Development of the Future Rail Freight System to Reduce the Occurrences and Impact of Derailment

D-RAIL

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D3.3

Guidelines on derailment analysis and prevention

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Executive Summary

The present deliverable, D3.3, is aimed at implementation of results from D3.2. To this end, it includes the following information for each derailment scenario studied in D3.2:

Extraction of the most important results from the studies detailed in D3.2.

- Brief description of the studies carried out
- Brief summary of most important conclusions

This part is mainly targetted towards operational implementation outside of the D-RAIL project.

To support further development of maintenance / monitoring solutions, the guideline provides information on

- What needs to be measured / monitored?
- Which measurement / monitoring accuracy is needed?
- Tentative limit magnitudes
- How measured data should be employed in operational control and maintenance planning?
- Parameters / scenarios suitable for test investigations / validations
- Input on key parameters in causing derailments and parameters that should be measured (and how these relate to the key parameters if the key parameters can't be measured directly)

This part is mainly targetted towards further investigations within the D-RAIL project.

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Glossary

IAL	Immediate Action Limit
RIV	Regolamento Internazionale Veicoli
UIC	Union Internationale des Chemins de fer
WP	Work Package

1 Introduction

The current guideline sets out from the top 8 derailment causes as identified in D-RAIL WorkPackage (WP)1 and listed in D-RAIL deliverable D1.1, see [1]:

1. Axle ruptures (dealt with in EURAXLES)
2. Excessive track width
3. Wheel failure
4. Skew loading
5. Excessive track twist
6. Track height/cant failure
7. Rail failures
8. Spring & suspension failure

In D-RAIL WP3, Task 3.1, a top-down analysis is carried out where cause–consequence chains are established together with matrices linking potential mitigating actions to their current level of implementation. Results are presented in D3.1, see [2].

A bottom-up approach has then been adopted in Tasks 3.2 and 3.3. Here numerical simulation have been adopted to facilitate detailed analyses of derailement scenarios. The aim has been to define threshold operational conditions for derailments. Details on these investigations are presented in D3.2, see [3].

The current deliverable (D3.3) contains an easy-to-access summary of the studies with focus on main results and input for further development of monitoring / maintenance solutions to prevent the occurrence of derailments and minimize the impact of any remaining derailments.

References

1. D-RAIL, **D1.1: Summary report and database of derailments incidents**, 71 pp + 1 annex (3 pp), 2012
2. D-RAIL, **D3.1: Report on analysis of derailment causes, impact and prevention assessment**, *in preparation*, 2013
3. D-RAIL, **D3.2: Analysis and mitigation of derailment, assessment and commercial impact**, *in preparation*, 2013

2 Flange climbing in line operations

2.1 Studied mechanisms and investigated parameters

A number of vehicle parameter studies were carried out to understand their relationship to derailment propensity within plain line operations.

Essentially they can be divided into three categories:

- Skew loading (longitudinal and lateral load offset)
- Bogie suspension variation:
  - Tare spring rate
  - Laden spring rate
  - Tare-laden spring clearance
- Bogie yaw resistance variation
- Analysis of measured wheel load data with respect to skew loading

2.1.1 Simulation results and parameter variations

A base vehicle model was configured using nominal parameters for a Y-series freight vehicle. The relevant parameters of the base vehicle were then modified as necessary to generate the derivatives for the parameter variation studies identified above.

For each set of parameters the vehicles were simulated through different derailment resistance cases.

A number of derailment assessments were used to establish the influence of parameter variation on flange climb derailment. To provide a reference framework and also to include a review of the current industry standards, the assessments were based on the Euro Norm EN 14363 and on the GB Railway Group Standard (RGS) GM/RT 2141. All the assessments were simulated using the Vampire vehicle dynamic simulation software. The scenarios, and relevant track geometry and simulations of laboratory tests studied were as follows:

- EN 14363 ΔQ/Q test – Simulation of a static laboratory test.
- GM/RT 2141 ΔQ/Q Appendix A – Simulation of a static laboratory test.
- X-Factor test (common to both standards) – Simulation of a static laboratory test.
- EN 14363 Y/Q test – Simulation of an on track flange climb assessment.
- GM/RT 2141 Appendix C, Y/Q simulations – Suite of transient simulations to assess flange climb.
- GM/RT 2141 Appendix D, Assessment of on-track ride; vertical and lateral body accelerations

Note that the conditions used to determine the ΔQ/Q quotient in GM/RT 2141 and EN 14363 are dissimilar, while the limit values imposed are the same.

The influence of a number of suspension failure modes on the derailment risk of Y-series bogies was also studied. These are summarised below:

- Failed Lenoir Link
- Failed Primary Spring
- Failed Sidebearer Unit
- Bogie Frame Twist

The method of assessing derailment risk was based upon the following analyses from the GB RGS GM/RT2141:

- GM/RT 2141 Quasi-static Assessment of Wheel Unloading (ΔQ/Q)
- GM/RT 2141 Low speed flange climb assessment (Y/Q)
- GM/RT 2141 On-track ride (Acceleration Peak Counting)

The above assessments provide an indication of the vehicle derailment propensity due to changes in dynamic wheel unloading, changes in quasi-static in response to low speed flange climb through high track twist or curvature.

In order to investigate the influence of isolated track defects which are based on the IAL's according to EN 13848-5, additional vehicle dynamics simulations are carried out using artificial sinusoidal track defects of variing length. The simulations consider runs in full curve sections of different curve radii and cant deficiencies. The selected parameter are studied for tare as well as skew loaded vehicle state.

2.1.2 Analysis of measured wheel load data

To provide further context to the results from skew loading simulations, the distributions of wheel load from in-track measurement sites were analysed. In addition the data analysis investigated the possibilities for detecting chassis and frame twist from the measured wheel loads.

On request, GB IM manager, Network Rail, provided on-track wheel load measurement data based upon the GOTCHA measurement system. In addition data from DB's axle load checkpoints "DafuR" was made available. Both measurement systems aim to record the dynamic and mean vertical wheel forces of all passing vehicles with the primary objective of detecting wheel flats and high axle loads.

The GB Gotcha data was post processed to obtain the distribution in skew loadings for 8 weeks of traffic. The raw data contains measurements for all passing vehicles, which includes non-bogie freight, passenger stock and locomotives. All non-bogied vehicles were removed from the data set, along with all bogied vehicles with a wheelbase greater than 2.2 m (the majority of passenger stock). The bogie semi-spacing was then used to identify intermodal vehicles.

The DB DafuR data set contained measurements from four different locations in the DB network within a time period of one month. All the selected axle load checkpoints are situated in straight track and the investigation was restricted to freight wagons with 2-axle bogies.

In both cases, the integrity of the mean wheel load data was checked by looking at the dynamic load ratio (dynamic force/mean force), and disregarding any vehicle with a wheel reporting a dynamic ratio greater than 0.75. A high dynamic ratio indicates the likelihood of a wheel defect, which in turn has been found to lead to uncertainty in the measured mean wheel load. 8,597 vehicles were disregarded from the GB data set due to high dynamic ratios, leaving 22,320 vehicles for analysis, while the filtered DB data set contained 55,262 vehicles.

2.2 Main conclusions from simulation and analyses

2.2.1 Main influencing parameters

The most influential parameter categories were found to be:

- skew loading, especially lateral load imbalance, can significantly affect both the bogie yaw resistance and static wheel loading, with a consequential reduction in the ability of a vehicle to safely negotiate track twist and curvature;
- high levels of lateral skew loading can cause contact within the side bearer vertical bumpstop assemblies. This can result in a step reduction in the ability of a vehicle to safely negotiate track twist and curvature (UIC centrebowl and pre-loaded side-bearer assembly);
- primary suspension stiffness, especially its transitional behaviour between tare and laden loadings;
- amplitude and length of isolated track defects.

2.2.2 Tentative limit values

From the skew loading investigations based on simulation and measurement data, it is apparent that the UIC RIV limit for lateral load imbalance of 1:1.25 is appropriate. The skew loading cases simulated with lateral load imbalance in excess of this limit were likely to exceed the established ΔQ/Q and Y/Q limits imposed in EN 14363, especially when coupled with longitudinal load imbalance.

From the investigations regarding bogie yaw resistance it was also apparent that the UIC RIV load imbalance of 1:1.25 is appropriate. Skew loading cases with lateral load imbalance beyond this limit were found to generate bogie X-Factors in excess of the 0.1 limit.

The dynamic simulations with varying isolated track defects showed that a worst case scenario with combined lateral and longitudinal skew loading in very small radius curves may lead to critical wheel climbing even if the vehicle meets the UIC RIV loading guidelines and the track geometry complies with EN 13848-5. Consequently a combination of extensive lateral and longitudinal load asymmetry should be avoided and adequate reglementations should be introduced.

Comparison of the results from the EN 14363 and GM/RT 2141 low speed flange climb simulations suggest that either the assessment criteria stipulated in GM/RT 2141 are too severe, or that the EN 14363 criteria are too lenient. In contrast the other derailment limits used in the analysis (ΔQ/Q, X-Factor, RIV skew loading limits and EN 14363 Y/Q) seem well aligned to each other.

It would appear that a review of the low speed flange climb assessments used in the EN and GM/RT2141 would be a valuable exercise.

Based on the analysis of wheel load data sourced from DB and Network Rail it was also concluded that the RIV limit values were a good starting point for understanding skew loading levels on railway infrastructure. However further work would be required to fully understand practical alarm levels, given that it has been shown from the NR data that 0.29% of intermodal vehicles excceded the RIV latreal loading limit.

From the NR data analysed, a hypothetical assessment of the impact of an alarm threshold was made. If a dynamic ratio threshold of 0.75 is applied the data showed that 12 axles exceeded a lateral imbalance of 1.70. It is not known how these highly imbalanced axles were distributed amongst different vehicles or consists. However a worst case may be found by assuming that no consist/train would have more than 1 highly imbalanced axle. The entire intermodal Gotcha data set contained 1268 consists. If an alarm threshold was implemented at an axle imbalance of 1.70, twelve of those consists would be stopped in order to remove the offending vehicle, which (in this worst case scenario) represents an impact on 0.94% of intermodal traffic. In a more optimistic scenario, where all the axle imbalances are the consequence of skew loading and four alarms may be attributed to a single vehicle this value may be quartered (to around 0.24%). To complete the range of possibilities, a third scenario may be posed where all the offending axles are distributed amongst a single consist. In this case the alarm limit would impact 0.08% of intermodal traffic.

If no dynamic threshold were to be applied, and implementing the same logic as above, it could be expected that between 0.16%, 4.4% and 17% of intermodal consists would be affected.

This demonstrates that setting an alarm threshold even well above the stipulated limit, where simulations suggests a high derailment risk, could have a potentially catastrophic impact on rail freight. The integrity of the measurement data used to trigger such an alarm is therefore paramount and should be investigated prior to further analysis work being carried out.

As not only the amplitude but also the length of isolated track defects was found to be important the track geometry assessment should take this into account too. It is shown that IAL track defects shorter than 8 m can become critical, especially in very small radius curves.

2.2.3 Potential preventive measures

From the skew loading investigations the effect of the increase in yaw resistance was found to worsen when the lateral load imbalance was sufficient to bring one of the side bearer bumpstops in to contact.

Initial simulations suggest that a reduction in X-Factor of up to 20% might be achieved by increasing the sidebearer vertical travel in these circumstances. The practical feasibility of this, when considering other constraints such as vehicle gauging requires further investigation.

It was found that the transitional behaviour of the primary suspension system for part-laden vehicles is important and should be optimised at the vehicle design stage to maximise not only ride and gauging performance but also derailment resistance.

Consideration could be given to multi-stage stiffness transition in the primary suspension or adoption of rubber components to increase the derailment resistance for part-laden/inter-modal traffic.

It was found that in general, the bogie rotational resistance (X-factor) is not a critical derailment control measure for the friction type arrangements studied. However, the behaviour of the centrebowl arrangement can significantly influence the part-laden and laden X-factor values. This can lead to approximately a 10-20% increase in Y/Q values within the curves studied and exceedence of the 1.2 limit value. Therefore good maintenance practice in this area is recommended.

With regard to suspension failures it was demonstrated that the fault with the largest effect on derailment propensity was a twisted bogie frame. Any twist in the bogie frame has a large effect on the vertical wheel loads with a lower wheel load posing a greater derailment risk. In the cases examined, a twisted frame resulted in up to 64% increase in ΔQ/Q and up to 30% increase in Y/Q. The results from the twisted frame analysis highlight the importance of maintaining the correct bogie geometry in manufacture, assembly and maintenance. In addition, effective bogie ‘twist’ due to incorrect suspension packing or component failure should be monitored and avoided.

Consideration should be given to the addoption of in-track based measuring systems to monitor wheel loads. The RIV limit values of 1:1.25 lateral load imbalance and 1:3 longitudinal load imbalance make a good basis for a monitoring limit. However, further work is required to determine practical alarm values which would not create to great a restriction on the movement of traffic or reduction in the competitivness of the rail mode of freight distribution.

2.3 Recommendations and concluding remarks

2.3.1 Recommended actions

What needs to be detected / measured / monitored?

- Wheel/rail contact loads
  - quasistatic vertical forces
  - quasi-static lateral forces
- Friction coefficient
  - sidebearer
  - centre bowl
- Track geometry
- Review of the off-loading capacity of the UIC and other sidebearer elements to ensure that acceptable X-factor values (<0.10) can be maintained at higher levels of load offset. Any study should include the wider ranginging aspects such as gauging, influence of body/chassis stiffness, overall risk and business case.

Which measurement / monitoring accuracy is needed?

- Wheel/rail contact loads
  Analysis has shown that given the observed levels of axle imbalance, an acceptable level of accuracy for analysis is typically +/-2%. This would require single wheel load measuremnent accuarcies of the order of +/-1% of the static wheel load (approximately). For low axle loads this may be difficult to acheive (due to load resolution and magnitude of error) but these levels should form the target.

- Friction coefficient
  Measuring friction coefficient is always a difficult task, therefore a good maintenance practice in this area is recommended in order to avoid excessive friction both in the sidebearers and in the centre bowl

- Track geometry
  The minimum requirements on track measuring systems are defined in EN 13848 series.

Tentative limit magnitudes

With regard to lateral and longitudinal load imbalance, UIC RIV limit values respectively equal to 1:1.25 and 1:3 are found to be appropriate, but a combination of extensive lateral and logitudinal load asymmetry should be avoided, see section 3.

Large friction coefficients should be avoided both in the sidebearers and in the centre bowl, in principle values higher than 0.4 should be avoided.

When assessing isolated track defects attention should be paid not only on their amplitude but also on the defect length. It is shown that IAL track defects shorter than 8 m can become critical, especially in very small radius curves.

How measured data should be employed in operational control and maintenance planning?

It is recommended that in-track measurement system data is used to establish the true population of offset loading levels. Based on analysis of this data and also cross-reference to skew loading levels of known derailed vehicles, alarm limits for skew loading should be defined. Based on the data analysis, the alarm limit value should capture only out-lying vehicles. It is recommended that a system is put in place that can initially be run in an off-line mode; this will allow the alarm threshold to be adjusted to ensure that it does not overly impact railway operations. Gathering of off-line information would also allow the potential commercial imapct of opertaing such a system to be established and thereby further assist in the setting of a practical alarm thresholds.

Parameters / scenarios suitable for test investigations / validations

It would be useful to test and/or develop a low cost modular wheel load measurement system which could be adopted across the rail market. This would allow efficient cross-border monitoring of skew loading. Alternatively, existing technology could be employed but uncertainty in measurements with respect to dynamic ratio found in this work should be investigated. (Dynamic ratio is the ratio of dynamic force component/static force).

Input on key parameters in causing derailments and parameters that should be measured (and how these relate to the key parameters if the key parameters can't be measured directly)

The simulation and analysis work carried out within this work package recommends that the primary mitigation measurement should be the level of lateral skew loading. This can be monitored by adoption of an in-track systems to measure vertical wheel loads.

Further derailment prevention mitigation could also be gained through the parallel measurement of the lateral wheel force, enabling the Y/Q ratio to be monitored. In-track measurements of lateral forces were not available to the project but alarm levels close to the Nadel quotioent limit of 1.2 could be considered as a practical start point.

2.3.2 Potential commercial impact

The commercial impact of detecting vehicles with skew wheel loads is difficult to determine prior to carrying out of a detailed study on the populations of skew loaded vehicles operating on the railways. The level of alarm limits and also the method of quaratining the vehicles will also impact on the commercial aspect. Commercial benefit may be realised when the costs associated with the system implementation and operational disruption are less than the costs associated with rectifying damage and disruption that would otherwise have been caused by the prevented derailments.

2.4 References

1. D-RAIL, **D3.2: Analysis and mitigation of derailment, assessment and commercial impact**, *in preparation*, 2013

3 Derailment in switches & crossings

3.1 Studied mechanisms and investigated parameters

The derailment scenario studied for S&C is flange climb derailment in switches. The traffic situation studied is traffic in the diverging route of a small radius (R=190m) right hand turnout. This scenario is chosen because simulations and measurements show that the Y/Q-ratios for traffic in the diverging route are larger than those recorded for the through route [1]. Further, simulations show that traffic in a small radius turnout generates larger Y/Q-ratios than a turnout with larger radius for typical speed limits (See Chapter 4 of D3.2). As the Y/Q-ratio correlates to the risk of derailment according to Nadal's criterion [2], the small radius turnout is considered to be the most critical case.

Four different parameter studies have been performed to evaluate a total of 25 track and vehicle parameters using design of experiments and response surface methods. All parameters are listed in Table 1.

Table 1 List of investigated parameters

Side bearer friction coefficient
Centre plate friction coefficient
Primary suspension friction coefficient
Side bearer play
Longitudinal play in primary suspension
Lateral play in primary suspension
Scale factor for vertical primary spring stiffness
Scale factor for horizontal primary spring stiffness
Bogie c-c distance
Axle c-c distance in bogie
Chassis torsional stiffness
Traffic move
Wheel polygonalization
Speed
W/r tread friction coefficient
W/r flange friction coefficient
Bogie load ratio for laden case
Side load ratio for laden case
Chassis twist
Car body CoG vertical for laden case

Lateral track irregularity amplitude
Vertical track irregularity amplitude
Track twist irregularity amplitude
Track gauge irregularity amplitude
Wheel profile

The major parameter group missing in these studies is the switch rail geometry itself. Efforts were made to create worn geometries based on Swedish maintenance limit templates. It was however found that a large number of assumptions were still needed regarding the actual shapes of the worn profiles. It is judged that a representative set of measured switch rail profiles on the current maintenance limit would be needed to perform this task, and that has not been feasible within the D-rail project.

3.2 Main conclusions from numerical simulations

3.2.1 Main influencing parameters

Based on the simulation based parameter studies, the most influential parameter categories were found to be

- Friction. The wheel rail friction coefficient is the most important in this category (as can be expected from Nadal’s criterion), but also the friction coefficients in the primary and secondary suspensions are important. They affect among other things the yaw stiffness of the vehicle. The magnitude of influence for the different friction coefficients depends on the load state of the vehicle.
- Skew loading. Especially combined lateral and longitudinal skew loading.
- Chassis twist. It was found that chassis twist and longitudinal skew loading can interact in a non-linear fashion when they both strive to unload the same wheels. According to DB measurements (Chapter 3 of D3.2), diagonal load imbalance for tare state wagons in traffic can be significant which indicates the existence of significant chassis twist for wagons in traffic.
- Track irregularities, especially track twist. A strong interaction was found between track twist and lateral alignment. If the track twist unloads the outer wheel (reduced Q) and the lateral alignment pushes the same wheel inwards (increased Y) the resulting increase in Y/Q is larger than what can be expected from the sum of the contributions that these parameters add if they are varied one at the time.

Also, traffic in the facing move of the turnout was found to record the largest Y/Q-ratios and therefore the derailment risk is expected to be largest for this direction of travel.

Studying the influence of load state, it was concluded that a tare state wagon generates larger Y/Q-ratios than a laden wagon when the payload is centred. If the skew loading is large, a laden vehicle can generate larger Y/Q-ratios than a tare state vehicle.

The exact magnitude of the influence for each parameter on the Y/Q can be found in D3.2.

3.2.2 Derailment surface

Based on the parameter screening, a bad case vehicle-turnout combination was defined where all parameters were set to their worst setting. By introducing a parameterization of the model in the most influential parameters (but friction) it was possible to estimate a derailment limit surface as a function of these parameters. The results suggest that a vehicle can derail within current standards if all parameters are set in their worst or close to worst position.

However, considering the studies presented in Chapter 3 of D3.2 on measured wagon skew loading and the relation between track irregularity amplitude and wave length, it should be noted that the parameter combinations leading to derailment in simulations are very unlikely in practice even if they are theoretically feasible within standards. Also, some of the variable levels used are based on engineering judgement.

Simulations suggest that the risk of derailment is larger in a curve featuring switch rail geometry compared to a plain line curve, everything else being equal. Maintenance tolerances for track irregularities in switches could therefore be tighter than those for plain line. In e.g. Sweden this is already the case for the gauge dimension [3].

Studying the influence of load state, it was concluded that a tare state wagon generates larger Y/Q-ratios than a laden wagon where the payload is centred. If the skew loading is large, a laden vehicle can generate larger Y/Q-ratios than a tare state vehicle. The derailment limits for laden and tare state vehicles can be compared more closely from the figures in Chapter 4 of D3.2.

3.2.3 Tentative limit values

In order to find tentative limits for the influential vehicle parameters longitudinal skew loading, lateral skew loading and chassis twist, a derailment surface was obtained for a parameterized vehicle running through a switch with bad but realistic track irregularities. Assuming that the simulation case is a realistic bad case, some tentative limits could be suggested for these parameters based on the performed simulations.

In order to set limits, distinction should first be made between nominal and observed vehicle and axle load imbalances as defined in Section 3.5. The nominal imbalances are of interest for loading guidelines while the observed load imbalances are relevant for the detection of derailment prone vehicles in track. Simulations show that a nominal lateral load imbalance (4) of 1:1.25 can correspond to an observed load imbalance for the vehicle (6) of up to 1:1.3 due to suspension compliance and other asymmetries that can arise when the vehicle is asymmetrically loaded in both the lateral and longitudinal direction. Longitudinal skew loading is less sensitive and a nominal load ratio (3) of 1:3 between bogies gives more or less the same result if calculated using observed wheel loads from simulation in (5).

For chassis twist the correlation between the applied twist and the diagonal load ratio (7) is good for a tare state vehicle. For a laden vehicle the correlation is poor as the diagonal load ratio can be affected by skew loading due to non-linearities in the suspension. The diagonal load ratio is dependent on the compliance properties of the specific wagon in question and is therefore not easily determined before the vehicle is loaded.

It is further shown that the maximum axle load imbalance (9) is a non-linear function of skew loading and chassis twist. It means that the observed axle load imbalances can be much higher than the nominal. See also Section 4.5.14 of D3.2

Limits for observed load imbalances

Studying the simulated derailment limit as a function of load imbalances (Section 4.5.13-15 in D3.2), it seems that a limit of the observed lateral load ratio (6) of 1:1.35 and a limit of the observed longitudinal load ratio (5) of 1:3 could be reasonable for the enforcement of skew loading limits. With these limits there is still some margin to derailment in each individual direction such that some chassis twist and skew loading in the other direction could be accepted in combination with an extreme value in one direction. Individual limits for lateral and longitudinal skew loading should work reasonably well in practice as measurements indicate that cases of combined extreme skew loading are rare. It is recognised however that combined skew loading is the worst case and a loading situation which is not assessed with individual criteria for the longitudinal and lateral direction.

Considering limits on the maximum axle load imbalance (9), the derailment surface study shows that all parameter combinations with an associated axle load imbalance above 1:2 derail for the given simulation set-up. In the light of these results, the SBB axle load imbalance limit of 1:1.7 is definitely reasonable as most vehicles with a maximum axle load imbalance above this limit derail. It is therefore suggested as a tentative axle load imbalance limit.

The load imbalance limits on vehicle and axle level are complementary. Sometimes the axle load imbalance is stricter than the imbalance on vehicle level and vice versa. It is therefore recommended that both limit types are used to detect derailment prone vehicles. Especially the maximum axle load imbalance and the longitudinal vehicle load imbalance are complementary. The lateral load imbalance on the vehicle level is almost rudimentary if the axle load imbalance is also in place according to this study. The axle load imbalance criterion was shown to be the better single imbalance criterion in this study as it is better at separating the derailing vehicles from the non-derailing vehicles for the investigated parameter space.

For the investigated derailment surface, it has been shown that it is possible to formulate criteria that can better discriminate between the derailing and non-derailing vehicles by taking the full load imbalance state of the vehicle into account. Criteria have been formulated using ellipsoidal, planar and line equations which are functions of the vehicle's load imbalances. Of these the line criterion showed the most promising result, and it is suggested as a load imbalance limit.

The criterion limits the maximum wheelset load ratio as a function of the observed longitudinal vehicle load ratio using a standard line equation. It is written as:

$$\varphi_{\text{axle,observed,max,limit}} = k_l \varphi_{\text{long,observed}} + m_l \qquad (1)$$

using the load imbalance definitions of equations (5) and (9). With k_l=-0.25 and m_l= 2.05, the criterion limits the $\varphi_{\text{axle,observed,max}}$ to 1.8 when $\varphi_{\text{long,observed}}$=1 and $\varphi_{\text{axle,observed,max}}$ to 1.35 when $\varphi_{\text{long,observed}}$=3.

Forming a quotient between $\varphi_{\text{axle,observed,max,limit}}$ as calculated for an observed $\varphi_{\text{long,observed}}$ and the corresponding observed $\varphi_{\text{axle,observed,max}}$, the line criterion l can be formulated as:

$$l = \frac{\varphi_{\text{axle,observed,max}}}{\varphi_{\text{axle,observed,max,limit}}} \tag{2}$$

A value above 1 would mean that the vehicle is beyond the estimated derailment limit. The details are given in 4.5.15 of D3.2.

Limits for nominal load imbalances

The simulations show that the RIV loading guidelines are appropriate, but it is a problem that they do not account for (extensive) combined skew loading which was found to be critical in simulations. It is therefore recommended that the loading guidelines are updated such that they also limit the amount of combined skew loading allowed. The planar derailment criteria of Section 4.5.15 in D3.2 could be a starting point for such a discussion. It can be simplified into two dimensional criteria that is a function of longitudinal and lateral skew loading. This creates a rombic loading limit surface instead of the rectangular which is the result of the RIV-limits.

Chassis twist

It is recommended that tare state vehicles with a diagonal load ratio (7) above 1:1.3 should be inspected for chassis twist. To reach this level of diagonal load ratio a chassis twist of 32 mrad was applied to a chassis modelled as rigid. If the diagonal load ratio is as high as 1:1.7, the wagon is estimated to be on the derailment limit and should be stopped then if not earlier. As there is no significant skew loading for a tare state vehicle, the stated diagonal load ratios correspond to the average axle load imbalance in each bogie. This means that an axle load imbalance of about 1:1.7 corresponds to the derailment limit for a tare state vehicle in this study.

3.2.4 Potential preventive measures

One way to mitigate excessive skew loading is to make sure that wagons are loaded correctly and solidly in the first place. E.g. containers can be checked for weight distribution before they are loaded onto wagons.

Uneven wheel loads due to chassis twist or other causes can be checked in maintenance workshops in case this exercise isn't performed already.

In track, skew loading and diagonal loading (due to for example a twisted chassis) can be detected by post processing of data for individual vehicles from wheel load measurement stations. Skew load measurements can detect vehicles with a poor load distribution while the measurement of diagonal loading is mainly useful to detect vehicles in need of maintenance in their tare state.

As it has been shown that the switch rail geometry slightly increases the risk of derailment compared to standard rail, everything else being equal, one potential preventive measure for switches is to prescribe tighter track irregularity tolerances in switches.

Friction management in both the wheel-rail contact and in the suspension of the Y25 bogie can be useful mitigation measures considering the importance of friction with regards to the risk of derailment.

3.3 Recommendations and concluding remarks

3.3.1 Recommended actions

It is recommended that cargo units such as containers are checked for load imbalances before they are loaded onto vehicles.

It is recommended that data from axle load check points are used to detect vehicle and axle load imbalances. The tentative limits and criteria given in 3.2.3 can be used as a starting point.

This work has highlighted that the derailment limit is a function of many influencing parameters which makes it difficult to set limits for each parameter separately. The suggested approach for derailment mitigation is that high limits that would stop only a very small part of the vehicle fleet are implemented and enforced for highly influential parameters such as skew loading and chassis twist. Through investigation of the stopped vehicles, more information can be obtained about the correlation between vehicle wheel load status and the vehicle condition and limits could then be tightened in steps if the stopped vehicles are in a very poor condition. Different levels of the limits could also be applied. E.g. above a certain threshold the vehicle isn’t stopped but has to be inspected within a specified time period. The objective in mind must be to increase the overall safety and efficiency of the railway freight sector, not to obstruct it.

Suggested alarm limits for lateral and longitudinal skew loading are load ratios of 1:1.35 and 1:3. The warning limits could be put at 1:1.3 and 1:2.5 respectively. The alarm limit for individual axles could be set to 1:1.7 which corresponds to the SBB limit which was found to be reasonable in simulations. If it is feasible to use a multi parameter criterion, the line criterion (2) is suggested as an alarm limit together with the limit on longitudinal load imbalance of 1:3. This criterion is believed to be a better criterion than the criteria that limit one load imbalance at the time as it was found to be better at separating the derailing from the non-derailing vehicles in this study.

For tare state wagons it is recommended that a diagonal load ratio above 1:1.3 should result in a chassis inspection and that wagons with a diagonal load ratio or axle imbalance ratio of 1:1.7 should be stopped as these ratios of load imbalance are on the derailment limit in simulations.

These limits do not consider the measurement accuracy of the axle load check points which needs to be accounted for. Especially the axle load imbalances are uncertain as high dynamic ratios caused by wheel defects could be difficult to separate from the quasi static values as discussed by UoH in chapter 3 of D3.2. If imbalance quantities are calculated for the whole vehicle, they are less sensitive to measurement noise and high dynamic loads from individual wheels as they include information from many wheels that create an averaging effect.

It is recommended that derailment studies are conducted for more freight vehicle types. Only Y25-based vehicles were studied in D-rail WP3.2.

3.3.2 Potential commercial impact

The commercial impact will depend on the enforcement strategy used. If rather lenient alarm limits for load imbalances are introduced at first together with warning levels at which the train operators is informed about the status of their vehicles, an awareness of the issue can be created. If train operators know that limits are enforced, they will probably make larger efforts to make sure that their vehicles comply with those rules. Then limits could be tightened over time. In this way the commercial impact counted as the number of stopped trains could be rather limited, but it might require additional efforts for train operators to ensure rule compliance. It is however expected that some extra care when loading vehicles and containers isn’t vastly expensive. The benefits expected are a lower derailment frequency when the most derailment prone vehicles are removed from traffic and less track deterioration as extreme individual wheel loads are mitigated.

3.4 Future Work

The parameter studies on track irregularities show that the simulated Y/Q-ratios are non-linear functions of the track parameters where there can be significant interaction between parameters. Especially a large interaction was found between track twist and the lateral alignment. These results suggest that track maintenance criteria could be a function of several track parameters instead of considering parameters one by one. A much more comprehensive study than the one performed here would be needed to confidently develop such limits.

Simulations show that traffic in a small radius turnout generates larger Y/Q-ratios than a turnout with larger radius for typical speed limits. These speed limits are based on the nominal uncompensated lateral acceleration for a point mass approximation of the vehicle and does not account for bogie yaw stiffness, rotational inertia and other relevant effects. It is therefore suggested that simulation models and track force measurements can be used to find speed limits that provides consistent track loading at different curve radii.

It was found that the risk of derailment is larger in a curve featuring switch rail geometry compared to a plain line curve, everything else being equal. Future studies could investigate the actual track geometry quality found in turnouts and compare it to plain line.

3.5 Load Ratios (Appendix)

3.5.1 Nominal load ratios

Typical loading guidelines for freight vehicles such as those of UIC (RIV limits) [4] limit the longitudinal and lateral offset for payload centre of mass. The offset limits are determined through simple studies of static equilibriums such as those presented in Figure 1 for a vehicle with two bogies and four axles which is the vehicle type of interest for the present study. In the longitudinal direction the full vehicle is considered while in the lateral direction half a vehicle is considered.

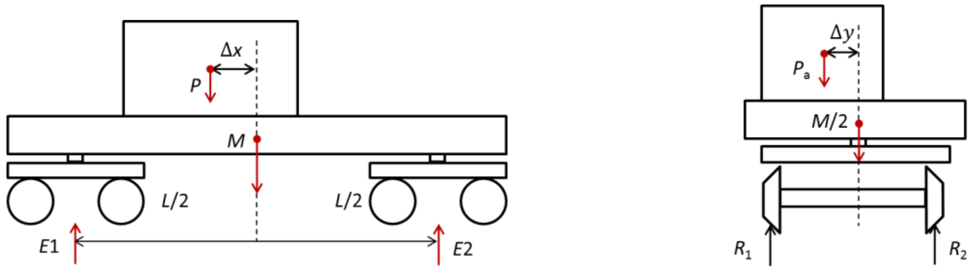

Figure 1 Illustrations of modeling for calculations of skew loading. Longitudinal (left) and lateral (right)

In Figure 1 the vehicle mass M is the mass of the tare state vehicle such that $M = E_1 + E_2 - P$. P is the full payload while P_a is the part of the payload for the studied half of the vehicle. $E_{1,2}$ are the bogie loads and $R_{1,2}$ the side loads for the studied half vehicle.

As there are only two unknown forces each in the studies of longitudinal and lateral load imbalance, the forces can be found using vertical force/load equilibrium and moment equilibrium.

The UIC loading guidelines then state that

$$\varphi_{\text{long,nominal}} = \max\left[\frac{E_1}{E_2}, \frac{E_2}{E_1}\right] \leq 3 \qquad (\leq 2 \text{ for two} - \text{axle vehicles}) \tag{3}$$

$$\varphi_{\text{lat,nominal}} = \max\left[\frac{R_1}{R_2}, \frac{R_2}{R_1}\right] \leq 1.25 \tag{4}$$

which imposes limits on the payload displacements Δx and Δy. As can be noted from the calculation set-up in Figure 1, only sums of wheel loads are included in the calculations as there isn't enough detail in the modelling to determine each individual wheel load. It can of course be assumed that the loads are distributed evenly, but such an approach doesn't account for e.g. non-linearities or compliance in the suspensions or any imperfections in chassis components that can be the case for real vehicles.

Especially for the lateral direction the observed lateral load imbalance in a bogie can be much larger than that predicted by the above model if there is a twist in the chassis which would apply a superimposed torque on the half-wagon model to the right in Figure 1. As long as any coupling torque between the vehicle halves isn't accounted for, it typically makes little difference for the offset limit whether the full or half vehicle is considered in the nominal calculations. If the payload is centred in the longitudinal direction it makes no difference. Also roll compliance in the suspension due to load offset can move the centre of mass of the vehicle and cause an increased load imbalance compared to the nominal.

It should therefore be recognised that even if a vehicle fulfils the nominal loading guidelines, it does not mean that the actual wheel loads that can be observed in track are also within those limits.

3.5.2 Observed load ratios

In this section vehicle load ratios for the longitudinal, lateral and diagonal direction will be defined. The ratios are based on observed vertical wheel loads that can come either from simulation or wheel load checkpoints. The longitudinal load ratio is defined as the sum of all vertical wheel loads for the bogie carrying the largest load divided by the corresponding sum of the bogie carrying the smaller load. As it is difficult to know which bogie carries the largest load beforehand, both versions of the ratio are calculated and the maximum value is used. Mathematically this can be written as

$$\varphi_{\text{long,observed}} = \max\left[\frac{Q_{21l}+Q_{22l}+Q_{21r}+Q_{22r}}{Q_{11l}+Q_{12l}+Q_{11r}+Q_{12r}},\frac{Q_{11l}+Q_{12l}+Q_{11r}+Q_{12r}}{Q_{21l}+Q_{22l}+Q_{21r}+Q_{22r}}\right] \qquad (5)$$

where $Q_{xy\,l/r}$ are the estimated quasi static wheel loads, index *x* the bogie number and index *y* the wheelset number within a bogie. *l/r* means left or right.

In the same way a lateral load ratio can be defined as

$$\varphi_{\text{lat,observed}} = \max\left[\frac{Q_{11r}+Q_{12r}+Q_{21r}+Q_{22r}}{Q_{11l}+Q_{12l}+Q_{21l}+Q_{22l}},\frac{Q_{11l}+Q_{12l}+Q_{21l}+Q_{22l}}{Q_{11r}+Q_{12r}+Q_{21r}+Q_{22r}}\right] \qquad (6)$$

It can be noted that $\varphi_{\text{long,observed}}$ and $\varphi_{\text{lat,observed}}$ are statically determined in the sense that for a given centre of mass position of the vehicle, the sum of torques about the centre of mass must be zero in both the lateral and longitudinal direction for the vehicle to stay at rest. This means that the load ratios observed correspond to the actual load ratio of the vehicle within the tolerances that can be expected due to e.g. asymmetric contact point conditions that can affect the length of the vertical wheel load levers about the centre of mass. The centre of mass position of the vehicle itself isn't statically determined however as it depends on e.g the state of the suspension.

For a mechanical system the term "statically determined" means that the forces and moments required for static equilibrium can be determined directly from static force and moment equilibrium equations. If the number of unknowns is larger, the deformation of the constituents of the system needs to be accounted for to obtain the equilibrium forces.

It should be noted that even if the sum of forces on each side of the vehicle and within each bogie of the vehicle are determined by the vehicle's load distribution, the individual wheel forces are not. Due to friction, suspension non-linearities, geometrical imperfections, load history etc. the load can be unevenly distributed between individual wheels and can also vary between different measurements. This makes measurements of the vehicle load ratios necessary if skew loading is to be estimated. It is not sufficient to study the load ratio of individual axles to draw conclusions about the vehicle's pay-load distribution.

In addition to the longitudinal and lateral load ratios, a diagonal load ratio which compares the sum of loads on the left front and right rear wheels to the loads on the right front and left rear wheels or vice versa is defined.

$$\varphi_{\text{diag,observed}} = \max\left[\frac{Q_{11r}+Q_{12r}+Q_{21l}+Q_{22l}}{Q_{11l}+Q_{12l}+Q_{21r}+Q_{22r}},\frac{Q_{11l}+Q_{12l}+Q_{21r}+Q_{22r}}{Q_{11r}+Q_{12r}+Q_{21l}+Q_{22l}}\right] \qquad (7)$$

The diagonal load ratio can be used to observe asymmetries in a vehicle such as a twisted chassis. The diagonal load ratio is statically undetermined and cannot be obtained directly from the payload distribution of the vehicle.

Even though it is difficult to draw conclusions about the exact skew loading state of a vehicle based on measured load ratios for individual axles, it is still a load imbalance measure that can be highly relevant for detection of derailment propensity. The observed axle load imbalance can be defined as

$$\varphi_{\text{axle,observed}} = \max\left[\frac{Q_l}{Q_r}, \frac{Q_r}{Q_l}\right] \quad (8)$$

As there are several axles in a vehicle, it is also useful to define the maximum axle load imbalance as

$$\varphi_{\text{axle,observed,max}} = \max\left[\frac{Q_{11l}}{Q_{11r}}, \frac{Q_{11r}}{Q_{11l}}, \frac{Q_{12l}}{Q_{12r}}, \frac{Q_{12r}}{Q_{12l}}, \frac{Q_{21l}}{Q_{21r}}, \frac{Q_{21r}}{Q_{21l}}, \frac{Q_{22l}}{Q_{22r}}, \frac{Q_{22r}}{Q_{22l}}\right] \quad (9)$$

3.6 References

[1] B. Pålsson, **Towards optimization of railway turnouts**, Licentiate Thesis, Department of Applied Mechanics, *Chalmers University of Technology*, Göteborg, Sweden, 2011

[2] UIC Code 716 R

[3] Spårväxel-Normalvärden och toleranser, Standard BVS 1523.004, Trafikverket (Swedish Railway Administration), Sweden, 2010

[4] RIV Loading Guidelines - Part 2: UIC, Paris, 01.01.1998

4 Derailment due to wheel failures

4.1 Studied mechanisms and investigated parameters

Thermal loads and mechanical loads can each cause fatigue damage and cracks to grow in wheels. Basically three scenarios for derailment have been studied in D-RAIL:

1. Wheel failure due to (excessive) tread braking
2. Wheel failure due to (excessive) wheel-rail contact forces
3. Wheel failure due to combined damage from tread braking and wheel-rail contact

Risks of fatigue and fracture have been assessed for two generic wheel designs where one represents the classical freight wheel with a slightly S-shaped web and one is a so-called low-stress wheel specifically designed for high thermal loads with a more elaborate shape of the wheel web. The wheel designs are studied for new and for fully worn down rims.

The aim is to find limiting parameters for both the loading of the wheels and for allowable cracks in wheels. The studied wheel designs (described above) have been chosen with aim to cover most wheels that are in use in Europe.

The evaluation of fatigue in the wheel web due to loads induced at the wheel-rail contact basically follows the standard EN 13979-1. In the standard, a set of fatigue loads and a method for assessing fatigue due to the stresses induced by the loads are given. The magnitudes of vertical and the lateral loads have been varied also outside the limits outlined by the EN standard.

Severe drag braking has been studied by use of a thermal model developed within CHARMEC for analysis of wheel and brake block temperatures. The rail chill is implemented as heat conducted from the hot wheel to the cold rail at rolling contact. The temperature field in a wheel during braking causes axial deflection of the wheel rim (in relation to the hub), which may cause a permanent change of the wheelset gauge. It may also induce high tensile residual stresses in the wheel rim after subsequent cooling. This may lead to initiation and growth of transverse cracks in the wheel rim. Sequentially coupled thermal-stress analyses have been performed by first solving the pure heat transfer problem. The temperature solution is then used in the stress analysis as a predefined field. Resulting stresses and displacements have been studied also for drag braking power levels higher than specified by standards. Two different material models with temperature dependent parameters are used for analysing the wheel behaviour at braking: One linear kinematic model and one novel (advanced) model that incorporates viscoplasticity and a combination of nonlinear isotropic and kinematic hardening. The viscous response of this model is primarily included in order to capture the observed relaxation of the material at elevated temperatures. The nonlinear hardening relations also include time recovery effects in order to capture slow processes (diffusion dominated) in the material leading to a decreased hardening. Using the advanced material model, also wheel behaviour for multiple braking cycles is studied to evaluate stability of wheel stresses and displacements.

The combined fatigue damage for the wheel web has been analysed from mechanical wheel/rail contact forces (as specified by the EN 13979-1) and by thermomechanical loading induced by drag braking and stop braking. It is assumed that the wheels cool down between brake cycles. In addition, to look at the influence of consecutive stop brake cycles also a braking load case that consists of two consecutive stops with time in-between only to allow for train acceleration is assessed. The damage is evaluated using a Coffin-Manson approach together with Palmgren–Miner damage accumulation. The damage from braking load cases is based on the stress–strain cycles during the fourth simulated brake cycle to allow the mechanical response to stabilize. The calculated lives of the wheels are determined for operational cases considering fully loaded wagons and for several prescribed drag and stop braking load cases. Positions of critical material points with respect to life are determined and the magnitudes of partial damage from the different load cases are studied. Moreover, in order to study the influence from an assumed deteriorated web surface as caused by e.g. severe corrosion or a surface scratch, it has been assumed that the fatigue limit is reduced. This reduction is assumed to affect mainly the high cycle fatigue life (by reducing the fatigue limit), whereas the (very) low cycle fatigue life is basically unaffected.

Analyses of the influence of track brakes, used on shunting yards, have been carried out for one wheel type. In the study, the combined damage on the wheel web is calculated from conventional wheel/rail contact forces (as specified by the EN 13979-1), unconventional forces induced by track braking and by thermomechanical loading induced by drag braking and stop braking. The damage is evaluated using a Coffin–Manson approach together with a Palmgren-–Miner damage accumulation. Track brake of two types is studied, one acting on both wheels of the wheelset and one track brake which acts on only one wheel in the wheelset.

To evaluate the wheels' susceptibility to crack initiation and growth, additional analyses were carried out. Circumferentially oriented elliptical cracks in the wheel web were presumed. Subsequent stress intensity factors and crack growth owing to mechanical loads were studied.

4.2 Main conclusions from numerical simulations

4.2.1 Main influencing parameters

Fatigue at mechanical loading has been studied for the two types of wheel designs. An assumed increase of the vertical loading (from 1.25P[1] to 2P or 3P), when the train is rolling on straight track, results in a minor increase in fatigue loading of the wheel web. However, if it is assumed that the increase in the vertical load also interacts with lateral loads at curving and negotiation of points and crossings, a quite substantially increase in the fatigue stresses can be seen at the transition between web and rim for the low-stress wheel design.

For the thermomechanical load case it is shown that there is a substantial difference between wheels with a slightly S-shaped web and a low-stress wheel. For single brake cycles at high power, all wheels have axial displacements even that are within requirements at the highest studied power level (80 kW). Moreover, for the residual axial displacements after braking, too high values are only obtained for the worn low-stress wheel. On the other hand, the low-stress wheel shows residual stresses that are within the regulations also for the worn wheel, whereas the slightly S-shaped shows too large stresses that can results in global wheel fracture. When analysing single brake cycles using the more advanced viscoplastic

[1] Wheels are designed according to the standards EN13979-1 with a vertical load of 1.25P, where P is the static load on the wheel

material model, it is found that all wheels except the worn slightly S-shaped wheel has axial rim displacements that are in line with the standards. However, the worn low-stress wheel shows too high residual displacements of the wheel rim for powers higher than 60 kW. Similar to the analyses with the linear kinematic model, the slightly S-shaped wheel has residual stresses in the rim that are lower than required by the standards also at the highest studied power level of 80 kW. The worn low-stress wheel shows too high residual stresses in the wheel rim already at 60 kW.

The wheels have also been assessed for multiple brake cycles (with intermediate cooling to ambient temperature) using the viscoplastic material model. These analyses confirms the results from the single brake cycle with respect to residual stresses after braking; the slightly S-shaped wheel does not fulfil the requirements at 60 kW while the low-stress wheel fulfils the requirements also at the highest tested brake power (80 kW). When it comes to the residual axial flange deflection, it is found that the low-stress wheel shows a substantial global ratchetting behaviour in the axial direction for power levels of 60 kW and higher. For each brake cycle, an additional displacement increment is then added to the total axial flange deflection. This makes the wheel rim gradually move outside of allowed limits (towards the field side). At the same time, the stresses in the wheel rim increase, but are kept within the allowed limits. For the slightly S-shaped wheel the assumptions made after single brake cycles still hold with only minor differences (in residual stress levels and displacements) as compared to when analysing single cycles.

The analyses of wheel web life with respect to combined mechanical loads and loads from tread braking generally show long lives (several millions of kilometres) for the assumed loading. The loading spectrum of the wheels is based on a train running for 300 000 km with 94% straight track, 5% curves and 1% switches / crossings, with additionally 10 high power drag cycles and 10 000 stops (one stop every 30 km). It is found that the lives of the low-stress wheel and the slightly S-shaped wheel are limited by mechanical fatigue. The largest damage levels occur in the web close to the hub. However, worn slightly S-shaped wheels are limited in life by a combination of damage originating from mechanical fatigue and thermomechanical fatigue from braking. The largest damage is then found to occur towards the wheel rim. Depending on the assumed braking load cases, thermomechanical fatigue (from braking) contributes with up to 65% of the total damage. It should here be noted that the braking load cases have been considered to occur separately (i.e. intermediate cooling to ambient temperature) but assuming severe braking load cases (drag braking up to 70 kW and stop braking as severe as resulting from braking 25 tonnes axle load at 1 m/s^2 from 120 km/h). In conclusion, the analyses of damage for combined mechanical loads (with forces in accordance with the standard EN 13979-1) and thermomechanical loads from braking are not restricting for wheel lives even when considering severe stop braking. Even the calculated lives of the worn wheels are higher than the expected total maximum service life of freight wheels (then having diameters ranging from new down to worn state).

Two cases of assumed deteriorated web surface (severe corrosion or surface scratch) that lead to fatigue limit reductions have been studied. In the first case a reduction by 25% is studied (corresponding e.g. to a change in surface roughness R_a=6.3 µm to 40 µm) and in the second case a reduction by 50% is studied (corresponding e.g. to a change in surface roughness R_a=6.3 µm to a non-machined forged surface appearance). These reductions are presumed to modify the high cycle fatigue life, while leaving the low cycle fatigue life mainly unaffected. The result is a dramatic reduction in calculated wheel lives. For the 25 %

reduction, the calculated wheel lives are still longer than about 800 000 km. For the 50 % reduction, the wheel lives now range from 90 000 km up to 150 000 km for all studied geometries. It can be concluded that the surface finish / appearance of the web is of utmost importance for wheel lives. Moreover, in the future, when stress optimized wheels will be more common these results indicate that wheel webs should be checked regularly at maintenance to avoid surface deterioration that substantially lowers the fatigue limit. An intact wheel web paint system for the duration of the wheel life is hence essential not to increase the risk of fatigue damage.

The wheel web life has also been analysed when assuming that each stopping event consists of two consecutive stop brake cycles (time between for train acceleration only) at each stop-braking event in the load spectrum. For this case, the resulting strain amplitude in the web is almost twice the one given by a separate stop. For this (degenerate) case it is found that the lives of the new wheels still are controlled by the mechanical fatigue, but that the lives of both wheel designs in the worn states are controlled by predominantly stop braking damage. However, even when considering double stops, the calculated lives of the wheels is still of the wheels are still more than one million km.

The analyses of the wheel life when also considering loads from track brakes show that the wheel lives are controlled by a combination of mechanical fatigue from conventional load cases and thermomechanical fatigue induced by tread braking. Here it should be noted that fairly short wheel lives are calculated, which is caused by the assumption that *three* consecutive stops are considered in these analyses. Nevertheless, the lives are only to a negligible degree controlled by the non-conventional mechanical loading cases that are induced by the track brakes. These findings indicate that there is no need to add loads from track brakes to the design load cases specified in standards. The study of wheel web cracks indicates that for normal running conditions crack growth rates are very low with stress intensity ranges only marginally above the crack growth threshold. When a crack continues to grow, the growth rate remains low and a substantial widening of the crack (in the circumferential direction) is required to for the crack to grow deeper into the web. Furthermore, it has been established that wheel web crack growth rates is fairly insensitive to increases in vertical and lateral loading.

A second type of wheel failure that may result in derailment is subsurface initiated RCF fractures. This has been previously studied see e.g. [1]. For this case, the main affecting parameters are vertical load magnitude (with a high influence also of high frequency content, see e.g. [2]), contact geometry, material defects, and contact close to the field side.

Further details are provided in [3].

4.2.2 Tentative limit values

Wheel web fracture

Concerning increased vertical loading at a lateral position corresponding to the rolling circle the risk of mechanical fatigue is low. However, the low-stress wheel design is sensitivity to large tread damage resulting in high vertical loads with an application point towards the sides of the wheel tread. For such a case fatigue of the wheel web could be an issue. However, to limit such loads more than vertical loads as generated at the rolling circle seems unrealistic. In general, no further limitations are required for such vertical loads than what is required for the safety of the rail. In detail, an assumed increase in vertical loading at the rolling circle of the two wheel designs does not increase the maximum fatigue stresses until the load is higher than three times the static wheel load. For an axle load of 25 tonnes, a vertical load of three times the static axle load corresponds to 370 kN, which is higher than the maximum allowed vertical load proposed for avoiding rail damage of 350 kN (or even down to 250 kN at very cold conditions), see section 5.

Note however that the risk of damaging bearings has not been considered in the current analysis. The reason is that the main influence of a damaged bearing would be increased heating leading to an axle fracture, a topic which is studied in the EURAXLES project and thus explicitly excluded from D-RAIL.

Analyses of severe drag braking (power levels up to 80 kW with a duration of 45 min) show that displacements during and after braking and build-up of residual stresses can be limiting for the wheels. The two studied different wheel designs have different types of response to high power drag braking. For worn slightly S-shaped wheel too high residual tensile stresses occur for power levels larger than 60 kW. For the low-stress wheel residual displacements become too high at power levels 60 kW and higher. For this case, the analyses also indicate incremental growth of the axial flange deflection of the wheel rim, while residual stress levels are still within allowed limits.

Subsurface initiated rolling contact fatigue

To keep the risk of subsurface initiated low, the operational scenarios should fulfil the requirement (cf [1])

$$FI_{\text{sub}} = \frac{F_z}{4\pi ab}\left(1+f^2\right) + c_{\text{dv}}\sigma_{\text{h,res}} < \sigma_{\text{e,dv}} \qquad (4.1)$$

where F_z is the normal force in the wheel–rail contact, a and b the semi-axes of the hertzian contact patch, f the traction coefficient (lateral load divided by normal load). Further, c_{dv} is a material parameter, $\sigma_{\text{h,res}}$ the hydrostatic part of the residual stress and $\sigma_{\text{e,eq}}$ the equivalent fatigue limit accounting e.g. for the effect of material defects.

For practical purposes, equation (4.1) can be simplified to

$$FI_{\text{sub}} = \frac{F_z}{4\pi ab} < \sigma_{\text{e,dv}} \qquad (4.2)$$

Where $\sigma_{\text{e,eq}}$ can be approximated to around 220 MPa. Note that F_z needs to account for load frequencies up to around 1 kHz, cf [2].

4.2.3 Potential preventive measures

Wheel web fracture

Limitation of braking power or excessive wheel temperatures cannot be handled by way-side monitoring since the brake cycle can be of relatively short duration. Hence, on-wagon monitoring is required to assess problems with build-up of too high residual stresses (slightly S-shaped wheel design) or too high residual axial displacements of the wheel rim (low-stress wheel design). The two studied designs have the same limiting brake power and hence also the same limiting temperature level. Classification of wheels according to their handling of different time–temperature scenarios could be a way forward. In the present study, the power limits for a 45 min braking duration have been assessed and for this case a bulk rim temperature limit of 550 ^{o}C corresponds to the limiting power level 60 kW. For a specific location of wheel temperature detection appropriate temperature limits can be defined. These wheel temperature limits might alternatively be “translated” into brake block temperatures for a specific block material.

A different type of preventive measure is to avoid having wheels in service that have been subjected to severe braking. This study implies that such wheels can be detected either by measuring residual stresses in the wheel rims (potential build-up of tensile stresses for slightly S-shaped wheel designs) or by measuring of the gauge of the wheelsets (low-stress wheel designs).

Furthermore, the study highlights the importance of proper maintenance of wheels, both regarding wheel tread damages (causing high wheel–rail contact forces) but also the increased risks of wheel web damage if allowing larger web surface deterioration that substantially lowers the fatigue limit. The management of the latter aspect is straightforward since in general 1) wheel webs are not prone to surface damage in the form of mechanical impacts / scratching and 2) protection from corrosion is given by a proper paint system.

Subsurface initiated rolling contact fatigue

Long-term exposure of wheels to excessive vertical load magnitudes and poor contact geometries should be avoided. The former is obtained mainly by avoiding high-frequency contributions from out-of-round wheels and/or corrugated rails. The latter is obtained by primarily by having (and maintaining) matching wheel and rail profiles. Note that as axle loads and speeds increase, the margins for poor operational conditions decrease.

4.3 Recommendations and concluding remarks

4.3.1 Recommended actions

Operational measures

- Wheels with excessive tread damage should be mitigated swiftly (naturally this not only relates to the risk of wheel failure, but also e.g. to rail breaks and bearing failures).
- Successive excessive brake cycles should be avoided.
- Wheel and rail maintenance should be performed as to avoid excessive wheel out-of-roundness and rail corrugation.
- Wheel treads should be manufactured so as to assure that no larger material defects are present.

Monitoring measures

- Implementation of on-wagon measurement of wheel or block temperatures for giving indications / warnings of tread brake malfunction. Such systems would only be required for vehicles not equipped with brake blocks with a "thermal fuse function" that protect the wheels by limiting wheel temperatures. Newly designed vehicles should always be equipped with such blocks, but older designs will normally continue to operate with their original set-up.

- Wheel–rail contact loads should be monitored. Since higher vertical loads can be allowed with respect to wheel fatigue than with respect the rail damage (and most likely bearing damage), the values given in Section 5 are sufficient.
- The wheel tread and rail head geometries should be monitored to assure conformity. This can be done either continuously in operation (e.g. laser measurements) or during inspections (workshop or track inspections). Usually wear is a fairly slow process, which means that in most cases the latter version should be sufficient.
- The absence of larger material defects should be assured by inspections during manufacturing.
- The rail / wheel gauge (and also the wheel/rail profiles, see above) need to be monitored to avoid contact close to the field side of the wheel tread.
- Detection of corrugated rail surfaces and out-of-round wheels.

Limiting values

Different limiting temperatures can be chosen for different classes of wheels depending on their capability to handle high brake power and temperature differences within the wheel. On-board measurement of wheel temperatures could be limited to about 550 ^{o}C (rim bulk temperature at centre of rim) or could possibly be reassessed for an allowed even higher temperature (at the surface of the wheel rim) in a chosen actual position for temperature measurement.

For subsurface initiated RCF, limiting parameters are defined by a limit of the magnitude of FI_{sub} ≤ 220 MPa, see [1, 2].

4.3.2 Potential commercial impact

Implementation of on-wagon temperature warning systems can substantially increase the derailment safety of freight vehicles equipped with general brake blocks (mainly older wagons). However the action is also fairly costly.

Limiting the vertical loading is already proposed in order to avoid rail breaks. Thus, there will be a synergy effect that increases the value of such detectors.

Regarding RCF-related fractures, most of the proposed monitoring actions are already employed for other reasons (e.g. track geometry measurements, defect detection of new wheels, some wheel load measurements, profile measurements in workshops and during rail grinding). Further monitoring actions will of course carry an additional cost. However the benefits are not confined to derailment prevention. As an example, corrugation identification from measurements of high-frequency contact loads will aid also in combating noise pollution.

Further, there is a major commercial impact in that D-RAIL aids in applying physically sound limit values e.g. on wheel temperatures, allowed load magnitudes etc. This will lead to savings due to a combination of decreasing risk of derailment (and deterioration, noise pollution etc), less unneeded operational disruptions and lower maintenance cost. Tentative magnitudes of such savings can be estimated from the evaluations made in WP1 and WP7 of D-RAIL.

4.4 References

1. Anders Ekberg, Elena Kabo & Hans Andersson, **An engineering model for rolling contact fatigue**, *Fatigue & Fracture of Engineering Materials & Structures*, vol 25, no 10, pp 899--909, 2002
2. Jens C O Nielsen & Anders Ekberg, **Acceptance criterion for rail roughness level spectrum based on assessment of rolling contact fatigue and rolling noise**, *Wear*, vol 271, no 1–2, pp 319–327, 2011
3. D-RAIL, **D3.2: Analysis and mitigation of derailment, assessment and commercial impact**, *in preparation*, 2013

5 Derailment due to rail failures

5.1 Studied mechanisms and investigated parameters

The study focuses on rail brakes. Primarily rail breaks owing to cracks originating from the gauge corner of the railhead (head checks) and from the edge of the rail foot have been investigated. The main conclusions should however (perhaps with some additional analyses) be roughly transferable to other types of cracks in the railhead (e.g. squats) and the rail foot (centre cracks).

As for other rail cracks (see e.g. [1]), such as cracks emanating at the rail web, from bolt holes at insulated joints, from welds etc, caution should be taken. The reason is that such defects are strongly related to local conditions (e.g. the run-down of a joint, the quality of a weld etc). Thus, the formation and growth of these cracks will depend more on these local conditions, then on global conditions (wheel impact load, track stiffness etc).

The analysis has set out from the presumption that:

- A rather large initial crack exists either in the rail head or the rail foot
- The crack is sufficiently large so that global bending (and uniform tension due to global thermal loading) will be the dominating crack driving force.
- Linear elastic fracture mechanics is valid, i.e. fracture is presumed when $K_I \geq K_{Ic}$, where K_I is the stress intensity owing to rail bending and uniform thermal loading and K_{Ic} is the fracture toughness of the rail (here taken as $K_{Ic} = 40$ MPa√m). Crack growth rates until fracture can be estimated using Paris law.

5.2 Main conclusions from numerical simulations

5.2.1 Main influencing parameters

The main influencing parameters are

- Wheel/rail contact load
  The vertical load is the main parameter in the propagation and fracture of the larger rail cracks, which have been studied here.
- Material defect sizes and locations
  “Material defect” here mainly implies defects that have formed during operation and handling of the rail, e.g. initiated head checks and check marks / corrosion pits on the rail foot.
- Rail temperature
  The key parameter here is the difference between rail temperature and the stress free temperature that will result in a tensile (or compressive) stress in the rail. Note that the (absolute) temperature may also alter the mechanical characteristics of the track (and the vehicles).
- Track stiffness
  This includes the global stiffness along the track, as well as the influence of local stiffness, e.g. hanging sleepers.

5.2.2 Tentative limit values

Taking the wheel/rail contact load as the primary parameter, and other parameters as "bad cases" (meaning values that will promote rail breaks, but are not as high as to be very rare), reasonable limit values of wheel/rail contact forces will be in the order of 350 kN down to around 250 kN for very cold conditions.

Corresponding crack sizes at fracture are in the order of 25 mm for a rail head crack and 5 mm for a rail foot crack.

The limit values account for the influence of hanging sleepers and not too soft ballast stiffness (presumed to be ≥ 30 MN/m per half sleeper).

5.2.3 Potential preventive measures

The most efficient preventive measure is to either prevent the formation of rail cracks (which in the case of head cracks is likely to be unrealistic) or to identify and remove rail defects before they are large enough to potentially cause a rail break.

Note that a large derailment risk usually corresponds to multiple rail breaks at a rather closely confined stretch of the track (in the order of a meter). This risk is larger for head cracks than for foot cracks since headcheck crackss have a tendency to form in clusters at highly loaded stretches of the track (usually in curves). Squats and similar crack forms are in this aspect usually in-between these two extremes.

Note also that if the rail break occurs in a signal rail (which is a 50% chance for a track with rail circuits), it will be detected as a loss os rail circuit. Thus, if rail circuits are removed (e.g. due to a change of signalling system) there is a need for improved / more common inspections.

Crack growth rates, and corresponding inspection intervals, can be estimated as outlined in [2].

5.3 Recommendations for continued studies in D-RAIL

5.3.1 What needs to be detected / measured / monitored?

- Wheel/rail contact loads
  - peak load
  - time series

- Material defect sizes and locations
  - rail head
  - rail foot
  - other cracks – not included in the analysis, but potential safety risks

- Rail temperature
  - preferably lowest temperature between to wheel load detectors
  - good prognosis important to plan for decreases in allowed impact loads

- Track stiffness
    - average stiffness
    - hanging sleepers

In addition, points on the line with increased load magnitudes and/or stress concntrations, e.g. insulated joints, welds, transition zones and switches and crossings should be subjected to additional monitoring.

5.3.2 Which measurement / monitoring accuracy is needed?

- Wheel/rail contact loads

  In general a reasonable accuracy would be in the order of 10 kN. However there are some potential pitfalls:
  - systematic errors should be avoided
  - measurements where trains are stopped will cause major costs; a lack of accuracy in such measurements may lead to legal complications
  For these reason, the accuracy may need to be higher.

- Material defect sizes and locations

The derived limits correspond, as mentioned, to a crack size of 25 mm for a rail head crack and 5 mm for a rail foot crack. However crack sizes that need to be identified are smaller since it must be assured that cracks do not grow to failure in-between inspections. As noted above, head cracks have a tendency to continuously form.

The size of an initial, small crack that can be identified **with certainty** will have a very high influence on evaluated inspection intervals.

The **certainty** of finding **large** cracks is crucial from a safety perspective and allowed wheel loads. Here the accuracy in size is not as important.

Thus, defect size measurement accuracy is in the order of one millimetre for initial, and in the order of 5 mm and 2 mm for large rail head and rail foot cracks, resepctively. The reliability must be very high (i.e. the detection failure rate must be very low).

- Rail temperature

A reasonable measurement accuracy is in the order of some 5°C. Note that this corresponds to the coldest section in-between to wheel load detectors. Further, there is a need to carry out prognoses some day(s) in advance to facilitate logistics.

- Track stiffness

The important features are to be able to identify (and mitigate) very soft tracks, and to identify (and mitigate) hanging sleepers.

5.3.3 Tentative limit magnitudes

Wheel impact loads in the order of 350 kN down to around 250 kN for very cold conditions.

Reasonable crack sizes that should be detected (and mitigated) are for a rail head crack some 2 mm for an initial crack and 2 cm for a final crack; and for a rail foot crack some 1 mm for an initial crack and 5 mm for a final crack.

5.3.4 How measured data should be employed in operational control and maintenance planning?

Required crack inspection intervals and resulting maintenance procedures need to be established based on results in the analysis. Further, communication and mitigating actions at high wheel loads (also below alarm limits) need to be established and harmonized (in the framework sense) across Europe. This is dealt with in the UIC-project HRMS.

5.3.5 Parameters / scenarios suitable for test investigations / validations

Test data on wheel impact loads exist in the literature (see e.g. [3]). Simulations have been calibrated / validated with these.

Test regarding relation between lateral forces and lateral bending in curves would be valuable. This is planned for D-RAIL WP6.

5.3.6 Potential commercial impact

Regarding potential reductions in derailment costs, please refer to [4, 5]. Generally wheel load detectors have additional value in that they can be used for operational monitoring (e.g. of amounts of transported cargo), vehicle maintenance planning etc. Also track stiffness measurements have synergy effects in that data can be used for maintenance planning. Temperature measurements can be employed also for other types of meteorological prognoses.

5.4 References

1. UIC, **UIC Code 712R: Rail defects**, 4th Edition, January 2002.
2. Anders Ekberg, Elena Kabo & Jens C O Nielsen, **Alarm limits for wheel–rail impact loads – part 2: analysis of crack growth and fracture**, *Chalmers Applied Mechanics*, Research report 2009:03, 53 pp, 2009
3. Jens C.O. Nielsen, **High-frequency vertical wheel–rail contact forces—Validation of a prediction model by field testing**, *Wear*, vol 265, no 9–10, pp 1465–1471, 2008
4. D-RAIL, **D1.1: Summary report and database of derailments incidents**, 71 pp + 1 annex (3 pp), 2012
5. D-RAIL, **D1.2: Report on derailment economic impact assessment**, 27 pp + 3 annexes (3+1+2 pp), 2012

6 Concluding remarks

In D-RAIL WP1, top derailment causes (regarding cost and impact) were identified as:

1. Axle ruptures
2. Excessive track width (section 2, 3)
3. Wheel failure (section 4)
4. Skew loading (section 2, 3)
5. Excessive track twist (section 2, 3)
6. Track height/cant failure (section 2, 3)
7. Rail failures (section 5)
8. Spring & suspension failure (section 2, 3)

The current deliverable contains a condensed report on investigations regarding all of these with the exception of axle ruptures, which is the topic of the EC-funded project EURAXLES and thus explicitly excluded from the scope of D-RAIL. The list also indicates in which section of the current report the topic is dealt with.

In addition to the listed derailment causes, a dedicated study on derailment in S&Cs has been carried out (as reported in section 3). The main reasons are that the categorisation in the list above (and also existing derailment statistics) does not really reflect how many derailment that occur on line vs in S&Cs. Further major derailments in S&Cs tend to be costly and operationally disruptive since S&Cs are key points in the infrastructure. Finally S&Cs represent rather extreme load conditions for an operating train; consequently, if the train can pass a S&C without derailing, chances are high that it will not derail.

The current report is intended as a summary of the (very) detailed and technically much more elaborate report D3.2 – Analysis and mitigation of derailment, assessment and commercial impact. Thus the report contains brief descriptions of the analyses, summary of the main results and pertinent conclusions. This is followed by recommendations on how to employ the results and also a first evaluation of the commercial impact.

The recommendation in implementing the results of the current study is to start with this report. If further details are needed, deliverable D3.2 can be consulted. Further details is provided by the references of this report and of D3.2.