Influential Factors on Adhesion between Wheel and Rail under Wet Conditions

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Study on the adhesion of wheel/rail system includes many research fields such as tribology, rolling contact mechanics, material science, structural dynamics, heat transfer and others. The authors focused on several parameters, which play very important roles in the adhesion coefficient of wheel/rail interface. Those parameters are running speed, water temperature, wheel load and surface roughness of the wheel and rail, which have a great influence on hydro-lubrication behavior of water film formed at the wheel/rail interface from the tribological point of view. This paper describes the relation between those parameters and their influence on the adhesion coefficient by means of both theoretical and experimental approaches. Numerical analysis was based on mix-lubrication theory and laboratory experiments were conducted with a twin-disc rolling contact machine. The numerical solutions and the experimental results indicated that the effects of running speed, water temperature and surface roughness of wheel/rail interface on the adhesion coefficient were significant.

Keywords: wheel, rail, rolling contact, adhesion, wet, contact mechanics

1. Introduction

Under dry conditions, the influence of running speed on the adhesion coefficient between wheel and rail is insignificant up to 300km/h for Japanese Shinkansen vehicles. Under wet conditions due to rain or snow, however, the adhesion coefficient drastically decreases with the increase of running speed in field tests and laboratory experiments as reported by Ohyama [1]. Low adhesion coefficient gives rise to some problems with driving force to achieve high speed, and braking force to stop safely within the distance regulated by railway standards. Furthermore, common wheel-tread surface damage, such as flats, skid marks and shelling due to low adhesion coefficient will occur, causing noise and vibration of the vehicle and deteriorating the riding quality. According to some questionnaires, wheel-tread machining to remove flats accounted for 75% of all urgent wheels machining work in Japan [2]. This may therefore be an important factor in increasing railway maintenance costs. In order to solve the low adhesion problem, many interesting and important findings about the mechanism of adhesion and methods of improving wheel/rail adhesion have so far been obtained. However, considering that common running high speeds today stand at over 300km/h in many countries and that the trend is for this speed to continue to rise in forthcoming years, clarifying the behavior of adhesion coefficients and its control, particularly in wet conditions are still very important issues. From a theoretical point of view, Ohyama et al. [3] have so far tried to shed greater light on the adhesion mechanism between the wheel and the rail under water lubrication by means of numerical analysis, applying the Elasto-Hydrodynamic Lubrication (EHL) theory. They adopted Herrebrugh’s integral equation to calculate the water film thickness between two smooth contact surfaces with various running speeds, and then discussed the adhesion coefficient. After that, Chen et al. [4] advanced the analysis of adhesion coefficients introducing a simplified numerical method directly in consideration of rough contact surfaces with the mixed lubrication analysis based on EHL theory and rough surface contact model which is Greenwood-Williamson’s (G-W) stochastic model for Gaussian distribution of roughness heights. Jin et al. [5] carried out numerical analyses to investigate the effect of wheel set rolling speed, axle load and creepage on the adhesion coefficient between wheel and rail with water contamination, adopting an improved Kalker’ s rolling contact theory. In their calculation, variable friction coefficients depending on rolling speeds, axle loads and creepage were taken into consideration, and the friction coefficients were given by functional expressed forms obtained statistically and regression of the experimental results. Using an experimental approach, Ohyama [1] carried out experiments with a high-speed rolling contact machine. He identified that the adhesion coefficient decreases with increasing rolling speed and wheel load, and that surface roughness had a great influence on the adhesion coefficient. Furthermore, Jin et al. [5] conducted experiments with a full-scale test facility to investigate the effect of running speed (80-240km/h) and axle load (44kN, 67kN) on the adhesion coefficient; they identified that the rolling speed had a great influence on the adhesion coefficient in the case of water contamination; and, when the rolling speed increases from 80 to 240 km/h, the adhesion coefficient decreases by a factor of about 2/5. Regarding the axle load, its increase also leads to a lower adhesion coefficient. Gallardo-Hernandez et al. [6] performed experiments with a twin disc simulation to study the adhesion coefficient. They indicated that water reduces adhesion compared to dry conditions.
In this paper, the authors firstly introduce a numerical model established to simulate the rolling contact between wheel and rail under wet conditions, and then show the calculated results of the adhesion coefficient under various conditions of running speed, water temperature, wheel load and surface roughness. In addition, the experimental results with a twin-disc rolling contact machine to investigate the effect of above-mentioned parameters on the adhesion coefficient as well as the comparison with numerically calculated results are described.

2. Theoretical analysis

Figure 1 shows a schematic model of numerical analysis established by the authors. (a) shows wheel/rail contact interface including water film; (b) shows water film geometry in a longitudinal cross section; (c) shows the typical contact area configurations for three different ranges of γ, which is a factor to characterize surfaces with directional patterns and defined as the ratio of the x and y correlation lengths of surface roughness asperity [7]. Water flow depends strongly on this parameter.

![Figure 1](image)

(a) Contact situation of wheel and rail (b) Water film geometry (c) Contact area configurations

Fig. 1 The numerical analysis model of wheel/rail contact with water film

In the above numerical analysis model, it is assumable that two contact bodies (i.e., wheel and rail) have elastic deformations when load is applied onto them and each δ₁ and δ₂ have a Gaussian distribution of heights with the mean of zero and the standard deviations of σ₁ and σ₂, respectively. The equations for numerical analysis are as follows:

(1) An average Reynolds equation in the case of rough surface contacts under the steady-state condition given by Patir and Cheng [8];

\[
\frac{\partial}{\partial x} \left( \phi \frac{\rho h^3}{12\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi \frac{\rho h^3}{12\eta} \frac{\partial p}{\partial y} \right) = \frac{u_1 + u_2}{2} \frac{\partial p_c}{\partial x} - \frac{\partial h}{\partial x}
\]

where \( \rho \) is the mean hydrodynamic pressure that occurs on the water film; \( h \) is the nominal lubricant film thickness (Fig.1 (b)); \( h_i \) is the average gap (Fig.1 (b)); \( u_1 \) and \( u_2 \) are the rolling speeds of surface 1 and 2, respectively, along the x direction; \( p \) and \( \eta \) are the density and the viscosity of lubricant, respectively as shown in bellow.

\[
\rho = \rho_0 + \frac{5.9 \times 10^8 + 1.34p}{5.9 \times 10^8 + p}
\]

(2) Average gap between two contact surfaces (i.e.; the nominal film thickness of water flow);

\[
h(x,y) = h_0 + \frac{x^2}{2R_x} + \frac{y^2}{2R_y} + d(x,y) - d(0,0)
\]

(3) Force compliance relation of the mean asperity contact with water film

\[
\frac{h}{\sigma} = \exp \left[ \sum_{i=1}^{6} (\gamma^T [G_i] [P_i]) \right]
\]

(4) Force balance condition;

\[
w = \int p(x,y) dx dy + \int p_c(x,y) dx dy
\]

where \( \lambda^2 \) is the auto-correlation length along the x direction, at which the auto-correlation function of profile in the x direction reduces to 10% of its initial value. \( G \) is the constant shown in Ref. [10].

(5) Force compliance relation of the mean asperity contact pressure and average gap between two contact surfaces given by Ren and Lee [10];

\[
\frac{h}{\sigma} = \exp \left[ \sum_{i=1}^{6} (\gamma^T [G_i] [P_i]) \right]
\]

(6) Force balance condition;

\[
w = \int p(x,y) dx dy + \int p_c(x,y) dx dy
\]

where \( \lambda^2 \) is the auto-correlation length along the x direction, at which the auto-correlation function of profile in the x direction reduces to 10% of its initial value. \( G \) is the constant shown in Ref. [10].

(7) Force balance condition;

\[
w = \int p(x,y) dx dy + \int p_c(x,y) dx dy
\]

where \( \lambda^2 \) is the auto-correlation length along the x direction, at which the auto-correlation function of profile in the x direction reduces to 10% of its initial value. \( G \) is the constant shown in Ref. [10].

(8) Force balance condition;

\[
w = \int p(x,y) dx dy + \int p_c(x,y) dx dy
\]

where \( \lambda^2 \) is the auto-correlation length along the x direction, at which the auto-correlation function of profile in the x direction reduces to 10% of its initial value. \( G \) is the constant shown in Ref. [10].

(9) Force balance condition;

\[
w = \int p(x,y) dx dy + \int p_c(x,y) dx dy
\]
where $\mu_{ad}$ is the adhesion coefficient; $\mu_s$ is the shear traction coefficient of lubricant determined by the following formula; $\mu_s$ is the friction coefficient between asperities contacts (i.e., boundary friction coefficient) estimated by the experiments.

$$
\mu_s = \frac{\eta \Delta u}{p_w h_0}
$$

where $\Delta u$ is the sliding velocity between two contact surfaces; $p_w$ is the mean pressure of water film obtained from the Hertzian pressure distribution over the contact region, $h_0$ is the compliance at the center of contact.

Regarding the numerical calculating method, the authors firstly changed all the equations to be dimensionless ones and then discretized those dimensionless equations on a rectangular uniform grids that were extended over the domain $[(x,y) \in \Omega]$. Finally, the discretized dimensionless equations are solved numerically by the multi-level method introduced by Venner and Lubrecht [11].

Several theoretical results on the relationship between the adhesion coefficients (involved the water film thickness) and the running speed, the water temperature, the contact pressure, the surface roughness amplitude and the parameter of roughness orientation under wet conditions were obtained. The common conditions for numerical calculations are shown in Table 1.

Figure 2 indicates the relations of running speed versus the adhesion coefficient as well as the nominal central water film thickness (i.e., the film thickness at the point of $x = y = 0$). It is obvious that the adhesion coefficient greatly decreases with an increase of the running speed, while the nominal central water film thickness increases against the adhesion coefficient.

Figure 3 shows the effect of water temperature on the adhesion coefficient as well as the nominal central water film thickness. With an increase of the water temperature, it is visible that the adhesion coefficient increases, while the nominal central water film thickness decreases against the water temperature.

Figure 4 shows the relationship between contact pressure caused by wheel load versus the adhesion coefficient as well as the nominal central water film thickness. It is visible that the adhesion coefficient does not change significantly with increasing contact pressure. On the other hand, the nominal central water film thickness changes slightly with increasing contact pressure. Since the adhesion coefficient changes associated with the nominal central water film thickness in general, the plot of adhesion coefficient was enlarged on the right side. It is possible to see a phenomenon that the adhesion coefficient initially increased at lower contact pressure, but then tended to decrease when contact pressure increases. This tendency is considered in relation to the ratio of supporting total load by water film and surface roughness asperities, respectively [12].

Figure 5 shows the effect of surface roughness on the adhesion coefficient as well as the nominal central water film thickness. Both the adhesion coefficient and the nominal central water film thickness increase with an increase of the surface roughness.

### Table 1  Common conditions in the calculations

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius of wheel, $R_w$, $R_i$</td>
<td>0.43m, $\infty$</td>
</tr>
<tr>
<td>Radius of rail, $R_o$, $R_{o1}$</td>
<td>$\infty$, 0.6m</td>
</tr>
<tr>
<td>Elastic modulus; $E_s$, $E_t$</td>
<td>$2.06 \times 10^5$ Pa</td>
</tr>
<tr>
<td>Poisson’s ratios of wheel and rail; $\nu_s$, $\nu_t$</td>
<td>0.3</td>
</tr>
<tr>
<td>Hertzian maximum pressure; $P_h$</td>
<td>750 MPa</td>
</tr>
<tr>
<td>Running speed; $u_1$, $u_2$</td>
<td>300 km/h, 0</td>
</tr>
<tr>
<td>Slip ratio; $\xi$</td>
<td>0.2%</td>
</tr>
<tr>
<td>Temperature; $T$</td>
<td>20 $^\circ$C</td>
</tr>
<tr>
<td>Density; $\rho_0$</td>
<td>0.998</td>
</tr>
<tr>
<td>Viscosity; $\eta_s$</td>
<td>$1.009 \times 10^{-3}$</td>
</tr>
<tr>
<td>Surface roughness; $\sigma_i$, $\sigma_f$ (r.m.s.)</td>
<td>0.44 $\mu$m</td>
</tr>
<tr>
<td>Roughness orientation; $\gamma$</td>
<td>1</td>
</tr>
<tr>
<td>Boundary friction coefficient; $\mu_b$</td>
<td>0.45</td>
</tr>
</tbody>
</table>
Figure 4 expresses the effect of contact pressure on the adhesion coefficient as well as the nominal central water film thickness. It is visible that the adhesion coefficient in the case where \( \gamma > 1 \) (i.e., the roughness asperity is longitudinally oriented), is higher than other cases, while a reverse tendency has presented in the nominal central film thickness. The presumable reason is that transversely oriented roughness \( (\gamma < 1) \) offers larger resistance to water flow than longitudinally oriented roughness \( (\gamma > 1) \). In Fig. 6, it is visible that the effect of \( \gamma \) on the adhesion coefficient is greater than other influential factors introduced above.

Summarizing all of the theoretical results mentioned above, it is known that increases in water temperature, surface roughness and surface pattern orientation, can increase the adhesion coefficient. Among these factors affecting adhesion coefficients, the most significant is roughness orientation.

3. Experimental investigation

Figure 7 shows a twin-disc rolling contact machine and a water spray system adopted in this study. The machine is equipped with two 150kW D.C. motors to control a 300mm diameter disc (termed driving-side wheel disc) and another disc of 170mm in diameter (termed the breaking-side rail disc), respectively. It is noted that for adhesion experiments, the DC motor on the rail side is disconnected, and the Eddy Current Brake (E.C.B.) installed in rail-disc side axle is employed to apply a braking torque onto the rail disc gradually producing a precise slip between the two discs. The wheel disc was made of actual wheel material based on the Japanese Industrial Standard (JIS) as adopted in Japanese Railways [13]. The rail disc was cut from an actual rail (JIS E1101: 60kg-rail) and formed as shown in Fig. 8. The contact configuration between the wheel and rail discs is described in Fig. 9. The cross section of the circumferential surface of the wheel disc is flat and that of the rail disc has a radius of 600mm on its crown, in order to simulate high contact pressure between actual wheel and rail. Table 2 gives functional specification of the twin-disc rolling contact machine.

To simulate wet conditions, water was sprayed from a nozzle with a flow rate of 650ml/min, which should be
enough for a water film to form between the wheel and rail discs. A thermometer was installed at the tip of the nozzle to measure the water temperature. The tolerance of measured data is \( \pm 2 \, ^\circ C \).

In the experiments, when both discs were accelerated to a target value of rolling speed, water was sprayed into the contact interface of the two discs, and then braking torque was applied by the E.C.B. The testing machine automatically stops if the slip ratio exceeded a pre-set value. Based upon the circumferential speeds of the two discs and the resistance torques on the axle of the wheel disc, the relationship between the traction coefficient and the slip ratio was estimated as given below. Supposing the resistance torque on the axle of the wheel disc is \( T_{R} \), and then it can be expressed as:

\[
T_{R} = F \cdot R_w - \omega_w \cdot I_w
\]  

where \( F \) is the traction force between the wheel and rail discs; \( R_w, \omega_w \) and \( I_w \) are the radius, rotational acceleration and moment of inertia of the wheel disc, respectively. The traction coefficient \( \mu \) can be described by the following formula:

\[
\mu = \frac{T_w + \omega_w \cdot I_w}{R_w \cdot W}
\]  \( (7) \)

where \( W \) is the radial load applied between the wheel and rail discs. If \( u_1, u_2 \) are the circumferential rolling speeds of wheel and rail discs, respectively, and \( u = (u_1 + u_2)/2 \) as the average circumferential rolling speed of the two discs, the slip ratio \( \xi \) can be expressed as:

\[
\xi = \frac{u_1 - u_2}{u}
\]  \( (8) \)

With the above equations (7) and (8), it is possible to obtain the relationship between the traction coefficient and the slip ratio. Since the maximum traction coefficient is defined as the adhesion coefficient in the railway system, the equation (7) can also be used to estimate the adhesion coefficient \( \mu_{ad} \) as the maximum value of the traction coefficient.

Table 3 gives the experimental arrangement. The water temperatures at 5, 20 and 50\(^\circ\)C were adopted to simulate filed environments (actual rail temperature) for winter, spring (or autumn) and summer. The magnitude of the radial load corresponding to wheel load was set 3.5kN (\( Ph = 751 \text{ MPa} \)) in reference to the appropriate contact pressure between worn wheel and worn rail in Japanese Railways. Before the experiments, the circumferential surfaces of wheel and rail discs were polished by various abrasive papers with grain size from P60 to P800 (JIS R6010) in order to make them close to the ones of new or worn actual wheel and rail. The surface roughness, expressed in root mean square (r.m.s.) in the table was measured by means
of a stylus profiler along the axial direction of the rail disc. The surface roughness measured on the actual worn rail surface was about 0.3μm statistically.

Figure 10 shows the variation of the maximum traction coefficient (almost the same as the adhesion coefficient) with the increase in rolling speed, obtained for three water temperatures of 5, 20 and 50 °C. In the figure, blue, green and red circles express all the values obtained in the experiments at water temperatures of 5, 20 and 50 °C, respectively, moreover, solid lines depict the curve fitted lines of the maximum traction coefficients of three kinds of water temperatures. Although the maximum traction coefficients shown in the figure show some variation even under almost the same contact conditions, it is not difficult to identify that the water temperature has a significant effect on the adhesion coefficient and in fact, the increase of water temperature contributes to the increase of the adhesion coefficient. The reason for some variation in the maximum traction coefficient is considered to be linked to the surface condition in rolling contact, such as oxide formed on the surface, surface roughness changed by plastic deformation and so on.

Figure 11 shows the effect of contact pressure on adhesion characteristics with the roughness parameter. In this figure, at a low rolling speed of 30km/h, the maximum traction coefficient decreases with an increase of contact pressure in spite of surface roughness. However, at high rolling speed of 80km/h, a quite different tendency between the grain size of P60 and P600 can be observed. At this stage, the reason for such large difference cannot be explained clearly. Plastic deformation of the surface roughness asperity related to solid contact area - or in other words, the real contact area - and the rate of water pressure and solid contact pressure against the applied load may contribute to this interesting tendency. Further study is needed in the future to understand this question.

Figure 12 shows the maximum traction coefficients under various rolling speeds and surface roughness. The surfaces of the wheel and rail discs were polished with abrasive papers in grain size of P80, P320 and P800, respectively, and the roughness configurations of the two discs were arranged longitudinally in relation to the rolling direction of the discs (i.e., γ > 1). It can be seen in the figure that the adhesion coefficient increases with the increase of surface roughness, but on the contrary, decreases with the
increase of rolling speed.

Figure 13 shows the relationship between the maximum traction coefficient and rolling speed for three types of roughness orientation. Here, the surface of the wheel disc was arranged longitudinally in relation to the rolling direction, and that of the rail disc was manufactured to three types of the roughness orientation with the abrasive paper in grain size of P80. The photos in the figure show the surface configurations before the experiments with the three types of roughness orientation $\gamma > 1$, $\gamma = 1$ and $\gamma < 1$ on the rail disc. It is clearly identified in Fig.13 that the adhesion coefficient was largest in the case of $\gamma < 1$, in the middle for $\gamma = 1$ and smallest in the case of $\gamma > 1$. Moreover, the difference in adhesion coefficient among the three types of roughness orientation grew as rolling speed increased. Such experimental results, however, were obviously opposite to the theoretical results as shown in Fig.6. The difference between experimental results in Fig.13 and analytical results in Fig.6 should be based on the fact that the analytical model, particularly EHL theory, assumes no lateral water flow and only longitudinal water flow. This means longitudinal orientation of roughness has a function of water flowing very easily in the longitudinal direction whereas the opposite is true for the lateral orientation of roughness where water flows with great difficulty in the longitudinal direction. However, in both real and experimental wheel/rail interface conditions the potential exists not only for longitudinal water flow but also for lateral water flow so that water flows laterally without any difficulty even if the lateral orientation of roughness is formed on rail surface. In fact, the influence of the lateral orientation of roughness on water flow should be quite different between the actual and experimental cases, and the analytical case. From this point of view, the analytical model should be improved.

4. Discussions

In summary of the results obtained by numerical analysis and experiments in this study, the tendency of the effects of running speed, water temperature, wheel load and surface roughness on the adhesion coefficient are recognized being identical between theory and experiment. The adhesion coefficient increases with the increase of water temperature and surface roughness. The influence of wheel load on the adhesion coefficient, however, may depend on surface roughness and running speed. The adhesion coefficient decreases with the increase of the wheel load in the case of large roughness and low running speed, but increase in the case of small roughness and high running speed. With regard to such interesting experimental results, more detailed study focusing on plastic deformation of the asperity of surface roughness and water pressure at the wheel/rail interface must be advanced. Finally based on the experimental and theoretical results shown in this study, it is considered that a method to increase water temperature or surface roughness is effective to increase or maintain the adhesion coefficient at a high enough level.

5. Conclusions

The conclusions obtained from this study are as follows:
(1) The effects of running speed, wheel load, water temperature and surface roughness on the adhesion coefficient have been investigated employing theoretical and experimental approaches.
(2) The effects of running speed, wheel load, water temperature and surface roughness on the adhesion coefficient were identified by theoretical analysis and experimental investigation.
(3) The adhesion coefficient can be kept high under wet conditions by raising water temperature or increasing the surface roughness.
(4) There are still difficulties in understanding clearly the effect of contact pressure on the adhesion coefficient.
(5) It was clearly identified that the adhesion coefficient was largest in the case of $\gamma > 1$, longitudinal orientation of roughness, middle in the case of $\gamma = 1$, no specific orientation of roughness, and smallest in the case of $\gamma < 1$. 

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\( \gamma < 1 \), lateral orientation of roughness, under parametric study based on an analytical model. But the experimental results were opposite to the analytical results. More detailed investigation is therefore required.

References


