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REDUCTION OF THE METAL CONTENT OF FRAMES OF BEARING SYSTEMS OF PNEUMATICS-WHEEL-RUNNING PORTALE MATS

ЗМЕНШЕННЯ МЕТАЛОМІСТКОСТІ РАМ НЕСУЧИХ СИСТЕМ ПОРТАЛЬНИХ МАШИН НА ПНЕВМОКОЛІСНОМУ ХОДІ

Portal lifting and transport machines are characterized by a large building height of the structure and the presence of scattered masses.

The main thing about the described bearing systems is that they fit into the calculation scheme, which includes a rectangular frame and four racks equipped with different numbers of wheels.

The article proves that the torsional rigidity of frames from closed profiles is two orders of magnitude higher than similar frames from open profiles, that is, it can be taken into account that torsional loads are perceived by frames from closed profiles.

The spars of the supporting systems of portal machines must be designed from closed-type profiles, which was proven by the authors, and the cross members of the supporting systems of these portal machines should be made from open-type profiles in order to reduce the torsional rigidity of the supporting system, which in turn leads to a significant reduction in metal consumption these frames.

***Keywords**: gantry machine, supporting system, moment of inertia, structural rigidity.*

Типи конструкцій портальних кранів в основному залежать від пристрою поворотної частини, яка може обертатися на поворотній колоні або опиратися на поворотний круг, укладений на порталі, і від типу стріловидного пристрою. Основними вузлами металевих конструкцій портальних кранів є стріли, колони (каркаси) і портали (напівпортали).

Конструктивно портали виконуються в даний час листовими. Коробчасті портали завдяки жорсткості коробчастого профілю на всіх напрямках не вимагають додаткових ребер жорсткості в бічних гранях, а іноді і затягувань. Це не тільки спрощує конструкцію, але і не створює перешкод для проїзду під порталом, що особливо важливо для портових кранів. Для коробчастих конструкцій використовуються гнуті і штамповані елементи, що додатково спрощує і здешевлює їх виготовлення.

Для портальних підйомно-транспортних машин характерні велика будівельна висота конструкції і наявність рознесених мас.

Головним для описаних несучих систем є те, що вони вписуються в розрахункову схему, що включає прямокутну в плані раму і чотири стійки, обладнані різною кількістю коліс.

У статті доведено, що крутильна жорсткість рам із закритих профілів на два порядки вища ніж аналогічних рам з відкритих профілів, тобто можна враховувати, що навантаження на скручування сприймають рами із закритих профілів.

Лонжерони несучих систем портальних машин необхідно проектувати з профілів закритого типу, що доведено авторами, а поперечини несучих систем даних портальних машин слід виготовляти з профілів відкритого типу з метою зниження крутильної жорсткості несучої системи, що в свою чергу призводить до суттєвого зменшення металомісткості цих рам.

***Ключові слова**: портальна машина, несуча система, момент інерції, конструктивна жорсткість.*

Problem's Formulation

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 researched extremely little, therefore the work of some researchers in this field was aimed at scientifically based selection of parameters of portal bearing systems, for the first time outlined the ways of creating rational structures of portal machines in terms of metal consumption. There was a need to investigate the influence of the torsional stiffness of the load-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, they can serve as an analogue of the car frame.

When designing the bearing systems of gantry machines in order to reduce their metal consumption, a greater number of characteristic load modes are taken into account: 1) static load when moving in a straight line; 2) load during braking; 3) load when the machine is brought in; 4) dynamic load when the front wheels hit a road obstacle; 5) suspension of one wheel.

Analysis of recent research and publications

The design of the supporting systems of lifting and transport gantry machines is based on the research of Beygul O.O. [1], on the basis of which a technique and an algorithm for the design calculation of the bearing systems of gantry machines were developed. This technique includes six calculated cases, as well as some of their combinations: 1) lateral roll when exceeding 300 mm; 2) axisymmetric overcoming of a frontal obstacle on the verge of breaking the clutch of the drive wheels with the support surface; 3) movement over bumps on the horizontal section of the road; 4) emergency braking on a horizontal section of the road; 5) obliquely symmetrical overcoming of a frontal obstacle on the verge of breaking the clutch of the drive wheels with the support surface; 6) frequency load. Design methods of gantry machines [1,2,3] provide for a reduction in the metal consumption of non-threaded systems due to the recommended transition to an open profile of the longitudinal and transverse power elements of the frame. Frequency load occupies a special place in determining the parameters of portal-type bearing systems, which are most prone to kinematic disturbances. This is due to the fact that the carrying capacity of the gantry machine is determined not only by strength, but also by rigidity. And the stiffness of the system directly affects its own dynamic characteristics on the one hand, on the other hand — external kinematic disturbances have a certain frequency spectrum. In this system, the frequencies of external disturbances and the natural frequencies of structures are very closely intertwined. Depending on the ratio of these frequencies, there is a corresponding load on the load-bearing structures.

An increase in the deflections of the elastic elements of the load-bearing structures is associated with an increase in the stresses in the power elements, which causes additional loads of non-frequency origin [4,5].

From the above analysis, it turns out that at present, the design of the supporting systems of lifting and transporting gantry machines on pneumatic wheels does not take into account a number of their load modes, the specifics of the operation of such machines in the conditions of the technological lines of factories when carrying out loading and unloading and transport work in ports and cargo- at these stations, which affects the correctness of determining the metal content of structures and their durability.

Formulation of the study purpose

The given review showed that even in the general automotive industry, there is no scientifically based approach to the design of load-bearing systems in terms of reducing the metal consumption of undercarriage vehicles, including gantry ones. In this regard, it is possible to formulate the main goal of this monograph, which is aimed at reducing the metal consumption of gantry lifting and transport machines on pneumatic wheels.

Presenting main material

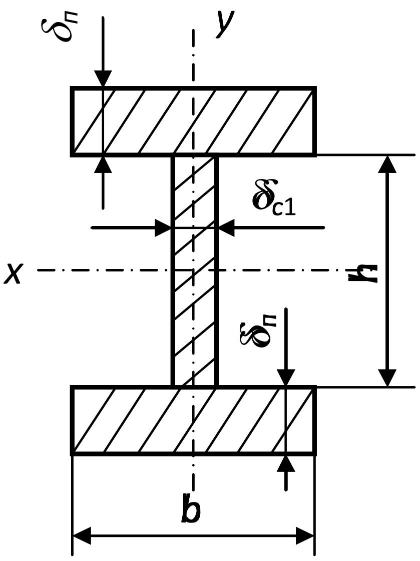

Fig 1. Cross-section of a double-breasted cross

Frames of gantry machines are complex, many times statically indeterminate flat-space systems. When revealing the static uncertainty, we will use the method of forces, where at the stage of design calculations it is necessary to know the relationship between the main geometric characteristics of the cross sections of the force elements [6].

In fig. 1 shows the cross section of the I-beam profile, which has the same moment of bending resistance as the box profile, which is determined by the formula:

$$J_u = \frac{2\delta h^3}{12} + 2\left(\frac{в\delta^3}{12} + \left(\frac{h}{2} + \frac{\delta}{2}\right)^2 в\delta\right), \tag{1}$$

where J_u — axial moment of inertia, m^4; y — shelf width, m; h — wall height, m; δ — wall thickness, m.

The reduced polar moment of inertia of the open section is determined by the expression

$$J_{KI} = \frac{1}{3}\sum_i \delta_i^3 S_i = \frac{1}{3}\left(\delta^3 2в + (2\delta)^3 h\right), \tag{2}$$

where i — number of plots; δ_i — thickness of the i-th section, m.

The reduced polar moment of inertia of a closed section is determined by the expression

$$J_{ко} = 4F^2 \Big/ \oint \frac{ds}{\delta} = \frac{4\left((в-\delta)(\delta+h)\right)_\delta^2}{2(в-\delta)+2(h+\delta)}, \tag{3}$$

where F — the area bounded by the middle lines of the walls, m^2; L — contour length, m.

In order to simplify further explanations, we accept the following relations, characteristics for thin-walled profiles: $h = 30\delta$, $h = 2в$, $в = 15\delta$. Then expressions (1)—(3) take the following form:

$$J_u = 11708\delta^4; \quad J_{кi} = 90\delta^4; \quad J_{ко} = 8371\delta^4. \tag{4}$$

Considering that the torsional stiffness of frames from closed profiles is two orders of magnitude higher than that of similar frames from open profiles, it can be concluded that the "twisting" loads are perceived only by frames from closed profiles, which allow one of the wheels to hang when the joint action is implemented bending and torsion [7]. In fig. 2 a, b show the calculated and equivalent schemes of the frame when one of the wheels hangs. In the general case, flat-space systems are many times statically indeterminate; in this case, we reduce the system to three times statically indeterminate. The canonical equations of the method of forces form a system:

$$\left.\begin{aligned} \delta_{11}X_1 + \delta_{12}X_2 + \delta_{13}X_3 + \delta_{1p} = 0; \\ \delta_{21}X_1 + \delta_{22}X_2 + \delta_{23}X_3 + \delta_{2p} = 0; \\ \delta_{31}X_1 + \delta_{32}X_2 + \delta_{33}X_3 + \delta_{3p} = 0. \end{aligned}\right\} \tag{5}$$

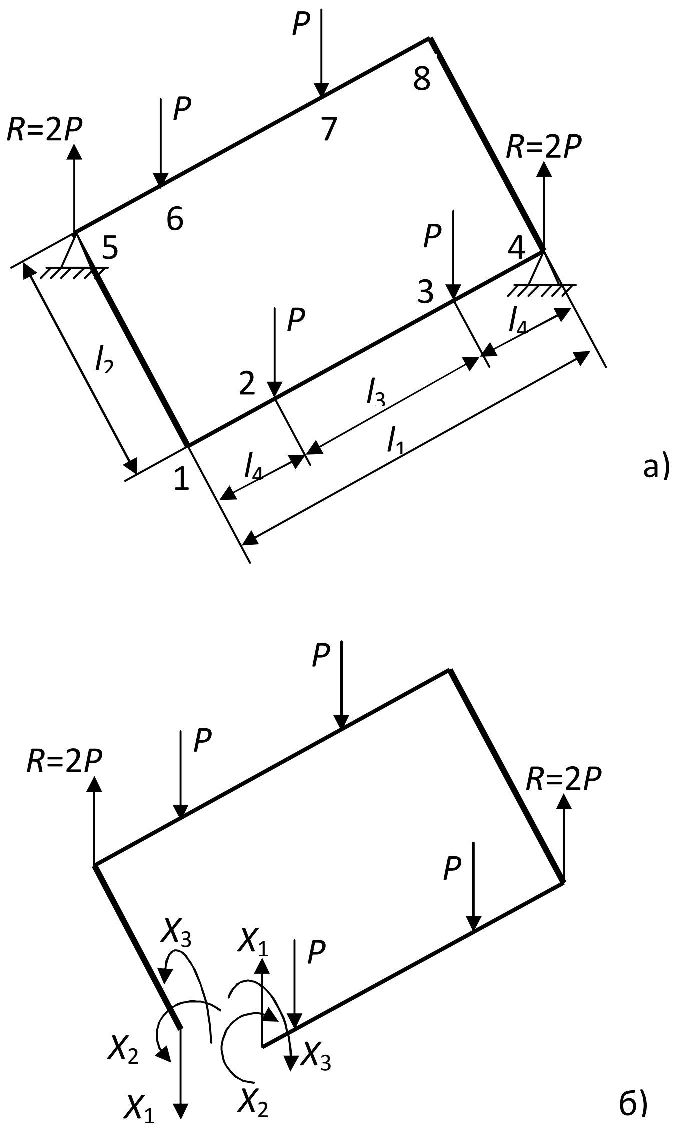

Fig 2. Calculation diagram (a) of the frame when one wheel hangs and the equivalent system (b)

To determine the coefficients and free terms of equations (5), plots of bending and torque moments from unit forces Xi, as well as from the external load in the main system, are constructed using Mohr's integral. Integration by the Vereshchagin method allows to obtain:

$$\delta_{11} = \frac{1}{EJ_и}\frac{2}{3}\left(L_1^3 + L_2^3\right) + \frac{2{,}6}{EJ_к}\left(L_1^2 L_2 + L_2^2 L_1\right); \quad (6)$$

$$\delta_{22} = \frac{1}{EJ_и}2L_1 + \frac{2{,}6}{EJ_к}2L_2\,; \quad (7)$$

$$\delta_{33} = \frac{1}{EJ_и}2L_2 + \frac{2{,}6}{EJ_к}2L_1\,; \quad (8)$$

$$\delta_{12} = \delta_{21} = \frac{1}{EJ_и}L_1{}^2 + \frac{2{,}6}{EJ_к}L_1L_2\,; \quad (9)$$

$$\delta_{13} = \delta_{31} = -\frac{1}{EJ_и}L_2{}^2 - \frac{2{,}6}{EJ_к}L_1L_2\,; \quad (10)$$

$$\delta_{23}=\delta_{32}=0\,; \tag{11}$$

$$\delta_{1p}=-\frac{1}{EJ_u}\left(\frac{2}{3}PL_3^3+4PL_3^2L_4+8PL_3L_4^2+\frac{16}{3}PL_4^3\right)-\frac{2{,}6}{EJ_\kappa}\left(2PL_1L_2L_4+PL_1L_2L_3\right); \tag{12}$$

$$\delta_{2p}=-\frac{1}{EJ_u}\left(PL_3^2+4PL_3L_4+4PL_4^2\right)-\frac{2{,}6}{EJ_\kappa}\left(2PL_2L_4+PL_2L_3\right); \tag{13}$$

$$\delta_{3p}=0\,. \tag{14}$$

Canonical equations (15) taking into account expressions (6)—(14) take the following form:

$$\begin{gathered}\left[\frac{2}{3}\frac{1}{EJ_u}\left(L_1^3+L_2^3\right)+\frac{2{,}6}{EJ_\kappa}\left(L_1^2L_2+L_2^2L_1\right)\right]X_1+\left[\frac{1}{EJ_u}L_1^2+\frac{2{,}6}{EJ_\kappa}L_1L_2\right]X_2-\\-\left[\frac{1}{EJ_u}L_2^2+\frac{2{,}6}{EJ_\kappa}L_1L_2\right]X_3=\frac{1}{EJ_u}\left(\frac{2}{3}PL_3^2+4PL_3^2L_4+8PL_3L_4^2+\frac{16}{3}PL_4^3\right)+\\+\frac{2{,}6}{EJ_\kappa}\left(2PL_1L_2L_4+PL_1L_2L_3\right);\end{gathered} \tag{15}$$

$$\begin{gathered}\left[\frac{1}{EJ_u}L_1^2+\frac{2{,}6}{EJ_\kappa}L_1L_2\right]X_1+\left[\frac{1}{EJ_u}2L_1+\frac{2{,}6}{EJ_\kappa}2L_2\right]X_2+0\cdot X_3=\\=\frac{1}{EJ_u}\left(PL_3^2+4PL_3L_4+4PL_4^2\right)+\frac{2{,}6}{EJ_\kappa}\left(2PL_2L_4+PL_2L_3\right);\end{gathered} \tag{16}$$

$$\left[\frac{1}{EJ_u}L_2^2+\frac{2{,}6}{EJ_\kappa}L_1L_2\right]X_1+0\cdot X_2+\left[\frac{1}{EJ_u}2L_2+\frac{2{,}6}{EJ_\kappa}2L_1\right]X_3=0\,. \tag{17}$$

We accept $L_2=L_1/2$; $L_3=L_1/2$; $L_4=L_1/4$. At the same time, the system of equations (15)—(17) is significantly simplified:

$$\left.\begin{aligned}&3{,}48L_1X_1+2{,}82X_2-2{,}07X_3=2{,}49PL_1;\\&2{,}82L_1X_1+5{,}64X_2=2{,}82PL_1;\\&-2{,}07L_1X_1+8{,}28X_3=0.\end{aligned}\right\} \tag{18}$$

Where do we get it from: $X_1=0{,}7P$; $X_2=0{,}15PL_1$; $X_3=0{,}175PL_1$.

We determine the parameters of power elements from the condition of strength under a complex stress state:

$$\sigma_e=\sqrt{\sigma^2+4\tau^2}=\sqrt{\left(\frac{M_u}{J_u}\left(\frac{h}{2}+\delta\right)\right)^2+4\left(\frac{M_{\kappa p}}{2F_\delta}\right)^2}\le[\sigma]. \tag{19}$$

Taking into account that $M_u=0{,}325PL$; $M_{\kappa p}=0{,}175PL$, based on the total curves of bending and torque moments for a rigid frame (closed profile), we obtain the following condition

$$\sigma_e=0{,}0006\frac{PL_1}{\delta^3}\le[\sigma], \tag{20}$$

where

$$\delta^3=\frac{0{,}0006PL_1}{[\sigma]}. \tag{21}$$

The linear mass of such an element, taking into account the accepted ratios, will be equal to:

$$m_0^{(\square)}=\rho F_{(\square)}=\rho(2\delta h+2\delta в)=2\rho\delta(30\delta+15\delta)=90\rho\delta^3, \tag{22}$$

where $m_0^{(\square)}$ — linear weight of the profile, kg/m; ρ — density of the construction material; $F_{(\square)}$ — cross-sectional area, m^2.

Or taking into account (21):

$$m_0^{(\square)}=0{,}054\rho PL_1\,/\,[\sigma]. \tag{22}$$

Due to the low torsional rigidity of frames made of open profiles, they scan the unevenness of the roads as it were, and practically do not perceive the twisting loads. For this reason, it is possible to neglect the redistribution of resistance reactions and consider them equal [8]. Then the frame is loaded only with bending loads.

We determine the parameters of power elements from the condition of bending strength. Taking into account that $M_u = 0{,}25PL_1$, based on the total curves of the bending and torque moments for the frame from the open profile, do I mean that the wall thickness is equal to δ, the following expression can be obtained

$$\delta^3 = \frac{0{,}0004PL_I}{[\sigma]}. \qquad (23)$$

The linear mass of such an element, taking into account the accepted ratios

$$m_0^{(I)}=\rho F^{(I)}=\rho(\delta h+2\delta в)=\rho\delta(30\delta+30\delta)=60\rho\delta^3, \qquad (24)$$

where $F^{(I)}$ — cross-sectional area of the open profile, m^2.

Taking into account (23), we have

$$m_0^{(I)} = 0{,}024\rho PL_1/[\sigma]. \qquad (24)$$

As follows from the above statements, the running mass of the frame of the gantry machine from open profiles is 2.25 times less than the corresponding frame from closed profiles.

Below in the tabl. 1 shows the data characterizing the reduction of the metal density of the frame of the supporting system depending on the external loads.

Table 1. Reducing the metal content of the frame of the supporting system of the gantry machine

P, кН	Mass of linear meter, kg		Mass of 2 crossbars, L=4 м		$\Delta_м$, kg
	Closed profile	Open profile	Closed profile	Open profile	
70	80,9	35,9	647,2	287,2	360,0
100	115,6	51,3	924,8	410,4	513,6
120	138,7	61,6	1109,6	492,8	616,0

Conclusions

Spars of the supporting systems of lifting and transport gantry machines should be made of closed-type profiles, as this allows to increase the critical speed of the machine. Cross-beams of the supporting systems of lifting and transport gantry machines should be designed from open-type profiles in order to reduce the torsional rigidity of the supporting system. The open profiles of the cross members of the supporting system must be optimized not only according to the criterion of minimum torsional rigidity, but also according to the condition of bending strength.

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