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STUDY OF THE DYNAMICS OF THE MOVEMENT MECHANISM OF AN OVERHEAD CRANE AS A COMPLEX ELECTROMECHANICAL SYSTEM

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ДОСЛІДЖЕННЯ ДИНАМІКИ МЕХАНІЗМУ ПЕРЕМІЩЕННЯ МОСТОВОГО КРАНУ ЯК СКЛАДНОЇ ЕЛЕКТРОМЕХАНІЧНОЇ СИСТЕМИ

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The article studies the dynamics of the crane movement mechanism to take measures for their reliable operation and reduce the wear of movement mechanisms. For this, the following assumptions are accepted: transverse motion is not considered, the dynamics of engines and electric drive is not considered; the crane is presented in the form of a seven–mass system, the load suspension is adopted as rigid.

It has been established that account of flexibility of the rope, with which the load is suspended, does not significantly affect (no more than 10%) the values of the largest dynamic loads in elastic connections; therefore, when determining the loads in the elastic connections of the movement mechanisms, it is possible to use a system with an elastic mechanism and a rigid suspension of the load. The processes in the movement mechanisms of load-lifting cranes are considered, using the calculated seven-mass dynamic scheme of structures. During the mechanism operation the oscillations occur in its metal structures and transmission shafts, in addition, the load sways (which, together with a rope of length L_s, forms a pendulum with a moving suspension point).

The graphs show the forces in the longitudinal, transverse beams of the crane and the linear speeds of wheels, with an external force applied to each wheel of 21700 N. All three types of oscillations have different frequencies, which allows them to be considered on simpler two-mass models. But when using synchronous rotation systems (SRS) on the movement mechanism, which are often prone to oscillations, the proposed model is better suited, because it takes into account the interaction of the SRS and the mechanism at different vibration frequencies. With these loads, the mismatch of the path of the sides Δx_c does not exceed 0,025 m, the relative swing of the load x_l is not more than 0,25 m.

The article found that according to the seven-mass model, when a driving force is applied, mechanical oscillations arise in which three frequencies of interaction are observed. For the crane designs discussed in the article, these frequencies are 0,2, 0,95 and 6,4 Hz; to study a more complete picture of the processes, it is necessary to consider the transverse displacement of the crane and the dynamics of electric drive movement.

The parameters of the calculated scheme for output data of the bridge are determined. Based on obtained results the calculated scheme of crane movement mechanism was constructed. The system is converted into a three-axle, the first of which is the rotor of engine, the second - the first gear together with the coupling, and the third - the axle with two gears.

Keywords: *overhead crane, elasticity, design, load suspension, dynamic load, oscillations, force.*

Introduction. Overhead cranes are widely used in mechanical engineering, metallurgical production, nuclear power plants and other industries. Overhead cranes are heavily loaded machines, their daily load reaches 70-80%, the load capacity is 75-85% of the nominal, the actual operating mode of the cranes is often overestimated compared to the standard. Despite the fact that the designs of overhead cranes have been improved for many years, premature failures of the crane system (crane-crane runway) still occur. In addition to operational and technological causes of failures (unscheduled maintenance, violations of operating rules, etc.), design imperfections of crane assemblies can serve as causes. The most characteristic causes and consequences of premature failures include the short service life of crane wheels and crane rails, fatigue failure of end beams, destruction of low-speed shafts of movement mechanisms with mounted gearboxes, loosening and wear of the track gauge, wheel derailment [1].

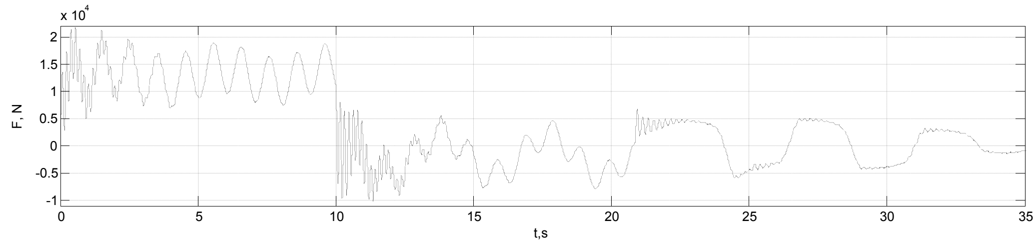
The objective. The role of technological cranes that presented in modern steelmaking can hardly be overestimated, their number in a converter shop can reach several dozen. To take measures for their reliable operation and reduce the wear of movement mechanisms, it is necessary to study the dynamics of the crane movement mechanism, which is an objective of this work.

Research results. The following assumptions are accepted: transverse motion is not considered, the dynamics of engines and electric drive is not considered; the crane is presented in the form of a seven–mass system, the load suspension is adopted as rigid. It has been established [2] that account of flexibility of the rope, with which the load is suspended, does not significantly affect (no more than 10%) the values of the largest dynamic loads in elastic connections; therefore, when determining the loads in the elastic connections of the movement mechanisms, it is possible to use a system with an elastic mechanism and a rigid suspension of the load.

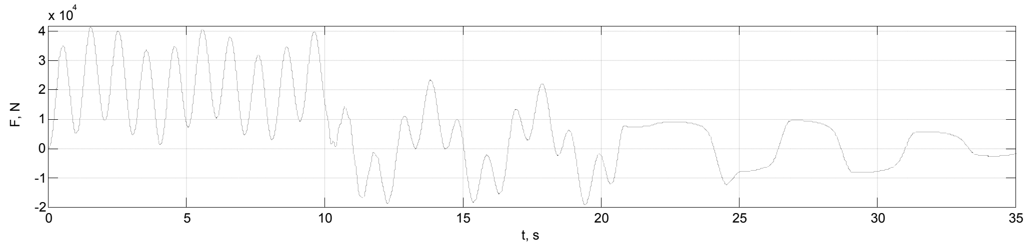
To consider the processes in crane movement mechanism, we will depict the structural elements in the diagram. The calculated seven–mass dynamic scheme of the movement mechanism is shown in article [3]. It contains: two reduced masses m_{1k}, m_{2k} of impellers (with electric motors), two reduced masses of idle wheels m_{3k}, m_{4k}, masses m_1, m_2 of the crane end beams (the mass of the crane cross beam is evenly distributed between m_1 and m_2, the mass of the cart is taken into account in mass m_2), mass of cargo m_3; elastic connections are taken into account by the stiffness coefficients: c_{12} – the transverse beam of the crane, c_1, c_2, c_3, c_4 – halves of the end beams of the crane; L_s – the length of the suspension of the load; F_{12} – elastic force in the cross beam of the crane; F_1, F_2, F_3, F_4 – forces in the end beams of the crane; V_{1k}, V_{2k}, V_{3k}, V_{4k}, V_1, V_2 – linear speeds of the crane wheels and central points 1, 2 of the end beams of the crane bridge; x_l – linear movement of the load relative to the point 2 of the load suspension; $\Delta\varphi$ – skew angle of the transverse beam of the crane.

During the mechanism operation the oscillations occur in its metal structures and transmission shafts (due to the presence of an elastic connection between the masses m_1 and m_2), in addition, the load sways (which, together with a rope of length L_s, forms a pendulum with a moving suspension point). Dynamic loads in elastic connections of movement mechanisms can exceed static loads by 3÷7 or more times [4, 5], and pendulum oscillations of the load cause uneven movement of cranes movement mechanisms or trolleys and create inconvenience during their operation.

To solve the above problems, an appropriate crane control system is needed, which would ensure the synchronism of the movement of the crane longitudinal beams. But often synchronous rotation systems tend to oscillate, so their interaction with the movement mechanism is of great interest. To study such interaction, a model of the movement mechanism is needed, built on the basis of the calculation scheme in one of the applied modeling packages, for example, in MATLAB and SIMULINK. The data of a bucket crane with a lifting capacity of 100 tons of the converter shop of the Alchevsk Steel Works (m_{1k} = m_{2k} = m_{3k} = m_{4k} = = 31250 kg, m_1 =42500 kg, m_2=127500 kg, m_3=100000kg, c_{12} = 2.5*10^6 N*m, c_1 = c_2 = c_3 = c_4 = 20*10^6 N*m, L_s = 5 m, friction coefficient k_{fr} = 0,0062). The graphs show the forces in the longitudinal (Fig. 1,a), transverse (Fig. 1,b) beams of the crane and the linear speeds of wheels 1, 2 (Fig. 1,c), with an external force applied to each wheel of 21700N.



a



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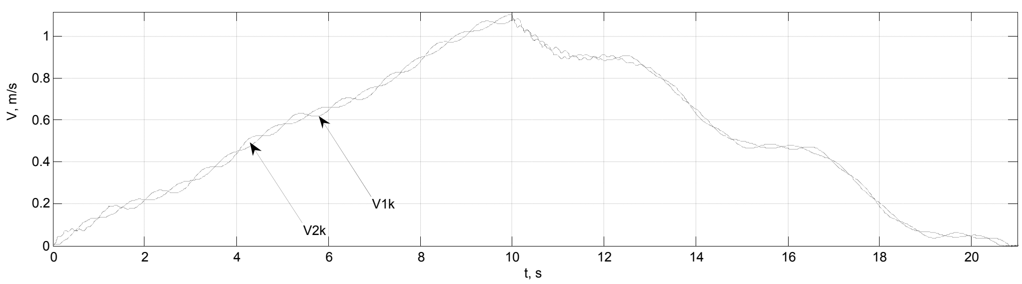

c

Fig. 1. Graphs:
a – elastic force F_1 in the longitudinal beam; b – elastic force F_{12} in the transverse beam; c – wheel speeds V_{k1}, V_{k2}

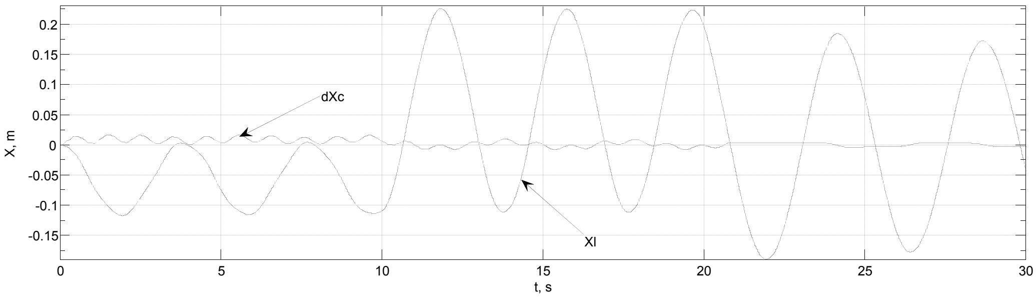

Fig. 2. Graphs of relative displacements of load x_l and sides of the crane Δx_c

It follows from the graphs that torque and velocity oscillations have three components: suspended load oscillations ω_0 = 1,25 rad/s; oscillations due to compliance of the transverse beam ω_0 = 6 rad/s; oscillations due to the compliance of the longitudinal beams ω_0 = 40 rad/s [3].

All three types of oscillations have different frequencies, which allows them to be considered on simpler two-mass models. But when using synchronous rotation systems (SRS) on the movement mechanism, which are often prone to oscillations, the proposed model is better suited, because it takes into account the interaction of the SRS and the mechanism at different vibration frequencies. Figure 2 shows the deviation of the load from the suspension point and the linear mismatch of the path traveled by the sides of the crane. With these loads, the mismatch of the path of the sides Δx_c does not exceed 0,025 m, the relative swing of the load x_l is not more than 0,25 m [3].

The mechanical part of the electromechanical system contains all the connected moving masses: the engine, the transmission and the actuator. A direct idea of the moving masses of the system and the mechanical connections between them is given by the kinematic scheme of the electric drive, one of the variants of which is shown in Fig. 3.

Engine M through the first gear z_1/z_2 drives a long intermediate shaft, which through two gears z_3/z_4 transmits energy to the wheels (runners) of the crane bridge. The total mass of the bridge m_M consists of the masses of the bridge itself, the trolley and the load. The bridge is moving at speed V. Shafts and axes of runners have limited stiffness, which is determined by specific coefficients C_i. In the elements of the movement mechanism of the bridge there are forces of friction (in gears, wheel axles, wheels on rails by flanges), and also forces of resistance of rolling on rails.

The considered scheme clearly shows the position that in the general case the mechanical part of the electric drive is a system of bound masses that moving at different speeds rotating or translationally. Under load, the elements of the system (shafts, supports, gears) are deformed, because the mechanical connections are not absolutely rigid.

The masses and stiffness of the bonds in the kinematic chain of the drive are different. The greatest mass and the least rigidity of connections have a decisive influence on the movement of the system. Therefore, one of the first tasks in the design and study of the electric drive is to compile calculation schemes of the mechanical part, taking into account the possibility of neglecting the elasticity of sufficiently rigid mechanical connections and approximate consideration of the influence of small masses. This is a graphical representation of a system in which the masses (moments of inertia) are represented by rectangles whose areas are proportional to these inertia, and the stiffness of the connections between them is represented by lines whose length is proportional to the pliability. Due to the presence of gears, different elements of the system move at different speeds, so compare directly their moments of inertia J_j, mass m_W, stiffness of connections C_j, C_W, displacement $\Delta\varphi_j$, ΔS_W, etc. is impossible. To compile the calculation scheme of the mechanical part of the electric drive, it is necessary to bring all the parameters of the kinematic circuit to one movement (engine or working body).

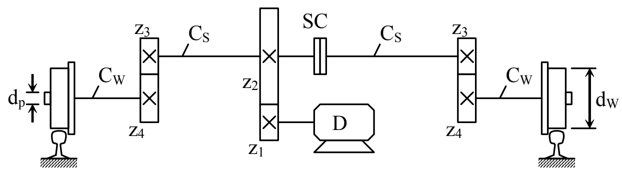

Fig. 3. Kinematic scheme of the movement mechanism of crane bridge

The condition for compliance of the given calculation scheme with the real mechanical system is the fulfillment of the law of conservation of energy (kinetic, potential), as well as the elementary work of all forces acting in the system, the moments of possible displacements.

When designing and researching electric drives, the moments of inertia, mass, stiffness of connections of real elements are usually known, and the forces acting in the system are either set or calculated based on the initial data of the mechanism and the conditions of its technology [6].

Definition of parameters of calculated scheme at such initial data of the bridge:

Bridge mass m_B=2600 kg;

Load mass m_L=20000 kg;

Movement speed $V = 75$ m/min;

Wheel diameter $d_W = 0,4$ m;

Diameter of pins $d_p = 7,5$ cm;

Coefficient of rolling resistance on rails f=0,05 cm;

Coefficient of friction when sliding in the $\mu = 0,07$ pins;

Friction of the flange of the wheels on the rails take into account by the coefficient K_f=0,07;

Gear efficiency $\eta = 0,95$;

Flywheel moments of gears and coupling elements:

$GD_1^2 = 0,2$ kgm^2; $GD_2^2 = 3,4$ kgm^2; $GD_3^2 = 2,4$ kgm^2; $GD_4^2 = 20$ kgm^2 (together with the wheel); $GD_m^2 = 0,4$ kgm^2.

Coefficients of rigidity of:

engine shaft $C_{ES} = 1,8 \cdot 10^6$ Nm;

shaft $C_S = 5,6 \cdot 10^6$ Nm;

wheel axles $C_W = 500 \cdot 10^6$ Nm;

The internal viscous friction in the transmission elements due to the complexity of its receipt is taken into account by the total reduced scattering coefficient ψ_c at a level corresponding to the logarithmic decrement of vibration damping $\lambda_d \cong 0,25$.

The engine has the following data:

$$P_H = 7,5 \text{ kW}; \; n_H = 945 \text{ rpm}; \; \lambda_M = 2,8 ;$$
$$J_o = 0,142 \text{ kgm}^2.$$

Gear ratios

$$i_1 = \frac{Z_2}{Z_1} = \frac{65}{16} = 4,06 ; \; i_2 = i_3 = \frac{Z_4}{Z_3} = \frac{67}{14} = 4,79 .$$

Total gear ratio from the engine shaft to the wheel axles

$$i = i_1 \cdot i_2 = 4,06 \cdot 4,79 = 19,45 .$$

Total efficiency coefficient of gears

$$\eta = \eta_1 \cdot \eta_2 = 0,95 \cdot 0,95 = 0,9 .$$

The total moment of resistance at the loaded crane that brought to engine shaft

$$M_W = r_K (m_B + m_L) \cdot g \cdot (\mu r_p + f) K_f / i\eta =$$
$$= \frac{0,2 \cdot (26000 + 20000) \cdot 9,81 \cdot (0,07 \cdot 0,0375 + 0,0005) \cdot 1,3}{9,45 \cdot 0,9} =$$
$$= 21 \; Nm.$$

Rated and starting torque of the engine

$$M_R = \frac{P_R \cdot 10^3}{0,105 \cdot n_R} = \frac{7500}{0,105 \cdot 945} = 75,8 \text{ Nm};$$
$$M_S = 0,8 \cdot \lambda_M \cdot M_R = 0,8 \cdot 2,8 \cdot 75,8 = 170 \text{ Nm}.$$

The moments of losses in wheel pairs that brought to the engine shaft

$$\Delta M_{WP} = \frac{\Delta M_W}{3} = \frac{(1-\eta) M_W}{3} = \frac{(1-0,9) \cdot 21}{3} = 0,7 \text{ Nm}.$$

Moments of inertia and mass are given.

Gears:

$$J_1 = GD_1^2 / 4 = 0,2 / 4 = 0,05 \text{ kgm}^2;$$
$$J_2 = GD_2^2 / 4 i_1^2 = 3,4 / (4 \cdot 4,06^2) = 0,053 \text{ kgm}^2;$$
$$J_3 = GD_3^2 / 4 i_1^2 = 2,4 / (4 \cdot 4,06^2) = 0,045 \text{ kgm}^2;$$
$$J_4 = GD_4^2 / 4 i^2 = 20 / (4 \cdot 19,45^2) = 0,013 \text{ kgm}^2;$$

Coupling

$$J_m = GD_m^2 / 4 i_1^2 = 0,4 / (4 \cdot 4,06^2) = 0,006 \text{ kgm}^2;$$

Bridge with load

$$J_B = (m_B + m_L) V^2 / \omega_o^2 = (m_B + m_L) r_W^2 / i^2 =$$
$$= (26000 + 2000) \cdot 0,2^2 / 19,45^2 = 4,86 \; kgm^2.$$

The values of the stiffness coefficients of the shafts and wheel axles:

engine shaft $C_{ES} = 1,8 \cdot 10^6$ Nm;

shaft $C_S^{/} = C_S / i_1^2 = 5,6 \cdot 10^6 / 4,06^2 = 0,34 \cdot 10^6$ Nm;

wheel axles $C_W^{/} = C_W / i^2 = 5 \cdot 10^8 / 19,45^2 = 1,32 \cdot 10^6$ Nm.

Based on the obtained results, the calculation scheme of the crane movement mechanism can be presented as shown in Fig. 4, a.

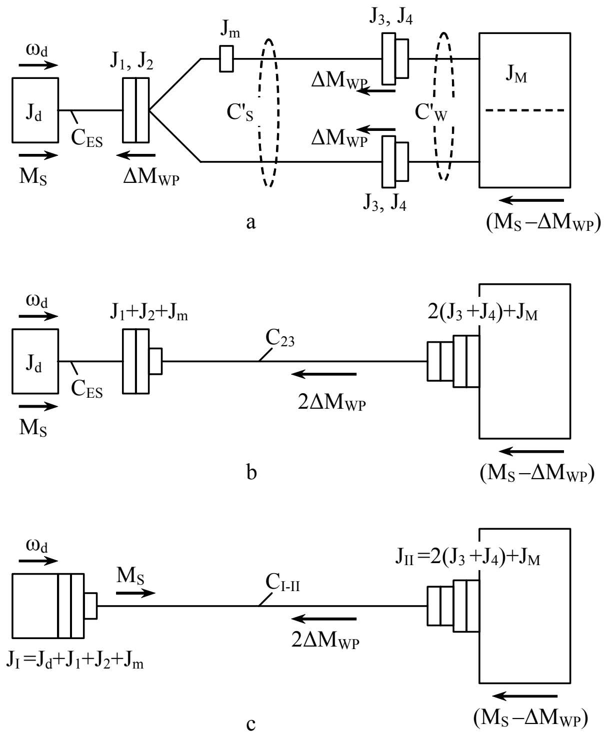

Fig. 4. Calculated scheme of the movement mechanism of overhead crane (according to Fig. 3)

As can be seen from the scheme, significant inertia in the system is the mass of the bridge J_M , the first gear pair $(J_1 + J_2)$ and the engine rotor J_d . Two shafts have significant susceptibility. Therefore $2(J_3 + J_4)$ can be connected with J_M , and J_m – $(J_1 + J_2)$ with, replacing two parallel branches with series-connected $C_S^{/}$ and $C_W^{/}$ on one elastic connection with the stiffness coefficient

$$C_{23} = 2\frac{C_S^{/} \cdot C_W^{/}}{C_S^{/} + C_W^{/}} = 2\frac{0,34 \cdot 10^6 \cdot 1,32 \cdot 10^6}{(0,34+1,32)10^6} = 0,54 \cdot 10^6 \text{ Nm.}$$

The system is converted into a three-axle, the first of which is the rotor of engine, the second - the first gear together with the coupling, and the third - the axle with two gears (Fig. 4, b). The first mass is connected to the second by an elastic bond with $C_{ES} = C_{12}$, the second to the third by a bond with C_{23}. The starting mass of the engine is applied to the first mass, to the second - the moment of losses in the first gear ΔM_{WP}, to the third - the moment $(M_W - \Delta M_{WP})$ of technological resistance and losses in two transfers. Further simplification of the scheme is obtained by combining the first two masses with each other and replacing all elastic bonds with a single one with a stiffness coefficient.

$$C_{I-II} = \frac{C_{12} \cdot C_{23}}{C_{12} + C_{23}} = \frac{1,8 \cdot 10^6 \cdot 0,54 \cdot 10^6}{2,34 \cdot 10^6} = 0,415 \cdot 10^6 \text{ Nm.}$$

Such a two-mass system with inertia $J_I = J_d + J_1 + J_2 + J_m$ and $J_{II} = 2(J_3 + J_4) + J_M$ has an elastic connection between them with stiffness C_{I-II} and scattering coefficients $\psi_c = 0,4$ ($\lambda_d = 0,255$), which reflects all the dissipative forces arising in such a system (Fig. 4, c). A moment is applied to the first mass is $(M_S - \Delta M_{WP})$, and a moment of resistance to the second is $(M_S - \Delta M_{WP})$.

Conclusions. When studying the dynamics of an overhead crane without transverse displacement, it was established that according to the seven-mass model, when a driving force is applied, mechanical oscillations arise in which three frequencies of interaction are observed. For the crane designs discussed in the article, these frequencies are 0,2, 0,95 and 6,4 Hz; to study a more complete picture of the processes, it is necessary to consider the transverse displacement of the crane and the dynamics of movement electric drive.

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Руднєв Є.С., Романченко Ю.А., Лінєвич А.О. Дослідження динаміки механізму переміщення мостового крану як складної електромеханічної системи

У статті проведено дослідження динаміки механізму пересування крана для вжиття заходів щодо їх надійної роботи та зниження зносу механізмів пересування. Для цього прийнято такі припущення: не розглядається поперечний рух, не розглядається динаміка двигунів та електроприводу; кран представлений у вигляді семимасової системи, підвіс вантажу прийнятий жорстким.

Встановлено, що облік гнучкості каната, за допомогою якого підвішений вантаж, несуттєво (не більше 10%) позначається на значеннях найбільших динамічних навантажень у пружних зв'язках; тому при визначенні навантажень в пружних зв'язках механізмів пересування можна використовувати систему з пружним механізмом і жорстким підвісом вантажу.

Розглянуто процеси в механізмах пересування вантажопідіймальних кранів за допомогою розрахункової семи-масової динамічної схеми конструкцій. При роботі механізму виникають коливання в його металоконструкціях і трансмісійних валах, крім того, відбувається розгойдування вантажу (який разом з канатом довжиною утворює маятник з рухомою точкою підвісу).

На графіках були показані зусилля в поздовжній, поперечній балках крана та лінійні швидкості коліс, при прикладених зовнішніх зусиллях 21700 Н до кожного колеса. Всі три види коливань мають різні частоти, що дозволяє розглядати їх на більш простих двомасових моделях. Але при використанні на механізмі пересування систем синхронного обертання (ССО), які часто схильні до коливань, краще підійде запропонована модель, так як вона враховує взаємодію ССО та механізму на різних частотах коливань. При даних навантаженнях неузгодження шляху сторін не перевищує 0,025 м, відносне розгойдування вантажу не більше 0,25 м.

У статті встановлено, що за семи-масовою моделлю при прикладенні рушійного зусилля виникають механічні коливання, в яких спостерігається три частоти взаємодії. Для конструкцій крана, розглянутого у статті, ці частоти становлять 0,2; 0,95 та 6,4 Гц. Для дослідження повнішої картини процесів необхідний розгляд поперечного зміщення крана та динаміки електроприводу переміщення.

Визначені параметри розрахункової схеми при вихідних даних моста. Виходячи з одержаних результатів, побудована розрахункова схема механізму переміщення крану. Система перетворюється на тримасову, першою в якій є ротор двигуна, другою – перша передача разом зі з'єднувальною муфтою, а третьою – міст разом з двома передачами.

Ключові слова: *мостовий кран, пружність, конструкція, підвіс вантажу, динамічне навантаження, коливання, зусилля.*

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