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STUDY OF VIBRATIONS GENERATED BY THE PASSAGE OF A TRAIN INFLUENCE OF THE TRACK DYNAMIC BEHAVIOR

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ABSTRACT.
The present paper deals with the aim of a numeric model development in order to predict the vibrations generated by the wheel-rail dynamic interaction, caused fundamentally by the profile irregularities of railway wheels. These imperfections are the main source vibrations generated by the passage of a train. In order to study the nonlinear behavior of the wheel-rail contact, a time-domain model is used to calculate the normal force produced by this interaction, which includes the nonlinear hertzian contact theory and a simplified modeling of the wheel and the track. The limitations of a linearized hertzian model and its application to an idealized contact are discussed. In certain railway networks, the train moves along a track supported by a concrete slab. In such way, a specific model is presented in order to include the slab dynamic behavior, with the aim of determining its interaction with the track. In this work a vibration generation global model is outlined, including the railway model, the rolling system model, the wheel-rail contact force model and a simplified slab model. The global model allows the study of the vibrations due to a passage of a train, and the study of the influence of the railway components in the vibration generation and transmission.
Keywords: wheel-rail contact, hertzian contact theory, slab impedance.

1. INTRODUCTION.

When a wheel rolls on a rail at a certain speed, wheel profile irregularities originate vibrations that are transmitted to the surroundings. According to [1], wheel profile roughness and mainly wheel flats can be considered the most significant irregularities, because they are responsible for high vibration levels.

In practice, although the wheel and the primary suspension can be modeled as a linear system, the elastic local deformation on the wheel-rail contact zone presents nonlinear stiffness characteristics. However, linear contact model is usually justified on the basis that these deformations are small.

With the aim to study the vibrations in a single point of the track, this work presents a vibration generation global model, which includes a track model, a wheel model and a wheel-rail contact force model. In order to calculate the contact force, hertzian contact theory is applied, using classical nonlinear approach and a linear version. The aim is to compare both versions and to know the availability limitations of the linear model. The global model will be as simple as possible, trying that the system will be well represented without loss of fundamental information. On the other hand, a simplified model of a slab, by means of a damper, is used to approximate the actual dynamic behavior of the track mounted on the slab.

2. WHEEL-RAIL INTERACTION DYNAMIC MODEL. CONTACT FORCE TIME HISTORY DETERMINATION.

According to the simplified track model outlined in [2], the aim of the proposed dynamic model is the determination of the contact force time history. This model consists of three parts: the track model, the wheel model and the contact force model. The track is modeled as a two-degree of freedom system which parameters are estimated from a modal analysis applied to the track continuous model described in [6]. The rail displacement so obtained is named y_r.

The wheel is modeled as a rigid mass with a primary suspension, formed by a linear spring-damper system. It is assumed that the sprung mass, the mass of the bogie frame and the coach, is not affected by the primary suspension movement. This statement is valid for frequencies over 10 Hz, which is the considered case because the frequency components of the vibrations generated by the contact force are above this value. The coach weight is equally distributed among the eight wheels of the two bogies of a coach. By means of this model, wheel center vertical displacement, y_w, is determined to be used in the contact force evaluation.

Contact force model is based on the nonlinear hertzian contact theory. Thus the contact force F is proportional to the elastic contact deflection δ to the power 3/2. This formulation depends on the wheel and rail vertical displacements, the wheel radius r, the wheel profile ε, expressed as a time function and therefore related with train speed, and the nonlinear elastic constant K_H. This constant is determined knowing the curvature radius of surfaces at the contact zone and the properties of its constituent materials.

$$F = \begin{cases} K_{\mathrm{H}}\left(\delta\right)^{3/2} & \delta = y_{\mathrm{r}} - y_{\mathrm{w}} + r - \varepsilon\left(x\right) > 0 \\ 0 & \delta = y_{\mathrm{r}} - y_{\mathrm{w}} + r - \varepsilon\left(x\right) \le 0 \end{cases} \qquad \ldots(1)$$

When the contact deflection δ is small, the linear hertzian contact theory represents and alternative for the contact force calculation. In reference [3], the authors establish limits to the use of this formulation, based on the wheel irregularities magnitude, expressed as an RMS value, as well as the static deformation magnitude in the wheel-rail contact zone. Equation 2 shows the linear formulation of hertzian contact theory:

$$F = W + dF \qquad dF = \frac{3}{2} K_H\, \delta_0^{1/2} d\delta \approx C_H\, d\delta \qquad \ldots(2)$$

where W is the eighth part of coach weight, dF is the fluctuating part of the contact force, C_H is the linear model hertzian constant, δ_0 is the static deflection and $d\delta$ is the fluctuating component of the contact deflection.

3. TIME HISTORY OF THE VIBRATION VELOCITY IN A FIXED POINT OF THE RAIL FOR DIFFERENT WHEEL PROFILES.

Two different kinds of wheel profiles have been used in order to check the model. The first one is related with rounded wheel flats, characterized according to the methodology exposed in [4]. On the other hand, the second one is based on the experimental results presented in [5], which describe the roughness of a wheel profile with random distribution along the wheel perimeter.

Using the results outlined in the mentioned work, it has been synthesized a theoretical wheel profile, which considers the contribution of the irregularities formed by wheel-rail contact, as well as those produced by rail corrugation. Wheel profile spectrums have been obtained by means of the power spectrums presented in [5], which are a function of the wavelength, and introducing a random phase for each component.

Using the proposed dynamic model, the contact force time history is obtained for linear and nonlinear hertzian contact versions for comparison purposes. In such way, different wheel-rail situations have been considered, which are related with the mentioned testing wheel profiles -see Figure 1-. It can be observed how the contact force obtained using the nonlinear hertzian contact version is greater than the force obtained by means of the linear version. So, it can be concluded that linear version is not suitable when the profile irregularities are severe.

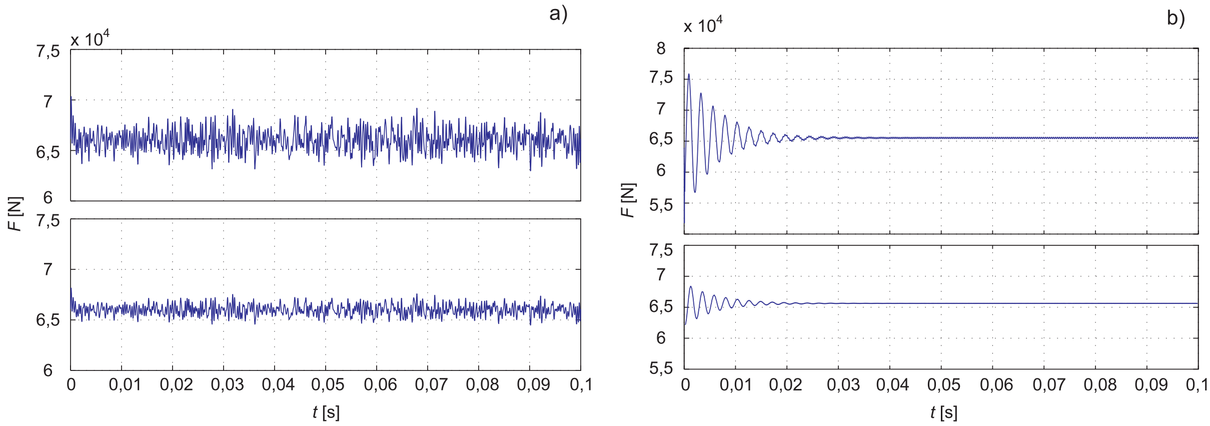

Figure 1. Contact force time history obtained with the nonlinear (top) and linear (bottom) hertzian contact. a) Random wheel profile. b) A wheel flat in the profile.

The time history of the vertical velocity in a fixed point of the rail, produced by the wheel-rail contact force, is computed by means of the convolution method with variable kernel, proposed in [6]. The basis of this method is the track cross impulse response function that relates the vertical velocity at one point in the rail with an impulse contact force located in another point. The change of the impulse response while the train is running along the track permits to consider the effect of the train speed in the track response. The cross impulse response is obtained from the cross vertical mobility for the track continuous model referenced in [2,6]. Figure 2 shows the vertical velocity time history in a single point of the rail due to the passage of a bogie, obtained with the contact force produced by the random wheel profile, using the linear and the nonlinear hertzian model. The magnitude discrepancies between both versions are the same than can be observed in Figure 1.

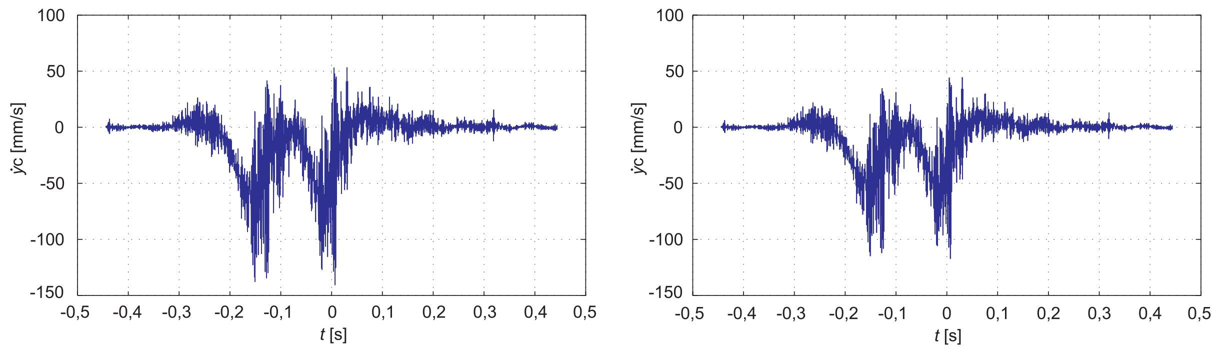

Figure 2. Vertical velocity of the rail due to the passage of a bogie produced by a random wheel profile. Left: nonlinear hertzian model. Right: linear hertzian model.

4. SLAB-TRACK INTERACTION DYNAMIC MODEL.

In this section a simple slab model is presented, which complements the two degrees of freedom model of the track presented in section 3. The track is assumed to be located on the surface of a horizontally layered space, which represents the slab. In this model, the slab behavior is described by means of a viscous damper and a constant force equivalent to the static loads in the system, the weights of their different elements. The damping constant is selected in order to quantify the vibration energy transmitted to the track surroundings, and so obtain a more realistic description of the dynamic track behavior. Figure 3 illustrates the mentioned time history considering wheel flats and wheel profile irregularities with random phase, and both hertzian contact versions. The results are comparable to the results in Figure 1, but there exist some magnitude differences.

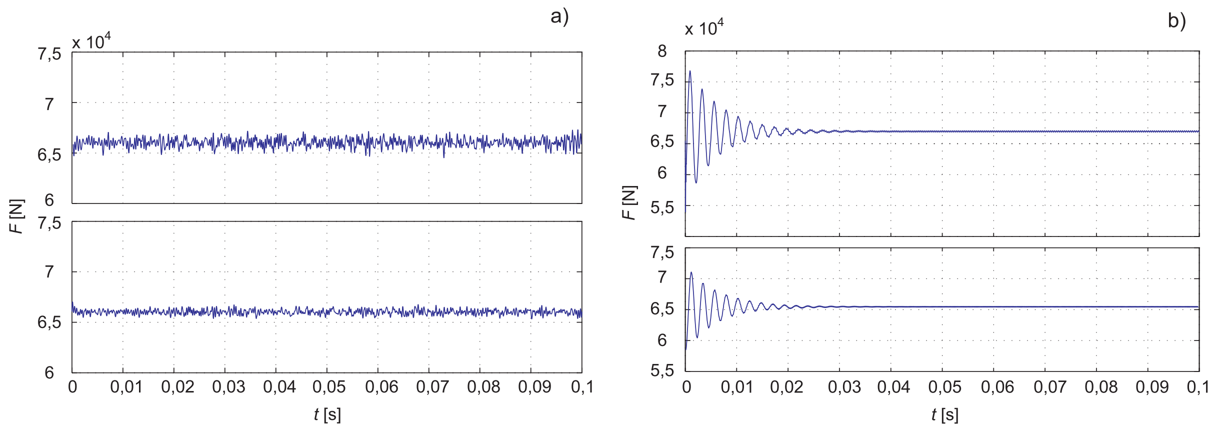

Figure 3. Contact force time history for slab-track interaction model obtained with the nonlinear (top) and linear (bottom) hertzian contact. a) Random wheel profile. b) A wheel flat in the profile.

5. CONCLUSIONS.

A global wheel-rail contact model has been developed, which permits the determination of the vibration in a single point of the rail due to the passage of a bogie, with different characteristics in the wheel profiles.

Comparing the nonlinear hertzian contact theory with its linear version, it has been observed that the contact force linear model presents some limitations, mainly when wheel profile irregularities are significant.

The presented model allows to study the influence of different kind of wheel profile irregularities on the vibration generation due to wheel-rail contact.

The simple slab model lets to obtain an estimation of the slab influence in the wheel-rail contact dynamics. At present, the authors are developing a more complete slab model to be included in the global model, based on a modal description of the slab.

This global model is the base for the study of the vibrations generated by the passage of a train.

6. ACKNOWLEDGEMENTS.

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7. REFERENCES.

[1] Wu, T.X., Thompson, D.J. A hybrid model for the noise generation due to railway wheel flats. Journal of Sound and Vibration 251(1), 2002, pp. 115 – 139.,

[2] Otero, J., Martínez, J., De los Santos, M.A., Cardona, S. Study of the vibrations generated by the passage of a train. A wheel-track system modelization. Tenth International Research/Expert Conference, Trends in the Development of Machinery and Associated Technology, Lloret de Mar, Barcelona, Spain 2006.,

[3] Thompson, D.J., Wu, T.X., Armstrong, T. Wheel/rail rolling noise – The effects of non-linearities in the contact zone. Tenth International Congress on Sound and Vibration, Stockholm, Sweden, 2003.,

[4] Newton, S.G., Clark, R.A. An investigation into the dynamic effects on the track of wheelflats on railway vehicles. Journal of Mechanical Engineering Science 21, 1979, pp. 287 – 297.,

[5] Dings, P.C., Dittrich, M.G. Roughness on Dutch railway wheels and rails. Journal of Sound and Vibration 193(1), 1996, pp. 103 – 112.,

[6] Martínez, J., De los Santos, M.A., Cardona, S. A convolution method to determine the dynamic response in a railway track submitted to a moving vertical excitation. Machine Vibration 4, 1995, pp. 142 – 146.