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INVESTIGATION OF BAND LOOP FRAME ARTICULATED CONTEINER TRUCK STABILITY BY MOTION ON THECHNOLOGICAL ROADS

Mathematical model of disturbance motion for articulated container truck with band loop frame on pneumatic wheels by different conditions of cross stabilization has been worked out by analytical mechanics method with Lagrange equation of second type. Analytical expressions of critical parameters of system, which define stability of articulated container truck in cross plane has been received.
***Keywords**: mathematical model, stability, articulated container truck, disturbance motion, band loop frame.*

У роботі розроблена математична модель збуреного руху зчленованого контейнеровоза з бугельною рамою на пневмоколісному ході при різних умовах поперечної стабілізації методами аналітичної механіки з залученням рівняння Лагранжа другого роду. Отримані аналітичні вирази критичних швидкостей, які містять у собі параметри системи, що визначають стійкість зчленованого контейнеровоза у поперечній площині.
***Ключові слова**: математична модель, стійкість, зчленований контейнеровоз, збурений рух, бугельна рама.*

The statement of the problem

The advancement of the metallurgical industry is closely linked to the continued growth of mechanization and automation of transport operations along technological lines, where new tasks are being set in the way of development of promising models of special vehicles. Thus, for carrying out operations related to the autonomous loading, transportation and unloading of containers, it is advisable to layout the carrier system with a tow frame.

The rowing layout of the carrier systems of technological vehicles is quite unconventional. As the review of the literature shows, only a few scientific works are devoted to such machines. If for the usual layout, especially in the general automotive industry, we have accumulated a wealth of experience in the development of mathematical models of perturbed motion, the formation of external loads, internal efforts, then for articulated container ships on a pneumatic wheel course with a tow frame everything has to be done for the first time.

For scientifically sound choice of such carrier systems, dynamic loads play a decisive role, the formation of which is described by a mathematical model of the process of perturbed motion of a container ship in conditions of technological roads of metallurgical production.

Analysis of basic research and publications

The development and construction of container transporters with a tow frame are in the initial stages of development, so we consider the problem from a more general point of view. Thus, works [1-3] are devoted to the study of special vehicles for the transport of goods in containers and packages in various industries. The work [4,5] dedicated to the design and construction of articulated skid steer carriers is of course preceded by a fundamental work [6] to investigate the dynamics of self-propelled swivel machines, including oscillation and movement stability. The works [7–9] are devoted to the dynamics of two-link mechanical systems, which according to the calculation schemes are similar to articulated containers with a tow frame. The work [10] is devoted to the study of the dynamics of perturbed motion of an articulated special vehicle with a U - shaped cargo frame.

Formulation of the purpose of the study

From the above analysis, it follows that the known mathematical models of the perturbed motion of technological special vehicles do not take into account the layout features of articulated containers with a tow carrier system. There are practically no materials in the scientific literature devoted to the research and development of metallurgical container ships with a tow frame, with the obvious relevance of such machines in technological lines of metallurgical enterprises.

Thus, it is possible to formulate the following research goal: to find out the influence of the stabilizer of lateral stability in the elastic suspension of the tow frame on the critical stability of the speed of movement of a container ship with different level of stabilization.

The main material

Shown in Fig. In Fig. 1 an articulated container vessel with a tow frame on the side, shown in Fig. 2 shows the design scheme of the container carrier on the rear view.

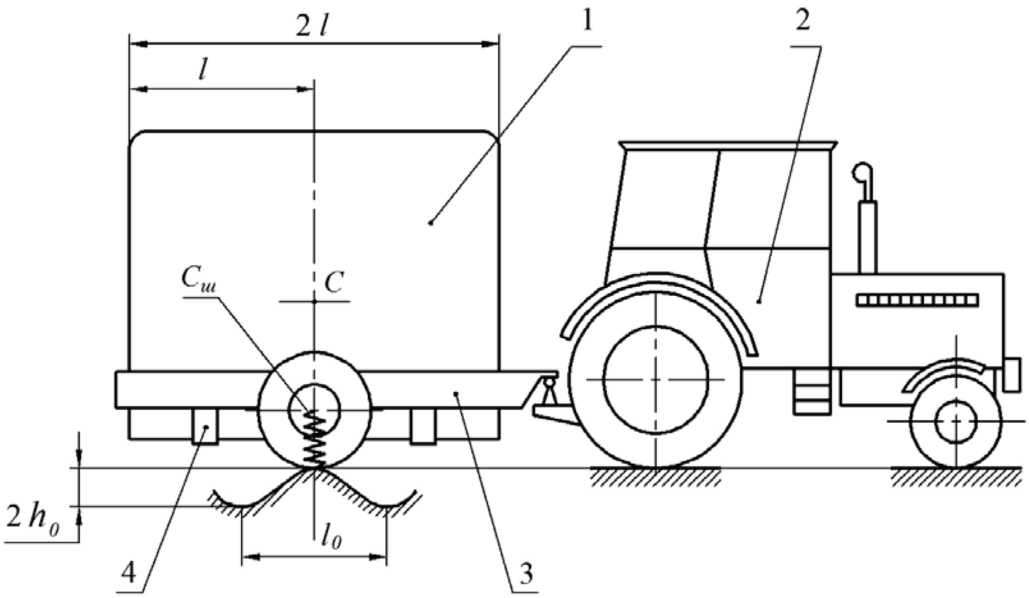

Fig. 1. Articulated container ship with a tow frame: 1 — container; 2 — tractor; 3 — semi-trailer with tow frame; 4 — lodging

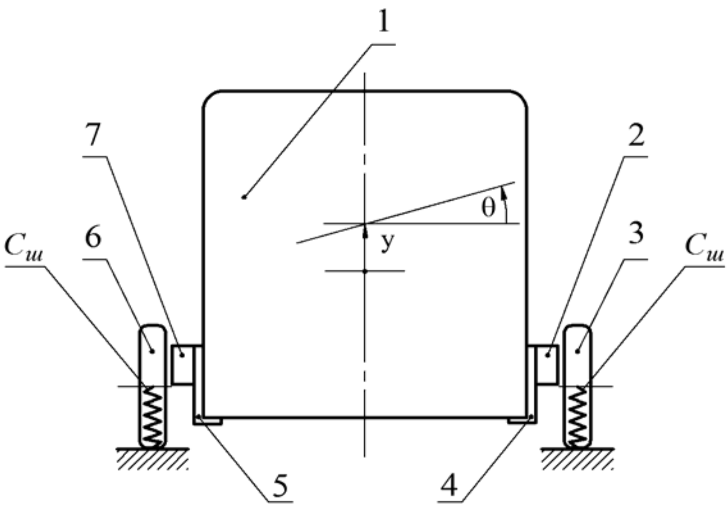

Fig. 2. The articulated container vessel on the rear view: 1 — container; 2 — right frame spar; 3 — right wheel of a semi-trailer suspension; 4 — right lodgment; 5 — left lodging; 6 — left semi-trailer suspension wheel; 7 — left frame spar

The perturbation equation is obtained in the form of the second-kind Lagrange equation [11].

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right)-\frac{\partial T}{\partial q_j}=-\frac{\partial \Pi}{\partial q_j}+Q_j\,, \tag{1}$$

where T — the kinetic energy of the system, J; Π — the potential energy of the system, J; q_j — j-a the generalized coordinate,m (rad); Q_j — j-a the generalized force of non-conservative origin, N (N·m).

In this case $q_1 = y$, $q_2 = \theta$, where y — vertical displacementof the semi-trailer, a θ — the angle of the semi-trailer in the transverse plane.

The kinetic energy of the system:

$$T=\frac{m_c\dot{y}^2}{2}+\frac{I_c\dot{\theta}^2}{2}+\frac{m_c v^2}{2}\,, \tag{2}$$

where m_c — the total mass of the semi-trailer, kg; I_c — semitrailer total moment of inertia relative to the longitudinal axis, which runs through the center of mass of the system, kg·m^2; v — system speed, m/s.

The potential energy of the system is realized by the energy of compressed air in the tires of the wheels, takes the following form:

$$\Pi=\frac{C_ш}{2}(y-l_к\theta)^2+\frac{C_ш}{2}(y+l_к\theta-h_n)^2\,, \tag{3}$$

where $C_ш$ — radial rigidity of a semi-trailer wheel tire, N/m; $l_к$ — half track of semi-trailer, m; h_n — lifting of the right wheel of a semi-trailer on inequalities of a sinusoidal profile, m.

In general form can be written

$$h_n = h_0\sin\frac{2\pi x}{l_0}\,, \tag{4}$$

where h_n — висота нерівностей, m; h_0 — the amplitude value of the inequality profile, m; x — abscissa of the approximate function of inequalities, m; l_0 — length of sine wave that approximates the function of inequalities, m.

When the system is moving

$$x = vt\,. \tag{5}$$

In view of expression (5), the lift of the right wheel of the semi-trailer is equal

$$h_n = h_0\sin\frac{2\pi v}{l_0}t\,. \tag{6}$$

Neglecting the forces of non-conservative origin, we substitute expressions of kinetic and potential energies in the second-kind Lagrange equation (1).

$$\frac{\partial T}{\partial \dot{y}}=\frac{\partial}{\partial \dot{y}}\left(\frac{m_c\dot{y}^2}{2}+\frac{I_c\dot{\theta}^2}{2}+\frac{m_c v^2}{2}\right)=m_c\dot{y}\,;$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{y}}\right)=\frac{d}{dt}(m_c\dot{y})=m_c\ddot{y}\;;$$

$$\frac{\partial T}{\partial y}=\frac{\partial}{\partial y}\left(\frac{m_c\dot{y}^2}{2}+\frac{I_c\dot{\theta}^2}{2}+\frac{m_c v^2}{2}\right)=0\,;$$

$$\frac{\partial \Pi}{\partial y}=\frac{\partial}{\partial y}\left[\frac{C_ш}{2}(y-l_к\theta)^2+\frac{C_ш}{2}(y+l_к\theta-h_n)^2\right]=$$
$$=\frac{C_ш}{2}\cdot 2(y-l_к\theta)\cdot 1+\frac{C_ш}{2}\cdot 2(y+l_к\theta-h_n)\cdot 1=$$
$$=C_ш y-C_ш l_к\theta+C_ш y+C_ш l_к\theta-C_ш h_n=2C_ш y-C_ш h_n .$$

Combining the expressions obtained, we write the first differential equation:

$$m_c\ddot{y}+2C_ш y=C_ш h_n . \tag{7}$$

$$\frac{\partial T}{\partial\dot{\theta}}=\frac{\partial}{\partial\dot{\theta}}\left(\frac{m_c\dot{y}^2}{2}+\frac{I_c\dot{\theta}^2}{2}+\frac{m_c v^2}{2}\right)=I_c\dot{\theta};$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\theta}}\right)=\frac{d}{dt}\left(I_c\dot{\theta}\right)=I_c\ddot{\theta};$$

$$\frac{\partial T}{\partial\theta}=\frac{\partial}{\partial\theta}\left(\frac{m_c\dot{y}^2}{2}+\frac{I_c\dot{\theta}^2}{2}+\frac{m_c v^2}{2}\right)=0;$$

$$\frac{\partial \Pi}{\partial \theta}=\frac{\partial}{\partial \theta}\left[\frac{C_ш}{2}(y-l_к\theta)^2+\frac{C_ш}{2}(y+l_к\theta-h_n)^2\right]=$$
$$=\frac{C_ш}{2}\cdot 2(y-l_к\theta)\cdot(-l_к)+\frac{C_ш}{2}\cdot 2(y+l_к\theta-h_n)\cdot l_к=$$
$$=-C_ш l_к y+C_ш l_к^2\theta+C_ш l_к y+C_ш l_к^2\theta-C_ш l_к h_n=2C_ш l_к^2\theta-C_ш l_к h_n .$$

Combining the expressions obtained, we write the second differential equation:

$$I_c\ddot{\theta}+2C_ш l_к^2\theta=C_ш l_к h_n . \tag{8}$$

Rewrite the equation (7) and (8) on the basis of expression (6):

$$m_c\ddot{y}+2C_ш y=C_ш h_0\sin\frac{2\pi v}{l_0}t; \tag{9}$$

$$I_c\ddot{\theta}+2C_ш l_к^2\theta=C_ш l_к h_0\sin\frac{2\pi v}{l_0}t . \tag{10}$$

We write down equations (9) and (10) with the introduction of new notation

$$\ddot{y}+p_y^2 y=\frac{C_ш}{m_c}h_0\sin\frac{2\pi v}{l_0}t, \tag{11}$$

where $p_y^2=\frac{2C_ш}{m_c}$ — the square of the natural circular frequency of the system in the generalized coordinate y, c^{-2};

$$\ddot{\theta}+p_\theta^2\theta=\frac{C_ш l_к}{I_c}h_0\sin\frac{2\pi v}{l_0}t, \tag{12}$$

where $p_\theta^2=\frac{2C_ш l_к^2}{I_c}$ — the square of the natural circular frequency of the system in the generalized coordinate θ, c^{-2}.

Equations (11) and (12) have the following form:

$$y=C_1\cos p_y t+C_2\sin p_y t+\frac{C_ш h_0}{m_c\left(p_y^2+\frac{4\pi^2 v^2}{l_0^2}\right)}\sin\left(\frac{2\pi v}{l_0}t-\psi_1\right), \tag{13}$$

$$\theta = C_3 \cos p_\theta t + C_4 \sin p_\theta t + \frac{C_ш l_к h_0}{I_c \left(p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2} \right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_2 \right). \quad (14)$$

The first two additions in expressions (13) and (14) describe the free oscillations of the system, which are determined by the initial conditions and, after a while, disappear. Remain forced oscillations, which are described as follows:

$$y = \frac{C_ш h_0}{m_c \left(p_y^2 - \frac{4\pi^2 v^2}{l_0^2} \right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_1 \right), \quad (15)$$

$$\theta = \frac{C_ш l_к h_0}{I_c \left(p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2} \right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_2 \right). \quad (16)$$

Directly from expression (16) we obtain the condition of loss of stability of the semitrailer in the transverse plane during asymmetric kinematic perturbation

$$p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2} = 0 \quad (17)$$

or after opening its own circular frequency p_θ

$$\frac{2C_ш l_к^2}{I_c} - \frac{4\pi^2 v^2}{l_0^2} = 0. \quad (18)$$

From where do we get the critical speed at which the stability of a semi-trailer container carrier in the transverse plane occurs:

$$v_{кр} = \frac{l_к l_0}{\pi} \sqrt{\frac{C_ш}{2I_c}}. \quad (19)$$

To increase the lateral stability of the system, due to the high position of the center of mass of the container carrier, we include a stabilizer of the transverse stability in the suspension of the trailer part, which is taken into account in the calculation model by introducing equivalent stiffness of the suspension during oblique symmetry perturbations.

Again, we incorporate the second-kind Lagrange equation (1) into the mathematical model.

In this case, the system has two degrees of freedom when $q_1 = y$, $q_2 = \theta$, where y — vertical movement of the semi-trailer, and θ — the angle of the semi-trailer in the transverse plane.

The kinetic energy of the system:

$$T = \frac{m_n \dot{y}^2}{2} + \frac{I_n \dot{\theta}^2}{2} + \frac{m_c v^2}{2}, \quad (20)$$

where m_n — the mass of the sprung parts of the semi-trailer, kg; I_n — moment of inertia of the sprung parts of the semi-trailer regarding to the longitudinal axis passing through the center of mass of these parts, kg·m^2; m_c — the total mass of the semi-trailer, kg; v — system speed, m/s.

Potential energy of the system:

$$П = \frac{C_e}{2}(y - l_к\theta)^2 + \frac{C_e}{2}(y + l_к\theta - h_п)^2 + \frac{C_\beta}{2}\theta^2, \quad (21)$$

where C_e — coefficient of equivalent rigidity of the elastic suspension, N/m; C_β — coefficient of angular stiffness of the stabilizer of lateral stability, N/m; $l_к$ — half track of semi-trailer, m; $h_п$ — lifting of the right wheel of a semi-trailer on inequalities of a sinusoidal profile, m.

The coefficient of equivalent rigidity is determined by the formula

$$C_e = \frac{C_{ш} C_{п}}{C_{ш} + C_{п}}, \quad (22)$$

where $C_{ш}$ — radial stiffness coefficient of suspension pneumatics, N/m; $C_{п}$ — the coefficient of rigidity of the suspension, N/m.

The angular stiffness coefficient of the lateral stabilizer is as follows:

$$C_{\beta} = C_c \frac{l_c^2}{2}, \quad (23)$$

where C_c — коефіцієнт лінійної жорсткості стабілізатора поперечної стійкості, N/m; l_c — the length of the stabilizer, which works on torsion at warps of the sprung parts, m.

In general form can be written

$$h_{п} = h_0 \sin \frac{2\pi x}{l_0}, \quad (24)$$

where $h_{п}$ — height of inequalities, m; h_0 — the amplitude value of the inequality profile, m; x — abscissa of the approximate function of inequalities, m; l_0 — the length of a sine wave that approximates the function of inequalities, m.

When moving the system

$$x = vt. \quad (25)$$

In view of expression (25), the lift of the right wheel of the semi-trailer is equal

$$h_n = h_0 \sin \frac{2\pi v}{l_0} t. \quad (26)$$

Neglecting the forces of non-conservative origin, we substitute the kinetic and potential energies in the second-kind Lagrange equation (1), and obtain the following system of equations

$$\left. \begin{aligned} & m_{п} \ddot{y} + 2C_e y = C_e h_0 \sin \frac{2\pi v}{l_0} t; \\ & I_{п} \ddot{\theta} + \left(2C_e l_{к}^2 + C_{\beta}\right) \theta = C_e l_{к} h_0 \sin \frac{2\pi v}{l_0} t. \end{aligned} \right\} \quad (27)$$

The first equation of system (27) is divided into $m_{п}$, other — into $I_{п}$, we get:

$$\left. \begin{aligned} & \ddot{y} + \frac{2C_e}{m_{п}} y = \frac{C_e h_0}{m_{п}} \sin \frac{2\pi v}{l_0} t; \\ & \ddot{\theta} + \frac{2C_e l_{к}^2 + C_{\beta}}{I_{п}} \theta = \frac{C_e l_{к} h_0}{I_{п}} \sin \frac{2\pi v}{l_0} t. \end{aligned} \right\} \quad (28)$$

We enter the following notation:

$$\frac{2C_e}{m_{п}} = p_y^2; \quad (29)$$

$$\frac{2C_e l_{к}^2 + C_{\beta}}{I_{п}} = p_{\theta}^2. \quad (30)$$

Rewrite the system (28) with the notation (29) and (30):

$$\left. \begin{aligned} & \ddot{y} + p_y^2 y = \frac{C_e h_0}{m_{п}} \sin \frac{2\pi v}{l_0} t; \\ & \ddot{\theta} + p_{\theta}^2 \theta = \frac{C_e h_0 l_{к}}{I_{п}} \sin \frac{2\pi v}{l_0} t. \end{aligned} \right\} \quad (31)$$

The solutions of equations (31) are given

$$y = C_1 \cos p_y t + C_2 \sin p_y t + \frac{h_0 p_y^2}{2\left(p_y^2 - \frac{4\pi^2 v^2}{l_0^2}\right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_1\right); \tag{32}$$

$$\theta = C_3 \cos p_\theta t + C_4 \sin p_\theta t + \frac{C_e h_0 l_k p_\theta^2}{\left(2C_e l_k^2 + C_\beta\right)\left(p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2}\right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_2\right). \tag{33}$$

In expressions (32) and (33), the first two additions describe the free oscillations of a fast-decaying system and cannot be taken into account. In this case, the solution of equations (31) can be written as follows:

$$y = \frac{h_0 p_y^2}{2\left(p_y^2 - \frac{4\pi^2 v^2}{l_0^2}\right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_1\right), \tag{34}$$

$$\theta = \frac{C_e h_0 l_k p_\theta^2}{\left(2C_e l_k^2 + C_\beta\right)\left(p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2}\right)} \sin\left(\frac{2\pi v}{l_0} t - \psi_2\right), \tag{35}$$

where ψ_i — phase angle, rad.

From these solutions, we write the condition of loss of stability of the system in transverse oscillations at skew symmetry perturbations.

$$p_\theta^2 - \frac{4\pi^2 v^2}{l_0^2} = 0 \tag{36}$$

or after disclosure p_θ^2:

$$\frac{2C_e l_\kappa^2 + C_\beta}{I_п} - \frac{4\pi^2 v^2}{l_0^2} = 0\,. \tag{37}$$

Where do we get the critical speed at which the system loses stability.

$$v_{кр} = \frac{l_0}{2\pi} \sqrt{\frac{2C_e l_\kappa^2 + C_\beta}{I_п}}\,. \tag{38}$$

Conclusions and prospects for further research

As a result of the theoretical study of the dynamics of a articulated container ship with a tow frame on a pneumatic wheel, a mathematical model of perturbed motion in the transverse plane was developed taking into account the different level of stabilization of the trailer on the basis of the mathematical apparatus of analytical mechanics with engaging.

The developed mathematical model of the perturbed motion in the transverse plane is the basis for determining the critical speed of movement of the container vessel in the absence and presence of the stabilizer of the transverse stability.

In further studies of container ships with a tow frame on a pneumatic wheel it is necessary to work in the direction of development of new types of stabilizers of cross resistance with rational metal content, structural perfection.

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ДОСЛІДЖЕННЯ ПОПЕРЕЧНОЇ СТІЙКОСТІ БУГЕЛЬНИХ НЕСУЧИХ СИСТЕМ ЗЧЛЕНОВАНИХ КОНТЕЙНЕРОВОЗІВ ПРИ РУСІ ПО ТЕХНОЛОГІЧНИХ ДОРОГАХ

Бейгул О.О., Грищенко Д.І., Бейгул В.О., Середа Д.Б.

Реферат

Мета роботи — отримання нових співвідношень у математичній моделі збуреного руху зчленованого контейнеровоза з бугельною рамою на пневмоколісному ході під час руху технологічними дорогами промислових підприємств. Основним джерелом збурень є кососиметричні кінематичні збурення з боку нерівностей покриття технологічних доріг, які апроксимуються гармонічними функціями вздовж колії. Основою математичного апарату теоретичного дослідження є методологія аналітичної механіки з використанням рівняння Лагранжа другого роду. Об'єкт дослідження зводиться до динамічної системи з двома ступенями свободи, де узагальнені координати представлені лінійними та кутовими переміщеннями несучої системи контейнеровоза, що призводить до складання двох диференціальних рівнянь руху. Права частина цих рівнянь описує параметри кінематичних збурень. З розв’язання диференціальних рівнянь збу-

реного руху контейнеровоза з бугельною рамою за наявності кососиметричних кінематичних збурень виходять власні динамічні характеристики системи, умова стійкості при поперечних коливаннях зчленованого контейнеровоза, отримується умова для визначення критичної швидкості, при якій система втрачає стійкість. Розроблена математична модель сприяє активному втручанню на стадії проектування у параметри коливань зчленованого контейнеровоза з бугельною рамою на пневмоколісному ході шляхом вибору відповідних жорсткісних характеристик стабілізатора поперечної стійкості. Отримані співвідношення демонструють алгоритм впливу на параметри системи на етапі розробки конструкції з метою створення ідеальної моделі зчленованого контейнеровоза з бугельною рамою на пневмоколісному ході, що гарантує велику практичну значимість дослідження.

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