A numerical model, based on FEM, for obtaining the vertical wheel-rail contact force.

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Abstract

The study of the wheel-rail contact is considered a major problem, because it seriously affects the dynamic behaviour of the railway vehicles. Frequently this type of contact is considered a very complex and nonlinear physical phenomenon. The main objective of the present paper is to contribute knowledge of the wheel-rail contact problem in the context of vibration generation, it also involves developing numerical model based on the Finite Element Method (FEM). The ANSYS language is used to model a wheel with primary suspension in contact with a half bi-block track design. A comparison between results from FEM model and from an analytical model widely used in the literature are presented. These results predict the vertical wheel-rail contact force caused by a wheel flat profile. The prediction model is useful during the development of a project as it facilitates the selection of the most suitable track for reducing vibration levels. The authors have also developed new numerical models of wheelset in contact with a complete railway track system, not presented in this paper, that overcome some disadvantages of the analytical models.

Keywords : *Wheel-rail contact, finite elements method, numerical results, rail vibration.*

1. Introduction

A considerable number of studies in the literature are related to the problem of the wheel-rail interaction [1, 2]. Several more or less complex theories have been developed to determine the transmitted forces between two bodies in contact, but these theories have limitations in practical use [3]. In this work, Hertz's nonlinear contact theory, generally cited by many authors, is used to describe the normal contact force, due to its simplicity. The numerical model simulate the contact between the rail of a longitudinal half track, which model can be found in [3, 4], and a wheel suspended from the bogie via the primary suspension; the model is adapted to the parameters of a bi-block half track. In this paper, the source of vibration is a wheel flat in the rolling band [4, 5]. The frequency band of the normal contact force falls into the range of 10–400 Hz.

2. Numerical Model

The 2D model specially developed to solve rapidly the problem of the wheel-rail contact in a twin block railway track is described in figure 1. The numerical model is elaborated by linking a half track model [3, 4], with a model representing the wheel and the primary suspension.

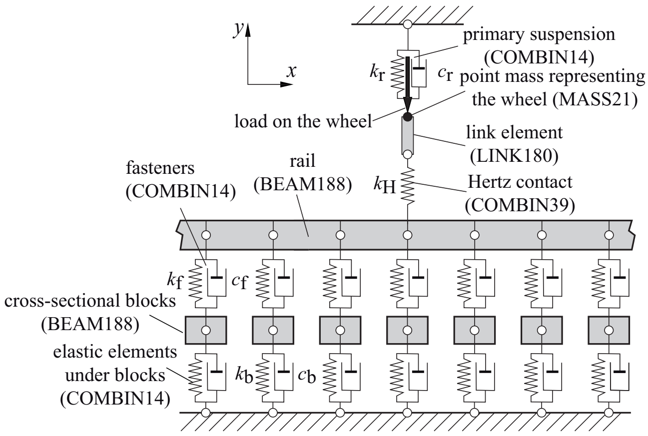

Fig.1: Wheel-rail contact numerical model of the bi block half-track. Longitudinal view.

Finally the COMBIN39 element from ANSYS is used to represent a nonlinear spring that describes the contact stiffness according to Hertz's nonlinear theory [2, 3]. In this way, the force $F(t)$ of the spring is proportional to the local deformation $\delta(t)$ of the wheel and the rail at the theoretical point of contact, raised to 3/2, according to the following equation

$$F(t) = k_H \left[\delta(t)\right]^{3/2} \qquad (1)$$

Where k_H is the stiffness constant of the Hertz spring. In the presented case, the rail is considered smooth and the irregularity of the wheel is a flat. The meshing process is done manually and assume the total number of elements constituting the structure under study. Referring to the boundary conditions, in order to approximate the real conditions of the contact, all the degrees of freedom of the nodes are constrained, except the vertical translation and the horizontal rotation , to assure the bending of rails and blocks in the vertical plane. Moreover, the upper node of the element representing the primary suspension has been blocked, thus maintaining the hypothesis concerning the bogie being considered as a rigid and fixed element to the reference [1, 2]. Finally, the condition of an infinitely long straight track is simulated with long enough track to guarantee that the track vibrations at both ends vanish due to damping. Table 1 summarize the geometric and material parameters used in the contact model.

Table 1. Parameters used in the contact models.

Wheel and primary suspension	
Mass of type S / 6000 fully loaded car	49850 kg
Wheel mass	311 kg
Wheel radius	390 mm
Primary suspension stiffness k_r	2,21 MN/m
Primary suspension damping c_r	165,96 kNs/m
Track UIC 54	
Mass of rail beam per unit length	54,4 kg/m
Density of steel	7850 kg/m^3
Young's modulus of steel	210 GPa
Poisson's ratio of steel	0,3
Bi block track.	
Rail pad (fasteners) stiffness	115,2 MN/m
Rail pad(fasteners) viscous damping	9 kNs/m
Mass of one concrete block	94,8 kg
Stiffness under blocks	17,58 MN/m
Viscous damping under blocks	8 kNs/m

The finite element analysis with ANSYS has need of three phases: pre-processing, resolution and post-processing. The first phase is about describing all the numerical elements and their physical and geometric characteristics. In the second phase, the transient dynamic analysis is employed to define the dynamic response of the system under the action of time dependent excitation forces. A constant vertical load must be applied to the wheel. Furthermore, the exciting force induced by the wheel flat is introduced into the model. In the transient analysis, the general system of equations is :

$$\boldsymbol{M}\ddot{\boldsymbol{u}}(t)+\boldsymbol{C}\dot{\boldsymbol{u}}(t)+\boldsymbol{K}\boldsymbol{u}(t)=\boldsymbol{F}(t) \tag{2}$$

where $\boldsymbol{M}$ is the mass matrix, $\boldsymbol{C}$ is the damping matrix, $\boldsymbol{K}$ is the stiffness matrix, $\boldsymbol{u}$ is the nodal displacement vector and $\boldsymbol{F}$ is the vector of forces. The system depending on time is solved using the Newmark method; see reference [3].

In the last phase, ANSYS offers the possibility to get the desired results of the developed study. In fact, the graphical and numerical revision of the results is carried out, besides generating the lists and graphs, to present the results [6].

3. Wheel with a flat

One of the most common defects appearing in the railway wheels tread are wheel flats. A newly formed flat present the form of a circle chord but soon degenerates into a flat with rounded edges. Many authors define a rounded flat according to the following equation [3]:

$$\varepsilon(x)=\frac{d}{2}\left(1-\cos\left(\frac{2\pi x}{l}\right)\right) \tag{3}$$

In order to excite vibration, a flat with the next parameters has been considered for the simulation. Flat depth d=0,2 mm. Flat length l=25 mm. Figure 2 displays the flat shape over time assuming a wheel speed of 18 m/s.

To introduce the wheel flat profile in the numerical model with ANSYS, a bar type element LINK180 was used. This element presents the attribute of a time varying length that can be associated with the flat profile.

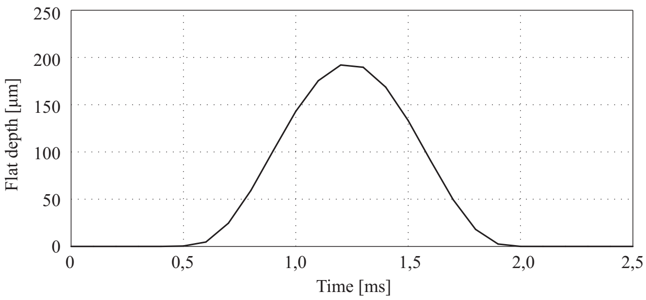

Fig 2. Flat shape depending on time. Time step 0.1 ms.

4. Comparison between the numerical and analytical model

A comparison between numerical and analytical results has been carried out for the case of the wheel flat described previously. Numerical results have been obtained using the model in section 2. The analytical model is a half track model with distributed parameters used commonly in the study of vibrations due to wheel-rail contact [1, 2, 3]. This analytical model is composed of three models listed in the next paragraphs, and is showed in figure 3. The track model is a continuous model and considers that parameters of all track elements are modelled as continuous longitudinal layers under the rail. The rail is modelled as an infinite Euler beam, that describes well the dynamic behaviour of the track for the frequency range of interest. The analytical model is detailed in [3, 5].

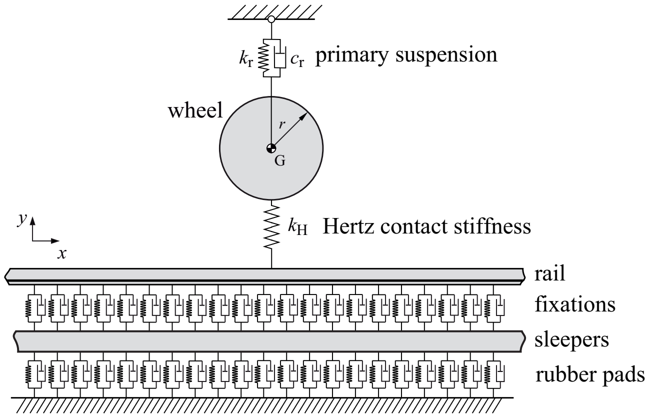

Fig 3. Analytical model of the wheel-rail contact.

The wheel and primary suspension model is a mass-spring-damper system with linear behaviour. As in the numerical model, it can be assumed that the sprung mass of the bogies and coach are not affected by the vibratory movement [5]. A constant load applied trough the primary suspension takes into account the effect of the bogie and coach weight.

The normal wheel-rail contact force model is again based on the nonlinear Hertz's theory. Corresponding to the previously observed equation 1:

$$F(t)=\begin{cases} k_{\mathrm{H}}\delta(t)^{3/2} & \text{if} \quad \delta(t)=y_{\mathrm{r}}(t)-y_{\mathrm{w}}(t)+r-\varepsilon(t)>0 \\ 0 & \text{if} \quad \delta(t)=y_{\mathrm{r}}(t)-y_{\mathrm{w}}(t)+r-\varepsilon(t)\le 0 \end{cases}$$

The contact deformation, $\delta(t)$, depends on the vertical displacement of the rail, y_{r}, the vertical position of the wheel's centre, y_{w}, and the wheel profile irregularity, $\varepsilon(t)$. It should be noted that when $\delta(t)\le 0$, the contact force disappears, what it means that the contact between wheel and rail has been lost. The dynamic equations of the model are solved in the time domain combining several convolution integrals. This integration procedure has been developed by members of the authors research team [5].. Figure 4 presents the compared results: the vertical contact force, and the vertical displacements of wheel and rail. A time shift has been introduced between numerical and analytical results to facilitate their comparison.

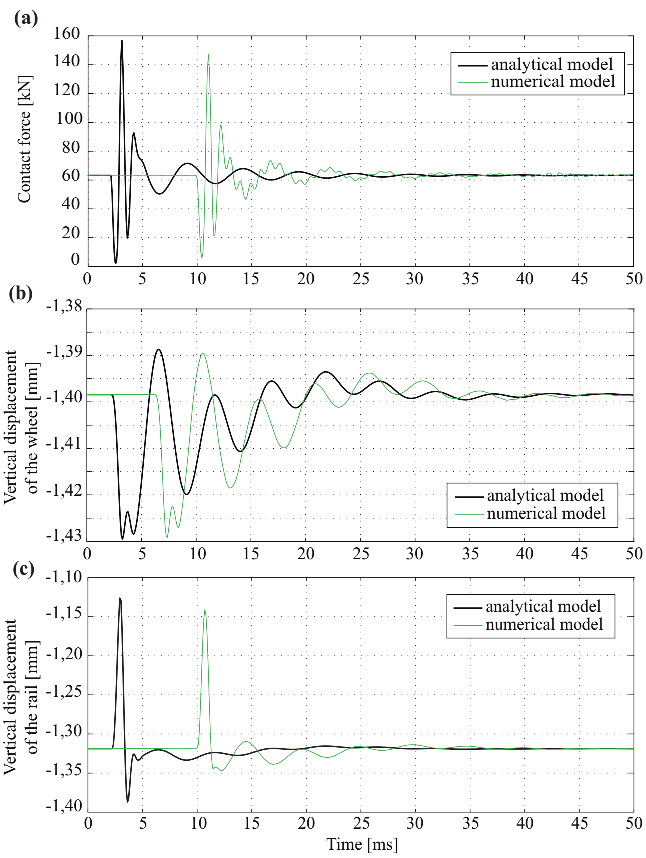

Fig 4. (a) Vertical contact force, (b) vertical displacement of the wheel, (c) vertical displacement of the rail.

It can be observed that when the wheel flat contacts with the rail, appears a sudden reduction of the normal contact force caused by the sudden change in the wheel profile, that practically reach to a null value, in the momentary separation of the wheel and rail.

The results from both models are adequately comparable, presenting some differences due to the fact that the analytical track model is a distributed parameter model unlike the numerical model.

5. Conclusions

The obtained results with the numerical half-track model agree satisfactorily with the analytical results. Finally, a numerical method provides a tool to predict vibration levels during the design and development stages of new railway projects, this method can be applied to different track models.

Although the analytical models have been widely used, they present some disadvantages as not considering the distance between fasteners or the interaction between vibrations in both rails. In fact the authors have also developed complete numerical models that take into account the simultaneous contact of both wheels of a wheelset and a complete track system.

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