A study of the contact of the wheel with the rail for various conditions of freight car

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Abstract. The paper presents the results of calculations of the parameters of contact interaction of the wheels of a freight car with R65 rails (wheel wear rate, areas of wheel contact spots with rails, friction power capacities in contact spots) for various operating conditions on the Russian Railways road network. The choice of groups of operating conditions was made on the basis of the methodology for ranking the characteristics of the main roads in the plan and profile. The quantitative values of the parameters of contact interaction were established, and their comparative analysis was carried out for the conditions under consideration.

1. Introduction

The aim of the work was to set up a full-scale experiment to determine the parameters of wear and lick formation of wheel steel for further evaluation of the life of the wheel. The article describes the solution of the problem of determining the parameters of the contact of the car wheels with rails (the size of the contact patch, creep forces, friction and pseudo-slip force, wheel wear) and ranking according to the crew's different operating conditions on Russian railways.

The task of determining the factors that most affect the indicators that assess the intensity of wear is relevant. This will help to further improve the crew system - a way by adjusting and managing the permissible levels of these factors. For work performed earlier on this subject, a characteristic approach to the selection of the most significant parameters is based on experience and intuition.

The object of the study was a universal freight gondola, installed on traditional trolleys of model 18-100 with wheels according to GOST 10791 [1], with a calculated static axial load of 23.5 ton, and a passenger car mounted on KVZ-TSNII type I trolleys (container of the car 60 t). The selection of representative sections of the path, characteristic of the existing regional divisions of the Russian Railways network, was made on the basis of an analysis of the statistical data of the main parameters for the main roads of the Russian Federation. The parameters of the contact interaction of wheels with rails were determined in the Medyna software package [2].

2. Formulation of the problem

As a result of the analytical review of the data [3-6] on the wear of wheel pairs, depending on the track width, curve radius, rail pivot, and outstanding acceleration. It was found that:

- an increase in the gauge leads to a decrease in the specific work of the friction forces and the intensity of wear in the contact wheel-rail. The narrowing of the track width by 4 mm (from 1524 to 1520 mm) leads to an increase in the specific work force of friction between the wheels and the rails of the crew by 3 - 10%, depending on the radius of the curves. And in the whole studied range of the gauge width (1510 - 1550 mm), the total specific work of friction forces can vary up to 3 times (with a curve radius of 650 m);

– in curves with a radius of 350 m when driving at a speed close to the equilibrium, the wear characteristics and the specific work of friction between the wheels and rails for loaded cars decrease: in the range of track width 1520 - 1530 mm - by 22%, with increasing track width over 1530 to 1,545 mm wear parameters are additionally reduced by 20–30%;

– in straight sections of the track with increasing track width in the range from 1510 to 1540 mm, the specific work of friction forces varies slightly. However, in the range of 1520 – 1530 mm there is its minimum. The influence of the track width on the wear rate of the rolling surface and wheel flanges largely depends on the crew’s inclination to wag and the degree of wear on the wheels;

– an increase in the bottom shaft causes an increase in the specific work force of friction and wear intensity in a pair of wheel-rail mainly during the period of running-in, since the contact conditions change: contact pressure and relative slip increase;

– the elevation of the outer rail has a certain effect on the wear of wheelsets and rails. The mechanism of this influence is determined by the inversely proportional relationship between the magnitude of the elevation and the unpaid acceleration, at a constant speed of the rolling stock, while reducing the unpaid acceleration by 0.3 m / s^2, which is allowed by regulatory documents, the wear rate decreases by 7-9%.

3. Theory

A method [4, 7] for joint modeling of the wear of wheel and rail profiles was proposed, taking into account the statistical variety of factors affecting the results of calculations. Due to the fact that any fragment of the railway network, taken as the object of modeling, has straight, transitional and curved sections of the route of different radius, and the rolling stock operated on this section, as a rule, varies in type and technical condition, the process of wear of profiles wheels and rails should be considered as partially deterministic. The size of the wear profile of the wheel and rail, therefore, it is advisable to determine, using the methods of probabilistic modeling. Thus, representative routes with heavy, normal and favorable conditions from the point of view of the wheel resource realization were identified from the entire railway network (see Table 1).

Table 1. Parameters of representative.

Curve radius (m)	Accepted for calculation radius (m)	Length (km)	Circular curve length (m)	Length of transition first (m)	Elevation of the outer rail in a curve (mm)	Crew movement speed (m / s (km / h))
Heavy operating conditions						
less than 400	323	240	237	80	100	20,83 (75)
400 -700	548	600	298	44	82	25,00 (90)
700 -1000	828	184	271	41	57	25,00 (90)
more than 1000	1604	381	245	32	27	25,00 (90)
Normal operating conditions						
less then400	310	151	199	80	78	19,20 (69)
400 -700	598	711	284	36	75	25,00 (90)

Curve radius (m)	Accepted for calculation radius (m)	Length (km)	Circular curve length (m)	Length of transition first (m)	Elevation of the outer rail in a curve (mm)	Crew movement speed (m / s (km / h))
700 -1000	842	314	251	33	59	25,00 (90)
more than 1000	1913	784	282	27	35	25,00 (90)
Favorable operating conditions						
less than400	335	32	189	80	15	16,39 (75)
400 -700	580	403	331	45	77	25,00 (90)
700 -1000	838	167	295	42	57	25,00 (90)
more than 1000	1663	507	297	35	29	25,00 (90)

To select representative path sections based on the analysis of statistical data on the characteristics of the main paths in the plan and profile, they were used to rank them for each of the following factors affecting the wear rate and directly related to the path parameters:

– the proportion of curves of small radius (less than 400 m);

– share of curves of average radius (from 400 m to 700 m);

– the proportion of curves of large radius (more than 700 m);

– the average length of both transition curves;

– the total share of curves of small and medium radius with an elevation of less than 40 mm;

– share of slopes;

– the average steepness of the slopes.

Thus, for example, routes that combined with areas with a minimum radius of curves, maximum length and elevation of the outer rail were assigned to severe operating conditions.

The speed of the crew in the curves was calculated on the basis of the value of the allowable outstanding acceleration of 0.7 m / s^2, and was taken no more than that specified for freight cars in table 84 [8].

The simulation of the motion of a loaded gondola on bogies of the model 18-100 was performed in the Medyna software package using a mathematical model similar to that described in [9]. The calculation of wear in the model is based on the theory of abrasive wear (Archard's theory [10]). The mass of worn material is proportional to the work of the friction forces in the contact (A), and the phases of weak and strong wear differ, for each of which a proportionality coefficient (kv) is established:

$$I = k_v \cdot A \tag{1}$$

The transition from strong to weak wear is taken into account through the ratio of the power of friction forces in the contact patch to its area.

To calculate the wear of the wheels of a freight car, the following parameters were used:

– wear coefficient of 2.2 • 10-6 g / N • m for the stage of severe wear;

–- wear coefficient of 1.4 • 10-6 g / N • m for the stage of weak wear;

– the ratio of the power of friction forces in the contact patch of the wheel with the rail to its area, corresponding to the transition from weak to strong wear, 7 MW / m^2;

– coefficient of friction on the surface of the wheel 0.25;

– coefficient of friction on the wheel flange 0.28.

The parameters of the contact interaction of the car with the track were calculated in accordance with the conditions specified in Table 1 as it moves along the rails R65 with irregularities according to RD 32.68 [11]. To account for the passage of the car at the same time the left and right curves for the left and right wheels of each wheel set, an identity was set.

The wear of the wheel profile for the rolling surface was estimated at a distance of 70 mm from the edge of the rim, for the wheel flange - at a height of 18 mm from its top. Figure 1 shows an example of the first worn wheel profile of the first wheel pair in the course of movement of a 26 thousand km car under difficult conditions in curves with a radius of less than 400 m.

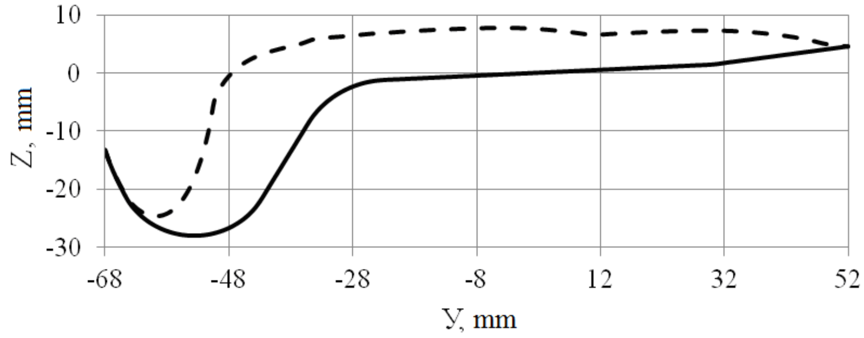

Figure 1. Worn wheel rim profile (dotted line) in comparison with the new one in accordance with GOST 10791 (solid line)

Comparison of the three operating conditions (heavy, normal and favorable) among themselves was made according to the reduced wheel wear rate (separately for ridges and rolling surfaces), taking into account the length of curves of different radii in the total length of each route:

$$P^{reduced} = \sum \frac{P_i \cdot L_i}{L}, \tag{2}$$

where P_i is the wear rate of the ridge / rolling surface of the wheel, mm / 10 thousand km, when the car moves in the curve of the i-th radius;
L_i is the length of the section of the path with the curve of the i-th radius, km;
L is the total length of the route, km.

4. Experimental results

Figures 2 and 3 present the results of calculations of the reduced rate of wear of the rolling surface and wheel flange, respectively, separately for each wheel set when the car moves in various conditions (in figures "k_p" - wheel pair, wheel pair numbers in the direction of the car).

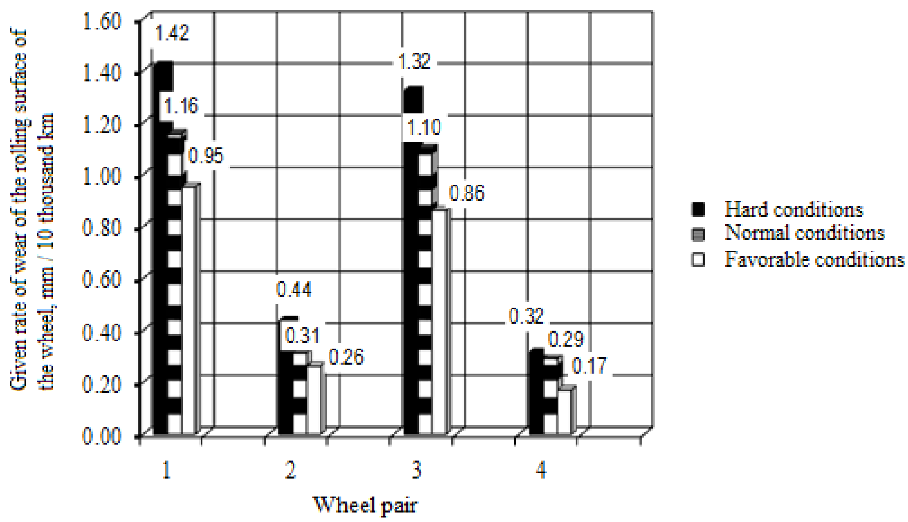

Figure 2. Diagram of the reduced rate of wear of the rolling surfaces of the wheels of the car depending on the operating conditions

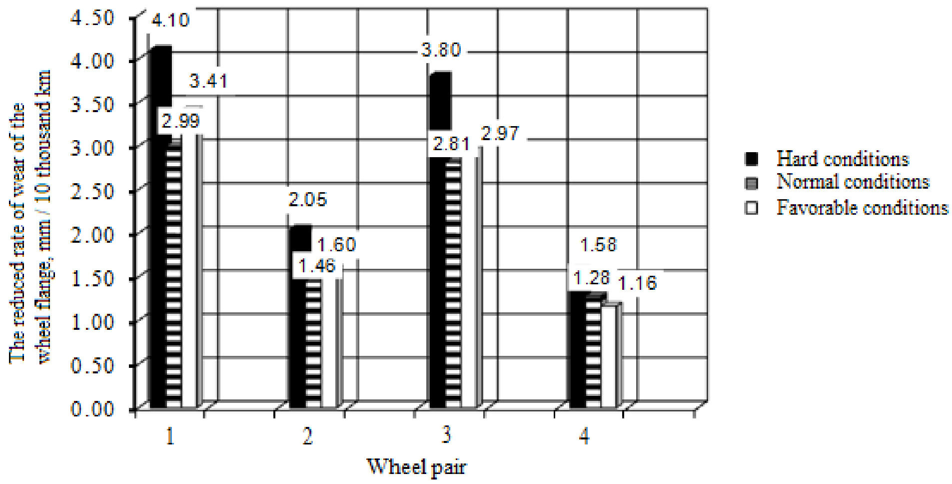

Figure 3. Diagram of the reduced rate of wear of the wheel flanges of the car depending on the operating conditions

In addition to the wear and wear rate of the freight car wheels for each wheel set, the areas of the wheel-rail contact spots and the friction forces in them when the car passed along curves of different radius were estimated (the car's mileage ranged from 25 km to 50 km for various calculated cases). Figures 4, 5 show the results of calculating the average area of the contact patch of the wheel with the rail (for the tread surface and the ridge, respectively) depending on the radius of the curve when the car goes under severe conditions (the largest areas of contact spots obtained from all the conditions considered). The maximum area of the contact patch corresponds to a small radius curve (323 m) and is 3.7 cm^2 and 0.76 cm^2 along the ridge along the tread surface, respectively.

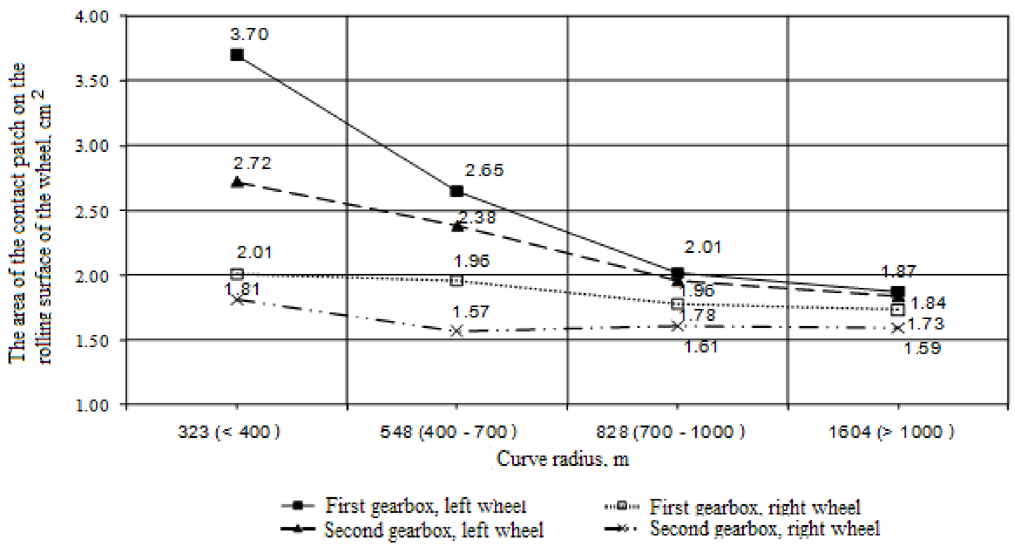

Figure 4. The area of the contact patch on the rolling surface of the wheel depending on the radius of the curve

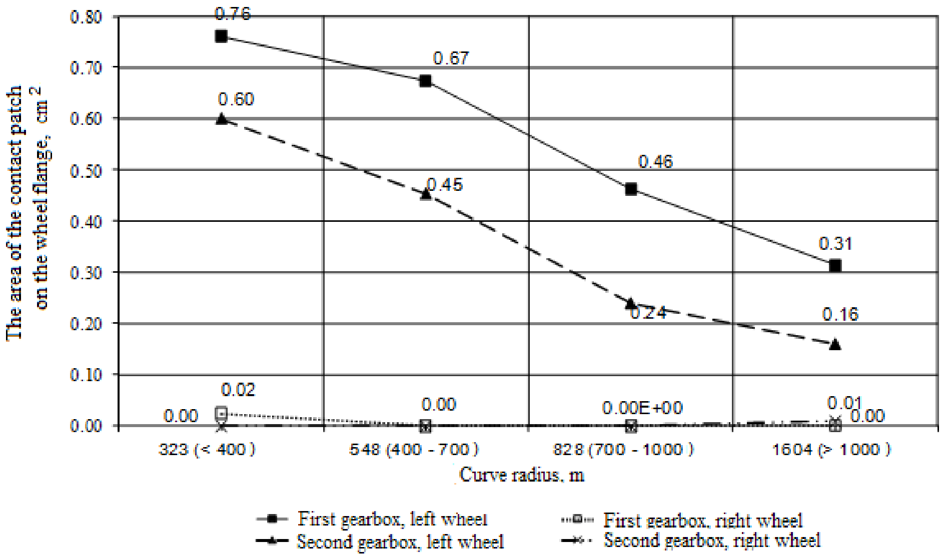

Figure 5. The area of the contact patch on the wheel flange, depending on the radius of the curve

Figures 6 and 7 show the results of the calculation of the path along the sections with different radii of average power of friction and pseudo-slip forces in the contact patch of a wheel with a rail (for a rolling surface and a crest, respectively) when the carriage follows in different conditions.

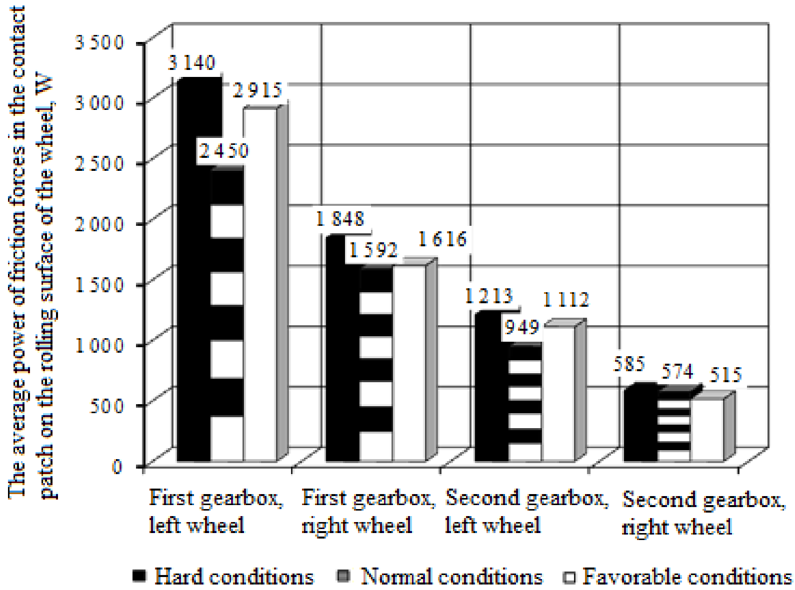

Figure 6. Dependence of the average power of friction forces in the contact patch on the rolling surface of the wheel on the operating conditions

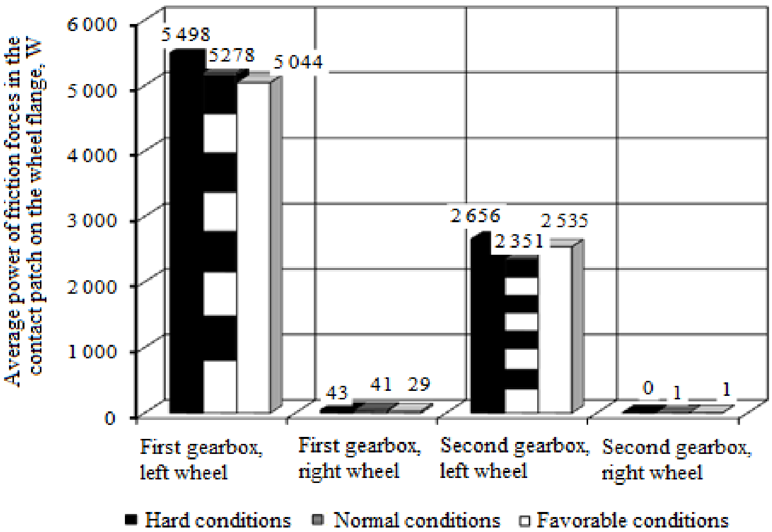

Figure 7. The dependence of the average power of the friction forces in the contact patch on the wheel flange on the operating conditions

5. The discussion of the results

Analysis of the results of the calculation of wear rates and parameters of contact spots showed:

<u>For loaded universal gondola</u>

The maximum wear of the ridge and the rolling surface was respectively 5.51 mm and 2.72 mm divided by 10 thousand km. for the oncoming left wheel of the first wheelset of the model 0110 with a curve radius of 301 m with a length of 151 km, with a valid distance of 25.4 thousand km.

The maximum reduced wear of the ridge and rolling surface for the rolling left wheel of the first wheel pair was 3.86 mm and 1.33 mm for severe operating conditions, for normal conditions – 2.81 mm and 1.09 mm and 3.2 mm and 0, 89 mm for favorable operating conditions, respectively.

The average contact area on the rolling surface and ridge is 2.49 cm^2 and 0.34 cm^2, respectively, for severe conditions, for normal conditions – 2.23 cm^2 and 0.18 cm^2, and for favorable operating conditions – 2.44 cm^2 and 0.36 cm^2.

The maximum average power values of the friction forces on the wheel surface are 3408.21 W for severe conditions, for normal – 2539.35 W and 2916.12 W for favorable operating conditions, respectively.

<u>For passenger compartment car</u>

The maximum wear of the ridge and the rolling surface was 6.53 mm and 1.37 mm, respectively, divided by 10 thousand km. for the oncoming left wheel of the first wheelset of the model 0110 with a curve radius of 301 m with a length of 151 km, with a valid distance of 16.18 thousand km.

The maximum reduced wear of the ridge and rolling surface for the rolling left wheel of the first wheel pair was 4.61 mm and 0.80 mm for severe operating conditions, for normal conditions – 3.65 mm and 0.68 mm, and 3.64 mm and 0,61 mm for favorable operating conditions, respectively.

The average contact area on the rolling surface and ridge is 1.81 cm^2 and 0.33 cm^2, respectively, for severe conditions, for normal conditions – 1.79 cm^2 and 0.31 cm^2, and for favorable operating conditions – 1.75 cm^2 and 0.32 cm^2.

The maximum average power values of friction forces on the wheel surface are 2858.45 W for severe conditions, for normal - 3633.23 W and 2723.30 W for favorable operating conditions, respectively.

The maximum average power values of the friction forces on the wheel flange are 3,694.51 W for severe conditions, for normal – 5,556.67 W and 3,023.76 W for favorable operating conditions, respectively.

6. Conclusions and conclusion

In this work, mathematical modeling was performed in the MEDYNA software package of a laden universal gondola motion mounted on 18-100 trolleys with an axial static load of 23.5 ts and a passenger compartment car mounted on KVZ trolleys – a Central Research Institute Type I with a wheel profile according to GOST 10791 paired with rails R65.

The simulation was carried out with the variation of such parameters as the speed of movement of the car and the length of the section of the track, the radius and length of the transition curves, as well as the elevation of the outer rail. A total of 12 freight car models and 12 passenger car models were considered.

The non-linear dependences of the creep forces on the relative slippage at the contact points, as well as the size and position of the wheel and rail contact spots, the pressure in the contact spots, the creep force, the power of the creep forces and wear are determined.

The highest wear rates of the ridge and the rolling surface were observed for the first and third (rolling) wheels.

The obtained data can be used in the selection of parameters of rollers on friction machines in order to simulate the contacts of the wheels of cars with P65 rails in various operating conditions.

References

[1] GOST 10791-2011 *Solid wheels. Technical conditions*
[2] Boronenko Yu P 1997 MEDYNA / Arge Care, Computer Aided Railway Engineering: User Guide (Saint Petersburg: National Exhibition Center "Wagons") 8 books.
[3] Sładkowski A. 2005 Rail vehicle dynamics and associated problems (Gliwice: Wydawnictwo Politechniki Śląskiej) p188
[4] Szabó A, Zobory I 1998 On Deterministic and Stochastic Simulation of Wheel and Rail Profile Wear Process. (Budapest: Technical University) pp 3–7
[5] Sładkowski A 2011 Finite element method for transport applications. (Gliwice: Politechnika Śląska) p 212
[6] Cherkashin Yu M, Pogorelov D Yu, Simonov V A 2010 Analysis of the influence of crew parameters and paths on indicators determining traffic safety (Moscow: Vestnik VNIIZhT) pp 3–9.
[7] Vorobev A.A., Konograi O.A. 2015 Comparison of the territorial branches of JSC “Russian Railways” on the conditions of operation of wheelsets of rolling stock (Bryansk: State Engineering and Technological Academy) pp 98–107
[8] Norms of permissible speeds of rolling stock on rail tracks of 1520 (1524) mm of federal railway transport (Order of the Ministry of Railways of the Russian Federation of November 12, 2001 No. 41) p 126
[9] Orlova A M, Vorobev A A, Saidova A V, Kerentsev D E 2015 Determination of the parameters of the wheel-rail contact for various operating conditions of a low-sided car (Saint Petersburg: Izvestiya PGUPS) pp 74-84.
[10] Archard J.F. 1957 Elastic deformation and the laws of friction (London: Proc. Royal Society. Publ.) pp 190-205.
[11] RD 32.68-96 Calculated unevenness of the railway track for use in research and design of passenger and freight cars