ГАЛУЗЕВЕ МАШИНОБУДУВАННЯ

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THEORETICAL RESEARCH OF OSCILLATIONS OF BEARING SYSTEMS PORTAL MACHINES ON PNEUMATIC WHEELS

In the article, theoretical studies of the disturbed motion of oscillation of the carrier system of the gantry machine on pneumatic wheels are carried out. Mathematical models of load formation on the frame of the load-bearing system of the lifting and transporting machine in the longitudinal and transverse vertical planes are presented. Equations of natural frequencies of the system have been obtained, which allow to determine at the stage of designing the elements of the supporting systems of gantry machines the operating modes of the machine in the pre-resonance or post-resonance zone.
Keywords*: frame, supporting system, moment of inertia, structure, coefficient.*

У статті проведені теоретичні дослідження збуреного руху коливання несучої системи портальної машини на пневмоколісному ході. Представлені математичні моделі формування навантаження на раму несучої системи підйомно-транспортної машини у подовжній та поперечній вертикальних площинах. Отримано рівняння власних частот системи, які дозволяють визначити на стадії проєктування елементів несучих систем портальних машин режими роботи машини у дорезонансній або зарезонансній зоні.
Ключові слова*: рама, несуча система, момент інерції, конструкція, коефіцієнт.*

Problem's Formulation

The portal-type structure is characterized by the presence of large building heights, long-dimensional load-bearing elements, and scattered masses, which can cause complex oscillations when moving on uneven roads. Natural for such a system is the existence of resonance zones where the amplitudes of forced oscillations increase sharply, creating prerequisites for the destruction of the load-bearing elements.

In order to create structures capable of stable operation under different load regimes, it is necessary to study a number of dynamic models, to obtain the own dynamic characteristics of the load-bearing systems, to justify such design parameters in which the structure never falls into resonance zones in real operating conditions.

Analysis of recent research and publications

In a number of works, for example, in [1—3], there are unambiguous recommendations for taking into account the oscillations of the gantry lifting and transport machine in the longitudinal plane when overcoming obstacles head-on and its oscillations in the transverse plane when it hits obstacles or uneven paths. In the scientific theoretical studies of Beygul O.O. [4,5] proves the need to take into account torsional vibrations determined by the coefficient of torsional stiffness when designing the frame elements of the bearing systems of gantry machines on pneumatic wheels, but in his works this coefficient is not reduced to sections of the lifting rods.

In the works of V.S. Loveykin [6,7] when carrying out theoretical studies of the loading of frame elements during disturbed movement of the gantry machine, a separate type of load — frequency, which affects the load on the gantry machine frame — was not taken into account.

Formulation of the study purpose

The article solves the problem of obtaining the own dynamic characteristics of portal bearing systems, in which, in real operating conditions, this structure will not fall into resonance zones.

Presenting main material

In the gantry bearing system, vertical loads are formed in the process of disturbed movement of the gantry machine in the longitudinal vertical plane. From the pallet with cargo, they are transferred to the spars with the help of load-lifting rods, then, after passing the crossbar, they are transferred to the racks of the supporting system. Thus, in this calculated case, the crossbar is not loaded, the struts work in compression, which is not deterministic, and the spars are in pure bending conditions under the action of calculated vertical forces. The task is reduced to the design calculation of the strength of the spar as a two-support statically determined beam, prone to pure bending [8].

The perturbed motion of the gantry machine from rest or uniform rectilinear motion is the oscillation of the system near the center of mass and fits into the analogy of an elliptical pendulum.

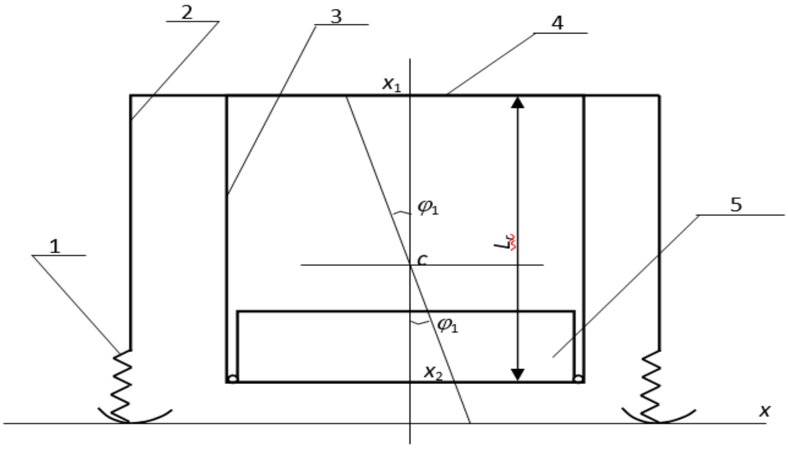

Fig. 1. Calculation diagram of the supporting system of the portal machine: 1— suspension; 2 — rack; 3 — lifting rod; 4 — frame spar; 5 — cargo

In fig. 1 presents the calculation scheme of the supporting system of the gantry machine. We accept the following generalized coordinates: x_1 і φ_1 — the horizontal coordinate of the frame and the angle of rotation of the lifting rods, respectively. Then the cargo coordinates take the following form:

$$x_2 = L_r \sin\varphi_1 - x_1, \tag{1}$$

$$y_2 = L_r\left(1-\cos\varphi_1\right). \tag{2}$$

Expressions of kinetic and potential energies are written as follows:

$$T = \frac{1}{2}\left(m_k + m_r\right)\dot{x}_1^2 + \frac{1}{2}m_r L_r^2 \dot{\varphi}_1^2 - m_r L_r \dot{\varphi}_1 \dot{x}_1 \cos\varphi_1; \tag{3}$$

$$P = m_r g L_r \left(1-\cos\varphi_1\right), \tag{4}$$

where m_k — mass of the supporting structure, kg; m_r — mass of the cargo, kg.

Performing actions according to the scheme of Lagrange equations of the second kind, at zero initial velocities, we obtain the following equations of motion:

$$x_1 = \frac{m_r}{m_k + m_r} L_r \sin\varphi_1, \tag{5}$$

$$\ddot{\varphi}_1 + \frac{m_k + m_r}{m_k}\frac{g}{L_r}\varphi_1 = 0. \tag{6}$$

It follows from equation (5) that the center of mass of the system in turbulent motion oscillates along the vertical axis y. It follows from equation (6) that at $m_k >> m_r$ there is a transition from the analogy of an elliptical pendulum to the analogy of a mathematical pendulum:

$$\ddot{\varphi}_1 + \frac{g}{L_r}\varphi_1 = 0. \quad (7)$$

Let's rewrite equation (6) with a new notation:

$$\ddot{\varphi}_1 + \omega_1^2\varphi_1 = 0, \quad (8)$$

where ω_1 — natural circular frequency, 1/s.

$$\omega_1 = \sqrt{\frac{m_k + m_r}{m_k}\frac{g}{L_r}}. \quad (9)$$

Given the initial conditi φ_{10} and $\dot{\varphi}_{10}$ it is possible to write the solution of the equation (8)

$$\varphi_1 = \varphi_{10}\cos\omega_1 t + \frac{\dot{\varphi}_{10}}{\omega_1}\sin\omega_1 t. \quad (10)$$

In fig. 2 shows the calculation scheme of the supporting system in a disturbed position during oscillations in the longitudinal vertical plane during the hinged fastening of the lifting rods.

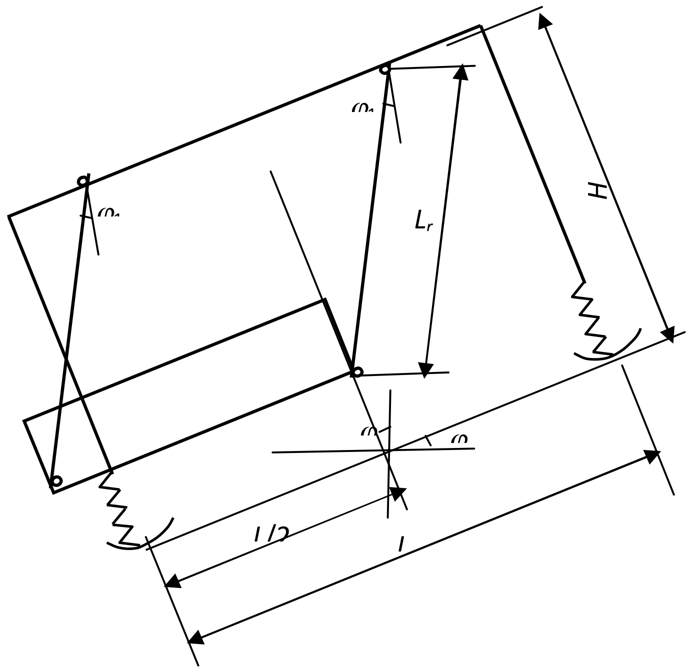

Fig. 2. Calculation diagram of the supporting system of the portal machine in a disturbed position

We accept as generalized coordinates y, φ i φ_1 — vertical coordinate, the angle of rotation of the supporting structure and the angle of rotation of the lifting rods, respectively [9,10]. Then the expressions of kinetic and potential energies take the following form:

$$T = \frac{1}{2}(m_k + m_r)\dot{y}^2 + \frac{1}{2}(J_k + J_r)\dot{\varphi}^2 + \frac{1}{2}m_k(h_{ck}\dot{\varphi} + V)^2 + \frac{1}{2}m_r(H\dot{\varphi} + L_r\dot{\varphi}_1 + V)^2; \quad (11)$$

$$P = C_e\left(y + \frac{L}{2}\varphi - h_1\right)^2 + C_e\left(y - \frac{L}{2}\varphi - h_2\right)^2 + \frac{1}{2}m_r g\left(L_r\varphi_1^2 - H\varphi^2\right), \quad (12)$$

where J_k — the moment of inertia of the supporting structure relative to the center of mass, kg·m^2; J_r — the moment of inertia of the load relative to the center of mass, kg·m^2; h_{cr} — the height of the center of mass of the structure, m; V — the speed of the gantry machine, m/s; C_e — the stiffness coefficient of the suspension is given, N/m; h_1 — the height of bumps under the front suspension, m; h_2 — the height of bumps under the rear suspension, m.

Performing actions according to the scheme of Lagrange's equation of the second kind, we obtain the following equations of disturbed motion [11]:

$$(m_k + m_r)\ddot{y} + 4C_{ey} = 2C_e(h_1 + h_2); \tag{13}$$

$$\left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right)\ddot{\varphi} + \left(C_e L^2 - m_r gH\right)\varphi + m_r L_r H\ddot{\varphi}_1 = \tag{14}$$
$$= -(m_k h_{ck} + m_r H)\dot{V} + C_e L(h_1 - h_2);$$

$$m_r L_r^2 \ddot{\varphi}_1 + m_r g L_r \varphi_1 + m_r L_r H\ddot{\varphi} = -m_r L_r \dot{V}\,. \tag{15}$$

Equation (12) is independent, equations (13) and (14) form a system.

We consider the solution of equation (12), which takes the canonical form after the transformation:

$$\ddot{y} + \omega_2^2 y = \frac{\omega_2^2}{2}(h_1 + h_2); \tag{16}$$

$$\omega_2 = \sqrt{2C_e/(m_k + m_r)}\,, \tag{17}$$

where ω_2 — natural circular frequency of vertical oscillations of the supporting system, 1/s.

Assuming the sinusoidal law of change of inequalities, we obtain the solution of the equation (16)

$$y = y_0 \cos\omega_2 t + \frac{\dot{y}_0}{\omega_2}\sin\omega_2 t +$$
$$+ \frac{h_0}{\left|1 - \dfrac{4\pi^2 V^2}{l^2\omega_2^2}\right|}\left[\sin\left(\frac{2\pi Vt}{l} - \psi_1\right) + \sin\left(\frac{2\pi(Vt - L)}{l} - \psi_2\right)\right], \tag{18}$$

where $y_0, \dot{y}_0$ — initial conditions, m, m/s; h_0 — amplitude value of the average statistical inequality, m; l — the length of the approximating inequality, m; ψ_i — phase angle, rad.

Given that the first two terms in expression (18) describe free oscillations that quickly decay in real conditions, the solution to equation (16) can be written as follows:

$$y = \frac{h_0}{\left|1 - \dfrac{4\pi^2 V^2}{l^2\omega_2^2}\right|}\left[\sin\left(\frac{2\pi Vt}{l} - \psi_1\right) + \sin\left(\frac{2\pi(Vt - L)}{l} - \psi_2\right)\right]. \tag{19}$$

We use the system of equations (14) and (15) to obtain the corresponding frequency equation

$$\left(\left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right)m_r L_r^2 - m_r^2 L_r^2 H^2\right)\left(\omega^2\right)^2 -$$
$$-\left(\left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right)m_r g L_r + \left(C_e L^2 - m_r gH\right)m_r L_r^2\right)\omega^2 +$$
$$+\left(C_e L^2 - m_r gH\right)m_r g L_r = 0. \tag{20}$$

Enter the notation:

$$A_1 = \left(\left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right)m_r L_r^2 - m_r^2 L_r^2 H^2\right); \tag{21}$$

$$B_1 = \left(\left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right)m_r g L_r + \left(C_e L^2 - m_r gH\right)m_r L_r^2\right); \tag{22}$$

$$C_1 = \left(C_e L^2 - m_r gH\right)m_r g L_r\,. \tag{23}$$

Then the solution (20) takes a more compact form:

$$A_1\left(\omega^2\right)^2 - B_1\omega^2 + C_1 = 0\,. \tag{24}$$

This is a well-known quadratic equation relative to ω^2, its solution is written as follows:

$$\omega_3 = \sqrt{\left(B_1 - \sqrt{B_1^2 - 4A_1C_1}\right) \Big/ 2A_1} \; ; \tag{25}$$

$$\omega_4 = \sqrt{\left(B_1 + \sqrt{B_1^2 - 4A_1C_1}\right) \Big/ 2A_1} \; , \tag{26}$$

where ω_3 і ω_4 — natural circular frequencies of the carrier system, 1/s.

Thus, during oscillations in the longitudinal vertical plane, the supporting system of the gantry machine has three degrees of freedom and is characterized by three natural circular frequencies, which are calculated according to formulas (18), (25) and (26) when the lifting rods are hinged.

In fig. 3 presents the calculation scheme of the supporting system in the disturbed position and hinged fastening of the load-lifting rods during transverse oscillations [12,13]. We accept as generalized coordinates y, θ і θ_1 — vertical coordinate, angle of rotation of the frame and angle of rotation of the lifting rods, respectively, then the expressions of kinetic and potential energies take the following form:

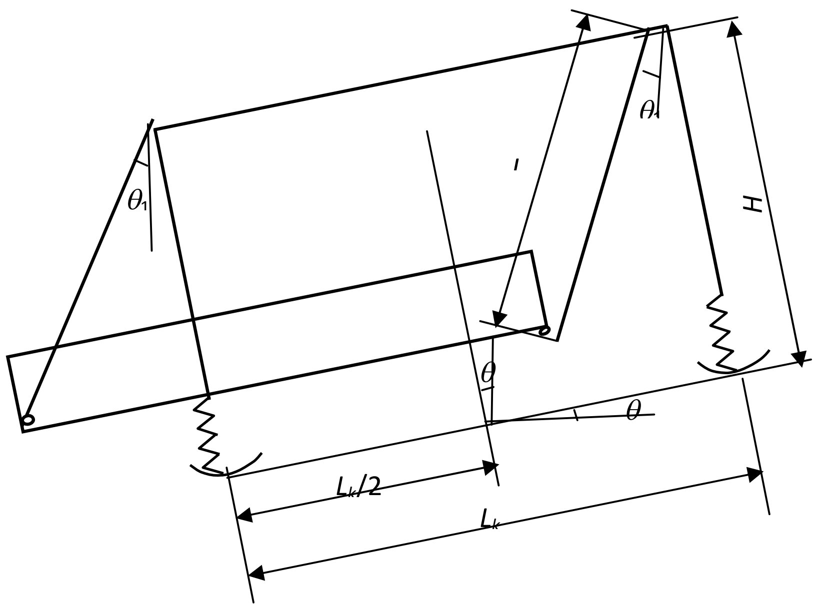

Fig. 3. Calculation scheme of the supporting system of the gantry machine in a disturbed position during transverse oscillations

$$T = \frac{1}{2}(m_k + m_r)\dot{y}^2 + \frac{1}{2}(J_k + J_r)\dot{\varphi}^2 + \frac{1}{2}m_k\left(h_c\dot{\varphi} + V\right)^2 + \frac{1}{2}m_r(\dot{x} + V)^2; \tag{27}$$

$$P = C_e\left(y + \frac{L}{2}\varphi - h_1\right)^2 + C_e\left(y - \frac{L}{2}\varphi - h_2\right)^2 + \frac{1}{2}C_x(x - (H - L_r)\varphi)^2, \tag{28}$$

where $J_{k\theta}$ — moment of inertia of the supporting structure relative to the longitudinal central axis, kg·m^2; $J_{r\theta}$ — the moment of inertia of the load relative to the longitudinal central axis, kg·m^2; L_k — portal machine track, m; h_3 — the height of bumps under the right wheels, m; h_4 — the height of bumps under the left wheels, m.

Performing actions according to the scheme of Lagrange's equation of the second kind, we obtain the following equation of disturbed motion:

$$(m_k + m_r)\ddot{y} + 4C_{ey} = 2C_e\left(h_1 + h_2\right); \tag{29}$$

$$\left(J_k + J_r + m_k h_c^2\right)\ddot{\varphi} + \left(C_e L^2 + C_x (H - L_r)^2\right)\varphi - C_x(H - L_r)x =$$
$$= -m_k h_c \dot{V} + C_e L\left(h_1 - h_2\right); \tag{30}$$

$$m_r \ddot{x} + C_x x - C_x (H - L_r)\varphi = -m_r \dot{V} . \tag{31}$$

Equation (29) does not depend on the other two, the corresponding natural circular frequency is determined by expression (18), and its solution can be written as follows:

$$y = y_0 \cos\omega_2 t + \frac{\dot{y}_0}{\omega_2}\sin\omega_2 t + \frac{2kh_0}{\left|1 - \dfrac{4\pi^2 V^2}{L^2 \omega_2^2}\right|}\sin\left(\frac{2\pi Vt}{L} - \psi_3\right), \tag{32}$$

where k — some coefficient, $k<1$; ψ_3 — phase angle, rad.

We use the system of equations (30) and (31) to obtain the corresponding frequency equation

$$\begin{gathered}\left(\left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right)m_r L_r^2 - m_r^2 L_r^2 H^2\right)\left(\omega^2\right)^2 - \\ -\left(\left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right)m_r g L_r + \left(C_e L_k^2 - m_r g H\right)m_r L_r^2\right)\omega^2 + \\ + \left(C_e L_k^2 - m_r g H\right)m_r g L_r = 0.\end{gathered} \tag{33}$$

Enter the notation:

$$A_3 = \left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right)m_r L_r - m_r^2 L_r^2 H^2 ; \tag{34}$$

$$B_3 = \left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right)m_r g L_r + \left(C_e L_k^2 - m_r g H\right)m_r L_r^2 ; \tag{35}$$

$$C_3 = \left(C_e L_k^2 - m_r g H\right)m_r g L_r . \tag{36}$$

We rewrite equation (33) taking into account the notation (34)—(36):

$$A_3\left(\omega^2\right)^2 - B_3\omega^2 + C_3 = 0 . \tag{37}$$

His decision:

$$\omega_7 = \sqrt{\left(B_3 - \sqrt{B_3^2 - 4A_3C_3}\right)\Big/2A_3} ; \tag{38}$$

$$\omega_8 = \sqrt{\left(B_3 + \sqrt{B_3^2 - 4A_3C_3}\right)\Big/2A_3} , \tag{39}$$

where ω_7 and ω_8 — natural circular frequencies of the carrier system, 1/s.

In this way, the natural circular frequencies of the supporting system of the gantry machine during disturbed motion in the transverse vertical plane in the case of hinged fastening of the lifting rods were obtained. The natural circular frequencies are determined by expressions (17), (38) and (39).

Torsional oscillations of the supporting system are realized with skew-symmetric disturbances in the longitudinal plane. The calculation scheme for hinged fastening of lifting rods is presented in fig. 4.

We accept as generalized coordinates $б_u$ and $б_1$ — angle of rotation of the frame and load, respectively [14]. Then the expressions of kinetic and potential energies take the following form:

$$T = \frac{1}{2}J_{ky}\dot{\alpha}^2 + \frac{1}{2}J_{ry}\dot{\alpha}_1^2 ; \tag{40}$$

$$P = \frac{1}{2}C_z L^2 \alpha^2 + \frac{1}{2}m_r g\frac{L_\delta^2 + L_r^2}{4L_r}\left(\alpha_1 - \alpha\right)^2 , \tag{41}$$

where J_{ky} — moment of inertia of the supporting structure relative to the central vertical axis, kg·m^2; J_{ry} — moment of inertia of the load relative to the central vertical axis, kg·m^2; C_{tz} — the coefficient of lateral stiffness of the wheel tire, N/m.

By performing actions according to the scheme of the Lagrange equation of the second kind, we obtain a system of equations that describe torsional oscillations:

$$J_{ky}\ddot{\alpha} + \left(C_z L^2 + \frac{m_r g\left(L_\delta^2 + L_r^2\right)}{4L_r}\right)\alpha - \frac{m_r g\left(L_\delta^2 + L_r^2\right)}{4L_r}\alpha_1 = 0 ; \tag{42}$$

$$J_{ry}\ddot{\alpha}_1 + \frac{m_r g\left(L_\delta^2 + L_r^2\right)}{4L_r}\alpha_1 - \frac{m_r g\left(L_\delta^2 + L_r^2\right)}{4L_r}\alpha = 0 . \tag{43}$$

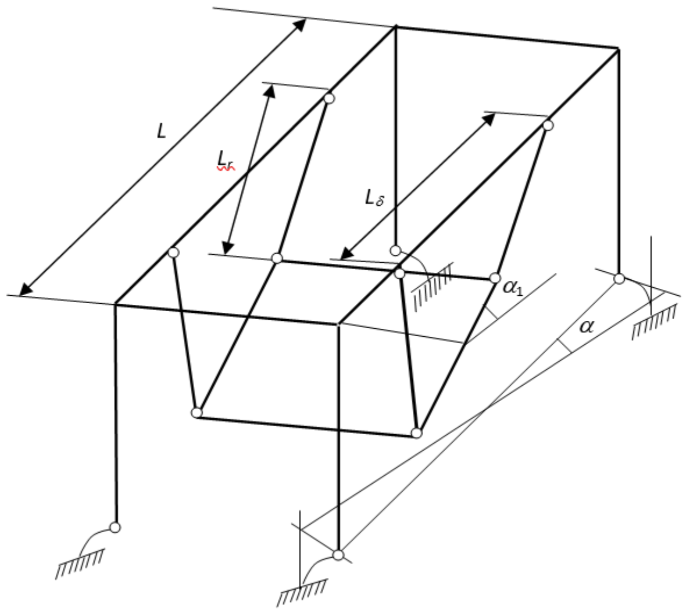

Fig. 4. Calculation scheme of the supporting system of the gantry machine in a disturbed position during torsional oscillations

We use the system of equations (42) and (43) to obtain the corresponding frequency equation

$$J_{ky}J_{ry}\left(\omega^2\right)^2-\left(\left(\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}\right)J_{ky}+\left(C_z L^2+\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}\right)J_{ry}\right)\omega^2+$$

$$+\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}C_z L^2=0. \tag{44}$$

Enter the notation:

$$A_5=J_{ky}J_{ry}; \tag{45}$$

$$B_5=\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}J_{ky}+\left(C_z L^2+\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}\right)J_{ry}; \tag{46}$$

$$C_5=\frac{m_r g\left(L_\delta^2+L_r^2\right)}{4L_r}C_z L^2. \tag{47}$$

Taking into account notations (44)—(46), equation (43) takes the canonical form:

$$A_5\left(\omega^2\right)^2-B_5\omega^2+C_5=0. \tag{48}$$

And its solution is written as follows:

$$\omega_{11}=\sqrt{\left(B_5-\sqrt{B_5^2-4A_5C_5}\right)\Big/2A_5}\ ; \tag{49}$$

$$\omega_{12}=\sqrt{\left(B_5+\sqrt{B_5^2-4A_5C_5}\right)\Big/2A_5}\ , \tag{50}$$

where ω_{11} and ω_{12} — natural circular frequencies of torsional vibrations of the supporting system, 1/s.

In this way, the natural circular frequencies of the torsional vibrations of the supporting system of the gantry machine during the hinged fastening of the lifting rods were obtained. The natural circular frequencies are determined by formulas (49) and (50).

The main source of forced oscillations of the supporting system of the gantry machine is its movement over road irregularities. In the case of axisymmetric frontal disturbances, oscillations in the longitudinal plane are excited, in the case of skew-symmetric disturbances — oscillations in the transverse plane, as well as torsional oscillations. Forced oscillations are characterized by a dynamism coefficient, which in the case of the most complex oscillations and the absence of dissipative forces is reduced to the following expression:

$$K_{\partial} = \frac{1}{\left|1 - \frac{4\pi^2 V^2}{l^2 \omega_i^2}\right|}. \tag{51}$$

where K_{∂} — dynamism factor; ω_i — natural circular frequency, 1/s; v – linear velocity, m/s.

Taking as the calculated dynamics coefficient equal to 1.5, and also taking into account that this dynamics can be realized both in the pre-resonance zone and in the post-resonance zone, we obtain two equations for each natural frequency:

$$1 - \frac{4\pi^2 V_-^2}{l^2 \omega_i^2} = \frac{2}{3}, \tag{52}$$

where e V_- — speed in the pre-resonance zone, m/s;

$$\frac{4\pi^2 V_+^2}{l^2 \omega_i^2} - 1 = \frac{2}{3}, \tag{53}$$

where V_+ — velocity in the resonance zone, m/s.

Equations (52) and (53) can serve as a basis for influencing the eigenfrequencies of the system during design, when the corresponding eigenfrequencies should be obtained at given speeds of movement. At the same time, it should be decided whether the machine will work in the pre-resonance or post-resonance zone.

$$\omega_i = \frac{2\pi V_-}{l}\sqrt{3}; \tag{54}$$

$$\omega_i = \frac{2\pi V_+}{l}\frac{\sqrt{3}}{\sqrt{5}}. \tag{55}$$

Conclusions

The conducted theoretical study of the disturbed movement of the gantry machine allows us to emphasize the need to allocate a separate type of load — frequency, which is especially relevant for gantry layout machines. Equations of natural frequencies are obtained, which largely characterize the specifics of such structures, ways of controlling dynamic systems including portal-type structures are shown.

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ТЕОРЕТИЧНІ ДОСЛІДЖЕННЯ КОЛИВАНЬ НЕСУЧИХ СИСТЕМ ПОРТАЛЬНИХ МАШИН НА ПНЕВМОКОЛІСНОМУ ХОДІ

Реферат

Конструкції портального типу характерна наявність великих будівельних висот, розгалужена просторова стрижньова конструкція, довгомірних несучих елементів, рознесених мас, які при русі по нерівностях доріг можуть чинити складні коливання. Природним для такої системи є існування резонансних зон, де амплітуди вимушених коливань різко зростають, створюючи передумови до руйнування несучих елементів. Такі конструкції зазнають складні просторові коливання при русі по нерівностях доріг в умовах промислових підприємств, деформації при виконанні штатних підйомно-транспортних операцій. У ряді випадків несуча здатність портальних машин визначається не тільки характеристиками міцності, але і жорсткісними характеристиками основних силових елементів.

Для створення конструкцій, здатних стійко працювати на різних режимах навантаження, необхідне вивчення ряду динамічних моделей, отримання власних динамічних характеристик несучих систем, обґрунтування таких конструктивних параметрів, при яких у реальних умовах експлуатації конструкція ніколи не потрапляє у резонансні зони.

Рами транспортних машин в загальному випадку є багатоконтурними плоскопросторовими рамними системами. Вони за своєю природою не можуть мати високу крутильну жорсткість, при русі по нерівностях дороги практично сканують ці нерівності, закручуються як пропелер, зазнаючи значні деформації.

Проведене теоретичне дослідження збуреного руху портальної машини дозволяє наголосити про необхідність виділення окремого виду навантаження — частотного, яке особливо актуальне для машин портальної компоновки. Отримано рівняння власних частот, які значною мірою характеризують специфіку таких конструкцій, показані шляхи управління динамічними системами, що включають конструкції портального типу.

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