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Numerical Analysis of Wheel/Rail Profile Pairings with Elastic and Plastic Material Behavior

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Abstract
To avoid excessive wear, reduce the probability of defects, and consequently reduce maintenance costs for rails and wheels, it is crucial to use rail and wheel profiles that fit together. But, rail configurations differ from country to country and the goal of an integrated European railway area requires trains to travel through different countries. This often makes it impossible, to use optimal rail/wheel pairings. Research has already been conducted to investigate such suboptimal rail/wheel pairings assuming an elastic material behavior. However, real steels will often be plastified during rolling contact. Therefore, this work considers the plastic material behavior of rails. The profile pairings are investigated using full scale explicit finite element simulations of a wheel rolling over a rail. The lateral position of the wheel is varied and all eight combinations of the wheel profiles S1002 and EPS with the rail profiles 60E1 and 60E2 and rail inclinations of 1/20 and 1/40 are investigated. It is shown that simulations using an elastic material model overestimate the contact pressure in most regions by not more than 30% compared to simulations using an elastic-plastic material model.

Keywords: wheel–rail contact, finite element, contact mechanics, wheel profiles, rail profiles, rail inclination, lateral wheel displacement

1 Introduction

The European Union aims to create an interoperable railway area that allows trains to pass through different countries [1]. However, each country develops their own

infrastructure specification with other rail profiles and rail inclinations. On the other hand, train wheels also have different profiles depending on the train operator. This variety of rail and wheel profiles significantly increases the number of possible wheel/rail pairings. Furthermore, even on a straight track, a train moves slightly left and right, such that the lateral wheel position on a rail varies [2].
Inappropriate wheel/rail pairings may cause exhaustive amount of wear or lead to crack like defects. Therefore, it is necessary to assess the contact situation for multiple possible pairings at several lateral wheel positions. Polach [2–4] computed contact quantities for several profile pairings and lateral wheel positions using the tool RSGeo, which computes the contact pressure for a given wheelset assuming an elastic material behavior of the wheel and the rail. Yan and Fischer [5] considered plasticity using a finite element method (FEM) model to investigate the wheel profile UICORE on the rail profile UIC60. They showed that simulations using an elastic material model overestimate the contact pressure compared to simulations with an elastic-plastic material. Meyer et al. [6] investigated the wheel profile S1002 on the rail profile UIC60 using a FEM model. In summary, studies found in the literature either analyze all wheel/rail pairings but neglect plasticity, or account for plasticity but only consider one wheel/rail pairing.

In this work, a full scale 3D explicit finite element (FEM) model is used to compute the contact pressure for eight pairings of wheel profiles (S1002, EPS) [7], rail profiles (60E1, 60E2) [8], and rail inclinations (1/20, 1/40). Furthermore, 13 lateral wheel positions are assumed for each wheel/rail pairing. For the comparison of the elastic and plastic contact pressure, the study is performed twice, once with an elastic material model and once with an elastic-plastic material model for the wheel. Consequently, a total number of 208 simulations have been performed. The FEM model consists of a wheel rolling about 0.7 m over a rail with a velocity of 50 km/h. At the end of each simulation, the maximum contact pressure aggregated over the x-direction is computed. The deviation of the maximum contact pressure between the purely elastic material and the elastic-plastic material is computed.
It was found, that neglecting the plasticity overestimates the maximum contact pressure and underestimates the width of the contact band. Nevertheless, the deviation of the maximum contact pressure was less than 30% in most region and the contact band width using an elastic material behavior was smaller by not more than 2 mm. Furthermore, the position of the contact band does not differ between the elastic study and the elastic-plastic study.

2 Methods

The following section describes the implementation of an automated Python script that imports the wheel and rail profiles as step files and builds an Abaqus explicit FEM model. After the simulation, another Python script extracts the contact pressure and compares the results between the elastic and the elastic-plastic study, as explained in 2.2 Postprocessing.

2.1 FEM model setup

Figure 1 depicts the 3D full scale model and Table 1 lists the corresponding parameters. The wheel rotates around the wheel center along the z-axis and moves down on the y-axis, whereas the rail moves in x-direction. The colors show different components with mesh sizes as follows: green=27 mm yellow=9 mm, orange=3 mm, red=1-3 mm. The red region of the rail becomes finer from the inside to the contact surface. The model is sweep meshed using linear hexahedral elements (Abaqus C3D8R) with reduced integration. Tie constraints connect different components.

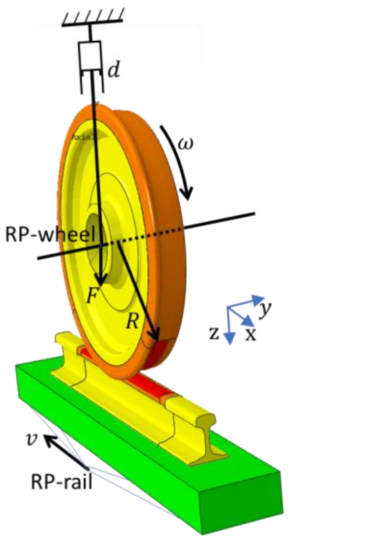

Figure 1: full scale FEM model of a rotating wheel on a moving rail.

Table 1: Parameters of the full scale FEM model

Velocity	v	13.89 m/s (50 km/h)
Wheel radius	R	0.5 m
Angular velocity	ω	27.78 rad/s
Force	F	90 kN
Damping coefficient	d	1000 kNs/m

The rail stands on a table (green) whose lower surface is connected using a kinematic coupling to a reference point (RP-rail). The table has a length of 2 m, a width of 0.3 m, and a height of 0.14 m. The rail has a length of 1.49 m. The orange region starts 0.03 m below the top of the rail and has a length of 0.944 m. The red region is 0.01 m below the surface and has a length of 0.844 m. Contact only occurs at the red region over a rolling length of 0.695 m.

The second reference point (RP-wheel) is connected to the inner wheel surface using a kinematic coupling. Furthermore, a damper with coefficient d on the z-axis reduces vertical oscillations. The wheel has a radius R, which is measured 70 mm inside of the flange. At a radius of 0.47 m the orange region starts. The red region extends from the wheel surface 0.01 m inside.The lateral wheel position u_{wheel} is the distance from the rail center to a point on the wheel 70 mm inside of the wheel flange.Figure 2 shows how the wheel is placed with a lateral wheel position u_{wheel} that is the distance from the rail center to the point on the wheel where the wheel radius is measured. The lateral wheel position u_{wheel} is held constant in a simulation. For every u_{wheel}, a new simulation is performed.

The load is applied as concentrated force F on RP-wheel. The angular velocity ω of the wheel is applied as initial condition on the whole wheel and as a boundary condition on the wheel center. The reference point RP-wheel can only rotate along the z-axis und move along the y-axis. All other degrees of freedom are constrained. The rail moves with a velocity v in x-direction. This is implemented as initial velocity condition on the whole rail and as boundary condition on the reference point RP-rail that can only move along the x-axis.

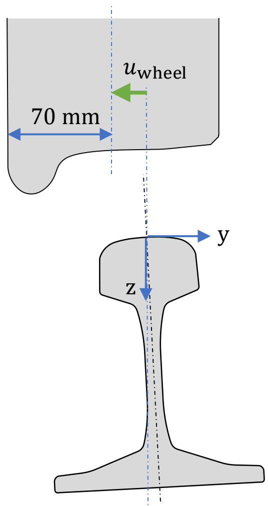

Figure 2: The lateral wheel position u_{wheel} is the distance from the rail center to a point on the wheel 70 mm inside of the wheel flange.

Surface to surface contact is defined between the red surface on the rail (main surface) and the red surface on the wheel (secondary surface). The contact uses an Abaqus hard penalty contact as normal behavior. The friction coefficient is $\mu = 0.3$.
All components except the red region of the rail are elastic with a Young's modulus $E = 206000$ MPa and a poisson's ratio $\nu = 0.3$. For the plastic study, the Chaboche elastic-plastic "R260" material model presented by Schnalzger et al. [9] is assigned to the red region of the rail. Furthermore, a density $\rho = 7800\,\text{kg/m}^3$ has to be defined for the explicit model. The density of small elements is increased at the beginning of the simulation using mass scaling to reach a stable time increment of at least 10^{-7} s.

2.2 Postprocessing

For each wheel/rail pairing and 13 lateral wheel position from -6 mm to 6 mm a model is built and simulated. In the next step, the contact pressure p of the last frame is extracted, as shown in Figure 3. The maximum contact pressure p_{max} is computed by taking the maximum along the x-axis. The contact band diagram plots p_{max} over y and u_{wheel}. Contact band diagrams are generated for both the elastic study and the elastic-plastic study.

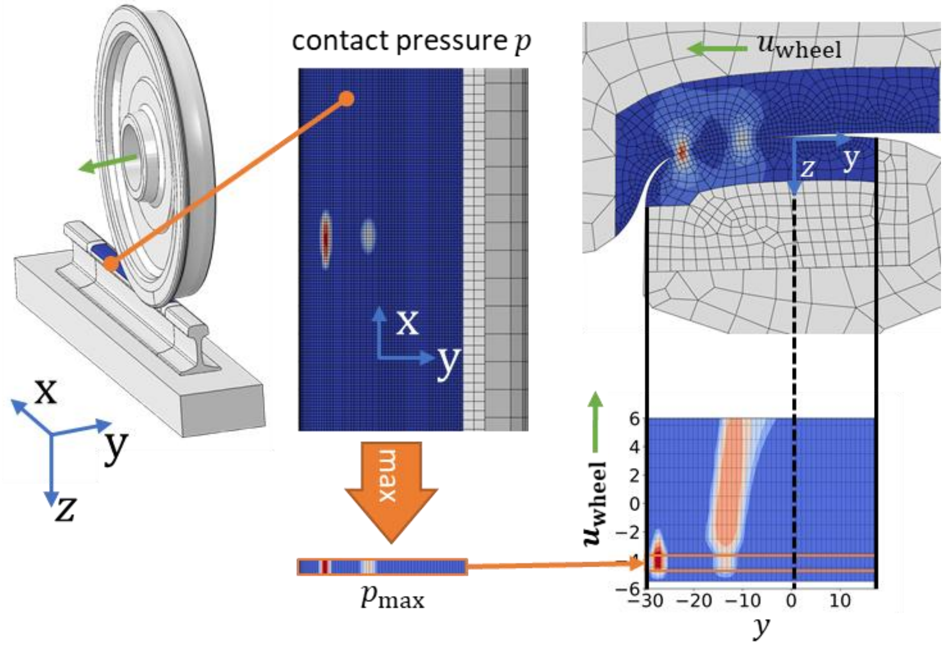

Figure 3: The contact pressure is extracted at the end of the rollover simulation. The maximum contact pressure p_{max} along the x-axis is computed. The contact band diagram plots p_{max} for various lateral wheel positions u_{wheel}.

3 Results

Simulations were performed on a workstation with AMD EPYC 9454 CPUs. Each model contained approximately 820000 elements and was simulated on 20 CPUs using the commercial FEM software Abaqus. Simulations using an elastic-plastic material model took between 4.5 and 5.5 hours. Simulations that considered only elasticity ran faster taking between 3.5 to 5 hours.

Figure 4 and Figure 5 show the contact band diagrams for all wheel/rail pairings. Figure 4 contains the pairings with a rail inclination of 1/20 and Figure 5 shows the simulation results with a rail inclination of 1/40. The name of each wheel/rail pairing can be found on the right side of the corresponding light-orange row. The first two columns represent the elastic-plastic and elastic studies, respectively. The grid corresponds to the mesh size. Consequently, the distance between two vertical grid lines is 1 mm on the rail. The rightmost column shows the comparison of the highest p_{max} over u_{wheel} of the elastic-plastic study from the first column (blue) and the elastic study from the second column (orange).

The following observation can be made:

- The shape of the contact band is similar for both the elastic and elastic-plastic studies. The width of the band varies by up to two elements, corresponding to 2 mm on the rail.

- There might be more than one contact point. Consequently, assuming a single elliptical contact is an oversimplification. This has already been shown by Knothe et al. [10].

- The elastic contact pressure is generally higher than the elastic-plastic contact pressure. However, the deviation is less than 30% except at a lateral wheel position of about $u_{\mathrm{wheel}} \approx -5$ mm for the pairings “EPS 60E1 1/40” and “S1002 60E1 1/40”. There, the maximum deviation is 54%. Furthermore, the elastic contact pressure shows the same trend as the elastic-plastic contact pressure.

- The contact band of the EPS wheel profile significantly deviates from the contact band of the S1002 wheel profile. For the S1002 profile, the contact pressure is higher and more concentrated to a smaller y-range on the 1/20 rail compared to the EPS profile. This effect is reversed on the 1/40 rail, where the EPS profile leads to a thinner contact band with higher contact pressures. Polach [2] introduced a contact concentration index to quantify this effect.

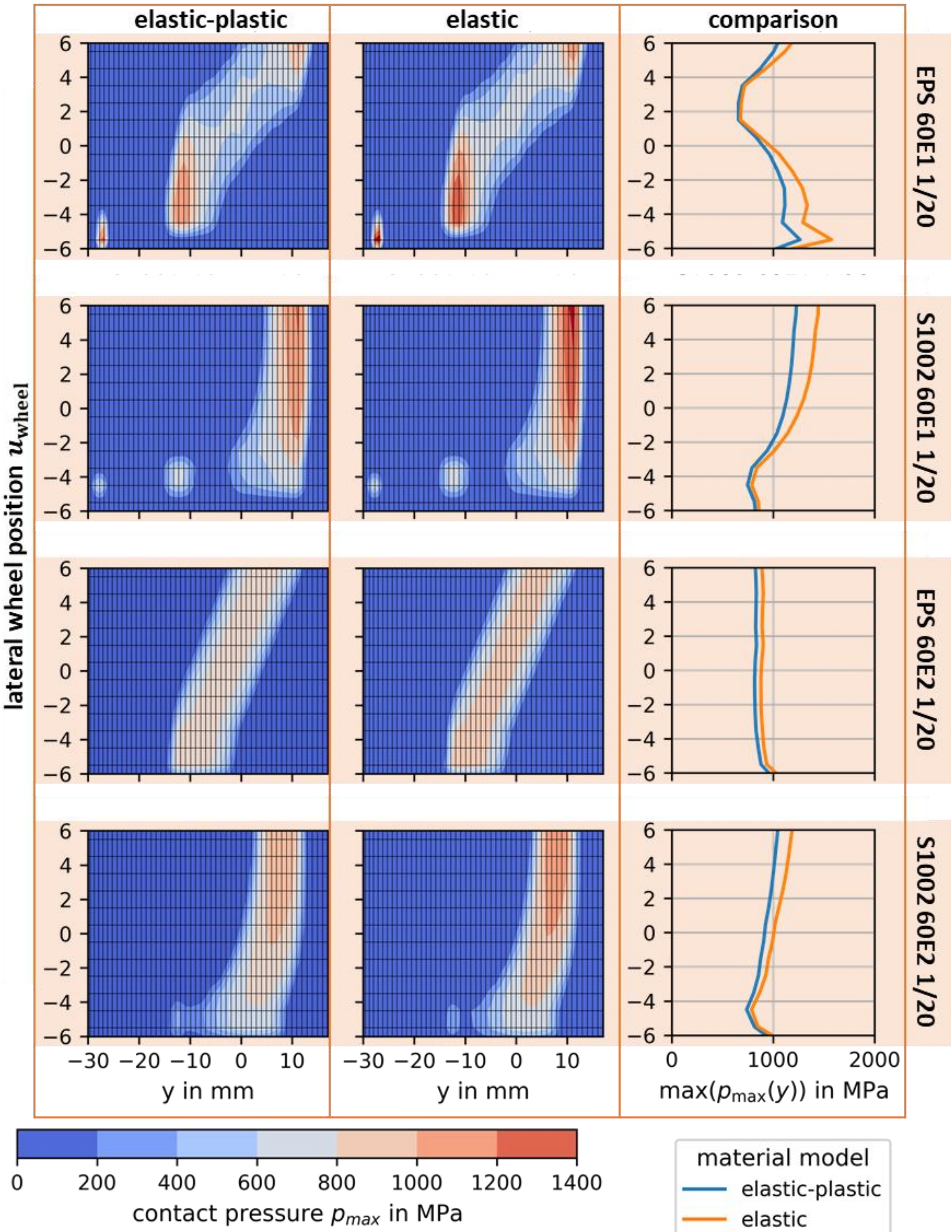

Figure 4: Contact band plots for various wheel/rail pairings with a rail inclination of 1/20. The left column considers plasticity, the middle column only elasticity. The right column plots the maximum p_{max} over y.

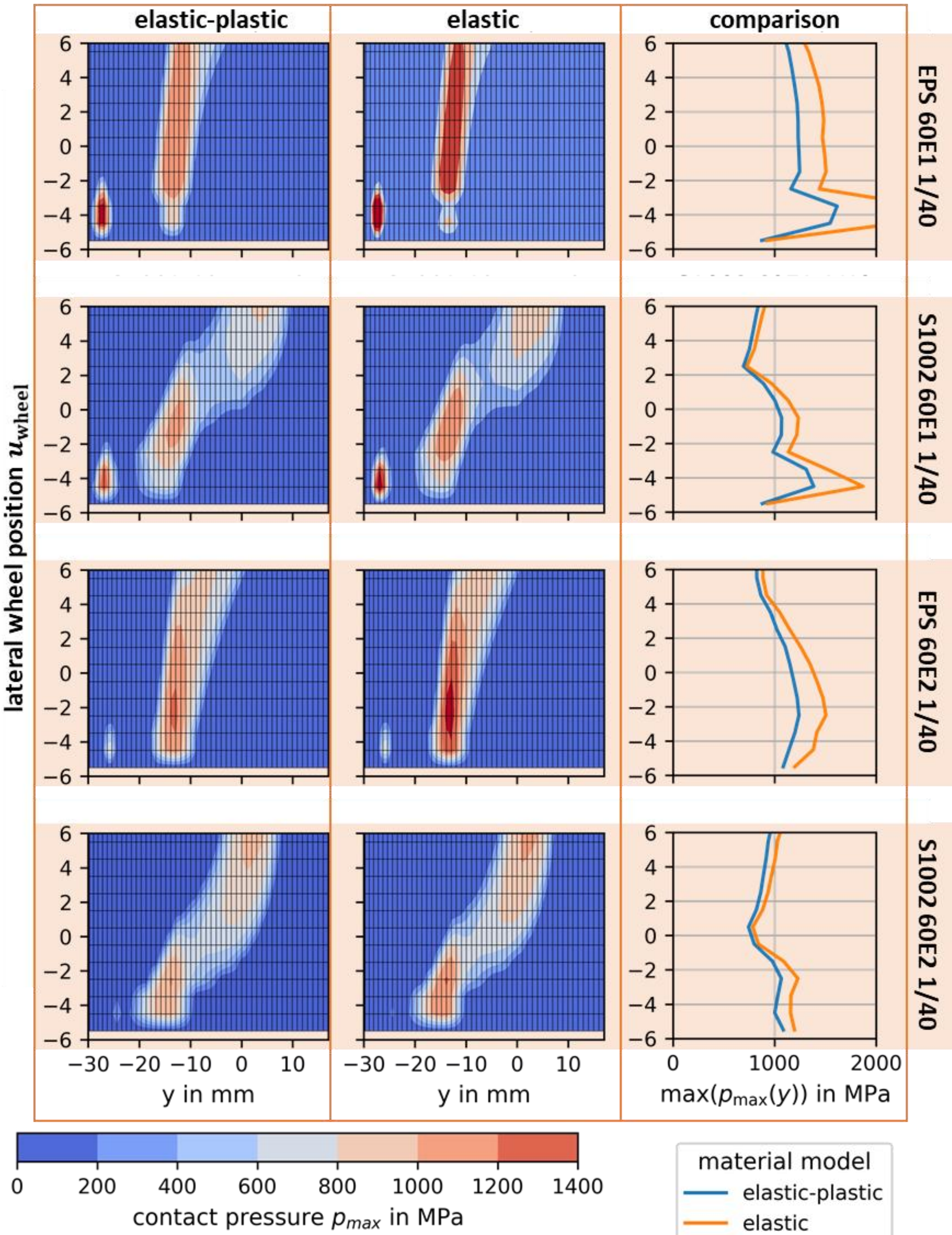

Figure 5: Contact band plots for various wheel/rail pairings with a rail inclination of 1/40. The left column considers plasticity, the middle column only elasticity. The right column plots the maximum p_{max} over y.

4 Conclusions and Contributions

A full scale explicit 3D FEM model of a rollover simulation was presented. The model was used to investigate the contact pressure for lateral wheel positions from -6 mm to 6 mm for all combinations of the wheel profiles S1002 and EPS with the rail profiles 60E1 and 60E2 and with the rail inclinations 1/20 and 1/40. This was computed once using an elastic material model (elastic study) and once a plastic material model for the rail (elastic-plastic study). The authors showed, that the shape of the contact band does not vary significantly between the elastic and elastic-plastic study. Furthermore, it was found that the max. contact pressure is overestimated by not more than 54%, if plasticity is neglected. In most regions the deviation was less than 30%. These findings allow to proceed with elastic materials if the shape of the contact band or the general trend of the contact pressure is of interest.

The study assumes a wheel on a straight track. For future work, the authors suggest considering an angle of attack of the wheel as it occurs in curves and a rotation of the wheel around the driving-direction, which might happen if one of the two wheels of a wheelset moves on its flange root and is hereby lifted.

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