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FREQUENCY ASPECT OF THE SELECTION OF PARAMETERS OF ELEMENTS OF PORTAL SYSTEMS OF LIFT AND TRANSPORT MACHINES

ЧАСТОТНИЙ АСПЕКТ ВИБОРУ ПАРАМЕТРІВ ЕЛЕМЕНТІВ ПОРТАЛЬНИХ СИСТЕМ ПІДЙОМНО-ТРАНСПОРТНИХ МАШИН

The article presents theoretical studies of loads on the load-bearing elements of a gantry lifting and transporting machine on pneumatic wheels when it moves along uneven paths. Analytical dependences of natural circular frequencies of the supporting system of the gantry machine under the condition of transverse oscillations during the moment fastening of the lifting rods were obtained. The given technique for determining the range of permissible velocities of the gantry machine during its disturbed motion provides an opportunity to influence the natural frequencies of the system by making changes in the torque characteristics of the main power elements of the structure.
***Keywords**: element, carrier system, circular frequency, structure, machine.*

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. Також необхідно враховувати, що в несучій системі підйомно-транспортних машин портального типу до загальної картини навантаження і деформації додається специфічна особливість, пов'язана з наявністю великої зосередженої маси у вигляді піддону з вантажем на довгих вантажопідйомних штангах, контейнера і т.п.

У статті наведені теоретичні дослідження навантажень несучих елементів портальної підйомно-транспортної машини на пневмоколісному ході при її русі по нерівностям шляхів. Отримані аналітичні залежності власних кругових частот несучої системи порталь-

ної машини за умови поперечних коливаннях при моментному кріпленні вантажопідйомних штанг. Наведена методика визначення діапазону припустимих швидкостей руху портальної машини при її збуреному русі дає можливість впливати на власні частоти системи шляхом внесення змін у жорсткісні характеристики основних силових елементів конструкції.

Розроблена у статті методика по визначенню жорсткісних характеристик силових елементів конструкції дозволяє ще на стадії проєктування портальних підйомно-транспортних машин отримувати оптимальні геометричні параметри елементів несучої системи, а також недопущення при експлуатації портальних машин входу власних кругових частот у резонансну зону, що може привести до руйнування.

***Ключові слова**: елемент, несуча система, кругова частота, конструкція, машина.*

Problem's Formulation

The structural features of the load-bearing systems of portal lifting and transport machines on pneumatic wheels include the fact that during their disturbed movement, the main loads are worked out by longitudinal power elements, i.e. spars, which for this reason are made of closed-type profiles. Transverse power elements during disturbed movement of the gantry machine perceive auxiliary loads, and therefore such frame systems work well for bending and work poorly for twisting loads.

Analysis of recent research and publications

On the basis of research [1], an algorithm for the design calculation of bearing systems of gantry machines was developed, which took into account six calculation cases, as well as some of their combinations. The design methods of gantry machines [2,3] provide recommendations for the geometric parameters of the bearing elements of the gantry frames, but some frequency characteristics, which are based on the types of fastening of power elements, were not taken into account.

Taking into account that the stiffness characteristics, no less than the strength of the elements, affect the load-bearing capacity of the lifting and transport gantry machine, it is necessary to pay special attention to its own frequency loads when performing design work. This was partially taken into account in works [4,5], but no analytical dependences of the movement speeds of the gantry machine were obtained, which make it possible to influence the natural frequencies of the system by making changes in the stiffness characteristics of the main power elements of the structure.

Formulation of the study purpose

Based on the above overview, when designing the frames of gantry machines, the important task of obtaining rational geometric parameters of the load-bearing elements arises. The purpose of this work is to determine the natural circular frequencies that influence the formation of loads on the supporting system and, in turn, constitute an essential stage of the design calculation of the frames of gantry machines.

Presenting main material

Fig. 1 shows the diagram of the support system of the portal lifting and transport machine on pneumatic wheels. This bearing system is a flat-space frame system, the characteristic feature of which is that the main loads are taken by longitudinal or spars. At the same time, auxiliary loads are perceived by transverse force elements. These flat-space frame systems work well for bending, but do not tolerate torsional loads [6,7].

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:

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

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

And expressions of kinetic and potential energies are written as follows:

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

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

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

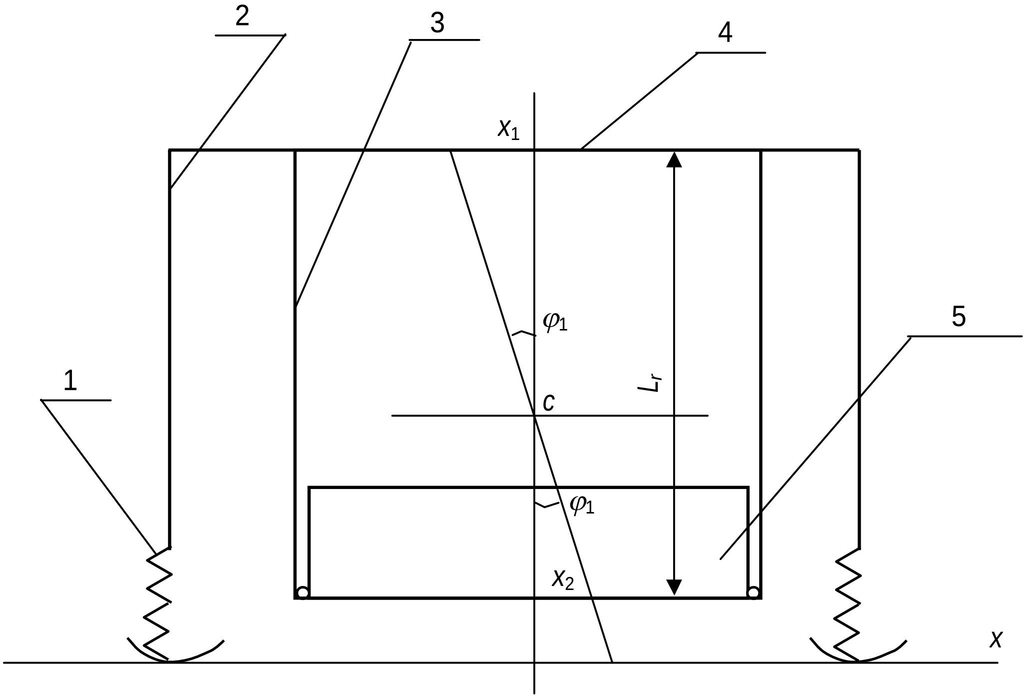

Fig. 1. Calculation diagram of the supporting system of the portal machine: 1 — suspension; 2 — rack; 3 — lifting bar; 4 — frame spar; 5 — cargo

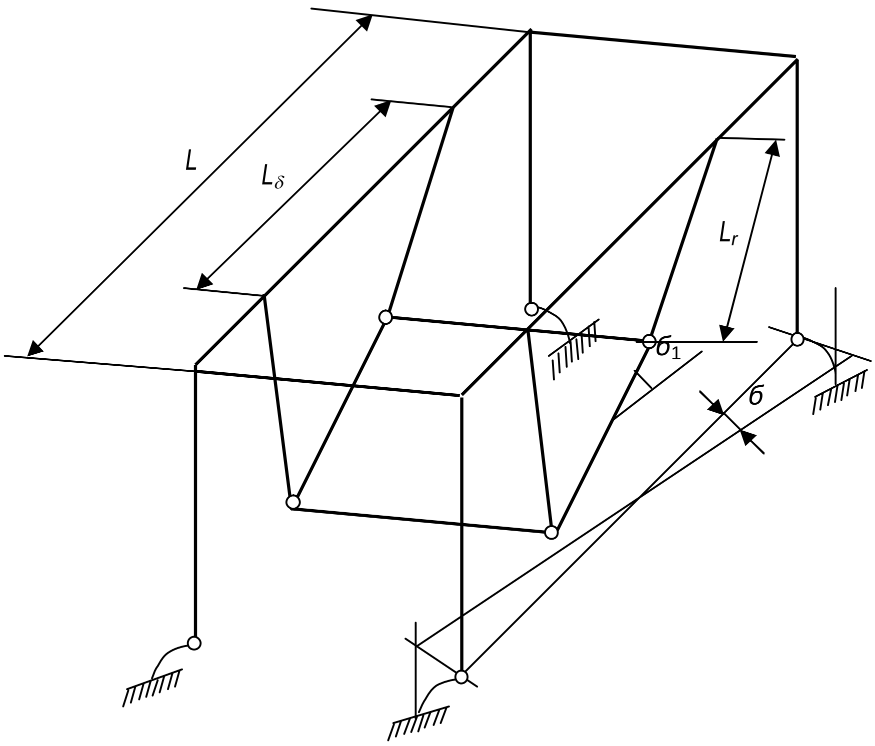

Fig. 2. Calculation scheme of the supporting system of the gantry machine in a disturbed position during torsional oscillations

The calculation scheme for the moment fastening of the lifting rods of the load-bearing system in the disturbed position is presented in fig. 2. As generalized coordinates, we take б and $б_1$ — the angle of rotation of the frame in the horizontal plane and the load, respectively. Then the expressions of kinetic and potential energies take the following form [8]:

$$T = \frac{1}{2} J_{ky} \dot{\alpha}^2 + \frac{1}{2} J_{ry} \dot{\alpha}_1^2; \quad (5)$$

$$P = \frac{1}{2} C_{tz} L^2 \alpha^2 + \frac{1}{2} C_\alpha (\alpha_1 - \alpha)^2, \quad (6)$$

where C_α — coefficient of torsional rigidity, reduced to the area of load-lifting rods, Nm.

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:

$$J_{ky} \ddot{\alpha} + \left(C_{tz} L^2 + C_\alpha \right) \alpha - C_\alpha \alpha_1 = 0; \quad (7)$$

$$J_{ry} \ddot{\alpha}_1 + C_\alpha \alpha_1 - C_\alpha \alpha = 0. \quad (8)$$

We use the system of equations (7) and (8) to obtain the corresponding frequency equation:

$$J_{ky} J_{ry} \left(\omega^2 \right)^2 - \left(C_\alpha J_{ky} + C_\alpha J_{ry} + C_{tz} L^2 J_{ry} \right) \omega^2 + C_\alpha C_{tz} L^2 = 0. \quad (9)$$

Enter the notation:

$$A_6 = J_{ky} J_{ry}; \quad (10)$$

$$B_6 = C_\alpha J_{ky} + C_\alpha J_{ry} + C_{tz} L^2 J_{ry}; \quad (11)$$

$$C_6 = C_\alpha C_{tz} L^2. \quad (12)$$

Taking into account notations (10)—(12), equation (9) takes the canonical form:

$$A_6 \left(\omega^2 \right)^2 - B_6 \omega^2 + C_6 = 0. \quad (13)$$

And its solution:

$$\omega_{13} = \sqrt{\left(B_6 - \sqrt{B_6^2 - 4 A_6 C_6} \right) \Big/ 2 A_6}; \quad (14)$$

$$\omega_{14} = \sqrt{\left(B_6 + \sqrt{B_6^2 - 4 A_6 C_6} \right) \Big/ 2 A_6}, \quad (15)$$

where ω_{13} and ω_{14} — natural circular frequencies of torsional vibrations of the supporting system during moment fastening of lifting rods, 1/s.

In this way, the natural circular frequencies were obtained during the moment fastening of the load-lifting rods. They are determined by formulas (14) and (15).

The main source of forced oscillations of the supporting system of the gantry machine is its movement over uneven roads. 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 [9]:

$$K_\partial = \frac{1}{\left| 1 - \frac{\Omega^2}{\omega_i^2} \right|}, \quad (16)$$

where K_∂ — dynamism factor; Ω — circular frequency of disturbances, 1/s; ω_i — natural circular frequency, 1/s.

When the car is moving:

$$\Omega = \frac{2 \pi V}{l}. \quad (17)$$

We receive:

$$V_- = \frac{l \omega_i}{2 \pi} \frac{1}{\sqrt{3}}; \quad (18)$$

$$V_+ = \frac{l\omega_i}{2\pi}\frac{\sqrt{5}}{\sqrt{3}}. \tag{19}$$

It follows from expressions (18) and (19) that when $V \le V_-$, $V \ge V_+$, the dynamism coefficient will not exceed 1.5. Given that in our case i = 1,...,14, we can write the following condition that characterizes the safe operation of the structure in terms of frequency loading:

$$V_{+\max} \le V \le V_{-\min}, \tag{20}$$

where $V_{+\max}$ — the maximum speed value from among those obtained by formula (19) at i = 1,…,14, m/s; $V_{-\min}$ — the minimum speed value from among those obtained by formula (18) at i = 1,…,14, m/s.

If the frequency load as a calculation case plays a very important role in the formation of loads on the supporting system of the portal type, then the determination of natural frequencies is a core part of the design calculation [10]. For this purpose, 14 natural frequencies are allocated, which are based on different schemes, in particular, on the type of fastening of power elements. For practical purposes, this division is to some extent conditional, because the real fastening conditions are between a rigid embedment and a hinge. The corresponding analytical expressions, written as an algorithm for calculating natural circular frequencies, have the following form:

$$\omega_1 = \sqrt{\frac{(m_k - m_r)g}{m_k L_r}}; \tag{21}$$

$$\omega_2 = \sqrt{\frac{4C_e}{m_k + m_r}}; \tag{22}$$

$$C_e = \frac{C_t C_n}{C_t + C_n}, \tag{23}$$

where C_e — equivalent stiffness of an elastic suspension, N/m; C_t — radial stiffness of the wheel tire, N/m; C_n — stiffness of the elastic element of the suspension, N/m.

$$\omega_3 = \sqrt{\frac{B_1 - \sqrt{B_1^2 - 4A_1C_1}}{2A_1}}; \tag{24}$$

$$\omega_4 = \sqrt{\frac{B_1 + \sqrt{B_1^2 - 4A_1C_1}}{2A_1}}, \tag{25}$$

where

$$A_1 = \left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right) m_r L_r^2 - m_r^2 L_r^2 H^2; \tag{26}$$

$$B_1 = \left(J_k + J_r + m_k h_{ck}^2 + m_r H^2\right) m_r g L_r + \left(C_e L^2 - m_r g H\right) m_r L_r^2; \tag{27}$$

$$C_1 = \left(C_e L^2 - m_r g H\right) m_r g L_r. \tag{28}$$

$$\omega_5 = \sqrt{\frac{B_2 - \sqrt{B_2^2 - 4A_2C_2}}{2A_2}}; \tag{29}$$

$$\omega_6 = \sqrt{\frac{B_2 + \sqrt{B_2^2 - 4A_2C_2}}{2A_2}}, \tag{30}$$

where

$$A_2 = \left(J_k + J_r + m_k h_c^2\right) m_r; \tag{31}$$

$$B_2 = \left(J_k + J_r + m_k h_c^2\right) C_x + \left(C_e L^2 + C_x (H - L_r)^2\right) m_r; \tag{32}$$

$$C_x = 12EJ_{zr}C_{tx} \Big/ \left(3EJ_{zr} + L_r^3 C_{tx}\right); \tag{33}$$

$$C_2 = C_x C_e L^2. \tag{34}$$

$$\omega_7 = \sqrt{\frac{B_3 - \sqrt{B_3^2 - 4A_3C_3}}{2A_3}}\,; \tag{35}$$

$$\omega_8 = \sqrt{\frac{B_3 + \sqrt{B_3^2 - 4A_3C_3}}{2A_3}}\,, \tag{36}$$

where

$$A_3 = \left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right) m_r L_r^2 - m_r^2 L_r^2 H^2\,; \tag{37}$$

$$B_3 = \left(J_{k\theta} + J_{r\theta} + m_k h_{ck}^2 + m_r H^2\right) m_r g L_r + \left(C_e L_k^2 - m_r g H\right) m_r L_r^2\,; \tag{38}$$

$$C_3 = \left(C_e L_k^2 - m_r g H\right) m_r g L_r\,. \tag{39}$$

$$\omega_9 = \sqrt{\frac{B_4 - \sqrt{B_4^2 - 4A_4C_4}}{2A_4}}\,; \tag{40}$$

$$\omega_{10} = \sqrt{\frac{B_4 + \sqrt{B_4^2 - 4A_4C_4}}{2A_4}}\,, \tag{41}$$

where

$$A_4 = \left(J_{k\theta} + J_{r\theta} + m_k h_c^2\right) m_r\,; \tag{42}$$

$$B_4 = \left(J_{k\theta} + J_{r\theta} + m_k h_c^2\right) C_z + \left(C_e L_k^2 + C_z (H - L_r)^2\right) m_r\,, \tag{43}$$

where

$$C_z = 12EJ_{xr}C_{tz} \Big/ \left(3EJ_{xr} + L_r^3 C_{tz}\left(1 + \frac{J_{xr}}{J_{x\delta}}\frac{3{,}9(L - L_\delta)}{L_r}\right)\right); \tag{44}$$

$$C_4 = C_z C_e L_k^2\,. \tag{45}$$

$$\omega_{11} = \sqrt{\frac{B_5 - \sqrt{B_5^2 - 4A_5C_5}}{2A_5}}\,; \tag{46}$$

$$\omega_{12} = \sqrt{\frac{B_5 + \sqrt{B_5^2 - 4A_5C_5}}{2A_5}}\,, \tag{47}$$

where

$$A_5 = J_{ky} J_{ry}\,; \tag{48}$$

$$B_5 = m_r g\left(L_\delta^2 + L_r^2\right) J_{ky} \big/ 4L_r + \left(C_{tz} L^2 + m_r g\left(L_\delta^2 + L_r^2\right) \big/ 4L_r\right) J_{ry}\,; \tag{49}$$

$$C_5 = m_r g\left(L_\delta^2 + L_r^2\right) C_{tz} L^2 \big/ 4L_r\,; \tag{50}$$

$$\omega_{13} = \sqrt{\frac{B_6 - \sqrt{B_6^2 - 4A_6C_6}}{2A_6}}\,; \tag{51}$$

$$\omega_{14} = \sqrt{\frac{B_6 + \sqrt{B_6^2 - 4A_6C_6}}{2A_6}}\,, \tag{52}$$

where

$$A_6 = J_{ky} J_{ry}\,; \tag{53}$$

$$B_6 = C_\alpha J_{ky} + C_\alpha J_{ry} + C_{tz} L^2 J_{ry}\,, \tag{54}$$

where

$$C_{\alpha} = \frac{3EJ_{xr}L_{\delta}^{2}}{L_{r}^{3}} \Bigg/ \left(1 + \frac{J_{xr}}{J_{k\delta}} \cdot \frac{3{,}9L_{\delta}\left(L - L_{\delta}\right)}{LL_{r}}\right); \tag{55}$$

$$C_{6} = C_{\alpha}C_{tz}L^{2}. \tag{56}$$

Next, we present the method of determining the parameters of portal machines according to the criterion frequency load.

With a known road profile, the ranges of permissible traffic speeds are determined from the system of irregularities:

$$\left.\begin{array}{l} \dfrac{l\omega_{1}}{6\pi}\sqrt{15} \le V \le \dfrac{l\omega_{1}}{6\pi}\sqrt{3}; \\ \dfrac{l\omega_{2}}{6\pi}\sqrt{15} \le V \le \dfrac{l\omega_{2}}{6\pi}\sqrt{3}; \\ \dots\dots\dots\dots\dots\dots\dots \\ \dfrac{l\omega_{14}}{6\pi}\sqrt{15} \le V \le \dfrac{l\omega_{14}}{6\pi}\sqrt{3}, \end{array}\right\} \tag{57}$$

where l — length of the approximating sinusoid, м; ω_i — natural circular frequency (i = 1,...,14), 1/s; V — the speed of the gantry machine, m/s.

In particular, if the obtained fields of permissible movement speeds turn out to be unacceptable for a number of reasons, then there is an opportunity to influence the natural frequencies of the system by introducing changes in the stiffness characteristics of the main power elements of the structure.

Conclusions

The frequency aspect of the selection of the parameters of the elements of the load-bearing systems involves the determination of their own circular frequencies, which are based on various types of fastening of the power elements. Analytical dependences of fourteen natural circular frequencies obtained in the work, as well as the field of permissible speeds of movement allow to make changes in the stiffness characteristics of the power elements of the structure at the stage of their design. The above makes it possible to reduce the material capacity of gantry machine frames without reducing their load-bearing properties and strength.

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