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Ways to improve the mathematical model of a freight car for the execution of forensic railway-transport expertises

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Ways to improve the mathematical model of a freight car for the execution of forensic railway-transport expertises

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Abstract. According to statistics, receipts of forensic railway-transport expertises related to the derailment of freight wagons annually grows. At the same time, an analysis of the available mathematical models of freight wagons for the study of such cases showed that at present they are not sufficiently detailed to the extent that it is necessary to analyze the causes of rolling stock derailment. Therefore, taking into account the main reasons that occurrence of such railway accidents over the last five years on the railways of Ukraine, the article selects the main directions for improving the mathematical model of a freight wagon. These directions, in turn, should cover the whole plural of factors (explicit and hidden) and identify the most significant ones regarding the circumstances of the derailment rolling stock off the track, established on the basis of a computer experiment. An algorithm and a mathematical model are proposed taken into account the influence on the dynamics of the wagon of process of closing gaps in the axle-boxes and slides, and the equations of motion of the wheelset are supplemented by a guiding force, whose value is one of the main indicators of the stability of the rolling stock.

1. Introduction

Traffic safety is a priority on the railways. To ensure it, the regulatory framework is being improved, measures are being taken to prevent the occurrence of railway accidents and the factors that pose a threat to the traffic safety and tend to grow as well as other necessary measures are being explored.

However, as statistics show, the number of the railway accidents is steadily increasing every year. Of the total number, the majority of cases are related to the rolling stock derailment. The reasons for this kind of railway accidents are, first of all are the deviations in the technical state of the track and the running gear of the rolling stock of their maintenance norms [1].

Many years of experience in conducting expertises shows that there are more and more cases when it is impossible to unambiguously determine the reasons for the rolling stock derailment. Such reasons are not obvious and require an in-depth analysis of the complex combination of many factors of the force interaction of the rolling stock and the rail track, which form the most unfavourable (worst) combination of circumstances occurring under particular state of the rolling stock and the railway track [2, 3].

Hence, there is the need to substantiate proposals for scientific approaches improvement in the study of the interaction of the rolling stock and the track, the calculation of the indices that evaluate the dynamic properties of its running gear and traffic safety indicators. This, in turn, will not only facilitate the investigation of the causes of the rolling stock derailment, but in most cases, it will allow to set the limit values of its technical state, at which there is a threat of an accident.

2. Purpose and objectives of the research

The purpose of the work is to highlight the main problems that arise when investigating the causes of the rolling stock derailment, and to develop ways to overcome them with the help of an advanced mathematical model of a freight wagon.

Research objectives:

- to analyse the main causes of freight wagon derailment on the railways of Ukraine over the last five years;
- for these reasons, to establish the main ways of improving the mathematical model of a freight wagon;
- to improve the mathematical model of a freight car for the possibility of its use in forensic railway-transport expertise.

3. Statement of the problem and review of scientific sources, which are devoted to its solution

During the period 2015 to 2020, the railways of Ukraine experienced about 165 cases of freight wagons derailment in train formations, of which 20% of cases was obvious running gear malfunction, 31% of other cases was malfunctions of the track, 15% was the rolling stock collisions [4].

The percentage of malfunctions of the running gear technical state that caused their derailment in train formations are shown in Figure 1

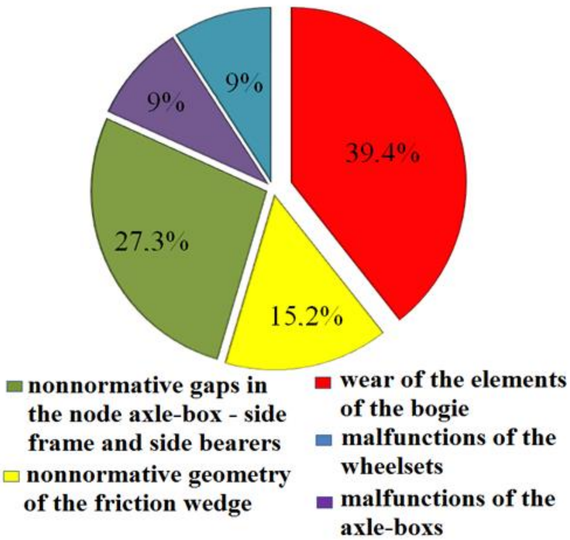

Figure 1. Diagram of the distribution of freight wagons running gear malfunctions that caused derailment for the last 5 years.

Other reasons for the loss of stability of the rolling stock are those that were not definitely established during railway investigations, and as a percentage, it accounted for about 7% of the total cases of the rolling stock derailment.

Obviously, the study of such cases is impossible without a detailed mathematical model of a freight wagon, which could give a reliable quantitative assessment of the parameters that affect the safety traffic, namely, friction wedges, spring suspension, gaps in the boxes and side bearers, etc.

Therefore, we analyzed the existing mathematical models of freight wagons and complex developments in the field of rolling stock dynamics, which served as a basis for further development of methods and means of this problem research.

The technique of constructing mathematical models of wagons spatial vibrations for the study of reasons of their stability loss during movement was substantially enriched due to works of M.L. Korotenko [5] and V.F. Ushkalov [6]. The researches performed by them allowed to solve the problem of optimization of the parameters of wagons running gears under condition of stability of their undisturbed movement.

In work [7], with the use of the developed mathematical model of a freight wagon, the influence on the stability of its movement a considerable difference in the base of bogies and wheel diameters was investigated. However, in the course of the research, the authors of this paper did not take into account the influence of horizontal irregularities of the railway track and a significant number of parameters of the bogies of freight wagons provoking their derailment.

In work [8], a detailed mathematical model of a freight wagon was developed, which allows to estimate the influence of the magnitude of angles of friction wedges, gaps in the boxes and side bearers of the wagon affecting its derailment. The disadvantage of this work is taking not into account the guide force, whose value is one of the main indicators of the stability of his motion.

The strategic method of analysis of operational safety of the rolling stock preventing its derailment was proposed in works [9, 10]. The mathematical model of a freight wagon offered in these works allowed to establish the influence of vertical irregularities of the railway track on the stability of wagon's movement. The disadvantage of the developed mathematical model of a wagon in works [9, 10] is the neglect of deviations influence on the parameters of the technical state of wagons' running gear from normative values in case of their derailment.

In work [2], many hidden causes of rolling stock derailment in the Russian Federation are outlined and a simplified model of a freight wagon is proposed to investigate such cases. The main disadvantage of this mathematical model of a wagon is that it gives the possibility to take into account only the irregularities of the track as a factor that impacts wagon's derailment.

With regard to the implementation of investigations of real cases of railway accidents, the work of Sokol E.M. should be noted [11], in which analytical expressions were derived to determine sufficient conditions for the rolling stock derailment on the basis of the developed mathematical model of motion a wheel-set of a freight wagon. However, this model allows to determine the influence of only a part of required parameters of a freight wagon and the railway track on wagon's stability. In addition, the equations of motion of a wheel-set of a wagon drawn up by Sokol E.M. do not take into account the action of creep forces that can have a significant effect on the stability of wagon's motion.

As can be seen, at the present time, many scientific works, both domestic and foreign, are devoted to the problem of the rolling stock derailment. However, each of the above works has its advantages and disadvantages when studying the dynamics of the rolling stock, and the mathematical models of freight wagons offered in them are not sufficiently detailed to investigate the reasons for their derailment.

4. Improved mathematical model of a freight wagon

Due to the fact that at present the experts of the railway research laboratory of Lviv Scientific Research Institute of Forensic Expertise undertake examination of cases of rolling stock derailment, which occur in the absence of obvious deviations of the rolling stock and the track from the requirements of the regulatory documents, there is a necessity to select such a mathematical model of a freight wagon that would take into account a larger number of parameters of its running gear than those currently available.

Therefore, the mathematical model of a freight wagon, which was developed in work [8], was selected as a basis, and in order to choose ways to improve such a model, the assumptions made during its creation were analyzed.

Of all the assumptions adopted in [8] when creating a mathematical model of a freight wagon, it should be noted from the fact that the rigidity of the rail track in the horizontal plane was not taken into account in the equations of motion of the wheelset. So, the authors of the mentioned work did not take into account the influence of the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them, which can lead to unreliable of the quantitative assessment influence of the parameters of the railway track and running gears of wagons on their derailment

With this in mind, the mathematical model of a freight wagon was improved by the authors of this article by taking into account the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them, in the equations of wheel-set motion. In addition, to create a computer model on its basis, the algorithm for determining the forces that arise during gaps closure in the side bearers and axle-boxes of a freight wagon was developed.

Since this mathematical model is quite cumbersome, we will give some of its equations (1-4) in which improvements and additions were made.

$$m_b H\ddot{y}_b + \left(m_b H^2 + J_{bx}\right)\ddot{\theta}_b + \sum_{i=1}^{2} R_{ti} e_{fti} + R_{ck1}B_1 - R_{ck2}B_2 + R_{ck3}B_3 - R_{ck4}B_4 = 0, \tag{1}$$

$$J_{xbl1}\ddot{\theta}_{bl1} + N_{z1}a_1 - N_{z2}a_2 + N_{z3}a_3 - N_{z4}a_4 + \sum_{j=1}^{5} P'_{fzj}b'_{fj} - \sum_{j=1}^{5} P''_{fzj}b''_{fj} - R_{ck1}B_1 + R_{ck2}B_2 - \sum_{j=1}^{7} P'_{fyj}h_{61} - \sum_{j=1}^{7} P''_{fyj}h_{bl1} = 0, \tag{2}$$

$$J_b\ddot{\psi}_b + \sum_{i=1}^{7} P'_{fyi}\left(L_i \pm e'_{fi}\right) + \sum_{i=1}^{7} P''_{fyi}\left(L_i \pm e''_{fi}\right) - \sum_{i=1}^{7} P'_{syi}\left(L_i \pm e'_{si}\right) - \sum_{i=1}^{7} P''_{fyi}\left(L_i \pm e''_{si}\right) + \sum_{i=1}^{7} T_{fyi}\left(L_i \pm d_i\right) -$$

$$-\sum_{i=1}^{7} T_{syi}\left(L_i \mp d_i\right) - X_9 \cdot T_{ck1}B_1 + X_{10} \cdot T_{ck2}B_2 - X_{11} \cdot T_{ck3}B_3 + X_{12} \cdot T_{ck4}B_4 - M_{fr1} + M_{fr2} = 0, \tag{3}$$

$$m_{wh1}\ddot{y}_{wh1} + F_{y1} + F_{y2} - X_{13} \cdot T_{py1} - X_{14} \cdot T_{py2} + R_1 n_1 - R_2 n_2 + N_{aby11} - N_{aby12} - N_{aby21} + N_{aby22} + N_{g1} - N_{g2} = 0, \tag{4}$$

where m_b, m_{wh1} – mass of the body and mass of first wheelset, *t*; H – height of center of weight of the body, *m*; y_b, y_{wh1} – transverse displacement of the body and wheelset, *m*; J_{bx} – body moment of inertia about an axis *x* $t{\cdot}m^2$; R_{t1-2} – vertical reactions on body center plate, *kN*; e_{fi}, e_{si} – transverse displacements relative to the center of application of the vertical reactions on body center plate, *m*; R_{ck1-4} – vertical reactions on the side bearers, *kN*; B_{1-4} – the distance from the center pin to the middle of the side bearer, *m*; $P'_{fzj}, P''_{fzj}, P'_{fyj}, P''_{fyj}$ – vertical and transverse reactions of the springs of the first and second spring sets of the first bogie, *kN*; P'_{syj}, P''_{syj} – transverse reactions of the springs of the first and second spring sets of the second bogie, *kN*; T_{pyi} – transverse friction forces between the side frame and the axle-box, *kN*; N_{abyij} – lateral forces, arising when closing transverse and longitudinal gaps in axle boxes, *kN*; N_{g1-2} – force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them, *kN*; R_{1-2} – reaction in wheel-rail contact, *kN*; n_{1-2} – taper of the wheel rim at the points of contact with the rail; F_{y1-2} – the transverse forces of the creep, *kN*; θ_b, θ_{bl} – generalized coordinates of the lateral oscillations of the body and the bolsters, *rad*; N_{z1-4} – projection on the axis of the normal forces acting between the bolster and friction wedges, *kN*; a_{1-4} – distance from the longitudinal symmetry of the bogie to the wedge spring, *m*; h_{bl1-4} – height of the center of gravity of the bolster above the upper supporting surface of the springs of the spring sets of the first and second bogie, *m*; T_{fyi}, T_{syi} – the friction force between the bolster and the friction wedge of the first and second bogie, *kN*; M_{fr1-2} – moments of friction forces in the bolsters, *kN ·m*; T_{ck1-4} – friction forces the side bearers, *kN*; X_{9-14} – auxiliary coefficients.

In the equations, the upper signs in the brackets at horizontal reactions $P'_{fyj}, P''_{fyj}, P'_{syj}, P''_{syj}$ ' correspond to the spring No 1, 2, 6 of the spring set of a bogie, low signs – for No 3, 4, 7, and for the forces of transverse friction T_{fyi}, T_{syi}, which arising between the friction wedge and the bogie bolster – the upper signs for friction wedges of a bogie No 1, 2, 6, 7, low signs – for wedges No 3, 4, 5 and 6.

As we see equation (4) was supplemented by the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them N_g, which acting on the right and left wheels of a wagon wheel-set, and her value proposed to be determined by the following formula:

$$N_g = C_y \left(z_a^{\varphi} + z_t \right), \tag{5}$$

where C_y – is the horizontal rigidity of the track, *kN/m*; z_a^{φ}, z_t – angular and transverse movement of the rail when being exposed to the wheel-set of a wagon, *m* [11].

To establish the influence of the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them, on the transverse movement of the wheel-set within the gap of the rail track, we will conduct study with and without respect of its value in Maple software environment.

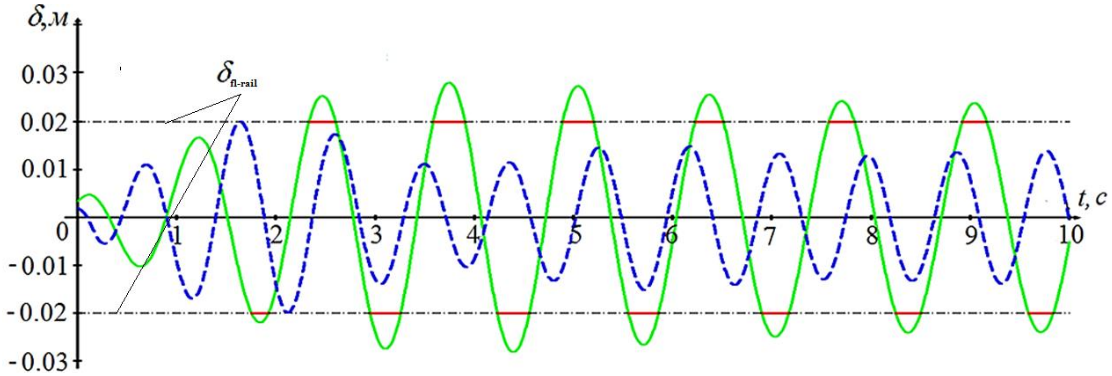

Figure 2. Transverse movement of the wheel-set with and without the guiding force.

As can be seen from Figure 2, the lack of regard of the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them, affects the value of the transverse movement of the wheel-set and may be greater than the value of the boundary clearance between the wheel flange and the rail, that is, leads to false results. When accounting the value of this force, the value of the wheel-set movement does not go beyond the limit gap.

Now let's consider the algorithm development for establishing the forces that arise during gaps closure in the side bearers and axle-boxes of a wagon. Since there are four gaps in the side bearers wagon and thirty-two gaps in the axle-boxes, we present formulas for determining several of them, and expressions for the remaining gaps are given in detail in work [8].

The gaps in the side bearers of a wagon are determined by the following formula:

$$\begin{cases} \delta_{ck1} = \delta_{ckn1} - \theta_b B_1 + \theta_{bl1} B_1, \\ \dots \\ \delta_{ck4} = \delta_{ckn4} - \theta_b B_4 + \theta_{bl2} B_4, \end{cases} \tag{6}$$

where $\delta_{ckn1}, \delta_{ckn4}$ – nominal value of the gaps in the side bearers, *mm*.

Transverse clearances in the axle-box of a wagon are determined by the formula:

$$\delta_{aby1} = \delta_{abn1} - y_{f1} + y_{wh1} - \psi_{f1} l_1, \tag{7}$$

where δ_{abn1} – nominal value of the operational transverse clearances in the axle-boxes, *mm*; y_{f1}, y_{wh1} –

lateral movement of the side frame of a bogie and the first wheel-set, *mm*; ψ_{f1} – generalized coordinates of the frame of the first bogie, *mm*; l_1 – half of the base of the first bogie, *mm*.

As can be seen from expressions (6, 7), the gaps closure in the side bearers and axle-boxes is possible when their value becomes equal to or less than zero, that is $\ddot{\delta}_{abyi}, \ddot{\delta}_{ckj} \le 0$.

To determine interaction forces in the side bearers when their gaps are closed, in work [8] a shock model was proposed, which assumes that the second derivative at the time of the strike, when inequality is fulfilled $\ddot{\delta}_{abyi}, \ddot{\delta}_{ckj} \le 0$, becoming zero, that is $\ddot{\delta}_{abyi} = 0, \ddot{\delta}_{ckj} = 0$.

Having differentiated expression (6) twice and having derived from equations (1, 2) $\ddot{\theta}_b$ and $\ddot{\theta}_{bl1}$, we obtain the value of reactions in the side bearers of a wagon when the gaps are being closed.

As an example, here are the analytical expressions for the forces that occur when the first and fourth gaps are being closed:

$$\begin{cases} R_{ck1} = X_2 \cdot R_{ck2}\left(\dfrac{B_2}{B_1 J_{xbl1}} + \dfrac{B_2}{B_1(m_b H^2 + J_{bx})}\right) - \dfrac{X_3 \cdot R_{ck3} B_3}{B_1(m_b H^2 + J_{bx})} + \dfrac{X_4 \cdot R_{ck4} B_4}{B_1(m_b H^2 + J_{bx})} + (A_1 - B_1)\cdot X_5, \\ \dots \\ R_{ck4} = X_3 \cdot R_{ck3}\left(\dfrac{B_3}{B_4 J_{xbl1}} + \dfrac{B_3}{B_4(m_b H^2 + J_{bx})}\right) + \dfrac{X_1 \cdot R_{ck1} B_1}{B_4(m_b H^2 + J_{bx})} - \dfrac{X_2 \cdot R_{ck2} B_2}{B_4(m_b H^2 + J_{bx})} - (A_4 - B_4)\cdot X_8, \end{cases} \tag{8}$$

where $$A_1 = \frac{\left(m_b H^2 + J_{bx}\right)\left(N_{z1}a_1 - N_{z2}a_2 + N_{z3}a_3 - N_{z4}a_4 + \sum_{j=1}^{5} P'_{fzj} b'_{fj} - \sum_{j=1}^{5} P''_{fzj} b''_{fj} - \sum_{j=1}^{7} P'_{fyj} h_{61} - \sum_{j=1}^{7} P''_{fyj} h_{bl1}\right)}{B_1 J_{xbl1}}, \tag{9}$$

$$B_1 = \frac{\left(m_b H \ddot{y}_b + R_{t1} e_{f1} + R_{t2} e_{f2}\right)}{B_1(m_b H^2 + J_{bx})}, \tag{10}$$

X_{1-8} – auxiliary coefficients; A_4, B_4 – coefficients are derived similarly to coefficients A_1, B_1.

Similarly, it is possible to determine the values of forces (reactions) that arise when the gaps are being closed in the axle-boxes of a wagon.

As noted above, the reactions in the side bearers and axle-boxes occur only when they are being closed, so in order to perform computer simulation of a freight wagon dynamics, the algorithm of their calculation should be fulfilled in the following steps (for example gaps in the side bearers):

- the value of the gaps is determined by the formulas (6) and if one of four gaps is closed, then the coefficient of X at its constituents A and B becomes equal to one, and the coefficients of X for the other reactions in the side bearers are zero;
- expression (8) defines the reaction in the closed side bearer;
- in case of closing of two gaps, the coefficients A and B at their constituents and their coefficients of X become equal to one, the other coefficients are equal to zero and the system of equations eventuates, from which the corresponding reactions in the side bearers are determined.

Since more than two gaps cannot be closed at the same time, then two of four equations (8) can be in the real situation.

It should be noted that when one or two gaps are being closed under the condition of relative movement of a body and bolsters, the friction forces T_{ckj} additionally arise and in equation (3) the value of coefficient X_i i becomes equal to one at these forces.

Therefore, the solution of the system of equations (8) will allow to evaluate how wagon movement dynamics is being affected by not only the forces that arise when the gaps are being closed in its side bearers, but also by the friction forces on their surface, which is especially relevant for curved sections of the track where their increased wear occurs.

Having done the research with the help of an advanced mathematical model of a freight wagon in Maple software environment, we give the graphs of the lateral oscillations of its body and bolster obtained during closed gaps in the side bearers.

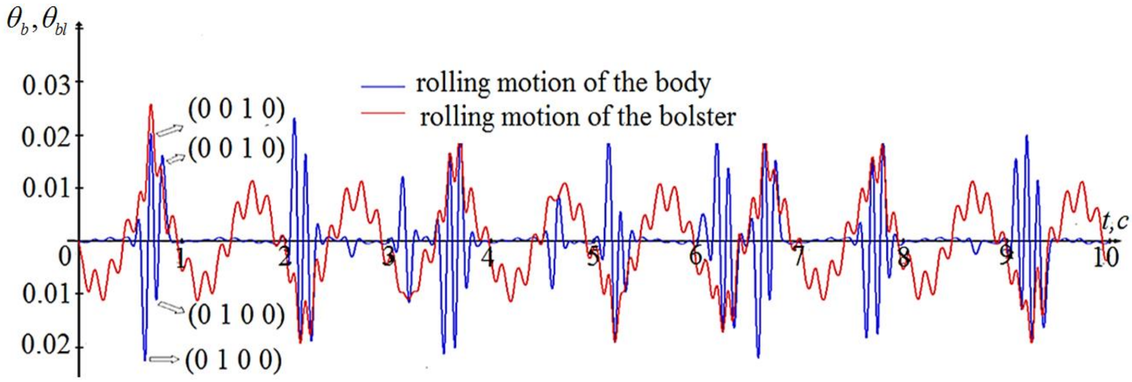

Figure 3. Lateral oscillations of a body and a bogie bolster when gaps are being closed in the slides.

As can be seen from Figure 3, the reactions occurring in the side bearers during their closure make a significant impact on the amplitude of a body and a bogie bolster movement. In this case, the moments of gaps closure can be traced from the matrix that reflects such a process ($\delta_{ck1}, \delta_{ck2}, \delta_{ck3}, \delta_{ck4}$) (Figure 3).

The similar algorithm and model can be used to determine the reactions that occur during gaps closure in the axle-boxes. However, unlike the gaps in the side bearers, when the gaps are being closed in the axle-boxes, the friction force between the axle-box and the side frame T_{pyi} becomes zero, since in this case they will move inseparably. Besides, in the system of differential equations (formula 4), the coefficients near the expressions, which determine the friction forces between the axle-box and the side frame T_{py1}, T_{py2}, which gap is depleted, must be equal zero, that is $X_{13}, X_{14} = 0$ (the case of two transverse gaps closure between the axle-box and the side frame).

Regarding the possibilities of the improved mathematical model of a freight wagon, it can be used for establishing causes of increased wear of the side bearers, side frames, axle-boxes, and in the study of the process of wheels of wheelsets rolling on the rail head with improper regulation of gaps under operational conditions.

Due to the fact that the main task in solving the problem of rolling stock safety is to find effective methods and tools that allow, with high probability, to describe the dynamic processes connected with rolling stock movement along the railway track. The proposed ways to improve mathematical models of freight wagons provide not only qualitative but also quantitative information about such processes and are adequate to real analogues of wagons movement in the real sections of the track.

5. Conclusions

1. The analysis of the reasons of rolling stock derailment on the railways of Ukraine over the past five years has revealed that in about 20% of cases they are obvious and in 7% are not obvious and implicit. The research of such cases in the course of an official investigation by the railway and during the performance of forensic examinations require the use of an advanced mathematical model of a freight wagon, which would allow quantify the influence of the parameters and malfunctions of

running gears of wagons on their derailment.

2. It was found that among the malfunctions of wagons' running gears, which caused their derailment, 39.4% were nonnormative wear of structural elements, and 29.3% – were nonnormative gaps in the axle-boxes and side bearers.

3. The main ways of improving the mathematical model of a freight wagon were proposed, namely, accounting when studying its dynamic behaviour of gaps in the axle-boxes and side bearers and the force, arising as a result of pressing the wheel flange to the rail head due to the exhaustion of the gap between them.

4. Performed research calculations with the use of the advanced mathematical model of a freight wagon showed that the lack of consideration of forces that arise during gaps closure in the slides, axle-boxes and guiding forces acting on the wheel-sets can lead to erroneous results when performing research in the course of forensic railway-transport expertises.

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