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Modelling of non-steady-state transition from single-point to two-point rolling contact

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ABSTRACT

By considering real wheel and track geometry and structures, as well as their coupled interaction, an explicit finite element method is applied to simulate the transition of wheel–rail rolling from single-point tread contact to two-point tread-flange contact. The evolutions of the contact position, stress magnitude and direction, adhesion–slip distribution, and wheel–rail relative velocities in the two contact patches are considered to investigate the transient dynamic effects during the contact transition. Important findings include that the transition to two-point contact can result in friction saturation and excite waves in the contact, causing local intensification and relaxation of compression, as well as turbulence of the micro-slip; the local relative velocity in the contact patch is a good measure of the dynamic effects.

1. Introduction

Wheel/rail rolling contact is more complex on curves than on straight tracks. Because the leading wheelset often fails to align itself tangentially with the rails, an angle of attack (AoA) arises, and non-Hertzian two-point contact may occur on the high rail: the wheel tread contacts the rail top surface at the same time that the wheel flange contacts the rail gauge corner.

Complex two-point contact is difficult to model, as it involves arbitrary wheel and rail geometries, large creepage and spin. Traditional approaches based on the half-space assumption, such as Hertz's analytical method, virtual-penetration-based methods [1,2], and Kalker's programs FASTSIM and CONTACT [3], are considered inaccurate because the radii of curvature of the contact bodies at the flange–gauge corner contact can be of the same size as the contact patch and the patch may be non-planar. ANALYN [4] and FaStrip [5], though also limited to the half-space assumption, were proposed as alternatives to the virtual-penetration-based methods and FASTSIM, with higher accuracies in terms of both creep force estimation and shear stress distribution, especially when the contact patch is non-elliptic. By extending Kalker's CONTACT algorithm to quasi-quarter-spaces, Li [6] presented the program WEAR to cope with conformal contact, including two-point contact. Its application to contact between a wheel and the switch blade of a turnout is presented in [7].

As CONTACT and WEAR are based on the boundary element method [6], they can only treat linear-elastic materials and idealised contact geometry, limiting their accuracy in analyses of wear and rolling contact fatigue (RCF) with large plastic deformation and arbitrary contact geometry. The use of the finite element method (FEM) was proposed to overcome these problems. The elastic half-space contact solutions calculated by the FEM correspond well with those obtained from Hertz contact theory [8]. The FEM was also proven to be applicable to two-point wheel–rail contact, as satisfying solutions have been obtained for contact points on the rail top and gauge corner [9–12]. A good integration of 3D FEM and multibody modelling is archived in [13] to have a full description of vehicle dynamics.

The models in [8–13] all assumed quasi-static state contact between the discretised wheel and rail models. However, dynamic effects may play significant roles under certain circumstances, such as impact contact at geometric irregularities of the wheel and rail surfaces. To consider such dynamic effects, the explicit FEM was adopted to simulate the fatigue and wear of the rail joint [14], the growth of squat [15] and damage to the crossing nose [16,17]. The transient elastic–plastic stress–strain states in the contact were calculated by employing elastoplastic materials in the models. By accounting for the normal load and creep force as well as a realistic contact geometry, the transient explicit FEM has been proven to be effective and accurate for solving single-point frictional rolling contact between the wheel tread and rail with small [18] or large spin [19] by comparing its quasi-static state solutions with those of Hertz and CONTACT. In comparison with the implicit FEM, the explicit integration scheme avoids the regularisation of the friction law [20] required to treat the no-slip condition in the adhesion area and the convergence difficulties caused by the demanding contact conditions [21]. Moreover, the computation

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efficiency is considerably improved when considering high-frequency vibrations.

In addition to impact, dynamic effects may also need to be considered when unstable vibration of the wheel (e.g., during curve negotiation) is present. In such a situation, increased wear, RCF and corrugation are expected [22], and a squeal noise may occur [23]. The occurrence of unstable wheel vibration can be significantly influenced by the transition from single-point to two-point contact during wheel curving through curved tracks.

The explicit FEM has been applied to the analysis of wear and RCF in wheel curving with two-point contact [24]. The effects of the AoA were considered, and the contact positions with certain AoAs and wheel/rail profiles could be determined. Another application of the explicit FEM to two-point contact was presented in [25] to simulate the wear process on a laboratory twin-disc rig; good agreement was achieved by comparing the simulation results with the experimental results. However, in both cases, no dynamic effects were considered in the simulations, as the wheel motion was considered to be in the steady state and no unstable vibration occurred.

In this paper, the transition from single-point to two-point rolling contact is studied by employing an explicit FEM. The transition is a non-steady state process and occurs in a short period of time. The appearance of the second contact point may cause sudden changes in the normal and tangential forces at the two contact patches, exciting waves and friction-induced unstable vibrations. First, a FE model is presented to reproduce such a transient dynamic process numerically. Then, the contact solutions are carefully examined in terms of the evolutions of the contact position, the stress magnitude and direction, the adhesion–slip distribution, and the wheel–rail relative velocities in the two contact patches. The analyses show that waves are indeed excited in the contact patches when two-point contact occurs; to the authors' knowledge, this conclusion has not been previously revealed by either quasi-static or dynamic wheel–rail contact solutions. The ability to analyse the generation and propagation of the waves in rolling contact will enable a better understanding of dynamic rolling and its resulting wear (including fretting [26]) and RCF and may provide a basis for the study of the 'elusive' short-pitch corrugation and 'erratic' squeal noise, both of which are consequences of frictional contact.

2. Modelling and validation of the wheel modes

As shown in Fig. 1(a), a 3D FE transient wheel–track interaction model is developed in which a 10 m length of half-track and a half-wheelset with the sprung mass of the car body and bogie are considered. The wheel, rail and sleepers are modelled using 8-node solid elements. To achieve a highly accurate solution with a reasonable model size, regular discretization is allocated at the wheel–rail contact areas and non-uniform meshing is used. The mesh size around the initial position of the wheel–rail contact and the 150-mm length of the solution zone is 1 mm. The lumped mass of the car body and bogie are modelled as mass elements connected to the wheelset by the primary suspension of the vehicle with parallel linear springs and viscous dampers. Each sleeper contains 12 solid elements, and the ballast is simplified as vertical spring and damper elements, with the displacements constrained in the lateral and longitudinal directions. The parameters involved in the track model are mainly taken from [27]. The wheel–rail contact is defined with real geometry, including the wheel flange and rail gauge corner, as shown in Fig. 1(b), enabling the creepage and spin motion caused by the flange rubbing to be fully considered. The wheel geometry corresponds to a passenger car wheel of the Dutch railway with the standard profile

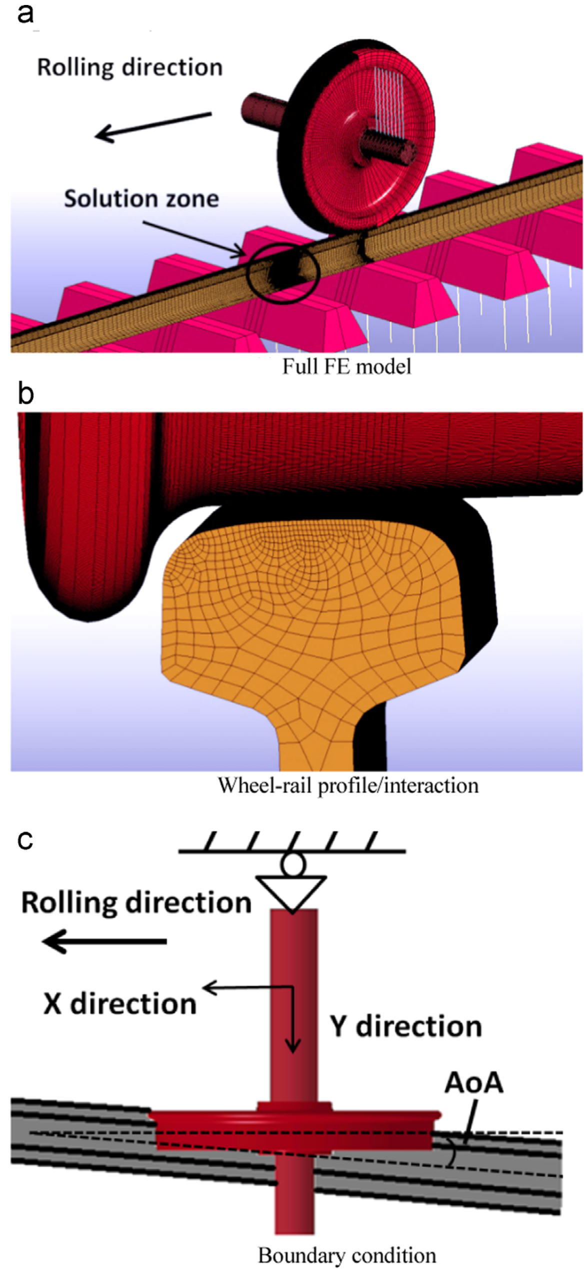

Fig. 1. Wheel–rail interaction model.

of S1002; the rail is UIC54E1 with an inclination of 1:40. No geometry irregularities are considered on the surfaces of the wheel and rail. The friction coefficient is set to 0.6 to represent dry friction.

The non-steady-state transition from single-point to two-point contact is simulated by applying the boundary condition to the wheel in the manner shown in Fig. 1(c). The lateral displacement is constrained at the inner side of the wheel axle, and the outer end of the axle is free. In the transient dynamic simulation, the wheel rolls from its initial position to the solution zone with an initial speed of 80 km/h along the X direction, forming an AoA with respect to the rail longitudinal direction. It is driven by a torque applied on the axle, thus generating a longitudinal creep force between the wheel and rail, which satisfies the requirement that the traction coefficient be below the friction coefficient. The wheel rolls with single-point contact between the wheel tread and rail crown before entering the solution zone. The damping in the system dissipates the initial kinetic and potential energy