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# Analysis of wheel–rail interaction using FE software

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**Abstract**

In this article, research into the influence of interacting wheel and rail profiles on the distribution of contact zones and stresses is presented. The influence of contact forces on the deformation of rolling carload wheels and rails, and the influence of this deformation on the redistribution of the contact stresses is also investigated.

The quasi-Hertz method as well as a finite element (FE) method were included as basis of mathematical simulation. With these tools, distributions of contact zones for different angles of attack of the wheelsets were defined. The problem was solved in three-dimensions. The offered technique to solve the contact problem has been used for the improvement of operating wheels.


*Keywords:* Wheel–rail interaction; Finite element method; Quasi-Hertz method; New profiles

## 1. Introduction

One of the most important questions relating to the exploitation of cars and locomotives on the railways of the Central and Eastern Europe countries is the increased wear of the working surface of the wheels. In particular, this concerns their flange zones. In addition, wear also occurs on lateral surfaces of the rail heads. The amount of wear varies for different track locations and for different rolling stock. It involves both plastic deformations, and abrasion of surfaces. The cause of this phenomenon may be high dynamic forces at the wheel–rail contact that as caused by bad maintenance, both of a rolling stock, and track and absence of effective and appropriate use of lubrication.

However, the most important questions are related to the choice of wheel and rail profiles. It is quite obvious that these profiles should correspond to each other and that a change of one of them should not occur without change of the other. Current practice of railways in some countries says the opposite.

In this work, an attempt to analyze the wheel–rail contact interaction for various profiles of wheels and rails, including severely worn ones, is made.

## 2. Technique of the solution of the contact problem

The problem was analyzed using the FE method. This method has been successfully applied by various authors to research of contact interaction of wheels with rails, points and crossings. In the literature, the influence of roughness, defects and cracks have been studied [1–5,13]. For the solution of the specified problem, geometrical modeling was carried out in a CAD-environment. The geometry was imported to a FE-code. Discretization was carried out in a semi-automatic mode. It was not possible to completely automate the discretization since a FE mesh with a high number of degrees of freedom was created, and a solution with a satisfactory level of accuracy was very difficult to find with the FE-grid created by a mesh generator.

Fig. 1 shows an FE model of the wheel–rail contact interaction and the irregular grid of a P65 rail. Construction of the grid is executed as follows: the area of possible contact of a

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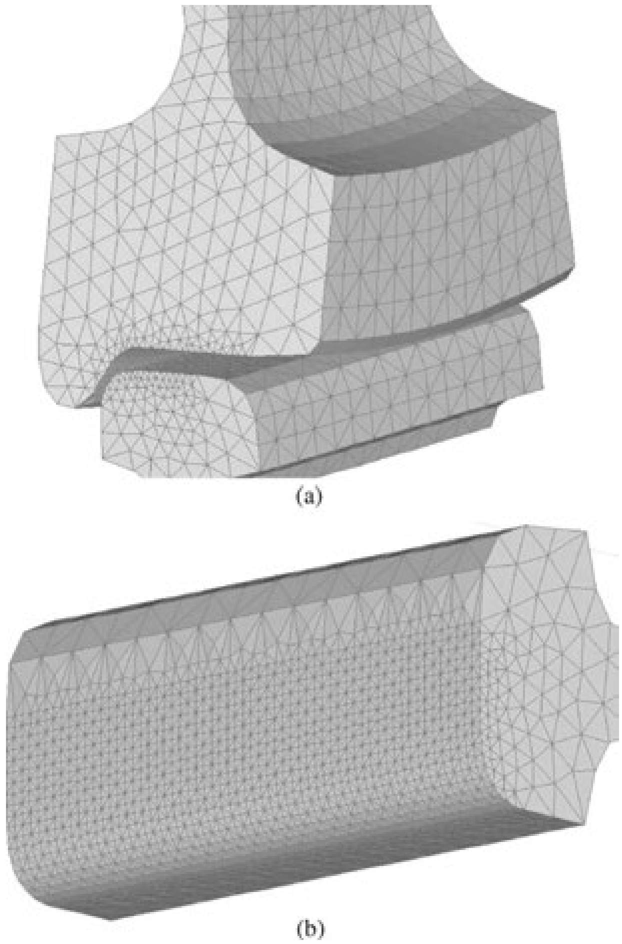


Fig. 1. FE model of contact between wheel and rail (a), irregular FE grid of rail head (b).

wheel and rail is allocated on the surface where generation of a surface grid is carried out; the specified grid was regular with a constant element size. In addition, the automatic generator in MSC.Visual NASTRAN for Windows created the remaining FE grid of the rail head. This grid became irregular.

The FE grid for a wheel was created in a similar way.

Unfortunately, current versions of MSC.Visual NASTRAN do not allow the solution of contact problems using arbitrary FE grids.

The package MSC.MARC gives some opportunities in this direction. Here, there are no special contact elements, groups of preliminary elements are simply united in contact bodies. However, the results may show major inaccuracies. To resolve this issue, the following method has been used. The data for the interacting wheel and rail profiles were modeled in AutoCAD allowing the modeling of both worn and new profiles. A restriction was that profiles needed to be described by pieces of straight lines or arcs. These restrictions are connected with the opportunities of exporting the geometry. FE grids of the wheel and rail were then prepared and boundary conditions and load were prescribed. The FE model was exported into the package MARC where contacting bodies were defined, and parameters such as required accuracy of the solution, used operative memory, etc. were set.

Unfortunately, the described technique did not provide the prescribed accuracy of the solution. Problems arose owing to the irregularity and inconsistency of the FE grids [6]. To illustrate this issue, the distribution of contact nodal normal forces (analogue of contact stresses) in the contact area are shown in Fig. 2. This distribution has been evaluated for a central arrangement of a wheel with a new profile GOST 9036–88 [7,8] in contact with a new rail P65 [9,10].

To eliminate similar errors, it is proposed to create regular FE grids for all zones near the wheel and rail contact [6]. The grids should take into account real surface profiles and are created for corresponding sections of wheel and rail.

To create matching, FE meshes an in-house code developed by the authors was employed for mesh generation in the flat section of the wheel and rail surfaces. For the rest of the wheel and rail automatic mesh generation was employed, which resulted in an irregular mesh. The FE meshes for the wheel and the rail were then adjusted to match.

The transformation of surface grids to spatial was made by the FEA software. It is then necessary that the three-dimensional FE grids are matching along the depth.

A separate question is the assignment of boundary conditions for the considered areas of a wheel and a rail. It is obvious that it is impossible to examine a full grid for a wheel and a rail. Such grid would have a very large number of degrees of freedom. Since the contact problem is nonlinear, it demands significant processing time. Taking into account that it is necessary to carry out calculations for various positions of the wheel and rail, as well as for different profiles of the interacting surfaces, it is expedient to consider the deformation of wheels and rails separately under the action of the whole set of loadings. Thus, FE grids can be sparse. These grids should be such that it is possible to allocate contact areas for which a regular discretization, as described above, can be carried out.

The problem has been considered in the following statement. The wheel is fixed at the nodes on a contour of a hole in the centre of the wheel. Node forces, which can be determined, for example, by dynamic modeling of the carriage using ADAMS are enclosed on a contact surface. Deformation of a wheel under the influence of the thermal loadings, caused by block braking, can also be considered. Deformation of the rail was considered in a similar way.

## 3. Discussion of the results received with the help of FEM

A three-dimensional FE model of the contact interaction has been developed using the technique described above. This model is shown in Fig. 3.

In Fig. 4, the results of the calculation of the contact nodal normal forces (analogue to contact stresses) for a new car wheel with a standard wheel profile GOST 9036–88 [7] (for the countries of the former USSR) in interaction with new rail

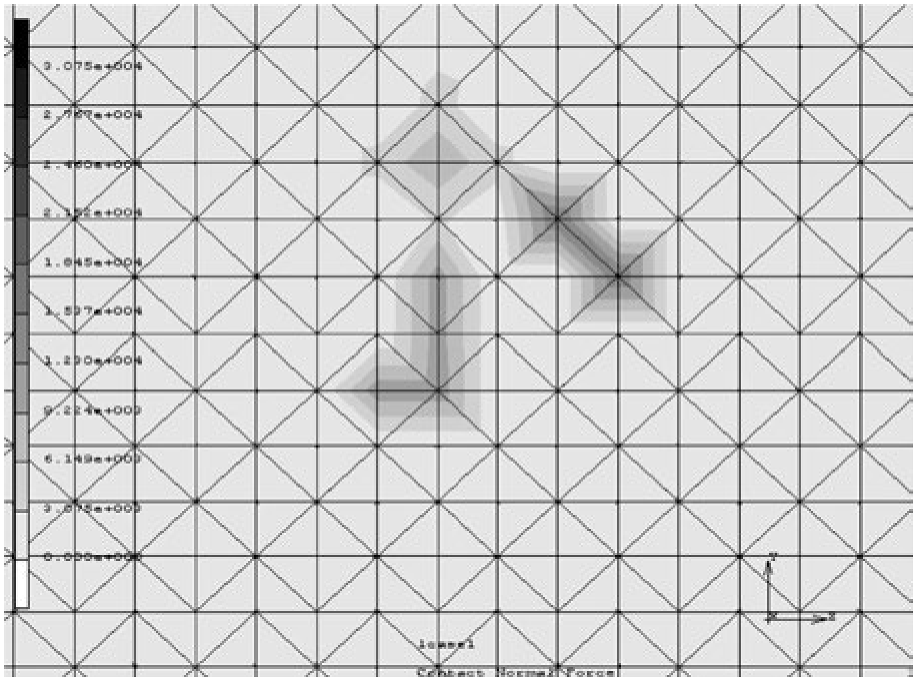


Fig. 2. Distribution of contact nodal forces (analogue to contact stresses) for irregular grids.

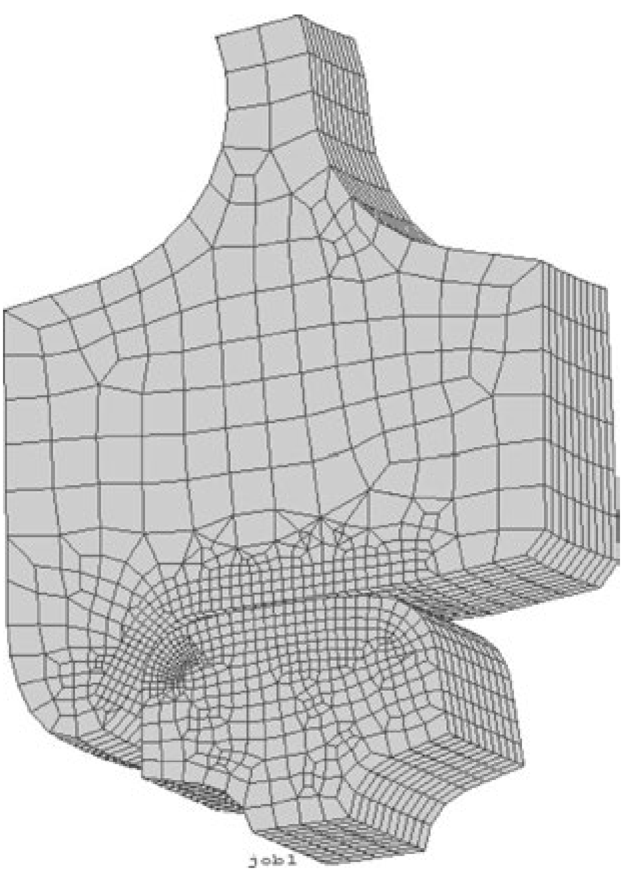


Fig. 3. Considered FE model of contact interaction of a wheel and a rail.

P65 [9] are shown. The mechanical properties of the wheel and rail were chosen according to standards [8,10]. At first, the wheel and the rail with sparse meshes were considered separately. Thus, various kinds of global loads were taken into account, for example thermal loading or wheel–axle press-fit [15,16]. As a result of these calculations, the displacements on the boundaries of the considered areas of the wheel and rail were defined. Then, the contact problem of interaction of the wheel and rail was analyzed using a dense FE mesh with the obtained boundary conditions. Next, the considered (near contact) areas of the wheel and rail were positioned. Vertical and lateral force were applied on the borders of these areas and the location and size of the contact zones was derived along with the distribution of stresses in an iterative manner. The vertical force imposed to the wheel was chosen as 125 kN (static load) or 200 kN (quasi-static). The size of the lateral force was chosen from 10 kN to 100 kN. A second load case was similar to the first with the difference that the mutual contact displacement between wheel and rail was set, and not the forces. To evaluate the total forces in this case, it is necessary to solve the problem iteratively for different relative displacements. Still the time of analysis was significant shorter for this case.

In order to show the contact zone, only the wheel is shown in Fig. 4. Simulations were carried out for various relative positions of the contacting bodies. As an example, Fig. 4 shows the wheel–rail contact in a tread zone. Here, one-point contact takes place. The shape of the contact patch is close to elliptic, i.e. the stress distribution is close to Hertzian. Nevertheless, there are differences as compared to a Hertzian distribution. This is caused by the deformation of the wheel disk.

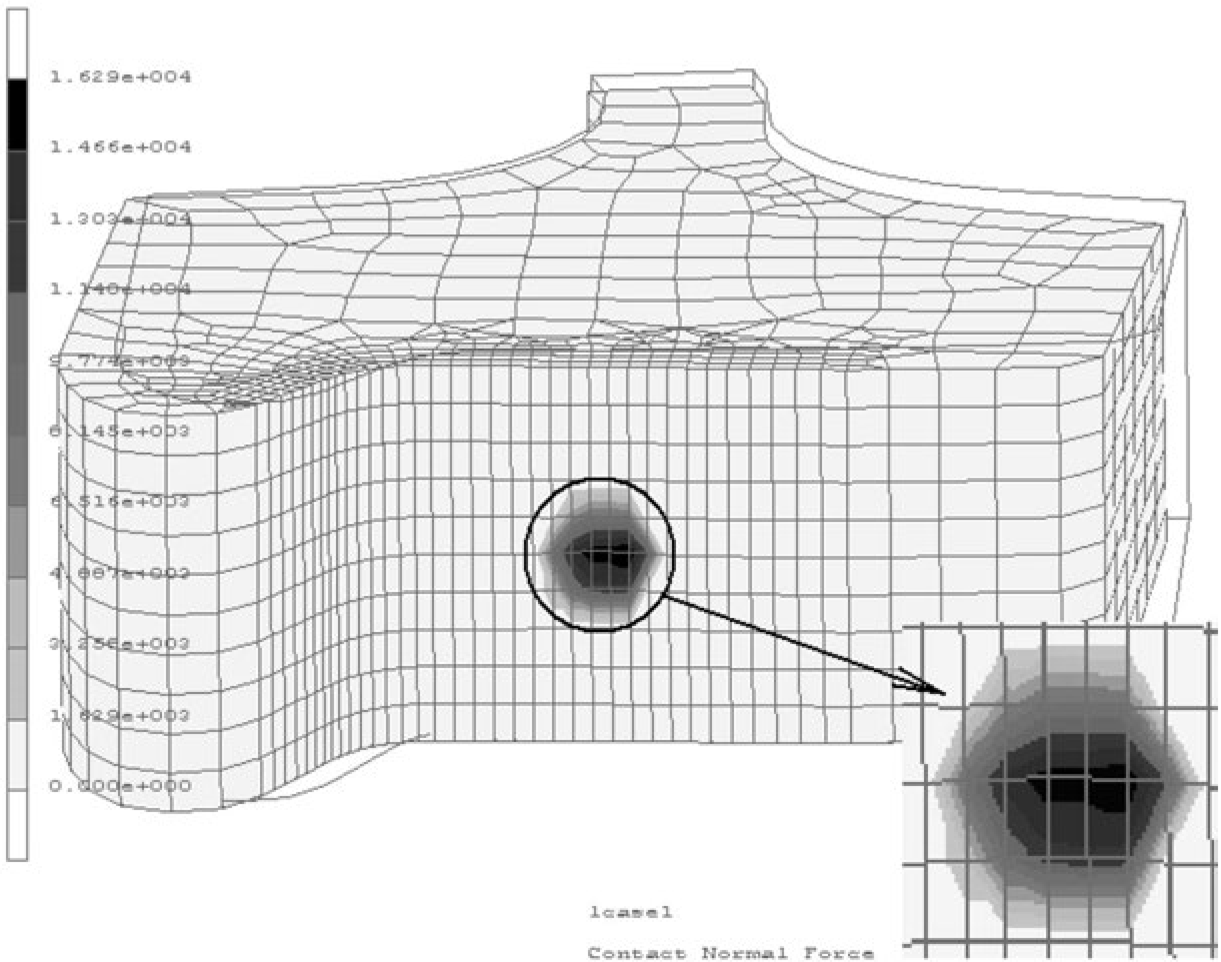


Fig. 4. Distribution of contact nodal normal forces in the case of one-point contact.

A comparison of a Hertzian calculation and FE-simulation is presented in the article [6] for test models where the influence of disk flexibility has been excluded. In Fig. 4, the undeformed arrangement of a wheel is also shown. As we can see in the magnified plot, the wheel rim becomes slightly warped under the influence of a normal force. Thus, the nodes in the left part move to the left and downwards, and nodes in the right part move to the left and upwards. This causes a redistribution of contact forces. This distribution differs from the parabolic one in that nodes located in the left part are more heavily loaded than nodes in the right part.

In Fig. 5, the distribution of contact nodal normal forces for two-point is shown. Such relative arrangement of a wheel and a rail takes place in the case of the displacement of a wheel to the lateral side of the track relative to its initial position. Here, the wheel flange comes in contact with the lateral surface of the rail head. Loading of the flange zone of the wheel results in significant lateral forces. There is also a redistribution of stresses between the contact zones. If the wheel is displaced in the lateral direction, the majority of the lateral force will affect the flange. Thus, the size and magnitude of contact stresses increases in the flange zone. As a consequence the size of the central zone decreases.

## 4. The validation of contact stresses

To validate evaluated contact stresses, a quasi-Hertz approach developed by the authors [11] has been employed. In this approach, contact deformations have been evaluated using Hertz–Beliaev theory, with the use of the local contact geometry at the centre of the contact zone as evaluated by the FE-simulation.

The problem arises when two-zone contact takes place. In this case, the distribution of forces between the two contact zones is not known. To analyze this distribution, compatibility condition of deformations sets and distribution of contact forces are identified by iterative way. The design procedure has been described in [12].

The use of FE-simulations in contact problems has advantages and drawbacks. Advantages include the possibility to analyze various relative positions of the wheel and rail at nonzero angles of attack in the presence of sev-

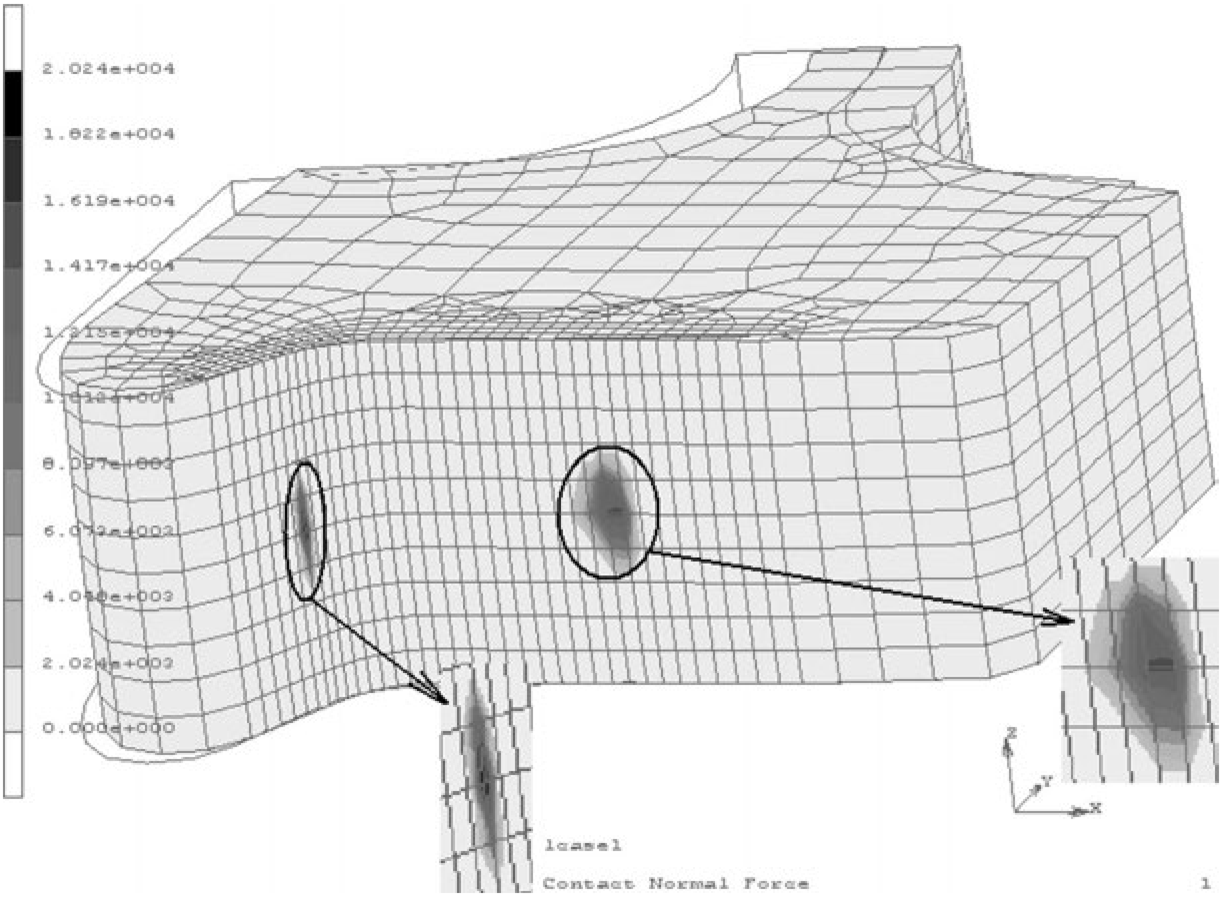


Fig. 5. Distribution of contact nodal normal forces in the case of two-point contact.

eral contact zones and in the presence of plastic deformations. However, FE-simulations also have essential drawbacks. To increase the accuracy there is a need for a large number of finite elements. This results in a substantial growth of the number of degrees of freedom of the model. In view of the non-linearity of the problem, it can lead to non-physical solutions. The complexity of generating a FE grid and to assign boundary conditions is also a drawback.

In Figs. 6 and 7, a comparison of the distribution of the contact zones in the case of two-point contact for various profiles is shown. The first figure (Fig. 6) is used to show the case of contact of new standard wheels and rails. The two contact zones are separated by a significant distance. They also have a significant difference in local radii. Consequently, there are significant relative slip in the flange zone of the wheel and rail. On the other hand, the maximum contact pressure in the flange zone reaches 3600 MPa, which considerably exceeds the yield point (the problem is solved elastically). Thus, plastic deformations and wear of the surface occurs. This phenomenon is well known in practice for rolling stock on railways in the countries of the former USSR. The comparison of calculations as made by a FE-simulations and the quasi-Hertz approach show a good conformity.

## 5. Introduction of an improved profile

In Fig. 7, contact zones for a new wheel design [14] is shown. The new profile has been developed using the method described in the current paper. The developed profile reduces the level of contact pressure by 50% in the flange zone. The two contact zones are closely located. This approach of the contact zones decreases the wear in that the relative slip between wheel and rail decreases due to the smaller difference in local radii.

The intensity of wear, primarily, in the flange zone, has decreased with the new wheels. The new profiles are now effectively used for locomotives in Ukraine, Russia and other countries. According to various depots, the wear intensity of these wheels has decreased with between 20% and 50%. (Wear intensity of a flange is defined as the ratio of change in flange per tens of thousands of kilometers.) Similar results have been achieved using the new profile for a railway tank car. However, the greatest efficiency is found when using the new profile in industrial railway transport. This is caused by the fact that in conditions of ore mining and processing enterprises wear has an essentially abrasive character. The main reason are curves of small radii where there is constantly a flange contact resulting in relative slip. The presence of a mining dust, sand and

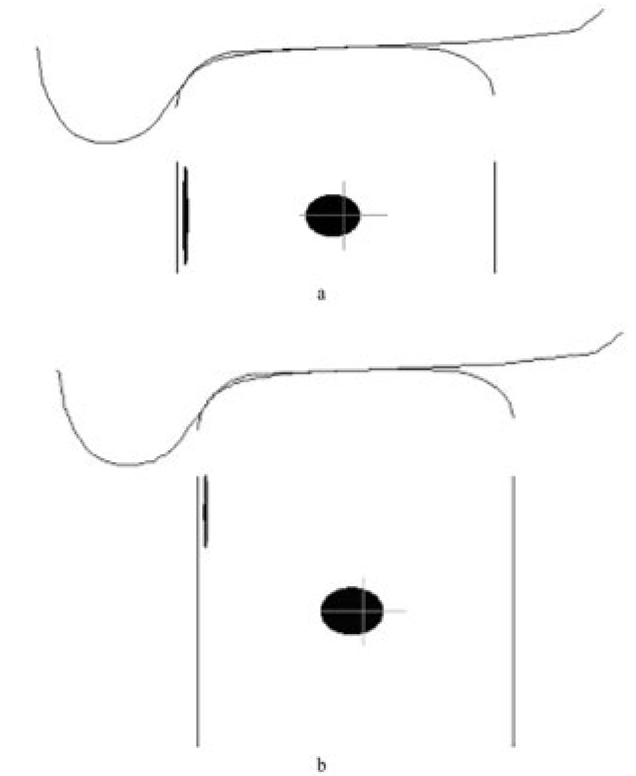


Fig. 6. Distribution of contact zones at the contact between standard profiles of new wheels and rails: (a) angle of attack is 0°; (b) angle of attack is 2°.

crushed stones also result in additional wear. Further, high-axial load and poor maintenance also result in an increase of wear that does not exclude plastic deformation. A reduction of sliding in the flange zone reduces the intensity of such wear.

Today, two profiles are used for cars and locomotives. They are shown in comparison with a standard profile in Fig. 8. For the first type, the flange thickness is the same as for a standard profile, i.e. 33 mm. The second type has a reduced flange thickness (30 mm). This profile is recommended for regenerative repair in depots. To restore a wheel to this profile, the requirements of metal removal are almost halved.

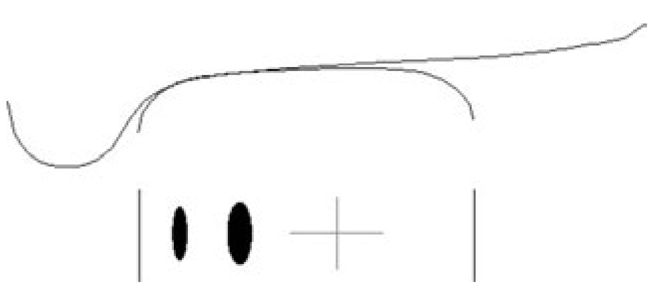

Fig. 7. Distribution of contact zones at the contact between standard rail and new wheel profiles.

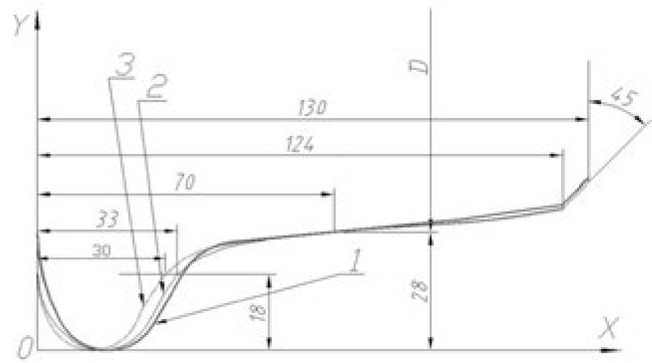


Fig. 8. Comparison of profiles of car wheels: 1—standard profile; 2—new profile with flange thickness of 33 mm; 3—new profile with flange thickness of 30 mm (repair).

## 6. Conclusions

As a result of the research, distributions of contact zones and stresses for various wheel and rail profiles have been determined. These investigations are used as the basis for the development of new wheel and rail profiles from which recommendations on improvement of profiles may be given. In particular, new designs of the rail heads are developed, rolled and are now tested on railways in Ukraine. The first results have shown that the stability of work of such rails in particular sites of a route has increased.

Directions of the further research is proposed to consider dynamic processes in a wheel–rail pair, as parts of the track–carriage system. Thus, it is planned to use dynamic dependencies for the contact forces and the relative arrangements of the wheel–rail pair as defined by means of the program ADAMS. The first steps in this direction have already been taken by the authors. It is considered that such an approach is necessary to design wheels and rails for a high-speed rolling stock.

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