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Traction-creepage curve identification at the wheel/rail interface: A fast experimental approach

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ABSTRACT

The development and refinement of theoretical and empirical models of the traction-creepage relationship in rolling contact (and more specifically wheel/rail contact) over many years by numerous researchers has resulted in a comprehensive understanding of this relationship, and a corresponding ability to incorporate a sophisticated representation of rolling friction in applications such as vehicle dynamics simulation and wheel/rail damage prediction. These tools can be highly useful when considering and examining the potential impacts of various materials used to control friction between the wheel and rail in pursuing a range of corresponding benefits. With that said, obtaining detailed traction-creepage data for specific combinations of wheel/rail materials and third-body layer contaminants (required to enable simulations) has remained somewhat involved and intensive (to the extent that the availability of this data is quite limited). In this paper, a novel measurement approach to identify the traction-creepage curve under various contact conditions is developed and evaluated. The approach has the potential to generate meaningful traction-creepage curves quickly (within a fraction of a minute) and with simplicity. A comparison with the results from previous studies using conventional methods suggests that the proposed approach warrants future consideration.

1. Introduction and background

Friction at the wheel/rail interface can have a significant impact on many aspects of railway operations including vehicle dynamics, wheel/rail wear, locomotive energy requirements and curving noise. As an example, by managing the friction level between the gauge face/wheel flange (using traditional lubricants) and the top of rail/wheel tread (using friction modifiers) significant improvements in wheel/rail life, noise levels and fuel consumption can be achieved [1–3]. In order to understand and quantify the benefits friction management can have on a railway's operations, vehicle dynamics simulations are increasingly employed to obtain accurate predictions of wheel/rail interaction forces. The utility of these simulation tools in evaluating the impacts of various materials (third body layer constituents) is dependent on the incorporation of an accurate representation of the underlying traction-creepage relationship, which dictates the development of friction forces (tractions) as a function of lateral, longitudinal and spin creep (microslip) at the wheel-rail interface.

The fundamental nature of rolling contact as it relates to wheel/rail interaction is well understood, owing much to the foundational work of Johnson [4] and Kalker [5], as well as many subsequent works to further explore and refine both the fundamental understanding (e.g. of aspects such as the third body layer) and corresponding mathematical formulae and models [6–9]. In addition, the incorporation of solutions to rolling contact problems into practical computational algorithms [10–13] has underpinned the subsequent development of a range of simulation tools that are regularly used in both research and industrial settings.

Although the mathematical models available to represent rolling friction are quite capable of representing the subtleties of the traction-creepage relationship under different frictional conditions, these subtleties are generally not well reflected or incorporated into vehicle dynamics simulations. This can, in part, be attributed to the level of experimental effort associated with generating data that accurately represents the traction-creepage relationship for a given set of conditions (including representative wheel/rail materials and third body layer composition) [14,15]. The lack of resolution in differentiating between frictional conditions when simulating wheel/rail interaction can lead to limitations in these simulations [16]. As rail vehicle dynamics simulations are increasingly relied upon for accurate predictions, the applied traction-creepage characteristics within the model need to represent the actual, physical conditions as accurately as possible.

Wheel and rail wear rates are strongly dependent on contact conditions at the wheel/rail interface including creepage and creep

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forces. Several computational methods have been developed to predict wheel/rail wear [17], with many assuming that the volume of material loss is proportional to the frictional energy dissipated in the contact patch, T_γ, [18,19]. Accurate calculation of the contact patch frictional energy requires creep curves for different contact conditions. In [20], it is shown that not considering friction variation with creepage leads to large errors in contact force calculations. On the other hand, relating the contact patch frictional energy to wear rate is also dependent on traction-creepage curve, [21].

Including accurate traction-creepage information in the wear rate calculation results in a more realistic evaluation of effect of friction modifiers on wear rate as discussed in [22].

This paper describes work developing a simplified methodology for extracting traction-creepage curves under these varying conditions.

2. Traction-creepage measurement methodology

The measurement apparatus consists of two discs where one is driven by an electromotor and the other is idle. The two discs are loaded against one another with a controlled contact force. The discs are equipped with precision encoders in order to measure their angular displacement and allowing for the derivation of their respective velocities and accelerations. Applying a torque (acceleration) to the driven disc will generate a traction force at the interface of the discs, resulting in a transferred torque on the second (idle) disc and a corresponding acceleration. The acceleration of the second disc will be determined by its rotary inertia and the magnitude of transferred torque, as well any resistive losses in the disc's bearing arrangement. The transferred torque will, in turn, be dependent on the longitudinal creepage (velocity difference) between the discs. By measuring the relative velocity difference between discs and calculating the transferred torque based on the idle disc's acceleration, the underlying traction-creepage curve can be extracted.

The system configuration is shown in Fig. 1. In the figure, θ_1 and θ_2 are the angular displacements of disc 1 and disc 2 respectively. Disc 1 is driven with a controlled torque while disc 2 is idle and in contact with disc 1 applying the normal contact force, N. The governing equations of motion for this system are:

$$J_1\ddot{\theta}_1 + c_1\dot{\theta}_1 = T_e - T_{f_1} \quad (1)$$

$$J_2\ddot{\theta}_2 + c_2\dot{\theta}_2 = T_{f_2} \quad (2)$$

where T_{f_1} and T_{f_2} are torques on disc 1 and disc 2 due to the friction at the interface. J_1 and J_2 are the angular moments of inertia of the discs. c_1 and c_2 are the angular damping coefficients and T_e is the external torque applied on disc 1.

The frictional torque on disc 2 is a function of the normal contact force, N, rolling traction coefficient, μ_T, and the radius of disc 2, r_2.

$$T_{f_2} = \mu_T N r_2 \quad (3)$$

Having measured $\theta_1(t)$ and $\theta_2(t)$, the first and second time derivatives of angular displacements, $\dot{\theta}_1(t)$, $\dot{\theta}_2(t)$, $\ddot{\theta}_1(t)$ and $\ddot{\theta}_2(t)$ are calculated.

Consequently creep percentage, $\nu(t)$ can be calculated as:

$$\nu(t) = \frac{r_1\dot{\theta}_1(t) - r_2\dot{\theta}_2(t)}{\frac{1}{2}(r_1\dot{\theta}_1(t) + r_2\dot{\theta}_2(t))} \times 100 \quad (4)$$

Combining Eqs. (2) and (3) yields:

$$J_2\ddot{\theta}_2 + c_2\dot{\theta}_2 = \mu_T N r_2 \quad (5)$$

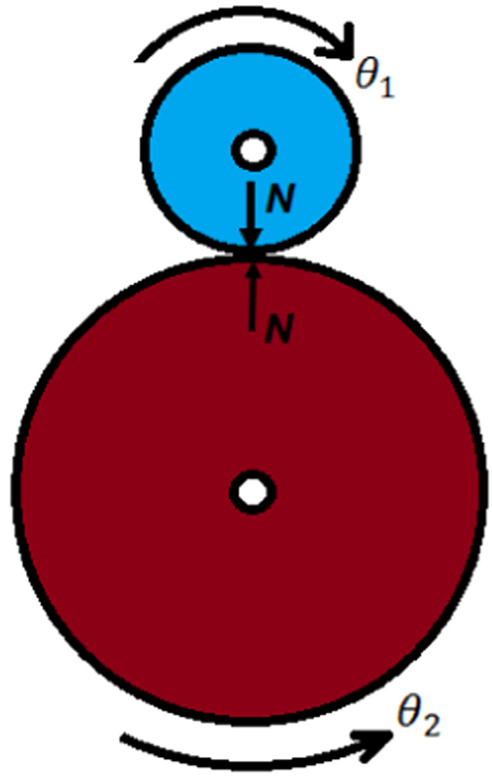

Fig. 1. Twin disc configuration.

Therefore the rolling traction coefficient in time, $\mu_T(t)$ can be calculated as:

$$\mu_T(t) = \frac{J_2\ddot{\theta}(t) - c_2\dot{\theta}_2(t)}{N r_2} \quad (6)$$

Using Eqs. (4) and (6), $\mu_T(t)$ can presented as a function of $\nu(t)$ to extract the traction-creepage curve.

3. Measurement system implementation

Shown in Fig. 2 is a twin disc machine equipped with two precision shaft encoders, which is used as a proof of concept tool in implementing the measurement methodology described above. The lower disc is coupled to an electromotor and can be driven over a range of velocities while the upper disc can rotate freely. The contact force at the interface of the two discs is controlled by a pneumatic cylinder. The applied force can be read and adjusted in the control interface of the machine.

3.1. Measurement system parameters, signals and calculations

To extract the traction-creepage curve, $\nu(t)$ in Eq. (4) and $\mu_T(t)$ in Eq. (6) need to be computed. The precision encoders noted previously are used to measuring angular displacements, $\theta_1(t)$ and $\theta_2(t)$. Velocities and accelerations are calculated by performing time derivation on angular displacements. However, the computed derivatives has to be low-pass filtered in order to remove discretization and computation errors. The first time derivative is calculated from the measured angular displacements and it is low-pass filtered using a zero-phase second-order section (biquad) filter. Filter parameters are obtained through a first-order Butterworth filter design with cut-off-frequency of 2.5 Hz. Zero-phase filtering is performed by processing the input data in both forward and reverse directions, [23]. The result of this filter has precisely