Influence of wear on the distribution of pressure and the state of tension at wheel-rail interface

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Abstract. Since no effective experimental approaches have been proposed to assess state of stress and distribution of pressures at the wheel and rail contact interface to date, numerical calculation methods are known as an alternative to approximate modelling of wheel-rail interaction. In this paper, a numerical procedure is proposed based on the finite element method of the complex tensions state on wheel-rail contact. This study includes the distribution of pressures and tensions on wheel-rail contact system with new and worn profiles, using a finite element modeling (FEM). Using a model of isotropic elastic-plastic material was obtained a FE model that can obtain the distribution of pressures and tensions at the wheel-rail interface for the real surfaces that are in contact, this model is necessary for any tribological study that requires data on the state of stress and pressure only by introducing input data: geometric characteristics, Young’s Modulus, Poisson’s Ratio, Tangent Modulus, Yield Strength, load on the wheel, lateral shift of wheelset.

1. Introduction

The profile of the wheel and rail affects their wear and tear. Interaction in the wheel-rail system will always be accompanied by wear and tear. The problem is its intensity, the rate of loss of metal on the thread surface of the wheels and on the lateral surface of the rail. There are many factors that influence this process. Dozens of comprehensive studies of the interaction between rolling stock and railways have been conducted to find a "cure" from it and rolling surface lubrication and hardening technologies have been introduced. But the intensity of wear remains high. Among them introduction of heavy-type rails and reinforced concrete sleepers, as well as an increase in load on the axis, excessive tilting of the rails, different profiles of wheels and rails. Contact interaction of a wheel and rail, there should be a low level of friction to ensure the movement of large masses, with low-resistance, while the level of friction should be sufficient to ensure the necessary traction. Constructed materials must be strong enough to ensure resistance to vertical forces resulting from high loads and dynamic reactions in wheel and rail interactions caused by accelerations of rolling stock elements, which are caused by the roughness and irregularities of geometric characteristics on rails/wheels. However, neither the rate of wear nor the rate of development of wear defects should be so high as to pose a threat to the safety of movement on the railway. One of the tasks of contact mechanics is wheel-rail interaction, the geometric parameters and material proprieties of the wheel-rail system, influence the distribution of normal, tangential tensions, relative skids, and friction forces on the contact surface. The contact load between two elastic bodies with approximately the same elasticity characteristics as in the case of the wheel-rail may be presented separately in the form of normal and...
tangential loads. The purpose of the first task is to determine shape size of the contact area, as well as the distribution of normal specific contact strains.

For understanding purpose task, involved in friction on the interaction between the wheel-rail, which is very complex and is characterized by loading, must be analyzed. Finding the contact area and distribution of pressures requires large computing resources and programming of complicated algorithms. This is why a very high effort has been put in the development of effective and efficient calculation procedures for contact problems between the wheel-rail.

1.1. Influence of normal and tangential contact stress on contact between rail and wheel

Due to the deformation of the wheel and rail during interaction, the wheel actually rests on the rail, on some surface. The process of deformation is very complex, and theoretical expressions describing the process of interaction of wheel and rail are obtained only for the simplest cases of combining wheel and rail profiles. The basis of the solution to the problem was laid by the physicist G. Hertz in 1882 in his work[1]. Based on the solution of hertz’s normal problem Pmax maximum contact tension can be calculated according to the formula:

\[ P_{\text{max}} = \frac{3 \cdot F \cdot E^2}{2 \cdot \pi^3 \cdot r_e^2 \cdot (1 - \nu^2)^2}, \]

where: \( E \) – Young’s modulus, \( F \) – normal load, \( r_e \) – equivalent radius in contact area derive from geometries of contact elements, \( \nu \) – Poisson’s ratio.

Thus, the normal tension on the surfaces of the wheel/rail depends of load on wheel, the properties of interacting materials, and the radius of the rolling-surface depends on the field of contact of the wheel and the rail. The contact area is extremely small and this causes high contact stresses[2, 3]. In a typical case, contact takes place on a quasi-elliptical site measuring area \( \approx 3\text{cm}^2 \) (Figure 1).

![Figure 1. Pressures distribution on the wheel force Q in the individual system components of the track [4].](image)

When the rolling stock moves, the position of the wheel pair in relation to the rails changes significantly [3], resulting in various combinations of contact areas of the wheel and rail (Figure 2).
Figure 2. Non-Hertzian concentrated contact situations in the case of wheel-rail interaction and definition of separations s and d specific to wheel-rail interaction.

Figure 3. Geometric elements defining wheel-rail contact[3].

Even with constant load on wheel, normal tension/strain and distribution of pressure will change significantly due to differences in the curvature radius of the contact surfaces of these zones. If there is one radius of surface curvature in the contact area, we can use the Hertz solution. If there are two or more curvature radii in the contact area, for example Figure 4 shows the geometry of the chosen profiles within initial contact points when the wheel is moved towards the rail and the radii of curvature at the contact points[4], Hertz's solution is unfair and a non-Hertz's solution should be used to determine the location of the contactFigure 5. This is especially important with a variety of combinations of worn wheel and rail profiles. When finding normal contact stresses for non-conforming and non-Hertz contact, various numerical methods and programs are used[5, 6, 7, 8]. The shape of contact area and the distribution of normal strain/tension depend on the normal load operating from the wheel on the rail, the profiles of the wheel and rail, the cross and angular positions of the wheel-seton the rails and canting of the rail, seeFigure 1, Figure 4and Figure 3.
Figure 4. Parameter definition of the contact setup for wheel with profile S1002 and rail with profile UIC 60, radii of curvature.

Figure 5. Non-Hertzian concentrated contact situations in the case of wheel-rail interaction[3].

When the wheel-set moves in a curve, at a certain angle of the attack the wheel can come into contact with the rail at two different points, first situation counting from the right seeFigure 5. Two-point contact leads to the formation of two contact areas on the surface of the rail and on the crown or shoulder side of the head of the rail and around gauge face. Due to the fact that the wheelset moves with some angle of attack, the contact area in the area of gauge face is shifted. Increasing the angle of the attack leads to an increase in the distances between the contact and to the instantaneous axis of rotation of the wheel-set, and thus to the increase relative slip and tangential force associated with it and increase the contact stresses which causes a plastic deformation on the rail head at the gauge face and on the flange face on the wheel rim.

When a new or worn rail encounters a new or worn wheel, the shape of distribution pressure area changes. The size of the contact area is significantly reduced, it shifts to the outer surface of the outer rail, leading to an increase in contact pressures, the level of which can reach the limit of yield strength, which causes a plastic deformation on the head of the rail and in the depth of the wheel tread.

The size and distribution of contact strains/tensions depend significantely of the wheel-rail profiles and on single-point or two-point contact interaction. Definitions for characterizing the wheel-path contact are the values of the separations s and d Figure 2, for situation were $s \cup d \leq 0.1$ mm, Wheel-path contacts are the tightly-conforming type, lead to intense wear and should be avoided[3].
According to Hertz’s theory[1], maximum static compression stresses take place on the surface of the contact elements, and maximum tangent stresses at a depth of 0.78 ∙ a, where a - half the length of the large axis of the elliptical contact shape. That directly under the contact area the material is in a three-tension state. The three components of the stress tensor are approximately equal, resulting in a high level of material carrying capacity. Further deep into the material these stresses become unequal and the level of maximum tangent stresses reaches its highest value. When tangential force is applied to the surface, the maximum tangent stress increases and this one being attained at smaller depth, [2].

Even if the normal deformation on the surface is elastic, near the surface can occur plastic deformations. Under the influence of rolling-stock moves on rail, under the surface there is a cyclical tension of compression-stretching, leading to the accumulation of subsurface plastic deformation and the appearance of residual stresses in rail/wheel material. This behavior of the material is the cause of various types of contact fatigue defects in the wheels and rails [9]. Two materials experience a major deformation. One of them is a very thin layer near the surface of the contact area, the other - subsurface volume near the place of maximum tangent strains. As the tangential traction force on the surface increases, these volumes approach each other and can form one area of potential material destruction.

1.2. Numerical solution procedure for the implicit FEM wheel-rail static interaction analysis
The modelling of wheel-rail contact is area of interest to many researchers, different numerical algorithms which can solve the normal and tangential forces for non-Hertz’s W/R contact, and it also provides a detailed description of the surfaces in contact and find the state of the strain/stress in the area of interest as mentioned above. This studies also explores damage mechanisms such as surface cracks, plastic deformations, wear, and fracture under fatigue. Proper for understanding and modeling of contact mechanisms, serves to clearly understand the detailed knowledge of the physical interaction between the W/R. In the literature, some of these issues are studied using some experimental observations and numerical calculations in different contexts. In this study, it is designed to demonstrate a well-known FEM procedure using the 3D model of W/R with new and worn profiles with a realistic 3D models and parametric study that gives a better understanding of the issues of interest.

For a good result from the modelling of contact between wheel and rail, need to use the Finite Element Method scenarios with the realistic 3D – solution for a better result, also on a good solution influence a parameter like a contact stiffness factor, contact damping, mesh size in contact area, material model and others. Contact and target surfaces are generated if two bodies are not attached to each other but still transmit forces. Since wheel pull on the disc, surface to surface contact was adopted. The wheel surface was modelled as a contact surface element and rail surface with the target element. Augmented Lagrange was adopted to represent nonlinear contact between the two bodies. The contact pressure ($F_{normal}$) was calculated using equation 2:

$$F_{normal} = k_{normal} \cdot X_{penetration} + \lambda$$

In Figure 6 as shown, contact stiffness was directly related to the penetration ($X_{penetration}$). Due to extra term, the normal force is less sensitive to the contact stiffness ($k$). In interface treatment, initial contact behaviour was defined as offset or adjust to touch,[10]. In Figure 7 adjust to touch determines the contact offset necessary for closing the gap and establish initial contact.
Physical contacting bodies do not interpenetrate, which is called “contact compatibility”, which is not numerically possible but as long as penetration is small enough or negligible, accurate solution could be guaranteed. Therefore, the program must establish a relationship between the two surfaces to prevent them from passing through each other during analysis. In Figure 8 shows schematic representation of wheel-rail contact interaction of the two contact bodies. For the contact to target, it is necessary to establish an attitude to rigidity between the two contact surfaces, this interaction is generated by a spring that is placed between two contact, where the contact force is equal to the product of contact stiffness, penetration and is a normal vector on the contact surface.

$$f_c = -s_l \cdot k_l \cdot n_l \text{ if } s_l < 0$$  \hspace{1cm} (3)

The stiffness factor \(k_l\) for the contact segments is given in terms of bulk modulus \(K_i\), the volume \(V_i\), and the face area \(A_i\) of the element that contains target segment as for brick elements. \(\alpha\) is the interface stiffness scale factor.

$$k_l = \frac{\alpha \cdot K_i \cdot A_i^2}{V_i}$$  \hspace{1cm} (4)

From equation (4), it can be notice that the variation of three factors, mesh size in zone of interest, interface stiffness factor and mechanical properties of material related to contact stiffness definition, might have an impact on the contact interaction.
Interface stiffness scale factor $\alpha$ represents the product of penalty scale factor and scale factor on contact/target stiffness, which is scale factor for the interface stiffness and is set to 1. If stiffness factor increases the oscillation of amplitude on the resultant force is low.

If substitute the element size variation from $l \cdot l \cdot l$ mm to smaller size $m \cdot l \cdot m \cdot l \cdot m \cdot l$ mm into equation (4), the contact stiffness on the small segment is:

$$k = \frac{\alpha \cdot K \cdot (m \cdot l)^2}{(m \cdot l)^3} \quad (0 \leq x \leq 1) \tag{5}$$

If the element size is reduced from $l \cdot l \cdot l$ mm to smaller size $m \cdot l \cdot m \cdot l \cdot m \cdot l$ mm, the force applied on the contact surface will decreased and we obtain the $m \cdot m \cdot f_s$. Introducing the $f_s$ and $k$ into equation (3):

$$m \cdot m \cdot f_s = -s_{ml} \cdot k \cdot n = -s_{ml} \cdot \alpha \cdot K \cdot m \cdot l \cdot n \tag{6}$$

and obtain the depth of penetration:

$$s_{ml} = \frac{m \cdot m \cdot f_s}{\alpha \cdot K \cdot l \cdot n} \tag{7}$$

The equation (7) means that to receive a lower penetration we need mesh elements with smaller sizes.

From equation (4) the contact stiffness is related to bulk modulus $K$. For the contact problem the hardness $K$ is important mechanical material property which affects the interface deterioration and play a important role in the distribution of stress/strain on wheel/rail contact. The hardness $K$ can be expressed as follow:

$$K = \frac{E}{3(1 - 2\nu)} \tag{8}$$

where $E$ is Young’s Modulus and $\nu$ -Poisson’s Ratio.

2. Modeling procedure

2.1. Input data for this study used is the described following:

Load on wheel is applied a vertical force with a magnitude of 100 [kN]. Rail is fixed at traverse location at all direction for prevented body motion. The effects of rotational wheel and lateral force neglected. The FE model expected to represent more realistic way for load transfer between the axle – wheel – rail. Monobloc-wheel with 460 [mm] radius with new profile S1002 [13] and worn profile shown in Figure 9.

![Figure 9. Wheel profiles used in modelling, generated in Inventor 2020.](image)

Rail profile: UIC 60[13] and worn profile shown in Figure 10.
Figure 10. Rail profiles used in modelling, generated in Inventor 2020.

Rail inclination: 0 rad
Distance between the inner faces of the two wheels: 1360 [mm]
Angle of attack: 0°
Load on wheel - 100 [KN]
Friction on the two directions (f_x = f_y = 0.1)
Lateral shift of wheelset: -3 to +5 mm

Material data for wheel, axle and rail is shown in Figure 11, Figure 12 and Figure 13

Figure 11. Material property for Wheel used in FEA.

Figure 12. Material property for axle used in FEA.
Figure 13. Material property for Rail used in FEA.

The geometry model transferred from CAD to CAE software and meshed with solid185 8-node hex mesh elements as shown in Figure 14 and Figure 17 for new and worn profiles. The contact zone is mesh very well and shown to in Figure 15 and Figure 18 for new and worn profiles. The contact is modelled with the conta174 elements placed on wheel and targe17 0 placed on rail. The finite element type information is presented in Figure 16 and Figure 19 for new and worn profiles.

Figure 14. Overview of 3D Model with new wheel and rail profiles meshes.

Figure 15. Mesh of wheel and rail with new profile at contact zone with the size of element on the tread zone with l=0,8 mm.
Figure 16. The finite element type information 3D Model with new profiles.

Figure 17. Overview of 3D Model with worn wheel and rail profiles meshes.

Figure 18. Overview of 3D Model with worn wheel and rail profiles meshes.
Figure 19. Overview of 3D Model with new wheel and rail profiles meshes.

Aspect Ratio information for new and worn profile of wheel/rail assembly is presented in Figure 20 and Figure 21.

Figure 20. Aspect Ratio information for new profiles of wheel/rail assembly.

Figure 21. Aspect Ratio information for worn profiles of wheel/rail assembly.
2.2. Results and Discussions:
The final FE model presented in section 2.1, is run and successfully post-processed to investigate the contact behavior under worn and new profiles and see pressure distribution and state of stress/strain.

2.2.1. Surface pressure
The results obtained on wheel/rail contact, for new and worn profile with the rail-cant 0-rad shall show that the wear change: the location of the contact Figure 22, the contact area Figure 24, the pressure distribution Figure 22, and the maximum pressure value Figure 23. Shown in Figure 22 of the non-Hertzian contact conditions, normal contact pressure distribution on the rail surface is presented with 2D contour plot and a 3D plot and the maximum inspected pressures for new and worn profiles. The non-Hertzian contact conditions, a very high normal contact pressure which values for new profiles is obtained 970.32 MPa and for worn profiles was obtained 1197.3 MPa at lateral shift of the Wheelset to -3 mm. The calculated pressure for the 3D model with new profiles compared to the worn profiles model is approximately 250 MPa, the maximum variation being recorded for the situation when wheelset lateral shift is -3 mm, with 264.08 MPa, showing an increase in pressure for worn profiles by about 27%, this scenario can be seen on in Figure 23. The contact area for the scenario where the wheel and rail profiles are worn, decreased compared to the situation when the profiles are new by about 30%, the maximum variation being inspected for the case when the wheelset lateral shift is -1 mm, with a value of 65.4 mm showing a decrease of 37%, this variation can be seen on in Figure 24.

![Pressure plot for model with new profiles](image1)

![Pressure plot for model with worn profiles](image2)

Figure 22. Normal contact pressure distribution on rail surface.
2.2.2. State of stress and strain

Results obtained for wheel/rail contact on 3D models of wheel/rail assembly, with new and worn S1002 profile for wheel and UIC60 profile for rail, show that wear change the location of the state of strain/stress and its maximum value. According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3 to +5 mm, state of equivalent Stress von-Mises was determined, for the interaction between a new S1002 wheel and a UIC60 rail with a rail tilt of 0, resulting in Figure 25.
Figure 25. Subsurface state of the Equivalent von-Mises Stress contact between wheel and rail for all scenarios with new profiles with variation of wheelset lateral shift -3 mm to +5 mm.
According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3 to +5 mm, state of Shear Stress was determined, for the interaction between a new S1002 wheel and a UIC60 rail with a rail tilt of 0, resulting in Figures 26.
Figure 26. Sub surface state of Shear Stress contact between wheel and rail for all scenarios with new profiles with variation of wheelset lateral shift -3 mm to +5 mm.

According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3, state of von-Mises equivalent Elastic Strain and von-Mises equivalent Plastic Strain was determined, for the interaction between a new S1002 wheel and a UIC60 rail with a rail tilt of 0, resulting in figures 26-27.

Figure 27. Equivalent elastic plastic strain distribution on wheel and rail for situations with new profiles with variation of wheelset lateral shift -3 mm.
Figure 28. Equivalent plastic strain distribution on wheel and rail for situations with new profiles with variation of wheelset lateral shift -3 mm.

According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3 to +5 mm, state of equivalent Stress von-Mises was obtained, for the interaction between an worn S1002 wheel and a worn UIC60 rail with a rail tilt of 0, resulting in Figure 29.
Figure 29. Sub surface state of the Equivalent von-Mises Stress contact between wheel and rail for all scenarios with worn profiles with variation of wheelset lateral shift -3 mm to +5 mm.

According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3 to +5 mm, state of Shear Stress von-Mises was obtained, for the interaction between a worn S1002 wheel and a worn UIC60 rail with a rail tilt of 0, resulting in Figure 30.
Figure 30. State of the Shear Stress distribution on wheel and rail for situations with new profiles with variation of wheelset lateral shift -3 mm to +5 mm.

According to isotropic elastic-plastic model, for lateral displacement values of the wheelset lateral shift from -3, state of von-Mises equivalent Elastic Strain and von-Mises equivalent Plastic Strain was obtained, for the interaction between a worn S1002 wheel and a worn UIC60 rail with a rail tilt of 0, resulting in figures 31-32.

Figure 31. Equivalent elastic plastic strain distribution on wheel and rail for situations with worn profiles with variation of wheelset lateral shift -3 mm.
3. Conclusions

1. As a result of operation, due to wear and tear, both the profile of the wheel and the rail profile undergoes changes, and for reasons related to traffic safety, the reduction of maintenance costs and not least the optimization of wheel and rail shafts, knowledge of the contact area and pressure distribution is necessary.

2. A FE model has been developed to obtain the 3D pressure distribution and state of stress/strain in non-Hertzian wheel-rail contacts subjected to both new and worn profile. The pressure distributions and state of strain/stress have been obtained considering isotropic elastic-plastic material model.

3. The results obtained on wheel/rail contact, for new and worn profile show that the wear change: the location of the contact, the contact area, the pressure distribution, and the maximum pressure value.

4. The results obtained on wheel/rail contact, for new and worn profile show that the wear change: the location of strain/stress, the maximum of strain/stress and also increase the plastic deformation.

5. The worn profile of wheel/rail are a major factor in influencing the shape of the contact area, the distribution of pressures, state of stress/strain and greater plastic deformation. FEA model showed the wear provide greater maximum pressures and stresses.

6. The finite element analysis of the elastic-plastic model has proven to be an effective instrument for determining the pressure distribution and state of strain/stress at the wheel-rail contact for new and worn profile. As well as for further work regarding the influence of passing wheel at rail-joint and interaction of fractured wheel/rail tread surface on distribution of pressure and state of stress at wheel/rail contact.

4. References

[1] Hertz H 1882 Ueber die Berührung fester elastischer Körper Journal für die reine und angewandte Mathematik (Crelle's Journal) 92 pp 156-171

[2] Spiridon C and Barbanta C I 2009 A Numerical Solver for the Wheel-Rail Contact Subjected to Combined Loads The annals of university “Dunărea de Jos“ of Galați VIII(XV) pp 73-78

[3] Spiridon C 2009 Contactul Concentrat Elastic-Plastic ( Iaşi: Polytehnium)

[4] Knothe K, Wille R and Zastrau B 2001 Advanced Contact Mechanics–Road and Rail Vehicle System Dynamics 35(4-5) pp 361-407

[5] Creţu S, Antalucia E and Creţu O 2003 The Study of Non-Hertzian Concentrated Contacts by a GC-DFFT Technique ROTRIB’03 Galaţi Romaniapp 39-47

[6] Barbinta C, Yaldiz S, Dragomir A and Cretu S 2010 An elastic-plastic solver of the wheel-rail
Proceedings of the ASME 2010 10th Biennial Conference on Engineering Systems Design and Analysis

[7] Enblom R and Berg M 2008 Impact of non-elliptic contact modelling in wheel wear simulation \textit{WEAR} \textbf{265} pp 1532–1541

[8] Ayasse B and Chollet Y 2006 Determination of the wheel rail contact patch in semi-Hertzian conditions \textit{Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility} \textbf{43}(3), pp 12-16

[9] Yongming L, Brant S and Sankaran M 2005 Fatigue crack initiation life prediction of railroad wheels \textit{International Journal of Fatigue} \textbf{28} pp 747–756

[10] Ansys 2014 Structural analysis Guide manual

[11] Ansys 2020 [Online]. Available: https://www.ansyshelp.ansys.com. [Accessed 25 February 2020]

[12] Profile UIC ORE S1002 Wheel

[13] BS EN 13674-1:2011+A1: 2017 Railway applications. Track. Rail. Vignole railway rails 46 kg/m and above