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Vladimir KOLESNIKOV
Rostov State Transport University
Rostov Strelkovogo Polka Narodnogo Opoltchenia sq.
Rostov on Don, 344038, Russia
Michail CHEBAKOV*, Andrey NASEDKIN, Dmitri SUKHOV
Southern Federal University
Miltchakova str., 8a, Rostov on Don, 344090, Russia
**Corresponding author*. E-mail: chebakov@math.rsu.ru

FINITE ELEMENT MODELLING OF CONTACT INTERACTION BETWEEN WHEEL AND RAIL WITH THREE-LAYERED THIN COATING

Summary. The stress-strain of thin multilayered coatings covered the surface of the rail is studied near the region of side contact in turning motion. The stress-strain state is studied for various geometric and mechanical coating parameters' values. A theoretical and a finite element models are developed. Stress-strain state analysis has been made based on models developed and recommendations were given for an optimal coating parameters' selection.

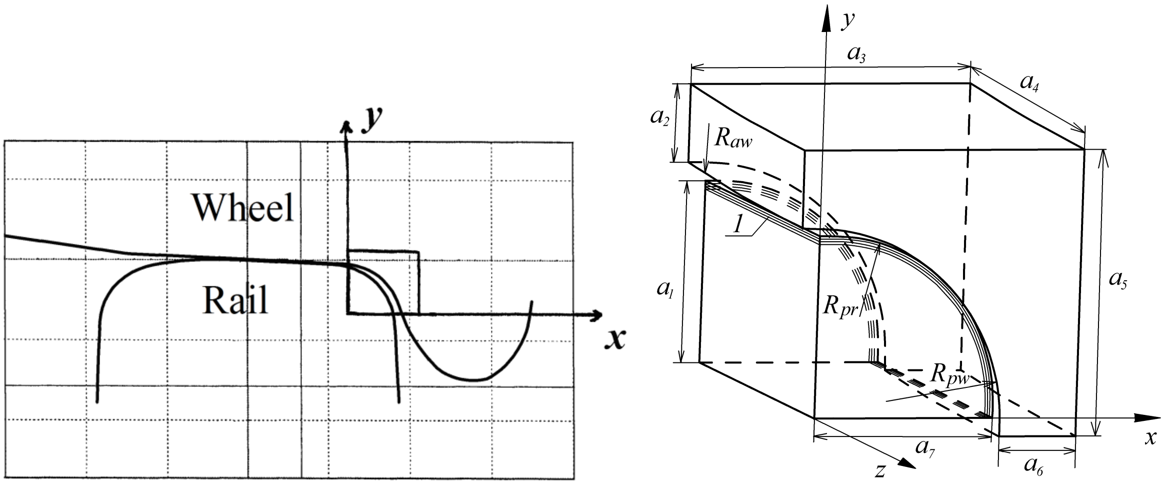
КОНЕЧНО-ЭЛЕМЕНТНОЕ МОДЕЛИРОВАНИЕ КОНТАКТНОГО ВЗАИМОДЕЙСТВИЯ КОЛЕСА И РЕЛЬСА С ТОНКИМ ТРЕХСЛОЙНЫМ ПОКРЫТИЕМ

Аннотация. Исследуется напряженно-деформированное состояние тонких многослойных покрытий, нанесенных на поверхность железнодорожного рельса в окрестности бокового контакта рельса и колеса, при движении в поворотах при различных значениях геометрических и механических параметров покрытий. Разработаны математическая и конечно-элементная модели, на основе которых проведен анализ напряженно-деформированного состояния и даны соответствующие рекомендации по оптимальному подбору параметров покрытий.

1. PROBLEM STATEMENT

In this paper the interaction between wheel and rail in turning motion is studied. In this case an additional contact region is appearing at the side surface of the rail apart from contact region at the surface of the roll. For the aim of side contact friction reduction the rail was covered by thin antifriction coating of complex structure which could be determined as three-layered coating with each layer differ by mechanical properties and thickness. For evaluation of the stress-strain state in the side contact region, including the thin coating, the investigation of corresponding contact problem is necessary.

Taking into account complicated geometry of rail and wheel system and for simplification of the problem we will use Saint-Venant principle [1] and investigate fragments of wheel and coated rail participating in side contact. This must be done taking into account real curvatures of corresponding surfaces. In Fig. 1a the rectangular region shows regions of wheel and rail that belongs to plane contained the point of initial contact and is located orthogonal to generatrix of the rail, for which a contact interaction problem is raised and a finite element model is build. Assume that z axis directed along the rail and Oxy plane contains the point of initial contact and is located orthogonal to rail generator. In this case the Oxy plane is a plane of symmetry. In Fig. 1b fragments of wheel and rail at $z \le 0$ are presented. Cut volumes are getting into contact after loading while forward approaching each other. With assumption that friction is absent in contact region and load is laid in Oxy plane we are getting a contact problem for bodies that are symmetric with respect to Oxy plane. Taking this into account it is possible to study the problem for regions which lay at one side of Oxy plane with symmetry boundary condition on Oxy plane and half load applied. It is assumed that three-layered thin coating is fixed on the cylindrical surface of the rail and each layer has different thickness and mechanical properties. Layers are assumed to be rigidly bound among themselves. The symbols for basic geometric dimensions of cut fragments of wheel and rail are presented in Fig. 1b.



b

a

Fig. 1. Problem geometry: a) Wheel and rail interaction scheme; b) Rail and wheel interaction model during side contact

Рис. 1. Геометрия задачи: a) Схема взаимодействия колеса и рельса; b) Модель взаимодействия колеса и рельса во время гребневого контакта

Note that dimensions a_i $(1 \le i \le 7)$ signed in Fig. 1b were chosen from the preliminary numeric experiments using a "negligible influence of borders to computation results in contact zone" criterion.

Let's introduce some notations: R_{pw}, R_{pr} are the radiuses of curvatures of wheel and rail surfaces, respectively, in contact region on Oxy plane; R_{aw} is the radius of wheel along the roll surface; h_{si} is the layer thickness of three-layered coating ($i = 1$ is the surface layer, $i = 2$ is the middle layer, $i = 3$ is the layer connected with rail body); E_r, ν_r are Young modulus and Poisson ratio of the rail, respectively; E_w, ν_w are Young modulus and Poisson ratio of the wheel, respectively; E_{si}, ν_{si} are Young modulus and Poisson ratio of the rail coating layer with number i, respectively.

We use a finite element method (FEM) for solution of this described problem. For this rail - wheel interaction model we accept the following boundary conditions:

– on the Oxy plane of rail and wheel fragments a symmetry condition is specified, i.e. normal displacement on the direction of z axis and tangential stresses are absent;

– edges of rail fragments for x=0 and y=0 are fixed and edges of wheel and rail with $z = -L$ $(L = a_4)$ are free from stresses;

– flat edges of wheel that parallel to $x = 0$ and $y = 0$ planes after interaction keep flat and have only translation movements; at the same time only shear stresses are allowed and friction is absent for these facets;
– the force laying in the Oxy plane with magnitude *P*=P/2* is applied to the wheel fragment in the point of intersection with two other flat edges; its direction is parallel to common normal of contact surfaces of wheel and rail that belongs to the point of initial contact.

2. FINITE ELEMENT ANALYSIS

The described problem was solved using ANSYS CAE package of version 11.0 [2 – 4]. The program system was developed using ADPL ANSYS language allows to solve considered problems with using parametric input data. With that, it is possible to conduct research of contact interaction of wheel and rail with different input values of geometric and physical parameters using the same programs.

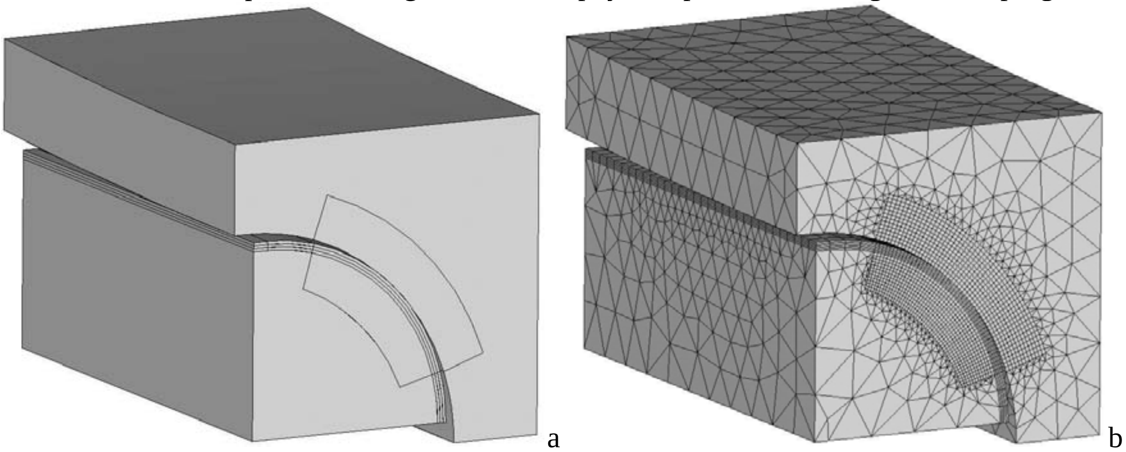


Fig. 2. Models of rail-wheel interaction: a) Solid model of contact problem; b) Finite element model of contact problem

Рис. 2. Модель взаимодействия колесо - рельс: a) Твердотельная контактная задача; b) Конечно-элементная модель контактной задачи

The solid geometric model of described problem is shown in Fig. 2a, and finite element model for one particular variant of finite element meshing is shown in Fig. 2b. According to methodology of FEM solution of contact problem having significantly non-homogeneous properties [5,6], the meshing thickness is increased in supposed contact region. For that purpose the geometric subregions were additionally defined; for them a regular less-sized finite element mesh was constructed. In regions occupied by elastic materials of wheel and rail the SOLID95 twenty-nodal 3D finite elements of siredipity type with quadratic approximations for each local coordinate were used. Paired quadratic elements TARGE170 and CONTA174 were used fs contact pairs [3].

3. ANALYSIS OF MULTILAYERED COATING’S STRESS-STRAIN STATE IN CONTACT REGION

Using finite element models a calculation of normal stresses distribution in contact region, contact region parameters and effective stresses were made for inner points of interacting bodies. For effective stresses, following formula was used:

$$\sigma_e = \frac{1}{\sqrt{2}}\sqrt{\left(\sigma_y - \sigma_x\right)^2 + \left(\sigma_y - \sigma_z\right)^2 + \left(\sigma_x - \sigma_z\right)^2 + 6(\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2)}\ .$$

During the analysis process following input data were taken: applied force P =70 КН, wheel radius R_{aw} =625 mm, rail surface curvature radius at the point of initial contact R_{pr} =13 mm, wheel surface curvature radius in Oxy plane on the point of initial contact R_{pw} =13.5 mm, Young modulus and Poisson ratio of rail and wheel $E_r = E_w$ =2.1·10^5 МПа, $\nu_r = \nu_w$ =0.3.

Three series of calculations were conducted depending on mechanical properties of the outer coating layer. In the first series of calculations (A) it was assumed: E_{s1} = 0.005·10^5 MPa, ν_{s1} =0.47; in the second series (B) – E_{s1} = 0.025·10^5 MPa, ν_{s1} =0.35; and in the third series (C) – E_{s1} =0.27·10^5 MPa, ν_{s1} =0.4. Simultaneously it was assumed that the outer layer's thickness h_{s1} =0.015 mm; total thickness of middle and inner coating layers 0.2 mm $\le h \le$ 0.35 mm ($h = h_{s2} + h_{s3}$) taking into account that $h_{s3} = h/6$; Young modulus and Poisson ratio of the middle coating layer E_{s2} =0.9 E_r, ν_{s2} =0.3; Young modulus of the inner coating layer $E_r \le E_{s3} \le 3E_r$ and its Poisson ratio ν_{s3} =0.3. We introduce also a notation $E = E_{s3} / E_r$.

The analysis was made by supercomputer system consisting of four T-Platforms Edge-8 computational clusters with top performance at 300 Gigaflops each (total peak performance of the whole system is 1.2 Teraflops). The ANSYS Multiphysics v. 11.0 licensed software with no restrictions of the number of degrees of freedom and with possibility to parallel computations and commercial use was applied.

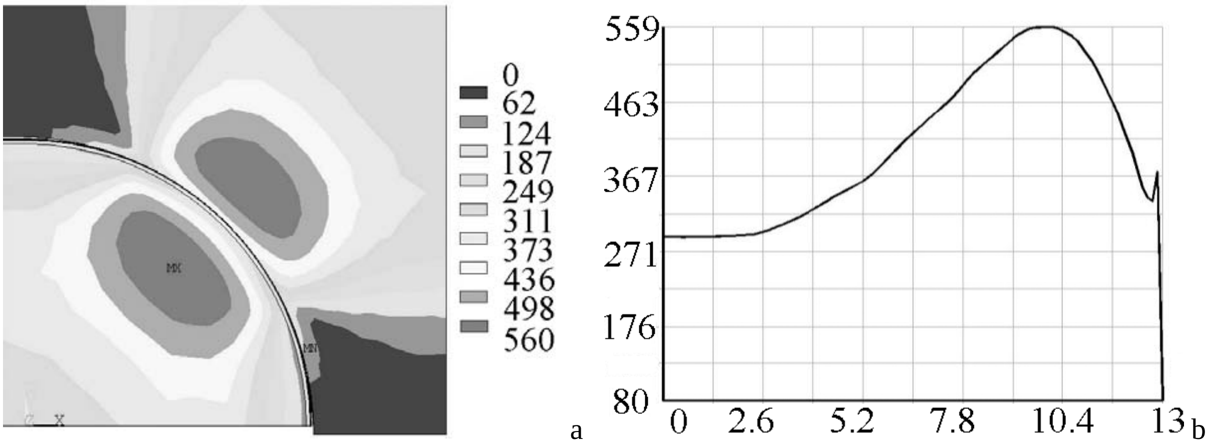


Fig. 3. Equivalent stress distribution (h =0.2, E =1.5): a) Equivalent stress distribution in *Oxy* plane; b) Equivalent stress distribution along normal

Fig. 3. Распределение эквивалентных напряжений (h =0.2, E =1.5): a) Распределение эквивалентных напряжений в плоскости *Oxy*; b) Распределение эквивалентных напряжений вдоль нормали

As an example a typical pictures of effective stresses in the Oxy plane are presented in Figs. 3a, 4a for series A with h =0.2 mm, E =1.5 and h =0.2 mm, E =3 ($E = E_{s3} / E_r$).

The typical diagrams of effective stresses distribution in Oxy plane of the rail on the normal to rail's cylindrical surface that goes through the initial contact point are presented in Figs. 3b, 4b with the same input values.

The effective stresses' distribution in Oxy plane of the rail fragment and in border surface between lower and middle coating layers are presented in Fig. 5a for the case h=0.35, E=3 for series A of numeric experiments. In this case, maximum effective stresses are located on the border between lower and middle layers at the some distance from Oxy plane.

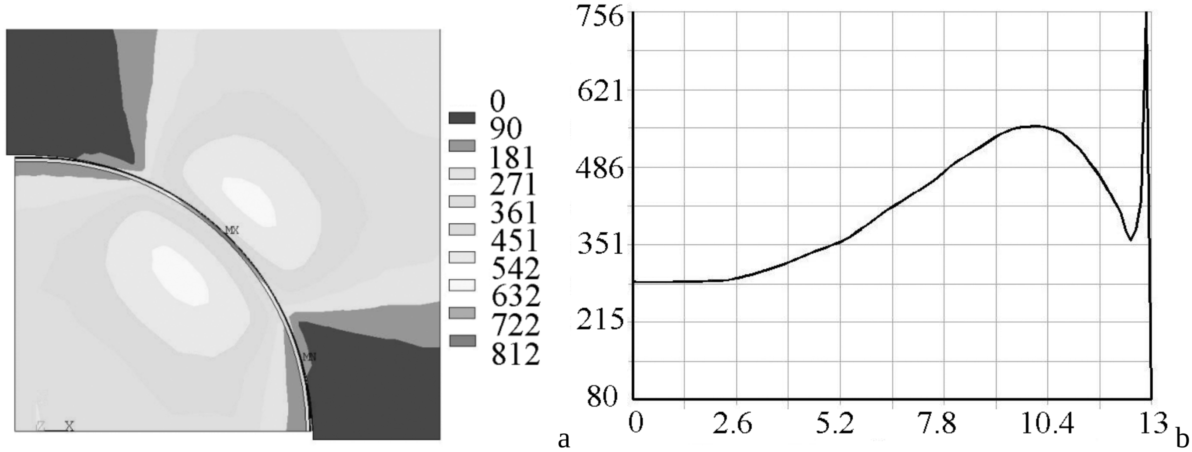


Fig. 4. Equivalent stress distribution (h =0.2, E =3): a) Equivalent stress distribution in Oxy plane; b) Equivalent stress distribution along normal

Fig. 4. Распределение эквивалентных напряжений (h =0.2, E =3): a) Распределение эквивалентных напряжений в плоскости *Oxy*; b) Распределение эквивалентных напряжений вдоль нормали

The contact stresses' distribution for particular parameters' values from series A is shown in Fig. 5b. The analysis have also shown that the behaviour of contact stresses' distribution doesn't change appreciably both in magnitude and structure for all numerical experiments of all series studied. From Fig. 5b one can also learn about shape and size of contact region, taking into account the height of rectangles that includes the contact region and shown at this figure, which is equal to 16 mm.

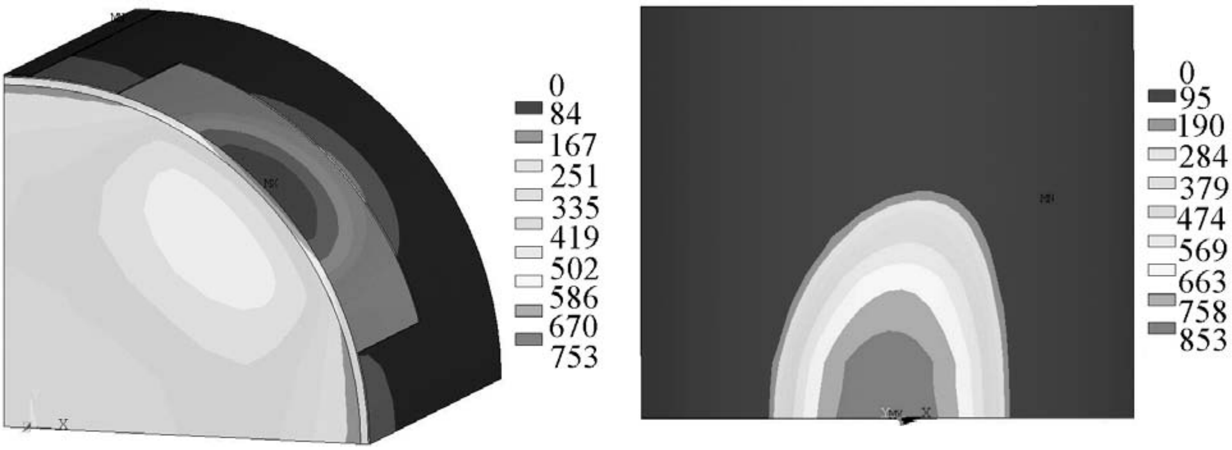


Fig. 5. Equivalent and contact stress distribution (A series, h =0.2, E =3): a) Equivalent stress distribution between lower and middle layers of coating; b) Contact stress distribution

Fig. 5. Распределение контактных напряжений (A series, h =0.2, E =3): a) Распределение напряжений между нижним и средним слоями покрытия; b) Распределение контактных напряжений

In the following table the results of effective and contact stresses' computations for some parameters' values of three series of analysis are presented. Following notation is used: $\sigma_e^{\max}$ is the maximal effective stress in the rail body; σ_e^n is the maximal effective stress in the rail body on the normal that belongs to Oxy plane and includes the initial contact point of rail and wheel; $\sigma_c^{\max}$ is the maximal contact stress.

Note that according to the results of analysis increasing of lower layer's stiffness leads to moving of the zone of maximal effective stresses towards the rail surface in the region of thin coating. At the same time, lowering inner and middle layers' thickness leads to higher effective stresses' values in the thin coating region, and maximal stresses will not belong to the Oxy plane but will lie in the same distance from it. The table 1 also demonstrates that contact stresses have relatively the same magnitudes independently to parameters of coating studied and they increase only slightly with outer layer's stiffness grow.

Table 1

Equivalent and contact stress values (MPa)

Series	E	1.0	1.0	1.5	1.5	2.0	2.0	3.0	3.0
	h	0.2	0.35	0.2	0.35	0.2	0.35	0.2	0.35
A	$\sigma_e^{\max}$	561	560	560	561	559	561	812	753
	σ_e^n	559	559	559	559	588	557	756	588
	$\sigma_c^{\max}$	846	846	849	850	850	852	852	856
B	$\sigma_e^{\max}$	558	557	557	557	556	558	812	753
	σ_e^n	556	556	554	556	554	553	756	589
	$\sigma_c^{\max}$	841	841	844	846	845	848	848	851
C	$\sigma_e^{\max}$	582	583	582	583	582	583	834	772
	σ_e^n	579	579	580	579	578	579	775	601
	$\sigma_c^{\max}$	878	878	881	884	884	887	887	891

From the obtained results the following conclusion can be formulated

1) Using the theory of strength and presented results, the stiffness of inner coating layer bonded with main rail body must be chosen as close to the stiffness of main rail body material as possible.

2) Thin three-layered coating with studied parameters do negligible influence to maximal contact stresses' values in rail and wheel side contact but the character of stress distribution in the inner points of coating depends on its parameters significantly.

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