi -$ twisting angle of the elastic link, $c' -$ is the angular stiffness of the elastic link, $M_p -$ driving torque. This equation (28) can be reduced to the following:

$$\ddot{\varphi} + k^2 \cdot \varphi = M_p / J, \;\; k^2 = c'/J. \tag{29}$$

Integrating the last equation (29) over time for the initial conditions:

$$\varphi|_{t=0} = M_p / c', \;\; \dot{\varphi}|_{t=0} = \omega, \tag{30}$$

where: $\omega -$ is the circular speed of the mechanism, we shall have:

$$\varphi = \frac{\omega}{k} \cdot \sin kt + M_p / c'. \tag{31}$$

The corresponding dynamic load of the crane rotating mechanism (torque from elastic forces) is defined as the product of the elastic deformation φ) of the machine to its angular stiffness c':

$$M_F = \frac{c' \cdot \omega}{k} \cdot \sin kt + M_p \Leftrightarrow M_F = \omega \cdot \sqrt{c' \cdot J} \cdot \sin kt + M_p. \quad (32)$$

The maximum dynamic load is:

$$M_{F\max} = \omega \cdot \sqrt{c' \cdot J} + M_p. \quad (33)$$

This load occurs at points in time t_n :

$$t_n = \left\{ \frac{(-1)^n \cdot \pi / 2 + \pi \cdot n}{k} \right\}, \ \ n \in N. \quad (34)$$

In order to reduce the aforementioned dynamic loads during the time that the locking process lasts τ_l (meaning $t \in [0, \tau_l]$), determine the law of motion $\varphi(t)$, at which the quality criterion for this movement will be met:

$$I = \sqrt{\frac{1}{\tau_l} \cdot \int_0^{\tau_l} (c' \cdot \varphi)^2 \, dt} \Rightarrow \min. \quad (35)$$

The physical meaning of the motion quality criterion (35) is that its implementation minimizes the mean-square dynamic load on the crane mechanism which takes part in the rotary movement during the time that the stopping process lasts.

Using equation (29), criterion (35) can be represented as follows:

$$I = \sqrt{\left\{ \frac{c'}{k^2} \right\} \cdot \frac{1}{\tau_l} \cdot \int_0^{\tau_l} \left(M_p / J - \ddot{\varphi} \right)^2 dt} \Rightarrow \min. \quad (36)$$

A prerequisite for realizing this criterion (provided that $M_p = const$) is the Euler-Poisson equation:

$$\varphi^{(IV)} = 0. \quad (37)$$

We will search for the solution of equation (37) under the following initial/end (so-called terminal) conditions, which make physical sense and follow from the adopted model of motion:

$$\varphi\big|_{t=0} = M_p / c'; \ \ \dot{\varphi}\big|_{t=0} = \omega; \ \ \ddot{\varphi}\big|_{t=0} = 0; \ \ \dot{\varphi}\big|_{t=\tau_l} = 0. \quad (38)$$

Let's feed the solution of equation (37) as a cubic spline on t:

$$\varphi(t) = a_0 + a_1 \cdot t + a_2 \cdot t^2 + a_3 \cdot t^3. \quad (39)$$

Uncertain constants (a_0, a_1, a_2, a_3) are easily found from (39) and the terminal conditions (38). We have:

$$a_0 = M_p / c'; \ \ a_1 = \omega; \ \ a_2 = 0; \ \ a_3 = -\omega / (3 \cdot \tau_l^2). \quad (40)$$

So, the law of motion $\varphi(t),$ at which the quality of movement criterion is realized I (35), (36), is of the form:

$$\varphi(t) = M_p / c' + \omega \cdot t - \frac{\omega \cdot t^3}{3 \cdot \tau_l^2}. \quad (41)$$

Value $M_F(t)$ for the law of motion $\varphi(t)$ (41) takes on the following meanings:

$$M_F(t) = M_p + c' \cdot \omega \cdot t - \frac{c' \cdot \omega \cdot t^3}{3 \cdot \tau_l^2}. \quad (42)$$

So, first of all, the advantage of the law of motion of the system (41) as compared to (31) is that the dynamic loads on the crane system (42) have a smooth character over time rather than oscillating as in (32).

Its maximum value $M_F(t)$ (42) acquires at a point in time $t = \tau_l$ and amounts to:

$$M_{F\max} = M_p + \frac{2}{3} \cdot c' \cdot \omega \cdot \tau_l. \quad (43)$$

Of course, under such circumstances, the value of $M_{F\max}$ decreases compared to (33), because by improving the locking mechanism (e.g. by using mechatronic control systems for this process), it is possible to reduce significantly τ_l $(\tau_l \to 0)$ and it always becomes an attainable condition:

$$\frac{2}{3} \cdot \frac{\tau_l}{k} < 1. \tag{44}$$

That is, if the condition is met (44) $M_{F\max}$ (43) is less than $M_{F\max}$ (33). This is another advantage of the mode of motion (41) in comparison with the mode of motion (31).

It should be noted that in the most general case this is the total reduced moment of inertia of all the moving parts of the crane drive located between the motor and the crane actuator, and M_p – the torque created by the crane motor.

If a safety coupling is installed in the crane drive, then the calculation scheme and all the above defined dependencies remain the same when the coupling is activated, but the value is equal to the total reduced moment of inertia of only those drive elements which are placed between the coupling slave part and the actuator of the crane mechanism (rotation), and the value M_p is equal to the moment at which the coupling slips.

The formulas (31) to (33) can also be used to determine the dynamic loads (on the rope) when the crane hoist is suddenly braked [3]. For this purpose, instead of M_p the static moment from the drag force (load), and for forward motion of the mass m_0 – static load (weight $G_0 = m_0 \cdot g$, $g = 9.81 м/c^2$); in the latter case the dynamic force F is determined according to the formula [3]:

$$F = \frac{c_0 \cdot v}{\beta} \cdot \sin \beta t + G_0, \tag{45}$$

where: $\beta = \sqrt{c_0 / m_0}$; c_0 – the stiffness of the crane rope material; v – lifting speed of the crane (normalized parameter).

To optimize the dynamic loads on the crane rope during sudden braking (braking time is τ_b, meaning $t \in [0, \tau_b]$) The above approach and considerations regarding the lifting mechanism of the crane (namely, to minimize the above loads) can be applied $M_F, M_{F\max}$, but already for F, $F_{\max}$, and make the following replacements:

$$c' \leftrightarrow c_0; \omega \leftrightarrow v; k \leftrightarrow \beta; J \leftrightarrow m_0; M_p \leftrightarrow G_0; \varphi \leftrightarrow Y; \ddot{\varphi} \leftrightarrow \ddot{Y}; \tau_l \leftrightarrow \tau_b. \tag{46}$$

(46) $Y, \ddot{Y}$ – symbolize the upward movement along the rope axis and the acceleration along this direction, respectively.

Conclusions:

1. Improved generalized force criterion for optimizing crane movements (start-up) is justified, minimizing the driving force during the transition process.

2. The law of a crane trolley movement and the law of change in time of its speed, which enable to fulfil the above criterion, have been determined analytically.

3. The law of motion of the crane rotation mechanism (drive of the crane) at its stopping and the law of motion of the cargo at its lifting by the proper mechanism of the crane and abrupt braking at which dynamic loads in the drive and in the crane rope are minimized, respectively, has been established.

3. The results obtained in the work can be further used to clarify and improve the existing engineering methods for calculating the starting modes of overhead cranes both at the stages of their design and in the modes of actual operation.

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ВДОСКОНАЛЕННЯ УЗАГАЛЬНЕНОГО СИЛОВОГО КРИТЕРІЮ ОПТИМІЗАЦІЇ РЕЖИМІВ РУХУ МОСТОВИХ КРАНІВ

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Анотація. При роботі мостових кранів часто спостерігаються маятникові коливання вантажу, що є причиною нерівномірного руху вказаних кранів, їх вантажних візків, навантажень на канати та силові елементи кранів, які, у свою чергу, створюють різноманітні незручності при їх експлуатації, зменшують надійність функціонування як крану у цілому, так і його окремих елементів. Зрозуміло, що всі ці фактори необхідно враховувати при уточнених розрахунках кранів (особливо в режимах їх оптимального (з мінімальною необхідною для цього рушійною силою) пуску/гальмування).

У роботі використана стандартна методика та схема розрахунку маятникових коливань вантажу на канатах мостового крану, які проводяться зазвичай у межах двомасової моделі кранової системи. Здійснені подальші уточнення й вдосконалення вказаної вище методики на основі обґрунтованого узагальненого силового критерію якості руху крану. Отримані залежності, що описують закон руху кранової системи та закон зміни у часі прикладеної рушійної сили на етапах пуску/гальмування, що задовольняють вищезгаданий силовий критерій і забезпечують якісний (плавний) рух системи у період її пуску чи гальмування.

Встановлений закон руху механізму обертання крана (приводу крана) при його стопорінні, а також закон руху вантажу при його підйомі відповідним механізмом крана й різкому гальмуванні, за яких мінімізуються динамічні навантаження у приводі та у канаті крана, відповідно. Отримані у роботі результати можуть бути у подальшому використані задля уточнення й вдосконалення існуючих інженерних методів розрахунку режимів пуску мостових кранів як на етапах їх проектування, так і у режимах реальної експлуатації.

Ключові слова: вдосконалення, узагальнений силовий критерій, оптимізація, режими руху, пуск, гальмування, мостові крани, двомасова модель.

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