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COUPLED METHOD TO INVESTIGATE PLASTIFICATION OF HEAVY HAUL RAILWAY WHEELS

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ABSTRACT

This work compares two methodologies to investigate plastic strain accumulation on a heavy haul railway wheel in pure rolling. Firstly, a finite element model was employed to simulate the wheel-rail interaction, producing plastic accumulation on a region of wheel tread after six passes. Then, a load distribution with elliptical shape estimated by analytical Hertz Theory was applied as distributed nodal forces on the wheel tread with a FE model. The simulations show the significant gain in computational cost with the semi analytical modeling because of the mesh reduction and the kind of element interactions of rail and wheel during the rolling in the first case. This allows for faster analysis of more complexes problems with this coupled approach.

Keywords: Finite Element Model, wheel-rail interaction, plastic accumulation, Hertz theory.

INTRODUCTION

The exchange or machining of railway wheels are among the main expenses in heavy haul railway transport. The analyzes of the internal stresses on the wheel during rolling can assist designers in identifying the optimum operating conditions and designs to avoid or at least to control rolling contact fatigue - RCF (Daves, 2016). Some authors report analytical studies comparing codes such as FASTSIM® and CONTACT® (Kalker, 1982) with Hertz's analytical theory, linear and non-linear, to validate methodologies to estimate the distribution of stresses in the wheel-rail contact region (Tao *et al.*, 2016). These studies investigate mainly the forces in the contact region, but do not allow the analysis of the wheel interior, through the calculation of the stress and strain distribution.

Investigations of wheel defects can be conducted using numerical simulation, but the precision of the results depends on the assumptions for the model (Gerlici and Lack, 2010). Some authors present simulations that couple Hertz's Analytical Theory and the Finite Element Method in order to develop more advanced and faster models for this investigation (Cuperus and Venter, 2017). These applications are commonly found in railway rails studies, as proposed by Srivastava (2017), but not in railway wheels. The latter is usually done with numerical simulations based on the resolution of the full contact problem, i.e. with the wheel and rail.

On the investigation conducted with numerical analysis, the adequate representation of the real phenomenon will directly reflect on the results. Naemi *et al.* (2018) presents an extensive and well discussed work, with different numerical models in finite element analysis of wheel/rail interaction - elastic, plastic, elastoplastic, and thermoelastoplastic approaches.

However, the authors work with a flat wheel geometry, which differs from the reality of railway wheels, which are conical. Wu *et al*. (2017) also conducted a 3-D analysis of thermal-mechanical behavior of wheel/rail sliding contact, considering the temperature variation of contact components. The results were developed in a wheel with a flat shape, simplifying the analysis.

Other authors also perform internal analyzes of crack propagation in simplified models, using two-dimensional meshes (Kracalik, 2016; Trollé, 2014). In these cases, some influence factors are omitted, such as the hoop stress from manufacturing process that cannot be inserted into the simplified model and will strongly interferes in the results. Moreover, the Shakedown formation cannot be analyzed with these two-dimensional simulations. This phenomenon is responsible for the variations in the stresses and internal deformations of the wheel after plastic strain accumulation that happens during the rolling process (Williams, 2005).

The present work investigated the plastic accumulation until the stabilization of the deformations and the formation of elastic shakedown through two different methodologies. The first one considers the pure rolling of wheel on a rail employing a three-dimensional elastoplastic model and the finite element method with Ansys® 18.0. The second one considers only the three-dimensional finite element model of the wheel, with the application of nodal forces in the wheel tread. These forces were determined by the contact pressure distribution estimated by Hertz theory and discretized for each element of the surface using a special developed MatLab® code.

NUMERICAL PROCEDURE

The numerical procedures adopted in this work are related with two different simulations: firstly, a method that is commonly found in the literature, consisting in a wheel/rail interaction with resolution of contact problem by FE analysis; furthermore, a second simplified coupled method using finite element analysis and Hertz's theory.

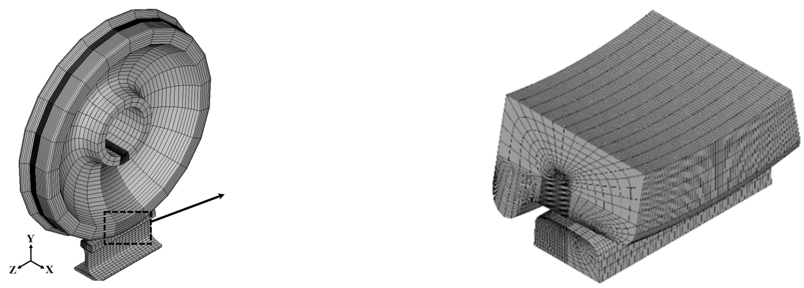
The study was developed with an AAR C-38-wheel geometry and the mesh used has 1 mm-thickness on the contact region, for both simulations. A hexahedral element type Solid185 was used; it is defined by eight nodes having three degrees of freedom at each node, which are the translations in the nodal x, y, and z directions. The material is considered to behave elastoplastically, with the mechanical properties given in Table 1. To simplify the analysis, dynamics effects are not considered. However, a correction value of vertical load was assumed to compensate this simplification as follows: considering a normal load of 16 Ton for an ore wagon, an increase of approximately 20% in this value is assumed to account for different possible oscillations; that will increment the vertical load to 19 Ton. A total of six rolling passes were performed in each approach. This number of passes is enough to lead to plastic stabilization. Only a wheel portion around the wheel/rail contact position was employed in the solid model in the contact position - the submodel. A multilinear kinematic hardening approach was adopted to describe the material behavior in both models.

Table 1 - Mechanical Properties

Property	Unity	Value
Young Modulus	E [MPa]	210
Yield Stress	σ_e [MPa]	750
Ultimate Tensile Strength	σ_u [MPa]	1220
Poisso's ratio	ν [-]	0.3

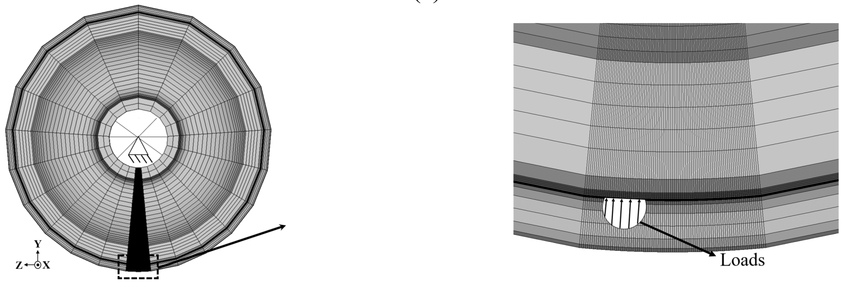
The 3D submodel of the wheel and rail is presented in Figure 1(a). It was developed to simulate the rolling movement of the wheel on the rail. This procedure demands significant computational cost because of the number of elements required to describe the contact interaction, however it is faster than using the whole wheel. Besides, thinner meshes are necessary to allow for the elastoplastic behavior. Initially, the simulation of rolling was made with a non-refined 3D full model; then the displacement results are extrapolated to the submodel boundaries. The reason is to obtain the displacements far from the contact region. With that, the effect of the boundaries will be considered in the stresses in the contact region. Hence, the whole wheel model can be simplified to a piece of the whole wheel and only the submodel mesh is refined, aiming to improve the results faster calculation. Augmented Lagrange formulation was used for contact interaction in the submodel. For the surfaces interaction, Targe170 (rail head) and Conta173 (wheel tread) contact elements were used, considering pure rolling, with a friction coefficient of 0.3 to resist the expansion in the contact.

Another method is proposed here, hereby denominate coupled method. It consists in the use of a 3D finite element model of a railway wheel, as indicate in Figure 1(b), and the Hertz's analytical theory. The analytical theory is used to calculate the contact pressure distribution and the contact area between wheel and rail. This pressure distribution is applied on the wheel tread surface as nodal forces in the contact area. The magnitudes of the forces are previously calculated using a program in Matlab® Software. The contact forces distribution moves on wheel tread surface, simulating the effect of rolling. Previously, the wheel tread was mapped to know what is the identification number of the nodes which will receives the distribution of loads at every step during each pass. Thus, an elliptical shape pattern representing the distributed load moves from one position to the next, step by step, of each time during the rolling process. An extensive data file is generated and then it is read by Ansys APDL interface, generating a dynamic application of loads on the wheel as a rolling movement, by steps of static analyses.



Full model Submodel

(a)



Full model Coupled model

(b)

Fig. 1 - 3D mesh submodel of pure rolling (a) and coupled method (b)

RESULTS

Figure 2 presents the Von misses stress after six passes, when the plastic stabilization was obtained. Pure rolling in 3D submodel results are shown in Figure 2 (a) and (b) and the coupled method results in Figure 2 (c) and (d). It is possible to observe the similarity between the results, which clearly shows that the simplification through the application of nodal loads can be used. For the pure rolling case, the stress field in the wheel and rail is similar, since the mechanical properties of wheel and rail were considered the same.

The maximum stress obtained in the wheel for each simulation is higher for the coupled method. The maximum stress is 810 MPa, against 790 with the submodel. This result can be understood by the fact that the applied nodal force is more concentrated in the coupled method than in the pure rolling, where the rail deforms plastically and the contact pressure slightly changes. A further study could employ the area after the stabilization in the solid model, to check the effect of its dimension and propose a correction factor.

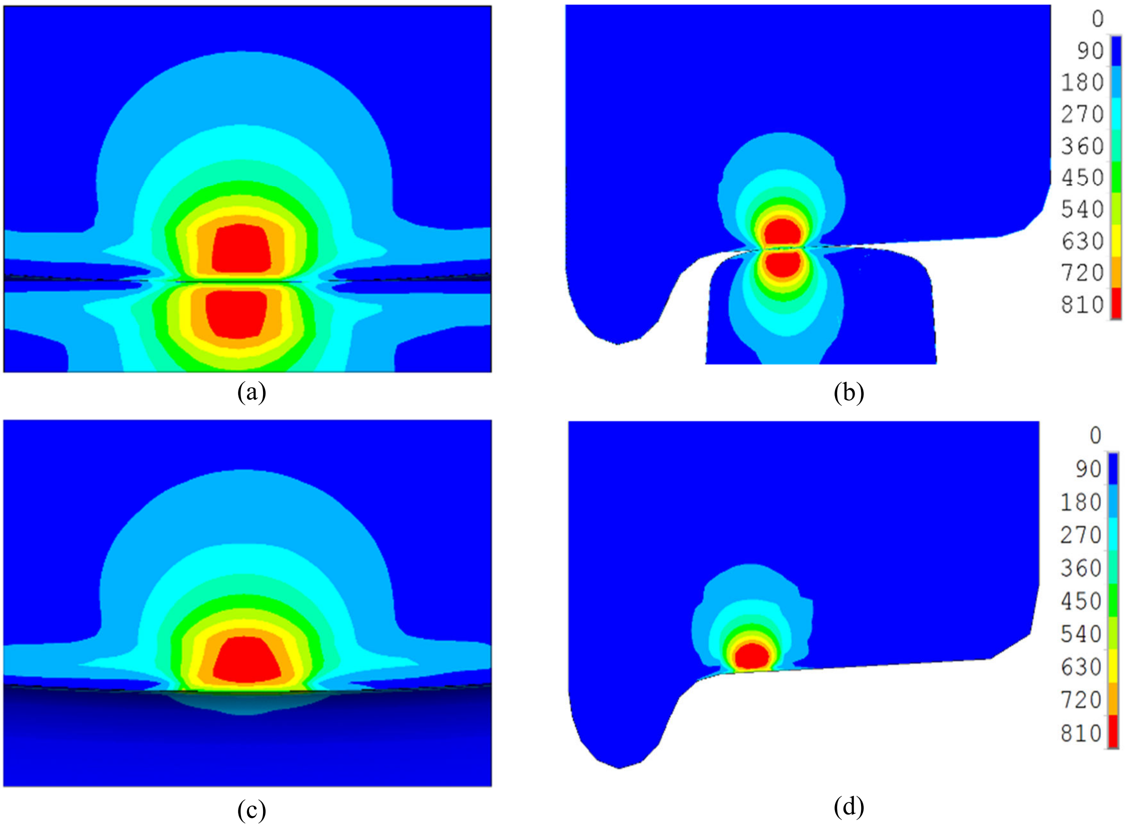


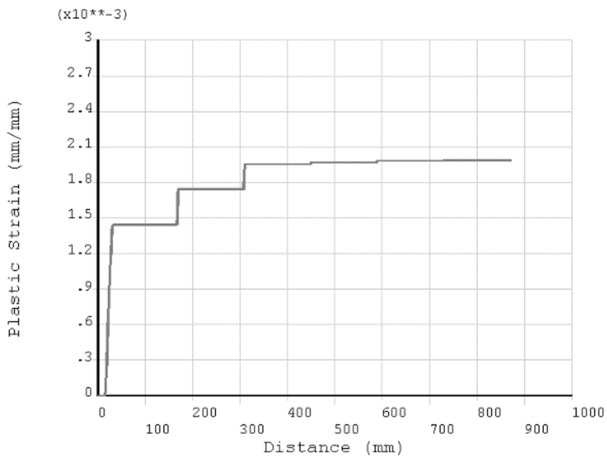
Fig. 2 - Von Misses Stress after plastic stabilization: (a) and (b) longitudinal and cross-section, respectively, in pure rolling and (c) and (d) by coupled method

The high loads applied on the wheel are supported by a little contact area and, consequently, the maximum stress exceeds the yield stress. To analyze the permanent deformations of the wheel during the rolling process, Figure 3 (a) shows the evolution of plastic strain as a function of wheel displacement for the pure rolling simulation and Figure 3 (b) shows the same for the coupled model. The selected position is located three mm above the wheel tread surface. Considering that both the wheel and rail undergo plastic strain in rolling model simulation, the ultimate strain is different from the coupled model. In the coupled method, the applied forces are distributed in an elliptical shape and this distribution keeps constant throughout the simulation. Besides, because of the elastoplastic behavior and noting that the stress levels are just above the yield stress, a small variation in the stresses could cause a significant increase in the plastic strain, resulting in the differences presented in Figure 3.

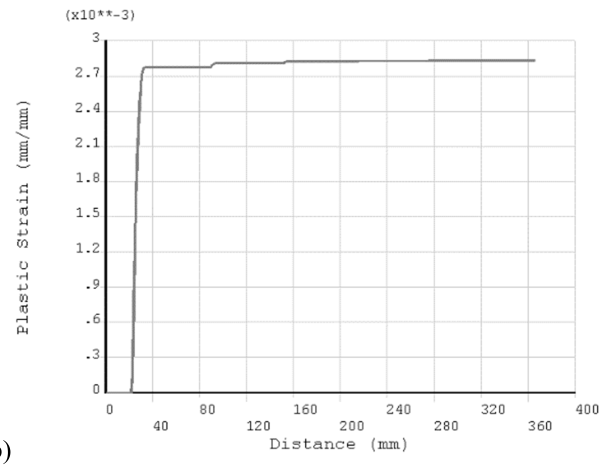
Lastly, the resulting Elastic Shakedown can be seen in Figure 4(a) for the pure rolling simulation and in Figure 4(b) for the coupled method. In first case, lower magnitudes of plastic strain are reached in the early passes; that is different for the second case. The reason again is that the plastic strain in the rail surface changes the contact area, which reduces the stress levels. For the coupled method, the first loading pass imposes a condition of high deformation and then the new deformations until the stabilization don't raise expressively. In the case of pure rolling, more cycles were needed until the Elastic Shakedown than for the coupled method.

The range of stresses after the stabilization is also different for both methods. This range is particularly important because one can calculate the life after the stabilization assuming that the stresses will variate between the limits presented in Figure 4. High cycle fatigue models are employed for this analysis, like Dang Van's criterium. The figure shows that the pure rolling model predicts a range from 160 MPa to 800 MPa, while the coupled model stresses range from 220 to 810 MPa. This difference could be important in fatigue life evaluations and will be the objective of future researches on this subject.

Though the differences can be important, the time to perform the calculation with the coupled method is about 40% of the total time for the pure rolling model. That suggests that the coupled method could be an adequate alternative after future developments.

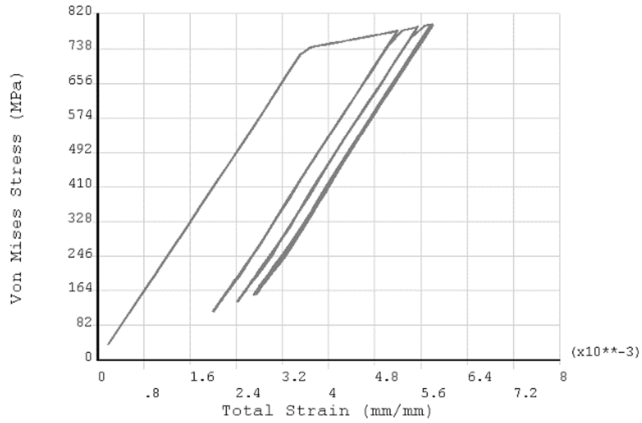


(a)

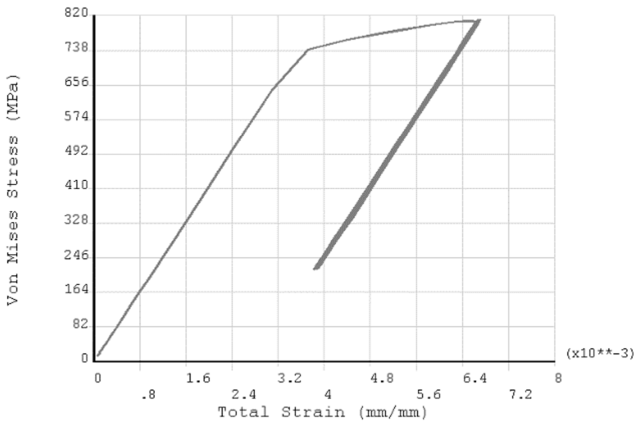


(b)

Fig. 3 - Plastic strain as a function of rolling distance and its stabilization during six cycles for pure rolling (a) and coupled method (b) models.



(a)



(b)

Fig. 4 - Elastic shakedown: Von misses Stress as a function of total strain during cycles for pure rolling (a) and coupled method (b) models.

CONCLUSIONS

This works compared two methods to obtain the plastic strain stabilization for heavy haul railway wheels. That condition is particularly important to estimate the life of the wheels until crack initiation. The conventional method, here called pure rolling model, represents both the wheel and the rail as solids, while the proposed alternative, here called coupled model, replaces the rail by a force distribution calculated from Hertz's Theory.

The results showed that there is a slight difference in the state of stresses after the stabilization, which is attributed to the contact area. In the coupled model, the same initial area is kept for all rolling passes of the wheel over the rail. For the pure rolling model, both the wheel and the rail undergo successive plastic deformation, which causes area variations and consequent stress changes. Both models reached stabilization before six loading passes, and the coupled one stabilized after just three passes.

Considering that the numerical simulation with an application of nodal forces by a contact ellipse lasted approximately 25 hours and that the simulation for pure rolling took about 60 hours, this work allowed to highlight the efficiency of the simulation with the application of the nodal forces in the semi-analytical approach - the coupled model. Besides, the differences in the stress ranges are not very high, since the mean stress changes from 480 MPa to 515 MPa and the amplitude of stress variation changes from 320 MPa to 295 MPa, for the pure rolling and coupled models, respectively. A future work will concentrate in solving the problem of the change of the contact area during the simulation using the coupled model. A model that considers this variation would represent better the final stress range and, from that, would allow for more accurate and quick life estimates for wheels and rails.

ACKNOWLEDGMENTS

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