REFINING THE PARAMETERS OF ARCHARD’S WEAR MODEL FOR CALCULATING WEAR OF WHEELS APPLIED FOR 25 T PER AXLE FREIGHT WAGONS ON RUSSIAN RAILWAYS

Alina Saidova, Anna Orlova
Department of Railway Cars, Petersburg State Transport University
Moskovskiy prosp., 9, 190031 St. Petersburg, Russia
e-mail address of lead author: a-orlova@yandex.ru

Abstract
The aim of research is a reasonable choice of wear coefficients for severe and mild phases in Archard’s abrasive wear theory and friction coefficients for wheel flange and tread that correspond to conditions of Russian track for wheel profile wear modelling using dynamic models of wagons. The paper considers the gondola wagon on bogies model 18-9855 that are the three-piece Barber S-2-R design. The wagon running test results on Experimental Loop Track in Scherbinka, the results of hardness measurements of wheel flange / tread and of wheel profile wear modelling are presented in paper.

1. OBJECTS OF INVESTIGATION AND AIM OF THE PAPER
The paper considers the gondola wagon on bogies model 18-9855 that are the three-piece Barber S-2-R design for Russian railways [1] (figure 1). The wagon has the wheel base of 8.65 m, axle load of 25 t and design speed of 120 km/h.

The wagon is equipped with wheelsets that have S-shaped disks and hardened surface. Hardness of the wheel profile surface is 320-360 HB. In operation such wheels have shown [2] to provide lower wear rate compared to traditional wheels with 248-285 HB hardness of the wheel profile surface.

Since hardened wheels have been introduced in operation in 2004, wear simulation models for them do not exist. The aim of the study presented in the paper is to implement Archard’s wear model [3] and to determine the parameters of wheel-rail interaction that correspond to conditions of Russian track and application of hardened wheels.

The choice was carried out on base of comparison of test measurements results with results of multivariant modeling in “MEDYNA” software [4].

2. AVAILABLE EXPERIMENTAL DATA
The mentioned gondola wagon was tested for reliability on Experimental Loop Track in Scherbinka. During the experimental run the fully laden wagon was moving with the constant speed of 70 km/h on circular track with 956 m radius in one direction. The rails on the test track had R65 profiles and average gauge width was 1522 mm. Right rail was inner, and left rail was outer. During measurements wagon sides (left and right, front and back) were accepted in direction of motion. Measurement of wheel profiles was done for all wheels in initial condition and after 53 300 km of mileage. Flange width was determined at a height of 18 mm from the flange top. The tread wear was a difference between measured height of flange at a distance of 70 mm from the external side of the wheel and its nominal value (28 mm). The average values of wheels’ flange and tread wear are presented in the table 1.

The measurement data was used for reference when choosing the wear model parameters as far as it refers to the well-determined test conditions compared to experimental data available from field tests [5] of freight wagons.
Table 1  Results of wheel profiles measurements after wagon 53 300 km mileage

<table>
<thead>
<tr>
<th>Wheelset</th>
<th>Linear wear, mm, on flange of left wheel</th>
<th>Linear wear, mm, on tread of left wheel</th>
<th>Linear wear, mm, on flange of right wheel</th>
<th>Linear wear, mm, on tread of right wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>3.8</td>
<td>0.3</td>
<td>1.2</td>
<td>1.1</td>
</tr>
<tr>
<td>Second</td>
<td>0.6</td>
<td>0.6</td>
<td>1.9</td>
<td>0.9</td>
</tr>
<tr>
<td>Third</td>
<td>4.1</td>
<td>0.4</td>
<td>1.6</td>
<td>0.8</td>
</tr>
<tr>
<td>Forth</td>
<td>0.7</td>
<td>0.5</td>
<td>1.0</td>
<td>0.8</td>
</tr>
</tbody>
</table>

3. DYNAMIC MODEL OF WAGON MOTION SUPPLEMENTED WITH ARCHARD’S WEAR MODEL

Simulation of wear was done using the gondola wagon model [6] developed in MEDYNA software that considered measured parameters of the tested wagon suspension. Model consists of 23 rigid bogies (1 carbody, 2 bolsters, 4 sideframes, 4 wheelsets, 8 sections of rails and 4 sections of a track) with interconnections between them.

The Archard’s abrasive wear model [3] is implemented in MEDYNA as a parallel simulation algorithm. The mass of worn material is proportional to the frictional work:

\[ V = k \cdot P \]

with \( V \) – wear mass, [g];
\( k \) – wear coefficient [g/N·m];
\( P \) – frictional work [N·m].

Phases of mild and severe wear have different coefficients of proportionality between volume of worn material and frictional work. Transition from mild to severe wear is accounted by using some value of power of creep forces divided by the area of the contact point (power density).

Frictional work in a segment of the profile is given by:

\[ P = P_k \cdot L \]

with \( P_k \) – wear number, [N];
\( L \) – loaded path length [m].

Wear number is given by:

\[ P_k = s_x \cdot T_x + s_y \cdot T_y \]

with \( s_x, s_y \) – longitudinal, lateral translational creepage;
\( T_x, T_y \) – longitudinal, lateral translational creep forces, [N].

Conditions of numerical experiment corresponded to mileage test conditions of Experimental Loop Track. Those initial parameters that could not be measured directly were varied in order to estimate their influence on wear of wheels and possible values in experiment.

4. SIMULATION RESULTS AND COMPARISON

4.1 Influence of irregularities on wheel wear in experimental curve

For the conditions of Experimental Loop Track the irregularities are not available from measurement. Therefore simulations were done for three cases: recommended by normative documentation irregularities [7], irregularities scaled by 1.5 and scaled by 0.5. The resulting profiles of left wheel of first wheelset after 50 000 km mileages in comparison with new unworn profile are shown in Figure 1.
Comparison of linear wear of treads and flanges showed that for three cases profiles have the difference within 5%. This allowed not to consider irregularity scale in experimental curve for further research.

### 4.2 Determining the switch point between severe and mild wear conditions

Gondola wagon was simulated in empty (6 t per axle load) and fully laden condition in curves with radii of 956 m (Scherbinka loop), 650 m (average curve) and 350 m (sharp curve) at 70 km/h. For average and sharp curves straight section length was 20 m, transition section was 100 m, superelevation was 150 mm.

For each contact point the average creep forces power density was computed. To determine the switch point between severe and mild wear an assumption was made that all wheels’ flanges and outer treads in curves experience severe wear, while inner treads are in mild wear conditions. On straight track flanges are in severe wear conditions and treads experience mild wear.

The results of calculating the average creep forces power density for the first wheelset of wagon (where the most intensive wear appears) in different cases of movement are presented in the table 2.

<table>
<thead>
<tr>
<th>Case of wagon movement</th>
<th>Value of average creep forces power density, MW/m², on the wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>left</td>
</tr>
<tr>
<td></td>
<td>tread flange (first contact patch) flange (second contact patch)</td>
</tr>
<tr>
<td>Laden wagon, straight track</td>
<td>6.3 119.0 0.0</td>
</tr>
<tr>
<td>Laden wagon, curved track with R=956 m</td>
<td>7.2 78.1 0.0</td>
</tr>
<tr>
<td>Laden wagon, curved track with R=650 m</td>
<td>20.2 78.7 1350.0</td>
</tr>
<tr>
<td>Laden wagon, curved track with R=350 m</td>
<td>40.1 133.0 2630.0</td>
</tr>
<tr>
<td>Empty wagon, straight track</td>
<td>6.7 67.0 0.0</td>
</tr>
<tr>
<td>Empty wagon, curved track with R=956 m</td>
<td>7.4 58.8 0.0</td>
</tr>
<tr>
<td>Empty wagon, curved track with R=650 m</td>
<td>9.0 150.0 0.0</td>
</tr>
<tr>
<td>Empty wagon, curved track with R=350 m</td>
<td>18.0 210.0 0.0</td>
</tr>
</tbody>
</table>

Analysis of results showed, that average creep forces power densities on the tread of left and right wheels are approximately equal to each other in the straight track and they are not more than 7 MW/m². In curves creep forces power density on tread of attacking wheel is higher than for unattacking wheel (7.2-20.2 MW/m² in comparison with 3.8-6.2 MW/m² in large and average curves and 18.0-40.1 MW/m² in comparison with 6.0-9.7 MW/m² for sharp curve). So, the switch point between mild and severe wear phases was determined at the power density value of 7 MW/m².

### 4.3 Results of wheels profile hardness measurement

Some researchers have found that good agreement between experimental data and results of wear modeling can be achieved by using hardness on wheel flange higher than the same on wheel tread [8]. Hardness determines the value of penetration of contact surfaces and thus it influences the volume of material in deformation.

During the modeling difference in hardnesses for different zones of wheel and rail can be specified with different friction coefficients for these zones. It is known from the standart [9] that for new unworn wheels difference of hardness, measured along the wheel perimeter, is not more than 20 HB (8%). The relationship of hardness for wheels that were in operation is determined in paper.

Samples without any hardenings, cracks and other defects that could influence on hardness measurement results were cut from extremely worn wheels.

Results of samples hardness measurements in three points of flanges and treads are presented in the table 3. Measurements were made with Rockwell method and results were counted in HB.

Analysis of results showed, that tread hardness was more than flange hardness by about 13% (the data for the second sample wasn’t considered because of its different value). Thus friction coefficient for flange was varied by 10-20% more than the same for the wheel tread.
4.4 Determining the influence of wheel-rail friction coefficient and wear coefficients

Multivariant calculations of wheels wear of model 18-9855 bogies for gondolla movement in curve with curvature 956 m at speed 70 km/h (imitation of conditions of running test on Experimental Loop Track in Scherbinka, mileage is about 50 000 km) were carried out for wear coefficients within a range from $1 \times 10^{-6}$ g/N·m to $3 \times 10^{-6}$ g/N·m and friction coefficients between wheel and rail within a range from 0.2 to 0.3. Track gauge was 1522 mm.

Based on modelling results diagrams of dependence of wheel tread and flange wear on wear coefficients and friction coefficients between wheel and rail are built. They allow to choose best fit pair of coefficient values. Diagrams of wheel wear of the first leading wheelset are presented on figures 2-3.

The best agreement between modeling and test results was when friction coefficient between flange and rail was 0.28, friction coefficient between tread and rail was 0.25, wear coefficient for the strong wear was $2 \times 10^{-6}$ g/N·m and for the mild wear it was $1 \cdot 10^{-6}$ g/N·m. The discrepancy between the results of modeling and measurements is presented in table 4.

In twelve control points discrepancy between the results of modeling and measurements was not more than 33%. But for the flange wear of not attacking wheels of first and third wheelsets and attacking wheel of forth wheelset discrepancy was more than 100%.

It is known, that the combination of tolerances on bogie manufacturing can increase wear rate up to 6.2 mm over 10 000 km, when the mean wear rate is 1.1 mm (i.e. more than 5 times). Tentatively this phenomenon was observed.

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### Table 3 Results of hardness measurements

<table>
<thead>
<tr>
<th>Number of sample and point on it</th>
<th>Tread hardness</th>
<th>Flange hardness</th>
<th>Mean hardness, HB</th>
<th>Difference of hardnesses for flange and tread, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>HRC</td>
<td>HB</td>
<td>HRC</td>
<td>HB</td>
</tr>
<tr>
<td>1</td>
<td>35</td>
<td>331</td>
<td>22</td>
<td>237</td>
</tr>
<tr>
<td>2</td>
<td>26</td>
<td>262</td>
<td>21</td>
<td>233</td>
</tr>
<tr>
<td>3</td>
<td>20</td>
<td>229</td>
<td>22</td>
<td>237</td>
</tr>
<tr>
<td>4</td>
<td>17</td>
<td>217</td>
<td>17</td>
<td>217</td>
</tr>
</tbody>
</table>

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![Figure 2](image-url)  
**Figure 2**  
Diagrams of dependence of tread and flange wear, mm, of the first attacking (left) wheel on wear coefficients and friction coefficients  
a) on flange; b) on tread

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5. CONCLUSIONS AND FURTHER RESEARCH

The comparison between available data on wear of wheels with hardened surface in well-defined conditions of experimental circular track allowed estimating the parameters of Archard’s abrasive wear model, such as mild and severe wear coefficients and the transition point between these phases depending on wheel-rail friction coefficient.

For the wheels with hardened surfaces the following parameters values were set:
- for severe wear phase wear coefficient is $2.2 \times 10^{-6}$ g/N·m;
- for mild wear phase wear coefficient is $1.4 \times 10^{-6}$ g/N·m;
- switch point between mild and severe wear phases is at the power density value of 7 MW/m²;
- friction coefficient for wheel tread is 0.25;
- friction coefficient for wheel flange is 0.28.

In future researches clarified wear model parameters can be used for development of new freight car bogies, providing lower wear of wheel’s profiles, and for estimation of the influence of different factors on wheel wear in existing designs.

References