SIMULATION OF WHEEL-RAIL WEAR BY NON-HERTZ ROLLING CONTACT

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Key Words Wheel, Rail, Wear, Rolling Contact, Simulation

Extended Abstract

Wheel-rail rolling contact exists in transportation systems such as railway, metro and tram. Rolling contact wear of wheel and rail profiles may cause a number of undesirable consequences. Increased wheel-rail interactive force, worsened vehicle dynamical performance and ride quality, freight damage, derailment, rolling contact surface fatigue, plastic flow, early wheel reprofiling, premature rail replacement, rail corrugation and deterioration of track are often related to wear.

There are a number of wheel-rail wear models relating material removal to various contact mechanical quantities. The most comprehensive and widely accepted is that the material removal V at a point due to wear is proportional to the material constant k and the sum of local frictional power P_i.

$$V = k \sum_{i=1}^{N} P_i \left(F, \nu_x, \nu_y, \varphi, f, \cdots\right) \tag{1}$$

where N is the contact frequency, namely the number of wheel-rail contacts which occur at a point on the wheel (or rail) profile after a distance is traversed by the wheel. F is the total contact force, ν_x and ν_y are the creepages in x and y direction, φ is the spin and f is the coefficient of friction.

In all those models wear is related to two important aspects. The first is the dynamics of the vehicle, which decides the wheel-rail interactive force, the creepage and the contact frequency. The second is contact mechanics, which , taking quantities from vehicle dynamics as inputs, determines the contact area, the traction, slip and frictional power distributions etc.

Accordingly the wheel-rail wear simulations carried out by now mainly fall in two groups[1,2] due to computational resource limitation. The first group pays more attention to vehicle dynamics, the operation conditions may also be included, while the treatment of contact mechanics is approximate. An example is [1] in which the material removal due to wear is assumed to be proportional to the Hertz pressure. The second group concentrates on contact mechanics, while the vehicle dynamics and the operational conditions are treated with some neglection. In [2] the contact frequency is simplified as a stochastic distribution plus a constant component and a sine component. Combing the two groups in one simulation is desired.

Even in the contact mechanics concentrated second group, the contact problem is assumed of Hertz type. Hertz contact is employed because its solution method is robust and its computing cost is low. Also its contact area can be determined easily, hence very suitable for automatical meshing in simulation. But in reality, not all the problems can be approximated by Hertz contact[3], and often non-Hertz contact is desired, especially when the wheel and rail are worn. Another drawback to carry out Hertz calculation is that the radii of curvature of the contact bodies must be known. In wear simulation, the profiles of wheel and rail are changing: they are updated after each wear step. To obtain the radii of curvature of the wheel and rail, the wear steps must be very small to avoid numerical instability, therefore the overall simulation speed is reduced.

In this paper, the wheel-rail wear simulation is carried out by using the precise full nonlinear non-Hertz rolling contact for the frictional power.

According to Kalker[4] a rolling contact problem can be expressed in the form of the maximisation of complementary energy in elastostatics, without body force, in surface mechanical form:

$$max\ C_{u,p} = -\int_{A_c} (h + \frac{1}{2}u_z)\, p_z\, dS - \int_{A_c} (W_\tau + \frac{1}{2}u_\tau - u'_\tau)\, p_\tau\, dS \tag{2}$$

$sub\ p_z \geq 0,\ |p_\tau| \leq f\, p_z$ in A_c, the tangential traction bound $g = f\, p_z$ is fixed

where C is the complementary energy. u is the displacement difference of contact particles. p is the traction. A_c denotes the potential contact area, which should include the real contact area. h is the undeformed distances, i and j indicate directions of the tractions or displacements, $i, j = x, y, z$. x, y, z are the coordinate directions of the curvilinear orthogonal coordinate system xyz. W_τ is the rigid body shift. τ indicates tangential (x, y) direction. dS is the area of a surface element.

Discretize A_c into a mesh of N equal rectangular elements with MX rows and MY columns, each element with area dS, (2) can be reduced into

$$C^* = -C;\ \ min\ C^*_{p_{Jj}} = \frac{1}{2} p_{Ii}\, A_{IiJj}\, p_{Jj}\, dS + \{h_J\, p_{Jz} + (W_{J\tau} - u'_{J\tau})\, p_{J\tau}\} \tag{3}$$

$sub\ \ p_{Jz} \geq 0,\ |p_{J\tau}| \leq f\, p_z,\ g_J$ is fixed

where A_{IiJj} is called the influence number. It represents the contribution of a unit load density in $j-$direction on element J to the $i-$component of the displacement difference in the center of element I. It can be shown that A_{IiJj} is symmetric about(Ii) and (Jj), that is $A_{(Ii)(Jj)} = A_{(Jj)(Ii)}$. If the bodies are fixed at infinity and their surfaces are free of traction outside the contact area, then $\frac{1}{2}\, p_{Ii}\, A_{IiJj}\, p_{Jj}$ is approximately the elastic energy, it is positive definite, and presumably $|A_{IiJj}| > 0$. It can be shown that C^* is strictly convex.

The constrained minimisation problem of equation (3) is reduced to a set of non-linear equations by Lagrange multiplier method. These equations are further decomposed into the normal and the tangential problems and solved iteratively, which not only reduces the computing time, but also satisfies the requirement that the tangential traction bound $g = f\,p_z$ is fixed in each step of the iteration. Under normal circumstances, a fast Gauss-Seidel method using block variants is employed. When this method goes wrong, the more reliable Gauss elimination method with LU decomposition is automatically switched on. Newton-Raphson linearization and Kalker's active set algorithm[4] are employed. The Newton-Raphson process and the active set algorithm may in some cases not converge, this is overcome by a perturbation method.

For non-Hertz contact, the contact area must be determined numerically in the solution process of the contact problem. A good initial estimation can significantly accelerate the convergence. The initial estimation is performed automatically based on the undeformed distance and the previously solved problems, satisfactory result is obtained.

In a wear step, the inputs to the contact problems are obtained from vehicle dynamical simulation. When the contact problems are solved, the material removal is obtained from (1) for every point along the wheel and rail profiles. The profiles are updated and the next wear step begin on the new profiles.

In summary, in this paper, the wheel-rail wear simulation is carried out by using the precise full nonlinear non-Hertz rolling contact for the frictional power. A number of numerical techniques are employed to speed up the algorithm. The overall simulation speed achieved is comparable to that by Hertz contact, and it can be improved even faster. Due to this it is now possible to combine both the full nonlinear non-Hertz rolling contact theory with vehicle dynamics to perform complete wear simulation.

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