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EFFECTS OF RAIL-WHEEL PARAMETERS ON VERTICAL VIBRATIONS OF VEHICLES USING A VEHICLE–TRACK-COUPLED MODEL

Summary. This paper presents an insight into mathematical analysis of dynamic models of the vehicle–track system. After identification of its advantages and disadvantages, an improved three-dimensional "vehicle–track" system of a mathematical model is presented. It not only assesses the influence of realistic track irregularities but also on all elements of the railway structure: rail, rail pads, sleepers, ballasts, and sub-ballast parameters on the wagon's movement smoothness. Based on the expanded mathematical model of the "vehicle–track" dynamic system, the dynamic process of the wagon was theoretically studied, and the effect of track with irregularities on the vibrations of the wagon elements was studied. The final conclusions and recommendations are presented.

1. INTRODUCTION

When running a train on rails, natural vibrations of the rolling stock occur, whereas the forced vibrations of the rolling stock are generated due to the track irregularities. Upon emergence of defects on the rolling surface of the wheel, dynamic forces operate during the wheel–rail interaction that place an additional load on the rail and the wheel. With an increase in vehicle speed, the dynamic forces caused by these disturbances also increase [1]. Due to the rolling stock vibrations induced by these forces, a traffic accident, noise, and a negative effect on the passenger comfort during a trip may occur [2]. The specialists involved in the operation and repair of passenger vehicles are faced with a problem of precise and reliable evaluation of the car body vibrations, induced by a wagon with wheel defects, and its impact on passengers [3]. Another important aspect is to determine the permissible driving speed by which the passenger wagon with wheel defects could move to the nearest station or depot without damaging the rails. It is a very expensive and long-lasting process for the specialists of the companies in charge of the technical maintenance of the railway vehicles to perform field testing of passenger vehicles (pilot running): the train traffic in the span should be closed and the vehicles in operation should be used for testing. Furthermore, a danger arises that during testing of vehicles, cars with wheel defects can irreparably damage both the rails and the wheelsets [4]. Therefore, researchers are striving to develop new and improved existing mathematical models of the "vehicle – track" system to replace very expensive natural experiments.

2. ANALYSIS OF DYNAMIC MODELS OF THE "VEHICLE–TRACK" SYSTEM

The "vehicle–track" system is usually simulated as two separate but interactive models of vehicles and tracks that assess vehicle and track damages. The interaction of these components can be analyzed using the respective mathematical equations based on reliable physical models [5]. Each railway vehicle is composed of three main components: body, bogies, and wheelsets.

During the development of dynamic vehicle models, it is necessary to evaluate the aforementioned vehicle components, their interaction, and connections. The elastic and damping devices that connect the wheelsets with a bogie frame constitute the initial stage of railway vehicle suspension. The secondary suspension stage connects the vehicle body to the bogie frame.

Vehicles can be simulated as one-dimensional, two-dimensional, or three-dimensional models. The one-dimensional model is used in many articles [6, 7, 8] that analyze the high-frequency oscillations in the wheel–track system. Two-dimensional models and models with more degrees of freedom [9, 10] describe vehicle–track interaction by assessing various damages to the wheel. Three-dimensional vehicle models usually include the 1/2 body, the entire bogie, and both wheelsets. Such models examine the oscillation of the vehicle elements by evaluating the track roughness, wheel damages, and suspension peculiarities [11, 12]. Later, these models were improved by introducing increasingly more elements that evaluate the dynamics of the transported cargo [13] and its influence on vehicle oscillations, as well as clarifying the dynamics of the vehicle suspension [14] with additional non-linear suspension models. The development of computer technology and software allows the development of powerful dynamic simulation software packages that include already manufactured, very complex multi-degree vehicle models. Multi-Body Simulation (MBS) Software SIMPACK, VI-Rail software of VI-GRADE company, or the "UNIVERSAL MECHANISM" software packages from the Bryansk State Technical University in Russia provide very extensive simulation capabilities. However, many of these software packages do not assess wheel damage and do not provide accurate track models. The stiffness and suppression of the track are usually taken into account, irrespective of the parameters of the sleepers, the ballast, and the railway bed, and their influence on movement evenness.

The track model portion of the "vehicle–track" system may consist of rails (Fig. 1), sleepers, and a railway bed depending on the desired accuracy. The railway bed consists of ballast, sub-ballast, and soil layers:
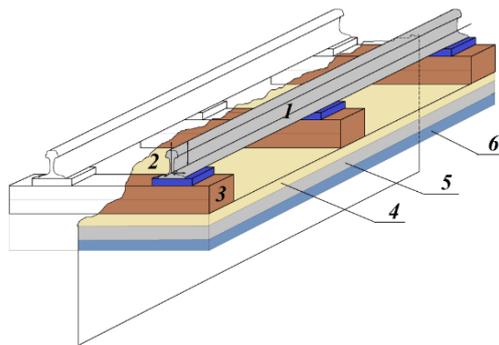
Fig. 1. Railway elements: 1 – rail, 2 – pad, 3 – sleeper, 4 – ballast layer, 5 – sub-ballast layer, 6 – soil layer

Railway track dynamic models can be discreet or continuous. Discreet models of railway track include separate rails with sleepers and ballast. They can be divided into infinite-length and defined-length models. Floquet's theorem [15 - 17] is applied for the analysis of these models. The finite element method can be used with success for the definition of elastic characteristics of pads under a rail and under a tie plate [18]. The continuous track model examined an infinite Bernoulli–Euler beam on an elastic basis of Winkler under static and dynamic loads. Later, the concept of the Bernoulli–Euler beam exposed by harmonious moving forces was extended to the Timoshenko beam [19, 20]. However, they use a simplified vehicle model of ½ mass.

Summarizing the dynamic model analysis of the "vehicle–track" system, it can be said that the complex dynamic vehicle model, including faults and wear of the vehicle during operation, is most commonly used in the analysis of the "vehicle–track" system, but a very simple track model is also often used. In other cases, the simplified vehicle model is used to analyze the railway track and its deficiencies. To obtain reliable results, it is necessary to equalize the precision level of all components of the models being examined. Many mathematical models of vehicle's vertical forces assess only a certain part (1/4, 1/8) of the calculated masses of vehicle elements and do not assess the dynamic effect of all vehicle elements on oscillations.

3. DYNAMIC MODEL OF A PASSENGER VEHICLE WITH THE WHEEL FLAT RUNNING ON THE RAILWAY TRACK

The mathematical model of the "vehicle–track" system developed by the authors is used to analyze the vertical interaction of vehicles and railway tracks by assessing the rotary fluctuations of the body, bogies, and wheelsets, damage of rolling wheels, and roughness of the railway track. One of the objectives is the oscillation analysis of the vehicle as a dynamic system to establish the impact parameter of the initial and secondary suspension for vertical body oscillations. This model provides an opportunity to analyze the effects of damage to the rolling wheel surface on system oscillations in case of few damages on the same and separate wheelsets. The developed mathematical model is different from other "vehicle–track" system models described in scientific publications sources since it uses the entire passenger vehicle with one or few damaged wheels on the rolling surface, instead of half or a quarter of the vehicle. The model assesses the impact of the parameters of all railway track structural elements (rail, pads, sleepers, ballast, and sub-ballast) on the evenness of vehicle movement. The expanded dynamic model assesses not only the vertical displacements of the main masses but also the angular displacements. The model contains 53 degrees of freedom. A passenger car model adapted to this research consists of the entire body, two bogies, and four wheelsets. This model, which contains 17 degrees of freedom, determines the vertical displacements of the body, bogies, and wheelsets. The rotation of the model elements around the x and y axes is also evaluated (Fig. 2).

A rail track model estimates the vertical displacements of the rail, sleeper, and ballast layer ($Z_{b_{ij}}$, Z_{pb_i}, $Z_{snk_{ij}}$) under each wheelset during the movement of the car. In the simulation of the rail track, the following separate elements with mass have been distinguished: rails, sleepers, and ballast layer – M_b, M_{pb}, M_{snk}. The rotation of sleepers is described by the value ϕ_{pb_i} (Fig. 3). All the main railway track elements interact as a system of stiffness and damping elements. The motion equations describing a model of vertical forces are written in matrix form without external forces:

$$[M_v]\{\ddot{z}\} + [C_v]\{\dot{z}\} + [K_v]\{z\} = 0 \tag{1}$$

where $[M_v]$ is the car model mass matrix, $[C_v]$ is the damping matrix, $[K_v]$ is the stiffness matrix, and z is the vertical displacement vector.

The above-described mathematical model of railway track assesses how the forces caused by wheel rolling damage are transferred not only to the rail but also to the entire track structure in the vertical direction. However, to assess the oscillations dispersing along the railway track (*x* axis), it is necessary to describe the rail with finite elements. It allows the mechanical effects transferred to the second wheelsets to be established via the rails, and the oscillations of the bogie and the body to be assessed with greater precision. The dynamic model of railway track described with finite elements is shown in Fig. 4. An assumption is made that the whole zone of impact of the wheelset with a flat consists of the central finite element L_e and two finite elements on each side L_{Le} and L_{Re}. Supports on the edges of the first and last finite element are absolutely stiff. According to the dynamic model of the track, a wheelset impact through the central finite element L_e is distributed on two sleepers and ballasts, and is further transmitted to the finite elements L_{Le} and L_{Re}. To assess the influence of L_{Le} and L_{Re} elements on the whole system, the displacements of the finite elements of the rails should be determined on the Z axis:

$$w(\xi) = [N(\xi)]\{q_e\} \ , \tag{2}$$

where $\xi = \dfrac{x - x_1}{5L_p}$, when $x_1 \le x \le x + 5L_p$ is the dimensionless coordinate; $5L_p$ is the finite element

length, $\{q_e\}$ is the displacement vector, x is the total covered distance, x_1 is the distance to the covered sleeper; and $[N(\xi_i)]$ – is the matrix of the finite element form functions.

The potential energy of the left-side rail finite element L_{Le} is equal to:

$$E_{pLe} = \frac{1}{2}\{q_L\}^T[K_{Le}]\{q_L\} + \frac{1}{2}\sum_{i=2}^{6}\{q_L\}^T[N(\xi_i)]^T k_{Li}[N(\xi_i)]\{q_L\} =$$

$$= \frac{1}{2}\{q_L\}^T[K_{Le}]\{q_L\} + \sum_{i=2}^{6}\frac{1}{2}k_{Li}w^2(\xi_i), \tag{3}$$

where $[K_{Le}]$, $\{q_L\}$ is the left-side finite element stiffness matrix and displacement vector, which equals: $\{q_L\}^T = [0, 0, q_1, \varphi_1]$.

The generalized force vector of the left-side finite element is equal to:

$$\{Q_{Le}\} = -\frac{\partial E_{pLe}}{\partial\{q_L\}} = -\left([K_{Le}]\{q_L\} + \sum_{i=2}^{6}[N(\xi_{c1})]^T k_{Li}[N(\xi_{c1})]\{q_L\}\right) = [K_L]\{q_L\}. \tag{4}$$

The potential energies and force vectors of elements L_{Re} and L_e are described in an analogous way, where displacement vectors equal $\{q_R\}^T = [q_2, \varphi_2, 0, 0]$ and $\{q_e\}^T = [q_1, \varphi_1, q_2, \varphi_2]$, respectively.

A system of equations for the movement of the rail finite elements L_e, L_{Le}, and L_{Re} is formed:

$$[M_b]\{\ddot{q}_b\} + [C_b]\{\dot{q}_b\} + [K_b]\{q_b\} = \{F_b\} \tag{5}$$

where $[M_b]$, $[C_b]$, and $[K_b]$ are the rail finite element mass, damping, and stiffness matrices, respectively; $\{q_b\}$, $\{\dot{q}_b\}$, and $\{\ddot{q}_b\}$ are the appropriate point displacement, velocity, and acceleration vectors.
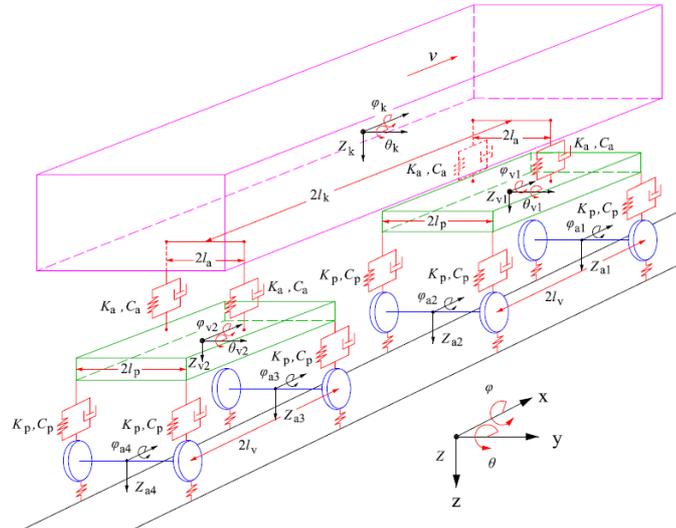
Fig. 2. Scheme of the vehicle model

The nonlinear system of equations of the rail movement is equal to:

$$\{\ddot{q}_b\} = \{F_{\Sigma b}(q_b, \dot{q}_b)\}, \tag{6}$$

where $\{F_{\Sigma b}(q_b, \dot{q}_b)\} = [M_b]^{-1}(\{F_b\} - [K_b]\{q_b\} - [C_b]\{\dot{q}_b\})$.

The novelty of this model lies in the fact that, upon integration of the derived nonlinear system of equations of the rail movement into a mathematical model of the "vehicle – track" system, it is possible to assess precisely the influence of the vibrations that propagate along the railway track on adjacent wheelsets.

Hertz theory was confirmed by the results of the research carried out during the last decades. Many of the latest scientific works describe contact problems between wheels and focus on the contact of wheels

and rails with defects of the running surfaces [21, 22]. All mathematical model equations of the "vehicle–track" system are solved together with the Hertzian contact. This makes it possible to define the interaction forces between all elements of the system under consideration and to find the accelerations.
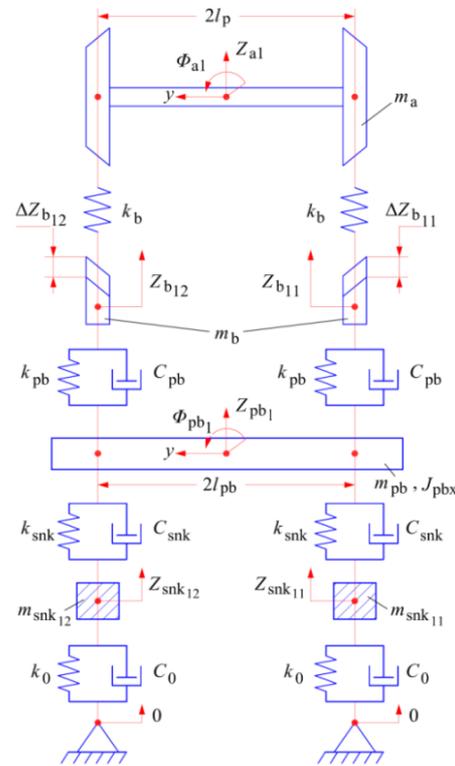
Fig. 3. Scheme of the track model

All the equations of the "Rail vehicle–track" system for calculating the fluctuation and contact forces described above were uploaded to the MATLAB/Simulink software package environment. The initial model parameters and their values are presented in Table 1.
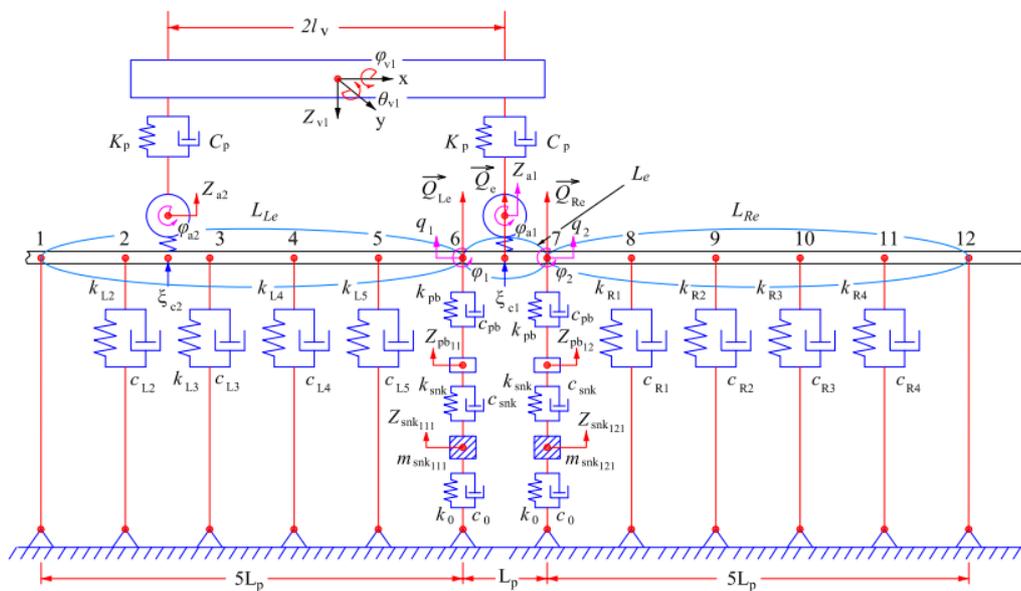
Fig. 4. A dynamic model for the interaction of railway track elements upon evaluation of the interaction of finite elements

Table 1

Input parameters of the system "vehicle–track"

Model parameter	Marking, dimension	Value
Vehicle mass	M_k, kg	38 000
Bogie mass	M_v, kg	2 600
Wheelset mass	M_a, kg	1 750
Rail mass (1 m)	M_b, kg/m	65
Sleeper mass	M_{pb}, kg	200
Ballast mass	M_{snk}, kg	300
Mass moment of inertia of the car body about the x axis	J_{kx}, kg×m^2	50 000
Mass moment of inertia of the car body about the y axis	J_{ky}, kg×m^2	2 310 000
Mass moment of inertia of the bogie about the x axis	J_{vx}, kg×m^2	1 800
Mass moment of inertia of the bogie about the y axis	J_{vy}, kg×m^2	2 000
Mass moment of inertia of the wheelset about the x axis	J_{ax}, kg×m^2	420
Mass moment of inertia of the sleeper about the x axis	J_{pbx}, kg×m^2	90
Primary suspension stiffness	K_p, N/m	1 100 000
Primary suspension damping	C_p, N×s/m	13 052
Secondary suspension stiffness	K_a, N/m	600 000
Secondary suspension damping	C_a, N×s/m	17 220
Herc contact stiffness coeficient	C_H, N/m$^{3/2}$	9.4×10^{10}
Rail stiffness	K_b, N/m	4,4×10^7
Railpad stiffness	K_{pb}, N/m	1.2×10^7
Railpad damping	C_{pb}, N×s/m	45 000
Ballast stiffness	K_{snk}, N/m	4×10^7
Ballast damping	C_{snk}, N×s/m	50 000
Sub-Ballast stiffness	K_0, N/m	5×10^7
Sub-Ballast damping	C_0, N×s/m	40 000
Half distance between the bogies pivots of the wagon	l_k, m	8,5
Half the distance between wheelsets of the bogie	l_v, m	1.25
Half the distance between the mounting points of the primary suspension elements	l_p, m	0.8
Half the distance between the mounting points of the secondary suspension elements	l_p, m	0.8
Half the width of the sleeper	l_{pb}, m	1.2
Wheel radius	R, m	0.4505
Flat depth	h, m	0.001
The radius of the curve of the flat	r, m	0.1

3.1. Research on vertical oscillations of passenger vehicle wheelsets

Since the extended model of the "vehicle–track" system provides an opportunity to assess the realistic track roughness of both rails separately, the vertical oscillations and angular accelerations of all vehicle wheelsets have been established when moving on even tracks and tracks with a realistic roughness. Figs. 5, 6, and 7 provide vertical accelerations of wheelsets, when the first wheelset has a 1 mm depth and a 20 mm flat length. The moving speed of the vehicle is 80 km/h.

Fig. 5 shows the wheelsets accelerations caused by the flat wheel and its change in value. The maximum vertical acceleration of the first set flat wheels can reach up to 240 m/s^2. The comparison of Figs. 5a and c shows a significant impact of road roughness on the vertical oscillations of the first and second wheelsets. As can be seen, the maximum vertical acceleration of the first set of wheels increases by a factor of 1.2, and the maximum values of the second set of wheels increase by a factor of 25 – up to 169 m/s^2.

The vertical accelerations, transferred to the second set of wheels via the vehicle bogies and the initial suspension stage, are provided in Fig. 6a.

The oscillations of the second set of wheels are low and their maximum values reach only 1.1 m/s^2 without assessing the mechanical impact of the first set of wheels that is dispersed along the track. However, after having assessed the mechanical impacts of the first set of wheels transferred along the track to the second set of wheels (Fig. 6b), the vertical accelerations of the latter increase by a factor of 6 - up to 6.4 m/s^2, and continuous stimulation is observed.

It is observed that the acceleration peak of the second set of wheels is reached slightly later than the impact caused by the flat wheel, which occurs due to the operation of the initial suspension elements during the transfer of bogie oscillations.

Even smaller accelerations are recorded in the 3rd and 4th wheelsets of other bogies. They are transferred from the 1st bogie to the body via the initial and secondary suspension, causing the oscillations of the 2nd bogie and transfer to the wheelsets via the suspension of the 2nd bogie. The vertical acceleration values of the 3rd and 4th wheelsets of the 2nd bogie are nearly the same and very small. They are 6×10^5 times smaller than the values of the first set of flat wheels. The rolling acceleration changes over time of the 1st and 2nd wheelsets are given in Fig. 7.

Fig. 7 b shows that the rolling accelerations of the second set of wheels are small and almost 100 times smaller than the accelerations of the first wheel set (maximum values do not exceed 0.8 m/s^2), but they systematically repeat the first wheel set. The rolling accelerations of the 3rd and 4th wheelsets are very small, and, thus not assessed. However, the introduction of track roughness changes the view of the roller wheelsets essentially (Fig. 7b and 7d). The additional stimulation caused by track roughness (Fig. 7b) and the significantly increased (up to 35 times) accelerations of the second set of wheels (Fig. 7d) can be clearly seen.

It can be concluded that impact accelerations caused by flat wheels have a strong influence on the vertical oscillations of the 1st bogie wheelsets; however, it has little impact on the 2nd bogie wheelsets. The railway track roughness has a huge influence on the oscillations of the wheelsets, exerting an additional mechanical impact on the wheelsets. The research shows that it is necessary to assess changes in railway track stiffness. These changes strongly affect the vertical oscillations of the wheelsets.

3.2. Research on vertical oscillations of passenger car bogies

The vertical oscillations of wheelsets caused by track roughness and wheel rolling surface are transferred to vehicle bogies via the initial stage of suspension. The oscillations of the 1st bogie are transferred to the 2nd bogie via the secondary stage of the suspension and the body. The vertical, roll, and pitch accelerations of these vehicle bogies are shown in Figs. 8 and 9 at a speed of 80 km/h.

Fig. 8a shows the vertical oscillations of the 1st bogie (up to 0.8 m/s^2) caused by the flat wheel, which are transferred to the 2nd bogie via the body and the chassis (Fig. 8b). However, the oscillations of the 2nd bogie are very low (1.1×10^{-2} m/s^2), and only the waves caused by body movement were observed. After having introduced the real roughness of the railway track (Fig. 8c), the accelerations of the vertical oscillations of the bogies increased significantly. The vertical acceleration values of the 1st bogie increased to 9.7 m/s^2.

During the vehicle movement on a railway, not only the vertical but also the angular displacements of bogies are assessed – θ_{y1}, θ_{y2}, and ϕ_{y1}, ϕ_{y2}. The time variations of these roll and pitch accelerations are shown in Figs. 9 and 10.

Fig. 9 shows that the roll and pitch accelerations of the 1st bogie on an even track vary greatly in comparison with the movement on the track with roughness. The maximum values of these accelerations increase from 7 times (pitch acceleration) to 3.5 times (roll acceleration).
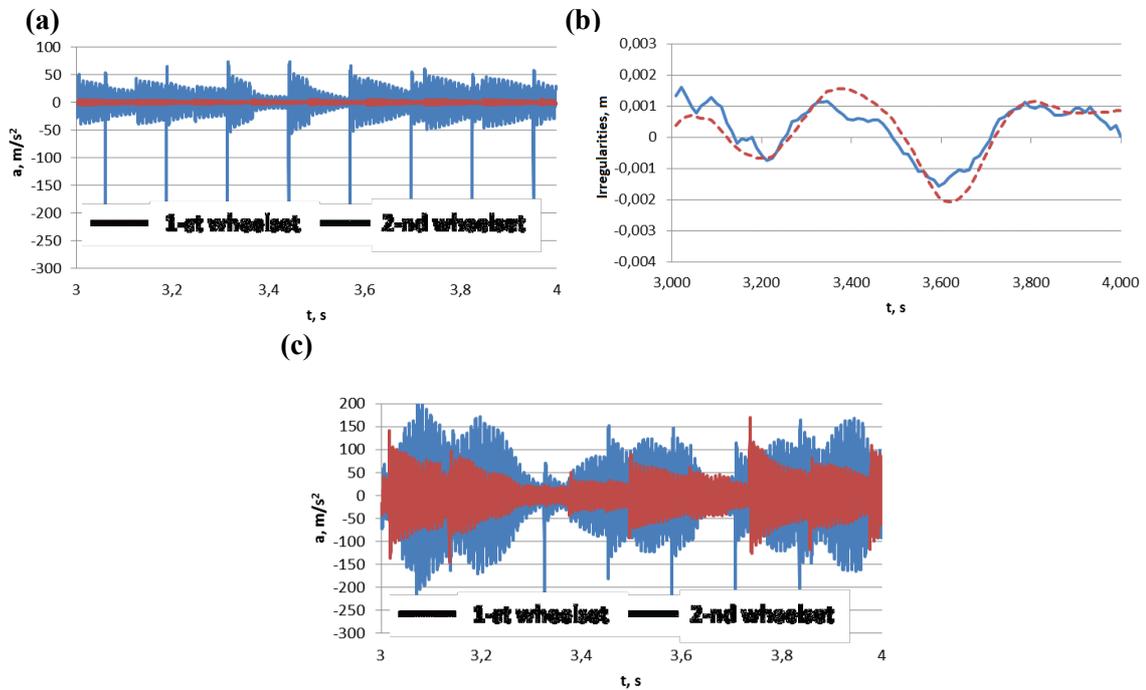
Fig. 5. Changes in wheelsets vertical accelerations of the 1st bogie over time 3.0–4.0 s: (a) the 1st and the 2nd wheelsets, track without irregularities; (b) changes in track irregularities (left and right sides); (c) the 1st and the 2nd wheelsets, track with irregularities
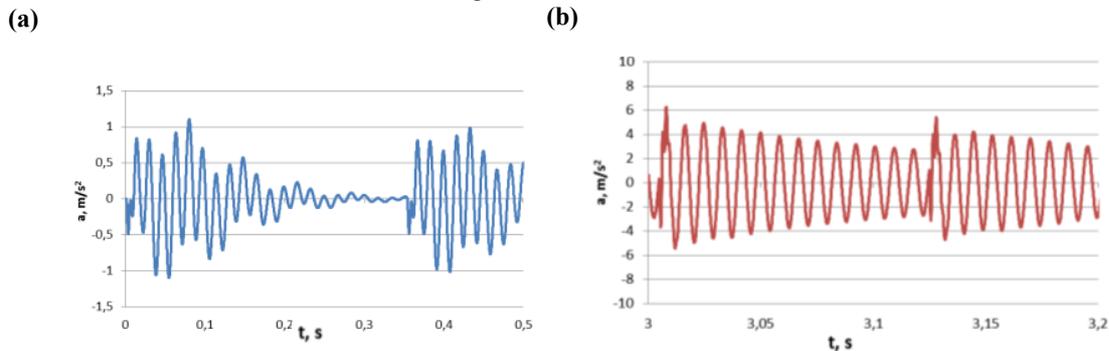
Fig. 6. Changes in vertical accelerations of the 2nd wheel set over time, track without irregularities: (a) the 1st wheel set mechanical impact, transmitted by track, was not evaluated; (b) the 1st wheel set mechanical impact, transmitted by track, was evaluated

The pitch and roll accelerations of the 2nd bogie on an even track are very small (the values do not reach 2×10^{-3} rad/s^2); hence, these were not evaluated.

The analysis of the oscillation graphs provided in Figs. 8-9 shows that the impact of the flat wheel on vertical and angular accelerations of the 1st bogie is sufficiently strong, but the oscillations of the 2nd bogie stimulated by the flat wheels are weakly noticeable in the oscillations caused by track roughness.
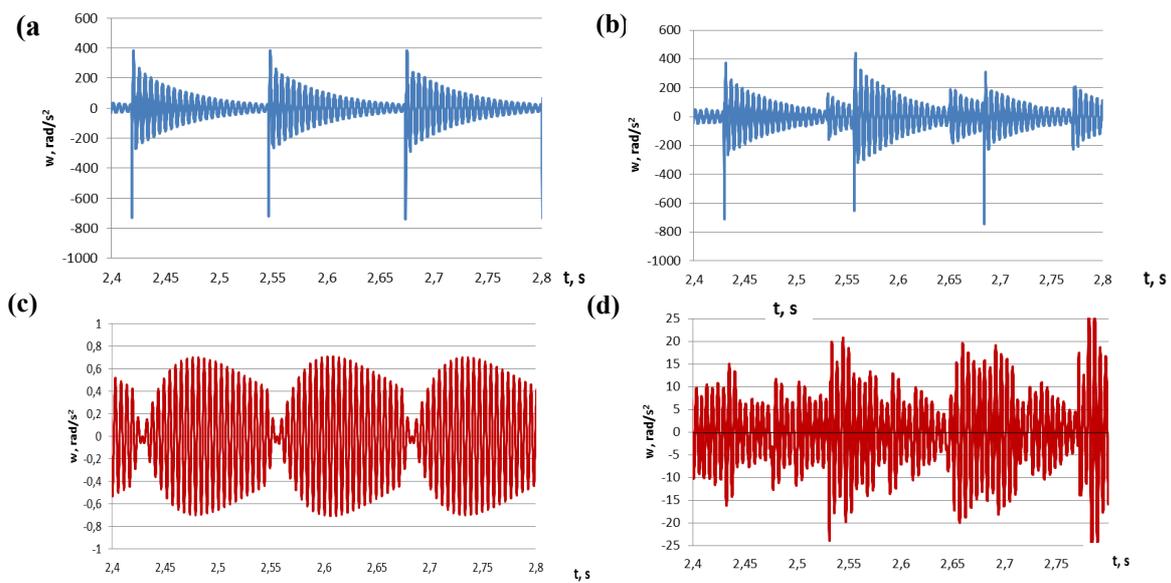
Fig. 7. Changes in roll accelerations of wheelsets over time: (a) the 1st wheel set, track without irregularities; (b) the 1st wheel set, track with irregularities; (c) the 2nd wheel set, track without irregularities; (d) the 2nd wheel set, track with irregularities
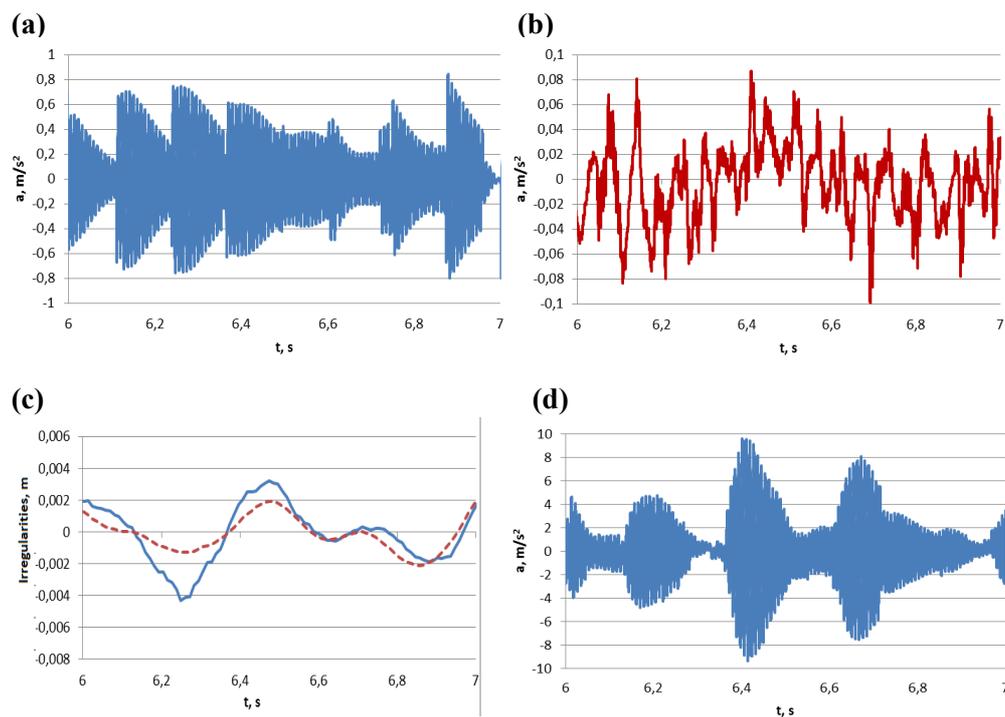
Fig. 8. Changes in vertical accelerations of bogies over time 6.0–7.0 s: (a) the 1st bogie, track without irregularities; (b) the 2nd bogie, track without irregularities; (c) changes in track irregularities (left and right sides); (d) 1st bogie, track with irregularities

(a)

(b)

(c)

(d)
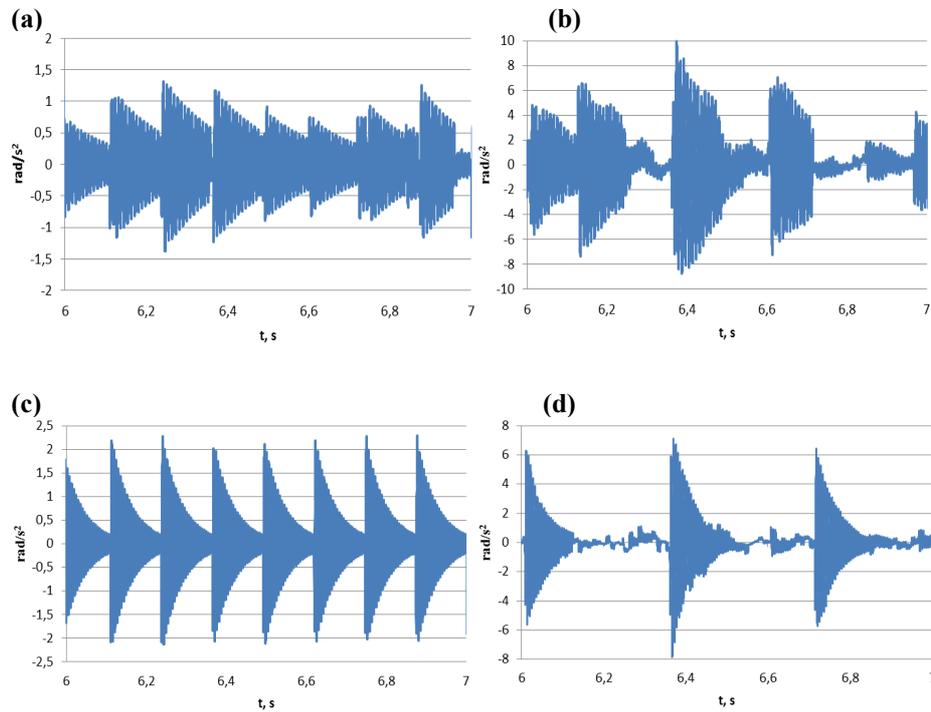
Fig. 9. Changes in roll and pitch accelerations of the 1st bogie of wagon: (a) $\ddot{\theta}_{v1}$, track without irregularities; (b) $\ddot{\theta}_{v1}$, when track with irregularities; (c) $\ddot{\phi}_{v1}$, track without irregularities; (d) $\ddot{\phi}_{v1}$, track with irregularities

It can be concluded that the flat wheel has a strong impact on the oscillations of the 1st bogie, but is barely noticeable for the 2nd bogie. The transferred oscillations are so low that they can be ignored.

3.3. Research on vertical oscillations of the passenger car body

During the vehicle movement on the railway track, the wheelsets with flat and track roughness cause the main vertical oscillations. These oscillations are transferred to vehicle bogies via the initial suspension and the bogie oscillations are transferred to the vehicle body. The developed mathematical model allows the determination of the impacts of track roughness and wheel damages on the vertical and angular accelerations of the body in various seasons. The calculation results of vertical body accelerations are given in Fig. 10.

However, after having introduced the track roughness, the body oscillations caused by the flat wheel basically disappear in the oscillations caused by the track. The vehicle body oscillations accurately repeat the changes in track roughness.

Table 2

Values of Sperling's ride index for the evaluation
of vehicle running smoothness

S/N	Vehicle running smoothness	Index value
1	Very good	2.0
2	Good	2.0–2.5
3	Enough for passenger wagon	2.5–3.0
4	Critical for passenger wagon	3.0–3.25
5	Critical for locomotive	3.5–3.75
6	Critical to human physiology	4.5

(a)

(b)

(c)
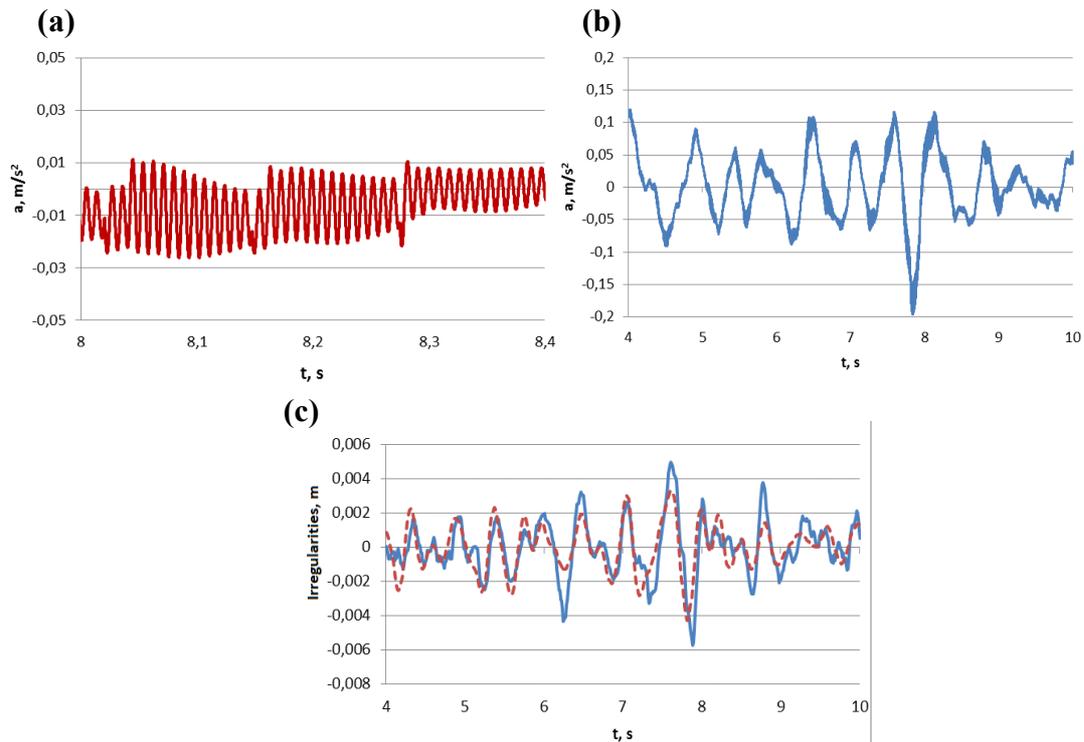
Fig. 10. Variation in the vertical accelerations of the car body: (a) on the even track in the period of 8.0–8.4 s; (b) track with irregularities in the period of 4.0–10.0 s; (c) changes in track irregularities (left and right sides) over time 4.0–10.0 s

After processing of the data obtained during the simulation by means of the Fourier transform method, the dependence of the vehicle body acceleration amplitude repetitions on the running time was obtained. These data make it possible to assess the comfort of the passengers through Sperling's ride index. The obtained values of indicators are shown in Fig. 11.
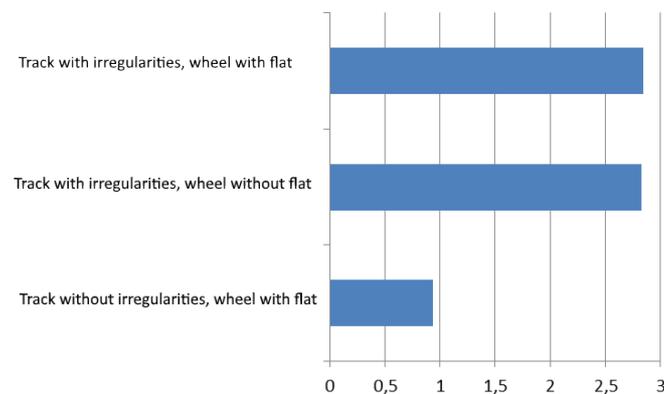
Fig. 11. Values of Sperling's index

Fig. 11 shows that the indicator values of smooth running are very good (see Table 2) when the passenger cars operate with a irregularity of 1 mm depth and a length of 20 mm on an even track, but after having introduced track roughness, Sperling's index values increased nearly 3 times and approached the limit values for passenger vehicles.

4. CONCLUSIONS

1. The developed extended mathematical model of interaction between the railway vehicle and the track structure evaluates the vertical oscillation parameters of the upper structure of the railway track, the bogies of the running vehicle, and the body. The mathematical model enables the establishment of the vertical interaction of rail vehicle elements, evaluating with greater precision the parameter changes of the track elements.
2. The significant impact of track roughness on the vertical oscillations of the 1st and 2nd wheelsets was established. After having introduced the track roughness, the maximum values of vertical acceleration of the first set of wheels increased by 1.2 times and the maximum values of the second set of wheels increased by 25 times.
3. If the mechanical impact of the first set of wheels dispersing along the track is not evaluated, the oscillations of the second set of wheels are small and their maximum values reach only 1.1 m/s^2. However, after evaluation of the mechanical impact of the first set of wheels transferred to the second set of wheels along the track structure, it was found that the vertical accelerations of the latter increased by a factor of 6 – up to 6.4 m/s^2, and continuous stimulation was observed.
4. After having introduced the real roughness of the railway track, the accelerations of the vertical bogie oscillations increased significantly. The values of the vertical accelerations of the 1st bogie increased by a factor of 12. The pitch and roll accelerations of the 1st bogie on an even track vary greatly in comparison with the running on the track with roughness. The maximum values of these accelerations increased from 7 times (pitch acceleration) to 3.5 times (roll acceleration).
5. After processing of the data obtained during the simulation by means of the Fourier transform method, the dependence of body acceleration amplitude repetitions on running time was obtained and Sperling's ride index was calculated. The results show that the indicator values of smooth running are very good if the vehicle moves with a depth of 1 mm and a length of 20 mm flat on an even track, but after having introduced track roughness, the values of these indicators increased nearly 3 times and approached the limit values for passenger vehicles.
6. The results obtained from the theoretical study on the running passenger car enable the expansion of calculation capabilities by assessing the impact of the additional railway track and vehicle parameters on the vertical dynamic processes of operation.

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