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INVESTIGATION OF STRESS LEVEL IN WHEEL-RAIL CONTACT

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Abstract: *Initial calculation of the wheel and rail contact problem was solved with analytical theories. These include some assumptions during computations. A finite element analysis was performed to compare results of the analytical tools because of the assumptions. In addition, various finite element models have been developed by researchers to examine material response during wheel-rail contact. Damage of the rail surface and its effects on the contact forces are a common research area in numerical computations of wheel-rail contact. Maximum stress level of the model is a key factor of the permanent deformations. Neutral position of the wheelset is commonly selected in the studies. In this study, a wheel-rail contact tool is developed to investigate effects of cant angle and lateral position of wheel on maximum stress level that occurs in the rail part.*

Keywords: Wheel-rail contact, Contact stress, Lateral shift, Cant angle

1. Introduction

Wheel and rail contact is commonly researched for determining the pressure distributions on the contact patch. When the wheel comes into contact with rail, different contact geometry may occur. In the standards, lateral radius of the rail surface has a constant curvature in the central part. In the case of cylindrical wheel shape, the profile of the wheel brings about an elliptical contact shape. In contrast, if the profile of wheel is curvilinear, non-elliptical contact area could be observed at the contact interface. For this reason, contact pressure values are changed when the wheel is supposed to shift laterally.

Telliskivi and Olofsson developed a finite element method based tool for wheel and rail contact. The analysis includes two different cases which are rail gauge corner contact and rail head contact. Maximum contact pressure and values of the total contact area are compared with the results of the Hertz contact theory and Contact software. In the first case, there is significant difference between the Finite element method and validation methods (Telliskivi & Olofsson, 2001).

Yan and Fischer analyzed the wheel and rail contact conditions with FEM. UIC60 rail and UIC-ORE wheel profiles were implemented in the research. Four different values of lateral wheel displacement are considered. Contact pressure distributions of the positions are given for linear elastic, elastic-plastic material models and Hertz contact theory. The results show that if the curvature of the rail surface is constant within the contact area, numerical calculations including elastic material properties are compatible with those of Hertz contact theory (Yan & Fischer, 2000).

The main idea of this study is to determine maximum Von-Mises stress values for various lateral wheelset positions. Also, since rails are normally mounted on the sleepers with an inclination (cant angle) towards the track vertical axis, the effect of rail cant is observed. Results of the numeric simulations are expected to serve as a guide for researchers dealing with analysis of elastic-plastic wheel and rail contact.

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2. Methods

Finite element simulations are performed via ABAQUS™ (Systèmes, 2013). Theoretical S1002 wheel profile (UIC, 2004) with 920 mm of diameter and UIC 60 rail (EN, 2011) profile with 200 mm of length are chosen for the numeric model. In addition, in order to assess effect of cant angle, two different wheel and rail contact assemblies are designed. One of them is characterized by a 1/40 canted rail whereas in the other one, the rail axis is vertical. The wheel is represented by a rim segment (without the disc or hub), but full rail geometry is utilized. Linear elastic material properties are considered in both of studies in which E = 210 GPa, v = 0.3 (Deng et al., 2015).

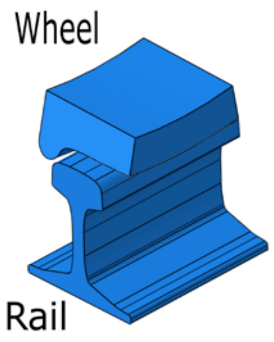

Fig. 1: Illustration of the wheel and rail contact model.

A vertical wheel load of 90 kN is applied at the center of the wheel. Transmission of the force from the reference point to the inner surface of the wheel body is set – constraint definition is used to connect the reference point with the surface of the wheel (Systèmes, 2013). All parts of the assembly are meshed with C3D8R type solid element whose edge size is 0.75 mm. Characteristic element size of the wheel and rail are 2 mm and 3.5 mm, respectively. Only measurement zones have finer FE discretization, so the assembly does not have uniform mesh structure. The FE models are presented in *Fig. 2*.

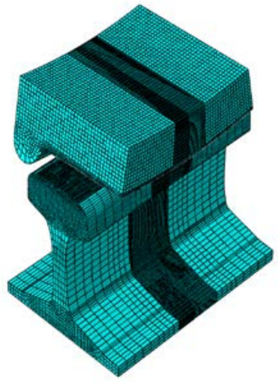

Fig. 2: 3-D meshed wheel and rail contact model.

Contact definition between the wheel and rail surface is defined as surface-to-surface (Systèmes, 2013). Contact definition includes only normal contact. Coefficient of friction is not defined as it does not play a role in the transmission of normal forces. During simulations, nine different lateral shift values (mm) are applied to the wheel part of the geometry. Maximum stress values are obtained in the rail part for each shifted position of wheel. Illustration of positions of the wheel are given in *Fig. 3*.

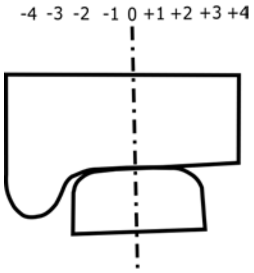

Fig. 3: Lateral wheelset positions [mm], positive values correspond to the wheel displacement away from the track centerline. .

3. Results

Figures 4 and 5 show maximum stress levels with respect to lateral positions of wheel. Two different inclination angles are examined in the study. The results present maximum equivalent stress in the rail part.

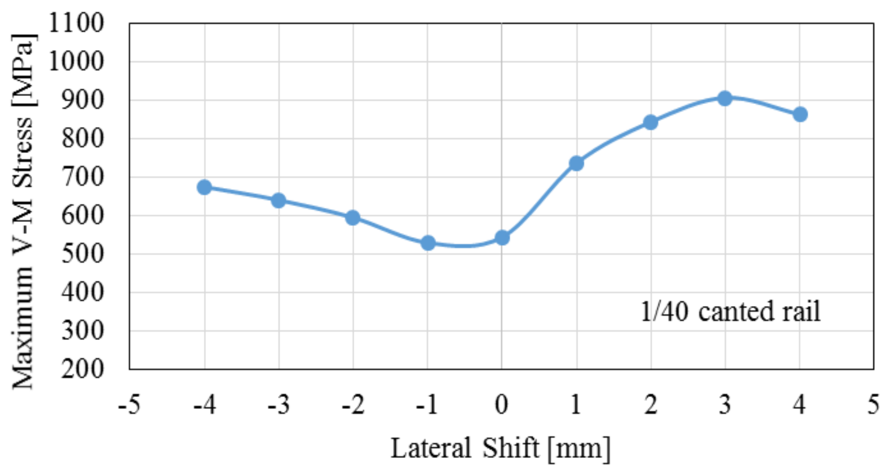

Fig. 4: Maximum Von-Mises stress levels according to lateral positions of the wheel for canted rail

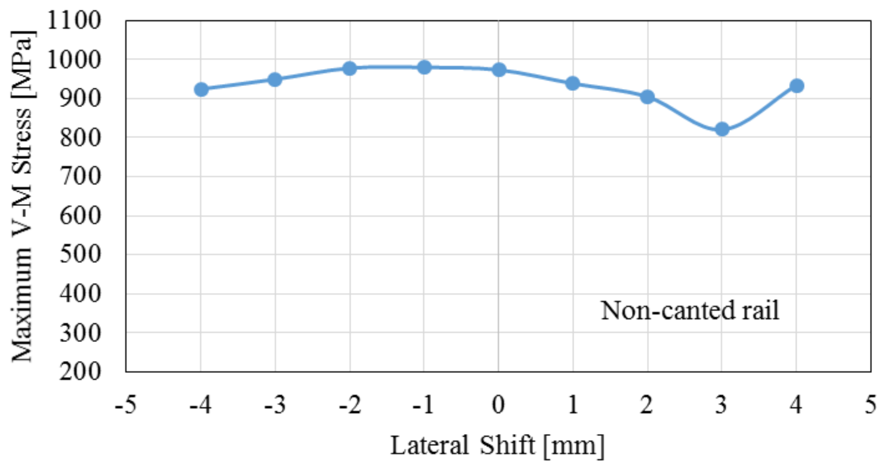

Fig. 5: Maximum Von-Mises stress levels according to lateral positions of the wheel for non-canted rail

For the canted rail, the maximum stress is observed in the +3 mm lateral shift position. In the positive side, stress levels are close to each other. However, when the wheel moves to negative direction, maximum stress in rail part decreases. Minimum values are seen between the zero and –1 mm positions, where the well-known non-Hertzian contact occurs, with the loading distributed in larger area of contact.

In *Fig. 5*, results of the non-canted rail are shown. The curve is different from that for the canted rail (*Fig. 4*) – particularly, the overall values are higher and there are not the peak levels which are seen in *Fig 5*. The results are close to each other between zero and –2 mm lateral positions of the wheel. Additionally, there is not big variation between 1 mm and –3 mm displacement values. Also, minimum stress emerges in the +3 mm position.

The effect of the inclination angle is clearly realized from *Fig. 4* and *Fig. 5*. Maximum stress has stable values for the non-canted rail, but canted rail does not have similar trend. The maximum stress peak values for canted and non-canted rail are close to each other.

4. Conclusions

Researchers focusing on material response of the wheel and rail contact can decide their working conditions according to *Fig. 4* and *Fig. 5*. Maximum equivalent stress level is an effective parameter for the plastic deformation, if the study in question considers elastic-plastic material behaviour. The results give an idea for deciding about lateral positions to analysis of plastic deformation, fatigue etc.

For the 1/40 rail inclination angle, working on the positive direction is better than the negative direction, because maximum stress levels are positioned in positive directions. As a result of that, more plastic deformation could be obtained in the positive direction. In addition, +3 mm lateral position of the wheel has maximum stress levels, so this should be considered.

For the non-canted rail, maximum peak does not appear clearly in *Fig. 5*. However, maximum level is located between zero and –2 mm shifted positions. The minimum stress level is shown in the +3 mm lateral position. Minimum plastic deformation is expected to be in this location. There is neither a significant increasing nor decreasing trend for the maximum stress distribution for the positive and negative directions.

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