FEM modeling of wheel-rail rolling contact with friction in an eulerian frame: results and validation

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1 Introduction

In the case of a tight radius curve, squeal can be generated due to the unstable slip of the wheel on the head of the rail. The mechanism causing the instability is however still controversial. A negative slope of the friction coefficient depending on creepage velocity is introduced in some models as a source of the instability [1–8] whereas squeal can also occur in the case of constant friction due to "mode coupling" instability in other models [9–11].

There are mainly two types of model used for wheel/rail rolling contact in squeal simulations. The "point-contact" models are based on analytic formulas which are heuristic laws of contact friction [13, 14]. In slightly nonlinear dynamic cases (close to pure rolling), equivalent point stiffness and damping can be used and give good results, especially for the rolling noise modelling [15]. In other cases (large slides with dynamic contributions), "surface contact" models, where the contact zone is discretized, are more adapted. The variational theory of Kalker [16] implemented in the CONTACT program is the most used model of this type [10]. However, a number of simplifications are generally performed in these models (elastic half-space assumption, contact and friction decoupling). The impact of these simplifications is unknown in the case of vibrations in curves where instability due to friction and excitation by surface irregularities (at different scales) may occur.

In this paper, we focus on the modeling of theses friction-induced vibrations which requires the nonlinear dynamics produced in the contact zone. By using finite element method, discretization of the contact surface, non-smooth frictional contact laws and model reduction techniques in an Eulerian frame, we perform first a stability analysis to predict unstable modes and frequencies, but which assumes full steady sliding in the contact area. Then, we propose a transient calculation which allows to introduce nonlinearities and to determine amplitude of vibrations. Reduction bases including free-interface modes and static attachment modes are included to reduce computing duration.

2 Finite Element (FE) formulation

2.1 Description of the problem- Equations in an Eulerian frame

A contact between the two bodies, lower and upper denoted in figure 1) is considered. The system is in two domains $\Omega = \Omega_1 \cup \Omega_2$ with boundary $\partial\Omega$. $\Omega_u = \Omega_{u1} \cup \Omega_{u2}$ is the part of Ω where displacement $\mathbf{u_d}$ is prescribed. $\Omega_f = \Omega_{f1} \cup \Omega_{f2}$ is the part of Ω where surface load $\mathbf{f_s}$ is applied. In Ω, volume load $\mathbf{f_d}$ is applied. $\mathbf{S_c}$ is the potential contact interface. $u(x,y,z,t)$ and $v(x,y,z,t)$ are respectively noted as the displacement and velocity fields.

The z-axis is chosen to coincide with the common normal to the two surfaces at 0. The longitudinal x-axis is the rolling direction. The y-axis is chosen to be the lateral direction. The surfaces move through the contact zone with tangential velocities δV_{y1}, δV_{y1} and longitudinal velocities δV_{x1}, δV_{x1} . The bodies may also have angular velocities ω_{z1} and ω_{z2} about their common normal. In the absence of deformation and sliding, material particles of each surface move through the contact region parallel to the x-axis with a rolling speed V.

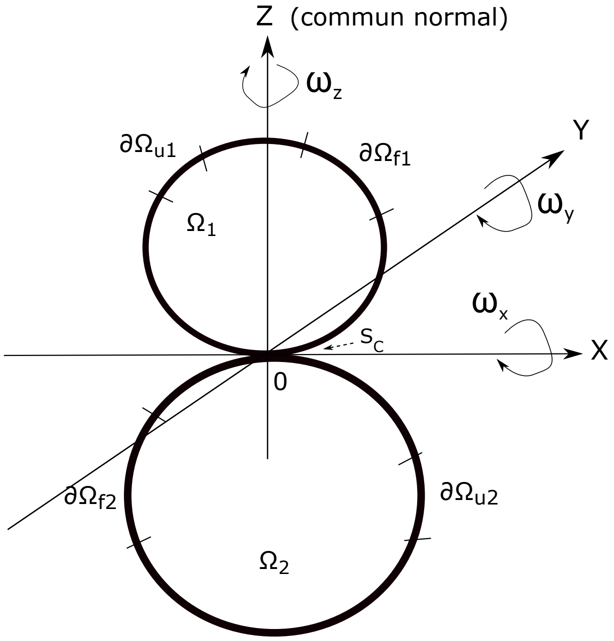

FIGURE 1 – Contact between two bodies

In the Eulerian frame which moves with the point of contact , according to [17], the creep velocities between contacting points are given by

$$\dot{s}_x(x,y,t) = v_{x1} - v_{x2} = \delta V_{x1} - \delta V_{x2} - (\omega_{z1} - \omega_{z2})y + V(\frac{\partial u_{x1}}{\partial x} - \frac{\partial u_{x2}}{\partial x}) + (\frac{\partial u_{x1}}{\partial t} - \frac{\partial u_{x2}}{\partial t}) \tag{1}$$

and

$$\dot{s}_y(x,y,t) = v_{y1} - v_{y2} = \delta V_{y1} - \delta V_{y2} + (\omega_{z1} - \omega_{z2})x + V(\frac{\partial u_{y1}}{\partial x} - \frac{\partial u_{y2}}{\partial x}) + (\frac{\partial u_{y1}}{\partial t} - \frac{\partial u_{y2}}{\partial t}) \tag{2}$$

2.2 Formulations and FE discretization

To deal with the unilateral contact and frictional contact , a non-smooth Signorini and a Coulomb law with a constant friction coefficient are chosen. The contact and friction laws can be rewritten in terms of projections on the negative real set $\mathrm{Proj}_{\mathbb{R}^-}$ and on the Coulomb cone $\mathrm{Proj}_{\mathbb{C}}$ [18]. The dynamics of the system is then given by the principle of virtual power and the contact laws are written in the weak forms [19]. Finite element discretization of theses equations gives then :

$$\begin{cases} \mathbf{M\ddot{U}} + \mathbf{C\dot{U}} + \mathbf{KU} = \mathbf{F} + \mathbf{P_n^T R_n} + \mathbf{P_t^T R_t} \\ \mathbf{R_n} = \int_{S_c} \mathbf{N_n^T} \mathrm{Proj}_{\mathbb{R}^-}(\mathbf{N_n}(\mathbf{H_n^{-1} R_n} - \rho_n \mathbf{U_n}))ds \\ \mathbf{R_t} = \int_{S_c} \mathbf{N_t^T} \mathrm{Proj}_{\mathbb{C}}(\mathbf{N_t}(\mathbf{H_t^{-1} R_t} - \rho_t \mathbf{\dot{S}_t}))ds \end{cases} \tag{3}$$

where $\mathbf{M}, \mathbf{C}, \mathbf{K}, \mathbf{U}, \mathbf{F}, \mathbf{R_n}, \mathbf{R_t}$ are respectively the mass, damping and rigidity matrix, are respectively the nodal displacement vector, the generalized force, normal contact reaction and tangent contact reaction. $\mathbf{N_n}, \mathbf{N_t}$ are shape function vectors on the contact interface, $\mathbf{H_n} = \int_{S_c} \mathbf{N_n^T N_n} ds, \mathbf{H_t} = \int_{S_c} \mathbf{N_t^T N_t} ds$ and $\mathbf{P_n}, \mathbf{P_t}$ are matrix allowing to pass the contact reactions from the local frame to the global frame. $\mathbf{U_n}$ is the gap and $\mathbf{\dot{S}_t}$ is the creep velocities at contact interface. ρ_n, ρ_t are are two arbitrary positive scalars called normal displacement augmentation parameter and tangential augmentation parameter respectively.

For the computation of the transient solution, with chosen no-smooth contact and friction laws , from the literature, a modified θ method is adapted. This is a first-order time integration scheme developed by Jean [18]. This scheme allows to impose inelastic shock. A fixed point statement on contact reactions is chosen for non-linear resolution of the system of equations eq. (3).

2.3 Reduction strategies

As the size of the system is often large and the nonlinear solving process implies several resolutions of a linear system at each time step, reducing the size of the system is necessary to obtain reasonable

TABLE 1 – Material and dimension data of the two cylinders

Young's modulus	205GPa
Poisson's ratio	0.3
Friction coefficient μ	0.3
Density (kg/m^3)	7800
Rayleigh's damping coefficients (α, β)	$(1, 10^{-6})$
Internal rayon	0.1m
External rayon	0.5m
Thickness	0.1m

computation time. An assumption is pursued. All the physical contact degrees of freedom are kept since there are strong nonlinearities on the contact zone. The proposed basis in the paper includes free-interface modes of the structure Φ and static attachment residual modes Φ_c. If the dynamics associated are then neglected, a total decoupling between the equations generalized corresponding to free-interface modes and the residual static contact equations is obtained. This model reduction gets close to the Pieringer's model [10]. The influence functions here are calculated by FE method.

3 Results

A contact between two identical cylinders with discretization of the contact surface is considered (Figures 2 and 3). The radii of cylinder transversal and longitudinal curvature is 0.5m. Rolling direction is the negative x direction. The material behavior assumed to be linearly elastic, isotropic and homogeneous solids undergoing small deformation. The material and dimension data of these two cylinders is listed in Table 1.

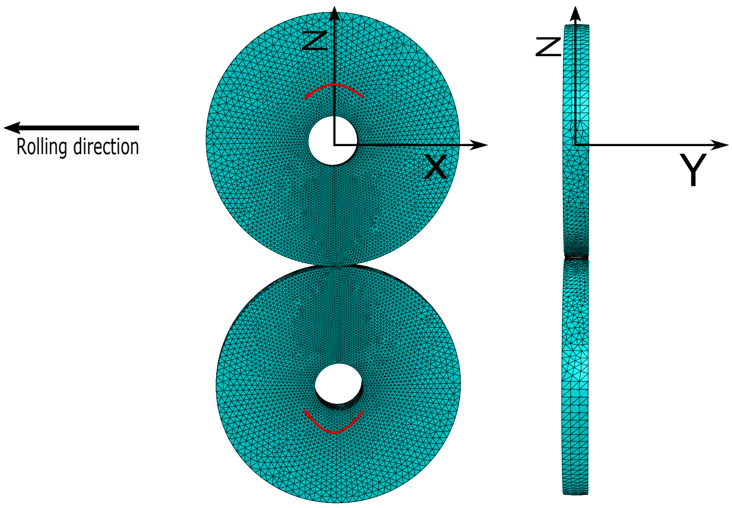

FIGURE 2 – Contact between 2 cylinders

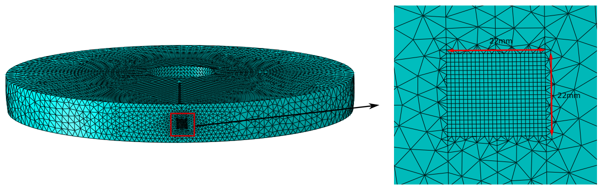

FIGURE 3 – Mesh of the 2 cylinders with discretization of the contact surface

The obtained results are first compared to Kalker's theory implemented in the CONTACT program in the quasi-static case. With an imposed lateral creepage of $s_{y0} = 0.3\%$ the tangential contact reaction at

$y = 0$ is presented in figure 4. The stick zone occurs at the leading edge of the contact and The slip zone occurs at the trailing edge of the contact. The comparison of tangent contact reaction obtained with both models shows that the FE model is in good agreement with CONTACT program.

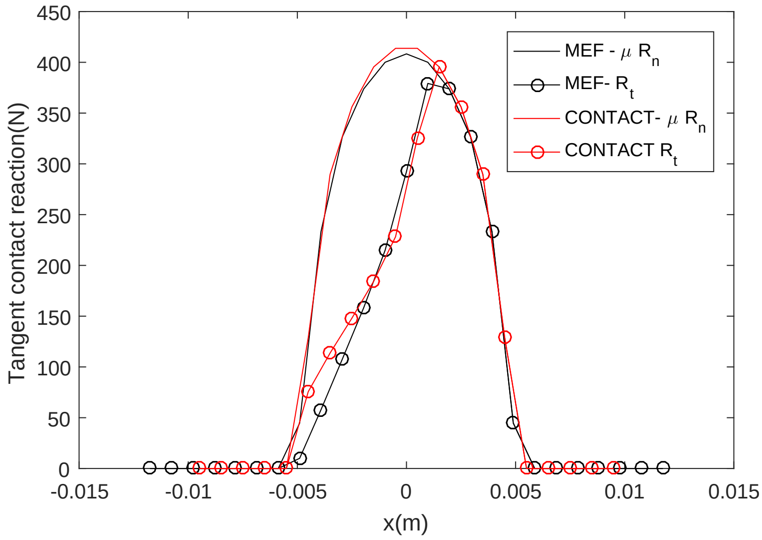

FIGURE 4 – Tangent contact reaction at $y = 0$

Application to narrow curves configuration shows that self-excited vibrations of the wheel / rail curve contact could occur due to mode coupling instability (with constant friction coefficient) . The results of transient dynamics exhibit are consistent with the stability analysis and exhibit the corresponding localized stick/slip oscillations in the contact zone.

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