



INFLUENCE OF WHEEL AND RAIL PROFILES ON RAILWAY VEHICLE DYNAMICS

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Abstract

This paper presents some results of investigation that combine measured data (track irregularities, wheel and rail profiles) and wide range of simulation studies. The object under discussion was a passenger car and bogies type VT72 on the Bulgarian Railways with typical inertial and elastic damping parameters. Simulations were done for 400m long sections and speed up to 200km/h. Three track sections of different geometrical imperfections had been analysed, and next their statistical and frequency characteristics have been performed. For each track section, wheel and rail profiles of diverse wear ratio were used. Obtained results such as vehicle accelerations and wheel-rail contact forces were analyzed in amplitude and frequency domain.

Keywords: railway vehicle dynamics, rail and wheel profile

1 Introduction

Wheels and rails are often contaminated with water, grease, coal, grain, sand, dust, wear debris etc. These contaminants invariably change the friction between wheel and rail. Some contaminants (e.g. iron oxide and sand) will increase friction, whereas others decrease friction (e.g. grease, coal and water). Inadequate friction can result in disasters. For example, low friction can cause poor braking. High friction on the gauge face/flange can cause low speed wheel climb derailments. Friction also causes numerous damaging effects. Abrasive wear, plastic flow, corrugation, shelling and spalling are obvious forms to wheel and rail rolling contacts. High friction, in particular, causes severe wear plastic flow and fatigue.

Geometric irregularities of a track such as alignment, cant or gauge, generate excitations in low frequencies of railway vehicle vibrations, commonly less than 30 Hz. This range of frequencies is responsible for dynamic behaviour of the vehicle on tangent track and for a good curving performance at medium or great values of curve radii, and at high speed. Also the rolling surfaces of the wheel and rail strongly influence on the vehicle performance, so that the effects of profiles changes formed by wear should be considered in detail, see e.g. [6], [8]. The wear of these surfaces influences also on the geometric parameters of a kinematic pair of wheel and rail and produces changes of contact forces between wheels and rails in comparison to nominal dimensions [2], [4]. This, in turn, can alter the safety and quality of a vehicle motion.

The experimental studies were performed on selected railway track sections. Measured track geometric irregularities were used as input data for the simulation tests. First of all, wheelset accelerations and wheel-rail contact forces were mainly analysed as vehicle dynamic response to those irregularities. The analysis of obtained results was performed in time and frequency domain as well as statistically. This analysis allowed us to formulate some important features of railway vehicle dynamic behaviour in relation to wheel and rail profiles wear.

2 Mathematical model of railway vehicle – track system

The mathematical model analysed in this paper is treated principally as a multi-body representation of a conventional passenger car. Its mechanical structure, inertial, elastic, and damping characteristics correspond to first class passenger car with Y72 bogies (Fig. 1.), which is in permanent operation on the Bulgarian Railways.

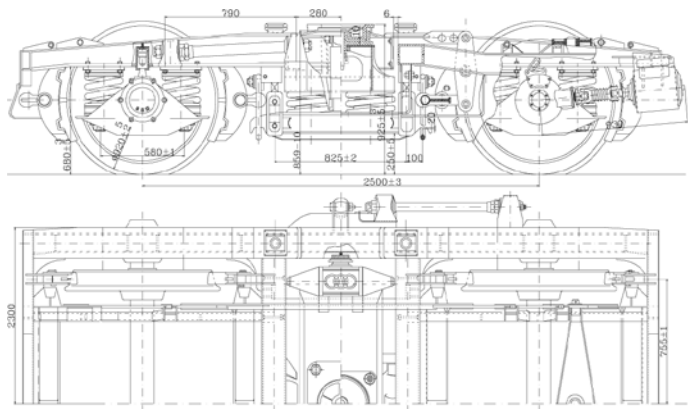


Figure 1 Bogies type Y72.

A mathematical model of vehicle motion along a track consists of two interacting subsystems, namely railway car model and track model. Railway car model consists of inertial solids in the form of four wheelsets, two bogie frames (including other parts rigidly attached to the frame), and a body. The bodies are connected by means of massless primary and secondary suspensions whose spring and damping characteristics may be introduced as linear or non-linear. The test includes studying vehicle movement in relation to a reference system that moves along the track axle with the speed of undisturbed vehicle movement. Track model is subordinated to the analysis of dynamics pertaining to low frequencies. In this sense its properties are described by means of geometric irregularities of wave lengths from ca. 3 to 30m. In the low-frequency model, the track is usually a rigid system, with the nominal layout geometry in the form of straight sections, transition curves, circular arcs, crossings and turnouts. Deviations from the nominal track dimensions are characterised here by means of geometric deviations. For a track of constant nominal dimensions usually four quantities (parameters) are used to describe the deviations [5]. These are as follows: level, alignment, cant, and track gauge deviations. The deviations are also taken into account while calculating geometric contact functions and wheel-rail forces. Therefore, assuming that the vehicle moves along a rigid track, the mechanical model of the system has 27 degrees of freedom, while the vehicle-elastic track system has 51 degrees of freedom [3]. Wheel-rail contact forces are calculated due to Kalker's theory [7]. Hypothesis of linear Kalker J.J. is based on the hypothesis of Carter, as the dependencies between creep and creep forces are given by linear expressions [1]:

$$F_x = -f_{33} \cdot \xi_x; \quad F_y = -f_{11} \cdot \xi_y - f_{12} \cdot \xi_{sp}; \quad M_z = f_{12} \cdot \xi_y - f_{22} \cdot \xi_{sp} \quad (1)$$

where: f_{11} , f_{12} , f_{22} and f_{33} creep coefficients determined by the dependencies:

$$\begin{aligned} f_{11} &= (ab) \cdot G \cdot C_{22}; \quad f_{12} = (ab)^2 \cdot G \cdot C_{23}; \quad f_{22} = (ab)^2 \cdot G \cdot C_{33}; \\ f_{33} &= (ab) \cdot G \cdot C_{11} \end{aligned} \quad (2)$$

where: ξ_x, ξ_y, ξ_{SD} - are respectively the longitudinal, transverse and rotating creep; a and b - axes of the ellipse of contact, defined by a formula of Hertz.

Characteristic of all existing assumptions and methods for determining the tangential force T_{kj} (fig. 2.) at the contact surface is that dependencies are expressed in two areas - the pseudoplasticity and dry friction and hence the need to work with disabled functions. Ex-

hibitions blemish is removed, if the function $\vec{T} = \vec{T}(\vec{u})$ can be approximated as a continuous function in both areas mentioned

Dependence is approximated [1] as follows:

$$|T_{kj}| = -f \cdot N \cdot \text{th}(2,727 \cdot \tau) \quad \text{where} \quad \tau = \sqrt{\xi^2 + \eta^2} \quad (3)$$

where: f – coefficient of dry friction; N - normal force at the point of contact.

The solution of the system of equations is achieved by means of the generated simulation program [2].

As input nominal data profiles BDZ-2 (Fig.3.) for wheels and UIC60 for rails.

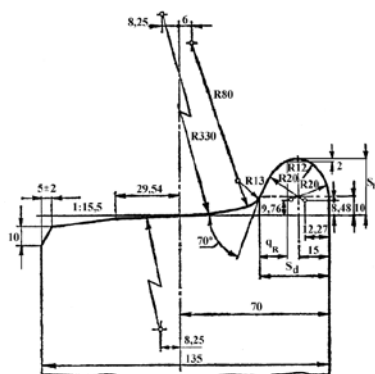


Figure 2 Force tangential.

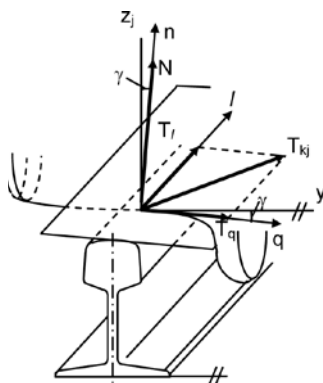


Figure 3 Fig.3. Profil wheel BDZ-2.

3 Geometry characteristics

In this section we will discuss geometric characteristics of the track irregularities (the statistical and spectral values) and rails and wheels profiles (the positions of contact points). The geometrical parameters were obtained using plasser-em120 recording car [5] and are valid for wavelength range from about 3 to 30m.

The track data chosen to computations, responds to track sections with different level of maintenance (different track quality) are called route I, route II and route III.

When we take into account the amplitudes of irregularities and their standard deviations, then the track section of the best maintenance level is route I, and the worst one is route III (fig. 4a and fig. 4b). The statistical parameters situate route II between route I and route III. For the routes under the discussion the standard deviations and the range of irregularities significantly change themselves.

In the frequency domain we can observe a considerable difference between the peaks positions (fig. 5).

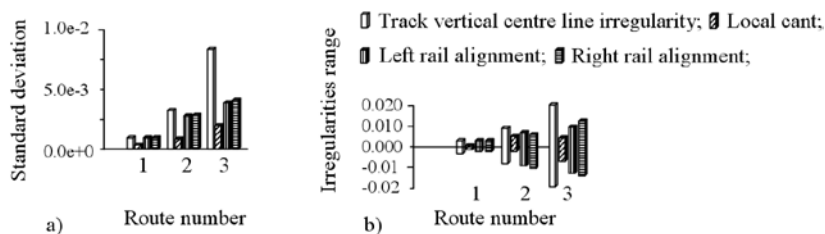


Figure 4 Track irregularities: a) standard deviations [m]; b) range of changes [m]

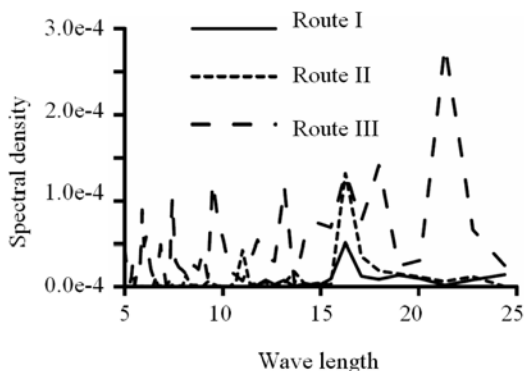


Figure 5 Power spectral density of track centre vertical irregularities for different routes

They are quite different for the route III. The dominating wave has there a length of over 20m in contrary to route I and II where dominating waves of irregularities are shorter. For simulations studies three rails profiles were chosen – a new one, and two worn profiles, as well as three wheels profiles with a different wear ratio (Fig.6, 7).

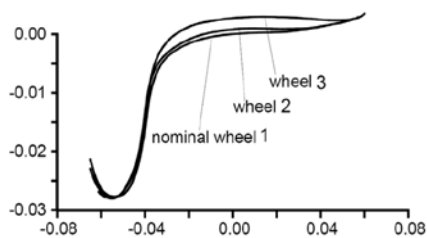
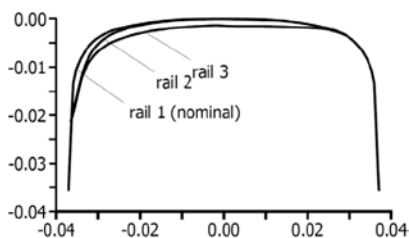


Figure 6 Rail profiles for simulation tests [m] Figure 7 Wheel profiles for simulation tests [m]

In order to characterise the system's geometry we will discuss the position of contact points between rail and wheel as the basis of geometry of contact.

4 Simulation results

While demonstrating the results, we will focus on the first wheelset dynamic behaviour. For route I simulations were done in a wide range of speed i.e. from 100km/h up to 200km/h. For other routes their state of maintenance does not allow such a high speed, and the simulations were performed from 80km/h to 160km/h for the route II and from 60km/h to 120km/h for the route III.

In figures 8–10 statistical characteristics of the wheelset motion are shown.

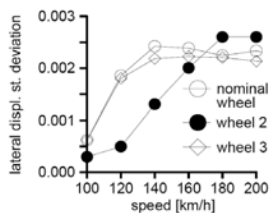


Figure 8 Standard deviation of the first wheelset lateral displacement

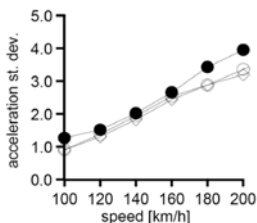


Figure 9 Standard deviation of the first wheelset lateral acceleration

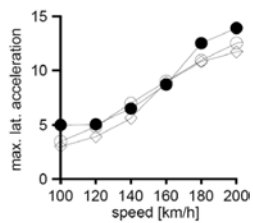


Figure 10 Maximum of the first wheelset lateral acceleration

For this route, when we compare differences in results for nominal profile and worn profiles of the wheelset we can state that they are not significant in standard deviations of the first wheelset lateral acceleration and maximum values of these accelerations. The apparent differences occur when we compare the results obtained from simulations with wheel profile signed as 2.

The analysis in frequency domain i.e. power spectral density of acceleration and lateral force does not show differences in peaks location but it should be noticed that for profile-2 the main peak value of power spectral density of lateral force is two times greater than for profile-3 and nominal.

Referring to displacements we can say that for the nominal wheel profile the standard deviation of wheelset lateral displacement increases with the speed up to 140km/h and next its value becomes nearly constant. Standard deviations of lateral accelerations for all the profiles increase almost linearly vs. speed. The differences between them exists and the results obtained for the nominal pair of wheel-rail profiles are nearly the average of results for worn wheel-rail pairs. The maximum values of wheelset lateral accelerations have very similar features as standard deviations, but for nominal profiles we obtain the highest values.

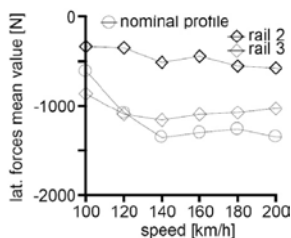


Figure 11 Mean value of resultant lateral force acting on wheel

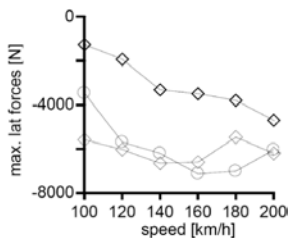


Figure 12 Maximum of resultant lateral force acting on wheel

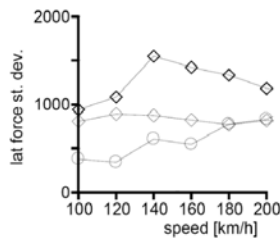


Figure 13 Standard deviation of resultant lateral force acting on wheel

Examples of forces statistical values are show in figures 11-13.

Concluding this section one can say that wear of wheel profiles can significantly change wheel-rail forces level, even on a track with good state of maintenance. It seems that frequency characteristics remain the same (in the sense of spectrum peaks location).

Finishing this paragraph we would like to say some words about results obtained for routes II and III, which were not described in details. In these cases the influence of profiles wear level on the vehicle response is not of prime importance. The track irregularities play the

Conclusions

The type of wheel and rail profiles interacting mutually during vehicle run principally has the substantial meaning for the vehicle dynamic behavior.

Performed calculations for chosen track sections using of the worn wheel and rail profiles point out visible changes of wheelset accelerations and wheel-rail contact forces. The changes are relatively greater on the sections of poorer quality of track maintenance. One can say that wear of wheel profiles can significantly change wheel-rail forces level, specially on a track with good state of maintenance. The frequency characteristics remain very similar in the sense of spectrum peaks location.

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10 RAILWAY MAINTENANCE

