ICSV13 - Vienna

The Thirteenth International Congress on Sound and Vibration

Vienna, Austria, July 2-6, 2006

THE INFLUENCE OF WHEEL AND RAIL PROFILES CONDITION ON LATERAL DYNAMICS OF RAILWAY VEHICLE WITH INCREASED LOAD CAPACITY.

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Abstract

Computer simulation has been proven in recent years to be one of the most efficient approaches to modelling of vibrations generated by a railway vehicle in different exploitation conditions. In this article the authors will focus on some basic aspects of lateral dynamics and corresponding low-frequency vibrations of the railway four-axle freight vehicle, utilizing the vehicle-track non-linear simulation model. The parameters of the vehicle model are chosen to match a typical open freight car construction with normal and increased capacity level (increased load capacity in this case denotes load of 25T per axle).

INTRODUCTION

Lateral dynamics of a railway car is inseparably connected to the conditions of interaction between the wheels and the rails, hence it is necessary to pay special attention during the simulation studies to contact conditions such as the actual shape of the wheel and the rail rolling surfaces, size of the contact area, interaction forces and resulting displacements. Wheel/rail interface research is even more vital in the case of putting increasing demands upon both a track and rolling stock in order to increase efficiency by using rolling stock with higher load capacity level. Moreover the simulation studies need to be placed in relation to the running safety issues.

SYNTHETIC DESCRIPTION OF THE MODEL

Analyzed model should be treated as a multi-degree of freedom system including stiff irregular track. Parameters of inertia, stiffness and damping elements as well as geometric structure are typical for a conventional two-bogie freight car.

General assumptions for the system model

We assume small magnitude of vibrations; relative motion of the system is modelled in a non-inertial reference frame, moving along the track centre line together with the vehicle. The vehicle model has 27 degrees of freedom and consists of stiff bodies (car body, 2 bogie frames and 4 wheelsets) moving at constant velocity. Every wheelset has 3 degrees of freedom: lateral displacement of mass centre, angle of rolling motion around lateral axis and attack angle. Bounds on wheelset motion depend on the geometry of the wheels and the rails rolling surfaces. Every bogie frame has 5 degrees of freedom: lateral displacement, bounce, roll, pitch and yaw. The car body has 5 degrees of freedom, which are the same as in the bogie frame. The bodies are coupled together through massless elastic-damping elements. Profiles of wheels S1002 and rails UIC60 used in the simulations are both nominal and slightly worn, lateral rail inclination is equal to 1:40.

GEOMETRIC CONTACT CONDITIONS BETWEEN A WHEEL AND A RAIL

Description of contact geometry between a wheel and a rail has been the subject of many publications where it has been considered as a fundamental problem in analysis of railway vehicle motion and estimation of contact forces e.g. [1, 2, 3]. Nevertheless the problems of contact geometry computation are still the subjects of many elaborations. Aspiration to extending revenue service time of tracks and railway vehicles, attempts to optimise profile shape and resulting minimisation of the maintenance costs which stem from the wear of rolling surfaces, analysis of derailment conditions [4, 5, 6] make significant impulse to continuing theoretical studies in this area. Analysis of the geometry of mutual interaction between wheels and rails is also the basis of studies of curving performance [7, 8] and running over a switch [9]. Moreover analysis of such negative phenomena occurring in vehicle motion as limit cycle oscillations connected to displacements of wheelsets and their hunting motion is largely based on studying the contact between a wheel and a rail with special attention paid to the flange contact [10].

There is a number of geometrical properties of rails and wheelsets, which strongly influence the dynamics of the railway vehicle–track system and consequently the safety of travel. In simulation studies they are accomplished through so called Geometric Contact Parameters (GCP) (fig.1). These objects represent geometrical quantities, which are characteristic for contacting the wheel and the rail rolling surfaces. Precisely, these objects depend on lateral displacement of the wheelset y, attack angle and on the position along the track section x (e.g. straight track or switch area), track gauge etc. Under these objects we can subsume: position of the contact point or points on the wheel and the rail, resulting such quantities as rolling radius, roll angle, derivative quantities such as contact angle in reference to the track plane or radiuses of curvature of the wheel and the rail at the contact point. GCP considerably change due to wear of the wheel-rail pair and resulting contact points location.

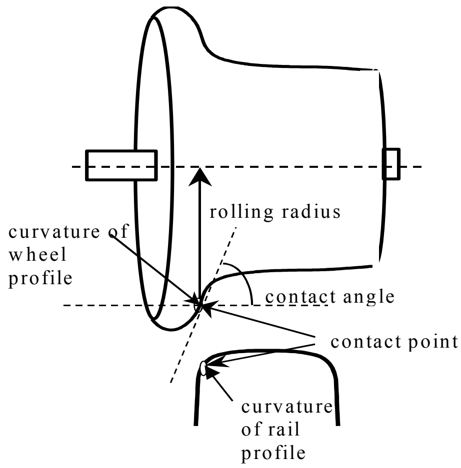

Figure 1 – Geometric Contact Parameters for one-point contact between a wheel and a rail

Seeing that linear models of contact between a wheel and a rail reveal lack of precision for larger displacements of the wheelset, the whole attention is focused on studying the influence of geometry on the task of choosing a method for calculation of tangential forces according to Kalker's theory (accomplished by program FASTSIM [11]). This algorithm as well as the others e.g. [12, 13] requires calculation of normal contact forces and creepages. Taking into account the fact that generally (in real exploitation conditions) a wheel and a rail profiles differ considerably from their nominal shape, as an exemplification we show analysis of the contact points location for the selected wheel and rail profiles at certain level of wear (fig. 2, 3, 4, 5).

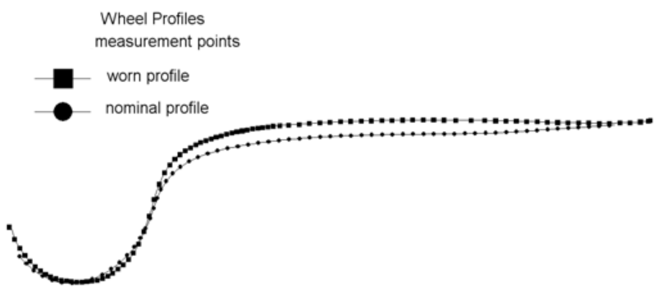

Figure 2 – Measurement points of the worn and the nominal wheel profile.

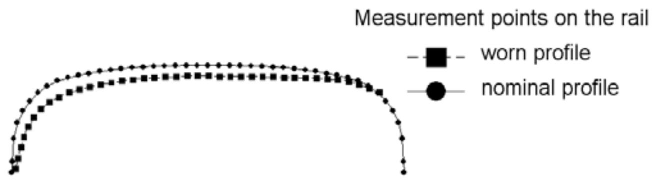

Figure 3 – Measurement points of the worn and the nominal rail profile

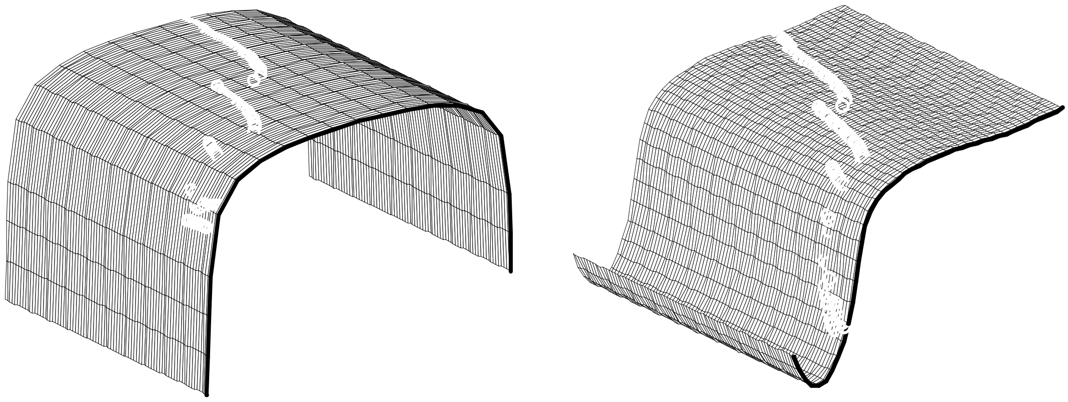

Figure 4 – Location of the contact points for the new rail and the new wheel for lateral wheelset displacement range from -8 mm do 8 mm, attack angle ψ=0.

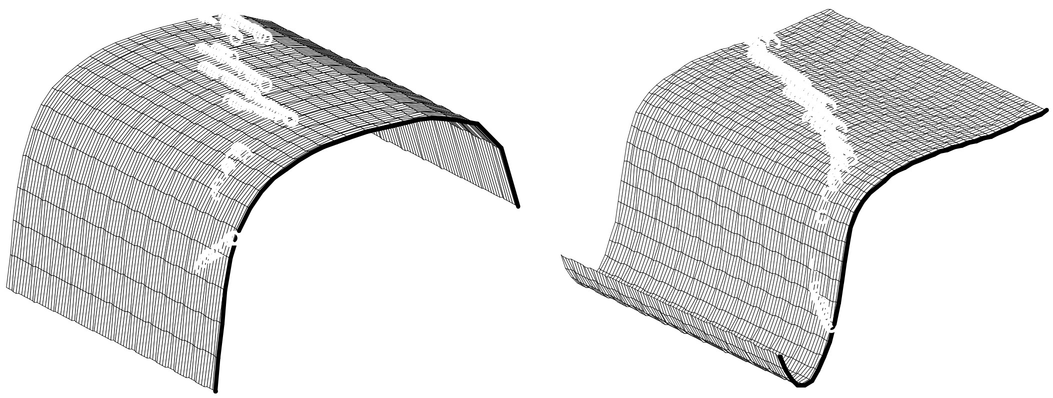

Figure 5 – Location of the contact points for the worn rail and the worn wheel for lateral wheelset displacement range from -8 mm do 8 mm, attack angle ψ=0.

Now we can characterize the above-mentioned cases. In the setup comprising the nominal wheel and rail we can clearly see a stepping change in the contact points location on the both rail and wheel with five distinguishable contact areas on the rail.

In the setup comprising the worn wheel and the worn rail we can distinguish several contact areas on the rail and stepping transition to the contact between the rail and wheel flange for larger lateral displacements of the wheelset.

SIMULATION STUDIES

Further exemplary results of the simulation studies are presented below. Figures 6-7 show the influence of the axle load on lateral displacement magnitude of the leading wheelset with nominal and worn profiles. At the level of a 180kN axle load (fig. 6) the wheelset with worn profiles has larger amplitude of oscillation than the wheelset with nominal profiles, whereas at the increased axle load of 250kN the amplitude of oscillation can be smaller for the wheelset with worn profiles. Similar behavior is also illustrated in fig. 9 where the leading wheelset with worn profiles loaded with a 180kN static force exhibits lateral motion of higher amplitude than the wheelset loaded with a static vertical load of 250kN. Next fig. 8 compares the oscillation amplitudes of the leading wheelset with nominal profiles. In the latter case varying axle load has less influence than in case of the wheelset with worn profiles.

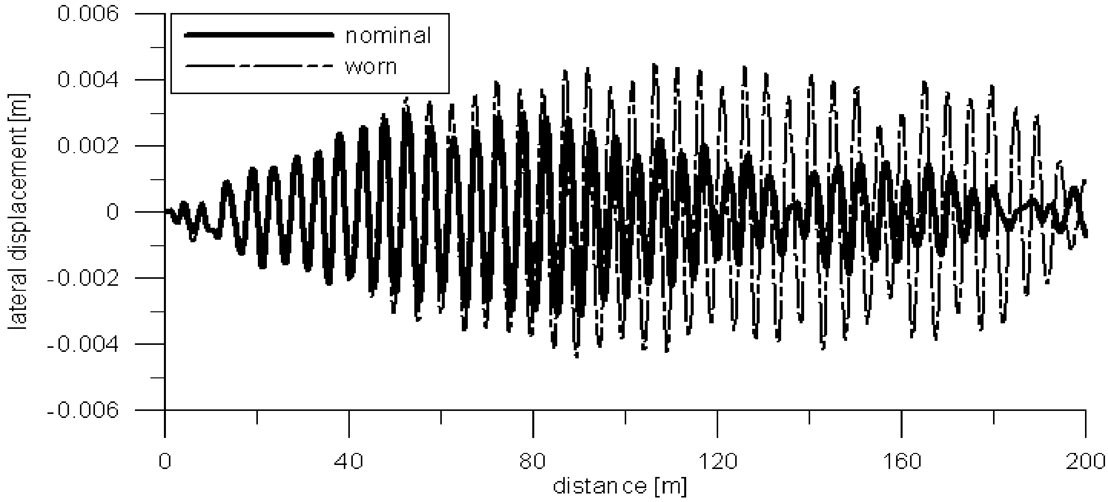

Figure 6 – Lateral displacement of the leading wheelset with nominal and worn wheel profiles. Vehicle velocity 100km/h, axle load ***180kN****.*

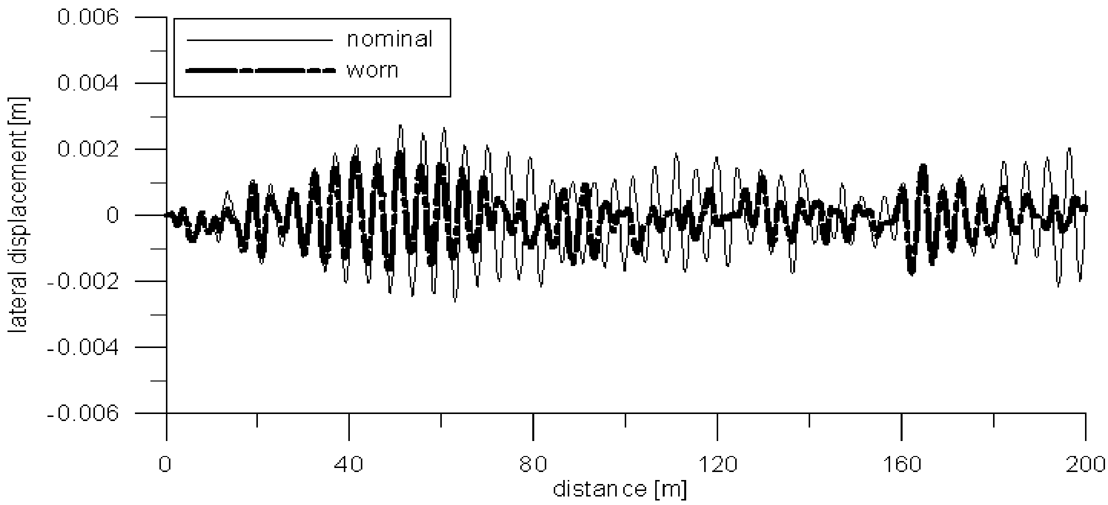

Figure 7 – Lateral displacement of the leading wheelset with nominal and worn wheel profiles. Vehicle velocity 100km/h, axle load ***250kN****.*

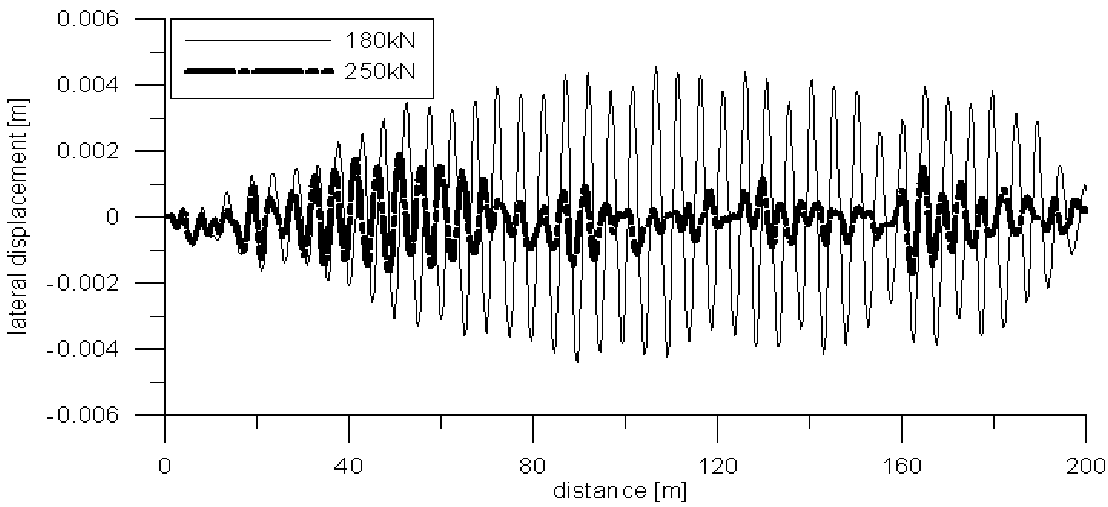

Figure 8 – Lateral displacement of the leading wheelset with **worn** *wheel profiles. Vehicle velocity 100km/h, axle load* ***180kN*** *and* ***250kN****.*

It can be noticed that the oscillation frequency of the wheelset lateral motion at different loads and different conditions of the wheels profiles has changed rather insignificantly while the amplitude change can be considerable.

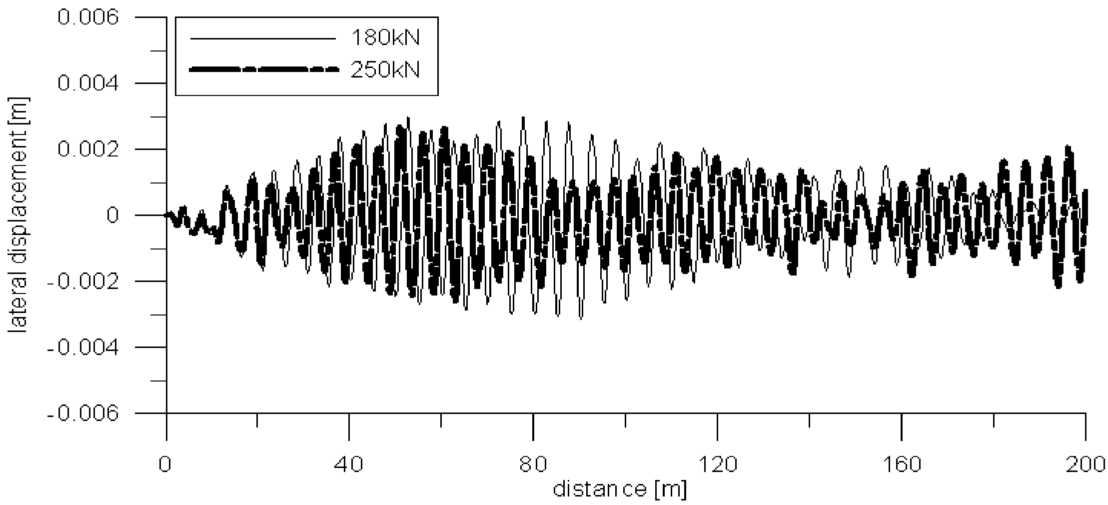

Figure 9 – Lateral displacement of the leading wheelset with ***nominal*** *wheel profiles. Vehicle velocity 100km/h, axle load* ***180kN*** *and* ***250kN****.*

The geometric quality level of the track model used in the simulation studies matches the category QN2-QN3 according to UIC518 code [14]. The authors have also carried out a simulation research using the track model of a better geometric quality, eventually it has been noticed that the influence of the wheels and the rails rolling surfaces condition on the vehicle dynamics (especially the lateral wheel-rail interaction forces and resulting displacements) is greater on the better track whereas on the poor quality track this influence is less distinguishable.

SUMMARY

It has been found that the wear of rolling surfaces significantly influences the location of contact points and consequently changes the system dynamics by generating tangential forces which are different than forces acting between nominal profiles. Exemplary results of the simulations studies show that varying static axle load may have significant influence on the stability of wheelset lateral motion and running safety.

REFERENCES

[1] Cooperrider N.K. Law E.H., Hull R., Kadalla P.S., Tuten G.M., "Analytical and Experimental Determination of Nonlinear Wheel–Rail Geometric Constraints", Int. Rep. on Contact DOT–OS–40018, 1985

[2] Decuyper M.: ,"Geometry of Wheel–Rail Contact: Extended Envelope Method", Int. Journal of Vehicle Design, Vol. 4. 1989

[3] Droździel J., Sowiński B. ,"Wheelset-track interaction influenced by variable rail profile", 7th Mini Conference on Vehicle Dynamics, Identification and Anomalies, VSDIA 2000, 6-8 November 2000, Budapest University of Technology and Economics

[4] Elkins J. Wu H., "Angle of Attack and Distance-Based Criteria for Flange Climb Derailment", The Dynamics of Vehicle on Roads and on Tracks, Supplement to Vehicle System Dynamics, vol. 33, 1999 pp. 293-305

[5] Gilchrist A.O., Brickle B.V., "A Re-examination of the Proneness to Derailment of the Railway Vehicle", Journal of Mechanical Engineering Science vol. 18 no 3 1976

[6] Aknin P., Ayasse J.B., Chollet H., Maupu J.L., Gautier B., Paradinas M., "Quasistatic Derailment of Railway Vehicle, Comparison between Experimental and Simulation Results", The Dynamics of Vehicle on Roads and on Tracks, Supplement to Vehicle System Dynamics, vol. 23, 1993

[7] Clark R. A., Eickhoff B. M., Hunt G. A., "Prediction of the Dynamic Response of Vehicles to Lateral Track Irregularities", Proceedings of 7th IAVSD Symposium 1981

[8] Knothe K., Theiler A., Guney S. ,"Invesigation of Contact stresses on the Wheel/Rail System at steady State Curving", The Dynamics of Vehicle on Roads and on Tracks, Supplement to Vehicle System Dynamics, vol. 33, 1999

[9] Knothe K., Zhang J. ,"Statistical Linearization of Wheel/Rail Contact Nonlinearities for Investigation of Curving Behaviour with Random Track Irregularities", The Dynamics of Vehicle on Roads and on Tracks, Supplement to Vehicle System Dynamics, vol. 25, 1995

[10] De Pater A. D. "The Geometrical Contact Between Track and Wheelset", Vehicle System Dynamics, vol 17, pp 127-140, 1988

[11] Kalker J.J., "Three Dimensional Elastic Bodies in Rolling Contact", Kluwer Academic Publishers, Dordrecht/Boston/London, 1990.

[12] Kik W., Piotrowski J.: "A Fast Approximate Method to Calculate The Normal Load At Contact Between Wheel and Rail and Creep Forces During Rolling", 2th Mini Conference on Contact Mechanics and Wear of Rail/Wheel Systems, Budapest, 1996

[13] Palach O. "A Fast Wheel–Rail forces calculation computer code", The Dynamics of Vehicle on Roads and on Tracks, Supplement to Vehicle System Dynamics, vol. 33, 1999

[14] UIC Code 518, "Testing and approval of railway vehicles from the point of view of their dynamic behavior – Safety – Track fatigue – Ride quality", 2003