ALLOWABLE WHEEL LOADS, CRACK SIZES AND INSPECTION INTERVALS TO PREVENT RAIL BREAKS

Anders Ekberg, Elena Kabo, Jens Nielsen
Chalmers University of Technology, SWEDEN

SUMMARY
Cost-efficient and reliable heavy haul operation requires a minimum of operational disturbances. To this end, the current study focuses on wheel load management with the aim to establish wheel load monitoring and mitigation actions (in terms of limits on allowable wheel defects) that minimize traffic disruptions.

As a first step, the relation between wheel impact load magnitudes (resulting from the out-of-round wheels) and critical rail crack sizes that would result in rail breaks is established. Variations in parameters such as track stiffness, rail temperature, impact load characteristics and hanging sleepers etc. are investigated and a “bad case scenario” that implies severe, but realistic operational conditions is established. In this manner allowable wheel impact load magnitudes can be linked to pertinent critical crack sizes that must be identified during inspections. An interesting finding is that thermal stresses have such a major effect that a seasonal variation in allowable wheel load limit seems justified.

Predicted critical crack sizes (presuming allowed wheel load magnitudes) are then contrasted to critical crack sizes at operational rail breaks. A large scatter in operational critical crack sizes is found and reasons for this fact are discussed. Finally operational aspects of implementing the suggested limit values are discussed.

INTRODUCTION

This study focuses on fracture of rails setting out from pre-existing long cracks. In contrast to shorter cracks, which mainly grow under the influence of the contact stress field, the long cracks mainly grow under the influence of the global rail bending induced by each passing wheel. In addition to the bending stress there is, for all-welded rails, a major influence of the rail temperature. This thermal stress is uniformly distributed over the rail cross-section and can be presumed as constant during a load cycle (basically corresponding to one wheel passage).

The aim of the study is to find a scientific basis for regulations regarding allowed wheel defects. These defects generate wheel–rail impact loads that in severe cases may promote fracture from pre-existing rail cracks. Previously, wheel removal criteria have mainly been related to the size (length) of a wheel flat. This is not an optimal situation since the correlation between the size of a wheel flat and the resulting bending moment of the rail is fairly week [1]. Further, it may be both difficult and dangerous to locate and measure the length of a wheel flat. This is not the least true for so-called rolling contact fatigue (RCF) clusters, which may cause very high impact loads [2].

In this study, the focus is therefore on relating known (measured) wheel–rail impact load magnitudes to the risk of rail breaks. Since fracture relates to long cracks driven by rail bending, the first step in such an approach is to relate (measured) wheel–rail impact loads to the induced rail bending moments. Such an analysis was carried out in a previous study, [3], which main results are briefly recapitulated. Then the influence of the rail bending moment on the risk of fracture is assessed through a fracture mechanics analysis. Finally the results will be scrutinised, a “bad case scenario” defined, and limit magnitudes for operations proposed. The paper concludes with a discussion on some operational issues related to implementation of the proposed limit magnitude framework. This discussion also includes a brief overview of some topics not captured by the current analysis of the influence of wheel–rail impact loads. In particular, potential approaches in addressing these topics are given.

It is believed that the current study progresses the state-of-the-art by providing a solid computational framework by which consequences of suggested limit magnitudes of wheel–rail contact forces can be assessed. This allows for harmonization of operational limits, and also for structured discussions leading up to the establishment (and potential revision) of such limits.
RAIL BREAKS – ANALYSIS FRAMEWORK

As mentioned, limits for allowable wheel loads are evaluated based on their influence on the risk of causing transverse fracture in 60E1 (UIC60) rails. This relates to the existence of long (in a fracture mechanics context) cracks. These cracks are presumed to have deviated into a transversal growth. It is further presumed that rail bending is the dominating loading mode (i.e. that head cracks have grown out of the contact zone) and that linear elastic fracture mechanics is valid.

Loading and Stresses

Established relations between impact load magnitudes and pertinent rail bending moments [3] are employed. These employ a worst-case (within reasonable operational limits) time history of the impact load, and have been derived for varying track stiffness magnitudes and vehicle types. Presuming Euler–Bernoulli beam theory to be valid, a rail bending moment $M_f$ corresponds to a maximum normal stress $\sigma_n$ in the rail of magnitude

$$\sigma_n = M_f h / I_y$$  \hspace{1cm} (1)

where $M_f$ is the bending moment with the sign convention that a positive bending moment generates a tensile stress in the rail foot. Further, $I_y$ is the corresponding cross-sectional moment of inertia and $h$ is $h_f$ or $-h_h$ as defined in Figure 1 for foot and head cracks, respectively.

Thermal loading (presuming a continuously welded rail) is incorporated as an additional (uniform normal) stress

$$\sigma_t = E\alpha \Delta T$$  \hspace{1cm} (2)

where $E = 210$ [GPa] is the elasticity modulus, $\alpha = 11.5 \times 10^{-6}$ [°C⁻¹] the thermal expansion coefficient and $\Delta T = T_0 - T$ where $T_0$ is the stress free temperature of the rail and $T$ the current temperature. Residual stresses are not included, which basically is a conservative assumption.

Magnitude and Influence of Wheel Impact Loads

Three vehicle types are considered

- A heavy haul operations at 60 km/h (axle load 30 tonnes)
- B freight operations at 100 km/h (25 tonnes)
- C passenger traffic at 200 km/h (21 tonnes).

Key data of the investigated vehicles are summarised in Table 1.

Table 1 Key data of investigated vehicles

<table>
<thead>
<tr>
<th>Train type</th>
<th>Axle load [tonnes]</th>
<th>Speed [km/h]</th>
<th>Axle distance in bogies [m]</th>
<th>Axle distance between bogies in the car [m]</th>
<th>Axle distance between wagons [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>30.0</td>
<td>60</td>
<td>1.78</td>
<td>4.4</td>
<td>1.8</td>
</tr>
<tr>
<td>B</td>
<td>25.0</td>
<td>100</td>
<td>1.80</td>
<td>7.0</td>
<td>3.2</td>
</tr>
<tr>
<td>C</td>
<td>21.4</td>
<td>200</td>
<td>2.50</td>
<td>17.5</td>
<td>6.4</td>
</tr>
</tbody>
</table>

Four different ballast stiffness magnitudes (per half sleeper), $k_b = 5, 10, 30$ and 100 MN/m have been accounted for. This ranges from very soft to very stiff ballast. For each operational scenario, rail bending moments have been derived for the case of nominal loads and for the case of a “worst case” wheel flat (in terms of time history of the impacting load, impact position etc). Details are presented in reference [3].

Influence of Hanging Sleepers

The influence of hanging sleepers on bending moments in the rail was also assessed in [3]. The results are compiled in Figure 2 where bending moments with no hanging sleeper(s), $M_{ha}$, are compared to bending moments in the presence of hanging sleepers, $M_{ha}$. In the study up to six hanging sleeper(s) (with a 2 mm gap to the underlying ballast) have been considered. The worst-case configuration (in terms of the location(s) of hanging sleeper(s) has then been employed. Full details of the analyses are presented in [3].

The relative influence of hanging sleepers is evaluated under the consideration that (long) rail cracks only grow in tension. Thus, the relative increase is here defined for foot cracks as

$$\max_i \left\{ M_{ha}(i) \right\} / \max_i \left\{ M(i) \right\} - 1$$  \hspace{1cm} (3)

and for head cracks as

Figure 1: Studied crack geometries. For nominal 60E1 profiles $I_y = 30.55 \times 10^{-6}$ m⁴, $h_h = 0.091$ m, $h_f = 0.081$ m.
For a head check factor is considered for the purpose of the current study, tension further as quasi as intensity factors are derived from these crack types are given in originating from a headcheck. Two crack locations are considered: An edge crack in the rail foot and a head crack for example originating from a headcheck. The geometry of these crack types are given in Figure 1. From the evaluated stresses, mode I stress intensity factors are derived from

\[ K_1(a,b,t) = f(a,b) \cdot \sigma(t) \sqrt{a} \]  

(5)

Here \( a \) is the crack size as defined in Figure 1 and \( \sigma(t) \) is the (time dependent) nominal stress taken as \( \sigma_n(t) \) according to Equation (1) for bending and as \( \sigma_t \) according to Equation (2) for the thermal loading. Note that the thermal stress is considered as quasi-static (as noted above).

Further \( f(a,b) \) is a geometry factor that for the rail foot crack can be approximated from the standard case of an edge crack in a plate under uni-axial tension, [4], as:

\[ f(a,b) = \frac{2b}{\pi a} \cos \left( \frac{\pi a}{2b} \right) \]  

(6)

\[ \left[ 0.752 + 2.02 \left( \frac{a}{b} \right) + 0.37 \left( 1 - \sin \left( \frac{\pi a}{2b} \right) \right)^3 \right] \]

For the purpose of the current study, this geometry factor is considered as a sufficient approximation for both bending and pure tension.

For a head check crack in a 60E1 rail, the geometry factor for bending and tension can be approximated, [5], as

\[ f_h(a,b) = 0.7 - 0.97 \left( \frac{a}{b_h} \right) + 2.6 \left( \frac{a}{b_h} \right)^2 \]  

(7)

\[ f_t(a,b) = 0.72 - 0.16 \left( \frac{a}{b_h} \right) + 1.4 \left( \frac{a}{b_h} \right)^2 \]  

(8)

The fracture criterion for a long crack can in general be expressed as

\[ \max \left\{ K_1(t) \right\} \geq K_{ic} \]  

(9)

Here \( K_{ic} \) is the fracture toughness of the rail material taken as \( K_{ic} = 40 \, \text{MPa}\sqrt{\text{m}} \) in the current study.

For the current case of combined bending and thermal loading, a fracture criterion can be expressed as

\[ \max \left\{ K_{ib}(t) + K_{tt}(t) \right\} \geq K_{ic} \]  

(10)

Since the thermal loading is presumed to be constant during a wheel passage, the fracture criterion can be reformulated as

\[ \max \left\{ K_{ib}(t) \right\} \geq K_{ic} - K_{tt} \]  

(11)

Magnitudes of \( \max(K_{ib}) \), corresponding to various impact load magnitudes, are evaluated for the studied operational configurations. Rail head crack sizes of \( a_h = 25, 30, 35 \) and 40 mm, and rail foot cracks of \( a_f = 5, 10, 15 \) and 20 mm (with \( a_h \) and \( a_f \) as defined in Figure 1) have been considered in the full analysis.

\[ K_{ib}(t) \] for a 5 mm foot crack in a 60E1 rail due to bending induced by impact loads of varying magnitudes. Influence of hanging sleepers is accounted for. Vehicle type: A—black, B—blue, C—red) and ballast stiffness by line type kb: 5—dotted, 10—dashed-dotted, 30—dashed, 100—solid). Fracture toughness reduced by thermal stresses is indicated by solid horizontal lines as \( \Delta T \) [°C]: 0 — red, 20°C — blue and 40°C — green. The thick black line indicates a “bad case” scenario.
Examples of results are presented in Figure 3 and Figure 4. Here the horizontal lines indicate the fracture toughness reduced by the stress intensity caused by the thermal stress (considered constant for one load cycle). The inclined lines indicate how the stress intensity increases with impact load magnitude for the different scenarios. It is seen that there is a more pronounced effect of the impact load magnitude on stress intensities for rail foot cracks (and consequently on tensile stresses in the rail foot) than on stress intensities for headcheck cracks.

![Figure 4: Stress intensities at a 25 mm head crack in a 60E1 rail due to wheel impact loads. Influence of hanging sleepers is accounted for. Line indicators as in Figure 3.](image)

**DEFINITION OF A "BAD CASE" SCENARIO**

Limit values of allowed wheel–rail impact loads should prevent vehicles that may cause risks of inducing rail breaks from continued operations. On the other hand, operational disturbances due to stopped trains should be minimized. This is especially the case since operational disruptions may shift traffic to road, which is some 50–100 times less safe than rail transportations [6]. The limit values thus need to consider a sufficiently severe, but still realistic scenario. In this context it is important to note that the suggested limit values are intended to be generally valid over a network. To account for extreme local conditions in such limit values is usually sub-optimal since the resulting very low limit values may cause significant operational disturbances. Instead, locally adopted actions, such as more frequent or more detailed inspections of critical points (e.g. switches, welds etc), are usually more (cost-) efficient.

The analysis presented this far has considered a broad range of vehicle types, a worst-case scenario regarding impact load position and hanging sleeper configuration, and a worst case (within operationally realistic limits, see [3]) time history of the impact load. As for ballast stiffness the study has covered stiffness magnitudes down to 5 MN/m. However these are extreme cases, especially since hanging sleepers are also accounted for. Thus, the proposed "bad case" scenario considers values from 30 MN/m per half-sleeper and above. Note that the soft tracks that thereby have been excluded generally correspond to high bending moments, however they are also in general less susceptible to the influence of hanging sleepers.

The peak bending moment in the "bad case" scenario can analytically be expressed as, see [3],

\[
\begin{align*}
\max_t \{M_x(t)\} &= \left(46 + 48 \left(\frac{F_{\text{max}}}{250} - 1\right) \right) \cdot 1.33 \quad (12) \\
\min_t \{M_x(t)\} &= \left(-19 - 15 \left(\frac{F_{\text{max}}}{250} - 1\right) \right) \cdot 1.68 \quad (13)
\end{align*}
\]

Here \(t\) is the time of a wheel traversal and \(F_{\text{max}}\) is the peak magnitude of the impact load. Further, the coefficients 1.33 and 1.68 basically account for the influence of hanging sleepers. Note that since the stress intensity (for a given crack) is directly proportional to the magnitude of the tensile stress (see Equation (5)), the increase due to hanging sleepers directly transfers to an increase in stress intensity factor magnitude, and consequently an increased risk of fracture.

Stress intensities for the proposed “bad case” scenario featuring different impact load magnitudes are indicated by the thick black line in Figure 3 and Figure 4. Fracture corresponds to exceedance of a horizontal line (indicating reduced fracture toughness at different temperatures). To clarify, combinations of impact load magnitude and rail crack size corresponding to fracture at varying temperatures for the "bad case" scenario are presented in Figure 5 and Figure 6.

![Figure 5: Combinations of impact load magnitude and rail foot crack size that result in fracture for the defined “bad case” scenario.](image)
Figure 6: Combinations of impact load magnitude and size of a headcheck crack that result in fracture for the defined “bad case” scenario.

SUGGESTED OPERATIONAL LIMIT VALUES

As mentioned above, operational limit values are inevitably a compromise between stopping vehicles that potentially may be a safety hazard and preventing operational disturbances. As seen from Figure 5 and Figure 6 this can be reformulated as a compromise between how high impact loads that should be allowed versus how small rail cracks that will be found and mitigated during inspection and maintenance.

There are however some caveats to such a discussion:

Firstly, high impact loads may cause also other types of (potentially safety related) damage, e.g. to bearings and wheels.

Secondly, inspections need not only find cracks of the specified sizes, but rail cracks that are in risk of growing to such a size before the next inspection. In this context it should be noted that cracks loaded close to the fracture toughness tend to grow fast.

Thirdly, rail foot cracks are commonly only subjected to visual inspections, which gives a lower probability of detection.

From this basis, proposed limit values set out from a 25 mm head crack and a 5 mm foot crack. In addition, the influence of thermal stresses is found to be so large that it would be unreasonable to account for the entire temperature range with one limit value. Finally, an overall limit of 350 kN is proposed to limit the risk of other types of damage not accounted for in the current study (e.g. damage to bearings, fastenings and sleepers, and large-scale plastic deformation of wheel and rail).

The suggested limit values for allowed wheel loads derived under these premises are presented in Figure 7.

Figure 7: Suggested limit values for impact loads from wheels.

Note that any modification of allowable wheel loads can readily be translated into modified critical crack lengths using the information in Figure 5 and Figure 6.

OPERATIONAL RAIL BREAKS

The suggested limit values set out from the presumption of a rail break at a head check crack size of 25 mm, and a foot crack size of 5 mm. This can be compared to critical crack sizes for operational rail breaks. Examples of operational rail breaks are presented in Figure 8 and Figure 9. Dimensions of the critical crack size that triggered the final fracture are indicated. Note that these cracks correspond to 50E3 rail, which will alter the critical crack size somewhat (cf analyses in [7]).

Figure 8: Operational rail break setting out from a foot crack. Photo courtesy: Anders Frick, Trafikverket.

In Figure 8 it can be noted that the influence of bending is reflected in the length of the critical crack, which is shorter at the upper face of the rail foot than at the lower face. This highlights the influence of the (conservative) simplification of presuming a uniform stress (corresponding to the stress magnitude at the lower rail foot face) that was made in the current analysis.
As for the dimensions of the critical cracks, these are found to be somewhat larger than presumed. If this generally is the case, it would imply an additional safety factor to the analysis in the sense that operational loading is not severe enough to cause fracture for the presumed critical crack sizes.

To evaluate if the critical crack dimension presented in Figure 9 is generally representative for operational rail breaks, an international comparison was carried out [8]. The conclusion from this study was that there are massive deviations in critical crack sizes around the world. A span in the critical crack size from roughly 10% up to roughly 80% of the railhead area was found. The consistency seemed to improve if only a single line was considered. However the statistical basis was too limited to draw any definite conclusions regarding this.

Using the developed analysis framework, it is possible to identify reasons for the wide variations in critical crack sizes. To this end, it should first be noted that there are stochastic components in the events leading up to a rail break. In particular the rail break is triggered by the combination of a large crack and a high stress as seen in Figure 6. Thus if high stress events are absent, the crack will grow longer until failure occurs. Consequently a critical crack size larger than the 25 mm presumed in establishing the alarm limits may correspond to the fact that no wheels inducing high impact loads have been operating during the last stage of crack growth and/or that the failure occurred during a warm season when thermal stresses were low or even compressive.

A critical crack size smaller than the presumed 25 mm is a bit more cumbersome. Analysing the influential parameters, it is seen that a smaller critical crack size can relate to a fracture toughness below the presumed 40 MPa√m and/or an impact load magnitude that is above the proposed limit value. As seen from Figure 6, the influence of load magnitude is however fairly moderate implying a massive increase in load magnitude for the smallest critical crack sizes. To rule out any such possibilities, a failure analysis related to a rail break could assess measured impact load magnitudes of trains passing just before the rail break. Naturally this requires that wheel load measurements were in operation on the line where the rail break occurred.

A second possibility is operational conditions outside the studied scope. This could relate to vehicle types, speeds and/or track stiffness. Regarding speeds and vehicles this is unlikely since the analysis covers a broad spectrum (with the obvious exception of very high speed trains that are considered outside the scope of the study). Locally very soft track support (including hanging sleepers with more than a 2 mm gap), is a more likely scenario. Identifying hanging sleepers at the location of a derailment is virtually impossible due to the massive disruption caused by the derailment. Still an investigation to ensure that the track stiffness in the vicinity of the rail break is acceptable is suitable, not the least to minimize the risk of future derailments in the same track segment.

The third, and probably the most likely, explanation is a (locally) high stress free temperature. Note that the (high) influence of the rail temperature $\Delta T$ does not relate to the absolute temperature, but to the deviation from the stress free temperature. Thus a high stress free temperature will result in high tensile stresses during cold periods. (Likewise a low stress free temperature would lead to high compressive stresses, and a risk of sun kinks, during a warm period.) When a rail break occurs with small critical crack sizes, an evaluation of the stress free temperature in the affected track segment should then be suitable to minimize the risk of future derailments. Note however that the derailment will affect the stress free temperature locally. Consequently, measured stress free temperatures in the vicinity of the rail break will not correspond to those before the derailment.

A fourth possibility for small crack sizes at fracture is that these relate to secondary rail breaks. In such cases the loading may be significantly enhanced by the influence of the primary rail break and does not reflect the presumptions underlying the derived alarm limits. Note however that secondary fractures imply the occurrence of multiple headcheck cracks at a limited track section. Such conditions may cause very severe derailments, cf. e.g. the Hatfield derailment [9].

Note that from the discussion above it is seen that a full clarification of underlying causes for a rail break (with the exception of the critical crack size and possibly likely impact load magnitudes) is difficult. Nevertheless, a thorough investigation should give good indications of the root causes.
and, perhaps most importantly, minimize the risk of further derailments on the affected track segment.

DISCUSSION

Harmonization vs Optimization

The suggested impact load limits comprise a balance between allowed impact load magnitudes and inspection/maintenance needs to limit the size of the largest rail cracks occurring in track. Naturally this balance can be skewed either way: Higher loads can be allowed at the expense of more rigorous inspections, or less frequent inspections can be allowed if lower impact loads are allowed. What is more beneficial depends on the operational conditions, and also on safety requirements: Compare for instance the case of a dedicated freight line to a line with mixed traffic.

In establishing suitable impact load limits for a network, the developed analysis framework can be used to give reliable predictions of the consequences. In combination and off-line analysis of the amount of stopped trains can be carried out (cf. the discussion below).

It should however be noted that optimization of alarm limits can have massive drawbacks. If scattered limits are introduced across an interacting network, vehicles that are allowed to operate in one part of the network may be stopped in another part. Depending on maintenance practices, logistics procedures etc. this can have very adverse and costly effects.

Required Measurement Precision

As noted above, the influence of the impact load magnitude on the magnitude of the stress intensity factor, and consequently on the risk of fracture, is moderate. This would imply that the needed precision in measuring load magnitudes is limited.

From a technical point of view this is correct. However from a legal perspective this may not be the case since a poorly calibrated wheel load detector may stop a train that is in fact within allowed load magnitudes. This might lead to claims for reimbursement etc.

In this context it should also be noted that the alarm limits relate to the peak load magnitude of an impact load. Thus, the load detector needs to be able to reliably include the effect of high-frequency load contributions at least up to 1 kHz.

Head Cracks and Foot Cracks

A comparison of Figure 8 and Figure 9 shows that the critical size of a rail foot crack is significantly smaller than that of a railhead crack. Furthermore, these cracks are normally not investigated in standard non-destructive testing (NDT) procedures.

However, as noted above, the severity of a rail break (i.e. the risk of derailment) generally increases significantly in the case of multiple rail breaks within a short distance (cf. the Hatfield derailment, [9]). This is less likely for rail foot cracks for two reasons:

1. Head cracks form during operation as a response to the local loading conditions in terms of frictional stresses and contact geometry [10]. Consequently, formation of large head cracks in clusters is likely along a curve where wheel loads and contact geometries are fairly uniform. In contrast, rail foot cracks are consequences of (more or less) stochastically distributed initiators such as scratches, cuts, corrosion pits etc combined with high local load magnitudes. Thus, the likelihood of rail foot crack initiation in clusters is less. It has however been shown [11] that lateral bending of the rail (i.e. imposing a bending moment $M_y$ following the coordinate system in Figure 1) increases the loading of a foot edge crack significantly. Thus, the risk of clustered rail foot cracks in (fairly) tight curves is not insignificant.

2. The influence of hanging sleepers is higher for railhead cracks. Due to this fact, it would be more likely that clustered head cracks that are initiated in the vicinity of a hanging sleeper propagate to failure than it is that clustered rail foot cracks initiated at a hanging sleeper do.

One should also note that, for a stretch of track with track circuits, there is (in theory) a 50% chance that a rail break is captured by the signalling system. Consider two cracks at a short track section. For a rail foot crack, the two cracks are likely to be initiated independently. The chance is thus roughly 50% that two cracks appear on opposite rails. For head check cracks, the chance is close to 100% that they are initiated on the same rail (the outer rail, since the load conditions are more detrimental for head crack initiation on this rail). Thus, the probability that at least one of two rail breaks (occurring at roughly the same time) will be detected by the signalling system is for the rail foot crack 75%, and for the head crack 50%, provided there is a 50/50 probability that the signalling rail is on the inner/outer rail. Note that the likelihood of detecting rail breaks increases significantly if the outer rail in a curve is always adopted as the signalling rail.

In relation to this discussion it should also be noted that the introduction of alternative signalling systems where rail breaks cannot be detected increases the risks of undetected rail breaks (and thus derailments).
What the Suggested Wheel Load Limits Do Not Cover

The suggested limit values for wheel loads account for the influence of impact loads on the risk of rail breaks. They do not account for the influence of these loads on other components such as fastenings or sleepers. The main reasons are that these components are less safety related, and that the influence on these components is more related to their detail configuration. As an example, the influence of sleepers is strongly dependent on the support conditions of the sleeper. More details on this can be found e.g. in [12].

Further, the focus of the current study is on fracture related to long cracks. Initiation and growth of small cracks are not addressed. It should be noted that mitigating initiation and early crack growth might lead to major cost savings. In addressing these phenomena it should be noted that they typically relate to other key operational parameters such as frictional forces and contact geometry, cf. [10].

Derailment due to other causes, such as flange climb is also not captured by the proposed limits. To this end, other criteria have been derived [13] that could be employed in parallel. These criteria are based on the imbalance between wheel loads within a vehicle and employ average (over time) load magnitudes for each wheel (in contrast to the impact load magnitudes employed in this study).

The case of general overloading of a vehicle is also not considered in the establishment of the wheel load limits. Instead it is generally prescribed as an allowed axle load for a line. With some exceptions (e.g. risks of bridge collapses and landslides) overloading is not directly a safety critical phenomenon. However it will increase deterioration of the track and also gain unfair advantages for operators who choose to ignore the prescribed axle load limits. To quantify this increased cost caused by overloading, a reasonable approach is to consider the damage as proportional to the cube of the overloading, cf [14]. A rough estimation of the additional cost for an overloaded axle will then be

\[ \Delta c = \left( \frac{Q}{Q_{\text{all}}} \right)^3 - 1 \]  

(14)

Here \( \Delta c \) is the additional deterioration cost due to an overloaded axle, \( Q \) is the (average) load of the overloaded axle, \( Q_{\text{all}} \) is the allowed axle load, and \( c_{\text{nom}} = c_{\text{tot}}/n \) is the nominal cost of the track with \( c_{\text{tot}} \) being the total (annual) cost of the track, including (accrued) investment and reinvestment costs etc., and \( n \) being the number of axles over the time \( c_{\text{tot}} \) was calculated.

Implementation strategies

Before the suggested alarm limits are implemented operationally they should be contrasted with existing alarm limits. Operational consequences (e.g. in terms of stopped trains) can then be analysed on a network-per-network level in an off-line analysis. Such a network analysis should also ensure that additional potential safety issues are not overlooked. Any actions needed to avoid unnecessary operational disruptions and/or safety issues can then be enforced before the alarm limits are put in action.

Recall that the alarm limits depend on the (rail) temperature. This requires some practical considerations. Tentatively, a two-limit approach can be employed where a first step is to ramp down to an intermediate limit of \( Q_{\text{max}} = 300 \text{ kN} \) at a temperature 30 degrees below the stress free temperature. The corresponding absolute temperatures need to be evaluated for the different regions of the network and kept available for operators. Note here that a large scatter in absolute temperatures within the network will lead to operational complications (cf the discussion above).

It is also important that operators and maintenance contractors know in advance what the operational limits are. To this end, temperature forecasts need to be made. Depending on the required level of advance notice, these can correspond to weekly or monthly prognoses, or even fixed seasonal dates decided once and for all.

As mentioned, the proposed limits are upper safety related limits. Damage to rail, sleepers, wheels, bearings etc. may be high even at the allowed levels of wheel forces. To address this, maintenance limits (and potentially limits where wheels are indicated for future trend analyses), may be defined. For wheel loads above these limits, operators are notified and can plan required maintenance of the wheels to avoid operational disruptions.

Finally, it should be noted by infrastructure managers that the proposed limits consider the risk of rail breaks only on nominal rails. In cases of insulated joints, welds, switchblades, heavily worn rails in curves etc., risks will be increased due to the local increase in loading. Inspection routines should thus account for such risks. Also in such an assessment, the current study should provide a good basis.

CONCLUSIONS

The current study investigated the influence of wheel impact loads on the risk of rail breaks. To this end, evaluated relations between impact load magnitudes and rail bending moments were employed. Resulting normal stresses in the rail
were evaluated using beam theory and combined with thermal stresses due to cooling of an continuously welded rail below the stress free temperature. Finally, stress intensity factors evaluated for different crack lengths and stress magnitudes were contrasted with the fracture toughness of the rail. The result was an identification of combinations of crack lengths and impact load magnitudes likely to cause rail breaks under certain operational conditions.

To extract limits for allowable wheel loads, a "bad case scenario" featuring detrimental, but not unrealistic, operational conditions was derived. By combining the identified "bad case" conditions with the presence of reasonably large (head check and rail foot) cracks, allowable impact load magnitudes were proposed as those that do not cause fracture in the presence of such cracks. The proposed impact load limits are presented in Figure 7. In particular it could be noted that they account for the high influence of the rail temperature.

Presumed critical crack sizes were compared to cracks triggering operational rail breaks and found to be of comparable magnitude. However a significant scatter of critical crack sizes from around the world has been reported. Reasons for this scatter were discussed in the light of the established analysis framework and deviations in the stress free temperature of the rail was indicated as a major potential cause.

Implementation related topics were then discussed. These include benefits / drawbacks of harmonized load limits and how the developed analysis framework can be employed to study consequences of modifying limits. Differences between head and foot cracks in terms of safety aspects were highlighted and the influence of factors such as the employed signalling system were discussed. Topics related to wheel/rail interaction and subsequent deterioration that are not addressed by the derived wheel load limits were especially identified and briefly discussed with a focus on how they can be accounted for.

Finally, the paper provided a brief discussion on some important factors related to the implementation of the proposed limit values. In particular this concerns a pre-evaluation of consequences of imposed wheel load limits, and how the temperature dependence of the limits can be addressed.

REFERENCES


7. Franklin F (ed), INNOTRACK Deliverable 4.2.5 – Improved model for the influence of vehicle conditions (wheel flats, speed, axle load) on the loading and subsequent deterioration of rails. 2009 (www.innotrack.eu)


11. Hejzlar L (ed). D-RAIL Deliverable D6.1 – Analysis of tests for the validation of numerical simulations. 2014 (d-rail-project.eu)


IHHA 2015 Conference
21 – 24 June 2015
Perth, Australia