Simulation of Contact Problems in Railway Engineering *

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Abstract

The analytical and numerical simulation exhibits dynamic reasons of the generation of wheel corrugations. First, the non Coulomb friction law causes the oscillation of the rolling moment. Second, self–excited vibration of the contact forces is generated by the wave phenomena. At the end the enforced steady motion of the wheel center leads to unsteady motion of the wheel surface.

1 Introduction

In the railway transportation both the load carrying capacity of carriages and speed of trains increase. It causes new problems of exploitation: faster wear of rail surfaces and wheel rings. Circular geometry of wheels and plane surface of rail heads lose their perfection. Both on the rail head and the wheel ring wave–shape deformations can be observed. They are called corrugations. Even in low speed motion and light trains the result of successful deformation of steel rail by the wheel can be seen by the naked eye and requires frequent intervention of technical services. The improper wear results in considerable increase of noise. In cities noise generated by tramways or even underground trains negatively effects the environment. In long distance trains can be tiring for passengers. From the technological point of view spurious effects of mechanical phenomena shorten the life of large steel parts of mentioned mean of transportation.

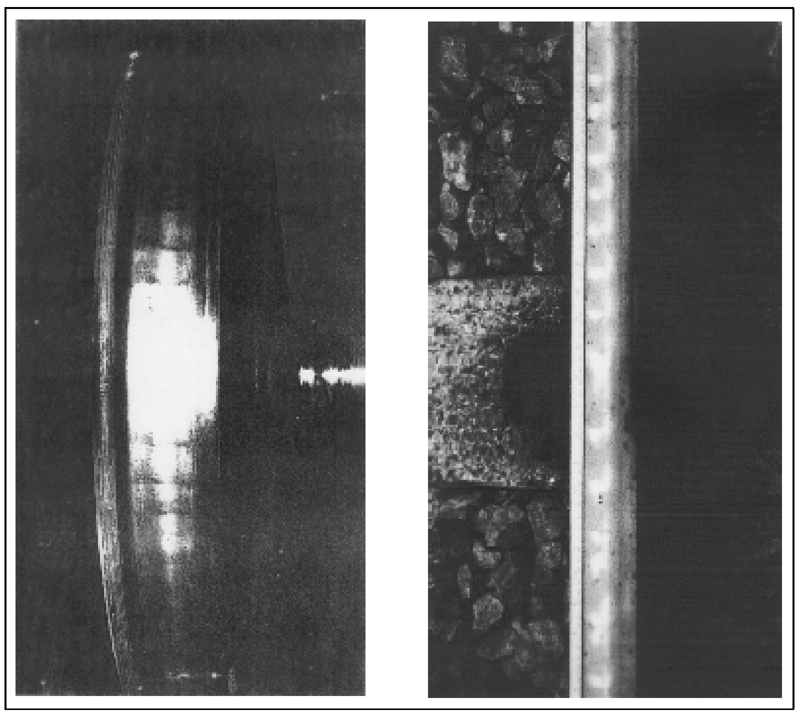

Figure 1: Corrugations of wheel and rail.

Another case where similar phenomenon occurs are vehicle breaks and clutches. High frequency oscillations generated between break shoes and disks or friction disks of the clutch considerably reduce the life time of elements. Besides a noise affects the environment by tones heavy to carry down.

The aim of the work is the simulation and investigation of generation of corrugations and its influence on the durability of rails. Particularly burdensome conditions will be in the scope: self excitation in higher (300 km/h) velocity range, influence of non–linear material properties (visco–plasticity), non-linear friction, torsional vibration of wheel/axle system, influence of plate bending

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state for cone–shaped wheel, approach to optimization of resulting parameters. The subject is wide and several research centers in the world work intensively in the field.

Both the polygonized contour of the wheel and the waved surface of the rail will be simulated in the selected tasks (Fig. 1). The process of the destruction of wheels and rails will be investigated.

The problem pointed is wide and has been undertaken in several research and technological centers in the world (USA, Japan, Germany, France). Different hypotheses were assumed as a base of investigation. Some of them can be easily rejected, others require intensive theoretical and numerical tests. In the literature the following cases are pointed as a source of corrugations:

- imperfections in rail joints,
- cone form of wheels which results in different linear speed of left and right wheel; it causes snaking of trains and generally, disturbs steady motion,
- periodical structure of rails (sleepers); instability of motion on the periodically placed supports [1],
- contact problems between wheel and rail; stick and slip sections which vary with high frequency (horizontally) generate waves which deform elastically, then plastically metal surfaces [2, 3, 4],
- residual stress caused by manufacturing and service of rails and wheels [5],
- non-linear friction law in the stick zone [3],
- influence of material hardening [6],
- deformation of elements of wheel/axle as results of impact during rolling motion,
- instability of wheelsets motion [7, 8].

2 Analytical modeling

There is a need to investigate the dynamical behavior and stability of the elastic model of wheelset in higher frequency range. One of the well known procedure used for higher frequency wheelset vibration analysis the finite element method. Unfortunately, a simple FEM approach yields to unsatisfactory description of the unstable regions and the interpretation of obtained results is unproper at whole.

A new analytical approach is presented in [9], In [10] the wheel tire is modeled as an elastic curved Rayleigh beam with constant curvature. In the application of the curved beam theory the cross–section of the tyre preserves its real shape. Taking into account the angular velocity enables to consider the beat phenomenon as well as bifurcation of frequencies during the rotating motion of wheelset. Similar phenomena occur in rails under action of oscillating moving loads [11]. The modelling of the wheel plates as a continuous, inertial, elastic foundation which carries loads in four directions (i.e. radial, longitudinal, torsional and lateral) is more difficult and limits the application of the model which depends on natural frequencies and eigenforms of the wheel plates.

The considered physical model consists of two elastic wheels joined together by a rigid axle. The wheels interact with the rails by means of linear Hertz springs, which also transfer the forces into three directions and the spin moment. The creepage during the motion of the wheelset is neglected in the presented paper.

The mathematical description of the considered model consists of the system of 14 nonlinear, coupled, partial differential–integral equations. They describe:

- wheel vibrations, including transverse vibration in two planes, circular and torsional vibrations of tyre (4 equations for each wheel),
- vibration of the wheelset mass centre (3 equations),
- rotational vibrations of the wheelset around its mass centre.

The elastic and inertial forces of the contact, including forces obtained from the Coriolis acceleration, are the source of coupling of the equations. If we assume the geommetrical symmetry of the wheelset and constant angular velocity, then the rolling motion along the rails can be described by the set of 9 equations. The analytical solution of the linearized set of equations is obtained by the Fourier method. The linear model is used to investigate the rolling motion stability.

Each component of Fourier trigonometric series satisfying the system of equations as well as boundary conditions is an eigenfunction of the boundary problem. From the physical point of view it is the mode of the wheel free vibration. In the considerations of wheelset dynamics the assumed conditions are the conditions of continuity. By means of the finite sine and cosine transforms the system of motion equations is transfered to the system of finite number of uniform algebric equations. The condition of existance of non trivial solutions has the form

$$\Delta(r) = 0 \ . \tag{1}$$

The roots of determinant (1) yield the frequencies of free vibrations of a wheelset rolling motion. In numerical investigations it is possible to take into account a finite number of modes for wheels.

3 Simulation of rolling contact process

Recently particular attention is payed to the problem of generation of corrugations. The proper solution requires the investigation of inelastic effects of the dynamic contact problem. The steady and nonsteady states of elastic strip rolling have been investigated in [12, 13, 3]. The rolling contact problem was formulated for large strain of material with isotropic hardening and various friction laws between a rigid roller and the strip surface. The updated Lagrangian approach was used to obtain the numerical solution. The problem was approximmated by minimization of a penalized functional which expresses the variation of virtual vork. The penalty parameters are associated with constraints of the contact and support conditions. A special penetration method was applied to the finite approximation of the deformed body in order to solve the nonlinear problem of two bodies coming into contact [12]. Results obtained for the case of the oscillating roller show the upper surface of the strip corrugated. The influence of the friction model on the tangent stress distribution and stick–slip area for various creepage was also studied in [3]. The results obtained for the case of generalized Coulomb model are shown in Fig. 2. The model is described by two parameters which corresponds to static and kinetic friction. Such a model is a limiting case of piece–wise linear dependence of the friction force on the velocity of relative motion. The numerical investigation of the rolling contact problem by means of the finite element method performed in [12, 13, 3] were not sufficient to determine the reasons for the phenomenon of corrugation formation. The high frequency stick–slip self–excited vibrations of roling motion are obtained for two models shown in Fig. 4. Only for the Coulomb model self–excitation is not visible. The roling moment for various friction models vs. rolling distance is depicted in Fig. 3.

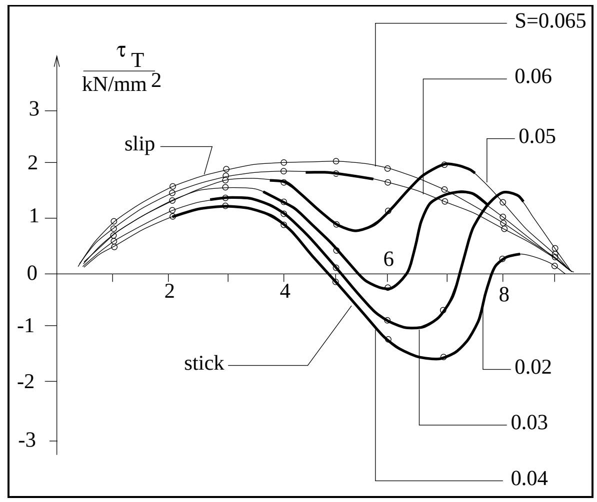

Figure 2: Tangential stress distribution in the contact area for various creepage S.

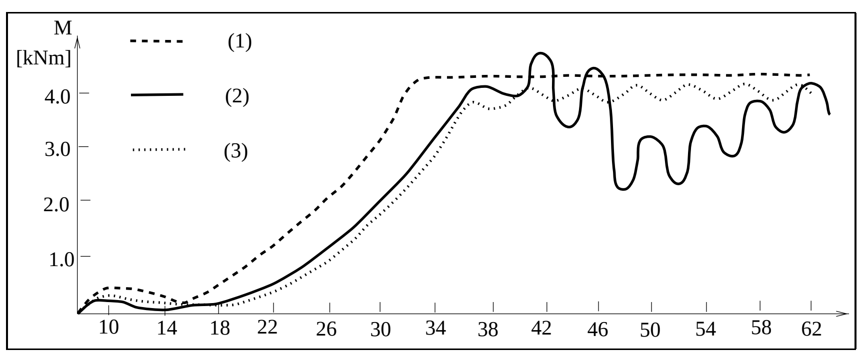

Figure 3: Rolling moment for various friction models vs. rolling distance.

4 Numerical analysis of the rolling contact

In the numerical analysis of the rolling contact problem we shall limit the investigation to the range where the contact occurs. The space–time finite element method was applied to the simulation of the corrugation generation [14]. Other factors such as friction, plastic deformation, hardening, can simply be added following the classical scheme. As an example we take the wheel with the radius R=10 cm, thickness 1 cm, made of steel (E=2.05·10^7 N/m^2, ν=0.3, ρ=7.83 g/cm^3). It rolls on the rigid base with an angular speed ω. The linear velocities taken into account were of the range 90–180 km/h. The elastic material in plane stress was assumed. The domain was discretized with 864 triangles and 469 nodes. The uniform mesh density was applied for the reason of wave nature of the process and stress concentration passing throughout the domain. To avoid multiple rotations of matrices effected by the rotation of the structure and in the same time the accumulation of round–off errors the rotation of the rigid base over the fixed wheel was assumed. All the forces arising from the circular motion were introduced. In the first stage the wheel, which turns is settled slowly on the rigid base (in numerical simulation the base which turns presses slightly the fixed wheel). The depth of penetration (flattening) reaches finally d=0.1 cm. In order to avoid the influence of the initial conditions and to reduce the effect of wave reflections and interference the comparatively large numerical damping was assumed. The value of the damping parameter γ was equal to 0.2 and it corresponded to the logarithmic decrement of damping $\Lambda = 0.03$. In practice it allowed to damp vibration according to the first eigenform and the period $T \approx 80$ μs in 95% during the first 1/4 turn of the wheel.

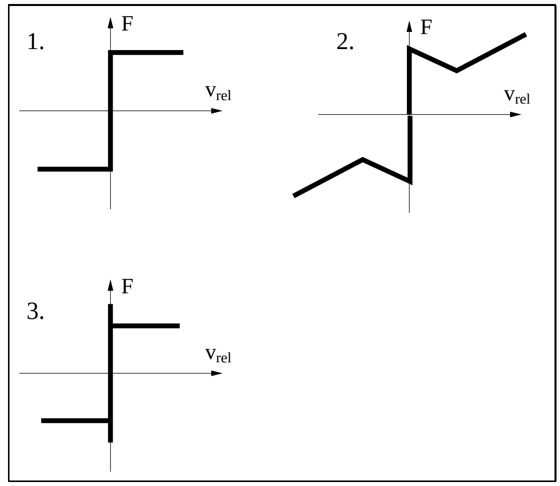

Figure 4: Various friction models.

Computation shows that the contact forces vary even when the motion is steady and well damped. A part of a turn of the wheel with ω=0.3·10^{-3} rad/s is presented in Fig. 5. The second invariant of stresses J_2 was integrated in successive phases of the full turn. It enables us to show the distribution of stresses in the material (Fig. 6). It exhibits the periodical distribution of the wear on the wheel surface which can occur during exploitation. The number of contact force oscillations decreases along with the increase of the speed. It was observed for example in [15] for a rubber wheel. However, in those publications the authors treat the problem as an eigenvalue problem. They do not solve the initial boundary problem. The value of the contact force increases with the increase of the velocity ω.

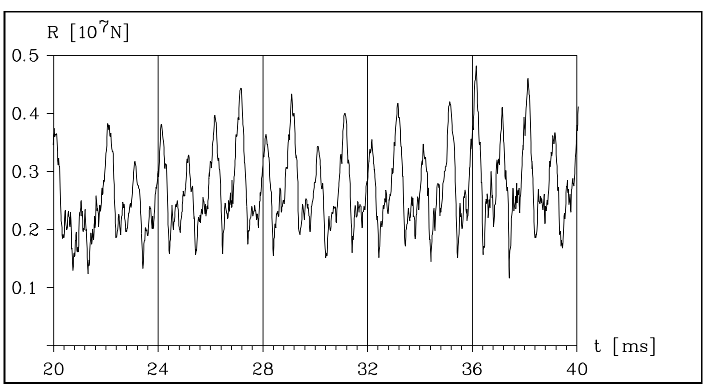

Figure 5: The part of the diagram of the contact force in time.

The investigation was performed for a full turn of the wheel. If the number of waves due to a turn is not an integer (i.e. the phase shift occurs after each turn), then the diagram is disturbed in the vicinity of the lower point of the wheel, from which the solution starts and on which is finished.

5 Conclusions

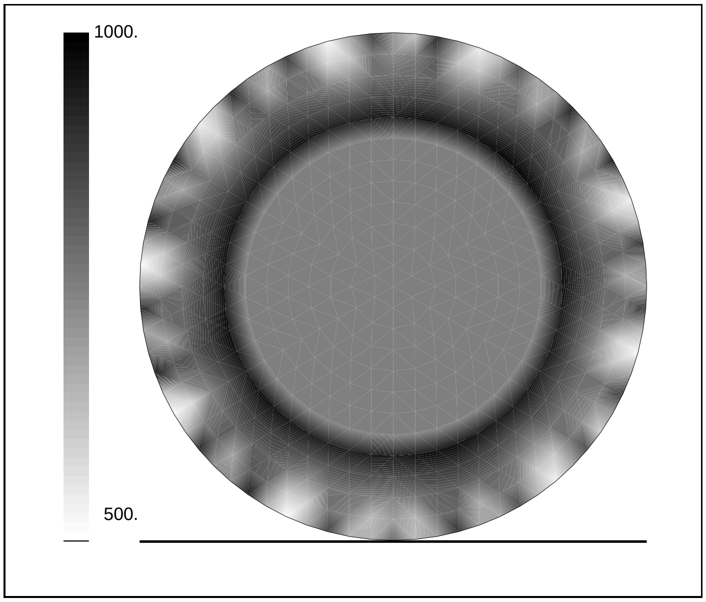

Figure 6: Averaged for the full turn stresses J_2 with the velocity ω=0,30$\cdot 10^{-3}$ rad/s [MPa].

Analytical and numerical investigations prove the complexity of the rolling contact problem. Nonlinearities, especially non Coulomb friction law, increase the oscillacions of rolling moment. The steady motion starts to pass into vibrating state. The phenomenon deepens by self–excited waves of Reileigh type on the wheel surface.

Bibliografia

[1] R. Bogacz, T. Krzyżyński, and K. Popp. Application of floquet's theorem high–speed train/track dynamics. In *Advanced automotive technologies, ASME Congres*, pages 55–61, 1995.

[2] C. Bajer, R. Bogacz, and C. Bonthoux. Adaptive space–time elements in the dynamic elastic–viscoplastic problem. *Comput. and Struct.*, 39:415–423, 1991.

[3] M. Brzozowski, R. Bogacz, and K. Popp. Zur reibungsmodellierung beim rollkontakt (on the modelling of wear in rolling contact). *ZAMM*, pages T678 – T680, 1990.

[4] R. Bogacz, M. Brzozowski, O. Mahrenholtz, and J. Rońda. Dynamic effects in a rolling contact problem. *ZAMM*, 67(4):T176–T179, 1987.

[5] R. Bogacz. Residual stresses in high–speed wheel/rail system; Shakedown and corrugations. In A.N.Koanadis, editor, *Proc. of the 1-st European Conference on Steel Structures EUROSTEEL'95*, pages 331–343, Athens, 1995. A,A Balkema.

[6] Cz. Bajer. Numeryczne modelowanie czasoprzestrzenne dynamicznych zagadnień kontaktowych. Prace IPPT, 5, Warszawa, 1997.

[7] P. Meinke and T. Szolc. On discrete–continuous modelling in the railway wheelsets for non-linear dynamic analysis in the medium frequency range. In *Proc. 2 Europ. Nonlinear Oscillation Conf., Euromech*, pages 135–138, Prague, 1996.

[8] R. Bogacz and S. Dżuła. Dynamics and stability of a wheelset in rolling contact motion on rails. In *Proc. of ITTG International Symposium on the Technological Innovation in Guided Transports*, pages 871–883, Lille, France, 1993.

[9] E. Bromundt. Oscillations and stability of a wheel roling on flexible rail. In R. Bogacz, J. Lückel, and K. Popp, editors, *Dynamical problems in mechanical systels*, pages 63–75. IPPT, Warsaw, 1991.

[10] K.G. Krapf. Der elastich gebettete Kreisring als model für den Gürtelreifen. Dissertation am Fachbereich Mechanik, Technischen Hochschule Darmstadt, 1979.

[11] R. Bogacz. On dynamics and stability of continuous system subjected to a distributed moving load. *Ingenieur Archiv*, pages 243–255, 1983.

[12] R. Bogacz, J. Rońda, and M. Brzozowski. Corrugations in rolling contact problems. *ZAMM*, 67(11):567–568, 1987.

[13] J. Rońda, O. Mahrenholtz, R. Bogacz, and M. Brzozowski. The rolling contact problem for elasto–plastic strip and rigid roller. *Mech. Res. Comm.*, pages 199–132, 1986.

[14] C.I. Bajer. The space-time approach to rail/wheel contact and corrugations problem. *Comp. Ass. Mech. Eng. Sci. (in print)*, 5(2), 1998.

[15] J.T. Oden and T.L. Lin. On the general rolling contact problem for finite deformations of viscoelastic cylinder. *Comput. Meth. Appl. Mech. Engng.*, 57:297–367, 1986.