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THE INFLUENCE OF THE TYPE OF TRACTIONAL COUPLING DEVICE ON THE REDISTRIBUTION OF BRAKING TORQUES WHEN THE OPERATIONAL AND DESIGN FACTORS OF THE TRAILER CHANGE

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A modern tractor is a complex mechanical system consisting of a large number of elements united into one whole by various types of connections. The relevance of this work lies in the fact that the tire-vehicle-driver system acts as an oscillating system when the tractor train moves on different road surfaces and off-road. At the same time, it must be taken into account that the entire variety of disturbing influences is taken into account by the "double" (dynamic and kinematic) and "two-frequency" nature of excitation. Both are mainly related to the unevenness of the road surface and the imbalance of the rotating and gradually moving masses.

It must be said that almost all research on vibration protection of a person and a transport unit was reduced to the study of high-frequency vibrations. However, along with high-frequency oscillations, low-frequency oscillations have a significant impact on the human operator and the transport unit.

The study of any oscillating system includes two stages:

1. Study of disturbing influences.

2. Study of vibration dampers.

The study of disturbing influences is mainly reduced to the study of the interaction of the micro-profile of the road and the drivers of the transport unit, to the mathematical description of the road profile and, at the same time, the mathematical description of the laws of change in the relative movement of the driver. It should be noted that the impact on the studied indicators of the road body and the laws of motion is given by the speed of movement, since the frequency and amplitude properties of the influences change. The state of the road, in turn, affects the traction forces, the nature of the interaction between the road surface and the driver in terms of power.

Thus, taking into account the peculiarities of the study of the road-tyre-vehicle-driver oscillatory system, in order to determine the influence of the traffic parameters of the transport-technological train and the characteristics of the road background on the performance indicators of the transport-technological train in general and the operator in particular, it is necessary to draw up a system of differential equations that describes the movement of the transport-technological train taking into account the type of traction-towing device used.

Key words: *transport-technological train, dynamics, oscillations, driver, trailer.*

Eq. 28. Fig. 5. Ref. 10.

1. Problem formulation

The high speed of the tractor train, maximum utilization of the tractor's power and the variety of transported goods, together with the characteristics of the road surface and the design features of the rolling stock, lead to an increase in dynamic loads. This causes increased oscillation processes in the soil-trailer-tractor system, which negatively affects traction and coupling properties, handling, stability, as well as ergonomics and acceleration and braking properties.

Therefore, reducing dynamic loads and improving the operational characteristics of tractor trains is an important task that affects the increase in productivity, fuel efficiency, and cross-country ability.

The study of tractor movement parameters, taking into account the factors and relationships between individual elements, is a complex task, the solution of which, using an effective computational model, will qualitatively improve the consideration of the movement parameters of a transport and technological train.

2. Analysis of recent research and publications

Research in the field of the road-trailer-tractor-operator system has been conducted over the years by scientists from Ukraine and many foreign countries.

In his work, Altomonte Andrea [1] conducts an exhaustive analysis of the performance of a heavy-duty diesel tractor engine used in port logistics to assess the potential of regenerative braking that can be used in an electrified vehicle configuration.

Hesham Rakha [2] discusses a model that uses these variables to build synthetic driving cycles for each road segment and then predicts average fuel consumption and emissions for four modes: deceleration, idling, acceleration, and cruising. In each mode of operation, fuel consumption and emissions are determined using ratios derived from instantaneous microscopic energy and emissions models. The model allows the user to calibrate two additional input parameters, namely typical deceleration and acceleration rates. It is shown that the proposed model successfully predicts fuel consumption and emissions of hydrocarbons, carbon monoxide, carbon dioxide, and nitrogen oxides.

In his work, Borawski Andrzej [5] tried to check how the brakes of a semitrailer heat up during emergency braking. For this purpose, a mathematical model of the braking process was developed and simulation tests were conducted for different initial speeds and different loads. The result is friction material temperature profiles that show how the drum and brake pads heat up during braking and cool down immediately after stopping.

In the article by Vadym Samorodov [6], the description of the criteria given is used to form changes in the control parameters of hydraulic machines (hydraulic pump and hydraulic motor) during acceleration and braking of a wheeled tractor. A rational change in the control parameters of a hydraulic machine at the stages of acceleration and deceleration of a wheeled tractor with a continuously variable hydraulic-volumetric transmission, which is carried out according to the scheme of the output differential, has been formed. In the course of studying the obtained dependence of the hydraulic machine control parameters, the change in such indicators as acceleration and braking time; braking distance of a wheeled tractor; efficiency of the hydraulic drive and hydraulic mechanical transmissions; and operating pressure drop in the hydraulic drive were determined. It has been established that when using a rational change in hydraulic machine control parameters instead of a linear one at the acceleration and braking stages, the zone of the highest value of the efficiency of the hydraulic-volumetric transmission is narrowed, which, in turn, indicates the load of the hydraulic branch of the hydraulic-volumetric transmission.

In [7], Xu Shiwei proposes a multi-mode composite braking control strategy for a five-axle heavy-duty vehicle with a distributed electric wheel drive. First, considering the differences in braking dynamics between two-axle vehicles and multi-axle vehicles, the braking dynamics characteristics of multi-axle vehicles are analyzed, and a vehicle dynamics model of multi-axle vehicles is constructed. Next, a multi-mode combined braking control strategy is proposed, which includes all-electric braking and hybrid electro-hydraulic braking, to improve braking energy recovery and braking stability.

3. The purpose of the article

The aim of the study is to simulate the influence of the design parameters of the coupling device on the braking dynamics of a transport and technological train using an electronic model.

4. Results of the researches

The speed and safety of transport and technological trains are largely determined by their braking properties, i.e:

- the ability to quickly reduce the movement to a complete stop with a minimum braking distance;
- the ability to maintain a given speed downhill;
- ability to remain stationary under the influence of random forces.

The braking efficiency at a certain initial speed is assessed by the length of the braking distance or the time it takes for a transport and technological train to come to a stop. The higher the braking efficiency, the higher the safe speed that the operator can allow, and the higher the average speed of the transport and technological train along the entire route.

In practice, the following braking modes are distinguished:

- emergency
- service braking;
- braking on road slopes (descents).

Since braking torques affect traffic safety, let's consider the process of their formation.

Braking torques with traction on the trailer axles are determined from the expressions:

- on the front axle of the trailer

$$M_{T\_3} = r_3 P_{T\_3} = r_3 \varphi N_3, \tag{1}$$

- on the rear axle of the trailer

$$M_{T\_4} = r_4 P_{T\_4} = r_4 \varphi N_4. \tag{2}$$

The optimal value of the coefficient of distribution of the total braking force, at which the wheels of the front and rear axles are simultaneously locked, is expressed by the following formula [3]:

$$k_\beta = \frac{\varphi h_{ПР} + a_4}{L_{ПР}}. \tag{3}$$

Formula (3) shows that with a decrease in the trailer base and an increase in the height of the center of gravity, the optimal value of the total braking force distribution coefficient increases, which means that the coordinates of the center of gravity, as well as the distribution of braking torques along the trailer axles, largely depend on the load, which in turn depends on the proportion of the load.

Let's determine the height of the center of gravity for any trailer load. We assume that the distribution of masses over the entire volume of the trailer body is uniform. Since the coordinate h_{c_p} of the trailer's center of gravity changes as a result of its loading, it is advisable to consider a flat model with a continuously distributed mass when determining it [4]. Then

$$h_{c\_пр} = \frac{\sum m_{пр\_i} g y_{пр\_i}}{\sum m_{пр\_i} g}. \tag{4}$$

According to equation (4), the coordinate of the center of gravity will take the following form:

$$h_{c\_пр} = \frac{m_{ПР} g\left(h_{пор\_пр} - \dfrac{m_{гр} g}{c_{пр}}\right) + m_{гр} g\left(h_{пл} - \dfrac{m_{гр} g}{c_{пр}} + \dfrac{h_x}{2}\right)}{m_{ПР} g + m_{гр} g}, \tag{5}$$

where $c_{пр} = \dfrac{4 c_ш c_p}{c_ш + c_p}$ – The stiffness of the tires and springs is shown.

Given that the volume of cargo V_g in the body can be expressed as:

$$V_g = abh_x \text{ та } V_g = \frac{m_g g}{\lambda g}, \tag{6}$$

where a and b – width and length of the trailer body.

The body loading height will be determined as follows:

$$h_x = \frac{m_{гр} g}{\gamma_{гр} ab}. \tag{7}$$

Substituting expression (7) into equation (5), we obtain the equation for the coordinate of the trailer's center of gravity depending on the specific gravity of the cargo and the trailer load:

$$h_{c\_пр} = \frac{m_{п\_ПР} g\left(h_{п\_пр} - \dfrac{m_{гр} g}{c_{пр}}\right) + m_{гр} g\left(h_{пл} - \dfrac{m_{гр} g}{c_{пр}} + \dfrac{m_{гр} g}{2\gamma_{гр} ab}\right)}{m_{п\_ПР} g + m_{гр} g}. \tag{8}$$

Given the obtained coordinate of the trailer's center of gravity, write down the normal reactions acting on the trailer axles:

- for the front axle of the trailer

$$N_3 = \frac{1}{L_{ПР}\left(m_{п\_ПР} g + m_{гр} g\right)}\Bigg[\left(m_{ПР} g a_4 + P_{кр} h_{сц}\right)\left(m_{п\_Н} g + m_{гр} g\right) + \left(m_{ПР} g\varphi - P_{кр}\right)\left(m_{g\_ПР} g\left(h_{п\_ПР} - \frac{m_{гр} g}{c_{пр}}\right) + m_{гр} g\left(h_{пл} - \frac{m_{гр} g}{c_{пр}} + \frac{m_{гр} g}{2\gamma_{гр} ab}\right)\right)\Bigg], \tag{9}$$

- for the rear axle of the trailer

$$N_4 = \frac{1}{L_{ПР}\left(m_{п\_ПР} g + m_{гр} g\right)}\Bigg[\left(m_{ПР} g a_3 + P_{кр} h_{сц}\right)\left(m_{п\_Н} g + m_{гр} g\right) + \left(P_{кр} - m_{ПР} g\varphi\right)\left(m_{п\_ПР} g\left(h_{п\_пр} - \frac{m_{гр} g}{c_{пр}}\right) + P_{гр}\left(h_{пл} - \frac{m_{гр} g}{c_{пр}} + \frac{m_{гр} g}{2\gamma_{гр} ab}\right)\right)\Bigg]. \tag{10}$$

Substituting the values of the normal reactions (9) and (10) into the braking torque equations (1) and (2), we obtain the braking torques:

- on the front axle of the trailer

$$M_3 = \frac{r_3\varphi}{L_{ПР}\left(m_{п\_ПР} g + m_{гр} g\right)}\Bigg[\left(m_{ПР} g a_4 + P_{кр} h_{сц}\right)\left(m_{п\_ПР} g + m_{гр} g\right) + \left(m_{ПР} g\varphi - P_{кр}\right)\left(m_{п\_ПР} g\left(h_{п\_пр} - \frac{m_{гр} g}{c_{пр}}\right) + m_{гр}\left(h_{пл} - \frac{m_{гр} g}{c_{пр}} + \frac{m_{гр} g}{2\gamma_{гр} ab}\right)\right)\Bigg], \tag{11}$$

- for the rear axle of the trailer

$$M_4 = \frac{r_4\varphi}{L_{ПР}\left(m_{п\_ПР} g + m_{гр} g\right)}\Bigg[\left(m_{ПР} g a_3 + P_{кр} h_{сц}\right)\left(m_{п\_ПР} g + m_{гр} g\right) + \left(P_{кр} - m_{ПР} g\varphi\right)\left(m_{п\_ПР} g\left(h_{п\_пр} - \frac{m_{гр} g}{c_{пр}}\right) + m_{гр} g\left(h_{пл} - \frac{m_{гр} g}{c_{пр}} + \frac{m_{гр} g}{2\gamma_{гр} ab}\right)\right)\Bigg]. \tag{12}$$

To analyze the effect of the coupling device on the distribution of braking moments along the trailer axles when the share of cargo, the displacement of the center of gravity in the longitudinal plane, and the weight load of a two-axle trailer change, we calculated and plotted graphical dependences of braking moments on trailer loading (Fig. 1). on the elimination of the center of gravity in the longitudinal plane (Fig. 2).

From the analysis of the dependencies in Figs. 1 and 2, it can be concluded that the use of the experimental traction coupling device improves the braking dynamics of the tractor-transport unit, which is expressed in a more favorable distribution of braking torques along the trailer axles when braking a transport and technological train, which is more clearly seen when the trailer load changes.

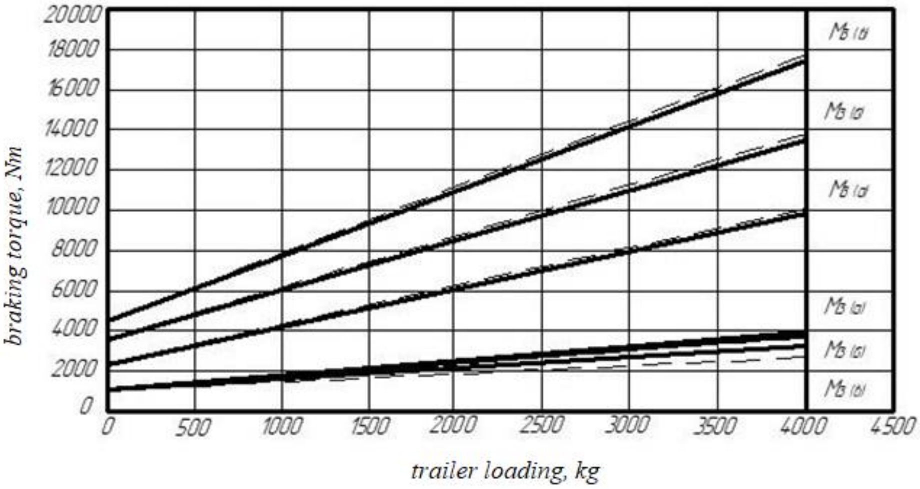

Fig. 1. Dependence of braking torques on trailer load

Thus, from the analysis of the influence of the coupling device on the distribution of braking moments, it can be concluded that the use of an elastic coupling device will allow smoother redistribution of braking moments along the axles of the transport and technological train, which makes the braking of the transport and technological train more stable.

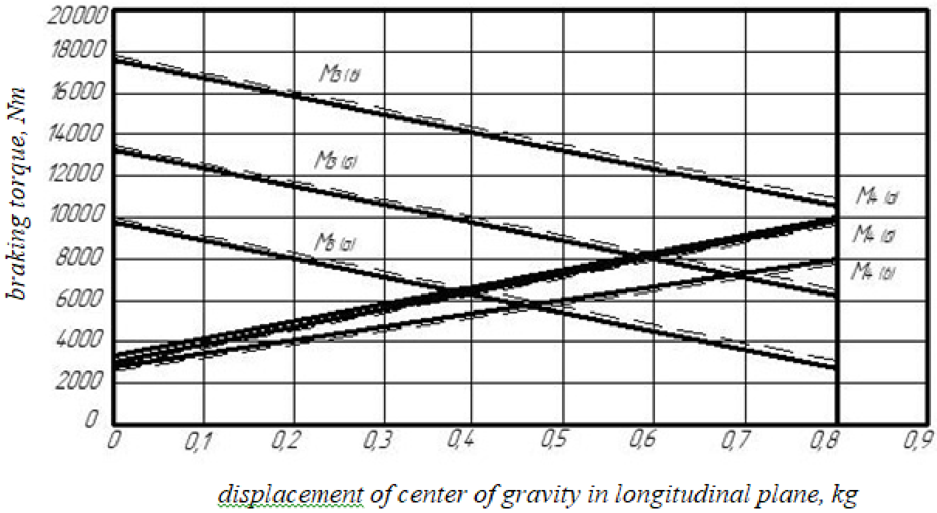

Fig. 2. Dependence of braking torques on the removal of the trailer's center of gravity in the longitudinal plane

Calculation of brake track of transport and technological train

The greatest interest from the standpoint of the theory of brake dynamics and the subsequent increase in active safety of the movement of a transport and technological train is an emergency type of braking. It is this braking mode that allows us to formulate the conditions for the full use of potential braking properties and, thereby, justify ways to further increase the active safety of the transport and technological train.

As you know, the main forces that cause a stop are braking forces. The distribution of the kinetic energy of the transport-technological train by types of resistance during braking to a stop is such that brake mechanisms account for more than 95%. Therefore, the braking of a transport-technological train can be considered under the influence of only braking moments.

When determining the braking path of a transport-technological train, assessing the braking efficiency, the level of improvement of the braking properties of the transport-technological train using an elastic traction coupling device, it is necessary to count the braking path from the beginning of the operator's pressing the brake pedal. Therefore, such a subjective factor as the reaction time of the operator will be excluded.

Let's calculate the brake track of the transport and technological train, consisting of a tractor with a mechanical brake drive and a two-axle trailer equipped with a pneumatic brake drive. In this case, it is impossible to describe the increase in all braking moments arising from a transport-technological train by the analytical dependence of one type.

At mechanical drive of brakes, increase of braking moment in time is described by direct proportional dependence, and operation of pneumatic drive of brakes is described by exponential dependence (Fig. 3) [5].

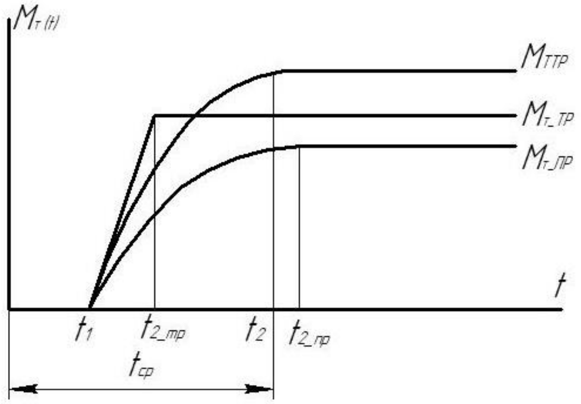

Fig. 3. Approximation of the law of increasing braking moments of a tractor-transport train and its links

For a mechanical drive, the dependence of the increase in the braking moment of the Mt_tr on time will be recorded in the form:

$$M_{T\_i}(t)=\begin{cases} 0 & \text{при } t \le t_{1\_тр\_i} \\ M_{T\_i} \; \dfrac{t-t_{1\_тр\_i}}{t_{2\_тр\_i} \; t_{1\_тр\_i}} & \text{при } t \le t_{1\_тр\_i} \le t \le t_{2\_тр\_i} \\ M_{T\_i} & \text{при } t > t_{2\_тр\_i} \end{cases} \tag{13}$$

$$M_{T\_j}(t)=\begin{cases} 0 & \text{при } t \le t_{1\_пр\_j} \\ M_{T\_j} \; k_{апр}\left(1-e^{-k_{cm}t}\right) & \text{при } t_{1\_пр\_i} \le t \le t_{2\_пр\_i} \\ M_{T\_j} & \text{при } t > t_{2\_тр\_j} \end{cases} \tag{14}$$

Due to the presence of elastic connections between the links of the transport and technological unit, it is no longer recommended to take the unit for a point mass. Therefore, to calculate the brake track of a transport and technological train, it is necessary to consider the following system of equations:

$$\begin{cases} m_{TP}\ddot{x}_{TP} = -\sum\limits_{i=1}^{n}\dfrac{M_{T\_TPi}(t)}{r_i} + P_{кр}(t) \\ m_{ПР}\ddot{x}_{ПР} = -\sum\limits_{j=1}^{k}\dfrac{M_{T\_ПРj}(t)}{r_j} - P_{кр}(t) \end{cases}. \tag{15, 16}$$

We rewrite the system of equations (15, 16) in the form:

$$\begin{cases} \ddot{x}_{TP} = -\dfrac{1}{m_{TP}}\sum\limits_{i=1}^{n}\dfrac{M_{T\_TPi}(t)}{r_i} + \dfrac{1}{m_{TP}}P_{кр}(t), \\ \ddot{x}_{ПР} = -\dfrac{1}{m_{ПР}}\sum\limits_{j=1}^{л}\dfrac{M_{T\_ПРj}(t)}{r_j} + \dfrac{1}{m_{ПР}}P_{кр}(t). \end{cases}. \tag{17, 18}$$

We find the law of change in the speed of the tractor and trailer by integrating equations (17, 18), respectively:

$$\begin{cases} \ddot{x}_{TP} = \vartheta_0 - \dfrac{1}{m_{TP}} \int\limits_0^t \sum\limits_{i=1}^n \dfrac{M_{T\_TPi}(t)}{r_i} dt + \vartheta_{0\xi} \dfrac{1}{m_{TP}} \int\limits_{t1}^t P_{\kappa p}(t) dt, \\ \ddot{x}_{\Pi P} = \vartheta_0 - \dfrac{1}{m_{\Pi P}} \int\limits_0^t \sum\limits_{j=1}^k \dfrac{M_{T\_\Pi Pj}(t)}{r_j} dt + \vartheta_{0\xi} - \dfrac{1}{m_{\Pi P}} \int\limits_{t1}^t P_{\kappa p}(t). \end{cases} \quad (19, 20)$$

Equating the expression (2.237) or (2.238) to zero, you can determine the braking time to stop tt.

The braking distance can be determined from equation (19) or (20) due to its secondary integration:

$$\begin{cases} x_{TP} = \vartheta_0 t - \dfrac{1}{m_{TP}} \int\limits_0^t \left(\int\limits_0^t \sum\limits_{i=1}^n \dfrac{M_{T\_TPi}(t)}{r_i} dt \right) dt + \vartheta_{0\xi} t_\xi \dfrac{1}{m_{TP}} \int\limits_{t1}^t \left(\int\limits_{t1}^t P_{\kappa p}(t) dt \right) dt, \\ x_{\Pi P} = \vartheta_0 t - \dfrac{1}{m_{\Pi P}} \int\limits_0^t \left(\int\limits_0^t \sum\limits_{j=1}^k \dfrac{M_{T\_\Pi Pj}(t)}{r_j} dt \right) dt + \vartheta_{0\xi} t_\xi - \dfrac{1}{m_{\Pi P}} \int\limits_{t1}^t \left(\int\limits_{t1}^t P_{\kappa p}(t) dt \right) dt. \end{cases} \quad (21, 22)$$

Since the coordinate of the transport-technological unit is taken when determining the braking distance of the tractor coordinate xtr, equation (21) allows you to determine the braking path of the transport-technological unit, taking into account the elastic connection of the links of the transport-technological train with the known nature of work and under any known law of increasing braking moments of time. The law of force change in the traction coupler is primarily determined by the law of increasing braking moments in the transport-technological train, that is, the nature of the change in the pressure of the working medium in the brake drive of the transport-technological train [8].

Enter the designation:

$$j_{TP} = \frac{1}{m_{TP}} \sum_{i=1}^n \frac{M_{T\_TPi}(t)}{r_i}; \quad (23)$$

$$j_{TP\_i} = \frac{1}{m_{TP}} \frac{M_{T\_TPi}(t)}{r_i}. \quad (24)$$

We solve equation (19) taking into account the expression (13). Replacing by places the operations of integration and summation in equation (19) we have:

$$\dot{x}_{TT\Pi} = \dot{x}_{TP} = \vartheta_0 - \sum_{i=1}^n \int\limits_0^t j_{TP\_i} dt + \vartheta_{0\xi} + \frac{1}{m_{TP}} \int\limits_{t1}^t P_{kp}(t) dt. \quad (25)$$

It follows from equation (25) that the first integral can be calculated in general form for one braking moment, and then, summing up at all points, get a final solution. To calculate the integral of equation (23), we use the method of phased integration. Consider the following characteristic intervals:

1. $t \le t_1$, $\Delta\vartheta_{I\_1} = 0$.

2. $t_{1\_i} \le t \le t_{2\_TP\_i}$, $\Delta\vartheta_{I\_2} = \dfrac{j_{TP\_i}}{t_{2\_TP\_i} - t_{1\_i}} \int\limits_{t_{1\_i}}^{t_{2\_TP\_i}} (t - t_{1\_i}) dt$,

3. $t > t_{2\_TP\_i}$, $\Delta\vartheta_{I\_3} = \dfrac{j_{TP\_i}}{t_{2\_TP\_i} - t_{1\_i}} \int\limits_{t_{1\_i}}^{t_{2\_TP\_i}} (t - t_{1\_i}) dt + j_{TP\_i} \int\limits_{t3\_TP\_i}^{t} dt.$

Або $\Delta\vartheta_{i\_3} = j_{TP\_i} t - j_{TP\_i} \left(\dfrac{t_{2\_TP\_i} - t_{1\_i}}{2} + t_{1\_i} \right).$

Substitute the last of the acquired integral values into equation (25):

$$\dot{x}_{TP} = \vartheta_0 + \vartheta_{0\xi} - j_{TP} t + \sum_{i=1}^n j_{TP\_i} \left(\frac{t_{2\_TP\_i} - t_{1\_i}}{2} + t_{1\_i} \right) + \frac{1}{m_{TP}} \int\limits_{t1}^t P_{kp}(t) dt. \quad (26)$$

Equating equation (26) to zero, we find the braking time by the example of P_{kr} = const, then [9]:

$$t_T = \frac{m_{TP}\left(\vartheta_0 + \vartheta_{0\xi}\right)}{m_{TP} j_{TP} - P_{kp}} + \frac{m_{TP}}{m_{TP} j_{TP} - P_{kp}} \sum_{i=1}^{n} j_{TP\_i}\left(\frac{t_{2\_TP\_i} - t_{1\_i}}{2} + t_{1\_i}\right) - \frac{P_{kp}}{m_{TP} j_{TP} - P_{kp}} t_1. \quad (27)$$

Formula (27) is valid only from the time when the brake system has fully worked and all braking moments have reached the maximum value.

Having integrated equation (26) on the segment from 0 to tt, we will find, using the method described above, the coordinate of the transport-process train during tt, the value of which at the initial coordinate of the transport-process train equal to zero will coincide with the value of the braking distance [10].

The decision will be recorded as:

$$S_{TTП} = x_{TTП} - x_{0TTП} = \frac{1}{2} t_T^2 \left(\frac{P_{kp}}{m_{TP}} - j_{TP}\right) + t_T \left(\vartheta_0 + \sum_{i=1}^{n} j_{TP\_i} t_{2\_TP\_i} - \frac{P_{kp}}{m_{TP}} t_1\right) - \frac{1}{2} \sum_{i=1}^{n} j_{TP\_i}\left(t_{1\_i}^2 - 2t_{1\_i} t_{2\_TP\_i} + 2t_{2\_TP\_i}^2\right) + \frac{P_{kp}}{2m_{TP}} t_1^2 + \vartheta_{0\xi} t_\xi \quad (28)$$

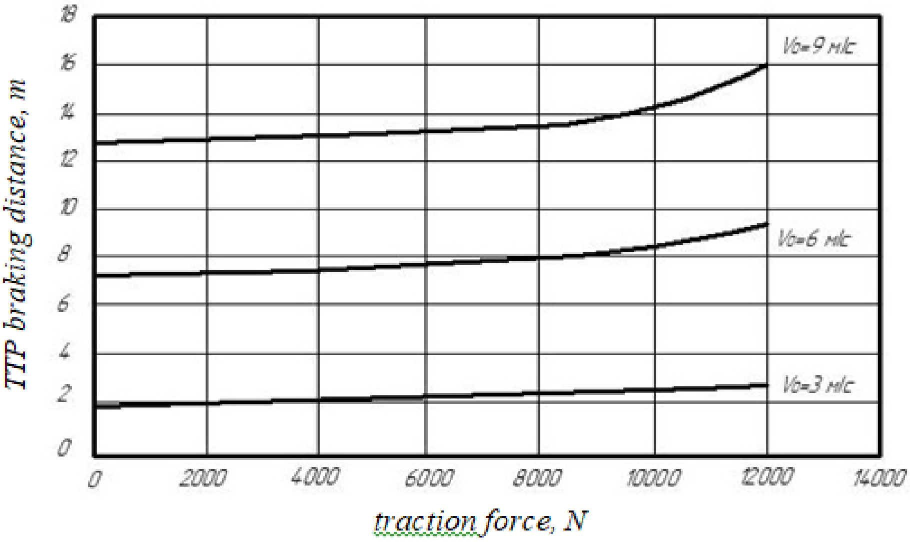

Fig. 4. Dependence of the brake track of the transport-technological train on the traction force

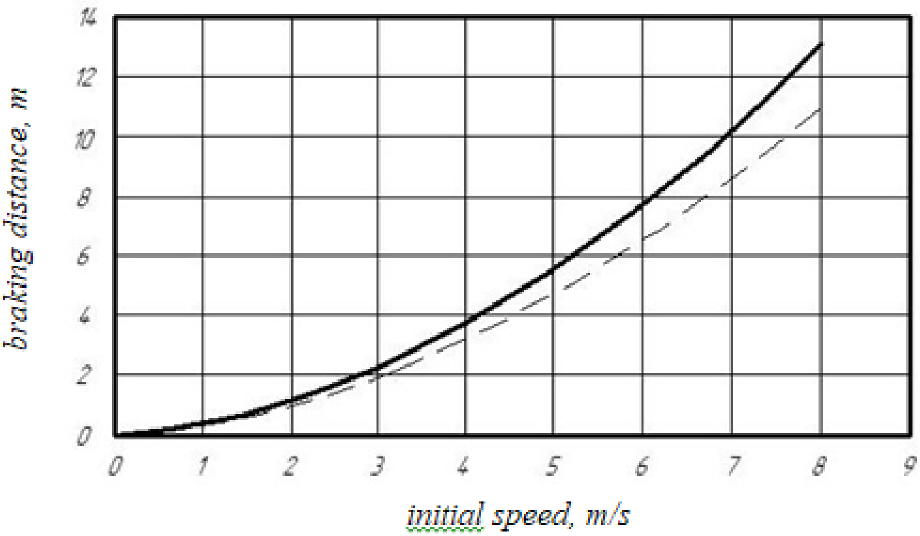

Fig. 5. Dependence of the brake track of the transport-technological train on the initial braking speed

5. Conclusions and prospects for further research

From the dependencies of the brake track on the traction force shown in Fig. 5, it can be seen that with its reduction from 11-12 kN to 7-8 kN, the brake track of the transport and technological train decreases to 26-29%.

To calculate the brake track of a transport and technological train using formula (26) (Fig. 5). If more accurate braking distance calculations are required, it is recommended to use formulas (13, 21) for mechanical (hydraulic) drive of tractor brakes or formula (14, 22) for pneumatic drive of brakes.

The analysis highlights the complex interactions within the "road-tire-vehicle-driver" oscillatory system, emphasizing the impact of dynamic and kinematic excitation from road irregularities and moving mass imbalances on transport performance and operator comfort. Considering both low- and high-frequency vibrations, the study underscores the importance of mathematical models for optimizing braking distances, traction forces, and vibration damping. Refining these models will enhance transport efficiency and operator well-being under diverse conditions.

References

1. Altomonte, A., Frasci, E., Di Ilio, G., Arsie, I., & Jannelli, E. (2023). Experimental testing of a heavy-duty Diesel engine at the dynamic test bench to assess the potential of regenerative braking. У *2023 IEEE International Workshop on Metrology for Automotive (MetroAutomotive)*. IEEE. DOI: https://doi.org/10.1109/metroautomotive57488.2023.10219129 [in English].
2. Rakha, H., Yue, H., & Dion, F. (2011). VT-Meso model framework for estimating hot-stabilized light-duty vehicle fuel consumption and emission rates. *Canadian Journal of Civil Engineering*, *38*(11), 1274–1286. DOI: https://doi.org/10.1139/l11-086 [in English].
3. Tucki, K. (2021). A Computer Tool for Modelling CO2 Emissions in Driving Tests for Vehicles with Diesel Engines. *Energies*, *14*(2), 266. DOI: https://doi.org/10.3390/en14020266 [in English].
4. Felez J., Lopez-Mendez A. (2021). Advanced steering stability controls for articulated vehicles by using bond graphs and model predictive control. Simulation Series 53, №3. International Conference on Bond Graph Modeling and Simulation, ICBGM 2021. San Diego, 10 November 2021. P. 98 – 110 [in English].
5. Borawski, A. (2020). EFFECTS OF INITIAL SPEED AND LOAD OF UNIVERSAL SEMITRAILER ON BRAKING SYSTEMS FRICTION PAIR HEATING. *Heat Transfer Research*, *51*(15), 1417–1428. DOI: https://doi.org/10.1615/heattransres.2020035553 [in English].
6. Samorodov, V., Kozhushko, A., & Pelipenko, E. (2016). Formation of a rational change in controlling continuously variable transmission at the stages of a tractor's acceleration and braking. *Eastern-European Journal of Enterprise Technologies*, *4*(7(82)), 37. https://doi.org/10.15587/1729-4061.2016.75402 [in English].
7. Xu, S., Zhang, X., Jiao, Y., Wei, L., He, J., & Zeng, X. (2024). Research on the Multi-Mode Composite Braking Control Strategy of Electric Wheel-Drive Multi-Axle Heavy-Duty Vehicles. *World Electric Vehicle Journal*, *15*(3), 83. DOI: https://doi.org/10.3390/wevj15030083\ [in English].
8. Ovsyannikov, S., Kalinin, E., & Koliesnik, I. (2019). Oscillation Process of Multi-support Machines When Driving Over Irregularities. У *Advances in Intelligent Systems and Computing* (с. 307–317). Springer International Publishing. DOI: https://doi.org/10.1007/978-3-030-19756-8_28 [in English].
9. Yan, Y., Chen, X., Wang, W., Hang, P., Chen, H., & Liu, J. (2022). Research on braking dynamics of multi-axle vehicle. *Journal of Physics: Conference Series*, *2246*(1), 012019. DOI: https://doi.org/10.1088/1742-6596/2246/1/012019 [in English].
10. Wu, X., He, R., Ge, H., & Chen, M. (2024). Electronic mechanical braking system executive mechanism design, calculation, and modeling based on dynamic control. *Frontiers in Mechanical Engineering*, *10*. DOI: https://doi.org/10.3389/fmech.2024.1368683 [in English].

ВПЛИВ ТИПУ ТЯГОВО-ЗЧІПНОГО ПРИСТРОЮ НА ПЕРЕРОЗПОДІЛ ГАЛЬМІВНИХ МОМЕНТІВ ПРИ ЗМІНІ ЕКСПЛУАТАЦІЙНИХ ТА КОНСТРУКТИВНИХ ФАКТОРІВ ПРИЧЕПА

Сучасний трактор є складною механічною системою, що складається з великої кількості елементів, об'єднаних в одне ціле різними видами зв'язків. Актуальність даної роботи полягає в *тому, при русі тракторного поїзда по різних дорожніх покриттях і бездоріжжю система шина-транспортний засіб-водій виступає в ролі коливальної системи. При цьому необхідно враховувати, що вся різноманітність збурюючих впливів враховується «подвійним» (динамічним і кінематичним) і «двочастотним» характером збудження. Те й інше пов'язано в основному з нерівностями дорожнього покриття і неврівноваженістю мас, що обертаються і поступово рухаються.*

Необхідно сказати, що практично всі дослідження вібрchoзахисту людини та транспортного агрегату зводилися до вивчення високочастотних коливань. Однак поряд із високочастотними коливаннями істотний вплив на людину-оператора та транспортний агрегат надають низькочастотні коливання.

Дослідження будь-якої коливальної системи включають два етапи:

1. Дослідження збурюючих впливів.

2. Дослідження роботи гасителів коливань.

Вивчення збурюючих впливів зводиться в основному до дослідження взаємодії мікропрофілю дороги і рушіїв транспортного агрегату, до математичного опису профілю дороги і одночасно математичного опису законів зміни відносного руху рушія. Слід зазначити, що вплив на досліджувані показники дорожнього тіла і закони руху надає швидкість руху, оскільки у своїй змінюються частотні і амплітудні властивості впливів. Стан дороги у свою чергу впливає на тягові зусилля, характер взаємодії дорожньої поверхні та рушія у силовому відношенні.

Таким чином, враховуючи особливості дослідження коливальної системи дорога – шина – транспортний засіб – водій, з метою визначення впливу параметрів руху транспортно-технологічного поїзда та характеристик дорожнього фону на експлуатаційні показники транспортно-технологічного поїзда загалом та оператора зокрема, необхідно скласти систему диференціальних рівнянь, що описує рух транспортно-технологічного поїзда з урахуванням типу застосовуваного тягово-причіпного пристрою.

Ключові слова: *транспортно-технологічний поїзд, динаміка, коливання, рушій, причеп.*

Ф. 28. Рис. 5. Літ. 10.

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