Remaining lifetime forecasting based on the dynamic simulation of used continuously welded rails

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Abstract: One of the basic principles of the modern track facilities policy is the maximum life extension of the permanent way elements less than 1.1 bln tonnage. This policy allows to increase the competitiveness of the railway transport. The article is devoted to the research of the dynamic load impact on the target remaining lifetime of a continuously welded rail. The article also contains the results of the research that describe how some rail segments work under the load of different types, including pre-stress that develops under fastening the rail in the track and under the accumulation of the residual stress in operation.

1. Introduction
Nowadays the alignment of track geometry deviations, the standard support of rails on the subgrade, reliable fastening of continuously welded rails that excludes creeping, accurate rail canting, careful loading of rails, careful laying and field operation of rails are important conditions for providing warranty and super-warranty life of rails.

The current policy of JSC «Russian Railways» and some large enterprises, that have in-plant and spur tracks, is focused on the use of resource-saving technologies in various spheres of operation, including the maintenance of track facilities. An important technology used to maintain assembled rails and sleepers is to “rescue” continuously welded rails by relaying them from curved track sections to tangent ones or relaying the inner and outer rail on the curved track sections with the certain load-strain and with the regulated lateral wear. They think that it is rational to relay continuously welded rails when they achieve pre-limit lateral wear of 13-14 mm [1-3].

Strong side wear of continuously welded rails is often developed on curved track sections due to the action of inertial forces and friction forces between the wheel edge and the rail head. The problem of the side wear of continuously welded rails on curves is important not only in the context of defects in the permanent way elements of the track, but also in the context of smooth running, traction, passenger comfort, and traffic safety [2, 4-5]. The tribowheel wear of a wheel-rail (figure 1) can be broadly divided into adhesive and abrasive.

Adhesive side wear causes the damage due to gripping of the contact surfaces of the rails and the wheel ridges during their mutual sliding. Abrasive side wear of rails and wheel ridges in their contact areas is characterized by the abrading of metal particles when exposed to abrasive grains, as well as when the rough hard surface of the wheel ridge slides on the gauge face of the head and vice versa. This problem is especially urgent on those track sections that have curves with R<600 m. On such sections, the rails are changed 3-4 times more often than on more tangent lines. The rails which have excessive side wear of more than 20 mm must be replaced first. We note a number of main factors that affect significantly on strong side wear:

• the intensity of wear depends on the radius of the curve;
• the growth of vertical and, particularly, horizontal track stiffness (the introduction of strong rails of heavy types, reinforced concrete sleepers and rigid fasteners);
• the replacement of the box plain friction bearings into the roller plain friction bearings on the rolling stock (in addition to the elimination of the natural lubrication of the rail with leaking grease, it has led to a high increase in the resistance to turning of the rolling stock bogies on curves);
• the implementation of electric braking from front of a train, which is accompanied by the forced car movement out of alignment;
• the use of heavy trains and trains of increased length, the increase of cargo transportation;
• coming into operation of new powerful locomotives;
• an overestimated value of wheel stiffness of the traction rolling stock (HB 550 and more) with plasma strengthening of the rail in relation to the rail head strength (HB 350) with an orange-peel effect of the rail surface;
• a quick decrease in the rail lubrication efficiency due to the abrasive particles in the contact area of the wheel and the rail, normally consisting of metal products of rail wear and rail wear, as well as quartz sand.

Figure 1. Schematic diagram of rail side wear.

To create an adequate model of the dynamic behaviour and operation under actual track conditions for a rail segment, it is necessary to take into account the above mentioned factors.

The monitoring system of temperature stresses in the continuously welded rails allows to determine more accurately both the stress state of the operating continuously welded rail and the state of the continuously welded rail when carrying out its unloading [6-7]. On fixing the anchor section on the stable end of the continuously welded rail with a hydraulic tension device (figure 2), the continuously welded rail is stretched to bring the gap in the joint where the device is installed to the standard value. At the same time, the monkey pile is struck at the end of the wedge, the force is transferred to the continuously welded rail and it is moved.

Figure 2. The adjustment of temperature stresses with the use of a hydraulic tension device.
After the replacement of the «rescued» continuously welded rail, it is planned how to relay this rail on the tangent line section according to the standard techniques and procedures. The dismantled used continuously welded rail without side wear is normally laid in the next curved section of the track. In this article it is proposed to conduct a number of research to understand how some rail segments operate under various types of load, including pre-stresses that develop when the rail is fastened in the continuously welded rail and when residual stresses accumulate during the operation. To solve this problem, a mathematical model of the rail segment dynamic behaviour is presented.

2. Methods
To create an appropriate model of the dynamic behaviour and operation of a continuous welded track element, put into the optimal temperature condition, we will highlight the main factors that affect the nature of the behaviour and state most of all:

- the radius of the curve on the railway section;
- the change of the vertical and horizontal stiffness of a track due to the usage of reinforced concrete sleepers, the alternations in their density and fastenings types, the existence of pipe-culverts, the sections with the variable stiffness, etc.;
- the increase of wheel hardness of traction rolling stock due to the modern methods of bandage strengthening (HB 550 and more) in relation to hardness of a rail head (HB 350) due to roughness of a bandage surface;
- reduction of the lubricator efficiency due to the penetration of quartz sand and abrasive particles in the form of wear debris of rails and wheel flanges in the contact zone between the wheel and the rail.

To describe the analytical dependence for the contact force and the track parameters, various models [8-10] are used, among which it should be mentioned those that contain several expressions for stages of loading and unloading (the initial stages of loading, the maximum output and unloading are most often defined) — the elastic-plastic model of Kilchevskiy (1) and Alexandrov-Kadomtsev (2); the model with unloading, where the beginning of the interaction before reaching the peak power is described with the classical formula of Hertz, and after — with the expression (3); the models with branch aliasing of loading and unloading - Abrate model (4), the linearized model with the varying mechanical parameters for different loading stages (5), Shimasu model (6), the empirical model (7).

\[
\alpha = \begin{cases} 
bp^{2/3}, & \text{for a loading stage} \\
bp^{2/3} + pd, & \text{for a loading stage} \\
bp^{2/3} + p_{\text{max}}d, & \text{for a loading stage}
\end{cases}
\]

\[
\begin{align*}
P &= P_m \left[ \frac{a - a_0}{(a_m - a_0)} \right]^{\alpha}, & \text{for a loading stage} \\
P &= P_m \left[ \frac{a - a_0}{(a_m - a_0)} \right]^{1.5}, & \text{for a loading stage} \\
P &= k_1 \alpha, & \text{for a loading stage} \\
\end{align*}
\]

\[
\alpha = \left( \frac{p}{k_H} \right)^{2/3} \left( 1 - \frac{\ln z p^{1/2} k_H^{2/3} k_0}{h} \right), 
\]

where

- \(P\) is the contact force,
- \(P_m\) is the maximum contact force,
- \(a, a_m, a_0\) are the parameters of the bandage surface,
- \(k_1, k_2, k_3\) are the coefficients of the bandage surface,
- \(P_m, k_1, k_2, k_3\) are the parameters of the bandage surface,
- \(Q, h\) are the parameters of the bandage surface.
\[ P = kw_0^3, \]  

in the expressions (1) – (7) the following symbols are adopted: the plasticity - \( \lambda = 5.7 \), \( \chi = \pi k_{pl} \), \( b = ((9\pi^2 (k_1 + k) / 16R)^{1/3} \), \( k_1 = (1 - \alpha_2^2) / R \), \( k = (1 - \sigma^2) / E \), \( R_f^{-1} = R^{-1} - R_f^{-1} \), \( P_i = \chi^3 (3R(k_1 + k) / 4)^2 \), \( b_y = R_f^{-1/2} (3(k_1 + k) / 4)^{2/3} \), \( c_1 = 3\chi^{1/2} (k_1 + k) / 8 \), \( R_f = (4/3 (k_1 + k)) p_{\text{max}} x^{-3/2} \), \( \alpha = \beta (P_{\text{max}}) = (1 - \beta) P_{\text{max}} (2\chi R_f)^{-1}, \) \( K_2 = 1.5K_H / a_c \), \( d = 1/2kR \), \( k_{pl} = \) the smallest of the plastic constants, \( \beta = 0.33 \), then the block of variables follows, which is often determined by an empirical way: \( P_m \) – the maximum value of the contact force before the beginning of the loading stage, \( a_m \) – maximum material crushing in the contact zone, \( a_0 \) – a current crushing value, \( a_u \) – an ultimate crushing value, \( k_H \) – contact stiffness, determined by the radius of a wheel \( R \), an effective contact modulus \( Q_H \) and the defects on the wage wheels (burns, scaled wheels, cracks and so on).  

3. Results  

Modeling of the track dynamic behaviour within two adjacent sleepers, under the operation of the wheel pair of rolling stock is proposed to perform with the use of the equations for the orthotropic flat element [10, 12], pre-stressed with the longitudinal force and two moments that, in general, corresponds to the segment condition of the continuously welded rail laid in the track according to the consolidation. In the di-mensionless form the ratio data can be written as follows  

\[ \frac{d^2\phi}{dr^2} + \frac{1}{r} \frac{d\phi}{dr} + \frac{1}{r^2} \frac{d^2\psi}{d\theta^2} - \frac{1}{r^2} \phi + \frac{c_2}{c_1} \frac{d^2\psi}{dr^2} + \frac{c_2}{c_1} \frac{d\psi}{d\theta} - \frac{c_2}{c_1} \frac{d\psi}{d\theta} + \frac{12c_4}{c_1} \frac{dw}{dr} - \phi = - \frac{d^2\phi}{dt^2} + M + \frac{M_\Delta \alpha_H}{c_1 \rho}, \]  

\[ \frac{d^2u}{dr^2} + \frac{1}{r} \frac{du}{dr} + \frac{c_3}{c_1 r^2} \frac{d^2u}{d\theta^2} - \frac{c_2}{c_1 r^2} \frac{du}{d\theta^2} - \frac{c_2}{c_1 r^2} \frac{du}{d\theta} - \frac{c_2}{c_1 r^2} \frac{du}{d\theta} + \frac{d^2u}{d\theta^2} + \frac{M_\Delta \alpha_H}{c_1 \rho}, \]  

\[ \frac{c_4}{c_1} (\frac{d^2w}{dr^2} - \frac{d\phi}{dr}) + \frac{c_4}{c_1} \frac{dw}{dr} - \frac{\phi}{r} = \frac{12c_4}{c_1} \frac{dw}{dr} - \psi = \frac{d^2\psi}{dt^2} + \frac{M_\Delta \alpha_H}{c_1 \rho}, \]  

Here \( c_1 = \frac{E_r}{1 - \sigma_2 \sigma_3}, c_2 = \frac{E_\theta}{1 - \sigma_2 \sigma_3}, c_3 = \frac{G_{r\theta}}{\rho}, c_4 = \frac{K_{rz}}{\rho}, c_5 = \frac{K_{r\theta}}{\rho}, q_1 = \frac{q_h}{\rho c_1}, M = \frac{12q_1 K_{z1}}{h^2 \rho c_1}, \]  

\[ R_1 \] – the radius of a wheel, \( q \) – reduced load, \( \alpha_1, \alpha_2 \) – angles of the direction of the greatest force in the vertical and horizontal plane, \( E_r, E_\theta \) and \( \sigma_1, \sigma_2 \) - the elasticity modulus and the coefficient of transverse deformations for the directions \( r, \theta ; G_{r\theta} ; \) \( \sigma_3 \) – the shear modulus in the planes indicated in the lower case; \( w(r, \theta) \) – the standard movement in the middle plane, \( u(r, \theta) \) and \( v(r, \theta) \) – the tangential movements of the middle surface respectively the coordinates \( r, \theta ; \varphi(r, \theta) \) and \( \psi(r, \theta) \) – the functions of the turning angles in the directions \( r, \theta \), \( \tau = \frac{\psi_0}{\theta_H}, \Delta _1 = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial}{\partial \theta} \right), \Delta _2 = \frac{1}{2} \frac{\partial^2}{\partial \theta^2} \), \( N \) – the external longitudinal force acting along the continuously welded rail, \( M_r \) – the external bending moment, the vector of which is directed along the rail, \( M_z \) – the external turning movement, the vector of which is directed along the sleeper.  

To solve the system of equations in the image space, we present the external loading function \( q(r, \theta) \), including the concentrated interaction force of the wheel and the rail \( P(t) \), as the expansion into the series according to spherical functions on the basis of Legendre polynomials  

\[ \tilde{q}_1 = \frac{P(r)}{\pi R_e^2} \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} (4n + 3) P_{2n+1} \left( \cos \frac{mr}{2R} \right) P_{2n+1} \left( \cos \frac{mr}{2R} \right) \cos(m\theta) \]  

(13)
Unknown linear and angular displacements are also presented as the expansions into the series according to Legendre polynomials
\[ \tilde{x} = \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} x_{2n+m} P_{2n+1}(\cos \frac{m\theta}{2R}) \cos(m\theta). \tag{14} \]

in the expressions (13) and (14), the tilde above the variable indicates that this value is used in Laplace space, the variable \( x \) takes the values \( u, v, w, \varphi , \psi , R \) — the characteristic linear size of a continuously welded rail segment, \( r_i \) — the coordinate of the point where the dynamic contact of the wheel and the rail takes place. This coordinate is counted along the continuously welded rail.

To determine the coefficients of the series (14), we use their representation near the desired point in the form of the Laurent series, in which \( \varepsilon = p^{-2} \)
\[ x_{2n+m} = x_{2n+m}^0 \varepsilon^0 + x_{2n+m}^1 \varepsilon^1 + x_{2n+m}^2 \varepsilon^2 + x_{2n+m}^3 \varepsilon^3 \tag{15} \]

By substituting the series (4) with reference to the ratios (15) into the system (8-12) and by equating the coefficients to the similar degrees \( \varepsilon \), we obtain the systems of linear algebraic equations, from which we define:
\[ \Phi^2_{2n+m} = \frac{\sin \alpha P_2(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) \cos \alpha P_2(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + C_{\varphi 2}(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4})}{C_{\varphi 2}(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + C_{\varphi 2}(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4})} + C_{\varphi 2}(A_{\varphi 2}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) \]
\[ \Phi^4_{2n+m} = \frac{\sin \alpha P_4(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + \cos \alpha P_4(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + \cos \alpha P_4(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + \cos \alpha P_4(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4})}{C_{\varphi 4}(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + C_{\varphi 4}(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + C_{\varphi 4}(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) + C_{\varphi 4}(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4})} + C_{\varphi 4}(A_{\varphi 4}A_{\varphi 3} - A_{\varphi 3}A_{\varphi 4}) \]

where
\[ A_{\varphi 3} = C_{\varphi 3} \frac{C_{\varphi 3}}{C_{\varphi 3}} - C_{\varphi 3} \frac{C_{\varphi 3}}{C_{\varphi 3}} - C_{\varphi 3} \frac{C_{\varphi 3}}{C_{\varphi 3}}, \]
\[ A_{\varphi 4} = C_{\varphi 4} \frac{C_{\varphi 4}}{C_{\varphi 4}} - C_{\varphi 4} \frac{C_{\varphi 4}}{C_{\varphi 4}} - C_{\varphi 4} \frac{C_{\varphi 4}}{C_{\varphi 4}}, \]
\[ P_s = P(p) \frac{4n + 3}{\pi R^2} \int_{2n+1}^{P_{2n+1}} \left( \cos \left( \frac{\pi R}{2R} \right) \right) P_{2n+1} \left( \cos \left( \frac{\pi R}{2R} \right) \cos (m\theta), i = 0,1,2,3. \]
The ratios $C_{fj}$ ($rde j = 1,2,3,4,5$) indicate the ordinal number of the equation in the system (8-12). They are defined as the amount of the coefficients in the corresponding equation for all displacements $x$.

The expressions (13) are calculated for a specific contact area of the wheel and the rail, in other words, $r$ and $\theta$ acquire certain values. After the inverse Laplace transformation, the displacements can be written as a function of time, two coordinates, and the impact force on the plate

$$x(\varphi, \theta, \tau) = \frac{(1-a+a\pi h)}{\pi R^2 E_r} \int_0^\tau \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} (4n+3)P(r_1)P_{2n+1} \left( \cos\left(\frac{\pi r_1}{2R}\right) \right) P_{2n+1} \left( \cos\left(\frac{\pi \tau}{2R}\right) \right) \cos(m\theta) \times$$

$$\left[ x_{2n+m}^0 \delta ac(\tau - \tau_1) + x_{2n+m}^1(\tau - \tau_1) + x_{2n+m}^3(\tau - \tau_1)^3 + x_{2n+m}^5(\tau - \tau_1)^5 \right] d\tau_1 \tag{21}$$

Having calculated all functional coefficients of two types of the series, it is necessary to pass to the space of originals.

To determine the dynamic characteristics of the deformed state of continuously welded rails, for example, the standard force of the interaction between a wheel and a rail contact force, it is necessary to consider the functional equation, describing the vertical movement of a wheel pair

$$m\ddot{y} = -P(t) \tag{22}$$

where $m$ – the reduced mass of a means of transport, acting on one axle, $y$ – the total vertical movement of a wheel, which depends on vertical movements.

On substituting the expressions for the contact force (1) – (7) and the expressions of Type (14) for various vertical movements of continuously welded rails under the deformation in ratio (15), we get the functional integro-differential equation, which can be solved using a computer.

4. Conclusions

The resulted dependencies of the contact force between the wheel and the rail on time, which can be applied to different models of the dynamic interaction, allow us to forecast the state of any used continuously welded rail with the determined failure probability in accordance with the optimal ratio of geometric, mechanical and structural characteristics of the track section and its operational parameters.

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