ARTICLE IN PRESS

Advances in Engineering Software xxx (xxxx) xxx–xxx

Contents lists available at ScienceDirect

Advances in Engineering Software

journal homepage: www.elsevier.com/locate/advengsoft

Research paper

Progression of plastic strain on heavy-haul railway rail under random pure rolling and its influence on crack initiation

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ARTICLE INFO

Keywords:
Rolling fatigue contact
Railway rail
Plastic accumulation
Finite element method
Hertz theory

ABSTRACT

The present work evaluates the plastic strain accumulation on railway rails for heavy-haul transportation and the effect of them on the life cycle and the stress-strain responses considering random loading passes of normal loads on the top of rail for wheel-rail pure rolling conditions. The normal pressure distribution on the contact region was estimated by the Hertz theory and applied on the surface of the rail in a tridimensional elastoplastic finite element model. Multiaxial Dang Van criterion was applied to estimate the life to crack nucleation. The results show that the level of plastic accumulation presents distinct magnitudes depending on the position analyzed, reaching similar plastic accumulation in analogous positions on the subsurface after several passes, for two different starting positions. That result is fundamental for life analyses, since indicates that, no matter the starting point, the computed life cycles always stabilizes after a few number of cycles. From that, the fatigue life to shelling can be estimated using high cycle models.

1. Introduction

Heavy-haul railways generate severe dynamic loading on the wheel-rail interface. Extremely high stress magnitudes can be reached in this contact region, which will plastically deform the rail and the wheel. In addition, the increasing need for mass transportation worldwide imposes continuous challenges to new designs of freight railways [1–3]. The exigence is always to increase the load by axle or by wheel and that creates the need for continuous improvement of materials and designs.

Rail damages can be generally associated to the manufacturing process, as impurities or unwanted phase formations; corrugation due to excessive load or vehicular dynamic issues; wear by wheel rolling; welding cracks; phase transformations in service by heating; and high cooling after braking or axle locking. Many of these problems can be predicted during the initial design and even controlled or attenuated during usage. Others, however, such as rolling contact fatigue (RCF), may emerge unexpectedly, regardless how advanced are the rail materials. RCF can lead to catastrophic failures of the rail and consequent derailment of freight cars and of the whole train, when associated with internal stresses. In spite of that, RCF will always occur to some extent and requires attention to be prevented [4–6]. In this way, fatigue analysis of the rail is crucial for the safety of the transport.

The rail life prediction methods differ from the classic fatigue analysis in several aspects, but mainly because they consider the multiaxial state of stress in the contact region during the rolling. To deal with this problem literature presents several approaches related with RCF studies, from new experimental methods to advanced mathematical tools, such as analytical formulations, numerical methods, or a combination of both [7–10]. The analytical solutions are fast but too simplified and can only be applied under very restricted conditions. Numerical solutions are complexes and frequently require more than the knowledge about the physics of the problem. It also requires some specialization on the numerical tools employed to develop the model. One common difficult arises from the need for very thin meshes in the neighborhood of the contact region, both on the wheel and rail. The difficulties related with that need is amplified for elastoplastic models.

Developing numerical models for contact stresses requires knowing the forces that are transferred between the wheel and rail. Those forces can be calculated from the dynamic of the vehicle and train. Mostly, multi-body codes are used for this task. The local contact stresses are obtained by applying the Hertz theory or employing validated computer codes, like FASTSIM [11–13], which makes use of a simplified elastic analysis of the problem. Meysam et al. use of the finite element method to investigate both the stress and strain distributions in the region of contact, including the subsurface layers [14]. This kind of approach demands the highest computational cost in comparison with the analytical one due to the number of elements required to model the geometry of the wheel and the rail, as mentioned. On the other hand,

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https://doi.org/10.1016/j.advengsoft.2018.07.003
Received 18 January 2018; Received in revised form 13 July 2018; Accepted 22 July 2018

combining analytic contact theories with numerical methods might simplify the analysis for the contact region. This can be done by applying normal and tangential forces previously founded by analytical methods on the surface of a finite element models [15]. This line of attack allows faster results, despite the simplified assumptions done when applying analytical formulation as a part of the whole numerical model.

Dealing with numerical simulations of contact stress-strain distributions on the rail, literature presents studies about how the loads are applied to the rail [16]; whether the model takes into account or not the tangential forces from the vehicular dynamics [17]; the temperature dependence of thermophysical properties [14]; and the hardening model for plastic deformation, which can be either kinematic hardening or isotropic, bi-linear or multilinear. Daves and Fischer [18] shows that a highly plastified region at the surface of the rail could arise after a few load cycles. However, deformations on the materials must reach a steady state in a few cycles, as otherwise it will not be possible to load a rail millions of times during its life without completely failing through low cycle fatigue. Furthermore, the authors show that microstructure can be changed drastically and the material parameters taken from experiments on a new rail may not be valid for investigations of large shear strains produced at the rail surface.

The phenomenon that describes the plastic accommodation during successive passes of wheels on the rails is called Shakedown and may be elastic or plastic, depending on the loading. Wu et al. [19] reveals that when the thermal effect for every wheel rolling is considered, the residual displacement, the equivalent plastic strain, and the residual Von Mises stress on the rail surface are higher than when disregarding the thermal influence. The same will happen with rails. Consequently, plastic strain and residual Von Mises stress can be significantly affected by local temperature until the stabilization is obtained.

The peak stresses in the wheel and rail contact region is found in the subsurface region of the track instead of on the surface. That happens because of the magnitude of the resulting load and its direction. Blanco-Lorenzo et al. [20] pointed out that the location of maximum equivalent stress moves towards the surface in cases which high levels of tangential tractions are considered. That is why the nucleation and crack growth are found to occur from a position below the surface in simulations of other rolling bodies, when the effects of the traction forces are neglected. Although it is difficult to observe in numerical simulations, the failure can also arise in any position of rail section when the effects of plastic shear flow are combined with some type of defect or impurity in the material in this region [21]. Besides, the maximum stress values at each of the contacting bodies tend to approach each other because this is mainly a local problem.

Meysam et al. [14] considered the analysis of contact in a rail simulating the rolling along just one straight path on the rail surface. They used 3D finite element model and included the temperature from friction between wheel and rail. Kracalik and Vo reduced the model to a 2D approach, but always considering the same path on the rail [22,23]. Studies about the behavior of stresses and deformations analyzing different rolling paths in a tridimensional model of the rail are scarce in the literature. Since previous plastic deformations near the contact region can directly affect the level of stress distribution during the passage of a wheel on a rail, this type of investigation is necessary. However, the computational cost can make this approach impractical. One solution could come from replacing the wheel by just a pressure or force distribution. That procedure is expected to reduce significantly the processing time, allowing for a greater number of evaluations in short time.

The aim of this work is to investigate the stress and strain distribution in a rail and correspondent fatigue life employing combined analytical and numerical simulations with finite element method. The approach considers distributed random passes of the wheel on the rail. Hence, every pass occurs along a line that is parallel to the previous pass but can be out of it. The loads over the rail are represented by the normal distribution of forces estimated by the Hertz analytical model. A 3D elastoplastic model is employed to describe the problem. ABAQUS software [24] is employed, considering kinematic hardening behavior for the rail material. A procedure in MATLAB was built to generate random paths on the top of the rail, aiming to produce plastic deformations in a distributed way, which is close to real conditions in freight railroads. The rail fatigue life is estimated with Dang Van criterion, widely used in RCF analysis [25–27], for high cycle fatigue.

2. Analytical and numerical combined analyses

The first step in the approach is to calculate the load distribution and the contact area using elastic assumption – Hertz theory. The difference in the area before and after plastic deformation is supposed to be small, as indeed it is. Instead of using the real wheel placed on the contact, the model employs only the pressure distribution, which can be moved from one position to the next one, during rolling. This procedure implicates in tremendous savings in processing time.

The next step is to use this pressure distribution as a load on the rail and calculate the elastoplastic stresses generated into it. The position of the contact area moves along the rolling path at every step of time. The third step is to move the contact path to another position, parallel to the previous one and close to it, to analyze the effect of the previous plastic deformation. This step is repeated until the stabilization of the plastic deformation in the rail. At last, the same procedure is repeated, but from a new starting position on the rail tread, aiming to verify if the initial position of the contact affects the maximum residual stress generated and so, the rail life. Also, the random nature of the procedure allows to verify whether the sequence of path positions will influence the final magnitude of the residual stresses or not.

2.1. Load distribution on the rail surface

The distribution of normal pressure $p(x,y)$ in wheel/rail contact surface is represented with an elliptical shape, based on Hertz's theory. The mathematical model is defined as:

$$p(x, y) = \frac{3N}{2\pi ab}\sqrt{1 - \left(\frac{x}{a}\right)^2 - \left(\frac{y}{b}\right)^2}, \quad (1)$$

where a and b are the half of the main axes of the contact area and N is the vertical load. Both rail and wheel are considered to have the same mechanical properties. For the following simulations, the load N is considered to be the load for current heavy haul ore wagons increased by 30% to take into account the dynamic effects. The parameters used for the calculations are found in Table 1.

Fig. 1 shows the contact pressure distribution in an elliptical shape calculated with MATLAB software. In real situations, changes in dimensions a and b of the contact area might occur during the rolling passes; however, they are neglected in the following analysis. For the investigation mainly aims to evaluate the effects of random passes on plastic strain, stresses, and RCF of the rail, the coefficient of friction, the lateral, and the longitudinal tangential forces are considered negligible to simplify the simulations. Once the hypothesis is proved, the effects of

Table 1
Parameters of analytical analysis.

Parameters	Values	Units
Young's modulus	210	GPa
Poisson's ratio	0.3	–
Principal radii – wheel	475	mm
Orthogonal radii – wheel	∞	mm
Principal radii – rail	254	mm
Orthogonal radii – rail	∞	mm
Angle between planes	90	
Vertical load	191,295	N