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# DYNAMIC LOAD EFFECT ON THE STRESS AND STRAIN OF RAILWAY CROSSING

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**ABSTRACT**

*Solicitations induced by the passage of trains (wheels) on rail tracks are difficult to analyze, as there are many kinds of mechanical phenomena: impact, sliding and rolling. In order to make studies, the contact wheels/rails was simplified by a rig test which encountered a rigid steel disk to determine each mode of wear and fatigue. The model was built in the form of a machine-test. This paper presents a numerical model of the rig test and results obtained are compared with the experimental data. A good correlation between numerical and experimental value of the dimensions of contact zone is obtained. The influence of contact parameters on the damage of the crossing material (Hadfield steel) was investigated by numerical model of the impact-sliding contact test.*

**KEYWORDS:** railway crossing, finite element simulation, rig test (impact-sliding contact test), dynamic load

## 1. INTRODUCTION

### 1.1. Motivation

For centuries, trains were considered the best means of transportation, and many researches have been made to improve them in many aspects (speed, safety etc.). Despite the progress of technologies with faster trains, rail tracks have not been improved for a long time and the damages of wheel/rails are still unpredictable.

Railway networks are subjected to more and more severe loading conditions requiring the use of steels with a high resistance to wear and good fatigue behaviour. Switches and crossings are important elements in a railway network as they provide flexibility to traffic operations. They consist in a switch and a crossing panel connected by a closure panel. In the crossing panel, there is a discontinuity in the rail, which leads to high dynamic loads. The process of a wheel passing a crossing panel is highly complex, including dynamic effects with contact and complex geometries. In service large contact and dynamic forces cause severe damage to crossing components, such as: wear, accumulated plastic deformation and rolling contact fatigue (Fig. 1).

a) b)

**Fig. 1.** *Examples of damage to switch components: (a) part of the switch rail is detached and (b) damage due to plastic deformation and wear on crossing noses and wing rail [1]*

### 1.2. Background

The surveys carried out on out of use equipment, such as rails or switches, show that these equipments fail by wear after quite a long period of use, but they can fail by fatigue in a substantially shorter time [2]. In service, crossings are submitted to rolling, impact and sliding stresses. The impact-sliding is the result of the wheel transition from rail wing to crossing nose. Very high contact forces act on the crossing nose while such wheels are passing over it. These large contact forces between wheel and crossing can cause severe damage at crossing nose and wing rail. It is very difficult to do the theoretical research on such rolling contact phenomena because of the non-linearity of contact boundary. The study on rolling contact started in 1926 with the two-dimensional theory of Carter [3]. In the development of new concepts or the improvement of existing ones, numerical models can help to shorten development times. This can be done both by predicting the performance of possible design concepts and improving the understanding of the mechanisms of loading [4].

On the other hand, it is known that maintenance works for railway elements generates important issues such as the cost of these works and major traffic disruption. Information provided by numerical and experimental studies thus gain great importance for planning maintenance works which leads to a decrease the in above mentioned drawbacks.

In order to study and predict all mechanisms of loading different tests are designed, performed and simulated. In the development of new concepts or in the improvement of existing ones, numerical models can help to shorten development times. This can be done both by predicting the performance of possible design concepts and improving the understanding of these mechanisms. Many studies are provided on this subject.

Carter used two cylinders to simulate a wheel and a rail, respectively, and gave a two-dimensional theoretical model that expresses a relationship between the creepage and the creep force in rolling direction. The theoretical model is still used to analyze the traction between locomotive wheels and rails today [5]. Kalker developed a linear model of three-dimensional rolling contact [6] and simplified the theory [7, 8]. In his theories, the longitudinal and lateral creepages and the spin between elements in rolling contact are taken into consideration simultaneously, so they are widely used in the numerical simulation of dynamical performance of railway vehicles now. Collette and al. [9] indicate that the hardness of the superficial zone of samples deformed by shocks is of the order of 500HV, and that of samples tested by friction (discs rolling one over another) exceeds 800HV. The same order of magnitude has been measured in rolling tests [2], with a superficial hardness reaching up to 1000 Hv. In railway service the superficial hardness can reach about 600 Hv, starting from 350 Hv [2]. Bahattacharyya [10] has realized rolling tests with and without sliding on Amsler wear machine on Hadfield steel (12Mn-1.2C). A. Ramalho, M. Esteves and P. Marta [11] investigate the effect of contact conditions on friction and wear behaviour by two-disc rolling-sliding tests. M. Pau, B. Leban and A Baldi [12] report the results of a series of experiments performed on realistic wheel-rail coupling, in which the wheel was characterized by the presence of subsurface defects.

An experimental method, based on the measurement of ultrasonic reflection, is used to solve the contact problem, together with a simplified theory of rolling contact algorithm [13] by A. Rovira, A. Roda, M.B. Marshall, H. Brunskill and R. Lewis.

Numerical simulation results of wheel rolling over rail at high-speeds [14] are obtained by Jiang X. and Jin X. considering the rolling contact of the wheel and rail as a two-dimensional rolling contact in a pure rolling and steady state. Three-dimensional elastic-plastic stress analysis of wheel-rail rolling contact [15], using ABAQUS, was performed by Z. Wen, L. Wu, X. Jin and M. Zhu. The finite element simulations were conducted to investigate the influence of partial slip condition and varying contact load on the rolling contact stresses and deformations. Impact and sliding wear resistance of Hadfield and rail steel are studied [2] by R. Harzallah, A. Mouftiez, E. Felder, S. Hariri and J.-P. Maujean considering a rig test machine and numerical simulation. The rig test is used to study the consequences of repeated impacts and sliding of a specimen made in Hadfield steel.

The influence of contact parameters on the damage of different materials was investigated by impact and impact-sliding tests. Maier C. and Duchambon M. [16] propose a numerical tool designed using the finite element method and simulating the impact-sliding test, one of the specific tests for this kind of study. This tool is used in order to study the influence of contact parameters on the: size and shape of the contact area, maximal values of the equivalent Von Mises stress and plastic strain, depth of the equivalent Von Mises stress propagation.

Lots of studies and researches have been performed to analyze the different kinds of wears that wheels and rails encounter during the passage of trains at high speed. Indeed, the contact between wheels and rails is quite difficult to model, and everyone who studied the case has their own model. Thus, it is very difficult to compare results as parameters often diverge from a model to another; only behaviors are comparable.

### *1.3. Purpose and Scope*

The large contact forces between wheel and rail, causing severe damages at the crossing noses, represent the reasons why great efforts should be taken in decreasing the dynamic response. Researches presented in this paper start from the previous numerical model [16] and different ways were used to improve it: material data and introduction of dynamic solicitation. The new numerical model was exploited in order to determine the influence of contact parameters on the damage of the crossing alloy and the effect of dynamic load on the stress and strain distribution in railway crossing. An experimental study using specific rig test machine [2, 16] is presented and the results were used to validate the new numerical model.

## 2. NUMERICAL STUDY

### 2.1. Finite element model

The numerical study of the rig test using MARC Mentat finite element code is made in order to study the impact behavior with and without sliding of rail use materials and dynamic load effect. Starting from the previous model [16] we improve it by:

*a) Correct definition of material data*

The previous numerical model was not complete since the data on the effective material of the rig was missing (Hadfield steel) and replaced by another very similar one, which was a HSLA (High-Strength Low-Alloyed) steel.

Studies and analyses were necessary to select useful data. Indeed, it was necessary to determine the material behavior $\sigma - \varepsilon$ of the Hadfield steel. But Hadfield steel held special properties which enable it to modify its behavior by modifying the deformation speed when doing the tensile test.

Moreover, many formulas have been studied because the only interesting part of the material behavior was the plastic domain.

The first analysis enables to determine the right curve. For the model, the selected curve was that which had a deformation speed of 0.0047s-1, which matches with the standard deformation speed for a tensile test (0.004s-1) [2] and provided the best results when comparing with those presented by R. Harzallah in his PhD thesis [17].

*b) Friction coefficient between the guide and the rig*

In addition to the material data, another parameter had a non-negligible influence on the results and it is the friction coefficient between the guide and the rig. So, the friction coefficient between the rig and the guide was used as an adjustment parameter; many simulations were run in order to get similar results with those presented by R. Harzallah in his PhD thesis [17], particularly the Von Mises pressure (we considered it primordial). These results are given in the table below for the case 300-200-0.4 and 200-100-0.4 (ω1 - ω2 - μ, where μ is the friction coefficient between the rig and the lower disc).

By running many simulations with different friction coefficient and considering the material data, the best one has a value of 0.05. The results can be seen in the table 1.

**Table 1.** *Optimization of the friction coefficient between the guide and rig*

| Friction coefficient | Von-Mises Stress (MPa) | |
|---|---|---|
| | **300-200-0.4** | **200-100-0.4** |
| 0.045 | 461,0 | - |
| 0.05 | 465.4 | 449.8 |
| 0.055 | 478.8 | 459.1 |
| 0.07 | 491,0 | - |
| R. Harzallah | 473.56 | 449.6 |

*c) Implement many cycles of passage of the rig on the disc*

The experimental test with the machine has been done by considering many passages of the rig on the disc (concretely, a single passage is impossible to do with the current machine). So, the results of the experiments are not too comparable with the numerical model.

In order to see the dynamic effect on the stress and strain distribution, implementing many cycles in the numerical model was the solution. The principle is still the same: the rig is let with a free translation in the guide by a touching contact; the rotation of the guide creates a centrifugal force which pushes the rig outside and the collision of the rig on the disc creates an impact, then a rolling with (or without) sliding.

What differs from the previous model are the added discs and the movement of the guide. The movement can be decomposed into three steps for two cycles (Fig. 2.a):

*Step 1* (In green): Original cycle, one passage of the rig on the first disc, the same as in the initial model.

*Step 2* (In yellow): Rapid rotation to get initial position for the other disc, so as to prepare the second cycle of passage.

*Step 3* (In red): Second cycle.

a) b)

**Fig. 2.** *Implementation of many cycles*
*a) Two cycles model; b) Undefined cycles model*

To do more cycles, a new model with six discs placed around the rotation center of the guide has been made (Fig. 2.b). Indeed, adding more cycles to the simulation only requires to repeat the three previous steps, so as the guide can do complete rotations; a complete rotation is characterized by six cycles of passages.

Many passages of the rig on the disc induced change of values such as Von Mises stress, plastic strain or contact zone dimensions, which can be found in the following table.

**Table 2.** *Results from the improved model*

| Model's parameters: | ω1 (r/min) | ω2 (r/min) | μ | | | | |
|---|---|---|---|---|---|---|---|
| | 300 | 200 | 0.4 | | | | |
| Number of cycles | Max stress (Mpa) | Plastic strain | Max stress depth (mm) | Strain depth (mm) | 2a (mm) | 2b (mm) | Surface ($mm^2$) |
| 1 | 476.7 | 0.023 | 0.45 | 0.725 | 0.9 | 0.7 | 0.4948 |
| 2 | 646.4 | 0.1241 | 0.35 | 0.9 | 2.4 | 1 | 1.885 |
| 3 | 789.9 | 0.1863 | 0.25 | 1 | 2.4 | 1.1 | 2.073 |
| 4 | 892.4 | 0.2315 | 0.175 | 0.975 | 2.4 | 1.075 | 2.0263 |
| 5 | 956.1 | 0.2639 | 0.1625 | 0.975 | 2.4 | 1.05 | 1.9792 |
| 6 | 1055 | 0.3043 | 0.1375 | 0.975 | 2.425 | 1.05 | 1.9998 |
| 7 | 1049 | 0.3165 | 0.125 | - | - | - | - |
| 8 | 1175 | 0.339 | 0.1125 | - | - | - | - |
| 9 | 1140 | 0.3445 | 0.1125 | - | - | - | - |
| 10 | 1189 | 0.3604 | 0.125 | - | - | - | - |

### *2.2. Results and conclusions*

In the contact zone of the rig, the equivalent Von Mises stress is bigger than the yield stress of the material. It results that plastic deformations are obtained.

- The value of maximal equivalent Von Mises stress increases with the number of cycles. Considering its evolution, it is possible to predict the value of the equivalent Von Mises stress function of the number of cycles.

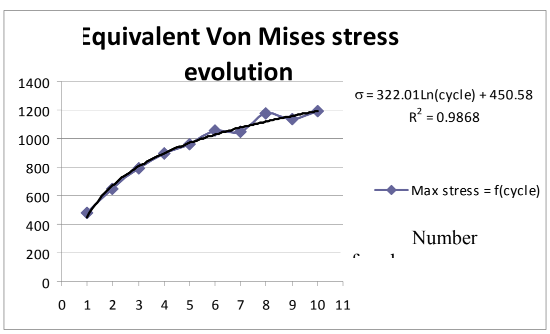


**Fig. 3.** *Evolution Equivalent Von Mises stress, function of the number of cycles*

After many cycles, the contact zone surface increases while the depth of maximal stress propagation decreases because the deformation diffuses in the direction where the value of the necessary plastic work is minimal.

After 6 cycles, the equivalent Von Mises stress exceeds tensile stress value of the tested material, when fracture can appear; this situation is experimentally confirmed.

Equivalent plastic strain value increases with the number of cycles due to the accumulation of plastic strain. Considering its evolution, it is possible to predict the value of the equivalent plastic strain function on the number of cycles.

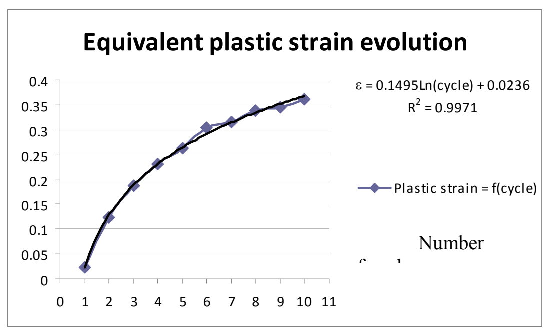


**Fig. 4.** *Evolution of Equivalent plastic strain, function of the number of cycles*

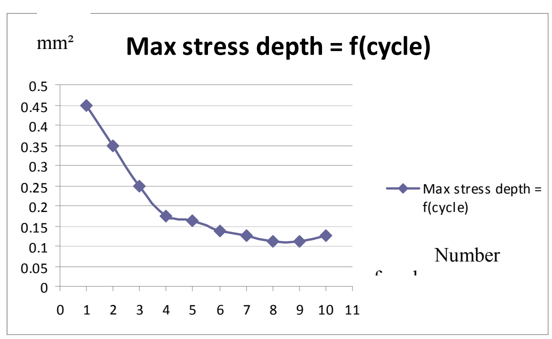


**Fig. 5.** *Evolution of maximal stress depth propagation, function of the number of cycles*

The hardness of the material in the contact zone increases with the number of cycles due to the increase of the equivalent Von Mises stress correlated with the stagnation of the size of the contact zone surface.

After 4 cycles, the increase of contact zone surface and depth of equivalent Von Mises stress propagation stops. Increase of these values is expected to continue due to the accumulation of the deformation energy.

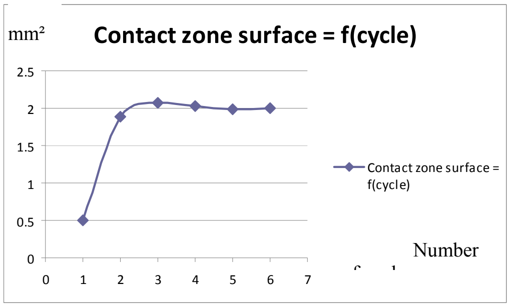


**Fig. 6.** *Evolution of contact zone surface, function of the number of cycles*

## 3. EXPERIMENTAL STUDY

The experimental study was done in order to validate the numerical results obtained after the presented improvements. The rig test machine is made of two rotating discs; the specimen can be fixed on the upper disc. During the rotation of the discs, the specimens impact the lower disc because of the centrifugal load created on them by the rotation of the upper disc.

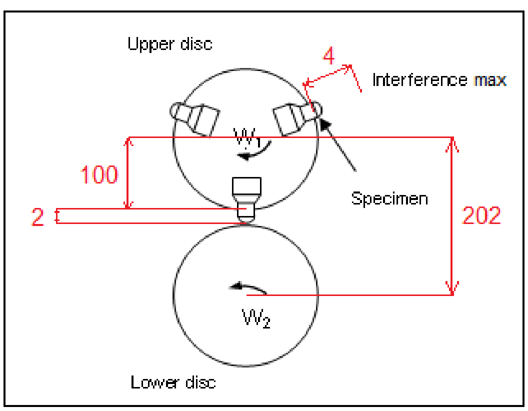


**Fig. 7.** *Plan of the rig test*

By changing the rotational speed, different combinations can be obtained. These combinations are described in the table below. In order to validate the numerical study, we consider only the following value of the machine parameters: $\omega_1$ = 300 r/min, $\omega_2$ =200 r/min and $\mu$ =0.4. The output parameter’s values obtained after first off possible stop (60 cycles) are: 2a = 2.5793 mm, 2b = 1.6516 mm and contact surface = 2.63994552 mm$^2$.

The same experiment was performed for three different states of the rig surface: without surface treatment (ND), surface hardening by explosion in one step (1T) and surface hardening by explosion in two steps (2T). The effect of dynamic load on the rig is very well expressed by its mass variation. The evolution of mass variation function of number of cycles, for all three cases of rig’s surface treatment, is presented in figure 8.

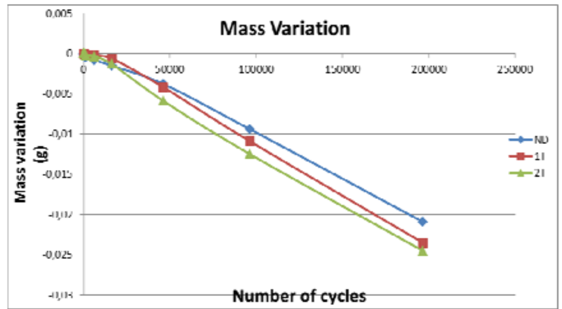


**Fig. 8.** *Mass variation for 3 state of the rig’s surface*

We can observe that a hardening treatment of rig surface decreases its mass variation.

## 4. CONCLUSIONS

The value of output parameters 2a, 2b and contact surface dimension, obtained after 10 cycles by numerical study and 60 cycles by experimental study are a little different. The experimental values of 2a, 2b and contact surface dimension are larger that numerical values with 0.0543 mm (2,1%), 0.6016 mm (36%) and 0.64015 mm$^2$ (24%) respectively. This difference can be justified by the data collection moment for the two studies.

The improvement of the finite element model can be continued by introduction of many cycles in order to obtain more accurate results.

Considering experimental results, a hardening treatment of rig surface seems to be a solution for decreasing the effect of dynamic load of crossing. The future researches will also consider this solution.

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