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DETERMINATION OF DYNAMIC LOADS IN THE CRANE LIFTING MECHANISM

ВИЗНАЧЕННЯ ДИНАМІЧНИХ НАВАНТАЖЕНЬ В МЕХАНІЗМІ ПІДІЙМАННЯ КРАНІВ

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Abstract. *To ensure trouble-free operation and improve the reliability of cranes when calculating the structures and components of their working equipment, it is important to take into account dynamic loads that are several times higher than static loads. Due to the fact that the lifting mechanism consists of a large number of elastic elements, the assembly and solution of equations for determining these coefficients is difficult. In order to simplify the equations and these calculations, the given calculation scheme according to which the remaining elements of the mechanism are brought to its first element (engine) is recommended. This allows you to greatly simplify the equation for solving and determine the values of the elasticity factors or stiffness of the elements of the dynamic loads of the crane lifting mechanism.*

The method of determining the coefficients of elasticity and rigidity of elements of dynamic loads, moments of inertia, durations of acceleration and braking of the load lifting mechanism allows to significantly simplify the solution of complex equations and determine their values with sufficient accuracy.

Keywords:. *crane, mechanism, indicator, element, load, moment.*

Introduction.

Loading and unloading works are an integral part of the technological process of construction. Cranes of different types are mainly used to perform these works [1].

Cranes as lifting machines are widely used in construction for the movement of goods and installation of structures.

The scientific and technological progress taking place in all countries of the world strongly requires an increase in productivity, load lifting and an increase in the working speeds of lifting machines, which leads to a reduction in transients, that is, to a decrease in the time of acceleration and braking of machines.

All this leads to an increase in the intensity of the load-lifting machine, causes additional forces on all elements of the machine, received in the technique the name - external dynamic loads [2].

On the other hand, any machine has structural features of its kinematics, deviations in the size of individual parts within the established tolerance, clearance in gear gears and couplings, deformability of the system - all this causes vibrational processes in the machine transmission and refers to phenomena - internal dynamics of the machine.

For safe operation of cranes, it is important to take into account the value of all types of dynamic loads operating when calculating their structures and selecting component elements [3].

Therefore, at present, the actual problem is the development of a technique for determining dynamic loads in the mechanism of lifting the cargo of cranes in case of lack of movement in order to simplify complex calculations.

Dynamic load diagrams of crane load lifting mechanism.

Many researchers have set and solved problems of dynamic analysis and synthesis of modes of movement of the cargo lifting mechanism. In their work, they used systems with concentrated and distributed parameters (crane bridge, rope) [4, 5].

Dependencies were given to determine the inertial loads in the mechanisms of self-propelled cranes when lifting (lowering) the load, turning the boom with the load, lifting (lowering) the boom and combinations of these movements [6, 7].

Dynamic loads were also considered in the rotary movement of the tower crane with a load that causes the spatial movement of the pendulum. For this purpose nonlinear mathematical models of load oscillation during turning motion were formulated, nonlinearity of rocking motion at large angles and nonlinearity of power transmission were taken into account [8, 9].

In all the above works and publications, the method of determining dynamic load indicators using the following schemes is not used.

In these schemes, any mechanism or any machine (Fig. 1) has elements or nodes of massive or rigid bodies that move as a whole in the course of the transition process. Such elements can be considered absolutely rigid bodies, and their entire mass can be concentrated at a point coinciding with the center of weight of this element or node.

Thus, the mechanism or machine consists of "point masses" which include: transported cargo, rotating parts of the engine, brake pulley, drum, gear wheels, etc.

These "point masses" are connected by elastic elements – shafts, ropes and other elements that determine, mainly, the deformation of the mechanism. These elastic elements have a relatively small mass, compared to "point masses", so they can, in the first approximation, be considered massless or absolutely elastic elements.

The elastic elements of the machine under its load are appropriately deformed. The amount of this deformation of the element is taken into account by the coefficient of elasticity or compliance.

The coefficient of elasticity or compliance is defined as the ratio of the value of linear deformation or the angle of twist of this element to the value of the force or torque acting on it:

$$k_l = \frac{y}{P}; \quad k_{cr} = \frac{\varphi}{M}, \tag{1}$$

where P – force causing linear deformation "y"; y – linear deformation; φ – angular deformation; M – moment that causes angular deformation "φ".

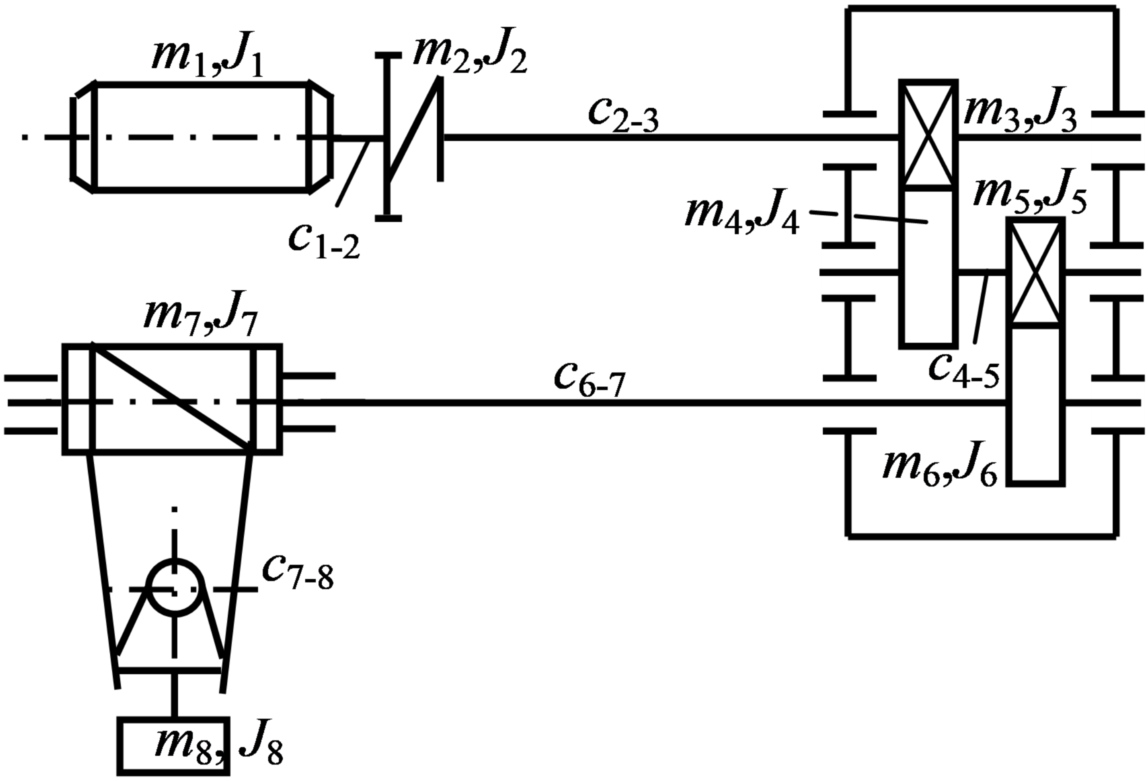


Figure 1 - Elements of dynamic loads of the lifting mechanism

In practice, more often use the value of the inverse coefficient of elasticity, which is called the stiffness coefficient.

The linear stiffness factor or linear stiffness is defined by:

$$C_l = \frac{P}{y}; \tag{2}$$

angular or torsional stiffness:

$$C_{cr} = \frac{M}{\varphi}. \tag{3}$$

Thus, the design scheme can be represented by a number of "point masses" connected by weightless absolutely elastic bonds. Figure 2 shows the design dynamic diagram of the lifting mechanism [7].

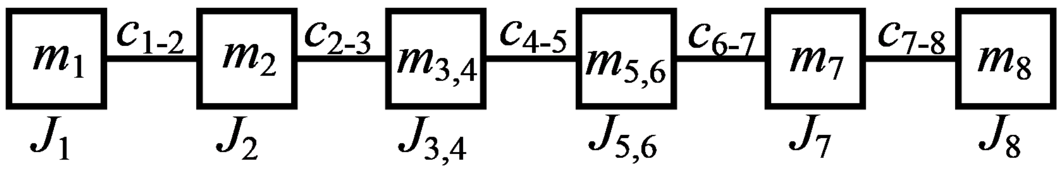


Figure 2 - Estimated dynamic scheme of the lifting mechanism

It follows from the analysis of this design scheme that if you take into account all the elements of the machine in the design scheme, then the scheme is very difficult, and the definition of dynamic loads is an intractable task. Therefore, in order to study dynamic processes in a mechanism or machine, it is advisable to use the so-called given calculation schemes that reflect the actual operation of the mechanism or machine and allow non-difficult decisions to obtain and analyze dynamic loads.

To illustrate the dynamic action of individual masses, depending on the task, they are led to some one elastic link located on one elastic link. Due to the fact that each mechanism has both rotating and progressively moving masses, two design drive schemes are possible.

If the drive is made to a certain shaft of the mechanism, then the given scheme of rotational motion is applied (Fig. 3).

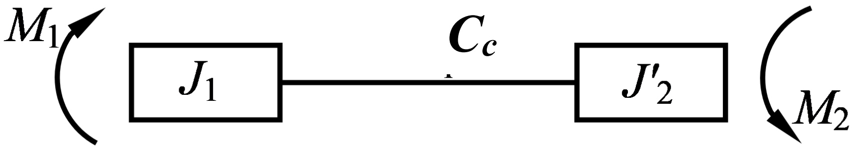


Figure 3 - The scheme of rotational motion is given

If the adjustment is made to the translational moving elastic element - rope, chain, rod, then the given scheme of translational stroke is applied (Fig. 4).

Figure 4 - The scheme of the translational propulsion elastic element is given

Let's say that we bring the weight of the load "m_7," then the mass "m_7" and the stiffness of the suspension "$c_{7\text{-}8}$" remain unchanged, the weight and stiffness of all rotating shafts and parts will have the reduced value:

$$m'_7 = m_7 + \sum_{i=1}^{i=7} m_{i,br} . \qquad (4)$$

When brought to the rotary motion scheme, it is allowed that the drive is carried out to the engine shaft, then the moment of inertia of the engine shaft remains unchanged, the moment of inertia of all the rotating shafts and parts has the reduced value:

$$I = I_1 + I'_2 = I_1 + \sum_{i=2}^{i=8} I_{i,br} , \qquad (5)$$

where I – total moment of inertia, I_1, I'_2 – moments of inertia respectively of the engine shaft and given all rotating parts and shafts of the circuit.

For example, consider the definition of moments in the given scheme of rotational motion:

$$M_d - M_s = I_1\varepsilon + I'_2\,\varepsilon = \varepsilon(I_1 + I'_2) = \frac{\pi n}{30 t_o}(I_1 + I'_2) = \frac{n}{9{,}55 t_o}(I_1 + I'_2), \qquad (6)$$

where M_d, M_s – moments of engine driving forces and static resistance forces respectively; ε – average angular acceleration; t_o – engine acceleration time; n – engine shaft rotation speed.

From where we will find the duration of acceleration t_o and braking of the engine t_b:

$$t_o = \frac{n(I_1 + I'_2)}{9{,}55(M_d \mp M_s)}; \tag{7}$$

$$t_b = \frac{n(I_1 + I'_2)}{9{,}55(M_d \pm M_s)}, \tag{8}$$

where the sign "+" – when braking on the rise and acceleration on the descent, the sign "-" – when braking on the descent and acceleration on the rise.

Write the equation for the second scheme - reduced to a translational move (Fig. 4):

$$P - W = (m_8 + m'_7)\frac{V}{t_o}; \tag{9}$$

where V – cargo lifting speed.

Where will we find the duration of acceleration t_o and braking of the engine t_b:

$$t_o = \frac{(m_8 + m'_7)V}{P \mp W}; \quad t_b = \frac{(m_8 + m'_7)V}{P \pm W}. \tag{10}$$

Thus, the calculated shown rotational motion and translational stroke schemes are the same in terms of both the volume of the calculated work and the consequences of the calculation, although the translational stroke scheme is more apparent than the rotational motion.

The parameters characterizing the inertial properties of the dynamic parts of the mechanisms are the masses at translational motion or moments of inertia (flywheels) at rotational motion. Bringing the moving concentrated masses of the mechanism to a certain shaft is carried out on the basis of the constant kinetic energy of the mechanism E_m before and after the reduction, taking into account the energy losses from the friction forces E_{fr}, which are proportional to the inertial forces:

$$E_{br} = E_m + E_{fr}. \tag{11}$$

On the basis of the law of energy conservation, the moment of inertia of the mechanism masses is brought to the engine shaft during start-up consists of several rotating masses and progressively moving:

$$\frac{I_{br}\omega_1^2}{2} = \frac{I_1\omega_1^2}{2} + \frac{I_2\omega_2^2}{2\eta_1} + \frac{I_3\omega_3^2}{2\eta_1\eta_2} + \ldots + \frac{I_n\omega_n^2}{2\eta_1\eta_2\ldots\eta_n} + \frac{mV^2}{2\eta_m}, \tag{12}$$

where $E = I\frac{\omega^2}{2}$ – kinetic energy of rotating masses; $E' = m\frac{V^2}{2}$ – kinetic energy of translational moving masses.

Dividing by $\frac{\omega_1^2}{2}$ and considering that $\frac{\omega_1}{\omega_2} = u_1$ we get:

$$I_{br} = I_1 + \frac{I_2\omega_2^2}{\omega_1^2\eta_1} + \frac{I_3\omega_3^2}{\omega_1^2\eta_1\eta_2} + \ldots + \frac{I_n\omega_n^2}{\omega_1^2\eta_1\eta_2\ldots\eta_n} + \frac{mV^2}{\omega_1^2\eta_m}; \qquad (13)$$

$$I_{br} = I_1 + I_2\frac{1}{u_1^2\eta_1} + I_3\frac{1}{u_1^2u_2^2\eta_1\eta_2} + \ldots + \frac{m}{\eta_m}\left(\frac{V}{\omega_1}\right)^2, \qquad (14)$$

where I_1 – moment of inertia of all rotating masses of the drive located on the first shaft of the engine; I_2, I_3 etc. – moments of inertia of rotating masses of the drive located on the second, and, accordingly, subsequent shafts of the drive; u_1, u_2 etc. – transfer numbers of the first and subsequent gears; m – weight of progressively moving drive elements (cargo, trolley, crane, etc.); V – speed of mass motion "m"; $\omega_1 = \omega_d$ – angular speed of engine rotor rotation.

Since in these expressions the terms taking into account the moments of inertia of masses on the shafts 2, 3, etc. contain the squares of the ratios in the denominator, the influence of these terms compared to the moment of inertia of masses located on the high-speed shaft of the engine "I_1" is relatively small. Therefore, when determining the reduced moments of inertia of the crane mechanisms, the moments of inertia of masses of slow-moving rotating shafts are taken into account by multiplying the moments of inertia of masses located on the fast shaft by the coefficient "c" equal to:

$$I_{br} = (I_1 + I_2\frac{1}{u_1^2\eta_1} + I_3\frac{1}{u_1^2u_2^2\eta_1\eta_2} + \ldots + I_n\frac{1}{u_1^2u_2^2\ldots u_n^2\eta_1\eta_2\ldots\eta_n}) \times (1{,}1 \div 1{,}2). \qquad (15)$$

Then the equation takes the form:

$$I_{br} = cI_1 + + \frac{m}{\eta_m}\left(\frac{V}{\omega_1}\right)^2 = c\frac{(m_1D^2)}{4} + \frac{m}{\eta_m}\left(\frac{V}{\omega_1}\right)^2, \qquad (16)$$

where (m_1D^2) – is the flywheel moment of all rotating masses located on the engine shaft.

Let's define the given moments of inertia:

for acceleration period - power mode

$$I_{br} = c\frac{(m_1D^2)}{4} + \frac{m}{\eta_m}\left(\frac{V}{\omega_1}\right)^2; \qquad (17)$$

for braking period – braking mode

$$I_{br} = c\frac{(m_1D^2)}{4} + m\eta_m\left(\frac{V}{\omega_1}\right)^2. \qquad (18)$$

Consider the reduced moment of inertia of masses to the drum for the lifting mechanism. If $V_c = \frac{\pi D_d n_d}{u_p u_r 60}$, $\omega_1 = \frac{\pi n_d}{30}$ then

$$V_c = \frac{D_d \omega_1 30}{u_p u_r 60} = \frac{D_d \omega_1}{2 u_p u_r \eta_c}, \tag{19}$$

where D_d – drum diameter; n_d – engine shaft speed; u_p – multiplicity of polyspast; u_r – gear gear ratio; ω_1 – angular speed of engine rotor; η_c – efficiency of load lifting mechanism.

In this case, the given moment of inertia in the power mode will be recorded:

$$I_{br} = c\frac{(m_1 D^2)}{4} + \frac{QR_d}{u_p^2 u_r^2 \eta_c}; \tag{20}$$

in braking mode:

$$I_{br} = c\frac{(m_1 D^2)}{4} + \frac{QR_d \eta_c}{u_p^2 u_r^2}, \tag{21}$$

where Q – weight of cargo.

Thus, the reduction of the moment of inertia of the masses is carried out through the squares of the radius of the drum, the gear ratio and the efficiency factor of the mechanism in the first stage. Moreover, the gear ratio and efficiency in the power mode are in one line, and in the braking mode in different lines.

In dynamic calculations, the forces in the ropes of lifting mechanisms lead to the direction of translational movement of the load. Forming the equation of energy equality, we get the expression of the mass of the load lifting mechanism reduced to this direction:

$$m_{br}\frac{V_c^2}{2} = m_c\frac{V_c^2}{2} + cI_{br}\left(\frac{\omega_1^2 \eta_c}{2}\right), \tag{22}$$

dividing by, "$\frac{V_c^2}{2}$" we get:

$$m_{br} = m_c + cI_{br}\left(\frac{\omega_1}{V_c}\right)^2 \eta_c. \tag{23}$$

Substituting the values "ω_1" and, "V_c" we get the expression:

$$m_{br} = m_c + cI_1\left(\frac{\pi n_d 60 u_r u_p \eta_c}{30\pi D_d n_d}\right)^2 \eta_c = m_1 + cI_1\frac{u_r^2 u_p^2 \eta_c}{R_d^2}. \tag{24}$$

This expression for force mode will be written:

$$m_{br} = Q + c\frac{\left(m_1 D^2\right)}{4}\cdot\frac{u_r^2 u_p^2 \eta_c}{R_d^2}; \tag{25}$$

for braking mode:

$$m_{br} = Q + c\frac{\left(m_1 D^2\right)}{4}\cdot\frac{u_r^2 u_p^2}{R_d^2 \eta_c}. \tag{26}$$

Thus, the weight of the load is also driven through the squares of the radius of the drum, the gear ratio and the efficiency of the mechanism in the first stage.

The main elastic elements of hoisting machines are shafts, ropes, elastic couplings, beams, arrows, etc. The rigidity of gear gears, splice joints and tongue joints is taken into account in the refined calculations.

The problem of stiffening elastic elements arises usually in the case when it is necessary to take into account the elasticity of several elements of the mechanism.

Stiffening is performed so that the potential energy of the reduced system is equal to the potential energy of the real elastic system, taking into account the available friction losses.

Consider a system with stiffness "c_1" and "c_2" (Fig. 5, *a*), which must be brought to stiffness "c_{br}" system (Fig. 5, *b*).

The potential energy of elastic deformation is defined by:
linear

$$P_l = \frac{c_l y^2}{2}; \tag{27}$$

angular

$$P_k = \frac{c_k \varphi^2}{2}, \tag{28}$$

but since the stiffness is linear $c_l = \frac{G_c}{y}$ and the angular $c_k = \frac{M}{\varphi}$, then finally the potential energy will be determined:

$$P_l = \frac{G_c y^2}{2}; \tag{29}$$

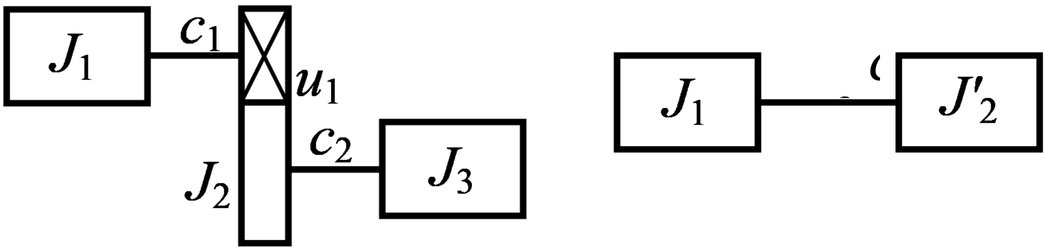
$$P_k = \frac{M\varphi}{2}. \tag{30}$$

In this system, the drive is carried out to the first shaft $M_{br} = M_1$.

The given potential energy is:

$$P_{br} = P_1 + \frac{P_2}{\eta_1}, \tag{31}$$

where η_1 – the efficiency factor taking into account the operation of friction forces of inertial masses in the power mode.



a *b*

Figure 5 - The scheme of reduction of system with rigidity:
***a* – the initial scheme; *b* – the given scheme**

Substituting the value of the potential angular energy, we get:

$$P_{br} = \frac{M_1\varphi_1}{2} + \frac{M_2\varphi_2}{2\eta_1} = \frac{M_1\varphi_1}{2} + \frac{M_1 u_1 \eta_1 \varphi_2}{2\eta_1} = \frac{M_1(\varphi_1 + u_1\varphi_2)}{2}, \quad (32)$$

where φ_1 and φ_2 – angles of shafts twisting under action of moments applied to them; u_1 – gear gear transfer number (Fig. 5).

On the other hand, the potential energy of the given system is:

$$P_{br} = \frac{M_{br}\varphi_{br}}{2} = \frac{M_1\varphi_{br}}{2}. \quad (33)$$

Equating the expressions (32) and (33) to find the reduced twist angle "φ_{br}":

$$\varphi_{br} = \varphi_1 + u_1\varphi_2. \quad (34)$$

Finally, the reduced stiffness will be:

$$c_{br} = \frac{M_1}{\varphi_1 + u_1\varphi_2}. \quad (35)$$

Stiffness "c_2" will be determined:

$$c_2 = \frac{M_1 u_1 \eta_1}{\varphi_2}, \quad (36)$$

from where

$$\varphi_2 = \frac{M_1 u_1 \eta_1}{c_2}. \quad (37)$$

Converting, we get:

$$c_{br} = \frac{c_1\varphi_1}{\varphi_1} + \frac{M_1 u_1 \eta_1}{c_2} u_1 = \frac{c_1 c_2}{c_2 + c_1 u_1^2 \eta_1}, \quad (38)$$

or

$$\frac{1}{c_{br}} = k_{cr1} + k_{cr2} u_1^2 \eta_1, \quad (39)$$

where $k_{cr1} = 1/c_1$, $k_{cr2} = 1/c_2$ – coefficients of steep rigidity or elasticity of shafts.

The reduced elasticity or compliance of the element in the power mode is equal to:

$$k_{cr\,br} = k_{cr1} + k_{cr2} u_1^2 \eta_1; \tag{40}$$

in braking mode:

$$k_{cr\,br} = k_{cr1} + k_{cr2} \frac{u_1^2}{\eta_1}. \tag{41}$$

Thus, the coefficient of stiffness or elasticity, as well as the moment of inertia, is driven by a square of the ratio between the shafts and the efficiency in the first stage.

For example, we find the stiffness of the rope polyspast brought to the shaft of the engine of the load lifting mechanism.

Potential energy of cargo suspension is equal to:

$$P_s = \frac{G_c y}{2\eta}, \tag{42}$$

where G_c – the weight of the moving cargo; y – flexible movement of cargo (deformation of polyspast ropes).

Determine the stiffness of the rope with a length "l":

$$c_k = \frac{P_r F_r}{l}, \tag{43}$$

where P_r – the potential energy of the rope; F_r – the pulling force of the rope, and stiffness of cargo suspension:

$$c_s = G_c c_k. \tag{44}$$

Bringing the load rope suspension stiffness to the shaft of the engine can be found on the condition of equality of the potential energy of the cargo suspension and the reduced system:

$$\frac{G_c y}{2\eta} = \frac{M_1 \varphi_{br}}{2}, \tag{45}$$

from where

$$\varphi_{br} = \frac{G_c y}{M_1 \eta}, \tag{46}$$

then, the load suspension stiffness brought to the engine shaft will be determined:

$$c_{br} = \frac{M_1}{\varphi_{br}} = c_k G_c \left(\frac{R_d}{u_p} \right)^2 \frac{1}{\eta}. \tag{47}$$

Thus, the transformation of the tensile stiffness factor into the torsion stiffness factor is carried out by means of the squares of the drum radius and the multiplicity of the polyspast and the efficiency factor of the suspension in the first stage.

Conclusions.

Application of the above method using the above diagrams allows to determine the dynamic load indicators of the crane load lifting mechanism with a lower difficulty in solving equations and accuracy sufficient for practice. This simplifies calculations and reduces their duration.

In the future, it is necessary to develop programs to perform these calculations using computers.

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Анотація. *Для забезпечення безаварійної роботи і підвищення надійності кранів при розрахунках конструкцій і комплектуючих елементів їх робочого обладнання важливо враховувати динамічні навантаження, які в декілька разів перевищують статичні навантаження. У зв'язку з тим, що механізм підіймання вантажу складається з великої кількості пружних елементів, складання і розв'язання рівнянь для визначення цих коефіцієнтів становиться складним завданням. З метою спрощення рівнянь і цих розрахунків в роботі рекомендована приведена схема розрахунків за якою решта всіх елементів механізму приводяться до першого його елементу (двигуна). Це дозволяє значно спростити рівняння для розв'язання та визначати величини коефіцієнтів пружності або жорсткості елементів динамічних навантажень механізму підіймання вантажу кранів.*

Наведена в роботі методика визначення коефіцієнтів пружності та жорсткості елементів динамічних навантажень, моментів інерції, тривалостей розгону та гальмування *механізму підіймання вантажу дозволяє значно спростити розв'язання складних рівнянь і з достатньою точністю визначати їх величини.*

Ключові слова: *кран, механізм, показник, елемент, навантаження, момент.*

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