NON-LINEAR ANALYSIS OF THE WHEEL / RAIL CONTACT

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ABSTRACT

In the present work a three dimensional non-linear finite element analysis of the wheel/ rail contact problem was performed using the software ABAQUS. The main goal was to analyse the rail stresses field during the passage of the wheel. The finite element model was built using standard rail and wheels profiles used by the Portuguese Railways company (CP), such as the UIC60 rail and the Alfa Pendular wheel. The commercial finite element software package ABAQUS was used, and a bilinear material model was adopted as an approximation for the simulation of the contact stresses between the wheel and rail. Aiming at the subsequent use of the Dang Van fatigue criterion, τ_{Tresca} versus σ_H data is required; this report presents the τ_{Tresca} *versus* σ_H loading path corresponding to the first cycle only. Further work will include the simulation of the number of passages necessary for achieving a stable loading path.

Keywords: contact stresses; Dang Van criterion; fatigue initiation in rail/wheel problems; rail/wheel contact.

1. INTRODUCTION

Fatigue is progressive damage occurring in materials subjected to cyclic loads. The study of this phenomenon assumes special importance in the design of machinery and structures, since this is the most frequent cause of service rupture.

As railways axle loads and speed increase, and wear prevention methods become more effective, it is crucial to implement solutions to prevent rolling contact fatigue. For example, increasing wear resistance of rails may imply that incipient surface cracks previously eliminated by wear, are no longer eliminated.

On wheels, fatigue cracks can be initiated not only on the surface but also under it. The initiation of surface cracks seems to be highly influenced by the presence of residual stresses and thermal loads, caused by a forced brake. According to elastic analyses the maximum shear stress appears between 4 and 5 mm under the wheel surface, however some cracks can be initiated at depths between 4 and 20mm [1].

The phenomenon of fatigue in rail is more complicated because of load randomness.

Maximum shear stress appears at a depth of 3mm and cracks initiate between 3 and 15mm below rail surface [1]. Initiation of cracks under rail surface is very common in heavy haul rail.

Dang Van proposed a fatigue initiation criteria based on the instantaneous value of shear stress $\tau_a(t)$ and hydrostatic stress $\sigma_h(t)$ [2]. This criterion states that fatigue failure will occur if the condition (1) is verified:

$$\tau_a(t) + a_{DV} \times \sigma_h(t) > \tau_{-1} \tag{1}$$

where:

$\tau_a(t)$: instantaneous shear stress value on a specific point;

$\sigma_h(t)$: instantaneous hydrostatic stress value on the considered point;

τ_{-1}: material fatigue limit in reversed torsion;

a_{DV}: adimensional constant, which represents the influence of hydrostatic stress, and can be determined by:

$$a_{DV} = \frac{\tau_{-1} - \sigma_{-1}/2}{\sigma_{-1}/3} \tag{2}$$

where σ_{-1} is the material fatigue limit in pure bending. Figure 1 shows schematically the application of this criterion. For the stabilized loading path represented in the figure, fatigue is predicted when the loading path crosses the boundary represented by equation (1).

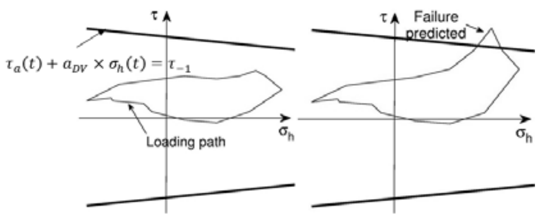

Figure 1: Dang Van criterion diagram.

The Tresca criterion can be used to calculate the maximum shear stress on the considered point [3], and the Dang Van criterion can be written as:

$$\tau_{\text{Tresca}}(t) + a_{DV} \times \sigma_h(t) > \tau_{-1} \quad (3)$$

where $\tau_{\text{Tresca}}(t)$ is the maximum shear stress.

2. FINITE ELEMENT MODEL

A 300mm long rail was used to simulate the passage of the wheel and special attention was given to the contact surfaces, where smaller elements where used to correctly define it, since contact stresses are highly dependent of contact surface geometry. Figure 2 shows the mesh used. In order to reduce computational time, only a small part of the wheel was used as showed in Figure 3 because only his surface geometry was needed. As shown in Figure 4, a refined mesh was used on the contact surfaces.

To build the finite element mesh 331 907 (256 208 to the rail and 75 699 to the wheel) 3D linear 4 nodes tetragonal elements (C3D4) were used. The wheel/rail contact surfaces were modelled using node-to-surface contact discretization and the Lagrange multiplier method was used for contact simulation.

With traditional node-to-surface discretization the contact conditions are established such that each "slave" node on one side of a contact interface effectively interacts with a point of projection on the "master" surface on the opposite side of the contact interface The Lagrange multiplier formulation add more degrees of freedom to the model in order to guarantee non penetration between contact bodies [4].

Figure 2: Rail mesh with 45 611 nodes and 256 208 C3D4 elements.

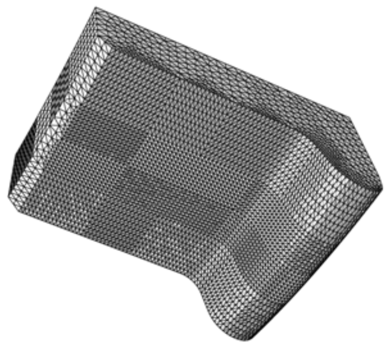

Figure 3: Wheel mesh with 15 042 nodes and 75 699 C3D4 elements.

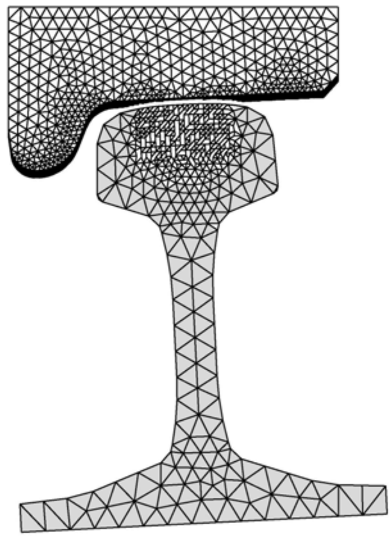

Figure 4: Model mesh.

The considered material has elastoplastic behaviour with linear-kinematic hardening and the considered properties, taken from reference [5], are shown in the Table 1. In Figure 5 the σ *versus* ε relation for the considered material is shown.

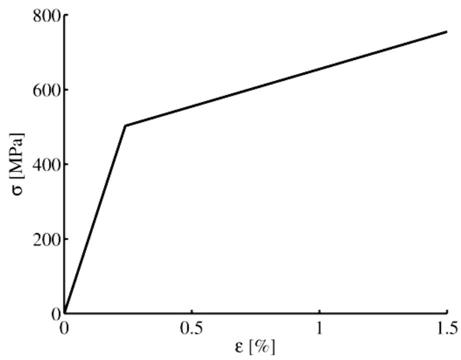

Figure 6: σ vs. ε relation for the considered material.

Table 1: Material properties [5].

E [GPa]	ν	k_e [MPa]	C [GPa]	σ_{-1} [MPa]	τ_{-1} [MPa]
210	0,3	237	20	460	270

The finite element model built was based on a UIC60 rail profile and a monobloc wheel with conic profile used in the train Alfa Pendular of the Portuguese Railways Company (CP). These two profiles are shown on Figure 6.

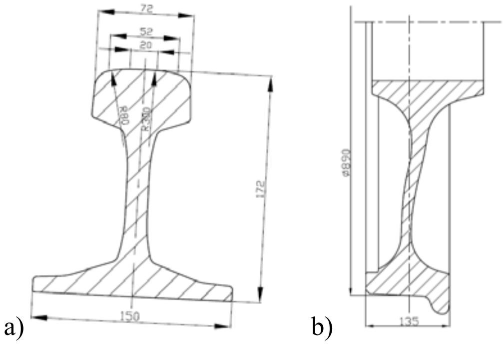

Figure 7: a) UIC60 rail profile; b) Alfa Pendular wheel profile.

In order to simulate the passage of the wheel on the rail, incremental displacements were applied to the wheel maintaining applied a normal force of 110kN, as shown in Figure 7. This load corresponds to a half of the maximum load permitted per axle.

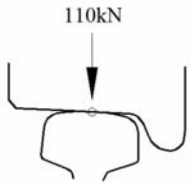

Figure 8: Load applied to FE the model.

3. RESULTS

Figure 8 presents the minimum principal strain on the rail after one passage. On Figure 9 Tresca stress distribution on the middle section of the rail after one passage can be seen. The wheel is not showed since it is not an interest of the study presented where.

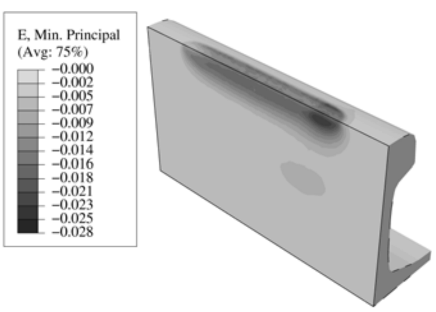

Figure 9: Minimum principal strain distribution after one passage.

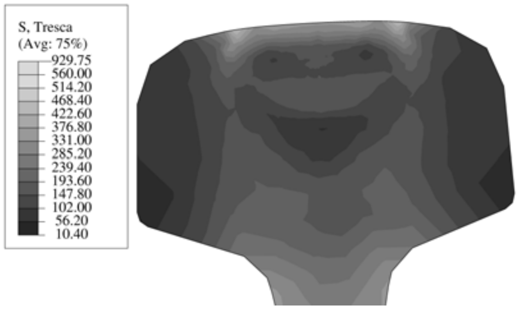

Figure 10: Tresca stress distribution on the midlle section of the rail after one passage.

Aiming at the subsequent use of the Dang Van fatigue criterion, τ_{Tresca} versus σ_H data is required. Exemplifying the data to be generated by the model developed and presented in this report, Figure 10 gives the τ_{Tresca} *versus* σ_H loading path corresponding to the first cycle only. In this figure the path Zs corresponds to the point where the maximum shear stress occurs.

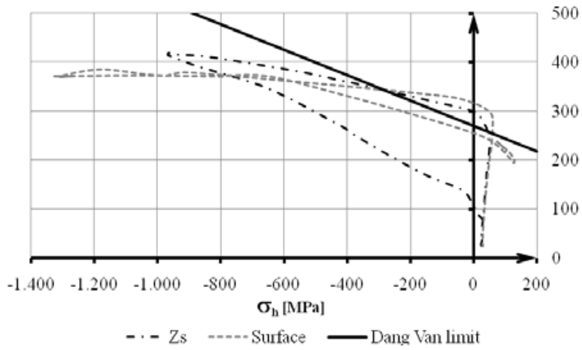

Figure 11: Application of the Dang Van's criterion to the rail after one passage

These results represent only the stress/strain evolution during the first passage.

4. CONCLUDING REMARKS

A non linear analysis of the stress/strain state resulting of the passage of one wheel on a segment of rail was performed using non-linear 3D finite elements and the commercial software ABAQUS. The work presented is a preliminary step based upon a simplified bilinear σ *vs.* ε relationship. The analysis is illustrated in the present paper through the data corresponding to the first passage; subsequent work will consist of a fatigue behaviour characterization of the rail using a more refined mesh and simulating the number of passages necessary to reach a steady state characterized by the repetition of the loading path.

ACKNOWLEDGMENTS

D. F. C. Peixoto acknowledges a Calouste Gulbenkian Foundation PhD grant (ref. number 104047).

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