2006

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.

In the analysis we will employ a two-dimensional, Cartesian coordinate system with its origin at the center of the contact patch and attached to the sliding wheel, so that the rail is moving with velocity V in the x-direction relative to this system. The thermal equilibrium for a continuum in which material is flowing with velocity vector $\mathbf{v}$ is described by

$$\rho c\left[\frac{\partial T}{\partial t}+\mathbf{v}\cdot\nabla T\right]=\nabla\cdot k\nabla T+Q \tag{6}$$

where T is the temperature which is a function of position vector $\mathbf{x}$ and time t, ρ the density, c the specific heat, k the conductivity, and Q is the heat generated per unit volume. To develop a finite element analysis, Eq. (6) is expressed in the weak form

$$\int_{\Delta t}\left\{\int_V\left[\rho c\left(\frac{\partial T}{\partial t}+\mathbf{v}\cdot\nabla T\right)-\nabla\cdot k\nabla T-Q\right]W\,\mathrm{d}V\right\}\mathrm{d}t=0 \tag{7}$$

where Δt is the time increment between steps and W is a weight function. Kennedy et al. [25] have observed that the finite element method is susceptible to difficulties involving numerical oscillations at high sliding velocities due to the dominance of convective diffusion terms. To overcome these difficulties we have adopted the approach introduced by Yu and Heinrich [26,27], which was developed for problems with significant convection. This involves using a special time-space quadrilateral element (DCC2D4) for convection/diffusion problems in the general purpose finite element program ABAQUS [28].

A finite element model of the wheel/rail system was constructed using 65,000 elements. A portion of the finite element mesh near the leading edge of the contact region is shown in Fig. 1. An artificial gap was created between the two bodies outside of the contact region. In the contact region the two bodies were joined by a thin layer of elements with artificially high conductivity in the y-direction and artificially low conductivity in the x-direction. This allows heat to flow freely in the vertical direction with negligible flux in the horizontal direction within the contact layer. The frictional heating was generated internally in the middle row of elements in the contact layer as a step function in time. The partitioning of the generated heat between the two bodies is governed by the physics of the problem rather than by an a priori assumption of its value. A graded mesh was used with high density in each body near the contact region where severe gradients in temperature are expected to occur. The outside boundaries were located a distance of 20ℓ from the contact region. This was found to effectively represent an infinite body. The validity of this approach for modeling heat conduction in sliding bodies was demonstrated in [24] where finite element results were found to be in good agreement with closed form solutions for special cases.

4. Results

To evaluate the analytical transient solutions, we have chosen the case presented by Sun et al. [14] which consists of a BR Mark II coach wheel with a wheel load of 42,000 N and a slide velocity of 40 m/s. The contact patch is a square area with sides of length 0.01 m. The coefficient of friction is 0.075. The thermal conductivity of the wheel and rail steel is 40 W/m°C, and the thermal diffusivity is 10×10^{-6} m^2/s. The heat generated is assumed to be uniform across the contact interface and given by $q=\mu pV$. A transient finite element analysis was performed for this case assuming an initial temperature of 0 °C. Fig. 2 shows the heat partition factor profiles at various values of time where $t=0$ corresponds to the start of the wheel slide (the contact patch

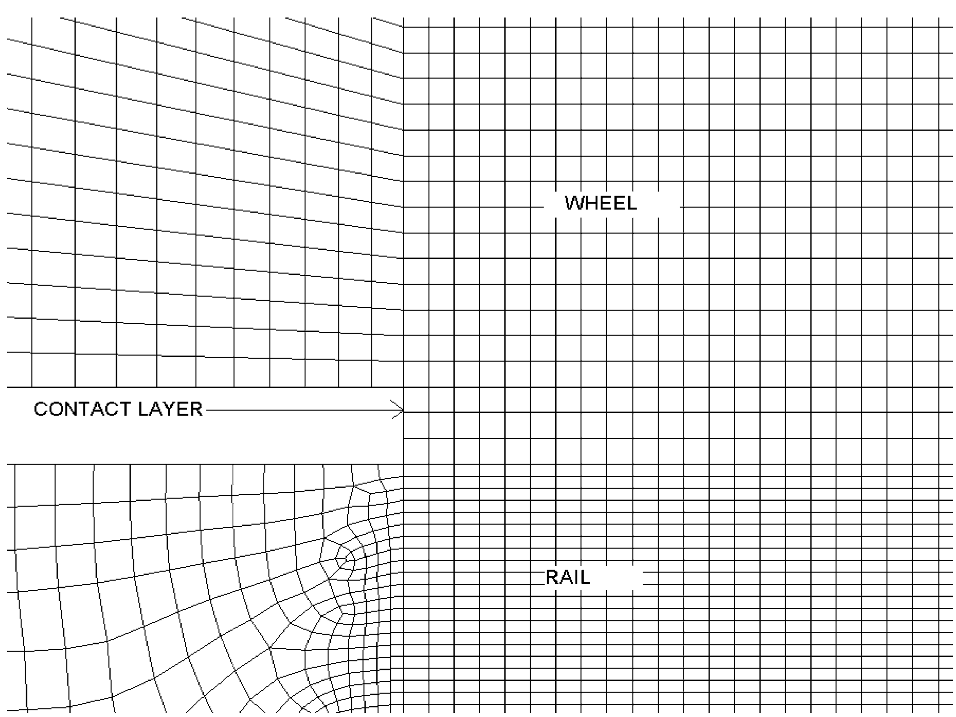

Fig. 1. Finite element mesh near the leading edge of the contact patch.

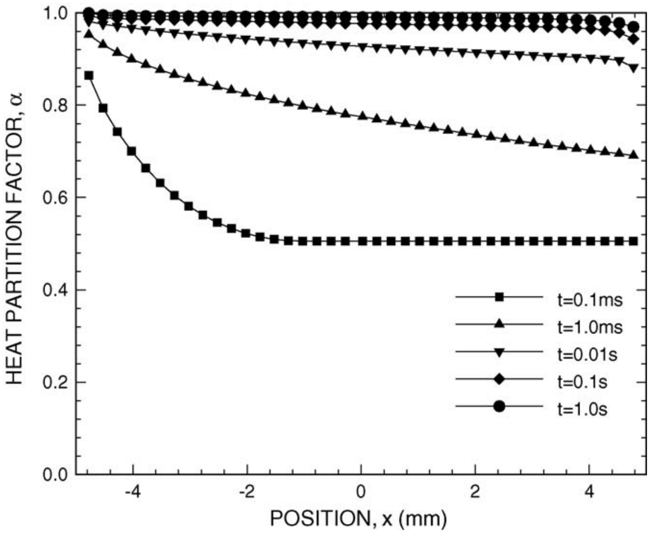

Fig. 2. Heat partition factor distribution along the contact patch at various times.

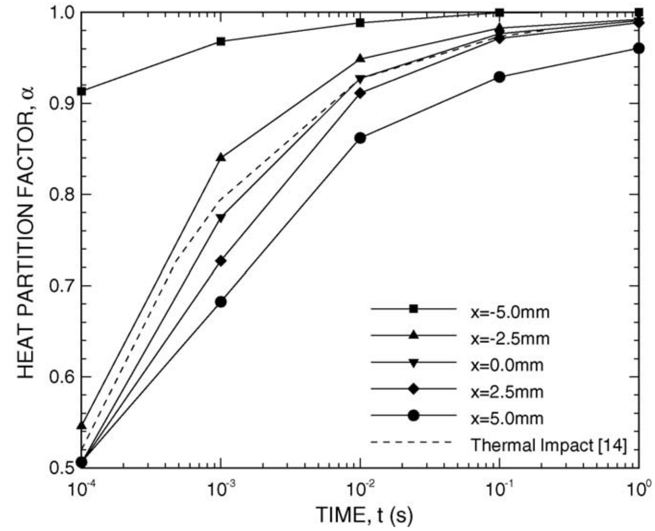

Fig. 4. Heat partition factor vs. time at various positions along the contact patch.

occupies the region (5 mm ≤ x ≤ 5 mm)). Initially, α is equal to one half so that the heat flux is evenly divided between the wheel and the rail. As time progresses, it increases, starting at the leading edge of the contact patch. After 0.1 s of sliding, almost all of the heat is entering the rail. This occurs because, relative to the contact patch, heat is being convected away by the moving rail.

Fig. 3 shows temperature profiles on the wheel surface near the contact patch at various values of time. At $t = 0.1$ ms, the temperature is approximately constant across the contact patch and is zero outside the contact patch. As time progresses the temperature becomes non-uniform with a peak value near the trailing edge. At $t = 0.1$ s the peak temperature is within a few percent of the steady-state peak value (∼1750 °C). A temperature in excess of 750 °C is required to cause the steel to transform from pearlite to austenite. It can be seen that such a temperature is reached very quickly, and a relatively brief slide could cause transformation to austenite, which upon cooling could be transformed into martensite and lead to subsequent spalling.

In order to compare these results to those from the simplified analytical solutions, the FEA results are recast as shown in Fig. 4 where the heat partition factor is plotted as a function of time at various positions on the contact interface. It is apparent that, for all points on the interface, α rises rapidly with time to a value close to one. The results of the thermal impact model by Sun et al. [14], in which α is treated as being uniform across the contact patch, are also shown in this figure. This simplified analytical solution is very close to the FEA results at the center ($x = 0$) of the contact patch. However, there is a fairly large difference between this solution and the FEA results at the leading and trailing edges of the contact patch.

Fig. 5 shows temperature as a function of time at various points on the contact patch (values at the leading edge, $x = -5$ mm, are omitted from the figure because they are essentially zero). Here, the temperature rises rapidly to very near the steady-state value within a period of 1 s of sliding. For this case the thermal impact solution of Sun et al. [14], who give the average temperature on the contact patch, is reasonably close

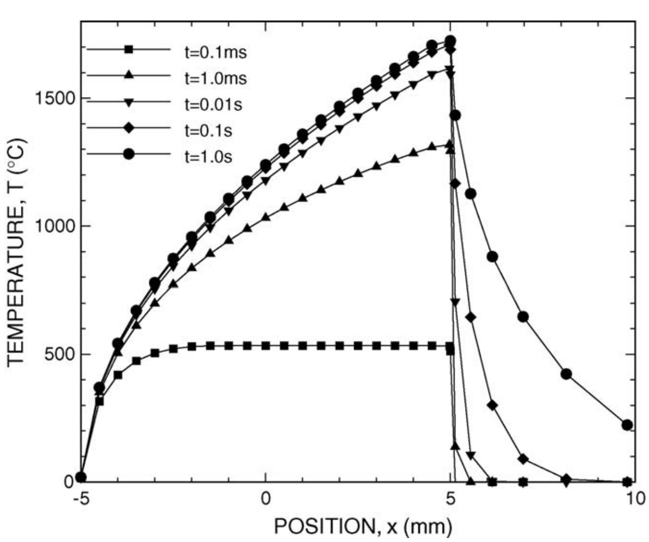

Fig. 3. Temperature distribution along the contact patch at various times.

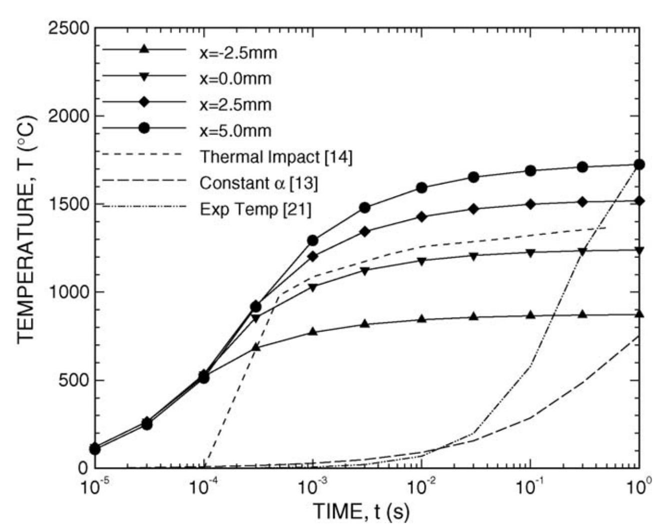

Fig. 5. Temperature vs. time at various positions along the contact patch.

to the FEA results at the center ($x = 0$) of the contact patch for time greater than 1 ms. Also shown is the constant-α solution of Iwand et al. [13] for this case. Here, the temperature rise is much slower than the FEA results. This occurs because of the use of the steady-state value for the heat partition factor. This large value (∼0.99) does not allow sufficient heat flux into the wheel at early times during the slide. In addition, the exponential-temperature solution of Ahlstrom and Karlsson [20], where the surface temperature is represented by an exponential function of time, is shown with λ set equal to -4 so that this solution would match the FEA results at $t = 1$ s. Here, the temperature rise is much slower than the FEA results. This indicates that the temperature history suggested by Eq. (5) may not always be a good choice for short sliding times.

5. Conclusions

Several simplified solutions for representing the temperature field in a sliding railcar wheel were evaluated by comparing them to the results of a finite element analysis that did not incorporate several of the simplifying assumptions. All were found to have deficiencies to some degree. The solution that relied on the use of a constant heat partition factor based on a steady-state solution did not allow the temperature rise quickly enough. This occurs because the heat partition factor starts out with a value of 0.5 so that heat is evenly divided between the wheel and the rail, but over time takes on a value close to one so that almost all of the heat enters the rail. It appears that this solution should not be used for high slide velocities and short slide times. Similarly, the solution that treats the surface temperature as an exponential function of time did not allow the temperature too rise quickly enough either. This solution also ignores the non-uniformity of the temperature distribution across the contact patch. The thermal impact solution, which is more mathematically complex than the others, provided a reasonable representation of the temperature history at the center of the contact patch. However, it did not account for the proper temperature distribution across the contact patch where the temperature reaches a maximum near the trailing edge. The finite element analysis approach presented here is reasonably straight forward to carry out and has the capability to account for temperature-dependent thermal properties, radiation, convection, and other complexities if needed.

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