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OPTIMIZATION OF GEOMETRIC PARAMETERS OF OPEN PROFILES CARRYING SYSTEMS OF PORTAL UPLOAD-TRANSPORTATION MACHINES

ОПТИМІЗАЦІЯ ГЕОМЕТРИЧНИХ ПАРАМЕТРІВ ВІДКРИТИХ ПРОФІЛІВ НЕСУЧИХ СИСТЕМ ПОРТАЛЬНИХ ПІДЙОМНО-ТРАНСПОРТНИХ МАШИН

The work solves an important task of researching the stiffness characteristics that determine the load of the load-bearing system of the gantry transport machine. 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. It has also been proven that the spars of portal bearing systems should be recommended from closed-type profiles, and the crossbars from open-type profiles.

***Keywords**: vehicle, frame, system, moment of inertia, structural rigidity.*

У статті теоретично та експериментально отримано розв’язання актуальної наукової задачі обґрунтування конструктивних параметрів несучих систем портальних підйомно-транспортних машин на пневмоколісному ході, яке забезпечує раціональну металомісткість конструкцій.

Характерною особливістю рами портальної системи є те, що лонжерони по умові динамічної жорсткості повинні мати закритий профіль і тоді компенсаційну функцію по закручуванню рами повинна брати на себе поперечини з відкритим профілем. Оскільки при цьому сумарна крутильна жорсткість рами неминуче збільшується, то зростають вимоги до крутильної жорсткості поперечин, які повинні мати мінімальний рівень. Для відкритих профілів плоско-просторових несучих систем необхідно проводити оптимізацію за критерієм мінімальної крутильної жорсткості.

Таким чином розв'язується задача оптимального проектування відкритих профілів силових елементів, яка формулюється як задача пошуку екстремуму цільової функції за наявності додаткових умов. Як цільова функція розглядається крутильна жорсткість профілів, яка однозначно пов'язана з приведеним полярним моментом інерції перетинів, як додаткова умова — умова міцності при вигині.

За розробленим алгоритмом була складена програма оптимізації двотаврового профілю по критерію мінімальної крутильної жорсткості і побудовані графіки залежності приведених полярних моментів інерції від товщини полиць профілів, де однозначно виражені екстремуми цільової функції, що дозволяє зробити висновок про те, що лонжерони портальних машин слід рекомендувати виготовляти з профілів закритого типу.

***Ключові слова**: транспортний засіб, рама, система, момент інерції, конструктивна жорсткість.*

Problem's Formulation

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.

Analysis of recent research and publications

The load-carrying systems of gantry-type vehicles in the vast majority are flat-space frame systems, which are usually called ladder frames [1,2]. A characteristic feature of such systems is that the main loads are perceived by longitudinal power elements, or spars. Auxiliary loads are received by transverse force elements, or crossbars. Flat-space frame systems work well for bending "out of the plane", work poorly for torsion [2].

As you know, it is structurally impossible to create a ladder-type bearing system with high torsional stiffness, therefore, they go for a conscious reduction of it, which is achieved by using open profiles of longitudinal and transverse power elements. At the same time, the frame twists freely when hitting road irregularities, and the stresses in it are kept within acceptable limits. Bending stiffness is provided by force elements, the parameters of which are selected from the condition of strength during bending [3].

A characteristic feature of the frame of the portal system is that the spars, according to the condition of dynamic stiffness, should have a closed profile, as shown above, and then the compensatory function of twisting the frame should be taken over by crossbars with an open profile. Since the total torsional stiffness of the frame inevitably increases, the requirements for the torsional stiffness of the crossbars, which must have a minimum level, increase.

Formulation of the study purpose

In this way, the problem of optimal design of open profiles of force elements is solved, which is formulated as the problem of finding the extremum of the target function in the presence of additional conditions [3,4]. The torsional rigidity of the profiles, which is uniquely related to the reduced polar moment of inertia of the sections, is considered as the target function, as an additional condition — the condition of bending strength.

Presenting main material

For further explanations, we will consider the I-beam profile, which, together with the channel, has become the most widespread for the power elements of the load-bearing systems of vehicles. In fig. 1 presents the evolution of the I-beam profile on the way to finding a profile with the least torsional stiffness. The following designations are accepted: y — shelf width; h — wall height; δ_n — shelf thickness; $\delta_з$ — is the wall thickness. Assuming that $в$ and h are constant, we will vary δ_n and $\delta_з$ in such a way that the moment of bending resistance W_z [4] remains unchanged.

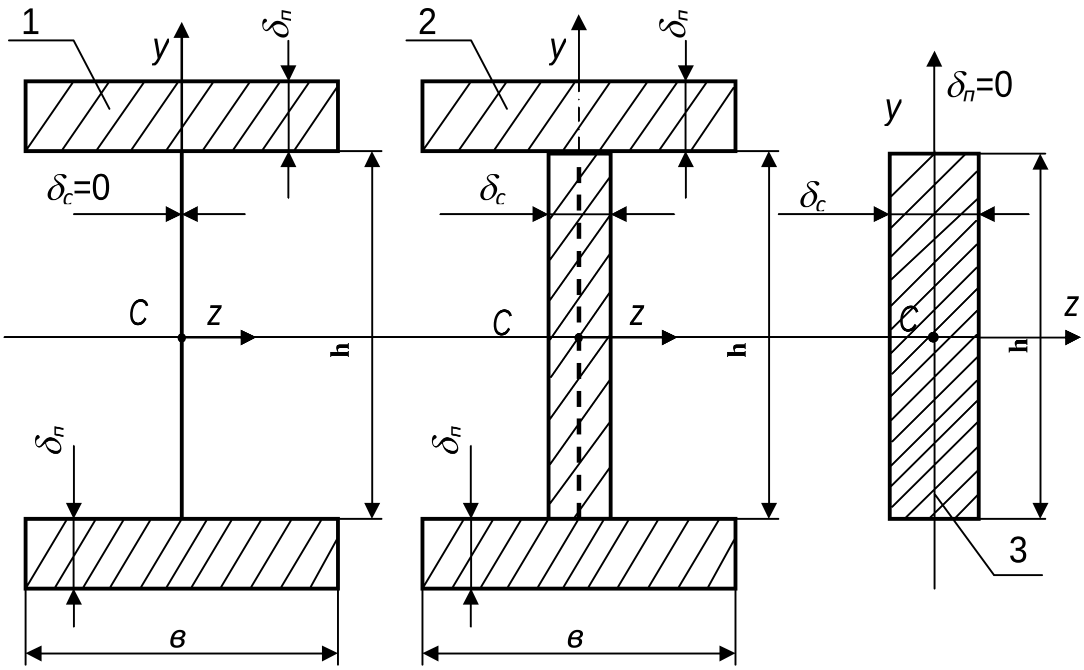

Fig. 1. Variants of an open profile that has a constant axial moment of inertia: 1 — J_z=const; $\delta_n=\delta_{n\,\max}$; δ_c=0. 2 — J_z=const; $\delta_n\neq0$; $\delta_c\neq0$. 3 — J_z=const; δ_n=0; $\delta_c=\delta_{c\max}$.

We take the first combination of thicknesses as $\delta_{n1} = \delta_{n\max}$, $\delta_{c1=0}$ (Fig. 1), which corresponds to the model of the section, where the shelves work only in tension-compression and perceive the bending moment, and the wall works only in shear [4, 5, 6].

In this case, the strength condition is written as follows:

$$\sigma = \frac{\mathrm{M_p}}{W_z} = \frac{\mathrm{M_p}}{\text{в}\delta_{\text{пmax}}(h+\delta_\text{п})} \leq [\sigma], \tag{1}$$

where σ — is the maximum normal stress, Pa; M_p — is the estimated bending moment, Nm.

From the strength condition (1), we obtain a quadratic equation with respect to the thickness of the shelf $\delta_{n\max}$:

$$[\sigma]\text{в}\delta^2_{\text{пmax}} + [\sigma]\text{в}h\delta_{n\max} - M_p = 0 \tag{2}$$

and its solution taking into account the selected sign

$$\delta_{n\max} = \frac{-[\sigma]\text{в}h + \sqrt{[\sigma]^2\text{в}^2h^2 + 4[\sigma]\text{в}\mathrm{M_p}}}{2[\sigma]\text{в}}. \tag{3}$$

For further thickness combinations, we take $\delta_n \neq 0$; $\delta_з \neq 0$ (Fig. 1). The thickness of the wall $\delta_з$ is expressed through the thickness of the shelf δ_n from the condition of bending strength:

$$\sigma = \frac{M_p}{W_z} = \frac{6M_p(h+2\delta_\text{п})}{8\text{в}\delta_\text{п}^3 + 12\text{в}h\delta_\text{п}^2 + 6\text{в}h^2\delta_\text{п} + h^3\delta_c} \leq [\sigma]. \tag{4}$$

Where do we get it directly from

$$\delta_c = \frac{6M_p(h+2\delta_\text{п}) - 8[\sigma]\text{в}\delta_\text{п}^3 + 12[\sigma]\text{в}h\delta_\text{п}^2 + 6[\sigma]\text{в}h^2\delta_\text{п}}{[\sigma]h^3}. \tag{5}$$

Knowing all the parameters of the intersection, we determine the reduced polar moment of inertia $J_к$ [5,7,8]:

$$J_\text{к} = \frac{2}{3}\delta_\text{п}^3\text{в} + \frac{1}{3}\delta_\text{c}^3 h.. \tag{6}$$

The following algorithm is developed to optimize profile parameters. Based on the known [σ], calculated calculated bending moment M_p, constructively accepted *в* and *h* we determine $\delta_{n1} = \delta_{n\max}$ according to formula (3). Given that $\delta_{c1} = 0$, we determine the reduced polar moment of inertia of the intersection $J_{к1}$ using formula (6). Combinations δ_{ni} and δ_{ci}, as well as $J_{кi}$ are calculated as follows.

The thickness of the shelf decreases and in the i-th combination of thickness is determined by the formula

$$\delta_{\text{п}i} = [1 - 0{,}05(i-1)]\delta_{n\max},, \tag{7}$$

where a coefficient of 0.05 with an accuracy sufficient for engineering calculations ensures good convergence of the approximate objective function with the exact one.

The thickness of the wall increases, and in the i-th combination of thicknesses it is expressed as follows:

$$\delta_{\text{ci}} = \frac{6\text{M}_{\text{p}}(h+2\delta_{\text{пi}})-8[\sigma]\text{в}\delta_{\text{пi}}^3-12[\sigma]\text{в}h\delta_{\text{пi}}^2-6[\sigma]\text{в}h^2\delta_{\text{пi}}}{[\sigma]h^3}. \tag{8}$$

For the i-th combination of thickness δ_{ni} and δ_{ci} the reduced polar moment of inertia of the section is determined

$$J_{\text{к}i} = \frac{2}{3}\delta_{\text{п}i}^3\text{в} + \frac{1}{3}\delta_{ci}^3 h. \tag{9}$$

Calculation cycle δ_{ni}, δ_{ci}, $J_{\kappa i}$ is repeated 21 times, which directly results from expression (7). For each set $[\sigma]$, M_p, $в$, h the operation of finding the minimum value of the reduced polar moment of inertia $J_{\kappa\min}$ is performed. According to the developed algorithm, the optimization program of the I-beam profile according to the criterion of minimum torsional stiffness was compiled. The calculations were carried out with the following initial data:

1) $[\sigma]$=220 MPa; B=80·10^{-3} m; H=150·10^{-3} m; M_p = 20 kNm.
2) $[\sigma]$=220 MPa; B=100·10^{-3} m; H=200·10^{-3} m; M_p = 50 kNm.
3) $[\sigma]$=220 MPa; B=115·10^{-3} m; H=250·10^{-3} m; M_p = 80 kNm.

Based on the obtained results, graphs of the dependence of the reduced polar moments of inertia on the thickness of the profile shelves were constructed (Fig. 2), where the extrema of the target function [4] are clearly expressed.

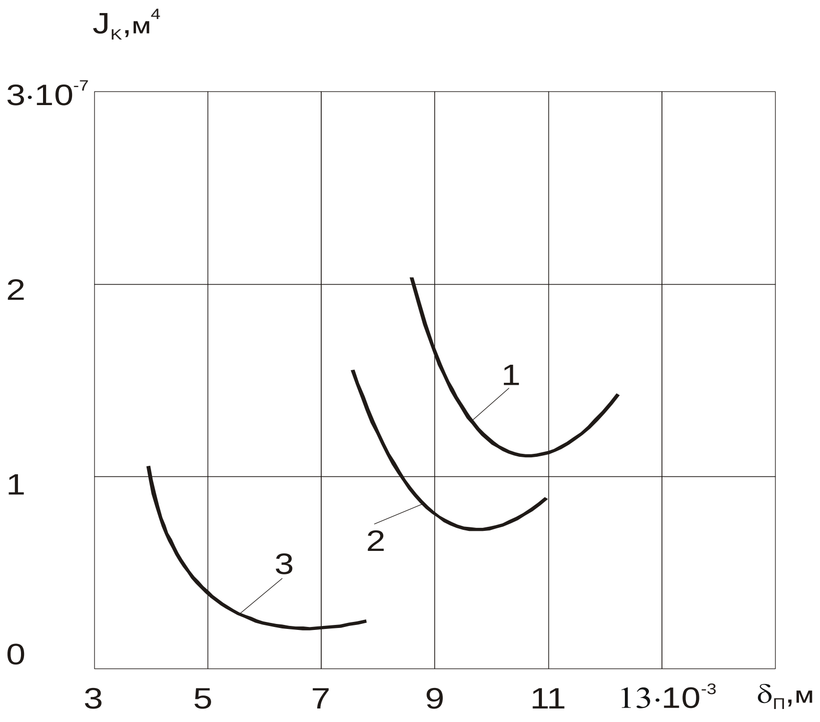

Fig. 2. Dependence of the reduced polar moment of inertia of the two-beam profile on the thickness of the shelves:

1 – M_p = 80·10^3 Nm; H=250·10^{-3} m; B=115·10^{-3} m.
2 – M_p = 50·10^3 Nm; H=200·10^{-3} m; B=100·10^{-3} m.
3 – M_p = 20·10^3 Nm; H=150·10^{-3} m; B= 80·10^{-3} m.

The transverse structural rigidity of portal bearing systems depends on the type of spars. Taking into account the low speed of movement of lifting and transport gantry machines, of the order of 1 m/s, at the design stage, it is necessary to bring the own dynamic characteristics to the zone of higher values, where resonant phenomena are detected at speeds that exceed operational speeds by an order of magnitude. It directly follows from this that the spars of lifting and transport gantry machines should be recommended to be made of closed-type profiles.

Conclusions

In order to reduce loads on the frames of the portal support systems, the crossbars should be made from open profiles, which can be objects of optimal design; the torsional stiffness of such profiles as an objective function has conditional extrema. For open profiles of flat-space bearing systems, it is necessary to optimize the parameters of the force elements according to the criterion of minimum torsional stiffness.

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