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LIFE ASSESSMENT OF RAILS BY CRITERION OF OCCURRENCE OF FATIGUE CRACKS

The paper presents modern concepts of the contact fatigue crack occurrence in the railhead. Numerical simulation of the contact interaction between rails and rolling stock wheels by finite elements method are presented. During the simulation, the problem was solved in elastic and elastic-plastic formulations. We considered R65 rail profile and standard railway wheels. The conditions for the rail–wheel interaction corresponded to train movement on the straight section of the track without slipping. The modern life assessment methodology involves the calculation of the material damage increment at each point of the element as the load varies over time, and subsequent summation of this damage. Upon reaching the ultimate value of the total damage, the structural element is believed to lose its load-carrying capacity, i.e. a crack is formed in it. Despite the substantial simplification of the real problem, the computational costs for the implementation of such methods for predicting the durability will be unnecessarily high. In this regard, we propose the simplified method of the durability calculation.

Keywords: *railway transport, deformation and fracture, strength of railway structures, failure analysis, contact stresses, contact-fatigue damage, numerical simulation, finite elements method*.

1. Introduction

The current stage of development of railway transport is characterized by a tightening of operating parameters (increase in the power, speeds, and load-carrying capacity of locomotives), which will inevitably lead to an increase in force and thermal effects on structural elements of the rolling stock and rail tracks. Extreme weather conditions lead to a greater number of failures of the main element of the track – the rail, the reliability of which affects not only traffic safety, but also the economic performance of railways. Contact-fatigue damage and wear of the railhead are among the main causes of rail failures *[1 – 3]*. Such damage leads to the formation of surface and internal cracks in the railhead, the formation mechanisms and kinetics of which differ from each other. The development of surface cracks, followed by peeling of particulate material, is one of the main mechanisms of the railhead wear upon contact with the wheels of rolling stock. As regards internal cracks, the mechanism of their formation and propagation depends entirely on the nature of changes in the stress-strain state of the railhead during operation, as well as the purity of the rail material, and the availability of technical defects. Such cracks (longitudinal, oblique, lateral ones) can emerge at a depth of 4-16 mm from the running surface. Most often this occurs in the areas of stress concentration caused by the presence of various types of inclusions or technical defects. It is now established that the main cause of the internal cracks is the accumulation of plastic strains under the action of cyclic stresses. Such cracks are of particular concern, as their further propagation and penetration into the railhead can lead to a complete destruction of the rail.

Considering that the presence of internal cracks at the early stages of their propagation can hardly be diagnosed with the applied diagnostics equipment *[4]*, the development of the methods to predict the durability of rails by the criterion of the occurrence of internal cracks is of great scientific and practical interest. Over the last 20 years, this problem has been an active area of research *[5 – 12]*.

In this paper, the basic elements of the method for predicting the durability of rails by the criterion of the occurrence of internal contact fatigue cracks based on the use of the kinetic theory of damage and the structural model of the elastic-plastic deformation of materials are considered. The proposed life assessment methodology involves the calculation of the material damage increment at each point of the element as the load varies over time, and subsequent summation of this damage. Upon reaching the ultimate value of the total damage, the structural element is believed to lose its load-carrying capacity, i.e. a crack is formed in it. Its further behaviour under load should be considered in the framework of the mechanics of cracks. To evaluate the durability of rails, this approach was first used in *[5, 6]*. Despite a number of obvious simplifications (in particular, the authors considered only cyclic loading of the contact area using distributed pressure calculated by the elastic theory of Hertz), the authors were able to calculate the kinetics of the stress-strain state of the rail for only eight load cycles. A forecast of durability was carried out by extrapolating the damage accumulation rate obtained in the last two load cycles. Given that in reality the number of load cycles until the occurrence of a crack can be from a few tens to hundreds of thousands, the accuracy of the forecast is, of course, low.

In view of the above, it becomes obvious that despite the substantial simplification of the real problem, the computational costs for the implementation of such methods for predicting the durability will be unnecessarily high. In this regard, we propose the following simplified method of the durability calculation.

2. Research Technique

The initial data for constructing the geometric model were rail R65 and a locomotive wheel with the tread diameter of 957 mm. A straight section of the track (6 m in length), along which a two-axel bogie CNII-H3-0 with the centre distance of 1850 mm moves at a slow speed (dynamic effects are not taken into account), was considered. The rails are laid with a canting of 1:20 on concrete sleepers, the laying pattern is 1680 pcs/km (a distance between sleepers is 60 cm), and the mount type is KPP-5. The axial load is 200 kN. The presented characteristics of the rail base stiffness in the longitudinal ($9{,}7\cdot10^7$ N/m) and vertical ($3{,}75\cdot10^7$ N/m) directions were taken from *[13]*.

With regard to the characteristics of the mechanical properties of the material, the existing standards have no data on the elastic-plastic properties of the rail and wheel steels even in simple tension, especially during cyclic deformation. Therefore, all the calculations were made using the mechanical properties of carbon steel of pearlite class BS 11 *[14]*, which is close in composition to the steel used in rails P65.

3. Numerical Results

Based on calculation of the stress kinetics within one loading block, which corresponds to the passage of one bogie through the reference section, the position of the dangerous point is located within the reference section of the rail. The dangerous point is the point on the rail cross-section, where the damage accumulated over one loading block is maximal. To calculate the damage, the combined criteria can be used *[6]*, which have worked well in the assessment of durability of structural elements operating in contact. The position of the dangerous point in the rail cross-section indicates the place of nucleation of the future crack. The history of variation of all components of the active strain tensor during the first loading block is recorded for the dangerous point. Assuming that the plastic strain increment is low, which is true for those areas of the railhead where internal cracks are nucleated, the block cyclic loading is later modelled only for the dangerous point, using the appropriate model of elastic-plastic deformation and the established law of changes in the components of the active strain tensor. It is also believed that the rolling stock is composed of identical elements.

In this case, it is expected that the position of the dangerous point will not change. The structural model of cyclically unstable materials developed by the authors, which allows taking into account the effect of deformation trajectory patterns on the regularities in cyclic hardening, was used as a model of deformation. A detailed description of the model, relevant mathematical formulations and the method for determining parameters from the results of basic experiments are given in *[15, 16]*. Based on these calculations, we can determine the kinetics of damage accumulation in the dangerous point until the start point of destruction. This approach will enable an acceptable solution in terms of accuracy without using any expensive software and high-performance computing systems.

Let us consider an example of using this technique on a straight section of the track. Railway track is a complex spatial structure, which includes not only rails, but also sleepers, pads, rail fasteners and the ballast. All these components of the permanent way have a different shape, very different sizes, in addition, they are made of different materials, have different mechanical and thermal properties, and have an impact on the stress-strain state of the rails. In principle, it is possible to create a geometric model of the track, which would include all of these elements. However, it should be borne in mind that the accuracy of the solution of the boundary value problem in a finite element method is highly dependent on the size of the finite element (the accuracy increases with a decrease in the size). At the same time, with a decrease in the size the number of elements increases sharply, resulting in a significant augmentation of computation costs or inability to solve the problem with the software used. In this connection, it seems appropriate to divide the problem of track deformation under the influence of loads from the rolling stock wheels into several stages, and use different geometric patterns for each stage. In particular, when calculating stresses and strains away from the contact zone between the rail and the wheel, other elements of the permanent track structure (pads, fasteners, sleepers, etc.) can be considered as a base with the averaged mechanical properties. Observations have shown *[1]* that the rail sections away from the contact zone (within three sleepers on the left and right of each bogie wheel) are subjected to buckling and torsion (induced by a vertical load eccentricity). Thus, to consider the impact of both bogie wheels on the rail deflections and ignore the influence of fixing scheme on the stress-strain state of the rail, the length of a geometric model of the rail must be at least 6 m. In addition, the finite element mesh can be quite tenuous. Elastic bending strain in the rail section was calculated by the finite element method using the said geometric model. In addition, it was loaded with the moving pair of concentrated forces that correspond to axial load. To calculate the stress-strain state of the rails in the contact zone between the rail and the wheel, the finite element method was used. The geometrical model was a 600 mm long rail with a pivot bearing on the ends, which allowed considering the effect of the rail deflection on the value of the contact patch. In the central part of the rail, an area was found that was three times the size of the expected area of contact. In this region, the finite element mesh was concentrated. The problem was solved in a three-dimensional elastic-plastic formulation in two phases.

In the first phase of the calculation, the optimal size of finite elements was determined. To this end, the problem was first solved in the elastic formulation, comparing the results of the numerical solution with those of the analytical solution obtained in the case of contact between two elastic cylinders with mutually perpendicular axes (Hertz solution). Only static vertical load on a wheel (100 kN) was considered, which was applied as a uniformly distributed normal pressure on the inner surface of the wheel hub.

It was found that with the average size of finite elements in the area of condensations (i.e. the average length of the tetrahedron edges) of the order of 0.5 mm, the difference between the resulting solution and the analytical solution does not exceed 4%. In addition, the maximum contact pressure was 988 MPa; the size of the contact patch (the size of semi-axes of the ellipse) was 13.6 and 13.58 mm. The shape of the contact area was close to the circle, due to the proximity of the radii of the railhead and the wheel (rolling circle). The center of the contact patch is located on the axis of symmetry of the rail cross-section, which is explained by the coincidence of the values of the wheel taper and the rail canting.

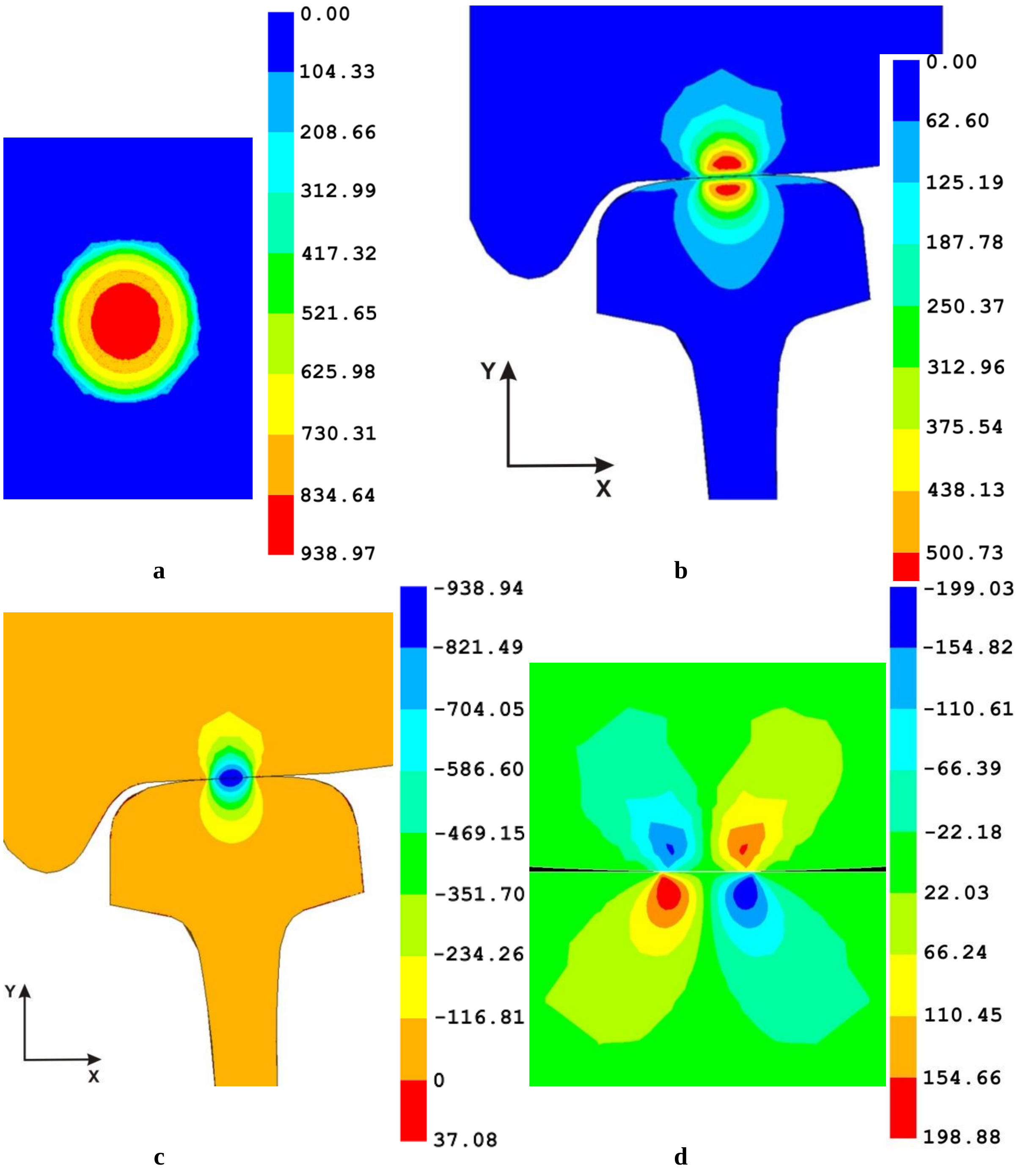


Fig.1. **Some results of elastic-plastic calculation – distribution of contact pressures (a), equivalent stresses (b), normal stresses σ_y (c) and shear stresses τ_{yz} (d)**

A further decrease in the size of finite elements provides for a slight improvement of the solution, but due to a significant growth in their amount, the time of the problem solution increases appreciably. In case of increasing the size of finite elements in the contact patch, the quality of the solution deteriorates greatly, and power surges can be observed, which are caused by poor surface smoothing and the appearance of false contact points.

Taking plastic strains into account leads to a certain increase in the size of the contact pad and a marked reduction of contact pressure (Fig. 1a) as compared with the elastic solution. The zone, in which plastic deformation takes place, is located inside the railhead within a short distance from the surface. The point, at which the stress intensity has the maximum value (dangerous point), is located at a depth of 3.32 mm. The nature of the distribution of equivalent stresses is shown in Fig. 1b. All normal stresses are negative in the contact zone; the greatest of them by the absolute value is $\boldsymbol{\sigma_y}$ (Fig. 1c). Shear stresses in the dangerous point are not large (Fig. 1d), so the direction of the principal axes of the stress tensor in the dangerous point is close to the direction of the coordinate axes. The value of the maximum shear stress $\boldsymbol{\tau_{max}}$ in the dangerous point is 269.9 MPa at the maximum shift of $\boldsymbol{\gamma_{max}}$ =0.412%.

A contribution to the total damage of the material is made both by the elastic cyclic strains resulting from the rail bending, and cyclic plastic strains in the contact zone. The latter can be accumulated due to a complex (non-proportional) loading.

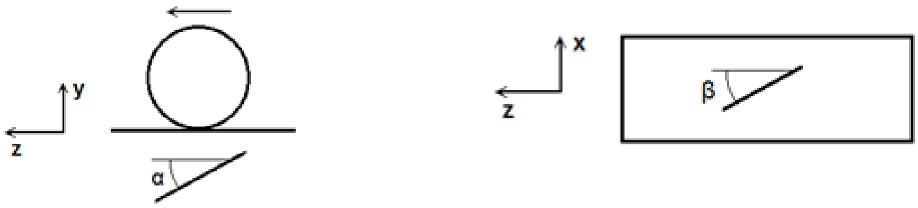
4. Discussion and Generalization

Since the computational costs for the implementation of the considered methodology for assessing the durability were sufficiently large, in a first approximation we considered only damage caused by low-cycle fatigue and the rule of linear summation of damage. In addition, damage accumulated in the $\boldsymbol{i}$-th cycle of loading *[5, 6]* is determined as $\Delta\omega_i = \frac{1}{(N_f)_i}$, where durability N_f at a current level of stresses and strains is found from the solution of the non-linear equation

$$(FP)_{max} = \frac{(\bar{\sigma}_f)^2}{E}\left(2N_f\right)^{2b} + \bar{\sigma}_f\bar{\varepsilon}_f\left(2N_f\right)^{b+c}, \tag{1}$$

where $FP = \langle\sigma_{max}\rangle\frac{\Delta\varepsilon}{2} + J\Delta\tau\Delta\gamma$ is the damage parameter. Here, brackets $\langle\ldots\rangle$ denote the operator $\langle x\rangle = \frac{|x|+x}{2}$; σ_{max} is the maximum stress, normal to the plane of crack propagation; $\Delta\varepsilon$ is the scope of the normal strain in the direction of σ_{max}; $\Delta\tau$ and $\Delta\gamma$ are the scopes of shear stresses and shear strains in the plane of crack propagation; J is the material parameter, which depends on the stress state type; E is the elasticity modulus of the material. The other parameters are determined by means of approximation of the curve of the fatigue life of the material obtained under uniaxial cyclic loading (tension-compression or torsion) with the control of the strain amplitude. The plane of crack propagation (critical plane) is defined as the material plane with the highest value of the damage parameterFP. The material plane, in which maximum shear stresses are active, will be a critical plane in this case, as all principal stresses in the dangerous point are compressive. For this plane, the fatigue parameterFP, estimated by formula (1), has a maximum value of 0.2161. The orientation of this plane in the dangerous point defines the initial position of the crack. Figure 2 schematically shows the position of the critical plane relative to the coordinate axes and the direction of the wheel motion.

As a first approximation, we assume that the position of the dangerous point is not changed during loading. The fact that plastic strains are small enough confirms this hypothesis, and the problem is solved in a geometrically linear formulation. Then for the dangerous point we obtain α=24.18°, β=88.14°, which is close to the results obtained in [5, 6].



a b

Fig. 2. **Position of the crack nucleation plane with respect to the coordinate axes and the direction of the wheel motion: a – side view, b – top view**

The durability of the rail (the number of cycles to failure) at a given value of $(FP)_{max}$ will be 12.2·106 cycles, which corresponds to the passage of approximately 245 million tons of gross cargo at a given axial load. This value is minimal for the durability under the operating conditions taken in this calculation, since for the cyclically hardening materials the durability will be maximal in the first cycles of loading. As the material hardening, the value of the increment of plastic strain per cycle decreases, and, therefore, the intensity of the damage accumulation declines. If we compare this estimate with a durability warranty (180 million tons of gross cargo for the rail type P65 produced by OJSC MK Azovstal), it should be noted that this estimate corresponds to the moment of the crack nucleation, but not its propagation up to the size limits. The authors are intended to consider other factors that affect the durability of rails in subsequent publications. Of course, this method of estimating the ultimate state of the rail is approximate, because it does not take into account many factors that are difficult to assess (for example anisotropy of the mechanical properties of the rail material, associated with the technology of rolling, heat treatment and straightening of the rails, the effect of vibration, the imperfections of the rail track, the parameters of the environment, and others.).

5. Conclusions

The main advantage of the developed method compared to traditional approaches (full-scale and model experiments) is the possibility to conduct numerical experiments dedicated to the effect of individual process and design factors or their combination on the theoretical life of the structural element. The results of such numerical experiments can be used to optimize the design and technological parameters of critical structural elements of railway track and rolling stock.

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ОЦІНКА ДОВГОВІЧНОСТІ РЕЙОК ЗА КРИТЕРІЄМ ВИНИКНЕННЯ ВТОМНИХ ТРІЩИН

У статті представлено сучасні концепції виникнення тріщини у головці рейки внаслідок контактної втоми матеріалу. Наведено результати чисельного моделювання контактної взаємодії рейок та коліс рухомого складу методом скінченних елементів. Задачу контактної взаємодії рейки типу Р65 із стандартним залізничним колесом розглядали у пружно-пластичній постановці. Умови взаємодії відповідали руху поїзда на прямій ділянці колії без ковзання. Сучасна методологія оцінки довговічності конструкцій передбачає розрахунок приросту пошкодження у кожній точці елемента конструкції, оскільки навантаження змінюється з плином часу, і подальше підсумовування цих пошкоджень. Після досягнення граничного значення загального пошкодження вважається, що елемент конструкції втрачає свою несучу здатність, тобто в ньому утворюється тріщина. Незважаючи на істотне спрощення реальної проблеми контактної взаємодії рейок з колесами, обчислювальні витрати на впровадження таких методів прогнозування довговічності будуть надмірно високими. У зв'язку з цим запропоновано спрощений метод розрахунку довговічності рейок.

Ключові слова: *залізничний транспорт, деформація і руйнування, міцність залізничної колії, аналіз руйнувань, контактні напруження, контактно-втомні пошкодження, чисельне моделювання, метод скінченних елементів.*