1.0 INTRODUCTION

Vehicles and track meet at a wheel/rail interface that sees large forces, and stresses much higher than the tensile strength of wheel and rail steels. The forces in this interface act positively to steer the vehicle. However, they also lead to rolling contact stresses that work negatively to cause wear and cracks, spalls, and shells produced by rolling contact fatigue. Vehicle steering forces depend on many factors including vehicle and bogie type, track curvature, the degree of wheel/rail lubrication, and the detailed transverse profile shapes of the wheels and rails. Inappropriate profile shapes can lead to vehicle instability and even derailment. Wheel/rail stresses also depend critically on the wheel and rail profiles and the way they interact.

North American railroads have about 48 million tons of rail installed in some 200,000 miles of track. Estimating the typical life of rail is difficult, but Table 1 gives approximate lives (in millions of gross tons or MGT) for rail in track varying from straight to 175-m radius. There are also about 12 million wheels in service in North America. Of these, about 1 million removed each year, generally for reasons other than wear such as spalls, shells, flat and out-of-round wheels, and built-up treads. A survey of approximately 6,950 wheel profiles from vehicles in North American service indicated that few wheels approach their theoretical maximum amount of allowable wear. The numbers demonstrate that wheels and rails are major railroad assets. A key element of improving railroad profitability and safety is improved management of wheel and rail profiles to reduce wear and rolling contact fatigue (extending asset life and reducing vehicle and track maintenance costs), and to improve vehicle stability.

This paper describes two projects Transportation Technology Center, Inc. (TTCI), a wholly-owned subsidiary of the Association of American Railroads (AAR), has undertaken as part of the AAR’s Strategic Research Program to improve the maintenance of wheel and rail profiles in North America. The paper will first describe work to define an economic-based removal criterion for wheels worn to a hollow tread condition, and then describe the development of procedures to assess the suitability of rail profiles for service.
Table 1. Estimated life of rail in North American track

<table>
<thead>
<tr>
<th>Curve radius</th>
<th>Estimated rail life, MGT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Straight</td>
<td>1460</td>
</tr>
<tr>
<td>1,750 m</td>
<td>1050</td>
</tr>
<tr>
<td>875 m</td>
<td>640</td>
</tr>
<tr>
<td>580 m</td>
<td>540</td>
</tr>
<tr>
<td>440 m</td>
<td>510</td>
</tr>
<tr>
<td>350 m</td>
<td>440</td>
</tr>
<tr>
<td>290 m</td>
<td>390</td>
</tr>
<tr>
<td>250 m</td>
<td>380</td>
</tr>
<tr>
<td>220 m</td>
<td>370</td>
</tr>
<tr>
<td>190 m</td>
<td>350</td>
</tr>
<tr>
<td>175 m</td>
<td>330</td>
</tr>
</tbody>
</table>

2.0 HOLLOW-WORN WHEELS

Wheels are said to be hollow when the tread center has worn such it is below the end-of-tread area, as shown in Figure 1 where the degree of hollowing is defined as the distance H. Mace et al. identified the deleterious effect of hollow wear on the steering performance of the standard three-piece freight bogie. The negative effect that hollow wheels have on steering increases lateral forces in curves and raises rolling resistance. Increased lateral forces accelerate damage to wood sleepers, and can lead to increased risk of derailment especially when the false flange at the end-of-tread area bears on the outer side of the low rail in curves. Higher rolling resistance translates directly to higher fuel costs and increased wear of the wheel and rail. The false flange also leads to rolling contact fatigue on the field side of the low rail in curves, and to damage at the wing rail-nose transfer area of crossings. This fatigue and damage typically needs repair by welding and/or grinding. The goals of this project on hollow-worn wheels were to determine whether there was an economic net benefit from removing hollow-worn wheels from service, and, if justified, to determine the most economic removal limit.
2.1 Methodology of Study
The methodology adopted to achieve the objectives is described:

1. A survey of wheel profiles on North American vehicles was undertaken to determine the distribution of hollow-worn wheels.

2. The potential benefits from removing hollow-worn wheels were identified, and dynamic modeling backed by track tests was undertaken to quantify the benefits for hollow wheel removal limits from 2 mm to 6 mm.

3. For each removal limit, the increased wheelset maintenance costs were calculated using typical and defined costs for wheelset maintenance operations in North America.

4. A cost/benefit sensitivity analysis was performed.

5. Finally the costs and benefits were compared to establish whether there was an economic case for removing hollow-worn wheels from service.

Table 2 shows the results of the wheel survey in which over 6,500 profiles were taken from a wide range of vehicles and were analyzed. While approximately 46 percent of the wheels measured showed virtually no hollow wear, about 2 percent of wheels showed more than 4 mm of hollow wear. Analysis also indicated that hollow wheels did not necessarily occur in pairs on the same axle. Thus, although 2 percent of wheels were more than 4 mm hollow, about 3.8 percent of axles had at least one wheel with 4 mm hollow wear or more. Very few wheels showed more than 6 mm of hollow wear. This is because the tread wear needed to achieve 6 mm hollow wear causes wheels to meet the removal criterion for a tall flange.
Table 2. Distribution of hollow wheels in North America

<table>
<thead>
<tr>
<th>Hollow wear</th>
<th>Percent of wheels</th>
<th>Percent of axles (worst wheel)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tapered to 0.25 mm</td>
<td>45.49</td>
<td>35.11</td>
</tr>
<tr>
<td>0.25 mm to 1 mm</td>
<td>21.37</td>
<td>22.17</td>
</tr>
<tr>
<td>1 mm to 2 mm</td>
<td>18.20</td>
<td>19.55</td>
</tr>
<tr>
<td>2 mm to 3 mm</td>
<td>8.69</td>
<td>11.85</td>
</tr>
<tr>
<td>3 mm to 4 mm</td>
<td>4.22</td>
<td>7.52</td>
</tr>
<tr>
<td>4 mm to 5 mm</td>
<td>1.64</td>
<td>3.81*</td>
</tr>
<tr>
<td>5 mm to 6 mm</td>
<td>0.36</td>
<td></td>
</tr>
<tr>
<td>More than 6 mm</td>
<td>0.03</td>
<td></td>
</tr>
</tbody>
</table>

* 3.81 percent of axles have at least one wheel more than 4 mm hollow

2.2 Technical Investigation

The perceived benefits from hollow wheel removal were as follows:

- An improvement in vehicle steering leading to reduced rolling resistance; hence, reduced fuel usage and lower rail wear. It was believed that overall wheel wear rates would also reduce with hollow wheel removal, but these potential benefits were not calculated.
- A reduction in gage-spreading lateral forces leading to reduced damage to wooden sleepers.
- A lowering of contact stresses on the field side of the low rail in curved track caused by the hollow wheel false flange, leading to a reduction in rolling contact fatigue and the grinding needed to remove cracks and spalls. Other work had established that the low rail in curves was a particular focus for rail grinding in North America.\(^{(3)}\)
- A reduction of impacts (and therefore damage) caused by the false flange at special track work. Damage, which needs to be repaired by extensive weld repairs, is particularly noticeable at the wing-rail/nose transfer points at crossing.
- A reduction in derailment risk. By increasing lateral force, and with the rail turning moment caused by the action of the false flange on the low-rail field side, hollow wheels increase the risk of rail rollover. However, means to calculate this benefit were not available.

Modeling using the dynamics program "NUCARS™" was undertaken to establish the effect of hollow worn wheels on wheel/rail interaction, principally rolling resistance and lateral force.\(^{(4)}\) This is a multi-body vehicle dynamics program that allows the detailed simulation of any rail vehicle running on virtually any track. It uses a full nonlinear creep theory for calculating wheel/rail forces, and wheel/rail
contact geometries are pre-calculated from files of measured and/or theoretical rail and wheel shapes. The basic model used a typical 119-tonne coal hopper equipped with standard three-piece bogies with constant contact side bearings and representation of all major suspension elements (including friction elements). Table 3 lists the major variables studied, and the common model outputs.

Small-angle axle misalignments were applied, since earlier work has shown that bogies on tangent track run with axles misaligned typically by 2 mrad.\(^5\) In the modeling, these misalignments had little effect on results from curved track. They were necessary, however, to give realistic rolling resistance predictions in straight track. (In the absence of misalignment, the model gave negligible rolling resistance in straight track.) Because these axle misalignments destroyed vehicle symmetry, modeling in curved track was undertaken using both left- and right-hand curves, and the final results were expressed as averages of these left- and right-hand runs. Typical results from the modeling exercise are shown in Figures 2 and 3, for the 0.45 wheel/rail friction case.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel/rail friction</td>
<td>Friction 1: 0.45</td>
<td>0.45 everywhere on both rails</td>
</tr>
<tr>
<td></td>
<td>Friction 2: 0.4/0.2</td>
<td>0.4 on rail top, 0.2 on gage face, both rails</td>
</tr>
<tr>
<td>Track curvature</td>
<td>Straight track to 175-m radius</td>
<td>Six track cases in total</td>
</tr>
<tr>
<td>Track roughness</td>
<td>Typical vertical and lateral roughness</td>
<td>To introduce small amplitude lateral and vertical dynamic response in the simulations</td>
</tr>
<tr>
<td>Wheel profiles</td>
<td>Tapered to 6-mm hollow in approximately 1-mm increments*</td>
<td>In any given simulation, all wheels in the vehicle were identical</td>
</tr>
<tr>
<td>Rail profiles</td>
<td>Typical straight and curved rail profiles</td>
<td>Two-point contact conditions were avoided</td>
</tr>
<tr>
<td>Center bowl friction</td>
<td>0.267</td>
<td></td>
</tr>
<tr>
<td>Axle misalignment</td>
<td>0, 1, 2 mrad</td>
<td>Imposed misalignment between bogie axles</td>
</tr>
<tr>
<td>Speed</td>
<td>48 km per hour</td>
<td>Constant for all simulations</td>
</tr>
<tr>
<td>Common outputs</td>
<td>1 Vertical, lateral, and longitudinal forces at each wheel</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2 Angle-of-attack at each wheelset</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 Rolling resistance of vehicle</td>
<td></td>
</tr>
</tbody>
</table>

* The wheel profiles had simulated worn flanges
Figure 2 shows the effect of track curvature on rolling resistance for the hollow-wear cases examined. (Note that hollow wear is expressed in mm, according to the definition of Figure 1. The only exception is the 0 mm case where this is used as shorthand to describe the nearly new tapered wheel condition.) Figure 3 shows how the gage-spreading force changes for the same track and hollow-wear conditions. (Gage-spreading force is defined as the sum of the outward lateral force on each rail, expressed as the average for all four axles.)

Principal results from the modeling exercise are:

- As expected, rolling resistance increases substantially with track curvature, for wheels in all conditions.
- Hollow wheels also lead to much higher levels of rolling resistance, even in straight track. Closer inspection of the modeling output indicated that this increased rolling resistance was associated with lozenging of the bogies, causing all four wheelsets to run with higher angle-of-attack.
- The same effect of track curvature and hollow wear was seen for gage spreading force. For tapered and low-hollow wheels the gage spreading force arose mainly from the lead axles in each bogie. The modeling cases with more-hollow wheels consistently showed relatively high lateral forces also for the trail axles. (For very hollow wheels, gage spreading force saturated approximately at the product of axle load and friction.)
- Rolling resistance and gage spreading force were little affected by hollow wear up to about 2 mm. Hollow wear above 2 mm led to significant increases in resistance and force. The cause of this cut-off value is not known with certainty, but it equates approximately with the onset of negative equivalent conicity. That is, for the wheel and rail profiles used, hollow wear up to approximately 2 mm gave positive conicity, while hollow wear of approximately 3 mm and above gave negative conicity with associated lack of steering.
Track tests were undertaken to validate the dynamic model used for predictions. In two sets of experiments, a five-vehicle test train was assembled and run under dry rail conditions round a loop with variable curvature at the U.S. Federal Railroad Administration’s Transportation Technology Center in Pueblo, Colorado. In the first experiment, the vehicles (all coal hoppers with standard three-piece bogies) were equipped first with tapered wheels (0 mm) and then with 2-mm hollow wheels. In the second experiment, the same vehicles were again equipped with tapered wheels and then with 3.7-mm hollow wheels. For each wheel condition an instrumented drawbar was used to measure overall rolling resistance.
in straight track and in curves of 581-, 437-, and 233-meter radius, from which wheel/rail rolling resistance was estimated by subtracting estimated resistance due to bearings and aerodynamic losses. (Low speeds were used in these tests to minimize aerodynamic resistance. In the same straight track and curves, strain gages attached to the rails were used to measure lateral forces, and optical measuring systems were used to measure wheelset angle-of-attack.

Additional work was undertaken to model the vehicle/test conditions, thereby allowing direct comparison between model predictions and test results. Good agreement was seen between predicted and measured rolling resistance, as shown in Figure 4, which shows values for two wheel cases for all four track test locations. Reasonable agreement was also seen between predicted and measured values for lateral force and angle-of-attack.

The modeling showed a correlation between total vehicle wheelset angle-of-attack (the sum of angle-of-attack of all four wheelsets, irrespective of sign) and rolling resistance. This is seen in Figure 5, for the two wheel/rail friction cases, where, for all track curvatures and hollow wheel cases, rolling resistance is seen to increase linearly with total angle-of-attack. Also shown in Figure 5 are results from the track tests run under a wheel/rail friction condition of approximately 0.5. It can be seen that the test results again agree reasonably well with predictions.

As a final test of the ability of the model to predict rolling resistance, estimated bearing and aerodynamic resistance was added to the predicted wheel/rail rolling resistance to obtain total wagon resistance. For the tapered (0 mm), 1 mm and 1.9 mm wheels this gave:

\[
\text{Total Wagon Resistance} = 1.7 \text{ lb/ton} + 0.98 \text{ lb/ton/degree of curvature} \quad (1)
\]

This is close to the commonly assumed formula of 1.8 lb/ton + 0.8 lb/ton/degree used by North American railroads to calculate locomotive traction requirements.

These model/test results indicate that the NUCARS™ dynamic model is able to predict wheel/rail interaction well.
Comparison of Rolling Resistance Model Results (Solid Points) and Test Results (Open Points) for Two Wheel Cases

Effect of Total Angle-of-Attack on Rolling Resistance, Comparison of Model Results (at Two Wheel/rail Friction Conditions) and Test Results (Solid Points)

In addition to predictions of resistance, force, and angle-of-attack, a pre-processor of the NUCARS™ program was used to calculate the wheel/rail contact stresses produced by all the wheels (tapered to 6.2 mm hollow) when matched against the model rails. These stresses were calculated for all possible lateral positions of the wheelset, and used a Hertzian calculation with all profiles approximated by a series of circular arcs. These predicted contact stresses were used to estimate the reduction in rail rolling fatigue achievable through the removal of hollow wheels.
2.3 Economic Modeling
The economic analysis models the incremental costs and benefits associated with hollow-wheel removal policies of 2, 3, 4, 5, and 6 mm. For each removal policy, the model calculates five benefits (fuel, rail wear, sleeper life, rail grinding, special track work welding) and the associated increased wheelset maintenance costs. The model considers traffic on two typical North American eastern and western coal-haul routes (with annual route tonnage of 60 MGT), using a standard train with 100 freight vehicles. The train is modeled both loaded (119-tonne) and empty. The following sections briefly describe components of the model.

2.3.1 Fuel Cost
Fuel is expended to overcome wheel/rail rolling resistance, and since hollow wheels increase rolling resistance, removing them from service gives savings in fuel. The rolling resistances of individual wheel profiles found from the dynamic modeling are used to establish the fuel consumption for each particular wheel profile as it traverses the model routes. The model first calculates the fuel needed to haul the train with the current wheel profile distribution found from the wheel survey. For each hollow wheel removal criterion (for example, 3 mm), a new wheel distribution is estimated and the new fuel use figure calculated. The resulting reduction in fuel consumption is translated to benefits using current average fuel costs.

2.3.2 Rail Wear
Rail wear is proportional to rolling resistance, and is reduced when hollow wheels are removed from service. The base-case calculation assumes current average North American rail lives (Table 1) and the survey wheel profile distribution. As with the fuel model, the reduction in rail wear is then estimated for each hollow wheel removal criterion. The incremental rail life improvements are translated to benefits based on the capitalized cost of rail installation.

2.3.3 Wooden Sleeper Life
Removing hollow wheels leads to reduced lateral forces, which in turn are expected to lead to longer wooden sleeper lives. As with fuel and rail wear, the model first calculates the life of sleepers for the train with the survey distribution of hollow wheels, and then calculates the increased life using the appropriate wheel distribution for each removal policy. These calculated lives are benchmarked against known lives for sleepers in wet and dry track locations. Increased lives are translated to benefits using known railroad costs.
2.3.4 Rail Grinding
Hollow wheels lead to higher wheel/rail contact stress, giving increased rolling contact fatigue damage that needs to be removed by grinding. The model assumes that 20 percent of the annual cost of rail grinding in North America can be saved by the removal of all hollow wheels, and calculates the saving from a given removal policy by considering the magnitude of the contact stresses associated with hollow wheels and the lateral wheelset shifts over which those stresses are applied to the low rail.

2.3.5 Special Track Work Welding
It is known that hollow wheels are a cause of much of the damage that occurs to special track work, principally in the area where the wheel transfers from the wing-rail to the nose and vice-versa. This damage can be repaired by welding, but also shortens component life. It was found impossible to determine the costs of damage using stress and impact calculations. Consequently, presentations on the types of damage caused by hollow wheels were made to railroad welding staff who were then asked to estimate the percentage of their time spent repairing such damage. These estimates were converted to benefits using standard railroad costs for welding.

2.3.6 Wheelset Costs
The wheelset cost model was developed to estimate the number of wheels requiring maintenance or replacement throughout a wheel life cycle, for each of the removal policies. Three wheel types were considered: 1-wear wheels, 2-wear wheels, and multi-wear wheels. Maintenance costs were estimated using data from the AAR’s Field and Office Manuals of Interchange Rules and known railroad costs. It was difficult to calculate the increased maintenance costs associated with a given removal policy, in large part because wheels tend to be taken out for causes other than wear. Many of these causes, such as wheelflats and shelled wheels, tend to occur on a random basis. It was eventually decided to undertake the analysis by assuming that, for a given hollow-wear limit, all wheels in the next-lowest hollow wear band reach the limit. For example, for the 3-mm removal case, the model assumes that all 2-mm hollow wheels eventually wear to the 3-mm limit and are replaced for hollow wear. This is conservative, since in practice many 2 mm hollow wheels would likely be removed from service for other reasons before reaching 4 mm hollow. The wheelset costs calculated using this model are termed Case 1 costs.

The Case 1 model assumes that one wheelset is taken out for every two hollow wheels removed. This is unlikely, since the wheel survey showed that hollow wheels rarely occur in pairs (Table 2). Therefore augmented wheelset costs (Case 2 costs) were calculated by increasing the Case 1 costs by 50 percent (equivalent to removing three wheelsets for every four hollow wheels removed). It is believed these Case 2 costs are highly conservative.
2.4 Economic Results

The net benefit results for the Case 1 cost model are shown in Table 4. This shows positive net benefits for all removal policies, and for all wheel types, with the highest net benefit found for the 3-mm removal policy. Lower net benefits are seen for the 2 mm policy, principally because of the larger numbers of these wheels and the reduced damage/costs they produce. Very low net benefits are seen for the very hollow wheels, mainly because there are so few of them. When Case 2 costs are used the maximum net benefits are reduced, but are still positive for all policies except the 2-mm policy, when the net benefits become highly negative.

An extensive sensitivity study was undertaken to test the net benefit results. As expected, increasing fuel costs increased net benefits substantially, while any changes that delayed benefits worsened net benefits. A particular finding of interest is that net benefits are significantly reduced (though still positive) for vehicles equipped with improved suspension bogies that are resistant to lozenging. In these vehicles hollow wheels produce lower increases in rolling resistance. Therefore the case for removing hollow wheels from vehicles with such bogies is less strong.

The results indicate that, for typical freight vehicles, there is an economic case for removing hollow wheels at the level of 3 to 4 mm and above for North American conditions. It is possible, though not yet proven, that removal of hollow wheels at this limit would also lead to a reduction in derailment risk. This figure of 3-4 mm compares with a removal value of 2 mm adopted recently by Cartier Railway Company in Canada and Spoornet in South Africa.\(^6,^7\) For policy reasons, these two railroads have much lower bearing-related costs than most of the North American railroads for wheelset re-truing.

**Table 4. Summary of cost benefit results for Case 1 costs**

<table>
<thead>
<tr>
<th>Assumed hollow wear removal limit</th>
<th>2 mm</th>
<th>3 mm</th>
<th>4 mm</th>
<th>5 mm</th>
<th>6 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-wear wheels</td>
<td>$20.14 m</td>
<td>$88.7 m</td>
<td>$44.73 m</td>
<td>$12.29 m</td>
<td>$1.02 m</td>
</tr>
<tr>
<td>2-wear wheels</td>
<td>$41.34 m</td>
<td>$103.72 m</td>
<td>$51.3 m</td>
<td>$14.93 m</td>
<td>$1.49 m</td>
</tr>
<tr>
<td>Multi-wear wheels</td>
<td>$57.35 m</td>
<td>$109.13 m</td>
<td>$55.97 m</td>
<td>$16.10 m</td>
<td>$1.70 m</td>
</tr>
</tbody>
</table>
3.0 RAIL PROFILE ASSESSMENT

The objective of this project is to lengthen wheel and rail life by reducing wheel/rail contact stresses through improved control of rail profiles. As will be seen, the software developed for this also potentially allows other improvements in wheel/rail interaction.

Manual wheel and rail profile measurement systems have been available for some years, as have automatic rail profile systems. Automatic wheel profiling systems have emerged recently, and are currently being used or tested in North America. Profiles traditionally have been used mainly to get an indication of the extent of wear. This is useful, but profiles hold much more information. As part of the AAR’s Strategic Research Program, TTCI is developing software designed to extract and analyze more information from measured rail profiles to reduce the stress state between the wheel and rail extending life and reducing costs.

3.1 Methodology

The methodology adopted is to match measured rails (in pairs) against a range of wheels typical of the vehicles passing over, calculate the wheel/rail interaction, and assess the calculated interaction against the preferred interaction. This has required the following tasks:

1. A range of wheel and wheelset profiles typical of those found in North America was defined using the wheel survey described in Section 2.0. Of the wheels, 49 of them could be used to describe about 90 percent of the survey wheels judged to be no more than 3 mm hollow. Forty-nine typical wheelsets were found by adding the mating wheels to the 49 typical wheels.

2. Four wheel/rail interaction parameters were defined (contact stress, contact angle, effective conicity, and degree of two-point contact). Work is continuing to determine preferred limits for each interaction parameter.

3. The wheel database holds wheel profiles idealized by a series of circular arcs. Software has been designed also to automatically represent the input measured rail profiles by a series of circular arcs. This method of profile representation was adopted to speed the calculation of contact stress using Hertzian analysis. The software enables the user to compare the measured profiles with the circular-arc profiles.

4. Software has been designed to match the wheels and wheelsets against a measured pair of rails for all possible positions of lateral wheelset shift. The software places the left wheel in flange contact with the wheel and moves the wheelset in incremental steps until the right wheel is in flange contact. At each step the software calculates the wheel/rail interaction. For calculation of contact stress, contact angle, and degree of two-point contact the software uses the 49 typical
wheels with identical wheels at each end of the axle. For calculation of effective conicity the software uses the 49 typical wheelsets. At the end of processing, the software assesses the interaction against the preferred interaction. Both graphical and numerical data output is available.

While the software was intended initially for the North American environment, it has been designed for general use. As such it can accept any database of wheels and wheelsets, and allows varying track gage. A goal has been to make the software sufficiently fast such that eventually rail profiles can be analyzed in real time. Currently, to input and curve-fit a pair of rail profiles, calculate all the interaction parameters between the rails and 49 wheelsets, and produce all outputs with preferred value comparisons takes about 15 seconds on a high-powered desktop personal computer.

3.2 Wheel/Rail Interaction Parameters

Four interaction parameters have been chosen, although the software has flexibility for more to be added. User-defined preferred values can be assigned to these parameters. Initial work has focused on contact stress and contact angle, and setting the interim preferred values for these parameters. Work continues to set preferred values on the degree of two-point contact and effective conicity.

3.2.1 Contact Stress ($P_0$)

This is the maximum compressive stress normal to and at the center of a given wheel/rail contact. Past work has shown that wear and rolling contact fatigue increase as contact stress rises.\(^8,9\) For Hertzian conditions the calculated contact is circular or elliptical and methods to calculate contact stress are readily available.

As an example of output, Figure 6 shows all the possible contact points between one of the wheels and a measured rail. Some of these contacts are less likely to occur in reality. In curved track for example, contact of the wheel flange on the low rail is unlikely even for the trailing wheelset in a bogie. Figure 6 also shows some contacts that can only occur when the wheel is climbing the rail gage face. If required the software can filter these contacts out since high contact stresses in these situations are not relevant.
All Possible Contact Points Between Wheel and Rail

Figure 7 shows all the possible contact stresses caused by the 49 wheelsets when matched against a measured high rail in curved track. It also shows the critical maximum contact stresses that, ideally, should not be exceeded. Stresses above these critical values are likely to cause rolling contact fatigue damage, while lower stresses are likely to be relatively harmless. The position of contact is given in millimeters with respect to the track centerline. In this example the high-rail gage face is at about -720 mm.

Example of Contact Stresses from 49 wheels on the High Rail in Curved Track

Calculation of these critical stress values is relatively straightforward, but many assumptions have to be made. The model used assumes, as a first approximation, that the critical stresses ($\sigma_c$) are equal to the shear yield stress ($\sigma_{SY}$) of the rail steel, multiplied by a number of factors, as shown in Equation 2.
\[ \sigma_C = \sigma_{sy} \cdot F_{WH} \cdot F_{SR} \cdot F_{SH} \]  

(2)

\( F_{WH} \) is a factor to account for work hardening of the near surface layers of the rail in service. Measurements indicate that this can typically be in the range of 1.5 to 2.0.\(^{(10)}\)

\( F_{SR} \) is a factor to account for the effect of strain rate on yield stress. Strain rates in wheel/rail contact are likely to be high, since a point on the rail moves through a typical contact in less than 1 ms. The rail strain rate factor is estimated as about 1.3.\(^{(11)}\)

\( F_{SH} \) is the shakedown factor. Under repeated rolling contact the material at and near the rail surface develops protective residual stresses that act to inhibit further plastic flow. The theory of shakedown applied to rail/wheel contact is reasonably well developed, and shakedown factors are available for circular contacts and elliptical contacts not over-elongated in the direction of rolling.\(^{(12)}\) The shakedown factor depends on the traction coefficient (the ratio of tangential force in the contact divided by the normal force), which itself depends on track curvature, vehicle characteristics, and whether the wheels are powered. The traction coefficient cannot exceed the prevailing level of wheel/rail friction; therefore, it is affected by lubrication.

A particular problem is the definition of shear yield stress, which can take on a range of values (for a given steel) depending on the method of measurement. The current project uses relatively low yield stress values determined by the author from unpublished work on the general fatigue performance of pearlitic steels. By coincidence, the use of these low yield stress values and the factors in Equation 2 give a final effective shear yield stress reasonably close to the plain yield stress divided by 2 (Tresca criterion) or \( \sqrt{3} \) (Von Mises criterion).

The critical stress values increase with rail hardness. They also increase greatly with lubrication. In Figure 7 it is assumed that there is good lubrication on the high-rail gage face that spreads onto the gage corner. This is why the high-rail critical values show a change from the gage face to the rail top. The figure illustrates a number of points about wheel/rail stresses and the analysis method.

- Stresses at the gage face are always high, and practically uncontrollable so long as gage face contact exists. High wear rates will always occur unless good lubrication is applied.
- For the particular high rail used, high stresses are predicted on the gage corner (at about -740 mm). These stresses are likely sufficient to cause spalls and shells.
• High stresses are also predicted at the field side of the high rail, caused by just a small number of wheels. These stresses may be unlikely to occur in practice, requiring a wheelset shift giving flange contact on the low rail.

3.2.2 Contact Angle (_)
The contact angle is the angle that the contact plane at the wheel/rail contact point makes with the horizontal to the track. The risk of flange-climb derailment increases as this angle is decreased. Based on previous work by TTCI, to reduce derailment risk the maximum contact angle at flange contact should preferably be greater than 68 degrees.

3.2.3 Degree of Two-Point Contact
During rail grinding to remove rolling contact fatigue damage, a past practice on North American railroads has been to remove metal especially from the high-rail gage corner. This produces two-point contact conditions, where the wheel contacts the high rail on the gage face and rail top. This is intended to reduce the incidence of shell defects that can lead to broken rails. When properly applied, such grinding can be effective in reducing defects (especially in softer rails), but it impairs vehicle steering leading to significantly increased wear and rolling resistance. This practice is not now generally favored, and recent work at test sites indicates that premium head-hardened rails are much less susceptible to breaks from shell defects. It was decided, however, that the software could usefully examine whether unwanted two-point contact conditions were present on the high rail in curves, as an aide to remedial grinding to reduce rolling resistance.

Early work in the hollow wheel project indicated that rolling resistance (hence, wear and fuel use) rises approximately linearly with the vertical distance between the two contact points. This distance would therefore seem to be a reasonable way of defining the degree of two-point contact. No preferred value for the maximum allowable degree of two-point contact, which influences railroad economics rather than safety, has yet been set.

3.2.4 Effective Conicity (C<sub>e</sub>)
Conicity is a measure of the combined geometry of a wheelset and rail pair. It is defined as one half the rate of change of wheel rolling radius difference with lateral wheelset shift, for shifts that do not give flange contact. For a coned wheelset on new rails the conicity is normally equal to the cone angle of the wheel. For a worn wheelset on new or worn rails the conicity generally varies with lateral wheelset shift, and the effective conicity is taken to be some average of the range of calculated conicity. The effective conicity affects the stability of the vehicle. Too low a value impairs the ability of the vehicle to steer
adequately in curves, since the wheelset cannot generate sufficient rolling radius difference to generate longitudinal steering forces. Too high a value can lead to hunting on straight track. For an idealized case of a single wheelset restrained by springs in a frame it has been shown that the critical speed for hunting is inversely proportional to the square root of the effective conicity.\(^{13}\)

Effective conicity is an automatic output of the software used to calculate wheel/rail contact stress (Section 3.2.1). Hence, work is in progress to see whether preferred ranges of effective conicity can be defined for specific vehicle/bogie combinations. The goal is to set limits on effective conicity, and thus on wheel and rail profiles, such that the speed of hunting can be reasonably guaranteed to be above the required operating vehicle speed.

Early work has concentrated on modeling the dynamic performance of an empty coal hopper equipped with standard three-piece bogies (containing many friction elements) and roller side-bearings. The model has examined the effect of effective conicity on hunting speed for a range of coned wheelsets running on new 136RE profile rails installed in straight track with vertical and lateral roughness typical of heavy haul freight track. (For these wheel/rail profile combinations the effective conicity is constant with wheelset shift, and varies from 0.01 to 0.2.) The model slowly increases vehicle speed, and the onset of hunting is assessed by examining outputs including axle and bolster lateral displacement, and lead and trail body lateral accelerations. For these ideal wheel/rail combinations the speed at onset of hunting is found to vary approximately as the inverse one-quarter power of effective conicity, as shown in Figure 8. Current work is examining the hunting characteristics of the same vehicle equipped with worn wheelsets that give variable conicity with wheelset shift.

![Image of graph showing the effect of effective conicity on hunting speed for a freight vehicle equipped with coned wheelsets.](image)

**Effect of Effective Conicity on Hunting Speed for a Freight Vehicle Equipped with Coned Wheelsets**
3.3 Testing of Contact Stress Model and Example of Use

By matching multiple wheelsets against a single pair of measured rail, the software is potentially capable of allowing rapid assessment of the suitability of the measured rail profiles in terms of contact stress for the wheels passing over. This would allow decisions to be made about the suitability of the profiles used for rail grinding, or the need for remedial grinding. It also enables those wheels that give damaging stresses to be identified. (In future work it is intended to “reverse” the software, and match databases of rail profiles against measure wheel pairs to identify damaging wheels.)

Before the software can be used, however, tests are needed to ensure that the information given is relevant for use in maintenance decisions. As a first step, the software has been applied to rail profiles measured over several years at a range of rail grinding sites in North America. Long-term observation and measurement at these sites has documented the profiles (before and after grinding), the degree of lubrication applied, and the surface condition (in terms of rolling contact fatigue) of the rails. The software has been used to assess the wheel/rail stress condition, which can be related to changes in surface condition.

Figure 9 shows results from one of the sites, a 270-m radius curve with head-hardened rails made to clean steel specifications. The x-axis gives the accumulated tonnage (first visit at 12 MGT, with further visits to 250 MGT), and the y-axis gives the percentage of wheels passing over that are judged by the software to apply damaging stresses; i.e., stresses above the solid line in Figure 7. This site had little lubrication until approximately 154 MGT (shown by the solid line in Figure 9), and was ground for the first time at 164 MGT, and at intervals thereafter. Before about 154-164 MGT the high rail showed significant rolling contact fatigue damage (cracks, flakes, spalls and shells) while after 164 MGT fatigue damage has been minimal. This damage history correlates with the results shown in Figure 9, in that before 154-164 MGT it is predicted that most of the wheels passing over were applying damaging stresses. In contrast, the application of lubrication and grinding caused the percentage of damaging wheels to fall significantly. Grinding helps reduce stress by giving more conformal wheel/rail contact. The effect of lubrication is subtler, in that it reduces tangential forces in the contact and therefore, by its effect on the shakedown factor, raises the critical stress for damage (that is, it raises the solid line in Figure 7).
Example Application of the Wheel/Rail Profile Software

The results of Figure 9 are encouraging, but further testing has demonstrated that the outputs from the software are more complex than first thought. First, the use of a single parameter, such as the percentage of damaging wheels, can lead to results that are highly sensitive to the choice of critical stress. Small changes in calculated stress can lead to large changes in the percentage of damaging wheels.

Second, and of more concern, is that the calculated contact stress is sensitive to the curve-fitting procedure used. This has been observed elsewhere in work that showed that apparently minor changes in smoothing technique could lead to very large changes (approximately, factor of two) in calculated contact stress. Such a large factor has not been seen in the present work, possibly because of the software employs a number of techniques to filter out abnormally high stresses. However, this problem with contact stress calculation is forcing further work to study improved ways of generating and interpreting contact stress data.

4.0 SUMMARY

As speeds and axle loads increase, railways are recognizing the importance of optimizing the wheel/rail interface to decrease maintenance costs and fuel use and improve vehicle steering. This optimization involves gaining better control of wheel and rail profiles. An example of the increased interest in wheel/rail interaction is the recent publication of wheel/rail interface guidelines for heavy haul railways.

This paper has described two North American projects to improve the wheel/rail interface. The first project is seeking to determine whether there are overall net benefits from removing hollow-worn wheels.
from service. Economic calculations using benefits calculated principally from dynamic simulations indicate that the removal of hollow wheels at 3 to 4 mm and above is justified. The main benefits arise from reduced rail wear and fuel use from decreased rolling resistance, and reduced welding of damage at special track work. The second project is an attempt to develop software that can be used to rapidly identify rails that have profiles unsuitable for the wheels passing over. The software assesses wheel/rail interaction in terms of contact stress, contact angle, degree of two-point contact, and effective conicity. Good progress has been made, but much needs to be done. In particular, the software has confirmed that calculation of contact stress is still a major problem.
References


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