Wear 265 (2008) 1156–1162

Contents lists available at ScienceDirect

Wear

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

Prediction of corrugation in rails using a non-stationary wheel-rail contact model

Luis Baeza[a,*], Paloma Vila[a], Alejandro Roda[a], Juan Fayos[b]

[a] Dpto. Ingeniería Mecánica y Materiales, Universidad Politécnica de Valencia, Camino de Vera s/n, 46022 Valencia, Spain
[b] Instituto de Biomecánica de Valencia, Universidad Politécnica de Valencia, Spain

ARTICLE INFO

Article history:
Accepted 14 January 2008
Available online 19 May 2008

Keywords:
Corrugation
Non-steady state rolling contact

ABSTRACT

Most of the models used for simulating the conditions existing in the wheel-rail contact are based on stationary theories. In such theories, the parameters associated with the wheel-rail contact are independent on the conditions applied on it previously. This supposition is a simplification of the real phenomenon, whose validity lies in the rapid convergence of the contact parameters to their stationary values. However, the conditions simulated by means of non-stationary theories may differ from those obtained by using stationary theories when external conditions vary rapidly. Certain types of rail corrugation may be related to high-frequency normal or tangential forces transmitted through the contact, which may determine the effect of the temporal history on the contact parameters, and consequently on the rail wear. In order to investigate the influence of the contact process on the results of models of corrugation calculation, a methodology for estimating the rail wear depth due to a wheel running on a stretch of rail is developed. The method implements an improved contact model where non-stationary hypotheses and an exact elastic model are taken into account. The results show the influence of the more realistic hypotheses adopted in the proposed method.

1. Introduction

Rail corrugation has been a problem for railway industry as well as a subject of investigation by engineers all over the world for more than a century. Rolling noise, vibration, deterioration of vehicle and track components, along with discomfort to passengers are its principal negative effects. Nevertheless, the present operating conditions of railway vehicles, with higher speeds in passenger services and greater axle loads in freight trains, together with a general increase in railway traffic, make rail corrugation an even more serious problem. There is thus an incentive to study the causes and development of this phenomenon in order to prevent or minimise its effects.

This defect appears in all types of tracks, due to the continual passage of wheelsets, as a periodic undulation with a certain wavelength on the running surface of the rail. Grassie and Kalousek initially classified rail corrugation into six types [1]. Later, finding the typology to be even more extensive, they proposed classifying the phenomenon according to a wavelength-fixing mechanism related to resonance either from wheelsets, rails or the coupled vehicle–track system, and a damage mechanism such as plastic deformation or wear [2]. These authors suggested that wear is the damage mechanism responsible for most types of corrugation.

Since the problem is complex, mathematical models to simulate the beginning and evolution of corrugation are an essential tool for its analysis. Some of the existing models combine three mechanical models: a vehicle–track dynamic interaction model, a rolling contact model and a wear model [3–6]. The irregularities initially present on the rail running surface cause variations in the wheel-rail contact forces, in creepage and in the size of the contact patch. The input data of the wear model is composed of the results given by the vehicle–track dynamic interaction models and the contact model. This approach was developed from the specially designed tools for classic railway dynamics, such as the FASTSIM algorithm [7], to calculate contact parameters.

FASTSIM provides excellent results in determining the relationship between creepage and contact-transmitted forces, also for calculating the stick and slip zones in the contact patch. It also has the advantage of a low computational cost since it employs a simplified elastic model. The elastic model considers that the displacements associated with the elastic deformations at a point in the contact patch are a linear function of the traction forces applied at that point (*Simplified Theory*), also known as Winkler's model or Elastic layer. The effect of this simplified assumption was corrected

* Corresponding author.
E-mail address: lbaeza@mcm.upv.es (L. Baeza).

0043-1648/$ – see front matter © 2008 Luis Baeza. Published by Elsevier B.V. All rights reserved.
doi:10.1016/j.wear.2008.01.024

so that FASTSIM provides the same results than Kalker's Linear Theory at low external force or creepage. However, the local slip calculation precision may be affected by the elastic model hypothesis since this parameter has not been adjusted. This problem could influence the calculations because the local slip is the input data in wear models. The enhancing of the contact model can therefore be a way of improving methods for estimating rail wear.

An approach in this sense was presented by Li [8] and Jin et al. [9–11]. Li proposed a method for calculating wheel wear through non-Hertzian normal contact models. In Jin's studies, an improved normal contact model (non-Hertzian) was considered which permitted greater precision in determining the coupled dynamics between track and vehicle. The last method was implemented in rail wear calculations. Another work that followed this approach, but applied to wheelflats, was that of [12] who made use of the dynamic model proposed in [13].

The non-stationary contact process was investigated by Kalker [7] and Gross-Thebing and Knothe [14–18]. Gross-Thebing et al. studied the influence of the external tangential force variation on creep coefficients due to non-stationary contact process. Kalker developed a model to calculate the non-stationary contact process based on an *Exact Theory* (non-simplified relationships between displacements and tractions in the contact patch). Nevertheless, he concluded that its applicability to practical cases was inadequate due to the rapid convergence of the contact process to its stationary values. The non-stationary model has a much higher computational cost than FASTSIM or the Linear Theory, a disadvantage that prevented it from being used when the model was developed.

The applicability of a non-stationary contact model could however be related to the calculation of the contact parameters associated with a corrugated rail. Nielsen [19] thus developed a methodology to solve the tangential contact problem according to a bi-dimensional theory applied to the case of a corrugated rail. From its conclusions, the influence of the non-stationary process on wear results is determined when the forces involved in the contact vary rapidly.

The aim of the present work is to propose a method for estimating rail wear in three dimensions by using an improved contact model. The model input data are the contact geometry, the mechanical properties of the materials and the resulting forces transmitted between rail and wheel. The output gives the wear depth on the railhead. Regarding the models used for estimating wear, the contact model is based on an *Exact Theory* and considers the non-stationary wheel-rail contact process.

In the following section, the calculation model of the contact parameters is described. These parameters include the local slip used as input for the wear model implemented (Archard [20]). Section 3 establishes the procedure for the prediction of rail wear. In Section 4, the developed calculation method is applied to cases that could be considered as real and the results are compared to those of FASTSIM. The parameters of the model used in the calculations are given in Table 1.

Table 1
Model parameters

Radius of the railhead curvature	300 mm
Radius of the wheel profile	409 mm
Wheel radius	500 mm
Vehicle speed	100 km/h
Friction coefficient	0.4
Young's modulus	2.1×10^{11} N/m^2
Poisson ratio	0.3
Normal force mean value	100 kN
Longitudinal force mean value	20 kN
K_w/H ratio for Archard model	3.40×10^{-14} m^2/N

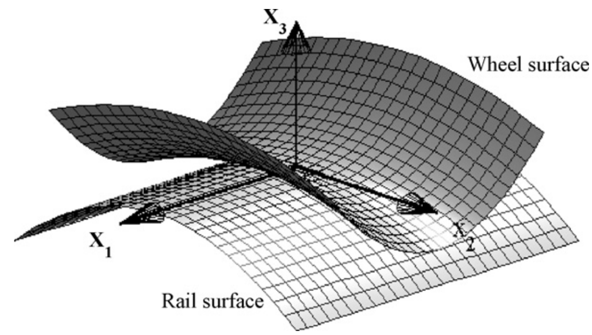

Fig. 1. Coordinate system.

2. Contact model

The method employed to calculate the contact parameters is based on the non-stationary model developed by Kalker [7]. With respect to the original model, a discretisation adapted to the contact area is used with the aim of enhancing the quality of the solution when variations occur in the normal contact forces. A wide description of the method can be found in [21].

The hypotheses associated with the non-conformal contact are considered, and it is also assumed that the normal (Hertzian) contact problem does not depend on the tangential contact problem. Its formulation is based on two wheel-rail contact models, one kinetic and one elastic. Both models will use a system of mobile coordinates $x_1x_2x_3$ (see Fig. 1) whose origin is situated at the centre of the contact patch; the x_1 axis defines the direction of the wheel movement and the contact plane corresponds with $x_3 = 0$.

To formulate the tangential contact problem, the following magnitudes are defined referring to the previously mentioned reference system

- The creepages ξ_1 and ξ_2, and spin ξ_{sp}, or the theoretical contact point velocities normalised in relation to the vehicle velocity V.
- The local slip $\mathbf{s}$, or the relative velocity between the contact surfaces.
- The displacements associated with the elastic deformations $\mathbf{u}$.
- The tractions transmitted from rail to wheel through the contact patch $\mathbf{p}$.

The relations between the magnitudes corresponding to the kinetic model are as follows:

$$\mathbf{s} = V \left\{ \begin{array}{c} \xi_1 - x_2\xi_{sp} \\ \xi_2 + x_1\xi_{sp} \end{array} \right\} + 2\frac{\partial \mathbf{u}}{\partial t} \tag{1}$$

And those corresponding to the elastic model ($\tau = 1, 2$) are

$$u_\tau(x_1, x_2) = \sum_{\kappa=1}^{2} \iint_{\text{Contact area}} A_{\tau\kappa}(x_1, x_2, y_1, y_2)\, p_\kappa(y_1, y_2)\, dy_1\, dy_2 \tag{2}$$

where $A_{\tau\kappa}$ are the elastic influence coefficients. The procedure requires a double discretisation of the problem. The first discretisation is done in the spatial domain and consists of dividing the contact area into a Paul and Hashemi mesh [22] of the type shown in Fig. 2. The local slip, the traction and the displacements associated with the elastic deformations are assumed to be constant inside each element and equal to the value they would acquire at the centre of the element. According to this simplified hypothesis,