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MULTIPLE-FACTOR ANALYSIS OF THE DYNAMIC INTERACTION BETWEEN RAILROAD CARS AND JOINT IRREGULARITY

Summary. Mechanical models of “railroad car-track” transport system for each phase of motion are developed. Particularities of dynamic interaction between the four-axle car and the track are studied by considering four phases of motion over joint irregularity. Methods for solving differential equations of the discrete-continuous system fluctuation are developed. Numerical analysis is used to determine deflections of the trailing rail under the first sleeper for each phase of motion depending on motion phases, and car load and speed.

Keywords: railway rolling stock; four-axle car; track; ballast; joint irregularity; trailing rail; facing rail

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1. INTRODUCTION

The issues of optimizing performance characteristics and developing efficient design solutions, which enhance reliability and increase the longevity of rail transport, are becoming of crucial importance nowadays under conditions of rapid development in transportation technologies. The operating life of railway rolling stock and track structure depends on the interaction between elements, which is affected by mechanical, design and geometrical characteristics [4,5,15].

The operating performance of the track under railway rolling stock depends on the type of rail fastening, the stiffness of the track structure elements and operational conditions. Reducing the parameters of dynamic interaction between the railroad car and the track, in particular, at the joints, allows for the transition to durable, reliable, comfortable and fast rail transport.

The ballast draft under sleepers in rail joint areas is the most informative index of mechanical interaction in the “wheel-rail” system [15]. It is at rail joints that the rail is typically exposed to the largest impact load, which leads to rail creeping and sagging, and the emergence of gapless and elongated joints. Thus, elastic and permanent residual deformations of the ballast under rail supports determine the operation and service life of the track.

2. LITERATURE REVIEW AND RESEARCH OBJECTIVE

According to the analysis of the research dealing with issues of mechanical interaction between the four-axle car and the track [15], insulated rail joints are the weakest link in the examined system. Besides, having to take into account phases of the car motion over joint irregularity is emphasized, as well as characteristics of the ballast stiffness [7,17]. In addition, an emphasis is placed on issues relating to the combined action of railway rolling stock and the track, which determines the particularities of their static and dynamic interaction [10] while the car passes over the joints.

As different factors, which change during the operation of the track, stipulate the application of the multiple-factor model, research that focuses on improving existing models of interaction between the car and the track structure is modern and of topical interest.

The load of rolling stock elements and track structure determine the parameters of longevity [188], durability and stiffness of the track [8]. These parameters have an impact on operation and service life. The experience of operating rail transport shows [145] that indices of reliability and longevity in the “car-track” mechanical complex greatly depend on particularities of interaction between the track and the rolling stock and operational conditions of the system above. Besides, the interaction above affects the system ability to withstand the destructive action of emerging impact and vibration loads [8], which are of a cyclically recurrent nature.

To analyse dynamic interaction between rolling stock and track structure, it is necessary to solve several interrelated problems, namely, static, impact and dynamic problems. Particular attention is being paid to the issues above, and new sufficient research in this field is being undertaken.

When we consider topical research in this field, when concerning the track, much of it focuses on particular aspects of design and operation. To ensure the application of a complex approach, it is necessary to use a generalized tool integrating different aspects of design and operational conditions, which allows for the complex assessment of track dynamics [9].

Several models have been developed to analyse and predict structural behaviour. The traditional approach is to apply the finite element method, involving commercially available software packages [16], which requires considerable financial costs in the course of designing. Besides, closed-source code does not facilitate the analysis of all interaction aspects [14].

The finite element method is also applied to the ballast model [11], which involves transmitting load from the wheel to the soil where the track is built [13]. Thus, some countries are using reinforced concrete when building roadways [2], but the cost of their construction is significantly higher compared to the standard ballast track design. Another approach is to increase ballast thickness, which leads to decreasing deflection under the load and causes lower stress in the soil. This improves the performance and lifespan of the track [1]. The large number of suggested design solutions prompts several questions as to the efficient ways in which to solve the set tasks. The analysis of modern research demonstrates that the application of a multiple-factor system analysis, which takes into account the phases of motion of the car over joint irregularity and characteristics of track stiffness, represents a current trend in the developing theory of mechanical interaction between the four-axle car and the track. Thus, we can state the need to create an adequate and easy-to-use model of the train-track interaction and the corresponding methods of their dynamic interaction analysis, where the railroad car is considered as a multidimensional discrete system and the track structure is viewed as a continual system.

The purpose of the present research is to study the dynamic interaction between the railroad car and the track structure to improve parameters of the discrete-continuous system by means of rational choice and optimization of the parameters of its components.

Thus, the objectives of the study are as follows: to develop a complex model and method of analysis of the interaction between the four-axle car and the track on the basis of a systemic approach and general correlations of the dynamics, taking into account phases of car motion over joint irregularity; to apply methods of numerical analysis in order to determine and analyse the interaction between the elements of the transport systemic discrete-continuous mechanical complex; and to determine new patterns of mechanical interaction between the four-axle car and the track when the four-axle car passes over joint irregularity with respect to motion phases in order to develop efficient design solutions.

3. MODEL AND METHODS FOR ANALYSING THE DYNAMIC INTERACTION BETWEEN THE TRACK AND THE RAILROAD CAR

A multiple-factor dynamic discrete-continuous model of the four-axle car is analysed in the present research. The four-axle car can be represented by either a tram car, or a passenger or freight wagon. The model takes into account the design parameters and load of the vehicle, connection conditions of the trailing and facing rails through a rail joint plate, and ballast stiffness. The car passing over joint irregularity is considered for all phases of motion. Thus, in the first phase, all the wheel sets are positioned on the trailing rail; in the second phase, three wheel sets remain; in the third phase, there are only two wheel sets; and, in the fourth phase, there is only one wheel set. In the present paper, a mechanical schematic, using the example of the first phase of motion, is given in Fig. 1.

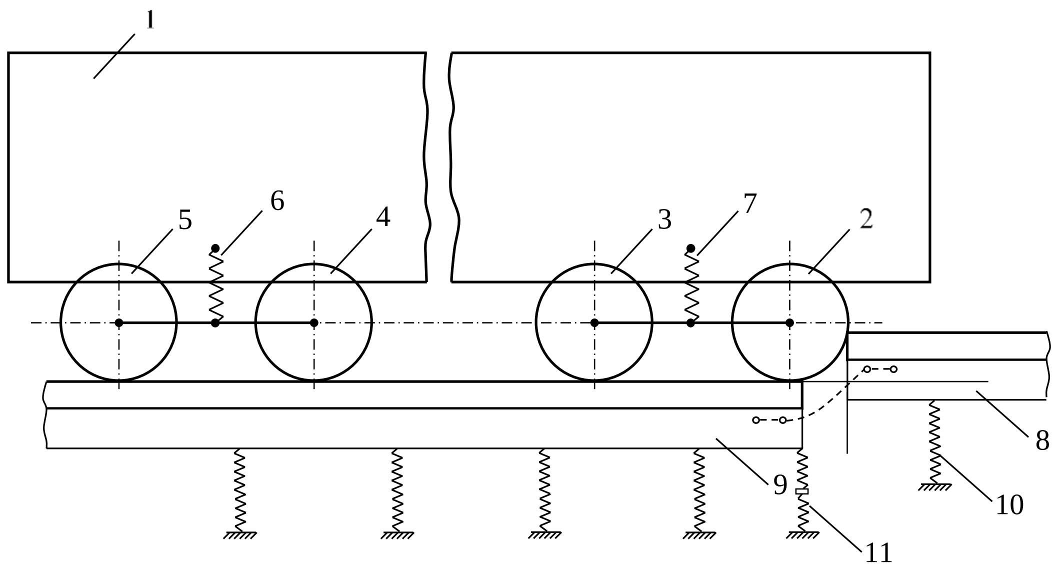

Fig. 1. Schematic for passing over joint irregularity

The label descriptions are as follows: 1) a railroad car of the transportation vehicle; 2-5) corresponding wheel of the wheel set; 6-7) central suspension of the car; 8) facing rail; 9) trailing rail; 10) elastic elements of the ballast under sleepers; 11) elastic element that simulates stiffness of the trailing rail at the end.

Calculation of the interaction at other phases differs according to the number of wheels on the facing and trailing rails. The results of the study concerning other phases of motion are presented as characteristic curves and finite analysis. The suggested integrated approach provides for the consecutive solution of interrelated tasks, as presented in the structural and logical scheme in Fig. 2, which implements the suggested scheme of a consecutive static and dynamic calculation method.

Block 1 - static calculation. Here mechanical characteristics of railway rolling stock and the track and the car load, as well as phases of the car passing and calculating error, are defined. As a result, we obtain the size of the step upstairs and the stiffness of the rail at the end.

Block 2 - calculation of the impact interaction of the wheel and the facing rail. At this stage, mechanical characteristics of the car, trolleys, wheels and rail are specified, as well as the size of the step upstairs, design speed and stiffness of the rail at the end, which allows us to define the after-collision speed of the joint motion of the wheel and the facing rail.

Block 3 - dynamic calculation. At this stage, mechanical properties of the examined system, initial conditions, after-collision speed, static deflection of the facing rail and the phase of the car passing over the joint irregularity are defined. As a result of the calculations, we obtain the maximum deflections of the facing rail under the first sleeper.

By conducting static calculations, we can define rail stiffness at the end as one of the elastic supports (Block 1 [15]). In total, static calculations enable us to define the deflections of the trailing and facing rails (transferred to Block 3), as well as the size of the step (transferred to Block 2). In Block 2, the impact interaction between the wheel and the facing rail is considered. In addition to motion parameters (speed and car load), these calculations define the size of the step. On examining the impact interaction, we can define the vertical speed of the wheel and the facing rail. These values are transferred to Block 3, in which the oscillations of the facing rail, during the phase of increasing its deflection under the first sleeper, are calculated. The data on static and impact interaction are used in this boundary value problem. Static deflection of the facing rail and its vertical speed, as calculated in Blocks 1 and 2, respectively, are used as initial conditions.

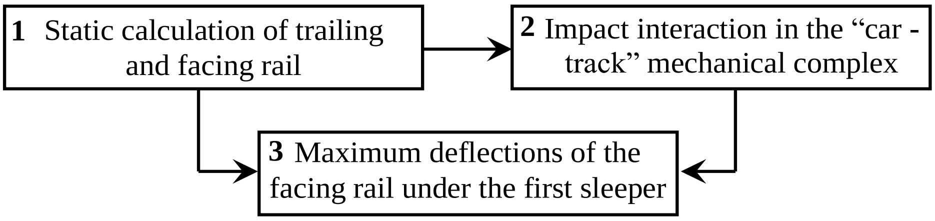

Fig. 2. Structural scheme for the calculation method

3.1. Impact interaction

Let us consider the impact interaction of the wheel and the facing rail in accordance with the structural scheme in Fig. 2. At the beginning of the calculations, the mechanical parameters of the transportation vehicle, and the track and connection conditions of the rails are defined as well as the results of the static calculations of the size of the step upstairs (Block 1 in Fig. 2). At this stage, the parameters of the impact momentum, to which the facing rail is exposed during its impact interaction with the wheel, are defined using the change-of-momentum theorem of the system [15]. Further, the assumption is that, on collision, the facing rail is bent at the same curve as in the case of static load. Then, the motion speed of the rail after collision at the contact point, depending on the load and phase of the car motion over joint irregularity, is defined. The obtained data are transferred to Block 3 (Fig. 2).

3.2. Dynamic interaction

The calculated mechanical schematic of dynamic interaction for the facing rail during the first phase of the car motion is shown in Fig. 3.

The label descriptions are as follows: c_1, b_1 - suspension stiffness and damping coefficient; c_2, b_2 - ballast stiffness and damping coefficient; $c_{p.k}$ - stiffness of the rail at its end (calculated in Block 1 in Fig. 2); m_1, m_2 - reduced masses of the wheel and the car, taking into account the load; y_1; y_2 - displacement of reduced mass of the wheel and the car; and l_i (i=1-24) - geometrical coordinates of the elastic supports.

While calculating the deflections of the discrete-continuous system, mechanical and geometrical characteristics, static deflections and after-collision speed for the facing rail are defined. Differential equations for oscillations in the mechanical schematic in Fig. 3 are given below [37]:

$$\begin{cases} \dfrac{\partial^4 w(t,x)}{\partial x^4} + \dfrac{\rho F}{EJ} \cdot \dfrac{\partial^2 w(t,x)}{\partial t^2} = \dfrac{c_1(y_2 - 0.5w(t,0))\delta(0)}{EJ} - \dfrac{c_{p.k} w(t,0))\delta(0)}{EJ} - \sum\limits_{i=1}^{22} \dfrac{c_2 w(t,l_i)\delta(l_i)}{EJ} \\ -\sum\limits_{k=1}^{K} \dfrac{m_1}{EJ} \dfrac{\partial^2 w(t,0)\delta(l_k)}{\partial t^2} + \sum\limits_{k=1}^{K} \dfrac{P_0 \delta(l_k)}{EJ} - \sum\limits_{i=1}^{22} \dfrac{b_2}{EJ} \dfrac{\partial w(t,l_i)}{\partial t} \delta(l_i) + \dfrac{b_1}{EJ} \left(\dfrac{\partial y_2}{\partial t} - \dfrac{\partial w(t,0)}{\partial t} \right) \delta(0); \\ m_2 \dfrac{\partial^2 y_2}{\partial t^2} + c_1(y_2 - 0.5w(t,0)) - b_1 \left(\dfrac{\partial y_2}{\partial t} - 0.5 \dfrac{\partial w(t,0)}{\partial t} \right) = 0. \end{cases} \quad (1)$$

where: δ(x) - impulse function; w - rail deflection; l_i - the coordinate of the relevant rail; l_k - the coordinate of the relevant wheel on the facing rail; K - the number of wheels on the facing rail corresponding to the phase of motion; F, ρ - the cross-sectional area and density of the rail material; and EJ - bending stiffness.

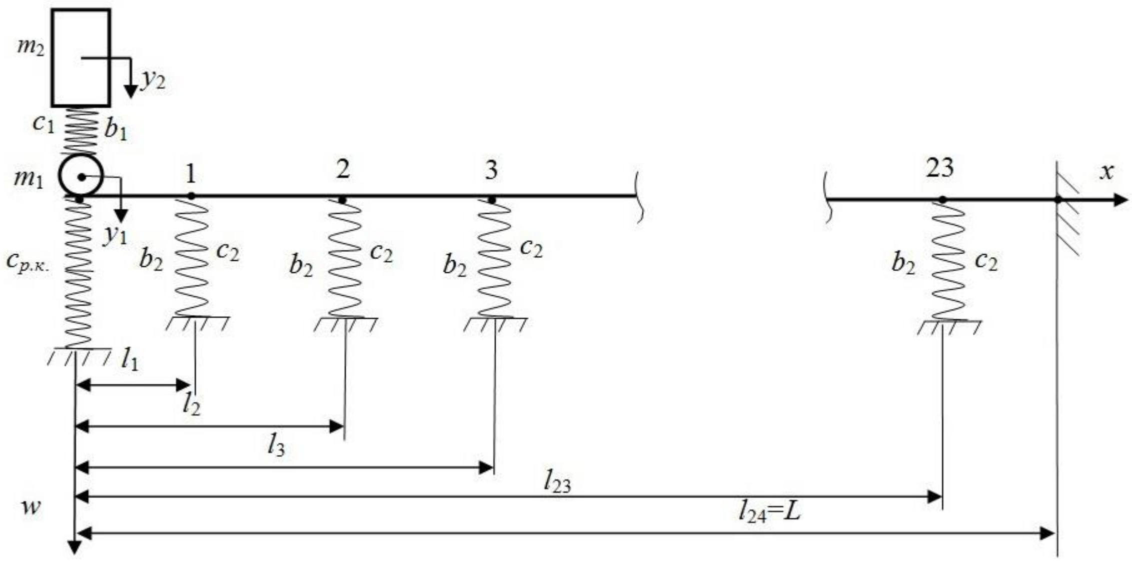

Fig. 3. Calculated mechanical schematic of dynamic interaction for the facing rail during the first phase of the car motion

The system deflections are viewed as a superposition of the first five eigenmodes. Using a model with energy dissipation is undoubtedly feasible under these conditions. However, the damping properties of the ballast are neglected in most calculations due to the fact that the rail deflection is only considered during the phase when it increases.

The solution to the system (1) is performed using the Fourier method for the separation of variables [3] and the Laplace-Carson transform. In the case above, the solution to the problem with the facing rail oscillation is reduced to a superposition of eigenmodes. As a result, the rail deflection, when considering the sprung mass and non-zero initial conditions taken from the static rail deflection (Block 1 in Fig. 2) and after-collision speed (Block 2 in Fig. 2), can be represented by the following expression [3,6]:

$$w(t,x) = \sum_{s=1}^{5} z^s(x) e^{-h_s t} D_s \sin \omega_s t \,. \quad (2)$$

where: D_s - coefficients from mode orthogonality [145] when jointly considering the after-collision speed of the rail with the wheel, $z^s(x)$; ω_s - eigenmode and natural oscillation frequencies of the system; $h_s = b_s / 2m_p$ - dissipation factors of the appropriate form; m_p - reduced mass of the rail; and b_s - reduced resistance factor.

All the above allows us to define the function of the deflections, which is used to obtain the maximum deflection under the first sleeper (x=l_1) of the facing rail at the initial time t=0. When increasing the time gap, the deflection starts to grow until it reaches its maximum value. After that, the deflection starts to decrease, which confirmed that the searching for its maximum values has ended.

4. RESEARCH RESULTS

In accordance with the suggested model, a numerical analysis of the parameters of dynamic interaction between a four-axle car and the track in the area with isolated joint irregularity of the “gap” type has been performed. Calculations have been made on the basis of the variation in operational factors, namely, car load and speed during the relevant phases of the car motion over irregularity, as shown in Figs. 4-7. As a result, we have obtained correlations of maximum deflection under the first sleeper by taking into account the fastening of the trailing and facing rails to rail joint plates, and the geometrical and mechanical characteristics of the rails, plates, sleepers and ballast.

The analysis has been performed in accordance with the calculation pattern in Fig. 2, which has allowed us to define the function of the deflections by solving the differential equations for discrete-continuous system fluctuation (1) in the case of mechanical energy dissipation (2). The results obtained show that the biggest deflections occurred during the second and third phases of the car motion over joint irregularity.

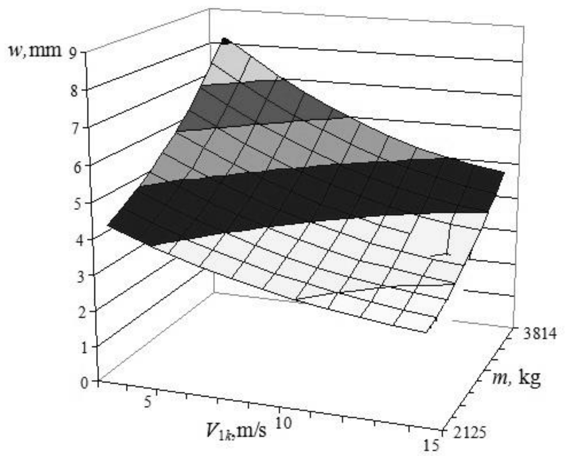

Fig. 4. Facing rail deflection during the first phase of motion

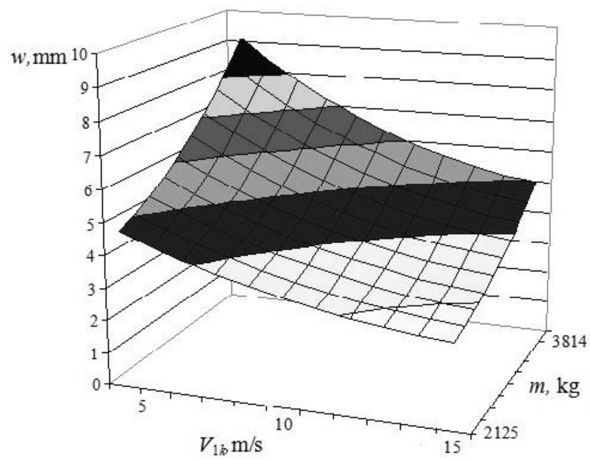

Fig. 5. Facing rail deflection during the second phase of motion

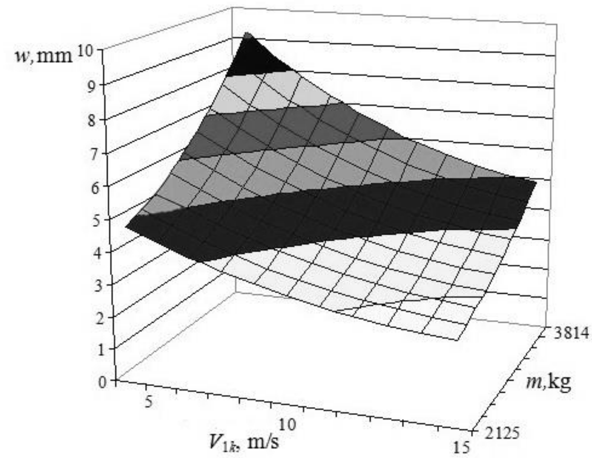

Fig. 6. Facing rail deflection during the third phase of motion

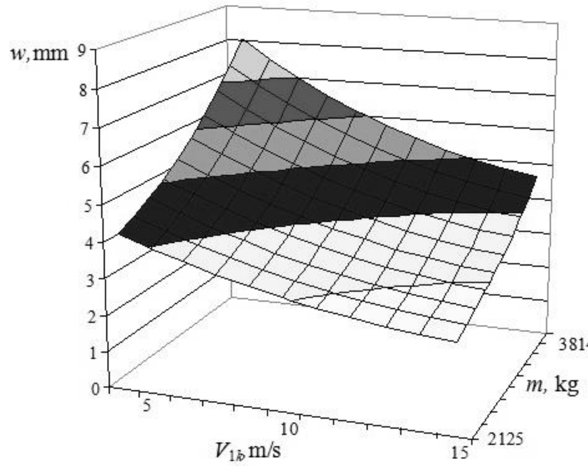

Fig. 7. Facing rail deflection during the fourth phase of motion

The calculations have been performed using the geometrical and mechanical characteristics of the P-65 rail, with the T-3 tram as an example [1], where: the modulus of elasticity of the rail material is E=2.6·10^{11} N/m^2; the moment of inertia of the rail cross section, relative to the neutral axis, is J=3573 cm^4; the suspension and ballast stiffness are c_1=4.225·10^5 N/m, c_2=1·10^8 N/m; and the resistance factors for suspension and ballast are b_1=24·10^3 kg/s, b_2=60·10^3 kg/s. The car's empty weight was reduced to one wheel (=m_1+m_2=2,125 kg) and the maximum weight of a loaded car, i.e., with 193 passengers (m=m_1+m_2=3814 kg; m_1=1,100 kg). Meanwhile, m_p =150 kg represents the reduced mass of the rail, which corresponds to the operational conditions and the design characteristics of the railway rolling stock, and the track and joint plates of real objects.

5. CONCLUSIONS

The numerical calculation results for the parameters of the dynamic car-track interaction in the area of joints are given, with reference to operational, mechanical and geometrical factors. The relevant models are also offered. The research allows us to define new patterns of interaction between the four-axle car and the track while passing over joint irregularity, as well as improving the operational performance and characteristics of the car and the track structure by means of rational choice and optimization. The findings can be applied when developing design solutions in order to improve track joints, defining operation modes of tram cars depending on the state of the track, and developing experimental and theoretical complexes for the purpose of researching, calculating and optimizing the parameters of rail transport knots.

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