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ANALYSIS OF THE CONDITIONS FOR THE EXHAUSTION OF THE STABILITY MARGIN IN THE RAIL TRACK OF FREIGHT CARS WITH THREE-PIECE BOGIES

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Abstract:

The research on improvement of methodical approaches to definition of the probable reasons of infringement of conditions of stability of freight cars from derailment is carried out. Using a basic computer model of the dynamics of a freight car, the influence of the characteristics of the technical condition of their running gear and track on the indicators of empty cars stability from derailment was studied through the computational experiment.
The article presents the main statements of the research methodology, which provides the analysis of probable causes of derailment of freight cars by conducting a series of numerical experiments with logging the progress of calculations and saving the results. Factor analysis was used to interpret the calculated data with an assessment of each of the factors influence or their combination on the probability of derailment.
The developed procedure of the simulation experiment provides a step-by-step study of the freight cars derailment conditions, including factors structuring and ranking, development of experimental plan, calculating coefficients of wheel pairs resistance to derailment from rails, provided that the wheel flange rolls onto the rail head, and determining the degree of influence of relevant factors on the dynamic stability of cars from derailment. A comparative analysis of the stability of cars in rail tracks was performed using the introduced concept of the combined coefficient of stability of wheel pairs against derailment.
Determining the probable causes of car derailment is based on scanning the parameter field. The results of the parametric study revealed the degree of influence on the freight cars stability of running gear technical condition characteristics. In particular, it is determined that the most dangerous in terms of stability loss of empty cars in the track is the exceeding of the wedges of the vibration dampers.

Keywords: *freight cars, derailment, computer simulation, traffic safety, dynamic performance*

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1. Introduction

Railway accidents involving the derailment of rolling stock depend on many factors, both objective and subjective. Due to the combined action of many factors, some of which are not recorded by objective means of control during the movement of the train, the analysis of emergency situations is not always possible to identify and explain the cause of the derailment. At the same time, the assessment of traffic safety indicators according to existing methods does not reflect the actual conditions that increase the risks of derailment.

Railway safety as a key issue includes a wide range of components, among which a leading place belongs to the dynamics of vehicle motion (Ashtiani, I., et al., 2017; Burdzik, R., et al., 2017; Dusza, M., 2014; Garg, V.K., Dukkipati, R.V., 1984; Kardas-Cinal, E., 2013; Wickens, A.H., 2003). In the mechanical sense, the level of operational safety of railway vehicles is mainly determined by the margin of stability in the track (Domin, R., et al., 2017; Fan, Y-T., et al., 2006; Molatefi, H., 2016; Opala, M., 2016). Therefore, in the field of mechanics of rolling stock, the role of research work to study the course of dynamic processes that affect the emergencies occurrence conditions associated with rolling stock derailment, remains acute at all stages of railway transport development (Domin, R., et al., 2019; Iwnicki, S., et al., 2015; Malcolm, C., 2016; Saviz, M.R., 2015; Wilson, N., et al., 2011).

A characteristic feature of the spatial oscillations of vehicles, the movement of which is directed by the rail track, is the tendency under certain conditions to self-excitation of auto-oscillations (Mazilu, T., 2009). The lowest value of the speed at which unquenchable lateral oscillations of a railway vehicle occur, is called the critical speed of hunting V_{cr}. Transverse hunting oscillations of bogies when moving at speeds exceeding the critical, cause excessive lateral forces of the wheels on the track, increase the damage of freights sensitive to dynamic loads, and lead to additional damage to rolling stock and railway infrastructure. In addition, the extra energy of the locomotive is spent on maintaining a constant speed of the train composed of cars, the movement of which is complemented with self-oscillations. Thus, the critical hunting velocities determine the limits of threshold changes in the dynamic properties of railway vehicles.

Self-excitation of lateral oscillations of railway vehicles is caused by their loss of stability of undisturbed motion. Effective use of methods of mechanical stability theory in relation to studies of the dynamics of the movement of railway vehicles, in particular the first approximation of A.M. Lyapunov (Lyapunov, A.M., 1956), first carried out by academician V.A. Lazaryan (Lazaryan, V.A., 1964).

2. Dynamic phenomena contributing to emergencies related to rolling stock derailment occurrence

Obviously, each transport event is associated with a coincidence of a number of adverse circumstances, among which, however, there is always a leading cause. The low freight cars stability margin from derailment is most often caused by their unsatisfactory dynamic properties which are mainly explained by design features and a technical condition of running gear (Galiev, I.I., et al., 2011).

According to the results of numerous studies and investigations of transport accidents, it turns out that the objects of emergency situations are increasingly freight cars in an empty state (Ge, X., et al., 2018; Ermakov, V.M., Pevzner, V.O., 2002). Empty cars with a high center of mass (bunker cars, tank cars, etc.) are most prone to loss of stability in the rail track.

2.1. Resonant movement modes of freight car in empty state

It is known that the spring suspension of freight cars of 1520 mm gauge in the empty state can partially or completely lose its damping properties due to the weakening or complete exclusion from the operation of wedge vibration dampers caused by the so-called exceeding of the wedges. At the same time rigidity of a spring suspension bracket of the bogie can decrease in 1.4 times. This situation leads to a decrease in the natural frequencies of the car, and hence a decrease in the speed at which the resonant mode occurs.

Periodic perturbations that cause resonant modes of rolling stock are associated with both periodic irregularities of the track and the existing defects on the rolling surfaces of car wheels. Therefore, in the spectrum of perturbations acting on a moving car, there are always components with the frequency of rotation of the wheel pairs. These components, even with the permissible defects of the wheels, are sufficient for the development of resonant phenomena, when the speed of the car reaches a critical value when the wheel pair rotation speed f_w with one of the natural frequencies f_i . These velocities are called resonant. So the resonant speed V_r is calculated by expression $V_r = L_w \cdot f_i$, where L_w – is the length of the wheel rolling circle.

Fig. 1 shows a diagram for determining the resonant velocities. Here, rays I and II show the dependences of the speed of wheel pairs with the full (I) and limiting (II) thickness of the wheel rims. Resonant velocities $V_r^{(i)}$ (i = 1…4) are at the points of intersection of lines I and II with the horizontal natural frequencies of vertical oscillations f_v and f_v^* respectively at nominal and reduced (due to the exclusion of underwedge springs) stiffness of the suspension. Thus, under operating conditions, the resonant velocities can take values in the range from $V_r^{(1)}$ to $V_r^{(4)}$.

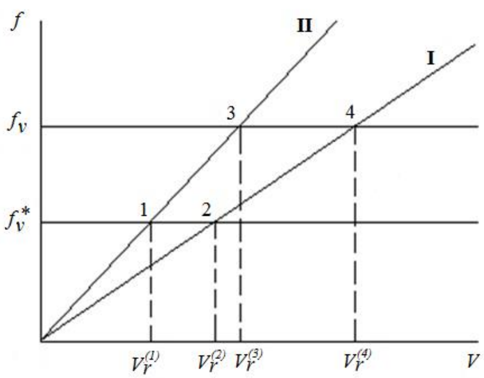

Fig. 1. Scheme for determining resonant velocities

The most dangerous for empty cars is the $V_r^{(1)}$ - $V_r^{(2)}$ speed range, which corresponds to cases of insufficient or absent damping of oscillations (Fig. 1). In the resonant modes of oscillations of bouncing and pitching at the moments of full unloading of wheels in case of horizontal forces cross there is a real threat of derailment.

The natural frequencies of oscillations of the freight car bodies of some types in the empty state are shown in Table 1. Oscillation frequencies of bouncing f_b and f_b^*, pitching f_p and f_p^* and rolling f_r and f_r^* calculated according to two values of the stiffness of the spring suspension, corresponding to the nominal stiffness of the spring sets (numerator) and reduced due to the exclusion from the normal operating state of the wedge vibration dampers (denominator).

Table 1. Natural frequencies of freight car bodies oscillations in Hz at the nominal and reduced spring suspension stiffness

Oscillation type	Car types			
	gondola car	covered car	hopper car	tank car
Jumping	5,61/4,74	5,22/4,41	5,50/4,60	5,16/4,36
Galloping	6,61/5,59	6,42/5,43	7,60/6,46	5,80/4,90
Lateral oscillation	5,02/4,24	4,00/3,38	4,14/3,50	3,99/3,36

According to the calculated frequencies, the resonant speeds of the considered types of cars with new and worn wheels are determined. For example, Fig. 2 shows the range of resonant velocities of the empty gondola car at full and maximum thickness of the wheel rim, respectively, at nominal and reduced stiffness of the suspension, taking into account the calculated frequencies (Table 1). Lines 1, 2, 3 correspond to frequencies f_b, f_p and f_r and lines 4, 5, 6 – to frequencies f_b^*, f_p^* and f_r^*. Dependences of frequencies of rotations of wheelsets on full $f_f(v)$ and limit $f_{ut}(v)$ the wheel rims thicknesses that corresponding to the wheel radii of 0.475 and 0.425 m are represented by graphs 7 and 8.

As can be seen from the graphical data, the values of resonant velocities for the gondola car are in the range of 41 – 71 km/h. According to similar calculations, the values of resonant velocities for other types of cars were obtained. Depending on the models of freight cars, the condition of the wedge dampers and the thickness of the rims of their wheels, the resonant speeds can vary in wide ranges: 32 – 69 km/h for a covered car; 33 – 82 km/h for a hopper car; 32 – 62 km/h for a tank car. Thus, it can be stated that these values of resonant speeds are covered by the operating speeds of freight trains on the railways of 1520 mm. From the point of view of traffic safety, this fact requires increasing the requirements for monitoring the technical condition of the spring suspension and wheelsets.

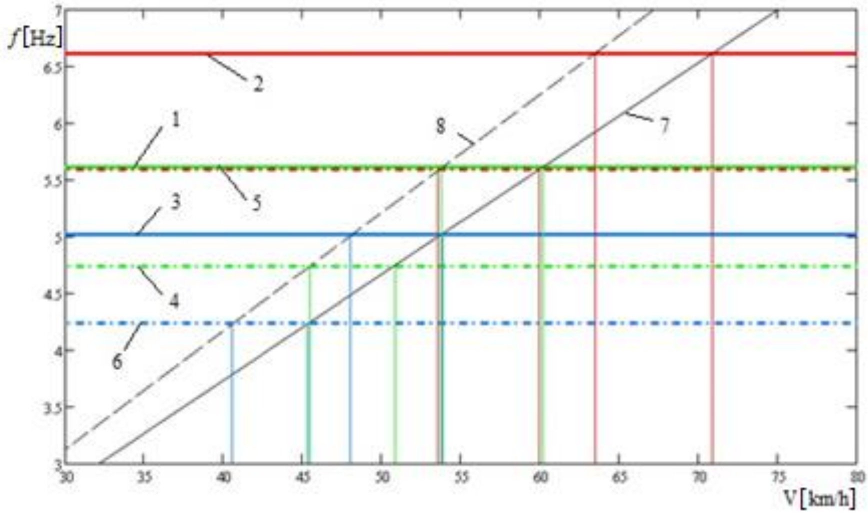

Fig. 2. Resonant velocities of the gondola car

2.2. Stability of rolling stock in rail track

Freight cars of 1520 mm track are equipped exclusively with three-element bogies, the characteristic feature of which, from the point of view of mechanics, is the saturation of units with open pairs of dry friction. This circumstance significantly complicates the analysis of stability conditions in the rail track of cars as essentially nonlinear systems.

Due to the presence of open pairs of dry friction in the combinations of bearing elements of the running gear between themselves and the body, it is possible to stop in the relative movements of individual bodies of the system, which includes the model of the freight car. Thus, the system may lose degrees of freedom and move from one structural state to another. Therefore, the original design system of the car can be considered as a system with a variable structure. The number of possible structural states of such system is equal to 2^i (i – number of friction nodes).

Based on the concept of fundamental variability of the output system, which simulates the dynamic behavior of a freight car, a method for determining critical velocities using linearization of discrete systems with dry friction units was proposed (Diomin, Yu.V., et al., 1994). The essence of this method is to replace the original nonlinear system with l linear subsystems ($l = 2^i$). Each of l subsystems corresponds to one of the possible states of the original nonlinear system. Such subsystems are built in accordance with the structural changes of the original system due to the alternate closure of connections with dry friction.

When constructing linear subsystems, the main thing is to determine the parameters of viscous friction, which replaces dry friction in open joints. According to the developed method of formation of linear subsystems for determining the coefficient of equivalent viscous resistance β^l_{i-j} in $i-j$ connection of a multi-mass self-oscillating system is carried out according to a formula similar to that used by S.P. Tymoshenko in the study of forced oscillations of the oscillator with dry friction (Weaver, W. Jr., et al., 1990). Regarding the model of operation of the body support devices on the bogies during their mutual turns, the mentioned formula has the form

$$\beta^l_{1-i} = 4W / \pi \cdot \Delta\psi_{1-j} \cdot \omega, \tag{1}$$

where: $\Delta\psi_{1-j}$ – amplitude values of angles of mutual turns of a body and bogies; ω – frequency of self-oscillations.

Values of $\Delta\psi_{1-j}$ and ω are determined by an iterative method based on the step-by-step solution and analysis of the complete problem of eigenvalues of matrices of coefficients of equations of motion of subsystems of type $\dot{x} = A^{(l)} \cdot x$. The stability of possible states of the system (subsystems) is estimated. The value is taken as an indicator of stability is $h_{\max}^{(l)} = \max(\mathrm{Re}\,\lambda_i^{(l)})$, where $\lambda_i^{(l)}$ - eigenvalues of matrices $A^{(l)}$. A critical speed is determined for each system $V_{cr}^{(lm)}$ as the value of speed V at $h_{\max}^{(l)} = 0$, or $h_{\max}^{(l)}(V_{cr}^{(l)}) = 0$. The smallest of the critical velocities of the obtained range is the speed of movement V_{so}, in which there are self-oscillations in the studied system with dry friction, thus

$$V_{so} = \min\left|V_{cr}^{(1)}, V_{cr}^{(2)}, \ldots, V_{cr}^{(l)}\right| . \tag{2}$$

The indicators of stability of the least stable of a number of subsystems, which approximated the original system, determine the conditions of self-oscillations of the studied railway vehicle. Thus, the method of structural linearization allows to extend powerful methods of linear algebra to a class of systems that are not fundamentally linearizable.

3. Study of influence on stability of cars from derailment of running gears elements technical condition characteristics

A basic computer model of the dynamics of a four-axle car with three-element bogies, developed with the help of a software package UM (Cherniak, A., 2013; Pogorelov, D.Yu., 2005) was used to study and identify the probable reasons for the derailment of freight cars. This model allows to obtain modifications of dynamic models of freight cars of the main types (gondola cars, covered cars, hopper cars, tank cars), which differ in the design of bodies, but have typical running gears (Domin, R., et al., 2016). The computer model of the dynamics of a freight car provides a detailed description of the technical condition of the running gears of freight cars that have derailed. This makes it possible to adjust the specialized parameters of the model, in particular those that characterize the technical condition of the running gears of the derailed car, and the characteristics of the derailment track section.

3.1. Estimation of stability margin of wheel sets from derailment

A necessary step in determining the prerequisites for derailment is to study the influence of certain factors on the characteristics of the dynamic processes that accompany the movement of the car. The assessment of dynamic characteristics should be the selected indicators of the margin of stability of wheel sets from derailment under the condition of rolling the wheel flange on the rail head, which comprehensively characterize the combination of both horizontal and vertical forces acting simultaneously on the wheel of each wheel set. On 1520 mm gauge railways, the main indicator of rolling stock safety is the so-called coefficient of stability of wheel set from derailment, provided that the wheel flange rolls onto the rail head. (Standards, 1996). Coefficient of stability of wheel against derailment k_{dr} when moving the car with the maximum speed on the straight track of good condition with combinations of deviations in the plan, skews and sags allowed, is calculated by the formula:

$$k_{dr} = \frac{tg\beta - \mu}{1 + \mu \cdot tg\beta} \cdot \frac{P_v}{P_h} \geq (k_{dr})_{\lim} , \tag{3}$$

where: β – the angle of inclination to the horizon of the generating conical surface of the wheel crest, for the wheels 1520 mm gauge cars β = 60º; μ – coefficient of sliding friction of interacting surfaces of wheels and rails (in calculations it is considered μ = 0,25); P_v – vertical component of the forces impacting from the wheels on the rails; P_h – the horizontal component of the forces of interaction of the wheel with the rail, which impacts simultaneously with the force P_v. For freight cars of 1520 mm gauge the maximum allowable value is $(k_{dr})_{\lim} = 1.3$.

Execution of the analysis of probable reasons of derailment of freight cars requires carrying out of a series of numerical experiments, according to the plan made in advance, with course of calculations logging and preservation of results. The general procedure of the simulation experiment to study the of freight cars derailment conditions is reduced to the following stages:

- structuring and ranking of factors influencing the stability of cars in the track;
- conducting an experimental plan;
- calculation of stability coefficients of from derailment of wheelsets on condition of wheel flange rolling on a rail head;

– determining the degree of relevant factors influence on the dynamic stability of cars from derailment.

The probable causes of car derailing are determined on the basis of computer simulations based on a scan of the parameters field. This method provides complete information about the objective function within the defined parameter sets. The number of computer experiments when scanning is calculated as $N = m^k$, where k is number of varying factors, m is the number of levels at which each factor varies. Depending on the number of factors and the levels of each of them selected for the scan, the number of options is growing rapidly. Thus, the time of scanning and computational costs, increases significantly.

In the task for scanning, in addition to the speed V, as study factors the following characteristics of the undercarriage were selected: *fp* – friction coefficients in center bearing nodes; *wp* – dislocation of center bearing nodes in the longitudinal direction; *fs* – friction coefficients in the side slides; *kl* – exceeding of wedges of bogies; *wb*1 and *wb*2 – clearances in the longitudinal direction between the axle boxes and side frames, respectively, for the first and the second bogies. For each of the selected factors, the determined levels are shown in Table 2.

Table 2. Factors levels

Levels	*fp*	*wp* [m]	*fs*	*kl* [m]	*wb*1[m]	*wb*2[m]
1	0,1	0	0,1	0	0	0
2	0,4	0,005	0,4	0,015	0,004	0,004
3					0,008	0,008

The full-factorial plan of the experiment as a plan of the conducted experiments takes into accounts all possible combinations of levels of each factor. According to the identified factors, a full-factorial plan of the experiment with the total number of variants 144 was formed (Table 3). An experiment with such a plan allows us to quantify the effects of both individual factors and the interaction of factors (Adler, Yu., et al, 1971).

Table 3. Estimated variants for the running gears technical condition

Variants numbers	*fp*	*wp* [m]	*fs*	*kl* [m]	*wb*1[m]	*wb*2[m]
1	2	3	4	5	6	7
1/2/3	0,1	0	0,1	0	0	0/0,004/0,008
4/5/6	0,1	0	0,1	0	0,004	0/0,004/0,008
7/8/9	0,1	0	0,1	0	0,008	0/0,004/0,008
10/11/12	0,1	0	0,1	0,015	0	0/0,004/0,008
13/14/15	0,1	0	0,1	0,015	0,004	0/0,004/0,008
16/17/18	0,1	0	0,1	0,015	0,008	0/0,004/0,008
19/20/21	0,1	0	0,4	0	0	0/0,004/0,008
22/23/24	0,1	0	0,4	0	0,004	0/0,004/0,008
25/26/27	0,1	0	0,4	0	0,008	0/0,004/0,008
28/29/30	0,1	0	0,4	0,015	0	0/0,004/0,008
31/32/33	0,1	0	0,4	0,015	0,004	0/0,004/0,008
34/35/36	0,1	0	0,4	0,015	0,008	0/0,004/0,008
37/38/39	0,1	0,005	0,1	0	0	0/0,004/0,008
40/41/42	0,1	0,005	0,1	0	0,004	0/0,004/0,008
43/44/45	0,1	0,005	0,1	0	0,008	0/0,004/0,008
46/47/48	0,1	0,005	0,1	0,015	0	0/0,004/0,008
49/50/51	0,1	0,005	0,1	0,015	0,004	0/0,004/0,008
52/53/54	0,1	0,005	0,4	0,015	0,008	0/0,004/0,008
55/56/57	0,1	0,005	0,4	0	0	0/0,004/0,008
58/59/60	0,1	0,005	0,4	0	0,004	0/0,004/0,008
61/62/63	0,1	0,005	0,4	0	0,008	0/0,004/0,008
64/65/66	0,1	0,005	0,4	0,015	0	0/0,004/0,008
67/68/69	0,1	0,005	0,4	0,015	0,004	0/0,004/0,008
70/71/72	0,1	0,005	0,1	0,015	0,008	0/0,004/0,008
73/74/75	0,4	0	0,1	0	0	0/0,004/0,008
76/77/78	0,4	0	0,1	0	0,004	0/0,004/0,008

Variants numbers	*fp*	*wp* [m]	*fs*	*kl* [m]	*wb*1[m]	*wb*2[m]
79/80/81	0,4	0	0,1	0	0,008	0/0,004/0,008
82/83/84	0,4	0	0,1	0,015	0	0/0,004/0,008
85/86/87	0,4	0	0,1	0,015	0,004	0/0,004/0,008
88/89/90	0,4	0	0,1	0,015	0,008	0/0,004/0,008
91/92/93	0,4	0	0,4	0	0	0/0,004/0,008
94/95/96	0,4	0	0,4	0	0,004	0/0,004/0,008
97/98/99	0,4	0	0,4	0	0,008	0/0,004/0,008
100/101/102	0,4	0	0,4	0,015	0	0/0,004/0,008
103/104/105	0,4	0	0,4	0,015	0,004	0/0,004/0,008
106/107/108	0,4	0	0,4	0,015	0,008	0/0,004/0,008
109/110/111	0,4	0,005	0,1	0	0	0/0,004/0,008
112/113/114	0,4	0,005	0,1	0	0,004	0/0,004/0,008
115/116/117	0,4	0,005	0,1	0	0,008	0/0,004/0,008
118/119/120	0,4	0,005	0,1	0,015	0	0/0,004/0,008
121/122/123	0,4	0,005	0,1	0,015	0,004	0/0,004/0,008
124/125/126	0,4	0,005	0,1	0,015	0,008	0/0,004/0,008
127/128/129	0,4	0,005	0,4	0	0	0/0,004/0,008
130/131/132	0,4	0,005	0,4	0	0,004	0/0,004/0,008
133/134/135	0,4	0,005	0,4	0	0,008	0/0,004/0,008
136/137/138	0,4	0,005	0,4	0,015	0	0/0,004/0,008
139/140/141	0,4	0,005	0,4	0,015	0,004	0/0,004/0,008
142/143/144	0,4	0,005	0,4	0,015	0,008	0/0,004/0,008

According to these options, the dynamics of the gondola car in the empty state were calculated by computer simulation. The comparative analysis of the received results is carried out on the combined indicator of stability from derailing of the gondola car k_{dr0}, which was calculated as the smallest of the minimum values derailment stability margin coefficients k_{dr1}, k_{dr2}, k_{dr3}, k_{dr4} respectively for each wheelset (min min). Fig. 3 shows the values of the combined coefficient of resistance to derailing k_{dr0} at speeds of the gondola 60, 70, 80 and 90 km/h on a straight track section of a satisfactory condition.

At speeds of 60 and 70 km/h, all calculated values of the coefficient k_{dr0} are higher than the allowable level. However, the margin of stability significantly depends on changes in parameters and characteristics of the technical condition of the running gears. For example, as obtained in the calculation variants 82-90 and 100-108, the exceeding of the wedges worsens the situation with derailing of the gondola only in combination with excessive friction in both central bearing nodes and on side bearings.

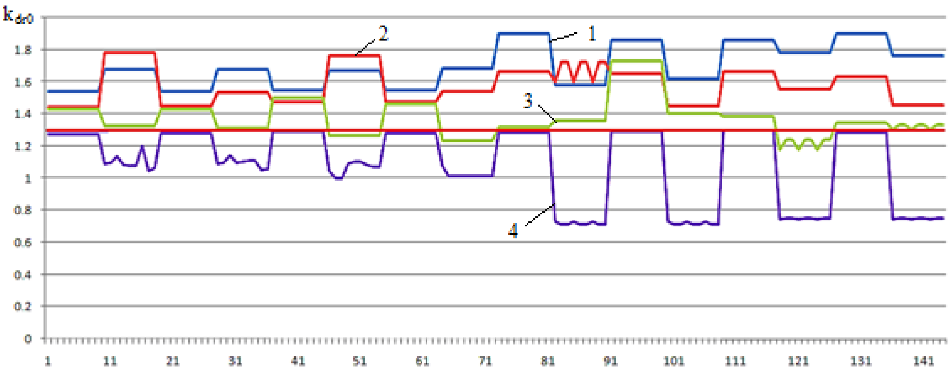

Fig. 3. Combined stability margin indicator k_{dr0} for all calculation options (curves 1, 2, 3, 4 correspond to the values V = 60, 70, 80, 90 km/h)

At a speed of 80 km/h, there is a depletion of stability margin in variants with a combination of three factors: increased friction and longitudinal wear in the central bearing nodes, along with exceeding of the wedges (variants 64-72, 118-126). For other calculation options, the obtained values of the coefficient k_{dr0} are above the maximum allowable level.

According to the results of calculations at V = 90 km/h it turns out that, depending on the options for a combination of factors, the values of k_{dr0} are either on the verge of permissible, or significantly lower. So, in the calculated variants corresponding to cases of movement of the gondola car with the excluded dampers of fluctuations (10-18, 28-36, 46-54, 64-72, 82-90, 100-108, 117-126, 136-144), gained values of k_{dr0} almost two times lower the permissible level when the wedge system is in good condition (1-9, 19-27, 37-45, 55-63, 73-81,91-99,109-117, 127-135). Thus, at a speed of 90 km/h, the situation of derailing the gondola becomes possible with a high degree of probability.

3.2. Stability of cars of different types in a rail track

Calculations for gondola car, covered car, hopper car and tank car were performed to determine the effect on the stability of the freight car type. In this case, given the fact that the clearances in the longitudinal direction between the axle boxes and side frames affected the resistance to the derailment of the empty gondola less than other changes in the technical condition, only 16 options were considered calculating the numbers $N = 5 + 9i$, where $i = 1\ldots15$ (Table 3). Fig. 4 and 5 show the values of the combined coefficients of stability k_{dr0} for four types of freight cars in the empty state at speeds of 60 and 80 km/h, respectively. Here the lines connecting the calculated values k_{dr0} are marked as follows: 1 – for gondola car; 2 – for covered car; 3 – for the hopper car; 4 – for the tank car.

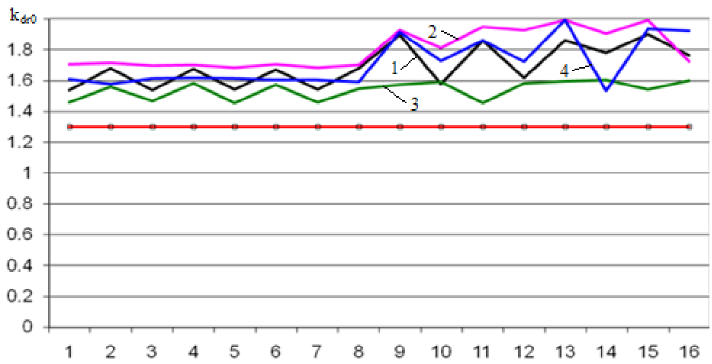

Fig. 4. The value of stability margin indicator k_{dr0} at V = 60 km/h

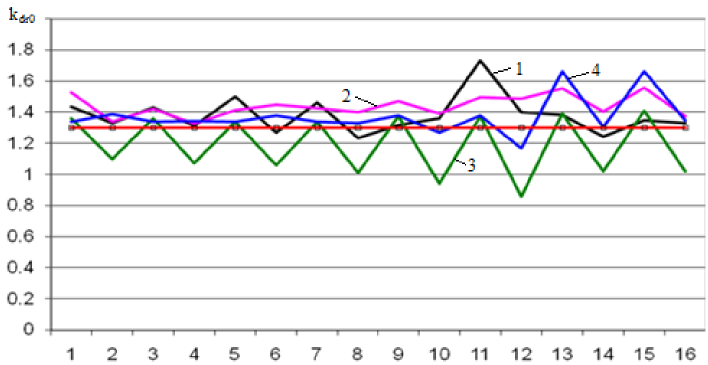

Fig. 5. The value of stability margin indicator k_{dr0} at V = 80 km/h

According to Fig. 4 data the stability margin level from derailing at a speed of 60 km/h for the specified types of freight cars is higher than the allowable value for all considered options. The covered wagon has preferably the best values of the stability margin, while the lowest values of k_{dr0} are obtained for the hopper car. In general, a greater influence of the technical condition of the bogies on the margin of safety is observed for gondola car and hopper car, at the same time, the increase in friction in cantral bearing nodes is reflected in the values of k_{dr0} for the covered car and the tank car.

According to the calculated data (Fig. 5) it turns out that at a speed of 80 km/h the values of the coefficients k_{dr0} are above the permissible value has only a covered car, and for options 14 and 32 (Table 3) the level of k_{dr0} values for a covered car is almost coincides with the allowable value. Depending on the design option for the condition of the running gears of the gondola car and tank car have a margin of stability from derailment or above the allowable value, or slightly below it. The safety margin of the hopper car for almost all variants is below the allowable level, and the technical condition of the running gears significantly affects the k_{dr0} value, changing it from 1.4 for variants with a working wedge system to 0.86 when the wedges are off (free).

According to the relevant calculations, the level of k_{dr0} values, obtained at a speed of 90 km/h for cars of the considered types below the allowable value for almost all calculated options. Especially low values of k_{dr0} were obtained for gondola car and covered car in variants when the increased friction in central bearing nodes and inoperable wedge system were simulated.

4. Conclusions

From the analysis of the results of computer-simulated studies of the derailment resistance of freight cars in the empty state, the conclusions are the following:

1. The dynamic phenomena that contribute to the threat of rolling wheels on the rails heads with the subsequent derailing of the wheelsets of freight cars, should include resonant modes, when at certain speeds in the operating range the least defects on the rolling surfaces of the wheels lead to intense vertical oscillations until the wheels are completely unloaded. Depending on the condition of the wedge system and the wear of the wheel rims, resonant (critical) speeds are in a fairly wide range of velocities - from 32 km/h to 82 km/h;

2. To ensure adequate display of the technical condition of the car as the initial data in the dynamic model, the parameters characterizing the deviation from the running gears normal technical condition are selected. The significance of the considered factors is established by means of the factor analysis of cars stability margin indicators. In particular, it is determined that the most dangerous in terms of the empty car in the track stability loss is the exceeding of the vibration dampers wedges;

3. The proposed procedure of reproduction by computer simulation of the situation related to the freight car derailment provides an opportunity to identify the most likely causes of derailment. This approach will deepen the search for investigations into the causes of traffic accidents and will help increase the reliability of predicted estimates of dynamic indicators of train safety.

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