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The Influence of Wheel Parameters on the Contact Forces in the Switch Panel

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Abstract

A 3D explicit FE model is utilised to analyse the contact situation in the switch panel. Especially, the diverging switch rail is subjected to high lateral contact forces, which leads to a degradation of this component. The location of the impact point at the switch rail is a decisive parameter which is highly affected by the turnout geometry and the passing vehicle. This work evaluates the influence of several wheel parameters, namely the wheel profile, wheel yaw angle and lateral wheel-set displacement on the initial impact position and the magnitudes of the lateral contact force. The analysed turnout geometry is a standard 60E1-500-1:12 turnout with a singe S1002 wheel rolling over it. It is shown that the critical region where initial impacts at the switch rail are likely to occur is located between 1.5 and 3 m from turnout entry for this turnout geometry.

Keywords: Switches and Crossings, FE modelling, Rolling-sliding contact, Switch rails.

1 Introduction

Switch rails are essential parts of railway turnouts and are exposed to significant contact loads which can lead to rail deterioration and break outs. The damage of switch rails is associated with considerable safety concerns and maintenance costs for the track operators [1].

In literature there are numerous investigations covering the simulation of turnout passages by means of multibody dynamics (MBD) simulations [2]. Recently, multidisciplinary approaches have been introduced using MBD in combination with sub-models for the discretisation of normal and tangential contact such as FASTSIM [3] or with 2D finite element (FE) simulations [4] to account for plastic deformation in a specific cross section. There are also approaches using explicit full-scale FE-models [5] with the drawback that they are usually computationally expensive.

The 3D-FE modelling approach provides the possibility to evaluate contact loads, stress distributions using advanced material models. It enables multiple contacts and this can be used for the contact between the switch and the stock-rail. The relative movement between switch rail and stock rail could facilitate crack initiation leading to rail break outs, which can be observed in Figure 1. Break outs are highly critical for the operational component safety and thus need to be analysed more in detail.

Figure 1: Heavily worn switch rail with characteristic break outs.

The analysis presented is conducted by means of a 3D half-track finite element model of a standard 60E1-500-1:12 turnout. The results obtained from this analysis provides insight about the influence of various wheel profiles and alternating initial wheel positions on the profile degradation.

The wheel positions can vary due to the hunting oscillation of the wheel-set and have a major influence on the initial impact position and the contact forces at the switch rail, which is schematically shown in Figure 2. Track irregularities and other imperfections can disturb this oscillation and lead to unfavourable combination of high lateral displacements and yaw angles [6].

Three different initial lateral wheel-set positions (0mm, -5mm, -10mm) and three yaw angles (0.0°, 0.3°, 0.6°) are analysed. These scenarios are considered the more critical ones as they translate the point of initial contact towards the switch rail tip. The nominal case corresponds to a passage with a nominal wheel profile and an ideal wheel-set position with 0mm and if not stated otherwise.

2 Methods

The developed FE model is utilised for the evaluation of the overall contact situation in terms of initial impact points, transition zones and contact forces in the switch panel. A half-track model is used considering a single S1002 wheel rolling over the switch rail and the stock rail in the diverging route. All parts are modelled with a linear elastic material model with a Young's modulus E = 206 GPa and a Poisson's ratio ν=0.3. The general contact formulation is used to model the contact between wheel, switch rail, stock rail and between the two rails with a coefficient of friction μ of 0.35 for all contact definitions. The discrete mass-spring-damper track model is defined by the lateral (k_{yt}, c_{yt}) and vertical (k_{yt}, c_{yt}) stiffness/damping coefficients and the corresponding sleeper mass ($m_{sleeper}$) is added. The vertical (k_{yv}, c_{yv}) and lateral (k_{zv}, c_{zv}) primary suspension of the bogie is included, while the secondary suspension is neglected. The vehicle and track parameters are adopted from [7, 8]. A static wheel load (F_z) of 125 kN is acting on the centre point of the wheel and the rotational velocity (v_r) around the wheel axes is prescribed. The yaw rotation around the vertical wheel axis is enabled to allow the wheel to be guided by the switch rail. The corresponding masses and inertias of the bogie ($m_{vehicle\_lat}$, I_{zz_bogie}) and the wheel-set ($m_{wheel\text{-}set}$, $I_{yy\_wheel\text{-}set}$, $I_{zz\_wheel\text{-}set}$) are applied to provide a characteristic response of the wheel after the impact.

The contact forces are compared to results obtained from an MBD simulation performed with NUCARS in order to validate the FE model. The vehicle and contact parameters are adopted from [8]. Figure 3a and b show the vertical and lateral contact forces for a passage in the facing diverging direction on the outer wheel obtained from the FE and the MBD model.

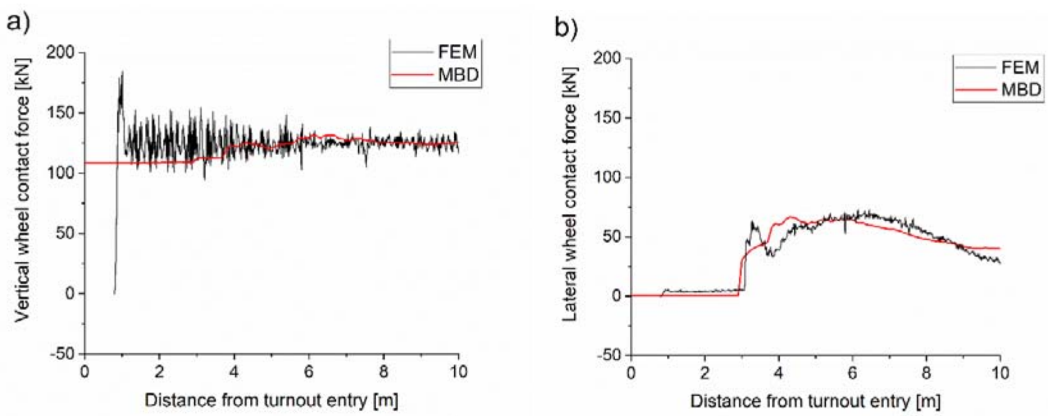

Figure 3: Comparison of a) vertical and b) lateral wheel contact force for a passage in facing diverging direction evaluated at the outer (left wheel) obtained from FE and MBD simulations.

The contact forces are generally in good agreement which shows that the half-track FE model is able to reproduce a characteristic loading of the switch rail. A summary of the utilised system parameters can be found in Table 1.

Parameter	***Vehicle***	***Track***
F_z [kN]	125	
v_t [km/h]	70	
V_r [rad/s]	38.88	
μ [-]	0.35	
k_{zv} [MN/m]	1.2	
c_{zv} [kNs/m]	30	
k_{yv} [MN/m]	20	
c_{yv} [kNs/m]	2	
$m_{wheel\text{-}set}$ [kg]	1090	
$m_{vehicle\_lat}$ [kg]	11000	
$I_{yy\ wheel\text{-}set}$ [kgm²]	135.5	
$I_{zz\ wheel\text{-}set}$ [kgm²]	778	
$I_{zz\ bogie}$[kgm²]	8500	
k_{zt} [MN/m]		60
c_{zt} [kNs/m]		535
k_{yt} [MN/m]		30
c_{yt} [kNs/m]		267.5
$m_{sleeper}$ [kg]		300
Δt_{crit} [s]	$1.2 * 10^{-6}$	

Table 1: Vehicle and track parameters utilised for the FE models

3 Results

Figure 4 shows the vertical and lateral contact forces on the wheel and both rails, respectively. The preliminarily evaluation of contact forces for a nominal wheel profile and a nominal initial wheel-set position with $\Delta y = 0$ mm and α=0° shows that during the first 3 m from turnout entry the entire load is carried by the stock rail. The initial point of contact between wheel and switch rail is located at 3.1 m from turnout entry. From 3.1 – 6 m a two-point contact region can be observed, which means the wheel contacts on switch rail and stock rail simultaneously. In this region the vertical contact force is distributed between the two rails. However, it can be observed that the lateral forces are mainly carried by the switch rail. Furthermore, the lateral contact force on the switch rail, even with this nominal set up, yields up to 75 kN which can cause significant damage.

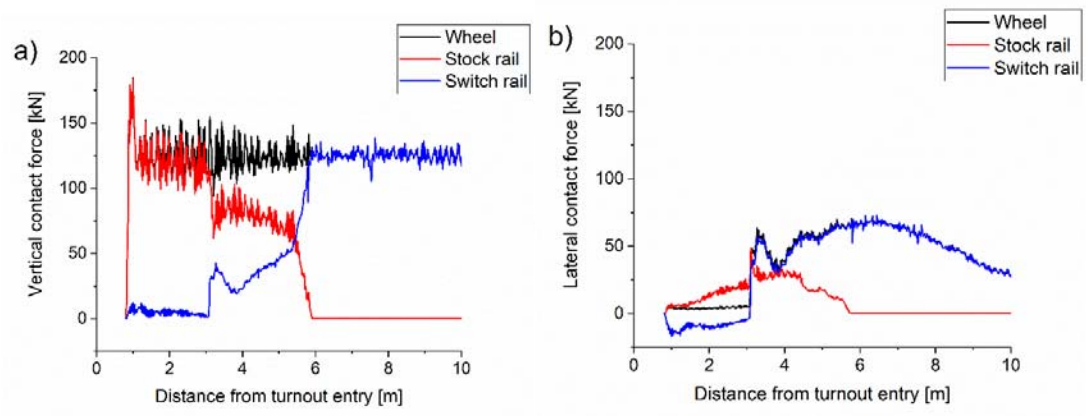

Figure 4: a) Vertical and b) lateral contact forces on wheel, stock rail and switch rail.

The parameter studies analyse the evolving contact forces and the area of the initial point of contact between wheel flange and switch rail. Three different wheel profiles are investigated (nominal S1002 profile, worn and hollow worn profile), see Figure 5. Furthermore, the initial wheel-set positions, regarding lateral displacement (Δy) and yaw angle (α), are varied and analysed.

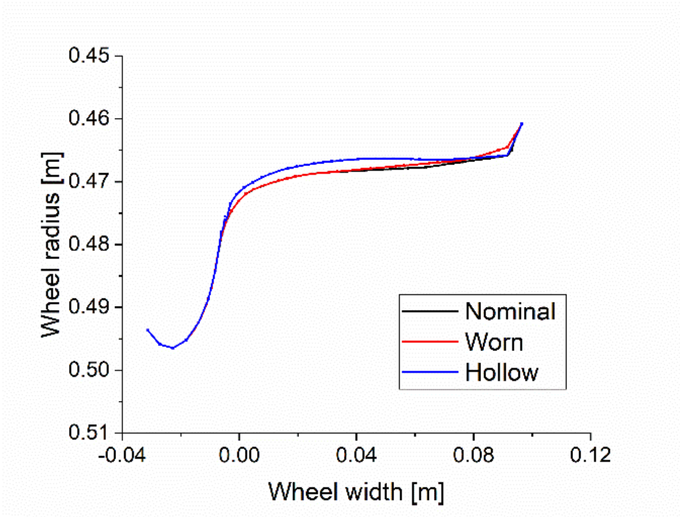

Figure 5: Wheel profiles used for the conducted parameter study.

The results in Figure 6 show that the analysed parameters affect the location of the initial impact point and the lateral contact force magnitude considerably. The hollow wheel profile contacts the switch rail earlier than a nominal and a worn profile. The lateral wheel-set displacement variation showed that a Δy of -10 mm can translate the initial point of contact up to 1 m closer to the beginning of the turnout. Although the contact force magnitude is slightly decreased this translation of the initial impact point is still critical as the switch rail cross section is significantly weaker in this region. An initial wheel yaw angle of α=0.6° shifts the initial point of contact up to 1 m closer to the beginning of the turnout with a simultaneous increase of the contact force magnitude.

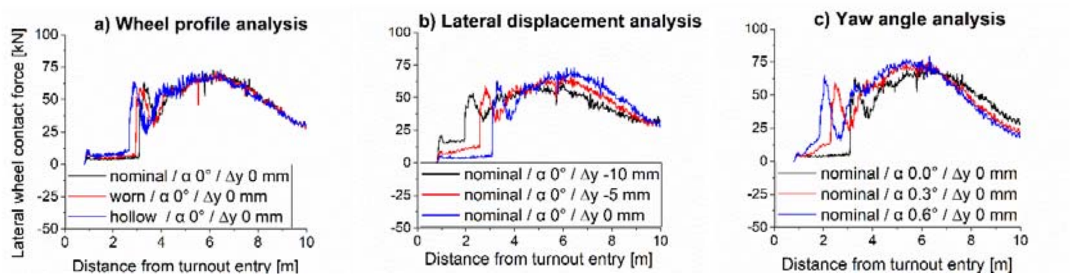

Figure 6: Lateral contact force over distance from turnout entry for a) three different wheel profiles, b) three initial lateral wheel-set positions, and c) three initial wheel yaw angles.

4 Conclusions and Contributions

The presented FE model is utilised to examine the general contact situation regarding multi point contact in the switch panel. Furthermore, the global contact forces acting on the switch rail are analysed for different wheel profiles, initial wheel set displacements and yaw angles. It is shown that the evaluated parameters significantly affect the location of initial contact between wheel flange and switch rail and the contact force magnitude.

- The parameter study shows that the hollow wheel profile contacts the switch rail earlier than the worn wheel profile
- The lateral wheel-set displacement can shift the initial impact point up to 1 m compared to a wheel set aligned in track centre
- A positive initial wheel yaw angle aggravates the initial impact on the switch rail significantly as it contacts earlier with increased contact force magnitudes
- For this turnout geometry the area with high lateral contact loading is located between 1.5-3 m from turnout beginning.

The results of the parameter study show that the initial impact point at the switch rail can vary considerably depending on the analysed wheel parameters. Furthermore, the investigated parameters have a significant influence on the lateral contact forces in the switch panel and thus can increase or decrease the S&C component deterioration considerably. It must be considered that in real loading cases the analysed parameters interact and vary simultaneously, so it is likely that inappropriate wheel profiles coincide with a high lateral displacement ($\Delta y < 0$) or an unfavourable yaw angle ($\alpha > 0$). This worst-case scenario can increase the damage even further, which means that a relatively low number of passing wheels can cause the majority of damage. It is recommendable to investigate and identify these unfavourable cases in order to facilitate a reduction of the contact forces in the diverging route. The lifetime of switch rails can be generally extended by a reduction of the high lateral contact forces in early regions of the switch panel. Other possible measures are (1) to reduce the amount of unfavourable wheel profiles passing the switch and to improve the steerability of the bogies to provide an enhanced alignment in track during the turnout passage.

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