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Short communication

Transient heat partition factor for a sliding railcar wheel

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Abstract

During a wheel slide the frictional heat generated at the contact interface causes intense heating of the adjacent wheel material. If this material exceeds the austenitising temperature and then cools quickly enough, it can transform into martensite, which may ultimately crack and cause wheel failure. A knowledge of the distribution of the heat partitioned into the wheel and the rail and the resulting temperature fields is critical to developing designs to minimize these deleterious effects. A number of theoretical solutions have appeared in the literature to model the thermal aspects of this phenomenon. The objective of this investigation was to examine the limitations of these solutions by comparing them to the results of a finite element analysis that does not incorporate many of the simplifying assumptions on which these solutions are based. It was found that these simplified solutions can produce unrealistic results under some circumstances.

Keywords: Wheel; Sliding; Heat; Temperature; Finite element

1. Introduction

The problem of freight car wheel spalling is governed by an intricate combination of physical and thermal phenomena. During sliding, a railcar wheel may develop a localized region of high temperature due to the generation of heat from friction between the wheel and the track. After the wheel starts rolling again, the rapid cooling by heat flow into adjacent wheel material may result in the formation of a brittle zone of martensite in this region. With subsequent rolling, this brittle material is broken free leaving a series of void spaces in the wheel tread known as spalls. Spall voids are deleterious to vehicle dynamic stability and safety, cargo ride quality, and track/train system component life.

A knowledge of the distribution of the heat partitioned into the wheel and the rail and the resulting temperature fields is critical to developing designs to minimize the deleterious effects due to spalling. Pioneering work on the basic problem of heat conduction in sliding bodies was carried out by Blok [1] and Jaeger [2]. On the contact interface between two sliding bodies, there should be continuity of temperature and conservation of heat fluxes. Instead of matching the surface temperature of the two bodies at all points along the contact interface, Blok determined the heat partitioned into each body by matching the maximum surface temperature. Jaeger [2] developed a steady-state solution by matching the average temperature of the two bodies at the contact interface. Ling [3] used a quasi-iterative method for solving integral equations matching temperature fields at all points of the contact interface. Barber [4] considered the case of multiple contact surfaces. Kennedy [5] developed a finite element analysis technique and applied it to a rotating shaft with a bearing and a labyrinth gas path seal. Yuen [6] used a Green's function formulation to develop asymptotic two-dimensional temperature fields for large Peclet numbers. He also examined the thermal penetration into the two bodies. Tian and Kennedy [7] developed analytical and approximate solutions for several sliding problems involving three-dimensional conduction including asperity contact. Bos and Moes [8] developed temperature distributions for uniform and semi-ellipsoidal heat sources acting over a square contact interface. They also investigated the case where the two bodies move in opposite directions. Komanduri and Hou [9] used a functional analysis approach to consider variable heat partitions along the interface between two bodies. They also present an extensive review of literature on the sliding problem. Komanduri and Hou [10] also used the functional analysis approach to determine the heat partition and

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temperature distribution at the tool–chip interface during metal cutting. Hou and Komanduri [11] also developed a general transient solution to the moving plane heat source problem in the form of multiple integrals that must be evaluated numerically. They present transient temperature distributions for the stationary source case.

The case of heat generation due to a railcar wheel sliding on a rail has been the subject of several investigations. Assuming that a fast moving heat source can be approximated as an instantaneous static source, Tanvir [12] determined the temperature rise due to slip between a wheel and the rail. Iwand et al. [13] developed a solution to this problem using the transient solution for the case of a suddenly applied heat source on a circular area. The partition of heat between the wheel and the rail was based on a steady-state formula. Sun et al. [14] developed a transient solution by assuming a one-dimensional heat flow in the non-sliding solid and equating average wheel and rail temperatures in the contact patch. Knothe et al. [15,16] used Laplace transforms to determine steady-state temperature fields for various types of pressure distributions resulting from wheel and rail contact. Gupta et al. [17] used finite element analysis to study heat generation through a combination of rolling and sliding. They assumed the heat to be equally partitioned between the wheel and the rail. Jergeus [18] also used finite element analysis to study the sliding wheel problem but assumed that the heat partition to be a function of temperature. He also considered phase transformations in the wheel. Kennedy et al. [19] performed a similar analysis, but considered a Hertzian pressure distribution over the contact area rather than a uniform one. This work was later extended by determining the heat partition factor based on matching the temperature between the wheel and the rail at all points on the contact patch [20]. Ahlstrom and Karlsson [21] developed a transient solution to the sliding problem assuming one-dimensional heat flow and a surface temperature with an exponential time dependence. Ahlstrom and Karlsson [22] also used the exponential temperature assumption in an axisymmetric finite element analysis that included a study of phase transformations in the wheel.

The transient solutions for the railcar wheel sliding problem described above are based on a number of simplifying assumptions that raise questions about their accuracy. The purpose of this investigation was to examine the limitations of these solutions by comparing them to the results of a finite element analysis that does not incorporate many of these assumptions.

2. Analytical solutions

As described above we will evaluate a number of transient solutions to the wheel sliding problem that are available in the literature. We begin with the work of Iwand et al. [13]. They assume that heat flows into the wheel as a constant flux over a circular area of radius a given by

$$q = (1 - \alpha) p \mu V \tag{1}$$

where q is the heat flux into the wheel, α the heat partition factor which is the fraction of the total friction heat generated that flows into the rail, p the surface pressure taken as uniform, μ the coefficient of friction, and V is the slide velocity. A steady-state value of α was used given by the formula

$$\alpha = \frac{1}{1 + 1.474\sqrt{\kappa/aV}} \tag{2}$$

where κ is the thermal diffusivity. The temperature field on the axis of symmetry is

$$T(y,t) = \frac{2q\sqrt{\kappa t}}{k}\left[i\,\mathrm{erf}\,c\left(\frac{y}{2\sqrt{\kappa t}}\right) - i\,\mathrm{erf}\,c\left(\frac{\sqrt{y^2+a^2}}{2\sqrt{\kappa t}}\right)\right] \tag{3}$$

where y is the depth into the wheel along the axis of symmetry, t the time, and k is the thermal conductivity. This solution was intended for low slide velocities and long slide times.

Next, we consider the solution of Sun et al. [14]. This solution assumes a uniform pressure and that heat flow in the non-sliding solid is normal to the surface (thermal impact). The wheel is treated as a semi-infinite body with a uniform heat flux acting on a rectangular area that represents the contact patch. By equating expressions for the average wheel and rail temperatures, they were able to develop a transient heat partition factor in the form

$$\alpha(t) = 1 - L^{-1}\left[\frac{J_1}{s^{1/2}+j_1} + \frac{J_2}{s^{1/2}+j_2} + \frac{J_3}{s^{1/2}+j_3}\right] \tag{4}$$

where L^{-1} is the inverse Laplace transform, s the Laplace transform variable, and J_i and j_i are the functions of the contact area dimensions, slide velocity, and thermal properties. They go on to present numerical results for a square contact patch.

Finally, we consider the solution presented by Ahlstrom and Karlsson [21]. They treat the wheel as a semi-infinite body with a surface temperature in the form of an exponential function of time

$$T = T_s(1 - e^{\lambda t}) \tag{5}$$

where T_s is the steady-state surface temperature and λ takes on a value between $-\infty$ and -2. From this the one-dimensional transient temperature field was determined from the solution in Carslaw and Jaeger [23].

3. Finite element formulation

We will employ several simplifying assumptions in developing a finite element analysis (FEA) of the wheel sliding problem. First, we will treat the problem as being two-dimensional. It has been shown [24] that the surface temperature distribution along the centerline of a square contact patch from a three-dimensional analysis is virtually identical to its two-dimensional counterpart for high Peclet numbers, where the Peclet number is defined as $Pe = V\ell/2\kappa$, and ℓ is the contact half-length. Also, the curvature of the wheel is ignored because the wheel radius is typically two orders of magnitude larger than the contact length. A uniform pressure distribution is assumed. The actual pressure distribution is expected to be somewhere between Hertzian and uniform depending on the amount of plastic flow, abrasion, etc. Our analysis is strictly thermal; i.e., no deformation effects are taken into account except to assume a size for the contact area.