N E W S
OF THE NATIONAL ACADEMY OF SCIENCES OF THE REPUBLIC OF KAZAKHSTAN
SERIES OF GEOLOGY AND TECHNICAL SCIENCES
ISSN 2224-5278
Volume 1, Number 433 (2019), 151 – 162 **https://doi.org/10.32014/2019.2518-170X.19**

UDC 625.001.05

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THE METHOD OF LIMITING SPEED WHEN PASSING TURNOUTS OF RAILWAY VEHICLES WITH BOGIES OF MODEL ZK1

Abstract. To improve the safety of freight trains traffic, regulatory documents of railway administrations establish the maximum permissible speeds when passing small radius curves and, including, transferable curves of switches of the most common brands. Transferable curves have a number of features – the absence of transferable inserts, elevation of the outer rail, blind intersections of the combined track, which causes the need for a sharp reduction in speed when entering the station through the transferable curve of the directional on the side track. To determine the maximum speeds, it is necessary to take into account the vertical, horizontal and transverse forces transmitted from the crew to the cores of the crosspieces of the turnouts, the coefficient of vertical dynamics, the value of the frame forces arising in the crews, in order to determine the ratio of these forces to the static load and the level of dynamic impact on the turnouts.

The task is solved by a combined method: analytical-computational and experimental. To solve the problem, an analytical method is used to draw up a calculation scheme, which takes into account the stiffness of spring sets, friction coefficients, angles of rotation of the body and wheel sets with respect to the transverse, vertical and longitudinal axes. On the basis of the calculation scheme and the assumptions, using the method of d'alembert were the equations of the second order. Additional dynamic forces of reaction and communication spring kits recorded through an additional equation. The solution of the equations was carried out by the numerical method of step-by-step integration. The experimental phase of the work was carried out on the stage of the Bel station of the Shuya branch of the road. Initially, the dynamic parameters of the vertical dynamics coefficients and frame forces arising during the passage of switches with different speeds were determined. Measurements of the level of impact and check the stresses in the edges of the foot rail of the switch. The estimated values of the processed processes were defined as the most probable values of the measured values for each individual velocity. The estimated values were estimated with a probability of 0.9985.

Keywords: railway carriage, railroad switch configuration, frame strength, stability, dynamic performance.

One of the criteria for establishment of the allowed speeds are the sizes of the vertical and horizontal forces which are transferred from wheels of crews to cores of blunt and sharp frogs.

The solution of the put objective was carried out by means of the spatial calculated scheme "carriage-way" submitted in figure 1. The scheme allows to investigatethe interaction of a way and the rolling stock in the vertical and horizontal planes in cases when roughnesses are both on one, and on both rail threads.

The carriage is presented by the four-axis freight gondola car of model 12-9920 manufacturing in the People's Republic of China on carts of model 18-9996 by ZK1 type in the calculated scheme, with conic cassette bearings. The movement of one cart of the gondola carisconsideredin details as the mutual influence of wheels of various carts is insignificant.

In work [1] it is shown that the division of mass of a wheel into various quantity of elements is not affectedon the sizes of dynamic forces. It allows to consider a wheel in the form of one mass.The transfer curveis simulated in the form of the concentrated masses connected by elastic connections and specified to the points of contact of wheels with rail threads. The lateral impact on a curve of the switch transfer is considered by movements of rail threads and elements of crews in the vertical and horizontal planes and the corresponding rigidity.

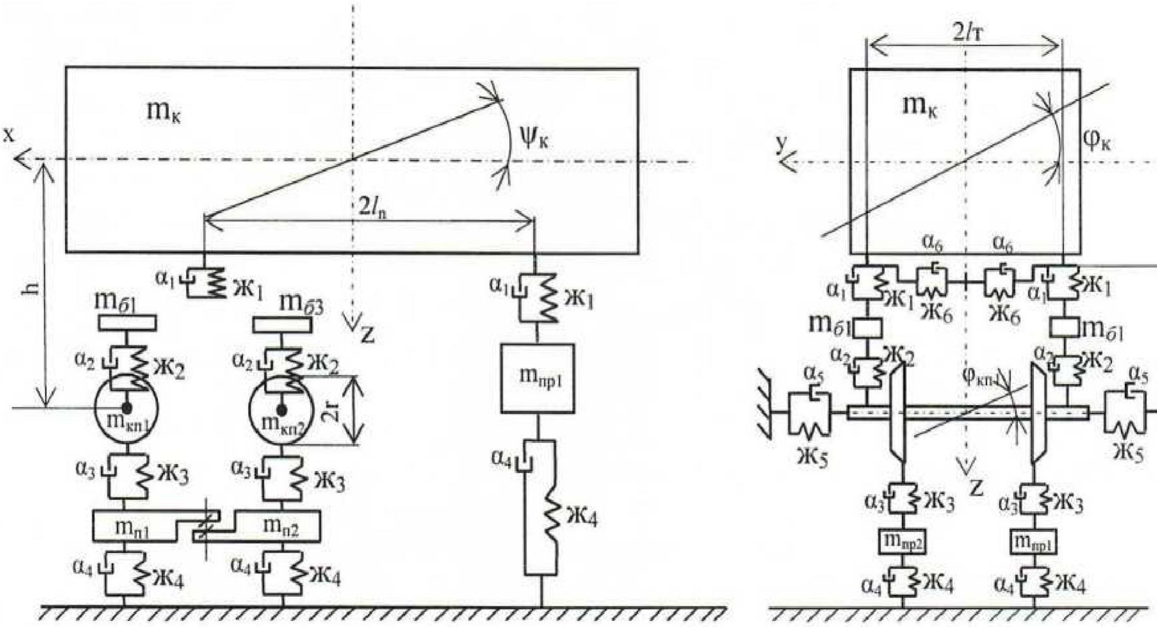


Figure 1 –The calculated scheme of системы «crew-ay» system

In the calculated scheme the following designations are accepted: $m_к$ - the mass of a cushioned part of the car; $m_{бi}$ - the mass of the wall falling on one wheel; $m_{кпj}$ - the mass of the wheel couple; $n_{пi}$ - the mass of the way specified to i-of a cart wheel; $ж_1$ - rigidity of springs of spring sets; α_1 - coefficient of viscous friction of spring knots; $ж2$ - rigidity of contact of a sidewall and wheel; α_2 - coefficient of viscous friction on contact of a sidewall and wheel; $ж3$ - contact rigidity of a wheel and way; α_3 - coefficient of viscous friction on contact of a wheel and way; $ж4$ -rigidity of the basis; α_4 - coefficient of viscous friction of the basis; $ж5$ - in pairs contact rigidity of rail threads; α_5 - coefficient of viscous friction in the place of contact of a wheel and rail thread in the cross direction; $ж_6$ - flexural rigidity of springs of a spring set; a_6 - coefficient of viscous friction of spring sets at a bend; $m_{прi}$, - the specified mass of the second cart falling on i- rail thread and relating to it the concentrated mass of a way; $\psi_к$, $\theta_к$, $\varphi_к$ - angles of rotation of a body concerning cross, vertical and longitudinal axis; $\varphi_{кп1,2}$ - angles of rotation of wheel couples concerning the axis parallel to a longitudinal axis of thecarriage; x, *y, z* - respectively, longitudinal cross and vertical movements; P_i - additional dynamic forces in the corresponding elements of «carriage-way» system.

The accepted calculated scheme consists of 13 solid bodies and has seventy eight degrees of freedom.

We make imposing and we find out connections:

$$\left.\begin{array}{c} x_к = x_{б1} = x_{б2} = x_{б3} = x_{б4} = x_{кп1} = x_{кп2} = x_{п1} = x_{п2} = x_{п3} = x_{п4} = x_{пр1} = x_{пр2} = 0 \\ y_{п1} = y_{п2} = y_{п3} = y_{п4} = y_{пр1} = y_{пр2} = 0 \\ \psi_{кп1} = \psi_{кп2} = \psi_{б1} = \psi_{б2} = \psi_{б3} = \psi_{б4} = \psi_{п1} = \psi_{п2} = \psi_{п3} = \psi_{п4} = \psi_{пр1} = \psi_{пр2} = 0 \\ \varphi_{б1} = \varphi_{б2} = \varphi_{б3} = \varphi_{б4} = \varphi_{п1} = \varphi_{п2} = \varphi_{п3} = \varphi_{п4} = \varphi_{пр1} = \varphi_{пр2} = 0 \\ \theta_{б1} = \theta_{б2} = \theta_{б3} = \theta_{б4} = \theta_{п1} = \theta_{п2} = \theta_{п3} = \theta_{п4} = \theta_{пр1} = \theta_{пр2} = 0 \\ \theta_{кп1} = \theta_{кп2};\ \psi_{кп1} = y_{б1} = y_{б2};\ y_{кп2} = y_{б3} = y_{б4}; y_{кп1} = y_{кп2}; \end{array}\right\} \quad (1)$$

There were 21 degrees of freedom: $z_к$, $z_{б1}$, $z_{б2}$, $z_{б3}$, $z_{б4}$, $z_{кп1}$, $z_{кп2}$, $z_{п1}$, $z_{п2}$, $z_{п3}$, $z_{п4}$, $z_{пр1}$, $z_{пр2}$, $y_к$, y_m, $\psi_к$, $\varphi_к$, $\varphi_{кп1}$, $\varphi_{кп2}$, $\theta_к$, θ_m;гдеу$_m$=$\varphi_{кп1,2}$ и θ_m=$\theta_{кп1,2}$.

In the calculated scheme the following assumptions are accepted:

- forces arising in connections are functions of the compressed spring and speeds of compression of shock-absorbers;
- rigidity between sidewalls and wheels are defined by deformations of sidewalls, axle-box knots and necks of axes of wheel couples and also deformations in a zone of their contacts;

- the rigidity between wheels and the specified mass of a way is defined by deformations of wheels and frogs (necks of rails) and also deformations in a zone of contact of wheels and frogs [2, 3];
- forces operating in the contact plane in the presence of a gap between the crests of wheels and rail threads are determined by the theory of pseudo-sliding [4, 5]; after the choice of a gap there appears the cross forces of interaction depending on the size and speed an space of rail threads [6, 7]. At the same time the space of rail threads take place, both in the presence of gaps, and at their absence;
- the connection between the rail threads is carried out only through a wheel couple;
- the connection between the specified mass of a way is carried out by introduction of coefficient P which considersmutual influence of the adjacent wheels of the cart on each rail thread. This connection is expressed by dependence:

$$P_i = \beta \cdot P_{i+1} \tag{2}$$

where P_i - force operating on the considered specified mass of a way; β - the coefficient of influence of the adjacent wheels considering the rigidity of railways; P_{i+1} - force which is transferred on the adjacent one with considered the mass of a way.

- there are no gaps between axle boxes and jaws of sidewalls, i.e. it is assumed that the angles of turns of axes of wheel couples and an axis of carts are equal: $\theta_{кп1}=\theta_{кп2}=\theta_m$.

Proceeding from the calculated scheme and considering the given assumptions, on the basis of Dalamber's principle we will write down the differential equations of «carriage-way» system:

$$\left.\begin{array}{c}
m_к \ddot{z}_к = -P_1 - P_2 - P_{19} - P_{20};\\
J_к^x \ddot{\varphi}_к = P_1 l_1 - P_2 l_1 + P_{19} l_1 - P_{20} l_1 - P_{23} l_6 + P_{24} l_6 - P_{25} l_6 + P_{26} l_5;\\
J_к^y \ddot{\psi}_к = -P_1 l_4 - P_2 l_4 + P_{19} l_4 + P_{20} l_4;\\
J_к^z \ddot{\theta} = P_{23} l_4 - P_{24} l_4 - P_{25} l_4 - P_{26} l_4;\\
m_к \ddot{y}_к = -P_{23} - P_{24} - P_{25} - P_{26};\\
m_{б1} \ddot{z}_{б1} = 0.5P_1 - P_3;\\
m_{б2} \ddot{z}_{б2} = 0.5P_1 - P_5;\\
m_{б4} \ddot{z}_{б4} = 0.5P_2 - P_6;\\
m_{кп1} \ddot{z}_{кп1} = P_3 + P_4 - P_7 - P_8;\\
J_{кп2}^x \ddot{\varphi}_{кп2} = P_6 l_1 - P_5 l_1 + P_9 l_2 - P_{10} l_2;\\
m_{п1} \ddot{z}_{п1} = P_7 - P_{11} + P_9\beta - P_{13}\beta;\\
m_{п2} \ddot{z}_{п2} = P_8 - P_{12} + P_{10}\beta - P_{14}\beta;\\
m_{п3} \ddot{z}_{п3} = P_9 - P_{13} + P_7\beta - P_{11}\beta;\\
m_{п4} \ddot{z}_{п4} = P_{10} - P_{14} + P_8\beta - P_{12}\beta;\\
(2m_{б1} + 2m_{б2}) \ddot{y}_m = -P_{15} - P_{16} - P_{17} - P_{18} - P_{23} - P_{24};\\
J_к^z \ddot{\theta} = P_{15} l_3 + P_{16} l_3 - P_{17} l_3 - P_{18} l_3 + P_{27} l_2 - P_{28} l_2 + P_{29} l_2 - P_{30} l_2;\\
m_{пр1} \ddot{z}_{пр1} = P_{19} - P_{21};\\
m_{пр2} \ddot{z}_{пр2} = P_{20} - P_{22};
\end{array}\right\} \tag{3}$$

The accepted designationshave the following physical sense: Z_i - vertical movements of the corresponding mass of the system; y_i - horizontal movements of mass of the system; $J_к^x$ - the moment of inertia of a body of the car concerning an axis x, passing through the body center of weight; $J_к^y$ - the moment of inertia of a body of the car concerning the cross axis at; $J_к^z$ - moment of inertia of a body of the car concerning the vertical axis z; $J_{кп1,2}^z$ - themoment of inertia of a wheel couple, concerning an axis x; J_m^z - the moment of inertia of the cart concerning the vertical axis passing through the center of its weight; $2l_1$ - the distance between the centers of axle-box sets of wheel couples; $2l_2$ - the distance between the circles of driving of wheel couples; $2l_3$ - rigidbase of the cart; $2l_4$ - the distance between the axes of pintles of a cart of the car; L_5 - the distance from the center of weight of a wheel couple to an over spring beam; l_6 - the distance from the body center of weight to an over springbeam.

The characteristics of the calculated «carriage-way» system are defined by selection at the solution of a problem of the movement of a carriage on the roughnesses received experimentally. The vertical forces determined in the course of the solution are transferred from wheels of the four-axle gondola car to cores of frogs, are compared at the same time with the similar forces received experimentally (figure 2). From figure 2 it is visible thatthe sizes of forces and also the nature of change on length of cores practically coincide with the forces received experimentally.

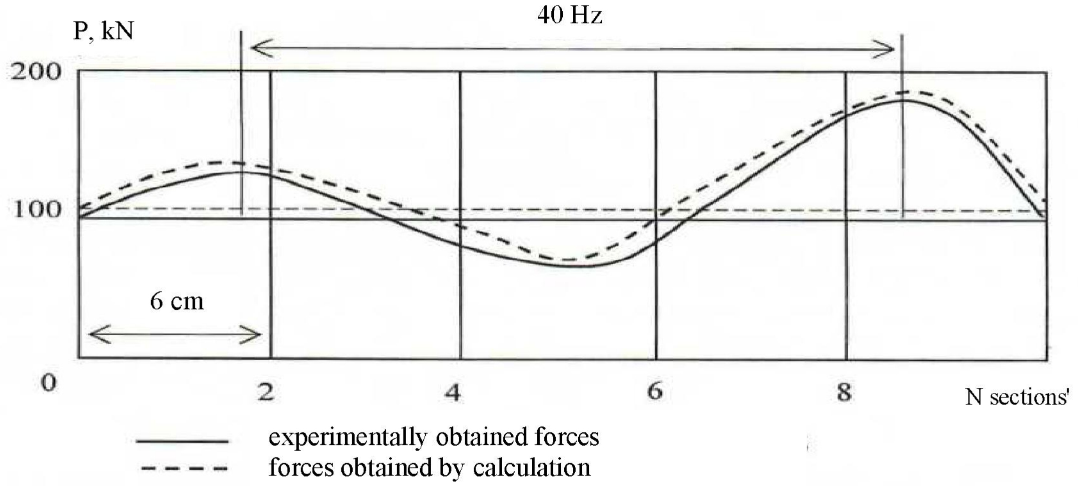


Figure 2 – Vertical forces of interaction of a wheel of the four-axle gondola car with the core of the frog of blind crossing

The oscillograms of primary measurements of the dynamic processes registered on railroad switches are provided on figures 3, 4.

Dynamic indicators of gondola cars on railroad switches are determined by the measurements executed at the passage by gondola cars of railroad switches №13 of brand 1/9 and №29 of brand 1/11 on the rails of P65 which are on the Bel.station. For a forward stroke of gondola cars it was accepted the direction when gondola cars 12-9920 moved forward by the 1st wheel couple from the station towards the entrance arrows.

The registration of dynamic processes in gondola cars in all arrivals began and came to an end on direct sites of a way.

The frame forces were measured by means of the graduated tensometric schemes pasted on side frames of carts of gondola cars.

The estimated values of the machined processes were defined as the maximum probable values of the measured values at each speed separately. The estimated values were counted on with probability 0,9985.

By the results of data processing at the given speed for each measuring scheme separate expanses of data were formed. According to these expanses there were estimated values of indicators. All the measurements were brokenon speeds and in the directions of the movement. In each arrival one maximum amplitude value of dynamic process was chosen. The values of frame forces were accepted taking into account a quastatic component, values of accelerations were processed without a quastatic component. For the assessment of size of frame forces the relations of frame forces to static load from a wheel couple of rails are considered. The indicators of influence of gondola cars on the way.

The tests on measurement of a level of influence of gondola cars on the way and railroad switches were made on the same sites of a way as testson the definition of dynamic indicators.

Forthemeasurement of a level of influence of gondola cars on the way and railroad switches the chosen sites have been equipped with tensoresistors for the measurement of tensions arising in the edges of a foot of rails, the vertical and side forces transferred from a wheel to rails. For measurement of tension in the lower edges of a foot of rails and the curvilinear pointed and forces transferred from a wheel to rails, the tensoresistors by base of 10 mm and with a nominal resistance of 100 Om which were gathered into measuring schemes have been pasted on rails. Signals from measuring schemes were given to the entrance of a measuring complex and were registered on the laptop.

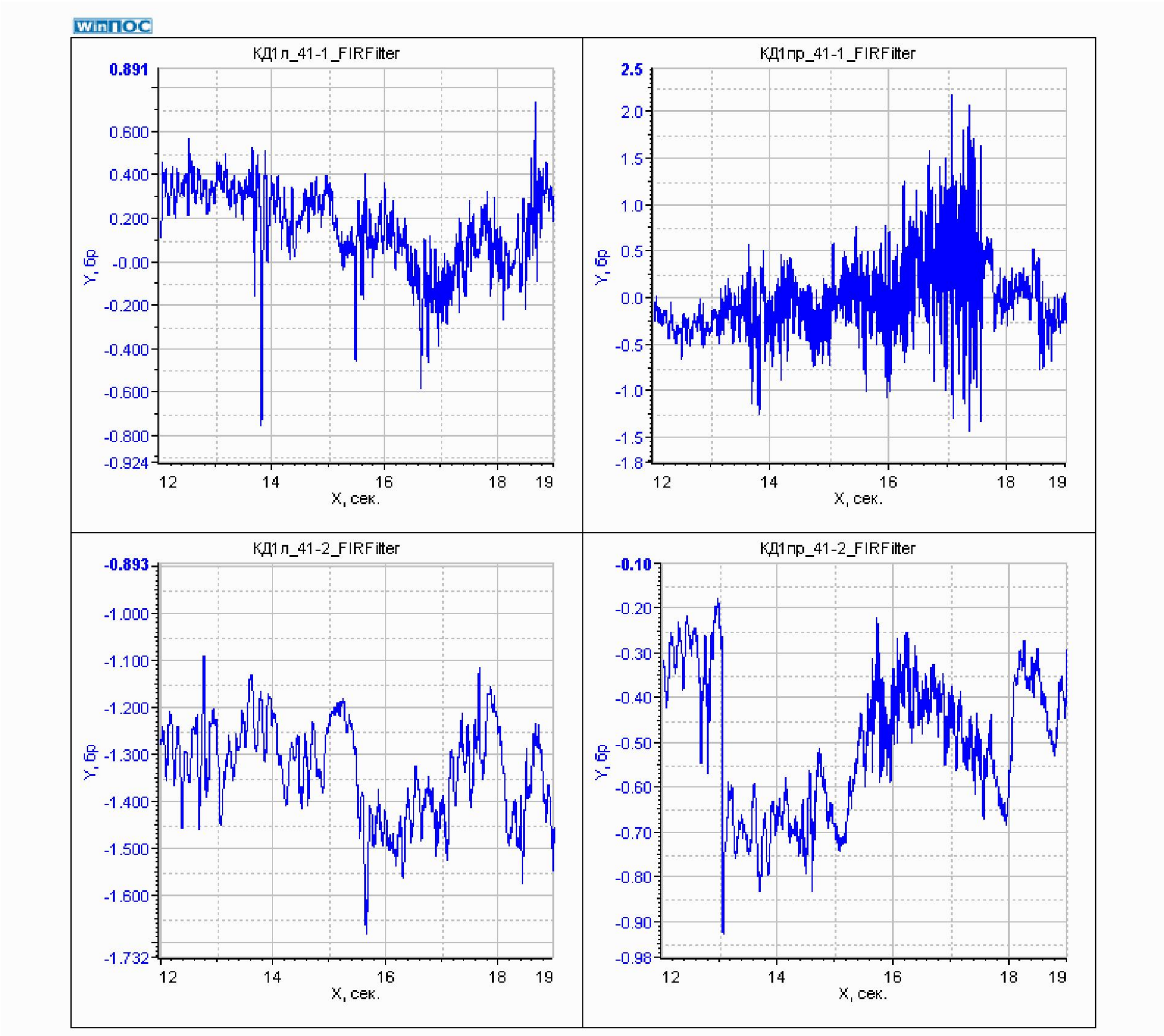


Figure 3 – Coefficient of vertical dynamics of the first step of spring suspension of gondola cars 12-9941 at the movement on railroad switches, mm

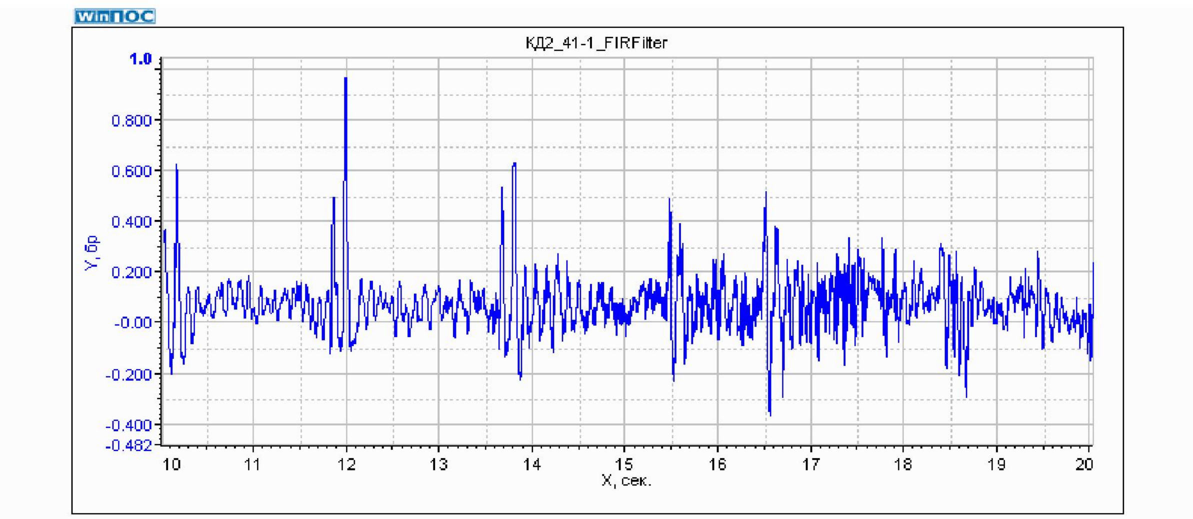


Figure 4 – Coefficient of vertical dynamics of the second step of spring suspension of gondola cars 12-9941 at the movement on railroad switches

The semi-bridge scheme, side and vertical forces from a wheel to rails – the bridge scheme is applied for the registration of tension in the lower edges of a foot of a rail. At the same time, the tensoresistor of a compensation run was pasted not on a rail, but on a separate steel plate and settled down near the rail section where the tension was measured in rail foot edges. The side forces from a wheel on rails were registered according to the bridge scheme by Schlumpf's method. The special load device with dynamometer sensors was used for graduation of tensometric schemes of measurement of side and vertical forces.

At testing on railroad switches the tensions in the lower edges of a foot of a rail were measured in a front extension of frame rails, in the curvilinear acute and in a switched curve. Forces transferred from a wheel to rails in the horizontal and vertical direction were measured in a front extension of frame rails and in a switched curve.

When calculating the following values of parameters of the gondola car have been accepted: μ - friction coefficient between a crest of the running wheel and a rail, μ =0,25; β - slope forming a wheel crest to the horizontal plane,β = 60º; Y_P - frame force taking into account a quasistatic component in curve sites of a way - the instant values registered at each measurement, кН; Q - the weight of a springed part of the gondola car coming to a neck of an axis of a wheel couple, Q=24,4 of kN for empty, Q=116,4 of kN for loaded; $K_{д1}$,$K_{д2}$ - coefficients of vertical dynamics in the first step of suspension (without dissipative forces and taking into account a quastatic component in curve sites of a way) respectively on the running and not running wheels of a wheel couple are the instantaneous values registered at each measurement; μ' - friction coefficient between the surface of driving of the running wheel and a rail, μ' =0,25; q - force of weight of the uncusioned parts falling on the wheeled couple, q=18,78 of kN; $2b$ - distance between the points of application of vertical loading to the necks of an axis of the wheeled couple, 2b = 2,036 m; a_1 - distance between the point of application of vertical loading to the neck of an axis on the running wheel and a contact point on a crest,a_1= 0,265 m; a_2 - distance between the point of application of vertical loading of an axis neck on not running wheel and a contact point on its surface of driving, a_2 = 0,228м; r - wheel radius around drivings, r= 0,479 m.

The dependence of the relation of frame to static load from the wheel couple to rails from the speed of movement is given in figure 5, a, b, c, d.

Here and in the subsequent figures it is designated:

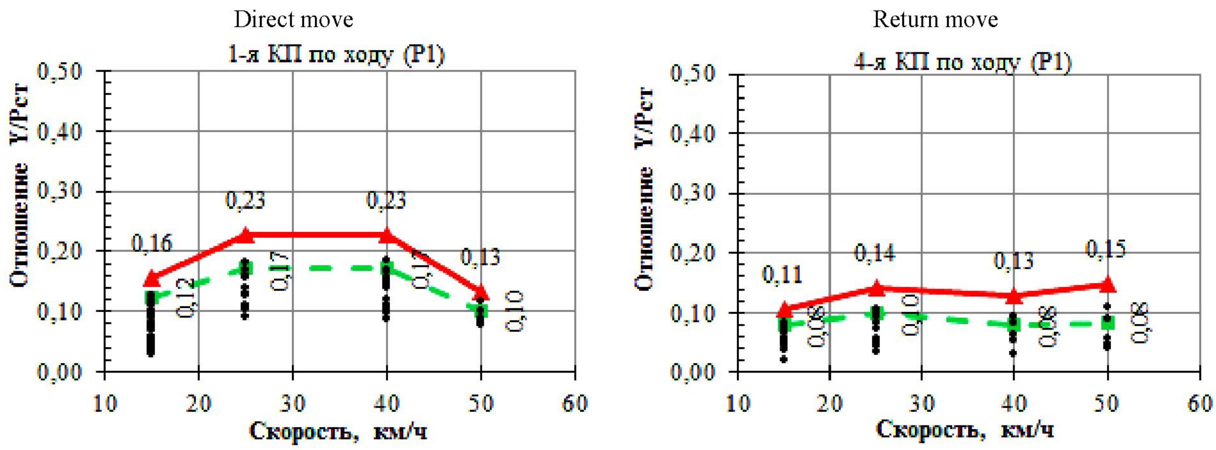
● – the measured values of indicators;

– estimated values (the maximum probable);

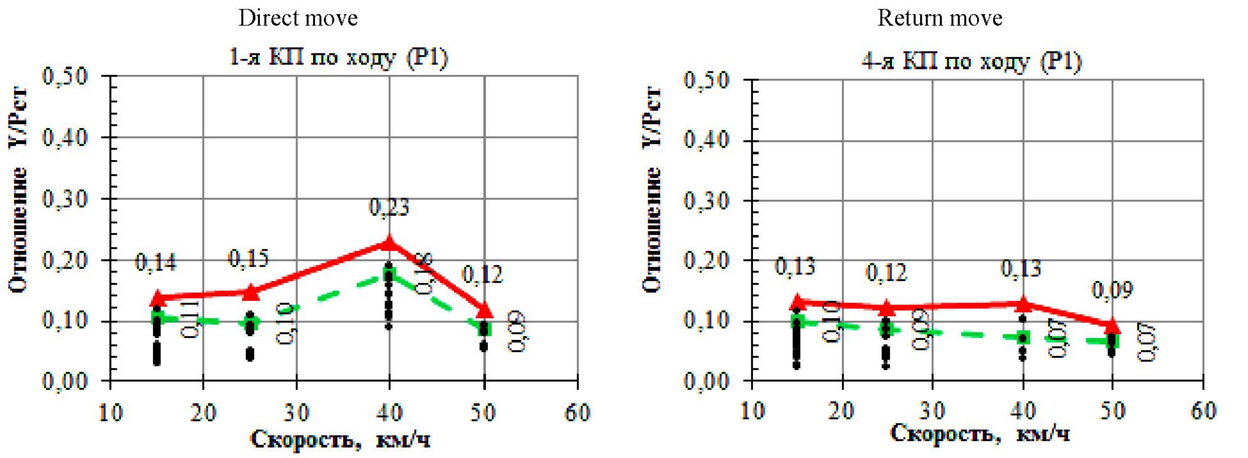
– an average from 3 maximum (for reference);figures on the schedules – estimated values.

In figure 5 it is visible that the relation of frame forces to static load from the wheeled couple to rails at the movement of gondola cars on railroad switches are in limits of admissible values. Also the big dynamics of the empty car is traced considerably.

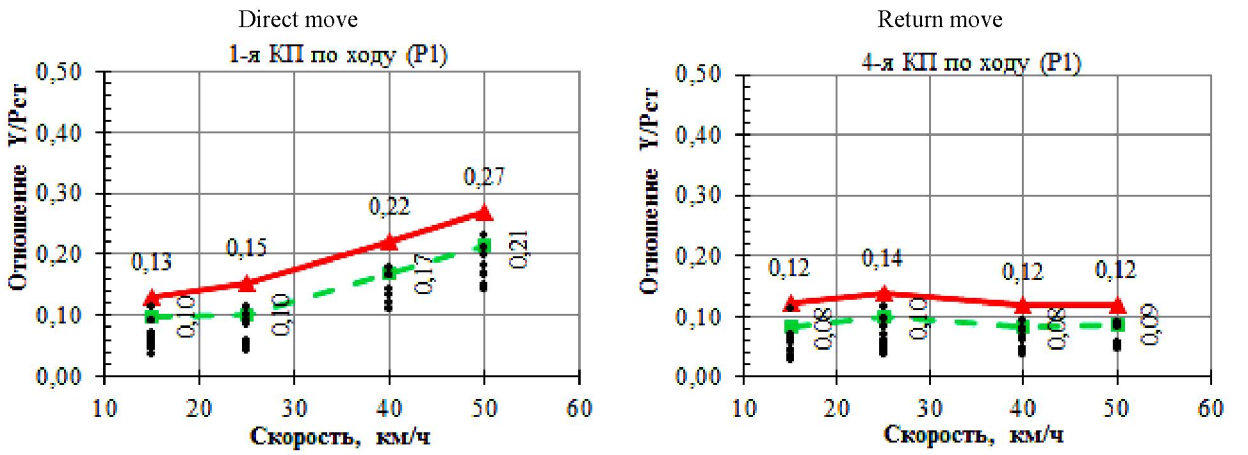
On instant values of frame forces and coefficient of vertical dynamics of the first step of spring suspension values of coefficient of a stock of stability against a wheel descent from a rail have been calculated (further in figures and in tables – KZU).



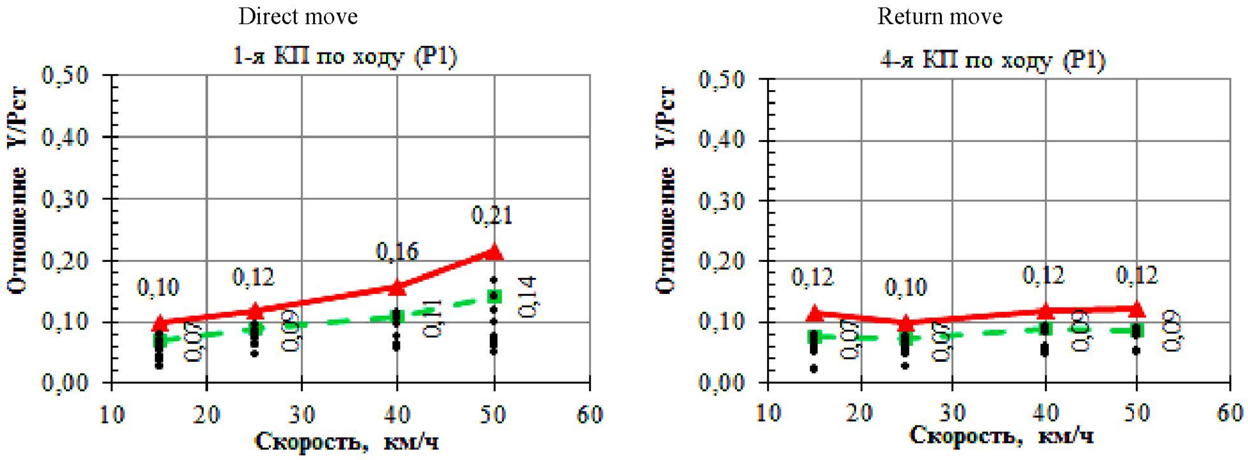
a) from the wheeled couple on rails to the empty gondola car of 12-9920 model
at the movement on P65 railroads witches of brand 1/9 to the side



b) from the wheeled couple to rails of the loaded gondola car of 12-9920 model
at the movement on P65 railroads witches of brand 1/9 to the side



c) from the wheeled couple to rails of the empty gondola car of 12-9920 model
at the movement on P65 railroads witches of brand 1/11 to the side



d) from the wheeled couple to rails of the loaded gondola car of 12-9920 model
at the movement on P65 railroads witches of brand 1/11 to the side

Figure 5 – Relation of frame forces to static loading

The minimum value of coefficient of stock of stability against a wheel descent from a rail at moving railroad switches is given in table 1.

Table 1 – Stability stock coefficient against a wheel descent from a rail

a) in railroads witches with a frog of brand 1/9										
Moving direction	KZUminimumvalue									
	Movementto a side				On a direct moving					
	At a speed, km/h									
	15	25	40	50	60	80	90	100	110	120
Empty gondola car										
Direct move	2,31	2,30	2,00	2,09	2,72	2,52	2,48	2,38	2,30	2,21
Return move	2,43	2,16	2,22	2,43	2,84	2,72	2,70	2,64	2,59	2,54
Loaded gondola car										
Direct move	2,27	2,48	2,36	2,53	3,16	3,00	2,80	2,72	2,61	2,49
Return move	1,83	1,70	1,82	1,65	3,13	2,74	2,58	2,39	2,20	2,02
The allowed value	1,4									
b) in railroad switches with a frog of brand 1/11										
Moving direction	The minimal value of KZU									
	Movement to a side				On a direct moving					
	At speed, km/h									
	15	25	40	50	60	80	90	100	110	120
Empty gondola car										
Direct move	2,71	2,77	2,24	2,26	2,79	2,75	2,73	2,71	2,69	2,67
Return move	2,84	2,68	2,93	2,87	2,91	2,72	2,63	2,53	2,44	2,34
Loaded gondola car										
Direct move	2,93	2,92	3,19	3,07	3,89	3,77	3,71	3,65	3,59	3,53
Return move	2,20	2,05	1,73	2,06	2,93	2,86	2,83	2,79	2,76	2,72
Admitted value	1,4									

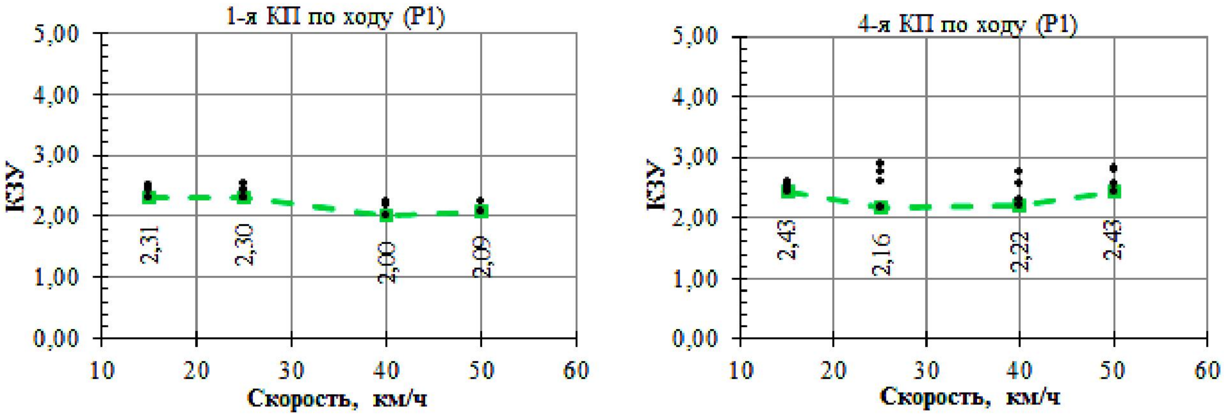
The received dependences of coefficient of a stock of stability against a wheel descent from a rail from the speed of the movement are given in figure 9, a, b, c, d. The coefficient of a stock of stability was estimated on the minimum calculated value. In figure 9 it is designated:

● – the measured values of indicators;

– ■ – – estimated values; figures on the schedules – estimated values.

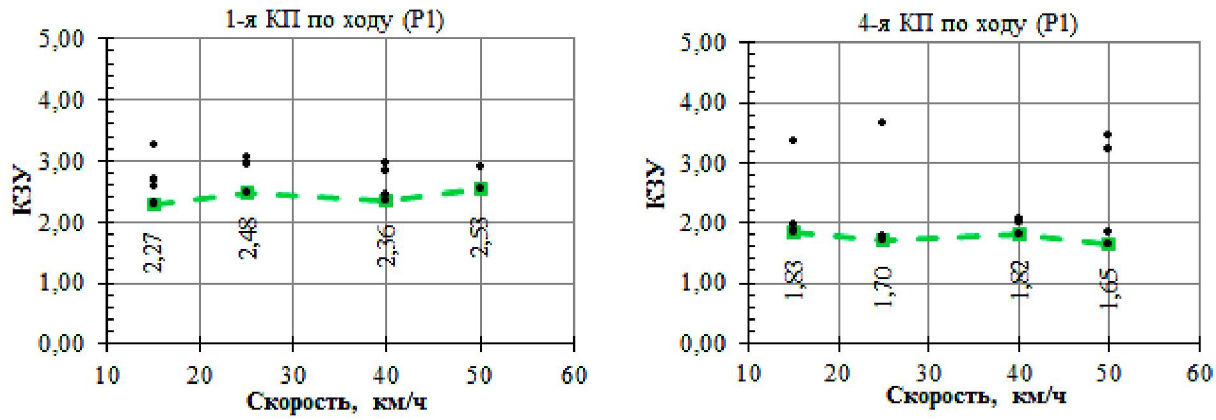
The data provided in table 2 and in figure 6 show that the stability stock coefficient against a wheel descent from a rail at transportation by gondola cars of railroad switches to the side up to the speed of 50 km/h and in the direct direction up to 120 km/h are in the admissible limits, i.e. accept the values not less than 1,4. The given results show that dynamic indicators of the gondola car of model 12-9920 in an empty and loaded state at moving railroad switches meet the requirements of "Norms of the allowed speeds of movement of locomotives and cars on railway tracks of a track of 1520 (1524) mm of railway transport of the Republic of Kazakhstan" (further Norms of the allowed speeds of movement).

Direct move Return move



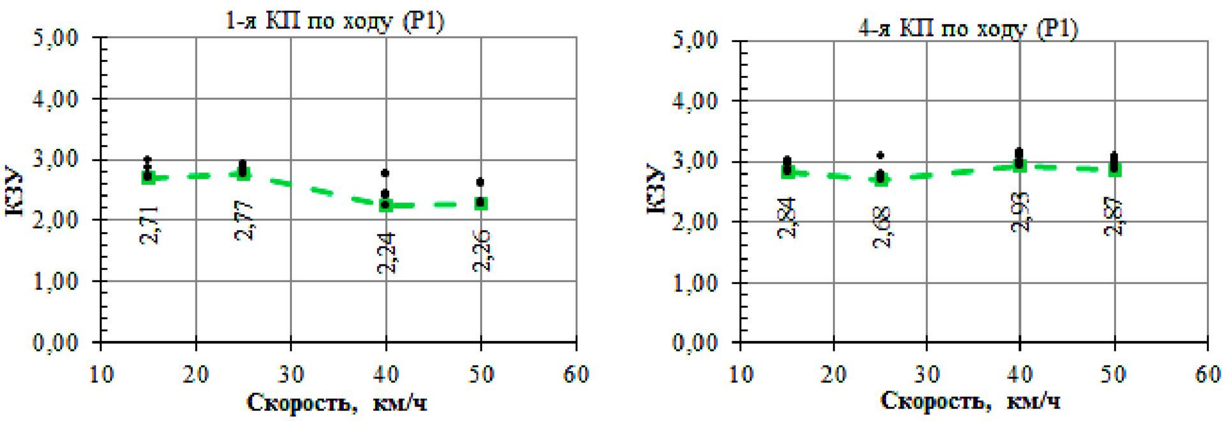
a) at moving by the empty gondola car of model 12-9920 of the railroads witch with the frog of brand 1/9 to a side

Direct move Return move



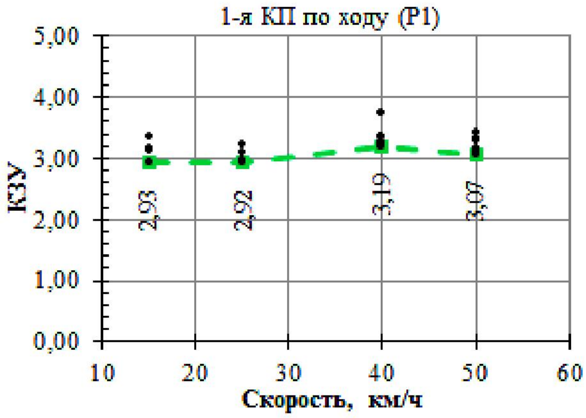
b) at moving by the loaded gondola car of model 12-9920 of the railroad switch with the frog of brand 1/9 to a side

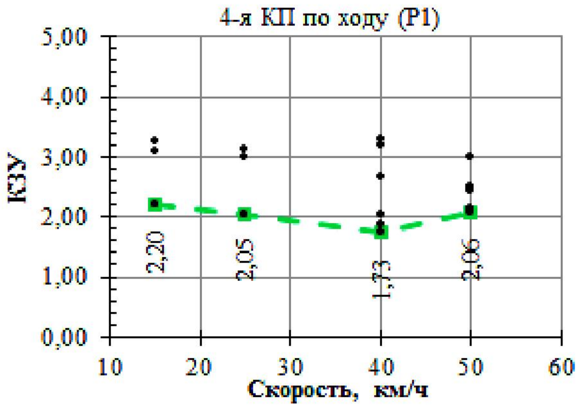
Direct move Return move



c) at moving by the empty gondola car of model 12-9920 of the railroad switch with the frogof brand 1/11 to a side

Direct move Return move





d) at moving by the loaded gondola car of model 12-9920 of the railroad switch with the frog of brand 1/11 to a side

Figure 6 – Stability stock coefficient against a wheel descent from a rail

Conclusions. As the conducted researches show, the sizes of the vertical and horizontal forces, which are transferred from the wheels of carriages to the cores of blunt and sharp frogs are one of the main criteria for designation of the maximum speeds of the movement on blank crossings of the combined track. And the spatial scheme used in calculations is rather reliable and can be used for carrying out analytical researches of interaction of a way and the rolling stock in the vertical and horizontal planes, including, in cases when roughness both on one, and on both rail threads. The executed pilot studies show that stability stock coefficient against a wheel descent from a rail at transportation by gondola cars of railroad switches to a side up to the speed of 50 km/h and in the direct direction up to 120 km/h are in the admissible limits, i.e. accept values not less than 1,4.

In the result of the executed researches it is proved that the dynamic indicators of the gondola car of model 12-9920 in the empty and loaded state at moving railroad switches meet the requirements of "Norms of the allowed speeds of movement of locomotives and cars on railway tracks of a track of 1520 (1524) mm of railway transport of the Republic of Kazakhstan" (further Norms of the allowed speeds of movement). Therefore, the speed of the movement on railroad switches according to Norms of the allowed speeds of movement on assessment of dynamic indicators is limited of 40 km/h at the movement on railroad switches to a side and constructional speed at the movement on railroad switches on a direct move.

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ZK1 МОДЕЛЬДІ АРБАШАЛАРЫМЕН ТЕМІРЖОЛ ЭКИПАЖДАРЫ БАҒЫТТАМАЛЫҚ БҰРМАЛАРДЫ ӨТКЕНДЕ ШЕКТІ ЖЫЛДАМДЫҚТАРДЫ АНЫҚТАУ ӘДІСТЕМЕСІ

Аннотация. Жүк поездар қозғалысының қауіпсіздігін жоғарылату үшін кіші радиустағы қисықтарды өткенде және соның ішінде кеңінен таралған маркадағы бағыттамалық бұрмалардың аудармалық қисықтарын өткен кезде рұқсат етілген шекті жылдамдықтар теміржол әкімшілігінің нормативтік құжаттарымен белгіленеді. Аудармалық қисықтардың бірқатар ерекшеліктері бар – аудармалық ендірмелердің болмауы, сыртқы рельстің жоғарыламауы, бүйір жолда аударманың бұрмалық қисығы арқылы станцияға кіргенде жылдамдықты шұғыл төмендету қажеттілігін шарттайтын біріккен жолтабанның тұйық қиылыстары. Шекті жылдамдықтарды анықтау үшін бағыттамалық бұрмалардың айқастырма өзектеріне экипаждан берілетін вертикалды және горизонталды-көлденең күштерді, вертикалды динамика коэффициентін, осы күштердің қатынасын анықтау мақсатымен экипаждарда пайда болатын рамалық күштердің шамасын және бағыттамалық бұрмаларға динамикалық әсер деңгейін ескеру қажет.

Қойылған мәселе құрама әдіспен шешіледі: аналитикалық – есептік және тәжірибелік-эксперименттік. Аналитикалықәдіспен шешу үшін рессорлар жинағының қаттылығы, үйкеліс коэффициенттері, көлденең, вертикалды және бойлық осьтерге қатысты шанақ және доңғалақтар жұбының бұрылу бұрыштары ескерілетін есептік сұлба құрылады. Есептік сұлба және қабылданған болжамдар негізінде, Даламбер әдісін қолдана отырып, екінші реттік теңдеу құрылды. Қосымша динамикалық реакция күштері және рессорлық жинақтар байланыстары қосымша теңдеулер арқылы жазылған. Теңдеулер шешімі кезең-кезеңмен интегралдаудың сандық әдісімен жүргізілді. Жұмыстың эксперименттік кезеңі Бел Шу бекеті жол бөлімшесінің өткелінде жүргізілді. Бастапқыда динамикалық көрсеткіштер: вертикалды динамика коэффициенттері және түрлі жылдамдықтармен бағыттамалық бұрмаларды өткенде пайда болатын рамалық күштер анықталды. Бағыттамалық бұрманың рельс табандарының жиектерінде кернеулердің әсер ету деңгейін өлшеу және тіркеу жүргізілген. Өңделінетін процестердің бағаланатын мәндері әрбір жеке жылдамдық үшін өлшенген мәндердің максималды ықтимал шамалары ретінде анықталды. Бағаланатын мәндер 0,9985 ықтималдықпен бағаланды.

Түйін сөздер: теміржол экипажы, бағыттамалық бұрма, есепік сұлба, рамалық күштер, тұрақтылық, динамикалық көрсеткіштер.

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МЕТОДИКА ОПРЕДЕЛЕНИЯ ПРЕДЕЛЬНЫХ СКОРОСТЕЙ ПРИ ПРОХОЖДЕНИИ СТРЕЛОЧНЫХ ПЕРЕВОДОВ ЖЕЛЕЗНОДОРОЖНЫХ ЭКИПАЖЕЙ С ТЕЛЕЖКАМИ МОДЕЛИ ZK1

Аннотация. Для повышения безопасности движения грузовых поездов нормативными документами железнодорожных администраций устанавливаются предельно допустимые скорости при прохождении кривых малого радиуса и, в том числе, переводных кривых стрелочных переводов наиболее распространенных марок. Переводные кривые имеют ряд особенностей – отсутствие переводных вставок, возвышения наружного рельса, глухие пересечения совмещенной колеи, что обуславливает необходимость резкого снижения скорости при заходе на станцию через переводную кривую стрелочного перевода на боковой пути. Для определения предельных скоростей необходимо учитывать вертикальные, горизонтально-поперечные силы, передаваемые от экипажа на сердечники крестовин стрелочных переводов, коэффициент вертикальной динамики, величину рамных сил, возникающих в экипажах, с целью определения отношения этих сил к статической нагрузке и уровень динамического воздействия на стрелочные переводы.

Поставленная задача решается комбинированным методом: аналитически - расчетным и опытно-экспериментальным. Для решения аналитическим методом составляется расчетная схема, где учитываются жесткости рессорных комплектов, коэффициенты трения, углы поворота кузова и колесных пар, относительно поперечной, вертикальной и продольных осей. Исходя из расчетной схемы и принятых допущений, используя метод Даламбера были составлены уравнения второго порядка. Дополнительные динамические силы реакции и связи рессорных комплектов записаны через дополнительные уравнения. Решение уравнений проводилось численным методом поэтапного интегрирования. Экспериментальный этап работы проводился на перегоне станции Бель Шуйского отделения дороги. Первоначально определялись динамические показатели коэффициенты вертикальной динамики и рамные силы, возникающие при прохождении стрелочных переводов с различными скоростями. Проведены замеры уровня воздействия и регистрация напряжений в кромках подошвы рельса стрелочного перевода. Оценочные значения обрабатываемых процессов определялись как максимально вероятные величины измеренных значений для каждой отдельной скорости. Оценочные значения оценивались с вероятностью 0,9985.

Ключевые слова: железнодорожный экипаж, стрелочный перевод, расчетная схема, рамные силы, устойчивость, динамические показатели.

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