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Application pin-on-disc method for wear rate prediction on interaction between rail and wheel

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Abstract. This paper aim to predict wear rate on interaction between rail and wheel using pin-on-disc method. The wear tests were performed on Ducom multispecimen test machine designed according to ASTM G99 standards. The applied loads selected were 40 N, 60 N, 80 N, and 100 N with a common rotating speed of 100 rpm. Pin-on-disc result was used to predict wear rate at real condition. The results show that stress distribution can be used to predict rail and wheel surfaces evolution. It also can be used in a specific condition for prediction at a real applied load. Depth of wear curvature indicate parabolic curve. Depth of wear are 89.33 µm, 127.25 µm, 263.01 µm, and 395.67 µm for 40 N, 60 N, 80 N, and 100 N applied loads respectively in steady state condition. Based on tangential contact stress, the surface shear stress increase in magnitude from leading edge to reach maximum value 668 MPa at longitudinal axis $x = -0.73$ mm. The adhesion part dominated the contact patch area. The maximum temperature $828.58^\circ C$ occurs at the contact surface.

1. Introduction
Interaction between rail and wheel can be divided into rolling-sliding contact and purely sliding contact [1]. All of contact produced damage, such as wear and fatigue. The maintenance cost of wear and fatigue of wheel and rail contacts is about 1.2 billion USD in China annually [2]. Wear prediction is important to be determined. It is well-known fact that sliding contact contributes more to wear than rolling-sliding contact. Sliding contact can be simulated using pin-on-disc.

Actual tests need huge resources. Since actual testing for rail and wheel contacts are expensive and time consuming, therefore simulation studies are selected well than actual experiment. Wear modelling in metals has been conducted by many researchers [3, 4]. Many models have been proposed for different materials and applications. Several models are co relational in nature and specific system. The model is based on the wear data of that system and only applied to that data set [5-8]. The present paper aimed to determine wear rate using pin-on-disc method and wheel and rail contact temperature.

2. Methodology

2.1. Interaction between rail and wheel
Interaction between rail and wheel can be divided onto two kinds of mechanism (figure 1(a)). The first one occurs on railhead and wheels tread contact, it sometimes called rolling-sliding contact. The second occurs on rail edge and wheel flange contact, it is pure sliding contact. In this research, contact between rail and wheel can be assumed non conformal contact.
A non-conformal contact of two bodies of revolution is considered in the general elastic contact. Depending on their modulus elasticity, the two bodies will deform to certain degrees. The problem may be reduced to that of a single equivalent elastic body of revolution with 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 [9, 10].

When at the top of an elastic pin of radius \( r \) and equivalent modulus of elasticity \( E_e \) (figure 1(b)) is pressed with a load \( N \) against a rigid plane disc surface, there is a contact area mark [11]. The equivalent modulus elasticity is given by equation (1).

\[
E_e = \frac{E}{1 - \nu^2}
\] (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 \( F_f \).

\[
F_f = N \mu \quad \quad \quad F_{Net} = F_{app} - F_f
\] (2)

The net force \( F_{Net} \) is equal applied force \( F_{app} \) minus frictional force \( F_f \) (equation (2)). The contact patch is defined as the region \( A \) in the \( xy \)-plane: \( A = \{(x, y): x^2 + y^2 \leq r^2\} \). The normal stress \( P_z \) is given by equation (3).

\[
P_z(x, y) = P_0 \sqrt{1 - \left(\frac{x}{r}\right)^2 - \left(\frac{y}{r}\right)^2} \quad \text{with} \quad P_0 = \frac{3N}{2\pi r^2}
\] (3)

Since pin-on-disc contact can be categorized as sliding contact, the Archard’s wear equation as shown in equation (4) can be applied.

\[
V_w = k_{(v_s, P)} \frac{N.s}{H}
\] (4)

where \( V_w \) is the wear volume (m\(^3\)); \( s \) is sliding distance (m); \( N \) is normal force (N); \( H \) is hardness of materials (Pa); \( k \) is wear coefficient (\( \sim \)); \( v_s \) (x, y) is relative sliding velocity (m/s); \( P_z \) (x, y) is contact pressure (Pa).

In the numerical analysis, Archard’s wear model is applied locally by partitioning the contact area into a calculation mesh and dividing equation (5) by the area increment, \( dx \ dy \). The wear depth, \( \Delta z \), at the center of the mesh element with co-ordinate \((x, y)\) thus becomes:

\[
\Delta z(x, y) = k_{(v_s, P)} \frac{P_z(x, y)s(x, y)}{H}
\] (5)
The wear depth is preferred to determine rail-wheel profile evolution [10]. Wear coefficients (k) are determined by experiment. Wear coefficient values are presented in maps as a function of contact pressure and sliding velocity. Based on available experimental results for relevant material combinations, a wear map for dry conditions has been defined. Jendel [11] presented the most important regimes for railway operation are the k2 and k3 areas although both higher contact pressures and larger slip may occur in narrow curves or at poor adhesion conditions, respectively.

A non-linear creep force theory has been proposed and developed [10, 12, 13]. The spin contribution in Kalker’s linear theory is neglected. The resultant creep force \( F \) was assumed to approach the traction bound \( \mu W x \) following a polynomial approximation in equation (6).

\[
F = \mu W x \left( k - \frac{1}{3} k^2 + \frac{1}{27} k^3 \right) \quad \text{for } k \leq 3
\]

and

\[
k = \frac{F}{\mu W}
\]

Many researchers investigated contact temperatures between rail and wheel using analytical method in numerous papers [14, 15]. Since pin diameter less than its length, one-dimensional approach can be used. In one-dimensional approach, the heat conduction with included heat transfer into the surrounding can be formed as in equation (7) [15].

\[
\frac{\partial T}{\partial \tau} = \kappa \frac{\partial^2 T}{\partial z^2} - b(T - T_a) \quad T = T(z, \tau), \quad z \in (0, \infty), \quad \tau \in (0, t)
\]

where, \( T \) is the pin temperature, \( z \) is the pin axis coordinate, \( T_a \) is the ambient temperature (20°C). \( \kappa \) is the thermal diffusivity coefficient.

Based on the one-dimensional heat conduction problem, the initial and boundary conditions are determined by equation (8) and equation (9).

\[
T(\tau = 0) = T_a \quad k \frac{\partial T}{\partial z}(z = 0) = q^-
\]

\[
\lim_{z \to \infty} T(z, \tau) = T_a \quad q^- = \mu P_v
\]

where, \( \mu \) is the coefficient of frictional. \( v \) is traveling velocity. The temperature distribution along the pin axis and the contact temperature are given by equation (10).

\[
T_{pin} = T(z, t) = T_a + \frac{\mu P_v}{k} \left( \frac{1}{\pi^2} e^{-\frac{b r^2}{4 k \tau}} \right) \quad T_{contact} = T(z = 0, t)
\]

where \( k \) is thermal conductivity (ksteel = 41 J/msK) [14].

3. Material and experimental
The rail steels were used to represented real materials in this research. The chemical composition of material is shown in table 1.

| Table 1. Nominal chemical composition of the studied steel. |
|-----------------|-------|-----|-----|-----|-----|--------|
| C        | Mn    | P    | S    | Si   | Fe    |
| Rail    | 0.71  | 0.87 | 0.02 | 0.01 | 0.29  | Balance |
| Wheel   | 0.53  | 0.69 | 0.02 | 0.01 | 0.28  | Balance |
Pin-on-disc is commonly used in wear test. The tests use Ducom multi specimen testing machine designed according to ASTM G99 standards. The wheel material was cut to form disc specimen, which was 42 mm in diameter and 5 mm in width. The rail material has been cut to form pin specimen. The 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.

4. Result and discussion

4.1. Normal load analysis

The samples used in this research to form pin materials and disc materials are rail steels UIC60 and wheel. It is assumed that normal load influence along contact area only. There is not normal load influenced outside contact area. It proved that wheel-rail contact is line contact. The dry sliding tests reported in Figure 2(a) are conducted using Ducom multi specimen tribometer. Rail steel has been cut to form disc specimen, which is 42 mm in diameter and 6 mm in thickness. Pin samples are prepared as 6 mm in diameter and 12 mm in length. The normal force of 40 N, 60 N, 80 N, and 100 N are selected during wear tests. Both pin and disc sample are polished using 120, 220 and 500 grit abrasive papers, and cleaned with alcohol and dried.

Figure 2(a) shows depth of wear are 89.33 µm, 127.25 µm, 263.01 µm, and 395.67 µm for 40 N, 60 N, 80 N, and 100 N applied loads respectively in steady state condition. Depth of wear curvature is exponential curve for transient condition.

![Figure 2](image)

**Figure 2.** Depth of wear, (a) Wear depth of rail and wheel material for various load after pin-on-disc results; (b) depth of wear prediction for 100 kN applied load.

The total weight of a boogie and passengers in a real operating railway-wheel system conditions are usually used 80 – 120 kN applied load [10]. In this simulation, normal load used 100 kN applied load to predict real condition. The prediction was used to fit the results of experiment, where k was identified to be 5.10-4 mm3/N.mm. Prediction depth of wear for 100 kN applied load is presented in figure 2(b). It is shown that depth of wear has parabolic curvature with maximum value at the center of applied load. This results can be used to predict rail and wheel profile evolution.

4.2. Tangential load analysis

The tangential contact problem is usually connected with friction, and a solution is needed for surface shear stress distributions and related area of adhesion and slip in the contact patch [16]. The slip-stick area distributions in contact patch when the traction coefficient 0.3 is shown in figure 3(a). The stick
area is located at the leading part of contact area and decreases in size with the increase of tangential load. The surface shear stress (figure 3(b)) increases in magnitude from the leading edge to reach maximum value 668 MPa at longitudinal axis \( x = -0.73 \) mm.

![Diagram of Slip-Stick area distribution](image)

**Figure 3.** (a) Slip-Stick area distribution for \( \mu = 0.3 \); (b) Surface shear stress.

4.3. Contact temperature

The distribution of contact temperature was calculated using equation (10). The results are shown in figure 4. The results show that the temperatures strongly decrease along the pin axis since the heat is conducted into the surrounding. The highest temperature 828.58°C occurs at the contact surface. This results do not consider cooling effects which caused by environmental contact and heat transfer between pin and disc materials.

![Diagram of Contact Temperature](image)

**Figure 4.** Comparison between experimental results and predicted results.

5. Conclusion

The sliding contact between rail gauge and wheel flange interaction have been investigated using pressure distribution study. The results showed that stress distribution can be used to predict rail and wheel surfaces evolution. It also can be used in a specific condition for prediction at a real applied load. The pin-on-disc method can be used to predict the depth of wear experimentally. The maximum pressures are 2.12 MPa, 3.18 MPa, 4.25 MPa, and 5.31 MPa for 40 N, 60 N, 80 N, and 100 N applied
loads respectively. Depth of wear curvature indicate parabolic curve. Depth of wear are 89.33 µm, 127.25 µm, 263.01 µm, and 395.67 µm for 40 N, 60 N, 80 N, and 100 N applied loads respectively in steady state condition. The surface shear stress reach maximum value 668 MPa at longitudinal axis $x = -0.73$ mm. The adhesion part dominated the contact patch area. The maximum temperature 828.58 ºC occurs at the contact surface.

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