Prediction of Contact Temperature on Interaction between Rail and Wheel Materials Using Pin-on-Disc Method

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Abstract: The interaction between rail and wheel materials produce wear damage on each surface and frictional heat. This study is aimed to predict contact temperature due to the interaction between rail and wheel for various applied load and velocity. The frictional heat was modeled as a solid contact pressure. The temperature rise was calculated by numerically integrating the equation using Adomian Decomposition Method. The paper also studies the influences of sliding speed on temperature rise. The pin-on-disc experiment was used to validate the results. The results show the peak pressure distributions increase linearly with the increasing applied load with an average slope of 0.053. The maximum temperature rise was 259.98 K at x = 1 mm for 5.3 MPa pressure and 3.14 m sec⁻¹ sliding speed. Prediction temperature rise results were successfully validated using pin-on-disc method. The average error between prediction and measurement results was 4.99 %.

Key words: Contact temperature, pin-on-disc, rail, wheel

INTRODUCTION

The contact temperature distribution at the interface of rail and wheel materials has been a topic for along time tribology research. Contact temperature distribution is critical for thermal stress analysis between two sliding bodies (Johnson and Greenwood, 1997; Chen and Wang, 2008) and thermal wear modeling (Bucher et al., 2006; Fischer et al., 1997). Contact temperature may influence material properties (Habeeb et al., 2008). Heat is generated due to frictional heating when a body slides on another body. The heat generated between two contact bodies is influenced of the thermal properties of the bodies, the contact geometry and the sliding speed.

One of the first models for estimating contact temperature was proposed by Blok (1963). The maximum temperature rises at the interface of two bodies are the same. The maximum temperature is independent of the speed at low Peclet numbers. The maximum temperature rise at the interface was calculated using a stationary heat source. Similar with Blok (1963) analysis, Carslaw and Jaeger (1938) and Gallardo-Hernandez et al. (2006) presented an analytical integral formula for the temperature rise on the half-space surface. However, their method was based on the average temperature rise instead of the maximum temperature rise.

The main objective of this study is to determine contact temperature on interaction between rail and wheel materials for various applied load and sliding velocity. The present model is based on pin-on-disc method. The model was applied to simulate the sliding contact of a half-space wheel material over a stationary sphere rail material. Results of pressure and contact temperature under different sliding speeds and applied load were discussed and compared.

MATERIALS AND METHODS

Contact problem: There are two kinds of mechanism contacts between rail and wheel (Fig. 1). First existing one is rolling-sliding contact which occurs on railhead and wheels tread contact. The second is pure sliding contact which occurs on rail edge and wheel flange. In this study, the contact between rail and wheel can be assumed as non conformal contact.

Fig. 1: Overview of wheel-rail contacts
A non-conformal contact of two bodies in revolution is classified in the general elastic contact. Depending on their modulus of elasticity, the two bodies will deform to certain degrees. The problem may be reduced to that of a single equivalent elastic body revolution on a plain infinitely rigid body. This assumption is widely adopted in the theory and application of Hertzian contacts so that the conversion formulas of geometry and elasticity are well established (Al-Bender and De Moerlooze, 2008; Enblom and Berg, 2008).

When at the top of an elastic pin of radius $r$ and equivalent modulus of elasticity $E_e$ (Fig. 2) is pressed with a load $N$ against a rigid plane disc surface, there is a mark on the contact area (Knothe and Liebelt, 1995; Olofsson and Telliskivi, 2003; Sundh and Olofsson, 2011). The equivalent modulus of elasticity is given by:

$$E_e = \frac{E}{1 - \nu^2}$$  \hspace{1cm} (1)

The pin is rotating with a fixed angular speed $\omega$. Due to the normal force, angular velocity and coefficient of friction $\mu$, a reaction force will occur. This reaction force is frictional force $F_r$:

$$F_r = N \mu$$  \hspace{1cm} (2)

\begin{table}[h]
\centering
\caption{Nominal chemical composition of the studied steel}
\begin{tabular}{|c|c|c|c|c|c|}
\hline
 & C & Mn & P & S & Si & Fe \\
\hline
Rail & 0.71 & 0.87 & 0.02 & 0.01 & 0.29 & Balance \\
Wheel & 0.53 & 0.69 & 0.02 & 0.01 & 0.28 & Balance \\
\hline
\end{tabular}
\end{table}

\begin{equation}
F_{net} = F_{app} - F_r
\end{equation}

The net force $F_{net}$ is equal applied force $F_{app}$ minus frictional force $F_r$ Eq. 3. The contact patch is defined as the region $A$ in the xy-plane: $A = \{(x, y): x^2 + y^2 = r^2\}$. The normal stress $F_r$ is given by (Al-Bender and De Moerlooze, 2008):

$$F_r = N \sqrt{\frac{E}{1 - \nu^2}}$$  \hspace{1cm} (4)

The magnitude of these axes depends on the radii and the radii of curvature of the contacting systems and on the normal load.

Since, the maximum heat flux occurs in the z direction, the problem can be simplified to two-dimensional contacts. The maximum pressure for this case is (Jendel, 2002):

$$\sigma = \frac{3W}{2\pi r^2}$$  \hspace{1cm} (5)

The materials used in this research are rail steels. The chemical composition of material is shown in Table 1.

Pin-on-disc is commonly used in wear test. The tests use Ducom multi specimen testing machine designed according to ASTM G99 standards. Rail steel was cut to form disc specimen which is 42 mm in diameter and 5 mm in width. Pin samples were prepared as 6 mm in diameter and 12 mm in length. During wear tests, the normal forces were applied. The normal force of 40 N, 60 N, 80 N and 100 N were selected. Both pin and disc sample were polished using 120, 220 and 500 grit abrasive papers and cleaned with alcohol and dried.

**Heat transfer equations:** The assumption is introduced an energy equation problem to determine the temperature field in the wheel-rail contact. The energy equation heat transfer equation will be governed based on Fig. 2:

$$\rho \epsilon_c \frac{\partial T}{\partial \tau} + \phi \left( \frac{k \partial T}{\delta x} \right) = \frac{\partial}{\partial y} \left( \frac{\partial \phi \left( \frac{k \partial T}{\delta y} \right)}{\delta y} \right) + \frac{\partial}{\partial y} \left( \frac{\partial \phi \left( \frac{k \partial T}{\delta y} \right)}{\delta y} \right) + q^*$$  \hspace{1cm} (6)

If the density of a solid is constant, $\rho = \epsilon v$. Dividing Eq. 6 by $k$ and replacing $k/\rho c_p$ by $\alpha$, the thermal diffusivity of the material:

$$\frac{1}{\alpha} \frac{\partial T}{\partial \tau} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + q^*$$  \hspace{1cm} (7)
The differential equation that governs the steady state temperature field is:

\[ \alpha \left( \frac{\partial T}{\partial x} + \gamma \frac{\partial^2 T}{\partial y^2} + \beta \frac{\partial^2 T}{\partial z^2} + q^0 \right) = 0 \]  

(8)

Mazidi et al. (2011) proposed that there is a heat partition at the contact surface of two sliding components, because of thermal resistance due to accumulation wear particles. If the frictional heat is the only internal heat generation source (Ertz and Knothe, 2002), then:

\[ q^0 = \mu P_v \]  

(9)

Where

\( v = \) Sliding velocity and within the contact area heat is generated owing to friction. The following assumptions were made:

- There is pure sliding within the whole contact area.
- The effects of elastic deformation are negligible. Consequently, the local relative velocity equals the global sliding velocity \( v \).
- The heat flow rate \( q \) (i.e., the product of the friction coefficient \( \mu \), the normal pressure \( P(x) \) and the sliding velocity) is transformed completely into heat.

**Temperature prediction of pin and disc:** The schematic diagram of the temperature at some point \((r, \theta, z, t)\) within a pin cylinder of length \(2L\) and radius \(R\), is shown in Fig. 2. The cylinder initially at temperature \(T_a\), at time zero, it is exposed to a fluid at temperature \(T_0\), with convection coefficient of \(h\). Temperature within the cylinder is a function of \((r, \theta, z, t)\), for a system with internal heat generation:

\[ \frac{1}{\alpha} \frac{\partial T_\infty}{\partial \tau} = \frac{1}{\alpha} \frac{\partial^2 T}{\partial r^2} + \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} + \frac{q^0}{k} \]  

(10)

The differential equation can be solved by the Adomian Decomposition method, with the assumption of coefficient of friction is constant. Solution for the temperature of any point located at the position \((r, \theta, z, t)\) of a cylinder at any time \(t\) is \(T = T_r (r, \theta, z, t)\) and \(k\) is thermal diffusivity. Equation 10 can be re-write in an operator form by:

\[ L_T T_r = (1 + \frac{1}{k} \mu \nu P_v \sqrt{a^2 - (r \cos \theta)^2 - (r \sin \theta)^2} + \frac{1}{k} \frac{L_T T_r}{L_T T_a} + \frac{1}{k} \frac{L_T T_r}{L_T T_a} ) \]  

(11)

where, the differential operators \(L_r, L_\theta\) and \(L_z\) are defined by (Wazwaz, 2002):

So that the integral operator \(L^{-1}\) exist and given by:

\[ L^{-1}(\alpha) = \frac{1}{\alpha} \int \alpha \]  

(13)

Applying \(L^{-1}\) to both sides of Eq. 13 and using the initial condition lead to:

\[ T(r, \theta, z, t) = \sum_{n=0}^{\infty} T_n (r, \theta, z, t) \]  

(15)

Substituting Eq. 14 into both sides of Eq. 15 yields:

\[ \sum_{n=0}^{\infty} T_n = f(r, \theta, z) + \alpha \]  

(16)

The components \(T_n (r, \theta, z, t)\), \(n \geq 0\) can be completely determined by using the recursive relationship:

\[ T_n (r, \theta, z, t) = \alpha L^{-1} \left( \frac{1}{k} \frac{L_T T_r}{L_T T_a} + \frac{1}{k} \frac{L_T T_r}{L_T T_a} \right) \]  

(18)

Where:

\[ T(r, \theta) = \frac{2 \alpha \mu \nu P_v \tilde{P}_0}{\pi k} \sqrt{a^2 - (r \cos \theta)^2 - (r \sin \theta)^2} \]  

(19)

**RESULTS AND DISCUSSION**

Figure 3 shows comparisons of pressure distribution for various applied load. The pressure distribution is calculated based on Hertzian contact solutions. Pressure distribution obtained from the purely elastic analysis well matches that of the analytical Hertz solution. The peak
The pressure distribution occurs at the centre of contact, \( x = 0 \). The peak pressure distributions increase linearly with the increasing applied load with an average slope of 0.053. These results are same trend with Windarta et al. (2011) conclusions. The pressure and wear rate increase linearly with increasing applied load.

However, the temperature rise due to frictional heating softens the material (degrades the yield strength) when the thermal softening effect is included which further decreases the von Mises stress intensity. Table 2 contains the circular contact parameters and material properties for temperature calculation throughout this study.

Figure 4 presents the comparison of contact temperature profiles along the \( x \) axis for various models. Chen and Wang (2008) Model assumes purely elastic contact based on Hertzian contact. There is not any temperature rise outside contact area. The contact pressure is approximately symmetric with respect to \( x = 0 \). Erzt and Knothe (2002) proposes a model using four degree polynomial. However, frictional heating distorts the contact pressure distribution under higher sliding speeds and shifts the pressure profile along the mating surface sliding direction. Proposed model modified (Erzt and Knothe, 2002) model, using assumption that frictional heating outside contact produce by conduction and fixed source from contact area. The maximum temperature rise is 259.98 K at \( x = 1 \) mm for 5.3 MPa pressure and 3.14 m sec\(^{-1}\) sliding speed. The peak temperatures rise with respect to sliding velocity present in Fig. 5.

The surface temperature rise model developed is evaluated based on a pure sliding circular Hertzian contact between an adiabatic moving surface and a stationary surface is considered. In this case, all of the frictional heat is transferred into the stationary surface.

Comparisons between prediction temperature rise and pin-on-disc measurement results with respect to sliding velocity are presented in Table 3. The measurement were taken 6 mm from the centre of pin and disc contact. Average error between prediction and measurement results is 4.95%. The highest error occurs at low sliding velocity (0.16 m sec\(^{-1}\)).

Figure 6 gives the variation of the surface temperature rise distribution along the \( x \)-axis with respect...
Fig. 6: Distributions of surface temperature rise along the x axis corresponding to various sliding velocities

<table>
<thead>
<tr>
<th>Velocity (m sec(^{-1}))</th>
<th>Prediction</th>
<th>Measured</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.16</td>
<td>3.52</td>
<td>3.6</td>
<td>7.36</td>
</tr>
<tr>
<td>0.31</td>
<td>4.90</td>
<td>4.7</td>
<td>4.26</td>
</tr>
<tr>
<td>0.47</td>
<td>6.03</td>
<td>5.8</td>
<td>3.96</td>
</tr>
<tr>
<td>0.63</td>
<td>6.98</td>
<td>7.3</td>
<td>4.38</td>
</tr>
<tr>
<td>1.57</td>
<td>11.02</td>
<td>10.6</td>
<td>3.96</td>
</tr>
<tr>
<td>3.14</td>
<td>15.59</td>
<td>14.7</td>
<td>6.05</td>
</tr>
</tbody>
</table>

Table 3: Temperature rise comparison for various sliding speed

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NOMENCLATURE

\(a_c, p_s\) = Hertz contact radius (mm) and peak pressure (MPa)
\(E_i\) = Young’s modulus two contact bodies (i = 1 and 2)
\(E^*\) = Equivalent Young’s modulus (GPa)
\(H\) = Hardness (GPa)
\(k\) = Thermal conductivity (W m\(^{-1}\) K\(^{-1}\))
\(l\) = Characteristic length (mm)
\(p_s\) = Pressure and shear traction (MPa)
\(q, q_1, q_2\) = Total heat flux and heat flux flowing to two bodies (W m\(^{-1}\))
\(\Delta T\) = Temperature rise (K)
\(T_m, T_a\) = Material melting point and reference room temperature (K)
\(V_s\) = Sliding velocity (m sec\(^{-1}\))
\(W\) = Normal load (N)
\(x, y, z\) = Space coordinates (mm)
\(\alpha\) = Linier thermal expansion coefficient (mm K\(^{-1}\) m\(^{-1}\))
\(\delta\) = Rigid body approach (mm)
\(\Delta\) = Mesh size (mm)
\(\mu_r\) = Coefficient of friction
\(\nu_{ij}\) = Poisson ratio of two contact bodies
\(\sigma_{ij}\) = Total stress component (MPa)
\(\sigma_{VM}\) = Von Mises equivalent stress (MPa)
\(\sigma_y\) = Initial yield strength (MPa)

CONCLUSION

The pressure distribution is calculated based on Hertzian contact solutions. Pressure distribution obtained from the purely elastic analysis well matches that of the analytical Hertz solution. The peak pressure distributions increase linearly with the increasing applied load with an average slope of 0.053.

The maximum temperature rise is 259.98 K at \(x = 1\) mm for 5.3 MPa pressure and 3.14 m sec\(^{-1}\) sliding speed. Prediction temperature rise results were successfully validated using pin-on-disc method. Average error between prediction and measurement results is 4.99%. The highest error occurs at low sliding velocity (0.16 m sec\(^{-1}\)).

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