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THE INFLUENCE OF STRUCTURAL RIGIDITY ON ITS DYNAMIC CHARACTERISTICS OF PORTAL BEARING SYSTEMS

ВПЛИВ КОНСТРУКТИВНОЇ ЖОРСТКОСТІ НА ВЛАСНІ ДИНАМІЧНІ ХАРАКТЕРИСТИКИ ПОРТАЛЬНИХ НЕСУЧИХ СИСТЕМ

For traditional pneumatic vehicles, the requirement of low torsional stiffness of the supporting systems fits into the static load pattern, dictated by the reduction of internal force factors, is realized by the use of ladder-type frames with open-profile power elements, then for gantry machines, the influence of torsional stiffness should be taken into account stiffness on its own dynamic characteristics, which determine the load pattern of the load-bearing systems when moving on uneven roads. The article theoretically investigates the effect of structural rigidity on the dynamic characteristics of portal bearing systems. It has been proved that the dynamic stiffness to a certain extent depends not only on the geometrical parameters of the elements of the supporting system, but also on the external disturbances observed during the dynamic movement of the gantry lifting and transporting machine.
***Keywords**: frame, supporting system, moment of inertia, structural rigidity.*

У статті теоретично досліджено вплив конструктивної жорсткості на власні динамічні характеристики портальних несучих систем. Доведено, що динамічна жорсткість певною мірою залежить не тільки від геометричних параметрів елементів несучої системи, а також від зовнішніх збурень які спостерігаються при динамічному русі портальної підйомно-транспортної машини.

Науково-технічний прогрес у промисловості тісно пов'язаний з розробкою та впровадженням нових технологічних ліній, для яких обов'язковими є транспортні операції. В умовах промислових цехів, що реконструюються і знов будуються, необхідні компактні, маневрені транспортні машини, які не вимагають великих виробничих площ. Цим вимогам цілком відповідають портальні машини на пневмоколісному ході, які завдяки своїй компоновці пристосовані для транспортування контейнерів, піддонів, термосів і інших вантажів. Виходячи з аналізу конструкцій підйомно-транспортних портальних машин, слід зазначити наступні умови їх експлуатації: необхідність маневрування, екстрене гальмування з вантажем на максимальній висоті підйому, бічний крен портальної машини, динамічне навантаження при фронтальному наїзді передніх коліс на дорожні перешкоди, кососиметричне подолання перешкод, рух по нерівностях на горизонтальній ділянці шляху.
***Ключові слова**: рама, несуча система, момент інерції, конструктивна жорсткість.*

Problem’s Formulation

The analysis of the methods of designing the supporting systems of gantry machines showed the following. The method of similarity can be used when creating a layout of power elements of subtank-transport gantry machines. The method of analogues allows the use of typical universities and aggregates. The analytical method of calculation and design involves the development of a mathematical model of the formation of external loads, based on the characteristic load modes of the main power elements of lifting and transport gantry machines.

Taking into account the structural transverse stiffness of this bearing system is an urgent task when designing the geometric parameters of the elements of the supporting system. The article deals with the solution of this problem, namely, conducting theoretical studies to determine the dynamic stiffness indicators of the elements of the load-bearing system of the gantry lifting and transporting machine.

Analysis of recent research and publications

The analysis of the methods of designing the load-bearing systems of gantry machines indicates a methodologically correct approach to this problem in most cases when a certain mathematical model of the formation of external loads is adopted, based on the characteristic load modes of the main power elements.

Thus, when calculating the load-bearing systems of portal machines models 7801 and 7803 in the Main Special Design Bureau of Truck Loaders (Lviv), the following characteristic load modes are accepted: 1) static load; 2) emergency braking with the load at maximum height; 3) turning with the load at maximum height. These characteristic modes are established as the basis of design calculations for the strength of all power elements of the supporting systems.

One of the main criteria for the structural perfection of lifting and transport machines is their metal content. Gantry machines on pneumatic wheels only in recent years began to be widely used in the technological lines of lifting and transport complexes; they have been studied very little, therefore the work of Lepetova G.L. [1], being directed to the scientifically based selection of parameters of gantry bearing systems, for the first time outlined the ways of creating rational structures of gantry machines in terms of metal consumption. In the works of Beygul O.A. [2] studied the effect of torsional stiffness of bearing systems on the formation of external loads. Since frames with low torsional stiffness are not significantly loaded by torques when moving over uneven road surfaces, the total load on such frames turns out to be less than on frames with high torsional stiffness. Reducing the load naturally affects the reduction of the metal content of the supporting systems. Experts from many branches of mechanical engineering deal with issues of metal consumption of machines. For the load-bearing metal structures of gantry lifting and transport machines can serve as an analogue of the frame of cars, the study of which is devoted to a number of works [3,4].

Formulation of the study purpose

The article solves the problem of determining the coefficient of transverse structural rigidity of the supporting system of the gantry lift truck when it moves over road bumps, skew-symmetric overcoming of obstacles and dynamic load when the front wheels collide with road obstacles.

Presenting main material

The bearing systems of lifting and transporting gantry machines are characterized by the presence of a branched spatial rod structure with a large building height, spread masses. Such structures are subject to complex spatial fluctuations when moving over uneven roads in the conditions of industrial enterprises, deformation during standard lifting and transport operations. In some cases, the carrying capacity of gantry machines is determined not only by the strength characteristics, but also by the stiffness characteristics of the main power elements.

If for traditional pneumatic vehicles the requirement of low torsional rigidity of the supporting systems fits into the static pattern of the load, dictated by the reduction of internal force factors, is realized by the use of ladder-type frames with power elements of an open profile, then for gantry machines the influence of torsional rigidity should be taken into account on their own dynamic characteristics, which determine the load pattern of the load-bearing systems when moving on uneven roads.

The behavior of load-bearing systems in the presence of harmonic disturbances is characterized by dynamic stiffness, which depends not only on the parameters of the system, but also on external disturbances, in particular, it decreases around resonance zones and drops to zero at the onset of resonance. This is especially relevant for spatial rod systems, which include portal bearing systems.

And, if in the longitudinal vertical plane it is possible to ensure sufficient stiffness both static and dynamic by structural means, for example, by introducing guide rods and braces, then in the transverse vertical plane it is difficult to introduce additional force elements that increase the stiffness of the system within the framework of a rational design. It remains to look for reserves to increase the stiffness of the system in the power elements of the supporting structure itself. This path goes through obtaining dependencies between the natural circular frequencies and the structural rigidity of the portal bearing system in the transverse vertical plane. Therefore, we will consider the differential equations of free oscillations of the gantry machine in the transverse plane, where these dependencies come from as a result of the solution of the frequency equation.

When deriving differential equations, some structural and structural features of lifting and transport gantry machines used in inter-shop transportation at ferrous metallurgy plants are taken into account. Portal machines, in contrast to portal cars considered from the general theoretical standpoint in works [5] and [6,7], have a low center of mass of the payload and long lifting rods, when the disturbed movement of the pallet in the transverse vertical plane is not causes a noticeable redistribution of the vertical reactions of the supports. The resulting transverse oscillations are determined not by the vertical, but by the transverse horizontal stiffness of the elastic structural elements and are shear in nature [8].

In fig. 1 presents the calculation scheme of the supporting system of the gantry machine. The following designations are adopted: 1 — spar of the frame of the supporting system; 2 — pallet with payload; 3 — load-lifting bar of the supporting system; 4 — crossbar of the frame; 5 — rack of the supporting system; 6 — tire of a pneumatic wheel.

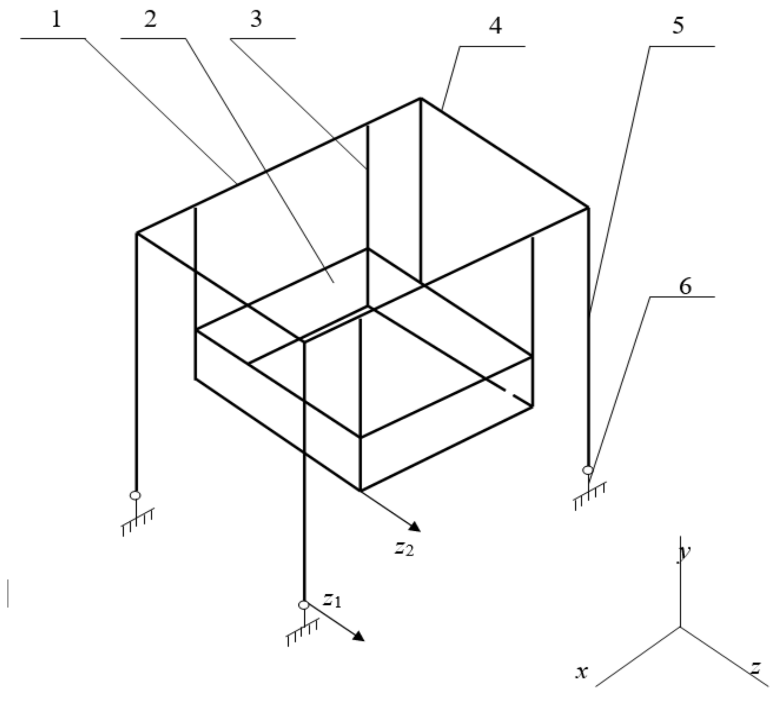

Fig 1. Calculation scheme of the supporting system of the metallurgical portal machine

The equations of transverse oscillations are derived in the form of a Lagrange equation of the second kind [9]. For this, the expressions of kinetic and potential energy of the system are recorded.

The kinetic energy is equal to:

$$T = \frac{m_\kappa \dot{z}_1^2}{2} + \frac{m_r \dot{z}_2^2}{2}, \tag{1}$$

where m_κ mass of the supporting structure, kg; $m_г$ — weight of the pallet with payload, kg; $\dot{z}_1$ — generalized speed of the supporting structure, m/s; $\dot{z}_2$ — generalized speed of the pallet with payload, m/s.

The potential energy is defined by the expression

$$\Pi = \frac{4C_{tz}z_1^2}{2} + \frac{C_{\kappa z}(z_2 - z_1)^2}{2}, \quad (2)$$

where C_{tz} — coefficient of lateral stiffness of the wheel tire, N/m; $C_{\kappa z}$ — coefficient of transverse structural rigidity of the supporting system, N/m; z_1 — generalized wheel coordinate, m; z_2 — generalized coordinate of the pallet, m.

Next, we perform actions according to the operators of the Lagrange equation of the second kind and, combining the obtained expressions, write down a system of differential equations:

$$\left.\begin{aligned} m_{\text{к}}\ddot{z}_1 + 4C_{tz}z_1 - C_{\text{к}z}(z_2 - z_1) = 0, \\ m_r\ddot{z}_2 + C_{\text{к}z}(z_2 - z_1) = 0. \end{aligned}\right\} \quad (3)$$

The system of differential equations (3.3) is solved in the form

$$\left.\begin{aligned} z_1 = A_1\sin(\omega t + \phi), \\ z_2 = A_2\sin(\omega t + \phi), \end{aligned}\right\} \quad (4)$$

where A_1 — amplitude of oscillations along the first generalized coordinate, m; A_2 — amplitude of oscillations along the second generalized coordinate, m; ω — circular frequency of oscillations, 1/s; t — time, s; φ — phase angle, rad.

After substituting expressions (4) into differential equations (3), we obtain a homogeneous system of equations for amplitudes A_1 and A_2

$$\left.\begin{aligned} (4C_{tz} + C_{\text{к}z} - m_{\text{к}}\omega^2)\mathrm{A}_1 - C_{\text{к}z}\mathrm{A}_2 = 0, \\ -C_{\text{к}z}\mathrm{A}_1 + (C_{\text{к}z} - m_r\omega^2)\mathrm{A}_2 = 0. \end{aligned}\right\} \quad (5)$$

System (5) has a non-trivial solution if its determinant is zero:

$$\begin{vmatrix} (4\mathrm{C}_{tz} + C_{\text{к}z} - m_{\text{к}}\omega^2) & -\mathrm{C}_{\text{к}z} \\ -\mathrm{C}_{\text{к}z} & (C_{\text{к}z} - m_r\omega^2) \end{vmatrix} = 0. \quad (6)$$

Expanding the determinant in equation (3.6), we obtain the frequency equation

$$m_{\text{к}}m_r(\omega^2)^2 - (C_{\text{к}z}m_{\text{к}} + 4\mathrm{C}_{\text{ш}z}m_r + C_{\text{к}z}m_r)\omega^2 + 4\mathrm{C}_{\text{ш}z}C_{\text{к}z} = 0. \quad (7)$$

Two natural circular frequencies of transverse oscillations are determined from equation (7).

$$\omega_{1,2}^2 = \frac{C_{\text{к}z}\left[\left(m_{\text{к}} + m_r + \frac{4\mathrm{C}_{tz}}{C_{\text{к}z}}m_r\right) \mp \sqrt{\left(m_{\text{к}} + m_r + \frac{4\mathrm{C}_{tz}}{C_{\text{к}z}}m_r\right)^2 - 4m_{\text{к}}m_r\frac{4\mathrm{C}_{tz}}{C_{\text{к}z}}}\right]}{2m_{\text{к}}m_r}. \quad (8)$$

The complete solution of the system of differential equations (3) describing the free transverse oscillations of the supporting system of the gantry machine has the following form:

$$\left.\begin{aligned} z_1 = A_{11}\sin(\omega_1 t + \varphi_1) + A_{12}\sin(\omega_2 t + \varphi_2); \\ z_2 = A_{21}\sin(\omega_1 t + \varphi_1) + A_{22}\sin(\omega_2 t + \varphi_2), \end{aligned}\right\} \quad (9)$$

where the constants $A_{j\kappa}$ and φ_j are determined from the initial conditions.

As follows from expression (8), the natural circular frequencies of the carrier system of the gantry machine significantly depend on the coefficient of transverse structural stiffness S_{kz} of the carrier system, which allows to shift the natural circular frequencies to the pre-resonance zone according to the dynamic stiffness index even at the design stage.

Conclusions

The article solves the theoretical problem of the effect of structural stiffness on the dynamic characteristics of portal bearing systems of lifting and transporting machines in the form of the obtained coefficient of transverse structural stiffness of the bearing system, which in turn allows shifting the inherent circular frequencies to the pre-resonance zone even at the design stage.

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