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Estimation of railway wheel running temperatures using a hybrid approach

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ABSTRACT

Accurate prediction of railway wheel temperatures, for a given running and braking history, is crucial for estimation of service induced thermal stresses which can affect wheel gauge and fatigue crack growth, particularly in tread region. In this work, a three step approach is adopted for estimating wheel temperatures. Firstly, a train running model employing brake block friction, wheel–rail traction-slip, and train running resistance characteristics are used to estimate heat generation rates at brake block–wheel and wheel–rail interfaces. Next, a two dimensional boundary element method, proposed in this work, is used to estimate heat partitioning at the said interfaces as a function of brake block type, geometry, and thermal properties. Lastly, finite element method is used to estimate wheel temperatures taking inputs from the train running model and the boundary element method. The adopted methodology is validated using wheel temperatures, on rim and rim–disc interface, from field trials of a locomotive fitted with cast-iron brake blocks. A good match between simulation and field trial results was obtained despite highly complex speed and braking patterns encountered in the field trial. For periodic braking with no stopovers, heat lost to the rail, cast-iron brake blocks, via radiation, and convection are seen to be comparable to one another. Enhanced convective cooling of railway wheels and use of higher diffusivity brake blocks are seen to be effective ways of reducing the wheel temperatures and resulting thermal stresses.

1. Introduction

Indian Railways is one of the world's largest rail networks with over 100,000 track kilometres. It carries over 8 billion passengers and 1 billion tons of freight annually and forms the country's economic backbone. For smooth operation of increasing traffic, there is a need to reduce component failures and maintenance issues. Most of the trains operated by Indian Railways employ tread braking which results in hot running locomotive and wagon/coach wheels. Ekberg and Kabo [1] showed that tread braking induced thermal stresses affect fatigue crack growth, particularly in tread region. In addition, Teimourimanesh et al. [2] showed that braking induced thermal stresses in wheels can cause wheel warping, which can in turn affect wheel gauge and result in derailment. Thus, accurate prediction of running temperatures for tread braked railway wheels is crucial for taking corrective measures to improve wheel life and prevent derailments.

Accurate estimation of wheel temperatures for any given train running and braking conditions, using an approach of the kind presented in this work, is crucial for the following: (i) Understanding wear mechanism in railway wheels subjected to tread braking. More specifically, correct estimation of wheel temperatures and resulting thermal stresses in tread region is crucial for identification of conditions that can cause/accelerate rolling contact fatigue in wheels [1], (ii) Identifying running and braking conditions that accelerate wear and damage of composite brake blocks. Under certain braking conditions, e.g. application of brakes only on locomotive wheels in a passenger/goods train, brake block temperatures can get quite high (600 °C) because of which several constituents in composite brake blocks vapourize and result in severe cracking and wear of brake blocks [3], (iii) Choosing lubricants to be used on curved segments of rail for reducing wear from wheel–rail flange contact [4] and for avoiding occurrence of rail corrugations [5]. Since lubricant properties are temperature dependent, they are expected to perform differently for different rail–wheel interface temperatures which are in turn affected by prior running and braking history, (iv) Studying formation of wheel flats and microstructural changes in wheel and rail. For example, for partial/full wheel locking, wheel–rail interface temperatures can be quite high and result in formation of wheel flats. White-etching layers are normally found around the wheel flats which signify formation of martensite from heating to above austenizing temperature followed by rapid self-quenching [6], and

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Nomenclature

BCP	brake cylinder pressure (bar)
b_t	brake block thickness (mm)
c	specific heat (J/kg K)
h	convective heat transfer coefficient (W/m^2 K)
h_{wr}	thermal conductance at wheel–rail interface (W/m^2 K)
h_{ax}	effective convective heat transfer coefficient used to model rail-chill effect in axi-symmetric analysis (W/m^2 K)
k	thermal conductivity (W/m K)
$\dot{Q}_{wr}$	heat generation rate at wheel–rail interface (W)
$\dot{Q}_{wb}$	heat generation rate per brake block at wheel–brake block interface (kW)
$Q^r(\tau)$	time varying surface power input (W/m^2)
r	radial distance (m)
R	wheel radius (m)
t_{train}	train running time (h)
T_0	initial wheel temperature (°C)
T_w	current wheel temperature (°C)
ΔT_w	temperature rise of wheel (°C)
v	tread line speed (km/h)
v_t	train speed (km/h)
w	width of heat source on translating body (mm)
α	thermal diffusivity (m^2/s)
$\Delta\beta$	angular segment (rad)
ϵ	surface emissivity (–)
η_w	overall heat fraction entering into wheel (–)
ϕ	angular coordinate (rad)
ω	angular speed (rad/s)
ρ	density (kg/m^3)
θ	angular displacement (rad)

(v) Investigating wear rates for rail, wheel and brake blocks that are affected by thermal stresses and operating temperatures [7].

Prediction of wheel running temperatures for a given train running and braking history is challenging for several reasons. First, brake block friction characteristics – which depend on sliding speed, sliding distance, and brake load [8,9] – along with wheel–rail traction-slip and running resistance characteristics need to be modelled to estimate heat generation rate at wheel–brake block and wheel–rail interfaces for the given train running and braking history. Second, the fraction of heat generated entering the brake blocks, rail, and wheel needs to be estimated, by solving a moving heat source problem. Lastly, based on the knowledge of net heat entering the wheel, wheel temperatures need to be obtained by using field data calibrated emissivity and convective heat transfer coefficients for locomotive and wagon or coach wheels. Several earlier works use 1D heat transfer analyses to estimate heat partitioning at wheel–brake block and wheel–rail interfaces. However, the efficacy of using 1D analyses has not been shown. Further, attempt to estimate wheel running temperatures based on braking and speed history, and brake block type has not been undertaken. This work has two main novelties. First, 2D boundary element method is used to estimate heat partitioning at wheel–brake block and wheel–rail interfaces and the obtained results are compared with that from 1D analyses to evaluate the accuracy of the latter. Second, effectiveness of the three step approach adopted here is demonstrated by comparing predicted wheel temperatures for a diesel locomotive, fitted with cast-iron brake blocks, with the observed temperatures in field trials. This methodology has for the first time given a clear estimate of heat lost to rail, brake blocks, and ambient air.

Several earlier works have used 1D boundary element method to study temperature distributions in sliding and rolling contact problems. Chang et al. [10] showed that Green's function can be used efficiently for solving steady-state and transient heat conduction problems in isotropic and anisotropic media. Beck [11] derived Green's functions for transient heat conduction and applied it to solve a simple 2D problem on a rectangular domain. Jaeger [12] obtained temperature distributions for plane, sliding bodies. Block [13] obtained the flash temperatures by considering lubricant films between two sliding bodies. Block [14] derived the expressions for flash temperatures at contact region of rubbing surfaces with stationary and moving heat sources. Cameron [15] obtained surface temperatures of sliding bodies after accounting for convection. Tanvir [16] estimated the temperature rise at wheel–rail interface due to slip. Expressions for temperature rise were derived using Laplace transforms, by assuming an elliptical contact patch at wheel–rail interface. Kennedy [17] studied the effect of sliding velocity and thermal properties of sliding bodies on surface temperatures at contact. Significant changes in surface temperatures were observed with change in velocity and thermal properties. Yuen [18] investigated variation of position at which maximum temperature occurs in the contact zone of sliding bodies as a function of Peclet number. Maximum temperature was shown to occur at the end of the contact zone for higher values of Peclet number. Tian and Kennedy [19] calculated the temperatures of sliding bodies with different contact patches (square, circular and elliptic).

Knothe and Liebelt [20] studied the effect of fluctuations in contact pressure, surface roughness and surface damage on maximum contact temperature of sliding bodies. An increase in maximum temperatures was observed with above said conditions. Gupta et al. [21] obtained wheel–rail contact temperatures and heat loss to rail due to frictional heat at wheel–rail interface. Laraqi [22] investigated the effect of contact size and velocity on thermal contact resistance of sliding bodies. A decrease in resistance was observed with an increase in speed and contact size. Ahlstrom and Karlsson [23] derived a 1D analytical model for calculating wheel temperatures at different depths of contact region during wheel–rail skid. A maximum wheel temperature around 900 °C was found during skidding. Hou and Komanduri [24] studied the effect of shape and pressure distribution in contact zone of stationary and moving heat sources. Vick and Furey [25] studied the effect of multiple contacts between sliding bodies instead of a single contact. A decrease in temperature rise was observed with an increase in number of contacts and distance between contacts. Ahlstrom and Karlsson [26] obtained temperatures in wheel–rail skidding using finite element analysis. These temperatures were further used to study the phase transformations in the wheel. High cooling rates were observed in short sliding times which lead to martensite formation in wheels. Kennedy and Traiviratana [27] obtained temperature profiles and heat partition at wheel–rail interface for different sliding times. Spiryagin et al. [4] formulated a mathematical model to understand temperature rises at wheel flange–rail interface. Contact between flange and rail leads to higher temperatures than regular wheel tread–rail contact.

Newcomb [29] obtained an analytical solution for estimating heat partition between wheel and brake block for disc braking. Petereson [30] obtained heat partition between wheel and brake block using 2D finite element model with different types of wheel–brake block contact pressure distributions. Analysis was conducted for drag braking at 80 km/h with composite brake block for a single braking event. Vernersson [31–33] studied heat