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Variation of Dynamic Wheel-Rail Forces in High Speed Trains

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Abstract: In this paper, variation of wheel-rail forces in dynamic train-track interaction at high speed track is investigated. To analyze track and train dynamically, a model of standard fleet and train is provided. To model the loads of track and train realistically, ADAMS / RAIL software is used. In modeling of a car by ADAMS / RAIL, an ERRI standard model of the car on a high speed track with corrugated rail (1 mm amplitude, 1 meter wavelength and total length of 5 meters) is provided. To verify the equations of dynamic load factors, offered in some codes, the software outputs and equations are compared to judge. The results of the dynamic analysis of the train shows that the equations offered in ORE manual are more applicable than those offered in the other codes.

Key words: Railway tracks, impact factor, wheel-rail interaction.

1. Introduction

In design of a track superstructure, to determine ballast thickness and limit stresses exerted on the rail, determination of dynamic forces between track and train is of most importance. In a railway track, at the beginning of operation the rail is smooth; hence, the magnitude of forces can be approximated as static weight of the train. But after a few years, rail corrugations will form, and wheel-rail impact increases the dynamic forces exerted on the rail. This problem appears more quickly in high speed tracks [1]. In case of exceeding maximum allowable magnitude, to reduce the dynamic forces, different methods will be applied, for instance, reducing the mass of primary suspension system [2] and providing optimum stiffness under the rail by means of rubber pads. In this paper, by modeling the track and train, dynamic forces between rail and wheel are investigated and the best formula to describe the dynamic forces is selected according to different standards and codes.

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2. Train Simulation

The software is capable of modeling the rear and the front bogies individually; hence, to create the model of a whole train, these elements can be assembled (Fig. 1). In order to achieve accurate and actual results, it is necessary to use a standard model. Here, ERRI model for the car is utilized. The specification of the bogie, car, and wheel-sets are detailed in Table 1. The wheel-rail contact is assumed according to Hertz theory [3]. In this theory, the elastic deformation of wheel and rail create an elliptical contact area. The dimensions of the ellipse diameters are calculated by normal force and area of the contact area [2]. Also, ratio between axes of ellipse depends on radius of wheel and curvature of rail cross section. Relation between the axes of ellipse can be calculated by:

$$2a = 3.04 \times (\frac{P.R \times 10^3}{2b.E})^{0.5}$$

Where:

P: vertical wheel load;

R: radius of wheel;

E: Yang's Modulus of rail;

2b: short axes of wheel – rail contact area;

2a: long axes of wheel – rail contact area.

Specifications of modeled vehicle are selected according to Table 1. The vehicle with two bogies is shown in Fig. 1.

After creating the vehicle model, to perform the dynamic analysis, one should define the specifications of the track.

3. Track Modeling

It is required to modeling of track with irregularity for determining dynamic force between rail and wheel. Irregularities of rail surface are assumed as sine form, which is simulation of rail corrugation. Wave length of corrugation is variable, which depends on loading condition such as axle load and train speed. In this paper, a track of 1000 meters length, UIC60 rails and corrugations of 1 mm amplitude, 1 to 5 meters wave length in 100 cycles is considered. In the software model, track (flexible) is assumed as a continuous Euler beam on visco-elastic foundation. The model is called Bushing model as shown in Fig. 2. In this model, spring/damper sets are used between rail-sleeper and sleeper- ballast.

4. Simulation Outputs and Analysis Results

In the software, initial loading of car weight on the track creates a transient variation of forces at the start time. Hence, in order to reduce the error, one should not calculate the magnitude of the force from zero. Due to this transition, the commencement of corrugations shall start form the length of about 100 meters and also 50 cycles are considered. To achieve the desired results, 250 iterations for the analysis are performed. In each analysis, time steps of 0.001 second are considered. Outputs are forces, vertical and horizontal displacements. For instance, in Fig. 3, the force variation of frontal axle of the rear bogie at the speed of 50 m/s and corrugations of 1 mm amplitude and wave length of 3 meters is illustrated. As it is shown, the average axle load of the last wheelset is measured 172.19 kN, which

Table 1 Vehicle specification [4].

Vehicle	Quantity
Mass of the full passenger car	59000 kg
Bogie mass	2615 kg
Wheel-set mass	1503 kg
Total mass	67236 kg
Wheel diameter	920 mm
Vehicle length	20 m
Vehicle width	2.6 m
Vehicle height	2.65 m

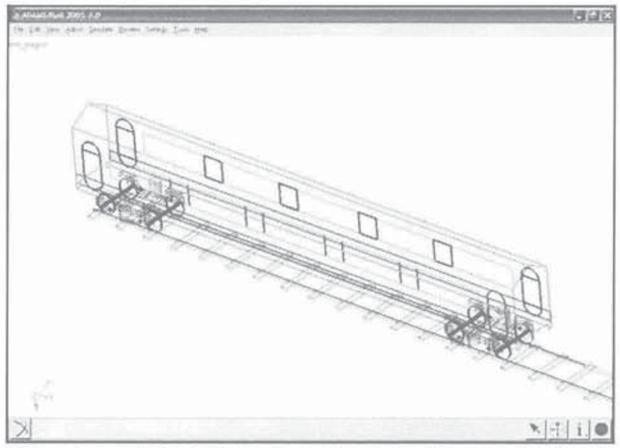

Fig. 1 Schematic diagram of the car-body in ADAMS/RAIL [4].

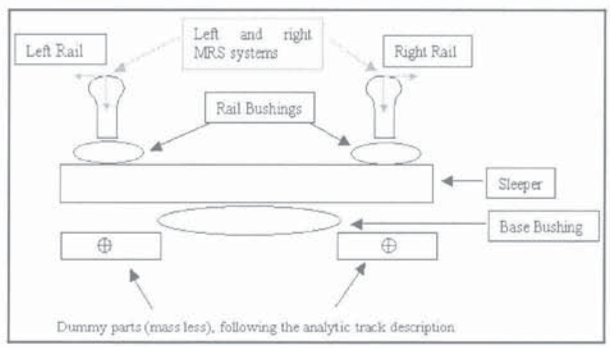

Fig. 2 Bushing model.

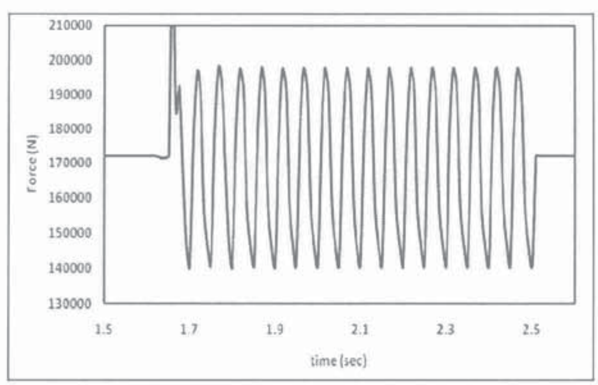

Fig. 3 Force variation of 1 mm amplitude and wave length of 3 meter.

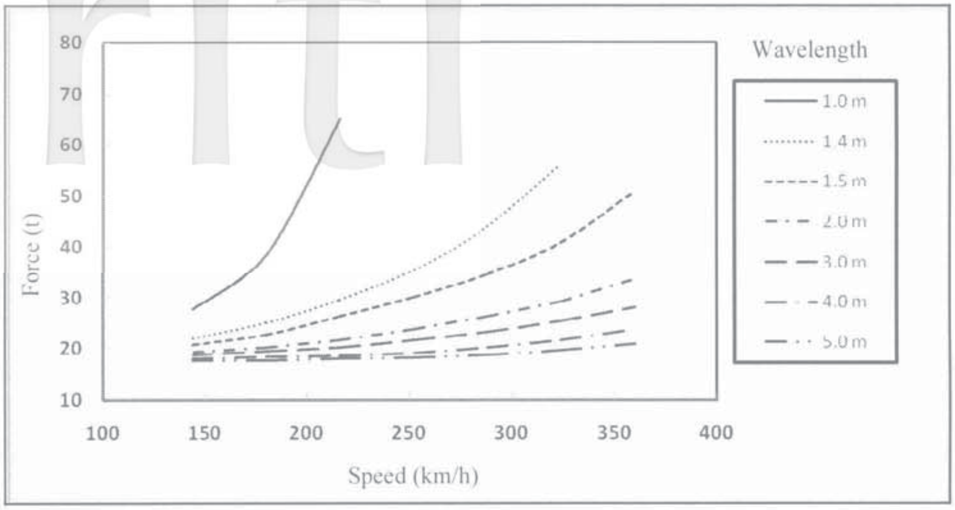

Fig. 4 **Variation of axle load with respect to speed, at wavelength of 1-5 meters and amplitude of 1 mm.**

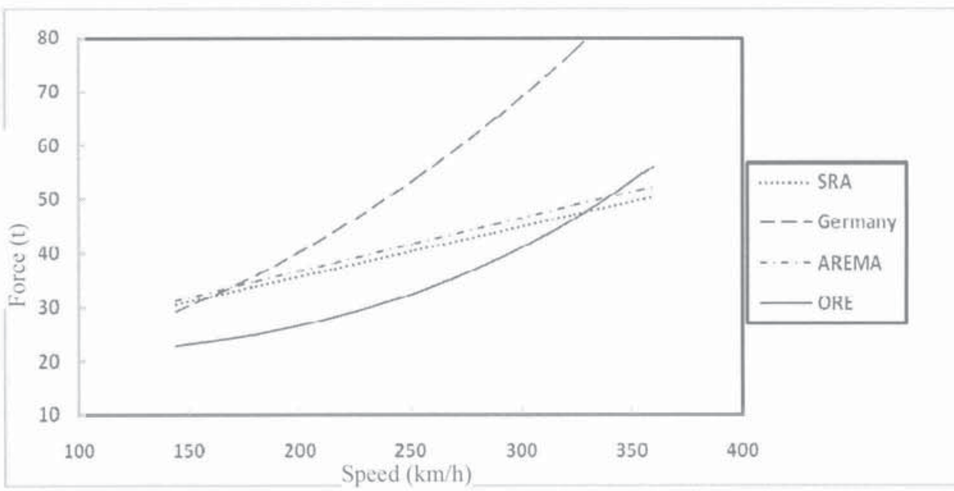

Fig. 5 **Variation of axle load with respect to speed in different codes.**

Table 2 **Impact factor in different codes.**

Impact factor	Code
$\alpha = 1.2 + 0.04(\frac{V}{100})^2(1.02 + 0.0034(\frac{V}{100})^3)$	ORE
Φ=1+5.21V / D	AREAMA [5]
Φ=1+V*V / (30000)	Schramm
Φ=1+5.21V / D	SRA

*parameter "v" denotes train speed in km/h and "D" denotes to wheel diameter in mm.

is slightly more than the average of all wheelsets, it is attributed to the position of the car-body mass center.

As illustrated in Fig. 4, at speeds of higher than 200 km/h with wave length of 1 meter and corrugations of 1 mm amplitude, force variation is extremely effected and indeed, derailment can be resulted.

To estimate the impact factor, there are some relationships offered at various codes. By comparison of these relationships with the results of ADAMS/RAIL, they can be verified, so that the best experimental relationship can be determined [4]. In Table 2, the equations to calculate the load factors are detailed.

5. Conclusion

By comparison of the results of the dynamic analysis with the experimental relationships to estimate dynamic forces, it is obvious that at speeds of higher than 140 km/h, those relations offered by ORE and Germany (Schramm) formula are of more accuracy than the others. Among these two, at corrugations of 1 mm amplitude and 1.5 meters wave length, relationships offered by ORE are more accurate (Fig. 5). Hence, to estimate dynamic forces, using ORE equations at typical high speed tracks is recommended.

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