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The influence of high temperature due to high adhesion condition on rail damage

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ABSTRACT

In wheel–rail contact, the locomotive adhesion variable characterizes the capability of the locomotive to convert available friction into traction at the interface. Recently developed AC (Alternating Current) drive induces a higher adhesion level compared to DC (Direct Current) drive. This can significantly affect the wheel–rail contact conditions such as high contact temperature due to the frictional rolling, wear and damage initiation of the rails.

The aims of this paper are to determine the temperature rise due to high adhesion contact and the thermal influence on the wear and rail life. Three-dimensional (3D) elasto-plastic finite element model was applied to evaluate the growth of temperature, residual stress and strain. The numerical model employed the moving heat source code developed by Goldak within ANSYS/LS-DYNA. The mechanical and thermal properties of the rail material were governed by temperature. The influence of multi-passes from multiple wheels attached to the locomotives on one point of the rail was also taken into account.

The results indicated that after six wheel passes, the temperature due to the high adhesion condition was sufficiently high (723 °C) to form the white etching layer (WEL) known to be associated to the rolling contact fatigue (RCF) on the rail surface. Moreover the rail material would be softened by high temperature, which resulted in the acceleration of wear process. Finally the results of thermal stress and strain from FE model were used as input to Kapoor's ratcheting model to determine the number of wheel rolling cycles leading to rail damage.

1. Introduction

The AC (Alternating Current) and DC (Direct Current) locomotives are electrical locomotives energized by electricity from the overhead lines, a third rail or an energy cache on the vehicle. The newer AC drive produces a higher adhesion level in comparison to the conventional DC drive because of the better design and the flexible current [1]. If using AC traction systems, a smaller number of locomotives are required to perform a specific task. Moreover they can help to reduce both fuel consumption and maintenance requirement [2]. However the high adhesion condition by AC drive possibly results in high stresses, heat and initiates damage prematurely [3]. In Australia, the research of locomotive adhesion was carried out in cooperation with industry within the framework of a railway research project R3.119 [4]; thus far, wheel–rail contact related problems under high adhesion condition have been the focus of the project. The adhesion induced heat generation and its influence on the rail life are reported in this paper.

Wheel–rail contact zone can be divided into stick and slip regions, and such divisions depend on the traction forces, creepage and the shape of the contact zone [5,6]. If the traction or braking forces exceed the adhesion strength, gross slipping occurs and generates significant heat at the contact spot [7]. Under high adhesion mode, the predicted temperature can be noticeably high and this level of temperature can cause detrimental structural alterations of rail material, wear and eventually failure of the wheel and/or rail.

Contact temperatures under actual railway operating conditions can hardly be measured, so experimental approaches such as pin-on-disc test [8,9], twin-disc experiment [10], braked block on wheel tread [11], or electrically-heated railway wheel set [12] have been utilized. Other than that, the temperature rise in the contact area can also be calculated either by the analytical method [13–17] or by the numerical method [18–23].

Ertz and Knothe [15] analytically and numerically calculated the temperature rise on and below the contact surface based on Hertzian contact, whereby the heat conduction from the wheel to the rail was taken into account. An approximate analytical solution for line contact model was further discussed in [17]. Fischer

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et al. [13] applied the Laplace transformer technique and thermal stress induced by the frictional contact in an analytical approach [24] to determine the temperature field. The combined analytical model proposed in [15] and 'brick' model [14] were applied by Widiyatar to examine the thermal influence on wear and rolling contact fatigue.

In addition to the analytical models mentioned above, there have been several numerical models of thermal calculation. Lunden [23] proposed a 2D thermo-elastic-plastic finite element model (FE) to study the plastic stresses and strains in the contact area of the wheel. Spiryagin et al. [22] recommended a mathematical model to calculate the temperature in the wheel–rail flange contact to detect the lubricant type at this contact location. Numerical algorithm was applied by Chudzikiewicz et al. [21] to build up a thermoelastic model with elastic graded materials to investigate the frictional heat generation and heat transfer across the contact surface. Another 2D FE model built up by Wu et al. [20] was utilized to investigate the residual deformation, plastic strain and residual stress at the rail surface during wheel–rail sliding contact. FE models employing moving heat source method to calculate the wheel temperature could be found in [19,25,26]. Very few 3D models were proposed [19,27]. These models were limited to either the temperature for wheel only or the finite element mesh was very coarse.

Studies of railroad have shown that the wheel operating temperatures can normally vary between 100 °C and 300 °C [12], and during sliding contact it reach 600 °C [24,28] in sliding contact. The rail temperature also depends on the sliding velocity and also is as high as 630 –1000 °C [20,29]. Furthermore, one point on the rail is normally subjected to a repeated loading from many wheels. The temperature field on the rail, therefore, increases with every cycle of rolling [18] which can result in the formation of a 10–100 μm thick layer on the contact surface [30]. This layer known as white etching layer (WEL) [31] due to the white appearance after etching in Nital is brittle, and can be the spot for crack initiation. This layer is formed at about 720 °C [32] but high hydrostatic pressure reduces the temperature at which this later can form [33]. Caroll et al. [34,35] simulated the WEL in the laboratory by two different ways: spot welding and rolling/sliding test machine. The investigation drawn a conclusion that the cracks initiated at the surface propagated rapidly within the WEL due to its brittle behaviour to the bulk material and caused the failure of the rail. Seo [31] conducted an analysis on the damage of WEL by two-dimensional (2D) FE model. The obtained results showed that the fatigue life at the end point of WEL is longer than those at the middle and starting sites.

It was found that temperature has a significant influence on the wheel–rail contact stress [24], and the combination of thermal stress and thermal softening enhances the rail wear rate [16]. Hence, temperature effects on the rail life need to be investigated thoroughly. From the literature review, most of the numerical approaches for thermal problems up to now have been carried out in two-dimension. Since wheel–rail contact is non-symmetrical problem, a 3D model is required to examine the behaviour of thermal stress by frictional rolling. This paper introduced a 3D thermo-elasto-plastic finite element wheel–rail contact model to calculate the temperature field and thermal stress induced by frictional rolling under high adhesion condition. The moving heat source code developed by Goldak et al. [36] was implemented in the current FE model and the temperature-dependent material properties were also applied. Repeated rolling of many wheels was considered to evaluate its influences on the temperature, thermal residual stress and the formation of white etching layer on the rail. From the obtained FE results, the discussion of thermal influence on wear of the rail material and the estimation of the rail life based on Kapoor's ratcheting model [37–39] were also provided.

2. Numerical modelling

In this study, the finite element model with moving heat source was developed with the finite element software ANSYS/LS-DYNA 14 and LS-PREPOST 4.2. Details of the model are presented in the following sub-sections.

2.1. Model of wheel–rail contact

When a wheel rolls on a rail, the contact area and the pressure distribution can be obtained by the Hertz theory [40,41]. The normal pressure distribution is determined as follows:

$$P(x', y') = P_0\sqrt{1 - \frac{x'^2}{a^2} - \frac{y'^2}{b^2}} \tag{1}$$

$$\begin{cases} x' = x - x_0 \\ y' = y - y_0 \end{cases} \tag{2}$$

where x' and y' are the local longitudinal and lateral coordinates, respectively; x_0 and y_0 denote the coordinates of the centre of the contact area; a and b are the semi-axis of the contact area in the x and y direction, respectively. The maximum pressure P_0 is calculated from the normal contact force F_Z by the following equation:

$$P_0 = \frac{3F_Z}{2\pi ab} \tag{3}$$

Based on the Coulomb's friction law [42], the distribution of the tangential stress is given as follows:

$$\tau(x, y) = \mu P(x, y) \tag{4}$$

where μ is the friction coefficient. The heat flux distribution within the contact patch due to friction is given as

$$q(x, y) = \mu v_s P(x, y) \tag{5}$$

where v_s is the sliding velocity. The heat partition δ=0.5 is assumed. This assumes that the heat generated is equally transferred to the wheel and the rail. However, the temperature of the rail can still potentially be increased under repeated wheel passes [20], resulting in material softening. The heat flux absorbed by the rail surface is calculated by

$$q_r(x, y) = \delta q(x, y) \tag{6}$$

The heat generation along the rail is similar to the moving of a heat source along a steel bar. Hence the moving heat source technique was applied in this study to model the thermal problem of wheel–rail contact. Goldak's heat source model [36] based on a Gaussian distribution of power density was chosen. The size and shape of the heat source are assumed to be an ellipsoidal energy deposition profile centred at the heat source, and decay exponentially with distance in this model. As the shape of the wheel–rail contact area is also elliptical [5,43], Goldak's model therefore is suitable to apply in this study. The mathematical expression of Goldak's model is described as follows [44]:

$$\dot{q}''_{(x,y,z)} = \frac{6\sqrt{3}\dot{q}}{r_x r_y r_z \pi\sqrt{\pi}} e^{-\left(\frac{3x^2}{r_x^2}\right)} e^{-\left(\frac{3y^2}{r_y^2}\right)} e^{-\left(\frac{3z^2}{r_z^2}\right)} \tag{7}$$

where r_x, r_y, r_z are the semi-axis of the heat source. Semi-axis in the z direction is the penetration depth of the heat source. The heat rate $\dot{q}$ is substituted by the frictional heat flux calculated from Eq. (6). The complete equations of Goldak's model can be found in [36].

2.2. Finite element model

The schematic of the finite element model is illustrated in Fig. 1. Simulation was carried out for 100-mm length, 50-mm