ГАЛУЗЕВЕ МАШИНОБУДУВАННЯ

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EXPERIMENTAL STUDIES OF LOADING MODES OF THE PORTAL MACHINE'S CARRIER SYSTEM ON PNEUMATIC WHEELS

The article presents the results of experimental studies of stresses in the load-bearing elements of a portal lifting and transport machine on pneumatic wheels.

The loading of the power elements of the portal supporting system was carried out in three characteristic modes: movement along the irregularities of the horizontal section of the factory road (design case for the frame spars); oblique impact on a high curb (design case for the frame crossbars and the supporting system uprights in the transverse vertical plane; frontal impact on a 200 mm high curb with one wheel at a speed of 5 km/h (design case for the frame against folding in its plane and for the supporting system uprights in the longitudinal vertical plane).

The results of the experiments confirmed the adequacy of the mathematical model developed in previous works and are within the limits of statistical error.

Keywords*: load-bearing system, frame, power element, load modes.*

У статті наведено результати експериментальних досліджень напруг у несучих елементах портальної підйомно-транспортної машини на пневматичних колесах.

Навантаження силових елементів портальної несучої системи здійснювалося на трьох характерних режимах: русі по нерівностях горизонтальної ділянки дороги заводу (розрахунковий випадок для лонжеронів рами); косому наїзді на високий бордюр (розрахунковий випадок для поперечин рами і стійок несучої системи в поперечній вертикальній площині; фронтальному наїзді на бордюр заввишки 200 мм одним колесом із швидкістю 5 км/год (розрахунковий випадок для рами проти складання в своїй площині і для стійки несучої системи в поздовжній вертикальній площині).

Результати експериментів підтвердили адекватність розробленої у попередніх роботах математичної моделі і знаходяться в межах статистичної похибки.

Ключові слова*: несуча система, рама, силовий елемент, режими навантаження.*

Problem’s Formulation

Since some parameters of the developed mathematical model can be determined only experimentally, the task of experimental research of a prototype of a lifting and transporting portal machine on pneumatic wheels was set.

As a prototype of a lifting and transporting portal machine on pneumatic wheels, a portal car with a carrying capacity of 30 tons from the company VALMET (Finland) was used in the experimental research. The experiments were carried out in the conditions of PrJSC "KAMET-STAL" on a number of technological roads.

Analysis of recent research and publications

In works [1—3], the results of scientific studies of the loading of the elements of the supporting system of a gantry machine are presented, which take into account vibrations only in the longitudinal plane when overcoming obstacles head-on, but do not take into account its vibrations in the transverse plane when it overcomes obstacles. In studies [4, 5], theoretical relationships between the geometric parameters of the supporting systems and stresses caused by dynamic loads during the movement of the gantry machine were obtained, but in these works the coefficient of transverse structural rigidity was not taken into account when calculating the loads on the elements of the supporting system.

In works [6, 7], when obtaining the results of experimental studies of the loads on the frame elements during the disturbed movement of the gantry machine, all possible types of road obstacles were not taken into account in the complex.

Formulation of the study purpose

In the developed mathematical model, calculated loads on the spars in the longitudinal vertical plane, on the cross members in the transverse vertical plane, and dynamic loads directed at the frame assembly in its plane were obtained. Therefore, the purpose of experimental research was to verify the adequacy of the mathematical model and calculation schemes to the actual loading modes of the machine during its operation.

Presenting main material

As a method of experimental research of the stress-strain state of power elements of structures, in particular, tests for the strength of structures, the electro-strain method was adopted. Strain gauge equipment when studying the load regimes of the supporting systems of these machines in road conditions of technological lines is in difficult conditions of vibration, shaking. At the same time, serious requirements are imposed on the equipment, in particular: vibration isolation and vibration resistance, when the reliability and operability of the equipment must be maintained up to fifteen-fold overload; power supply of all devices must be carried out from current sources located on board the machine under study; the layout and installation of strain gauge equipment must be simple and accessible to ensure its installation with minimal time costs. Regarding the lifting and transporting portal machine, the operator's seat must be reliably protected from the standpoint of safety; Regarding metallurgical portal machines used in areas with heated metal, the equipment must maintain its metrological characteristics at temperatures up to +60 єC. The current consumed by the equipment must not exceed 5A.

Standard strain gauge equipment was used to conduct the experimental study, which meets the above requirements. This equipment allows recording processes in the frequency range from 0 to 500 Hz.

The loading of the power elements of the portal supporting system was carried out in three characteristic modes: 1) movement along the irregularities of the horizontal section of the plant road (calculated case for frame spars); 2) oblique impact on a high curb (design case for the frame cross members and the support system uprights in the transverse vertical plane; 3) frontal impact on a 200 mm high curb with one wheel at a speed of 5 km/h (design case for the frame against folding in its plane and for the support system uprights in the longitudinal vertical plane).

Fig. 1 shows a diagram of the portal support system, indicating the places where the strain gauges were installed. Strain gauges 1, 2 were installed on the upper shelf of the spar and recorded the pure bending stresses in the longitudinal vertical plane in the first loading mode. Strain gauges 3, 4 were installed on the upper shelf of the crossbar and recorded the bending stresses in the transverse vertical plane in the second loading mode. Strain gauges 5, 6 were installed in the root sections of the uprights and recorded the bending stresses in the transverse vertical plane in the second loading mode.

Strain gauge 7 was installed in the root section of the upright that ran into the curb and recorded the bending stresses in the longitudinal vertical plane in the third loading mode. Strain gauge 8 was installed in the root section of the cross member, attached to the front wall of the cross member and recorded the bending stress in the plane of the frame at the third loading mode. Strain gauge 9 was installed in the root section of the spar, attached to the outer wall of the spar and recorded the bending stress in the plane of the frame at the third loading mode.

In the first loading mode, the stress recording took place on a characteristic horizontal section of the plant road with a length of 50 m, on which scrap was transported. The experiment was repeated three times.

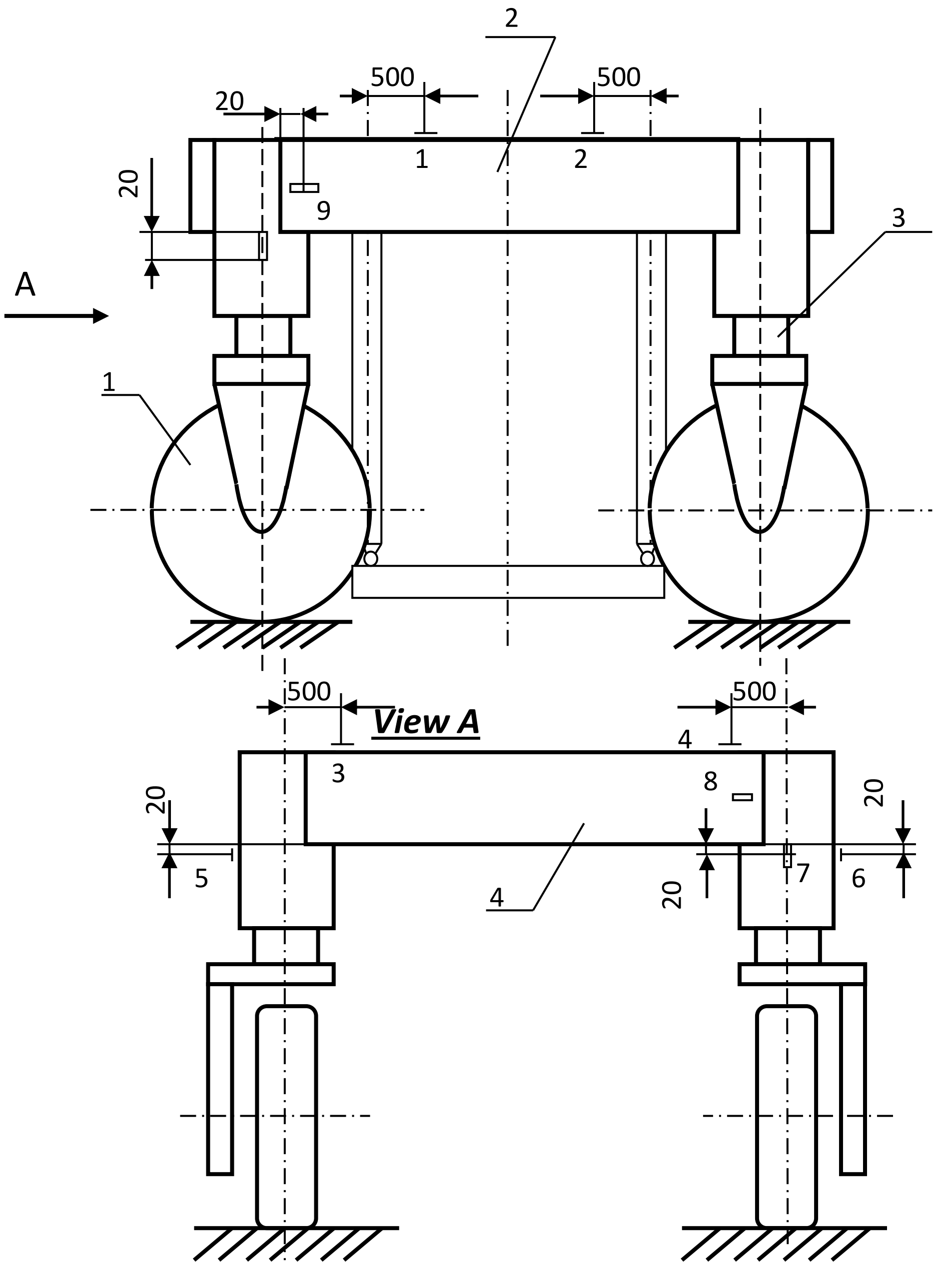

Fig. 1. Strain gauge installation diagram: 1 — wheel; 2 — spar; 3 — strut; 4 — cross member

In the second mode, the load was simulated through a 200 mm high road curb. The nominally loaded gantry machine approached the curb at an angle of up to 5є. The experiment was repeated three times.

In the third loading mode, the stress recording took place when the left front wheel hit a 200 mm high curb at a speed of υ_0 = 1 m/s. During the experiment in the third loading mode, the speed of the machine was also recorded. Since in the low-speed zone the standard speedometer does not provide the required accuracy of readings, a magnetoelectric speed sensor was developed and manufactured, which was installed on the front right wheel of the gantry machine.

The experimental study of the loading of the gantry bearing system was carried out in accordance with the above program and methodology. Oscillograms of stresses at points 1 and 2 of the spar (Fig. 1) are presented in Fig. 2, where the following notations are adopted: σ_{ci} is the static stress in the cross-section of the installation of the i-th strain gauge; $\sigma_{\partial i}$ is the dynamic stress in the cross-section of the installation of the i-th strain gauge. The dynamic coefficient is determined from the relations

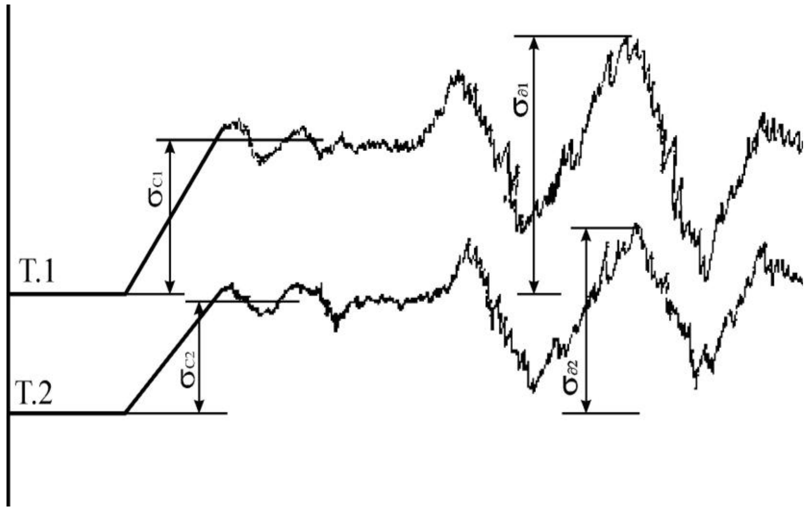

Fig. 2. Oscillograms of stresses in the spar at the first loading mode

$$K_\partial = \frac{P_\partial}{P_s}, \quad K_\partial = \frac{\sigma_\partial}{\sigma_c}. \tag{1}$$

where P_∂ — dynamic loads, MPa; P_s — static loads, MPa.

The adequacy of the mathematical model is assessed using the formula

$$\delta = \frac{K_{\partial t} - K_{\partial e}}{K_{\partial t}} \cdot 100\% , \tag{2}$$

where δ — adequacy indicator; $K_{\partial t}$ — theoretical dynamic coefficient; $K_{\partial e}$ — experimental dynamic coefficient.

When determining the theoretical dynamic coefficient $K_{\partial t}$, the dynamic load value P_∂ is calculated according to the expression [8], and the static load P_s is taken from the characteristic oscillograms of the static load stresses of the portal machine.

Experimental dynamic coefficients, some system parameters, and an assessment of the adequacy of the mathematical model are summarized in Tabl. 1.

Table 1. Assessment of the adequacy of the mathematical model at the first load mode

point	υ, m/s	σ_c, MPa	σ_∂, MPa	P_s, kN	P_∂, kN	$K_{\partial e}$	$K_{\partial t}$	δ, %
1	5,56	26,25	39,6	75	120	1,51	1,6	5,6
2	5,56	26,25	40,4	75	120	1,54	1,6	3,7

Fig. 3 presents characteristic oscillograms of stresses in the struts and crossbar of the supporting system in the second loading mode. The stress values averaged over three dimensions, as well as the assessment of the adequacy of the mathematical model, are summarized in Tabl. 2.

Table 2. Assessment of the adequacy of the mathematical model in the second load mode

point	P_z, kN	σ_T, МПа	σ_e, МПа	δ, %
3	72,0	120,6	112,8	6,4
4		120,6	111,9	7,0
5		100,3	94,1	6,1
6		100,3	95,2	5,0

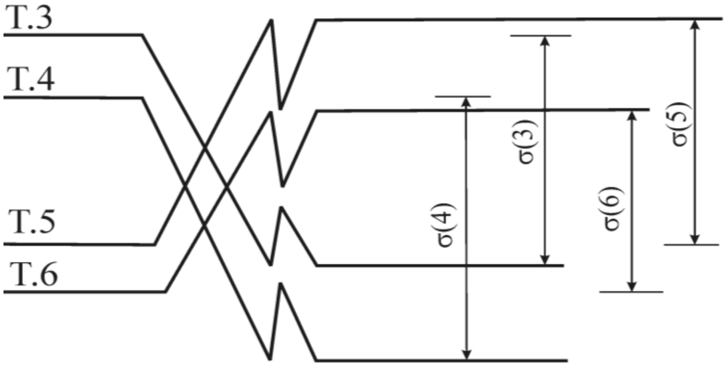

Fig. 3. Oscillograms of stresses in the uprights and crossbar in the second loading mode

The adequacy indicator is calculated using the formula

$$\delta = \frac{\sigma_{T}(i) - \sigma_{e}(i)}{\sigma_{T}(i)} \cdot 100\% , \quad (3)$$

where $\sigma_{T(i)}$ — theoretical value of stress at the i-th point, MPa; $\sigma_{e(i)}$ — experimental value of stress at the i-th point, MPa.

Fig. 4 shows the oscillograms of stresses in the strut that receives the impact when hitting the curb, as well as in the spar and cross member of the frame in the third loading mode. The stress values averaged over three dimensions, as well as the assessment of the adequacy of the mathematical model, are summarized in Tabl. 3.

Table 3. Assessment of the adequacy of the mathematical model for the third load mode

point	υ_0, m/s	P_p, kN	σ_T, MPa	σ_e, MPa	δ, %
7	1,0	191,0	166,4	154,1	7,0
8			79,2	74,4	6,0
9			47,7	45,8	3,9

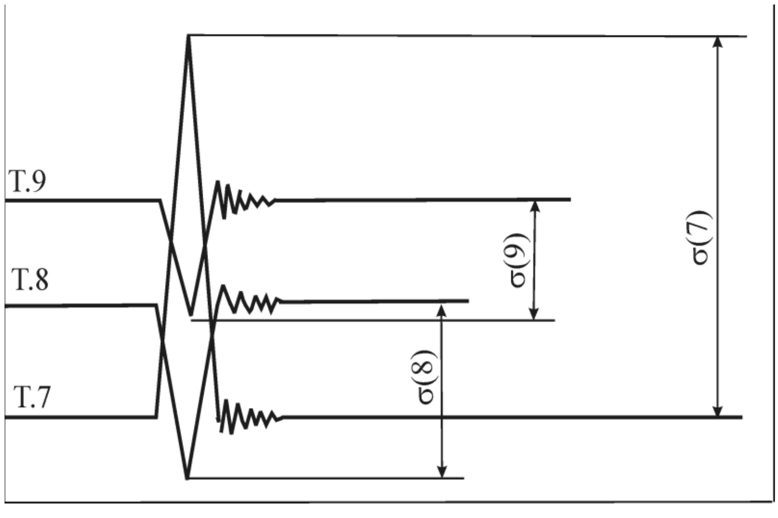

Fig. 4. Oscillograms of stresses in the strut, cross member and spar at the third loading mode

Frontal impact of one wheel of a portal machine on a high curb is likely during maneuvers related to loading and unloading of pallets or containers. The maneuvering speed in this case depends on a number of objective and subjective factors. Therefore, the test was carried out by three portal machine drivers at the beginning and end of the shift and three runs were performed (eighteen in total).

Experimental study of maneuvering speed was performed in accordance with the above program and methodology.

Based on the processing of experimental data on measurements of the maneuvering speed of portal machines, υ_0 = 1 m/s can be recommended as its calculated value during frontal impact on the curb.

The values of the theoretical bending stresses of the strut in the longitudinal vertical plane, the theoretical bending stresses of the cross member in the frame plane and the theoretical bending stresses of the spars were determined under the corresponding loading conditions of these elements during bending [9, 10]. The dynamic loads on the cross member and struts of the supporting system during the frontal impact of the portal machine on the curb and the calculated bending moments for the spars were determined in advance [8].

The values of the bending stresses of the strut, cross member and spars were obtained during the experimental study of the portal car of the VALMET company (Finland). In eighteen runs, the speed of the car varied at the moment of the frontal impact of one wheel on the curb from 0.75 m/s to 0.95 m/s.

The stresses at point 1 and point 2 of the spars (Fig. 1) in the first loading mode (Fig. 2) indicate that the maximum vertical loads on the supporting structure are realized during oscillations in the longitudinal plane, which confirms the correctness of the analytical expression [8] for calculating the vertical load. This phenomenon is explained by the layout of the portal machine, which has a large construction height, when the driver primarily parries oscillations, which at a high height of the cabin cause significant linear displacements in the longitudinal plane.

Conclusions

Experimental study of the lifting and transport portal machine confirmed the adequacy of the developed mathematical model [8] of the formation of external loads, the adopted calculation scheme of the supporting system. The discrepancy between the theoretical and experimental values of the profiling loads for the first calculation case does not exceed 5,6 %, for the second and third — 7 %, the obtained experimental values are less than the theoretical ones. This ratio is acceptable, since it is a margin of safety and reliability of the supporting systems of the portal machine as a whole.

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ЕКСПЕРИМЕНТАЛЬНІ ДОСЛІДЖЕННЯ РЕЖИМІВ НАВАНТАЖЕННЯ НЕСУЧОЇ СИСТЕМИ ПОРТАЛЬНОЇ МАШИНИ НА ПНЕВМАТИЧНИХ КОЛЕСАХ

Реферат

Деякі параметри розробленої у попередніх теоретичних дослідженнях математичної моделі можна визначити тільки експериментальним шляхом, була поставлена задача експериментального дослідження прототипу підйомно-транспортної портальної машини на пневмоколісному ході. Метою експериментальних досліджень є перевірка адекватності математичної моделі і розрахункових схем дійсним режимам навантаження портальної машини при її роботі.

Як прототип підйомно-транспортної портальної машини на пневмоколісному ході при проведенні експериментального дослідження був задіяний портальний автомобіль вантажопідйомністю 30 тон фірми VALMET. Експерименти проведені в умовах ПрАТ «КАМЕТ-СТАЛЬ» на ряді внутрішньозаводських шляхів.

Навантаження силових елементів портальної несучої системи здійснювалося на трьох характерних режимах: русі по нерівностях горизонтальної ділянки дороги заводу (розрахунковий випадок для лонжеронів рами); косому наїзді на високий бордюр (розрахунковий випадок для поперечин рами і стійок несучої системи в поперечній вертикальній площині; фронтальному наїзді на бордюр заввишки 200 мм одним колесом із швидкістю 5 км/год (розрахунковий випадок для рами проти складання в своїй площині і для стійки несучої системи в поздовжній вертикальній площині).

Експериментальне дослідження підйомно-транспортної портальної машини підтвердило правомірність розробленої математичної моделі формування зовнішніх навантажень, прийнятої розрахункової схеми несучої системи. Розбіжність між теоретичними і експериментальними значеннями профілюючих навантажень по першому розрахунковому випадку не перевищує 5,6 %, по другому і третьому — 7 %, отримані експериментальні значення менші теоретичних.

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