INVESTIGATION OF EFFECTS OF DIFFERENT BRAKING SYSTEMS ON RAIL WHEEL SPALLING

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Abstract

Wheel tread defects cost international railway industry several hundred millions of dollars annually for the repair and replacement, affecting freight and transit system alike. Based on their initiation mechanism, these defects can be classified as cracking, spalling and shelling.

In this study, the specific problem of wheel tread defect, observed in LHB coach wheel, was investigated. The problem was observed after the change in braking system from tread braking to disc braking. The problem identified was spalling damage and investigation was carried out to determine effect of braking system on wheel spalling. In this work, a multi-body parametric model of a LHB coach was simulated using ADAMS/Rail and the rail wheel contact dimensions were determined. After studying the rail wheel interaction, different mathematical models were utilised to calculate sliding velocities and interface temperature rise in the wheel during sliding. A Finite Element (FE) Model was developed for computation of temperature field in the wheel. Using this information and its effect on resulting phase transformation in the wheel material, the extent of damage was computed.

The outcome shows that the wheels with disc brakes have more tendency to slide due to higher braking force. High temperature generated at the rail-wheel interface during sliding causes the wheel to spall. The extent of damage was found to be dependent on sliding duration and the interface temperature.

Keywords: Wheel-Rail System, Railway Wheel Sliding, Spalling, Tread Defect, Contact Temperature, Heat Conduction, Phase Transformation

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>a</td>
<td>Contact patch semi-major axis, m</td>
</tr>
<tr>
<td>α</td>
<td>Deceleration, m/s²</td>
</tr>
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<td>Fb</td>
<td>Braking force, N</td>
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<td>Ft</td>
<td>Total opposing force, N</td>
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<tr>
<td>L</td>
<td>Wheelset inertia, kg m²</td>
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<tr>
<td>Ms</td>
<td>Martensite start temperature, °C</td>
</tr>
<tr>
<td>Mf</td>
<td>Martensite finish temperature, °C</td>
</tr>
<tr>
<td>M</td>
<td>Mass, kg</td>
</tr>
<tr>
<td>Nw</td>
<td>Wheel load, N</td>
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<tr>
<td>NA</td>
<td>Normal load on the wheel set, N</td>
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<tr>
<td>Po</td>
<td>Maximum Hertz contact pressure, N/m²</td>
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<td>q</td>
<td>Heat flow rate, W/m²</td>
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<tr>
<td>R</td>
<td>Wheel radius, m</td>
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<tr>
<td>s</td>
<td>Sliding distance, m</td>
</tr>
<tr>
<td>t</td>
<td>Time, s</td>
</tr>
<tr>
<td>Tm</td>
<td>Braking torque, Nm</td>
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<tr>
<td>V</td>
<td>Initial velocity of vehicle, m/s</td>
</tr>
<tr>
<td>Vₛ</td>
<td>Sliding velocity, m/s</td>
</tr>
<tr>
<td>α</td>
<td>Deceleration of vehicle through wheel radius, deg/s²</td>
</tr>
<tr>
<td>βw</td>
<td>Thermal penetration coefficient of wheel, Wₜₐₓ/°C.K.m²</td>
</tr>
<tr>
<td>βt</td>
<td>Thermal penetration coefficient of rail, Wₜₐₓ/°C.K.m²</td>
</tr>
<tr>
<td>ε</td>
<td>Heat partitioning parameter</td>
</tr>
<tr>
<td>λ</td>
<td>Thermal conductivity of material, W/m.K</td>
</tr>
<tr>
<td>µbt</td>
<td>Co-efficient of friction between brake and wheel tread</td>
</tr>
<tr>
<td>µwr</td>
<td>Co-efficient of friction between wheel and rail</td>
</tr>
<tr>
<td>σ</td>
<td>Thermal diffusivity, m²/s</td>
</tr>
<tr>
<td>ω</td>
<td>Angular Velocity, rad/sec</td>
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</table>

Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>BCT</td>
<td>Body Centered Tetragonal</td>
</tr>
<tr>
<td>FCC</td>
<td>Face Centered Cubic</td>
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<tr>
<td>FEA</td>
<td>Finite Element Analysis</td>
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<tr>
<td>IT</td>
<td>Isothermal Transformation</td>
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<tr>
<td>LHB</td>
<td>Linke Hofmann Busch</td>
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<tr>
<td>MBS</td>
<td>Multi-Body Systems</td>
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<tr>
<td>MMU</td>
<td>Manchester Metropolitan University</td>
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<tr>
<td>RWF</td>
<td>Railway Wheel Factory</td>
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<tr>
<td>TAZ</td>
<td>Thermally Affected Zone</td>
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</table>

1. INTRODUCTION

Wheel tread defects are a major cost to virtually all railway networks worldwide. Almost all the rail systems are subjected to some amount of spalling and shelling, thus limiting the service life of wheel. Several millions of dollars are spent worldwide for replacement and/or repair of spalled or shelled wheels. Along with high cost of repair and replacement, damaged wheels inflict tremendous load on rails and other components of coaches, causing undesirable noise and vibration, adversely affecting passenger comfort.

Although there are various types of tread defects, this study focuses on the specific problem of spalling observed in the wheels of LHB coaches of Indian railways. It is hypothesised that the problem observed is due to change in the braking system from tread brakes to disc brake system. So, this study is limited to investigating the effects of these two braking systems on wheel spalling.

Spalling is the wheel surface damage caused by the wheel sliding over rail, generally during braking. During sliding, as a result of frictional heating, wheel
Surface temperature increases tremendously (above lower critical temperature of steel). This causes pearlite in the contact patch of the wheel material to transform to martensite, which is brittle. This change in material structure leads to localized expansion of material in the contact patch. Resulting high residual stresses, combined with rolling contact load cycles cause martensitic patches to break out of wheel tread causing series of elliptical damage patches on the wheel tread [1-3].

Higher braking force causes wheels to get locked and slide over rail and this phenomenon is called sliding. Jian Sun and Sawley [2] derived the theory of wheel sliding, which predicts that when braking torque exceeds the torque generated due to adhesive force, wheel starts sliding. The insufficient adhesion between wheel and rail is cited as the main cause of wheel sliding by Magel and Kalousek [3]. They also observed that adhesion reduces drastically with the contamination like wet rails, leaves, and dust etc. causing variable friction condition. This study also indicates that the skid flats are more prominent in the empty wagons and coaches because of reduced adhesive force.

During wheel sliding, as a result of friction, large amount of heat is generated at the wheel rail interface. Fundamental work in the area of heat generation at the sliding bodies was done by Block [4] and Jaeger [5] with the theory of flash heating. Jaeger [5] proposed the theory of heat partitioning parameter under pure sliding between semi-infinite solids. The theory proposed by Jaeger [5] was extended by Archard [6] and gave the solution for the temperature rise in rubbing surfaces. Tanvir [7] gave the analytical formulation of sliding problem with the Hertz’s elliptical contact area and proposed an easy way for temperature calculation. Knothe and Liebelt [8] applied the theory of Jaeger and Archard and showed that the equations developed for maximum contact temperature agrees with the flash contact temperature theory developed by Block [4] and Jaeger [5]. In further development, Knothe and Ertz [9] compared the analytical and numerical method for calculation of wheel rail interface temperature and proposed a simple equations for interface temperature calculation which conformed with all the theories developed earlier.

On the other hand, determination of heat partitioning parameter using FE methodology was developed by Kennedy et al [10]. They studied the variation in heat partitioning parameter with respect to time and sliding velocity. They also showed that during sliding, maximum temperature in wheels is reached in the first 50 to 100 ms. and then was constant throughout the duration of sliding. The approximate solution for the heat partitioning parameter using simplified formula was developed by Swaley [11].

The study conducted by Ahlstrom and Karlsson [12] with full scale wheel flat experimentation defines the difference between Thermally Affected Zone (TAZ) and martensite zone. They showed that during sliding high stretching inclusion takes place in the highly deformed surface layers resulting in crack propagation around the martensite. Ahlstrom and Karlsson [13] modelled the phase transformation using FEA and observed that the thickness of transformed layer was strongly dependent on the axle load. The large sliding duration prolongs cooling while small sliding durations increases the cooling rate thus facilitating martensite formation. Hang et al [14] observed that the ferrite and martensite structures formed after cooling had strength lower than that of existing ferrite and pearlite structure. Therefore, it can easily become channel of crack initiation and propagation.

The initial wheel material is pearlitic in nature with FCC crystal structure. On the other hand martensite which is untempered has BCT crystal structure. So during perlite to martensite transformation in the contact patch, 0.5% volume expansion is observed, which leads to setting up of residual stresses in the contact patch [15]. When these residual stresses combine with the rolling contact loading cycle, hard and brittle martensite breaks out of wheel tread.

In this work an investigation was carried out on the effect of different braking systems on the wheel spalling observed in LHB coach.

2. MODELLING

2.1 Geometric Modelling

The aim behind geometric modelling of rail and wheel was to get profile data for the multi-body simulation. The wheel used in LHB coaches in Indian Railways was selected for analysis. The rail profile selected was IRS 52 rail used in the Indian railway network.

Fig. 1 The LHB coach model in ADAMS/Rail
2.2 Rigid Body Modelling of Railway Vehicle

A number of MBS packages are available in the market to carry out dynamic analysis of railway vehicle some of the examples are ADAMS/Rail, Vampire, Nucars and Simpack. Benchmarking of these softwares has been carried out in the MMU, Iwnicki [16]. In this study, specially designed MBS package ADAMS/Rail was used for the dynamic simulations.

The LHB coach and bogie were modelled (Figure 1), by taking European standard passenger coach as reference. The model contains primary and secondary suspension, vertical, lateral and yaw dampers, lateral and vertical bump stops. All the inertial properties, suspension parameters, gross weight were imputed to the model as per LHB coach specification.

The wheel and the rail profiles were defined using coordinate data obtained from the geometric model. The gauge, vertical and lateral alignments were defined in the system as per Indian railway specification.

2.3 Analytical Solution for Sliding Velocity

Forces acting on the wheel during braking are shown in Figure 2. The wheel slides over rail if braking torque exceeds the adhesive torque. The wheel gets locked when Eqn. 1 is satisfied.

\[ F_b \mu_b R_b > N_w \mu_{wr} R \]  

For calculation of sliding velocity, the following assumptions are made:

- Wheels roll at the centerline
- Track irregularities are negligible
- Braking force and coefficient of friction between brake disc and brake pad do not change during sliding

Angular deceleration of wheel-set depends on the balance of braking and wheel rail adhesion (Eqn. 2).

\[ \alpha = \frac{(2F_b \mu_b R_b - N_w \mu_{wr} R)}{I} \]  

Once the angular deceleration of the wheel-set is known, vehicle deceleration is calculated using Eqn. 3

\[ a_x = \frac{F_{w}}{M} \]  

Sliding velocity is calculated using Eqn. 4

\[ V_s = V - (a_x \cdot t) \]

where \( t \) is the time required to decelerate the wheel to zero velocity and is given by the relation

\[ t = \frac{\alpha}{a} \]  

2.4 Mathematical Model for Interface Temperature Rise

To analyse the condition where velocities of wheel and rail with respect to contact patch are different, a dimensionless parameter, \( L \), called “Peclet” number, which is the ratio of the surface speed to the rate of diffusion of heat, is used. Parameter \( L \) has to be taken into account for heat conduction problem. As given by Knothe and Ertz [9], if \( L > 10 \), the heat conduction occurs only perpendicular to the contact plane. In this situation, heat flow in longitudinal and lateral directions can be neglected.

The analytical solution for interface temperature rise was based on the following assumptions:

- There was pure sliding in whole contact area
- Coefficient of friction between wheel and rail was constant.
- Contact patch dimensions were constant and not varying with slide duration and temperature.
- The effect of elastic deformation was negligible
- The kinetic energy was transformed completely into heat.

According to Knothe [9] maximum temperature rise in the wheel rail interface is given by equation

\[ T_{max} = \frac{1.276 \epsilon \mu V_a P_v}{\beta_w \sqrt{V_s}} \]  

Average temperature rise in the wheel rail interface was calculated from Eqn. 6 by integrating with respect to pressure variation in the elliptical contact area and is given by the relation

\[ T_{avg} = \frac{0.853 \epsilon \mu V_a P_v}{\beta_w \sqrt{V_s}} \]  

Heat generated at the interface flows through the wheel and rail. Amount of heat flow is governed by the surface temperature of the wheel and rail. Amount of heat flow to the wheel and rail with respect to sliding duration is governed by the heat partitioning parameter [11], and the relation is given in Eqn. 8.

\[ \epsilon(t) = \frac{t^{0.5}}{t^{0.5} + 0.863 \frac{a}{V_s}^{0.5}} \]

Thus the heat flow through the wheel is

\[ q_w = (1 - \epsilon) q_f \]

and heat flow through the rail is

\[ q_r = \epsilon q_f \]
2.5 FE Modelling for TAZ and Cooling Rate

Various simplifying assumptions were used for development of finite element model. The problem was treated as two dimensional based on the study of Kennedy et al [10]. A uniform pressure distribution was assumed instead of Hertzian contact pressure. The model developed was strictly for thermal analysis. The deformation was not taken into account and size of the contact patch was taken from MBS.

The finite element model for the analysis of TAZ and cooling rate was constructed (Fig.3). The commercial meshing software ‘Hypermesh’ was used for meshing of the model. Since the depth of thermally affected zone is expected to be very small, a fine mesh was used in the contact patch area. Away from it the mesh was made coarser to limit the size of the problem.

![Fig. 3 2D FE model for thermal analysis](image)

2.6 Thermal parameters and transformation characteristics

The thermal properties of the material used were functions of temperature and phase only and were calculated based on the law of mixture from the data of individual phases.

- The thermal conductivity ($\lambda(T)$), specific heat ($c_p(T)$), and density were taken from the ASM handbook [17].
- The Iso-thermal diagram for material SAE 1060 steel was reproduced from Atlas of time-temperature diagrams (Fig. 4) and was used to describe the whole phase transformation sequence [18].

![Fig. 4 Iso-thermal diagram for SAE 1060 steel](image)

3. RESULTS AND DISCUSSIONS

3.1 Rail Wheel Contact Patch and Vertical Forces

The axisymmetric model was meshed with 2-D, 8-node element. Plane 77,5547 elements were used in the model.

The experimental results and the analytical model developed showed that the contact surface was initially heated very fast and stabilised at a temperature 1000°C to 1300°C. With this observation the boundary condition (Fig. 3) were applied in such a way that the surface temperature rapidly rises and stabilises during the sliding.

![Fig. 5 Semi-axes of elliptical contact patch (a) in longitudinal direction; (b) in lateral direction](image)
on contact patch were extracted and are shown in Fig. 5(a), 5(b) and 6.

Gross weight of the vehicle (66 Tonnes) was observed to be distributed equally on all the eight wheels (80932.5 N at each wheel) and was not varying with the running speed.

3.2 Wheel Lockup and Sliding Velocities

The wheel lockup conditions were analysed based on Eqn. 1 and the results are summarised in Table 1.

<table>
<thead>
<tr>
<th>Braking System</th>
<th>Braking Force per axle (N)</th>
<th>Normal Load on axle (kN)</th>
<th>Braking Torque (N m)</th>
<th>Adhesive Torque (N m)</th>
<th>Wheel Sliding</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tread</td>
<td>8000</td>
<td>162</td>
<td>3000</td>
<td>3845</td>
<td>No</td>
</tr>
<tr>
<td>Disc</td>
<td>14225</td>
<td>162</td>
<td>6757</td>
<td>3845</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Table 1. Wheel lockup condition

It is observed that in case of tread braking system, the braking torque is less than the adhesive torque. This condition represents no wheel lockup thus, no sliding and no spalling. On the other hand the braking torque with the disc brake system is more than the adhesive torque resulting in wheel lock up. As wheel is locking, there is possibility of spalling damage of the wheel tread because of wheel sliding. So, further analysis was restricted to the disc braking system.

3.3 Interface Temperature Rise

The temperature rise because of frictional heat at the contact patch was distributed to the wheel and rail according to the heat partitioning parameter. Figure 7 shows the variation in heat partitioning parameter with respect to sliding duration at various sliding velocities (41.31 m/s, 25.82 m/s and 12.91 m/s).

The heat partitioning parameter is the function of contact patch semi-axis length in rolling direction and the sliding velocity of the vehicle. As the sliding velocity decreases from 41.31 m/s to 12.91 m/s, heat partitioning curve takes more time to reach its steady state level. But, instead of large deviation in sliding velocity, the heat partitioning parameter reaches its steady state value in 50 to 100 ms.

Figure 8 shows the plot of rise in temperature with the sliding duration. From the plot it is observed that if the sliding velocity decreases the interface temperature also decreases. Moreover, the temperature rises drastically in the first 50 ms and is stable throughout the sliding duration. The temperature rise with 41.31 m/s and 25.82 m/s sliding velocity was observed to be 1267°C and 989°C, higher than the lower crystallisation temperature (LCT) of steel. Hence, there is a possibility of formation of martensite at the two sliding speeds. The temperature rise at 12.91 m/s sliding speed would reach a marginally higher level than LCT (726°C). This temperature is not sufficient to completely austenitise the TAZ. After cooling, the martensite would appear to be a very thin surface layer on the tread.

Based on the above results, the wheel was analysed for different sliding durations at 41.31 m/s and 28.52 m/s sliding velocity.

3.4 Thermally Affected Zone and Cooling Rate

The initial microstructure is pearlite in nature. The phase transformation was predicted by superimposing the cooling curves on the Isothermal Transformation (IT) diagram. The exact amount of martensite present in the composition can only be determined by experiments. The ambient condition was taken to be 30°C and convection on wheel tread was neglected.

Case 1: sliding velocity = 41.31 m/s, sliding duration = 3 sec, interface temperature = 1297°C

During the heating period the material on the tread surface follows the boundary condition applied. The material at a depth of 2.46 mm lags and reaches a temperature of 790°C (Figure 9(b)). The cooling rates at the surface and at a depth of 2.46 mm differ, with slower cooling observed below the surface.
Figure 9(a) shows the austenite zone when heating stops. It is observed that the austenite zone is 2.48 mm deep (above 750°C) followed by a 0.01 to 0.03 mm of transient layer (between 750°C and 723°C). Figure 10 shows the cooling curves superimposed on IT diagram of steel. The time for start of cooling curve for transformation was selected as 2.56 sec, after heating ends, i.e. when material reaches 750°C. This time was selected because the material near the surface was within appropriate temperature range for the diffusive phase transformation to occur. The cooling curve of the material close to the surface just touches the pearlite start line, which means that it is completely martensitic in nature. While the material at the depth of 2.48 mm crosses the pearlite line, the transformation is not completed. Hence, the zone contains martensite, pearlite, bainite and ferrite. This indicates the gradual reduction of fraction of martensite from the tread up to TAZ. But, still the zone contains un-tempered martensite.

In Case 2, most of the parameter values were same as in case 1. Only the heating time was extended to 5 sec. The temperature contours for this case are shown in Fig. 11(a). The same trend in cooling curves, as in Case 1, was observed (Fig. 11(b)). The material at a depth cools slowly as compared to the material at the surface. The TAZ was about 2.46 mm deep. The cooling curve when superimposed on IT diagram (Fig. 12), shows that only the fraction of austenite is converted to pearlite, bainite and ferrite, which indicates that the TAZ contains more untempered martensite.

In this case also the parameters similar to Case 1 were used. But, the heating was extended to 10 sec. The temperature contours at the end of the heating period are shown in Fig. 13(a). It is observed that the TAZ is 2.96 mm deep. The cooling rate curves, as shown in Fig. 13b...
indicate slow cooling rate as compared to Case 1. This is because longer heating time results in smaller temperature gradient. The local cooling rate is also affected by the local temperature distribution, which, in turn, is affected by the latent heat due to the formation of ferrite/pearlite/bainite.

The phase transformation taking place is shown in Fig. 14. Due to slow cooling rate, the austenite region shows complete transformation of austenite to pearlite at the TAZ depth, while the surface cooling rate is still high enough to form martensite.

In this case, the temperature rise corresponding to 25.82 m/s sliding velocity was considered. The peak temperature is observed to be 1019°C. The temperature history contours are shown in Fig. 15a, indicating about 1.64 mm TAZ. The TAZ is much smaller because of lower saturation temperature. The cooling curves, shown in Fig. 15(b), indicate that the cooling rate is much faster compared to Case 1.

Calculation of the amount of developing phases from IT diagram and the cooling curves from austenite decomposition temperature are given in Fig. 15(c). The results indicate that the cooling curves just touch the pearlite start curve. This indicates very small amount of ferrite/pearlite/bainite transformation. The TAZ, therefore, is completely martensite.

In this case, the parameters similar to Case 4 were used. Only the heating time was increased to 10 sec. From the temperature history counters (Fig.17), it is observed that due to increase in heating period, the TAZ has increased to 2.13 mm. The cooling curves (Fig. 16c) indicate that the material near tread surface is completely transformed to martensite, while the material near TAZ boundary at a depth shows almost 80% martensite.

**Fig. 13 Case 3 (a) Temperature distribution in the material below contact area (b) Heating and cooling cycles**

**Fig. 14 Case 3: Phase development during slide**

**Case 4: sliding velocity = 25.82 m/s, sliding duration = 5 sec, interface temperature = 1019°C**

**Fig. 15 Case 4 (a) Temperature distribution in the material below contact area (b) Heating and cooling cycles (c) Phase development during slide**

**Case 5: sliding velocity = 25.82 m/s, sliding duration = 5 sec, interface temperature = 1019°C**

Case 5 is very similar to Case 4 in terms of input parameter values. Only the heating time was increased to 5 sec. In accordance with the observation from temperature history contours, shown in Fig.16(a), austenite is almost 1.97 to 2 mm thick layer for the heating time of 5 sec. The delay in the cooling rate, shown in Fig.16(b), is first due to formation of ferrite/pearlite/bainite and secondly due to martensite formation.

The phase transformation, shown in Fig.16c indicates that the material near tread surface is completely transformed to martensite, while the material near TAZ boundary at a depth shows almost 80% martensite.

**Case 6 sliding velocity = 25.82 m/s, sliding duration = 10 sec, interface temperature = 1019°C**

In this case, the parameters similar to Case 4 were used. Only the heating time was increased to 10 sec. From the temperature history counters (Fig.17), it is observed that due to increase in heating period, the TAZ has increased to 2.13 mm. The cooling curves (Fig. 16c) indicate that the cooling curves just touch the pearlite start curve. This indicates very small amount of ferrite/pearlite/bainite transformation. The TAZ, therefore, is completely martensite.
18(a)) indicate slower cooling rate as compared to Case 4. This is due to higher saturation temperature at wheel rail interface causing low temperature gradient for heat flow.

The resulting microstructure is a mix of ferrite/pearlite/bainite, martensite and retained austenite.

Fig. 16 Case 5 (a) Temperature distribution in the material below contact area; (b) Heating and cooling cycle; (c) Phase development during slide

The phase transformation curves (Fig. 18(b)), indicate that some fraction of austenite is transformed to ferrite/pearlite and bainite for the material near tread surface, while the fraction is more at the TAZ depth.

Fig. 17 Case 6: Temperature distribution in the material below contact area

Fig. 18 Case 6 (a) Heating and cooling cycle; (b) Phase development during slide

The results obtained through Case 1 to Case 6 are compared and it is found that the TAZ depth increases with sliding duration as well as with sliding velocity (Fig. 19). The results are validated against the experiments carried out by Ahlstrom and Karlsson [13]. It shows similar range of TAZ and the cooling characteristics. All the cases analysed show that the pearlite wheel would be transformed to either martensite or the composition of martensite, ferrite/pearlite/bainite and retained austenite.

Fig. 19 Depth of TAZ vs sliding duration and sliding velocity
4. DISCUSSION

The one-dimensional problem of temperature histories during wheel sliding was studied using analytical models. As the models were based on average value of the thermal parameters on the contact area, these were able to predict the influence of various parameters like heating period, saturation temperature, initial temperature etc. when compared with the three dimensional model of Ahlstrom and Karlsson [19]. The one-dimensional model predicts slower cooling rate. The difference is marginal and also due to the fact that the divergent flow in the three dimensional model offers efficient heat transfer.

The general advantage of the present methodology is that it can predict the wheel slide very close to the observed results. Though the method is based on numerous assumptions, it is still more efficient for representing actual phenomenon by averaging out various parameters. The FE model for heat conduction has advantage of freedom to define various parameters like conduction and phase with respect to temperature. Moreover, the results obtained are in good agreement with the experimental data of Ahlstrom and Karlsson [13].

The martensite formed during sliding followed by cooling is considered purely dependent on the temperature. The amount of phase transfer is very critical based on the fact that it depends on various factors like cooling rate, time spent in transformation zone, the diffusion of martensite, initial temperature etc. The actual phase present in the TAZ can be predicted only by experimentation, but the present model can predict the change in phase in the TAZ.

The present FE model enables modelling of convective and radiation cooling. It is very difficult to estimate how much heat is dissipated from the skid surface as the velocity of fluid (air) flowing over wheel flat changes drastically with the wheel rotation. Also, the radiation from wheel surface is difficult to describe.

The methodology developed for the investigation, and consequent analysis, indicates its capability of modelling all the phenomena, viz. wheel sliding, generation and flow of frictional heat, resulting temperature rise in wheel, subsequent cooling and resulting phase transformation in TAZ.

Using this methodology, the events of braking of LHB coaches using disc brakes are analysed and the results are presented.

5. CONCLUSIONS

Based on the investigations carried out, the following conclusions are drawn:

- The wheel with the disc brake has a higher tendency to slide due to higher braking torque.
- Wheel is locked very fast (1-3 sec) during hard braking due to lower inertia of wheel set.
- Very high temperatures are rapidly developed when railway wheel slides on the rail (800-1300°C), followed by the rapid cooling resulting in material transformation in thermally affected zone.
- Depth of TAZ increases with increase in saturation temperature, and sliding velocity.
- Surface cracks take years to appear, then grow deeper leading to an uncontrolled crack growth and finally result in wheel shelling.
- The extent of wear with the tread braking is higher than the crack propagation rate. Thus, surface cracks generated get cleaned off during tread braking.

As for the given conditions with disc brakes, wheels have a higher tendency to slide and absence of tread cleaning i.e. magic wear rate, makes it prominent for tread damages.

6. REFERENCES


