RIVAS
Railway Induced Vibration Abatement Solutions
Collaborative project

Guideline for Design of Vehicles Generating Reduced Ground Vibration

Deliverable D5.5

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1. EXECUTIVE SUMMARY

Based on the work conducted in RIVAS WP5, a guideline for the design and maintenance of railway vehicles leading to reduced generation of ground-borne vibration is presented.

For a given combination of train speed and axle loads, it is concluded that the **unsprung mass** and **wheel out-of-roundness (OOR)** are the key vehicle related parameters determining the generation of ground-borne vibration.

Guidelines on reducing the unsprung mass are presented, including alternative wheelset designs (optimisation of geometry, material and production process) and the design concept and suspension of the mechanical drive system. For a given combined level of wheel and track irregularities, reducing the unsprung mass leads to a reduction of vertical dynamic wheel/rail contact forces and ground-borne vibration in a wide frequency range. The unsprung mass of a powered wheelset consists of the wheelset mass and part of the drive system mass. A reduction of the wheelset mass will directly translate to the same reduction in unsprung mass, while the effect of altering the drive system design will depend on the drive suspension concept. Since the conceptual design of a bogie is carried out at an early stage in the vehicle design process, a low unsprung mass has to be a clearly defined design target from the beginning.

Based on an extensive field measurement campaign carried out in Switzerland, where the influence of several different types of vehicle on vibration levels was measured, it is concluded that the maximum vibration levels are induced by freight locomotives. For several freight locomotives, a high variance in measured vibration levels was observed indicating a significant scatter in wheel tread conditions (OOR). High vibration levels are also induced by trailer bogies with OOR (such as wheel flats). Single orders of wheel polygonalisation can induce large contributions to the vibration level in a narrow frequency range even if the track irregularity level is high. For many types of wheel OOR (wheel polygonalisation), the causes for the initiation and growth are unknown. The most commonly applied mitigation measure is therefore early detection of out-of-round wheels by a track based detection system, followed by wheel turning (reprofiling). Aspects of vehicle design, such as the selection of brake system and primary suspension, and/or tolerances in production and maintenance of wheels have an impact on the deterioration of the wheel tread, and hence an indirect influence on the excitation of ground-borne vibration.

Table 1.1 shows the potential reduction in ground-borne vibration level associated with reducing the vehicle unsprung mass and the wheel OOR level. The given values are based on results from field measurements, numerical simulations and a technology assessment carried out in RIVAS. It is concluded that locomotives, which can be equipped with heavy unsprung drives, have the largest potential of reducing the unsprung mass by an alternative vehicle design. Reducing the unsprung mass of a trailer bogie has a more limited potential and can only be accomplished by optimising the wheelset geometry and material. Locomotives have shown a larger measured spread in wheel tread conditions compared to trailer vehicles, which may be caused by high traction and braking forces. This implies that the potential improvement in terms of lower ground-borne vibration level by reducing the wheel OOR is higher for locomotives compared to trailer wagons.

Table 1.1. Possible reduction in ground-borne vibration level by reducing unsprung vehicle mass and wheel OOR

	Feasible reduction of unsprung mass	Vibration reduction due to lower unsprung mass	Possible vibration reduction by reduction of OOR
Trailer bogies	15 %	< 1 dB	< 5 dB
Powered bogie	25 %	1 – 2 dB	< 5 dB
Locomotives	35 %	2 – 4 dB	< 20 dB

To reduce ground-borne vibration induced by railway vehicles, the following measures on rolling stock need to be implemented:

- Network based monitoring stations for wheel tread conditions (OOR) allowing for condition based and prompt wheel maintenance.
- Improved brake system design, wheel slide protection and wheel material quality to avoid wheel flats and other discrete wheel tread defects.
- Reduction of unsprung mass, in particular for locomotives, by application of suspended drive design concepts.
- Radial steering of wheelsets to reduce wear and wheel polygonalisation in small radius curves.

These measures reduce ground-borne vibration by mitigation at source while reducing LCC of wheelset and track. Bogies with low unsprung mass and radial steering of wheelsets are also track friendly. In the UK, track friendly bogies are subjected to lower track access charges. Thus, the application of a more sophisticated bogie design and drive concept could be regarded as both a profitable and environmentally friendly approach.

Whenever modifying the vehicle design aiming to reduce ground-borne vibration, it must be ensured that the changes do not conflict with safety and vehicle dynamics performance.

2. TABLE OF CONTENTS

3. INTRODUCTION AND SUMMARY OF DELIVERABLES

The RIVAS project aims at reducing train induced ground-borne vibration by measures applied on track superstructure, ground and vehicle. Measures applied for the ground address the propagation of vibrations while measures applied on track superstructure and vehicle aim at controlling the excitation and the response of the vehicle-track system. The perceivable ground-borne vibration has a frequency content ranging from a few Hz up to around 80 Hz [1-3]. The ground-borne noise is containing frequencies in the interval 16 – 250 Hz and is generated by vibration propagating in the ground which is radiated as noise from for example the building walls [1].

The present deliverable concludes the work of RIVAS WP5. A brief summary of all of the other RIVAS WP5 deliverables is given below.

3.1 D5.1 AND D5.2

A state-of-the-art survey on the influence of rolling stock on induced ground-borne vibration was presented in RIVAS deliverable D5.1 [4]. The potential of optimising various vehicle design parameters to reduce ground vibration emissions was discussed. When introducing new vehicle designs and performance requirements, it was concluded that these must be achieved without jeopardising the compliance with other existing requirements. For example, bogie parameters such as wheelset mass, drive concept and suspension properties need to be designed in order to meet requirements on ride comfort and stability as well as on curving performance. Thus, when modifying the vehicle aiming to reduce ground-borne vibration it must be assured that such changes do not conflict with safety and the vehicle performance in other areas.

It was concluded in RIVAS D5.1 [4] and D5.2 [5] that unsprung wheelset mass and wheel out-of-roundness (OOR) are the key parameters in the generation of ground-borne vibration. Aspects of vehicle design, such as the selection of brake system and primary suspension, and/or tolerances in production and maintenance of wheels may have an impact on the deterioration of the wheel tread (generation and growth of wheel OOR) and hence an indirect influence on the excitation of ground-borne vibration.

3.2 D5.3

RIVAS D5.3 [6] introduced a database (part of the main RIVAS database set up in WP1) containing data from field measurements at the three test sites in Switzerland which were also assessed in D5.2. The database also includes wheel OOR data from 120 locomotives of two different types (Re620 and Re420).

3.3 D5.4

In RIVAS D5.4 [7], an overview of present knowledge on causes for OOR growth was presented. Various types of OOR were classified and discussed. For many types of OOR the causes are unknown and the most commonly applied mitigation measure seems to be early detection of out-of-round wheels followed by wheel turning (reprofiling). Potential mitigation measures include more wear resistant wheel materials, optimisation of brake systems, improved process for wheel profiling with stricter tolerances, and improved bogie steering.

Further evaluation of field test data from Switzerland showed that vibration spectra are dominated by two mechanisms: the sleeper passing frequency and the resonance frequency of the unsprung mass on the track stiffness (often referred to as the P2 resonance). It was shown that significant levels of ground vibration are generated by freight locomotives of the types Re420, Re474, Re620 and TRAXX F140. Several wheels on these locomotives have large levels of OOR although there seems to be a large spread between different wheels. The unsprung mass of TRAXX and Re420 is 4500 kg and 3200 kg, respectively. The Re420 has a semi-suspended drive. In contrast, the Re460 passenger locomotives do not generate severe ground vibrations related to wheel OOR while vibration generated at sleeper passing frequency are at levels comparable to the other locomotive types. The Re460 reaches long reprofiling intervals due to the slow OOR growth, has a fully-suspended drive and a significantly lower unsprung mass of 1900 kg.

Based on numerical simulations, it was concluded that the most effective means to reduce ground-borne vibration by vehicle optimisation is by minimising the unsprung mass as this increases the vehicle receptance in a wide frequency range. Reducing the stiffness of the primary suspension could have a positive effect in reducing ground vibration for soils where the free field mobility is high at frequencies near the wheelset resonance on the primary suspension. However, modifying the properties of the primary suspension and/or the wheelset and bogie distances may simply shift the problem of ground-borne vibration from one octave band to the next.

A technology assessment (including functional constraints and cost efficiency) of vehicle and wheelset designs was presented. Different means of reducing the unsprung mass were discussed, such as wheel design, axle design and the design concept and suspension of the mechanical drive system. Reduction of wheel OOR by wheel material selection, improved traction and braking control as well as considering the curving ability of the vehicle were suggested. The unsprung mass of a powered wheelset primarily consists of the wheelset mass and part of the drive system mass. A reduction of the wheelset mass will directly translate to the same reduction in unsprung mass, while the effect of the drive system will to a large extent depend on the drive suspension concept. Since the conceptual design of a bogie is carried out at an early stage in the vehicle design process, a low unsprung mass has to be a pronounced design target from the very beginning.

3.4 D5.6

RIVAS D5.6 [8] summarises the results from an extensive field measurement campaign performed in Switzerland. The test train was assembled to include a sufficient spread in unsprung mass and wheel OOR among the different wagons, such that the influence of these parameters could be observed in the ground-borne vibration measured at different vehicle speeds. Both wheel and rail irregularities were measured by direct methods.

It was concluded that both unsprung mass and wheel OOR have a significant influence on ground-borne vibration, and that a reduction in combined wheel/rail irregularity level by removing wheel OOR will roughly lead to the same level of reduction in ground-borne vibration. Furthermore, high OOR levels due to wheel defects on a locomotive were measured already a few months after wheel reprofiling. The measurements of wheel OOR showed differences of up to 20 dB between good and bad locomotive wheels.

Due to the scatter in wheel tread conditions, the analysis of the influence of unsprung mass was not as straight-forward. The measured vibration levels induced by two different types of freight bogie (with unsprung mass 1610 kg and 1060 kg) showed a difference of up to 10 dB. However, there was a simultaneous difference of up to 5 dB in wheel OOR level between the two bogies. By analysing both the vibration and OOR spectra, it was concluded that the influence of the present difference in unsprung mass was 2 – 4 dB. According to simulations reported in previous deliverables (for the corresponding difference in unsprung mass), the predicted effect was 3 dB which is in good agreement with the measurements.

4. GENERAL ASSESSMENT OF GROUND-BORNE VIBRATION INDUCED BY IN-SERVICE VEHICLES

Case 1

Problem description:

High levels of ground-borne vibration are induced by pass-bys of many different types of locomotive, wagon, coach and EMU.

Mitigation measure:

If a large variation of vehicle types generates high levels of vibration at a given site, mitigation measures on the vehicles will have a limited effect. Considering the wide spread in wheel tread conditions that can be expected for a mixed fleet of trains it is probable that the fleet includes some vehicles with good performance in terms of vibration emission. In this case, measures should rather be addressed to the track and/or the track substructure and surrounding soil. For frequencies below 10 Hz, it is primarily measures in the soil that may be effective, see design guideline of RIVAS WP4 [9].

Case 2

Problem description:

High levels of ground-borne vibration are induced by pass-bys of specific vehicle types, locomotives or wagons.

Mitigation measure:

The types of vehicles that give rise to high levels of vibration, and the types that do not, need to be identified. Then any significant difference between the two vehicle categories which could have an influence on the vibration generation should be determined. For vehicles inducing high vibration levels, three examples of feature to consider are:

- High *unsprung mass*, which is expected on locomotives and powered bogies depending on the suspension of the drive system, see Table 5.4 on page 21. The reduction of a high unsprung mass in an existing vehicle design is unfortunately not feasible to change by any measures. For further information, see Section 5.1.
- *Malfunctioning primary suspension* on a trailer bogie could lead to a higher unsprung mass. If the suspension is locked, parts of the bogie mass will be unsprung. A locked suspension is a maintenance issue and could be detected by the use of monitoring stations such as the wheel load checkpoints described in [5]. For further information, see Section 5.2.
- *Wheels in poor condition* due to wheel flats or high levels of OOR (wheel polygonalisation) may appear on individual wheels and bogies or could be found on all vehicles of the same vehicle type. For further information, see Section 6, Appendix A and the failure mode effect analysis presented in Section 4.2 of D2.7 [10].

5. VIBRATION MITIGATION BY VEHICLE DESIGN

5.1 UNSPRUNG VEHICLE MASS

5.1.1 Wheel and axle geometry design

Wheel

For a given steel grade, the mass of the wheel can be minimised by optimising its cross-sectional geometry using finite element (FE) models that are today considered reliable for simulation of structural mechanics. In the case of thermo-mechanical modelling of tread braked wheels, an experimental validation is necessary.

The optimisation of the wheel geometry (normally of the web) can give a benefit that is normally limited to a 5 – 10 % mass reduction. The thickness of the rim is linked to the required life of the wheel and its reduction will have a direct impact on the LCC of the vehicle. A more effective solution to reduce wheel mass would be to reduce the diameter of the wheel but this has a direct impact on the design of the bogie and the rest of the vehicle.

The design criteria for the wheel diameter are determined by the loads applied on the wheel (the static wheel load and dynamic wheel/rail contact forces), maximum vehicle speed, wear rate, maintenance intervals and the dimensions of the vehicle and the bogie. For a given axle load, the wheel/rail contact stresses decrease by increasing the wheel diameter.

Axle

For a given steel grade, the reduction of axle mass is normally limited to the possibility of machining a bore with a diameter of 30 – 65 mm (even more in some specific designs). Another reason to have a bore in the axle is to enable ultrasonic inspection to detect internal defects (cracks) during the axle service life. This solution can reduce the axle mass by 20 – 50 kg depending on the bore diameter.

Some further mass reduction may be obtained by designing axles with a conical profile between the wheel seats (smaller diameter at the axle centre). This optimisation is not common and can be effective only on trailer axles without disc seats in between the wheel seats. In this case, the mass reduction may be around 10 – 20 kg.

For further information, see D5.4 [7], Section 7.2.

5.1.2 Material

Wheel

EN 13979 defines one unique fatigue limit (240 MPa) for the standard wheel materials ER7, ER8 and ER9. The standard also defines a procedure to evaluate the actual fatigue limit of the applied wheel material using full-scale fatigue tests (staircase test) to take advantage of higher strength steel grades, but in practice this solution is never adopted.

Normally the wheel steel grade is selected depending on the service application and on the railway operator's experience related to the specific vehicle and service line in terms of wheel life related to wear, OOR growth and RCF (Rolling Contact Fatigue) damages.

Shifting from ER7 to ER8 or ER9 (increasing hardness) will lead to reduced wear. However, the reduced wear may also lead to increased tread surface cracking due to RCF. Subsequent pitting and spalling will make the rolling surface irregular and this might induce ground-borne vibration.

In cases where ER8 or ER9 steel grades are not able to solve RCF problems, special pearlitic-silicon and manganese carbon steels could improve the performance. Such kinds of steel grade (for example the material Superlos produced by the RIVAS partner Lucchini RS) are presently being introduced in the revision of the EN 13262 (under the name of ERS8), which will be published in 2014.

Axle

EN 13261 defines the standard axle steel grades (EA1N and EA4T) and their fatigue limits. Also a higher strength material called 30NiCrMoV12 is used successfully on high speed vehicles like the tilting Pendolino trains developed by the RIVAS partner Alstom.

For these three steel grades, reference fatigue limits and possible mass reductions that could be obtained with reference to a standard A1N axle are summarised in Table 5.1.

For further information, see D5.4 [7], Section 7.2.

Table 5.1. Possible axle mass reductions by use of different materials

Steel grade	Body fatigue limit (MPa)	Axle type	Mass (kg)	% mass reduction
A1N	200	Solid axle	505	
		Bored axle	455	10%
A4T	240	Solid axle	455	10%
		Bored axle	405	20%
30NiCrMoV12	300	Solid axle	397	21%
		Bored axle	347	31%

5.1.3 Production

Production process parameters that may influence OOR growth in service are:

- Wheel rim material homogeneity can normally be ensured by correct heat treatment after the rolling process (uniform temperature during the heating and uniform water spray quenching of the rim rolling surface). Hardness measurements at various angular positions around the rolling circle can be used to check the homogeneous characteristics, but such measurements are not required by the EN 13262.

  The accuracy of the wheel machining affects the balancing of the wheel. Once the wheels are fitted on the axle, wheelset dynamic imbalance must be limited according to Figure 5.1 (based on EN 13260), depending on the maximum vehicle service speed. The balancing of the wheelset is measured by a machine where the wheelset is rotated supported by the journals (the wheel treads are free). The imbalance is corrected by grinding off material from the rim or web side.

- The accuracy of the rolling surface profile machining affects the radial run-out of the rolling surface when the wheelset is rotating supported by the journals. For new wheelsets the radial run-out error should be less than 0.5 mm for maximum vehicle speeds up to 200 km/h and 0.3 mm for speeds above 200 km/h. During maintenance, when the wheels are reprofiled, the radial run-out limits can be higher. The reason can be that in some cases low floor lathes are used to turn the wheels while they are still mounted under the vehicle and thus the machining accuracy could be lower.

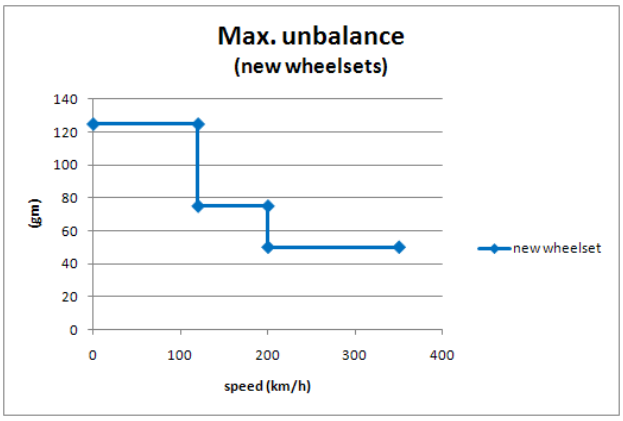


Figure 5.1. Maximum acceptable unbalance for new wheelsets depending on service speed (gm = gram·metres)

5.1.4 Cost benefit analysis

Indicative estimations of the influence of wheelset design and material selection on mass reduction and market cost are presented in Tables 5.2 and 5.3. The much higher cost for using alloy steel grade for the axle is partly due to the fact that not many manufacturers produce this type of axles and the use is not as common as A4T.

Table 5.2. Estimation of market cost for different wheel designs (shown as percentual decrease or increase in relation to the ER7 wheelset with medium wheel diameter)

Steel grade	Wheel diameter [mm]		
	700 – 850	890 – 920	1000 – 1200
ER7	- 9	0	37
ER8		3	
ER9		7	
Anti RCF steel grade		20	
Resilient wheel		230	

Table 5.3. Estimation of market cost for different axle designs (shown as percentual increase in relation to the solid axle with steel grade A1N)

Steel grade	Fatigue limit [MPa]	Axle type	Mass [kg]	Mass reduction [%]	Axle cost increase [%]
A1N	200	Solid	505	0	0
		Bored	455	10	20
A4T	240	Solid	455	10	40
		Bored	405	20	70
Special high strength alloy steel grade	300	Solid	397	21	225
		Bored	347	31	250

5.1.5 Discussion – passenger coaches and locomotives

The unsprung mass of different passenger vehicles may vary substantially depending on if the considered bogie is powered or not. For a trailer bogie, the unsprung mass will essentially consist of the wheelset mass, possibly with a contribution from brake discs and acoustic wheel dampers. In a powered bogie or in a locomotive, the wheelset mass will still dominate the unsprung mass but here a significant contribution may also come from the drive system consisting primarily of the motor and the gearbox. Depending on how the drive is suspended the significance of this contribution may vary, see Section 5.2. Locomotives generally have a larger and heavier drive system compared to powered coach bogies and hence the potential reduction due to a better suspension of the drive is higher for locomotives. It is primarily the material and the geometry of the wheelset that influence the unsprung mass. The material is selected to comply with requirements on fatigue and wear while the geometry is also influenced by the amount of removable material during re-profilings requiring a sufficient thickness of the rim. The diameter of the wheel will also be influenced by the design of the drive system. The rpm of the motor and the ratio of the gearbox should together with the wheel diameter ensure that the vehicle can reach its top speed at the coasting rpm of the motor. High-speed vehicles therefore tend to have wheels with a larger diameter compared to e.g. regional trains or metros.

A passenger vehicle may either consist of wagons with distributed power or by a locomotive pulling trailer coaches. A locomotive generally has a significantly higher unsprung mass compared to powered wagons and hence the ground vibration generated by the pass-by of the locomotive is expected to be higher compared to the vibration excited by any individual powered bogie. However, in a train with powered bogies, the average unsprung mass will be higher compared to a set of non-powered coaches. This means that the maximum vibration level during one pass-by is expected to be higher for a train with a locomotive while the equivalent level might be higher for a train with distributed power.

In addition to the reduction of unsprung mass by wheelset design, Section 5.2 describes the potential reduction of the unsprung part of the drive mass. Taking both parts into account it is concluded that the potential reduction of the mass is in the range of 15 % for a trailer bogie, 25 % for a powered bogie and 35 % for a locomotive with a bogie design where the drive is more or less completely suspended from the wheelset (fully suspended drive). Numerical simulations have shown that a reduction of the unsprung mass by 50 % may lead to a reduction of 6 dB on the overall ground vibration level [11]. Measurements have shown that a mass reduction of 35 % gives a reduction in overall ground vibration level of 2 – 4 dB [8]. Based on the current assessment, these results indicate that the feasible mass reduction would give a vibration reduction in the range of 2 dB on the equivalent vibration level of a complete train and up to 3 dB for a single locomotive.

5.1.6 Discussion – freight wagons

Due to the lack of a drive, the variation of unsprung mass between different freight wagons is relatively small compared to for locomotives.

The unsprung mass of one wheelset in a Y25 bogie is 1355 kg (wheel diameter 920 mm). In the RIVAS field test campaign in Switzerland [8], two other bogie types were tested. One bogie was the Y33 with smaller diameter wheels (730 mm) and unsprung mass 1060 kg. The other bogie was a Y25 bogie with axle-mounted disc brakes and unsprung mass 1610 kg. The measured insertion loss at 4 m from track and at four train speeds are presented in Figure 5.2. The insertion loss is 5 – 10 dB. However, the wheel tread conditions of the two compared bogies in the related wavelength range differed by 0 – 5 dB. From a further analysis of the results, the quantification of the unsprung mass influence was estimated to 2 – 4 dB when modifying the mass from 1610 kg to 1060 kg.

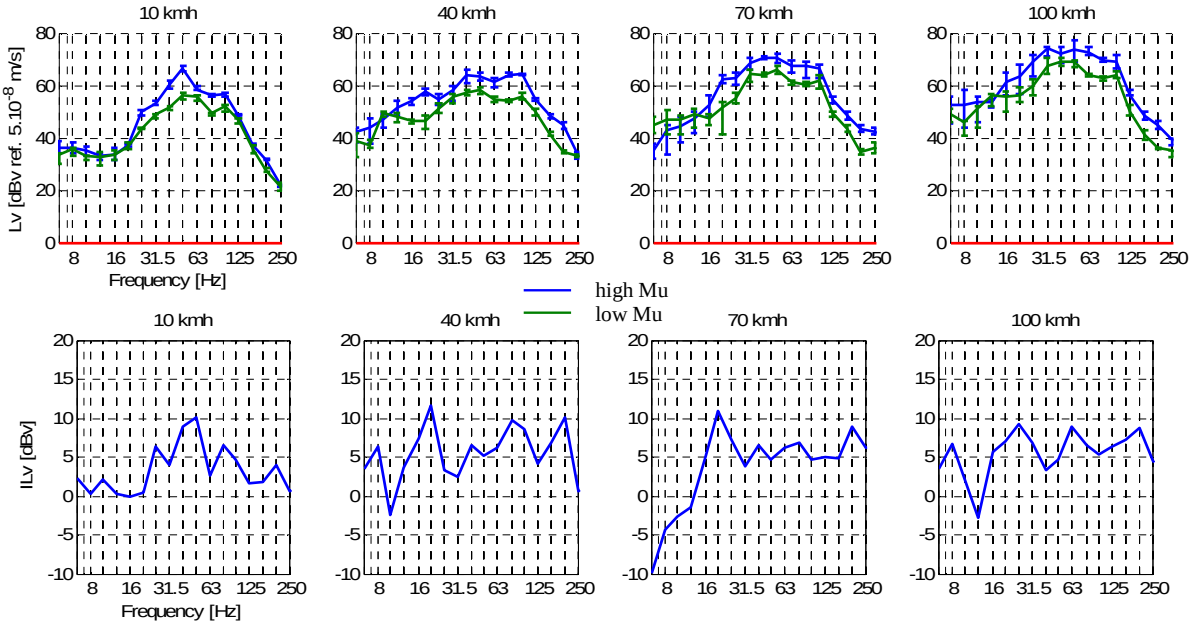


Figure 5.2. Measured influence of unsprung mass at four different train speeds. Upper figures: ground-borne vibration at 4 m from track (green curves – low unsprung mass 1060 kg, blue curves – high unsprung mass 1610 kg), lower figures: insertion loss when applying low unsprung mass instead of high unsprung mass

The unsprung mass could be further reduced by using even smaller diameter wheels. For example the RoLa (Rollende Landstrasse) bogie has wheels with diameter 360 mm. A statistical evaluation of measured ground-borne vibration at 8 m induced by regular trains on the SBB network (speed 60 – 70 km/h) showed a reduction of 10 dB at 31.5 Hz (sleeper passing frequency), but probably this was also influenced by the low static load and a short axle distance, see Figure 5.3. On the other hand, it should be noted that such small diameter wheels can carry approximately three times less wheel load than standard wheels. The small diameter wheels also have a negative cost driving influence on maintenance because of higher wheel/rail contact stresses and more wheel revolutions per travelled distance (RoLa wheels generate high wheel maintenance costs), which leads to worse wheel tread condition.

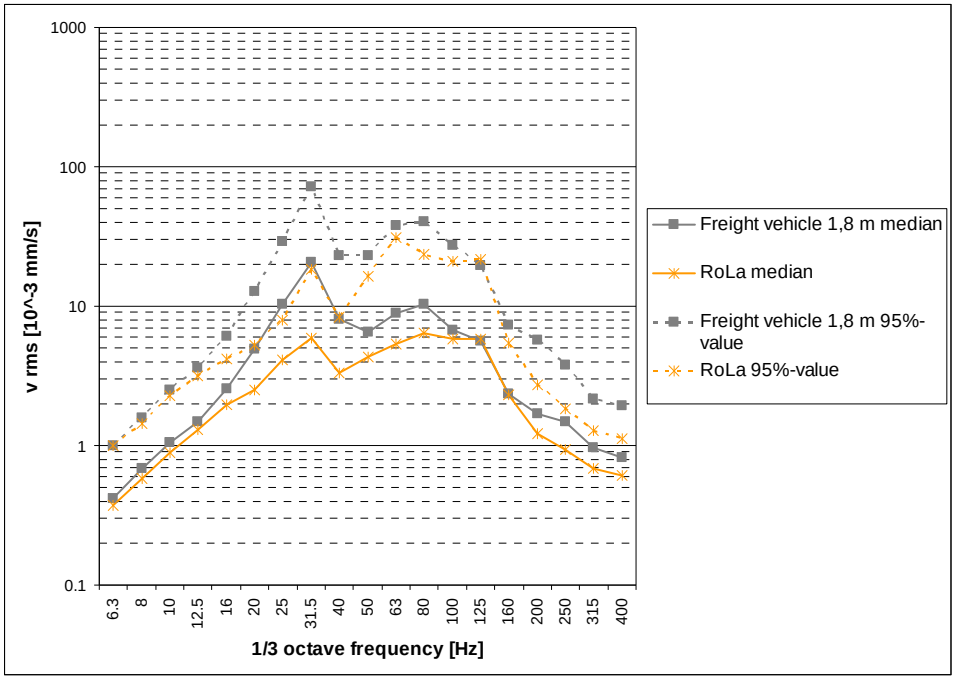


Figure 5.3. One third octave band spectra of measured ground-borne vibration induced by standard freight vehicles and RoLa, Thun, Switzerland, speed 60-70 km/h. From [12]

In [7], it was shown by numerical simulations that the most effective means to reduce ground-borne vibration by vehicle optimisation is minimising the unsprung mass as this increases the vehicle receptance in a wide frequency range. This effect can be illustrated by studying the combined vehicle and track receptance for two of the different vehicle parameter combinations studied in [7], see Figure 5.4. It can be shown that the combined vehicle and track receptance relates the dynamic component of the wheel/rail contact force to the track irregularity. Thus, for a given track irregularity spectrum, reducing the combined vehicle and track receptance (increasing the dynamic stiffness of the coupled vehicle/track system) in a certain frequency range increases the dynamic axle loads and ground-borne vibration in the same frequency range. The combined receptances for two vehicle parameter combinations (10 and 29) are compared in Figure 5.4. Vehicle 10 with a stiffer primary suspension shifts the wheelset resonance on the primary suspension to a higher frequency. The combined receptance is lower in the frequency range up to about 10 Hz which leads to higher ground-borne vibration levels in the 1/3 octave bands below 10 Hz, see Figure 5.5(a). On the other hand, the lower unsprung mass of vehicle 10 reduces the dynamic stiffness at higher frequencies and thus lower ground-borne vibration is generated for frequencies above 10 Hz. The influence of the resonance due to the lowest symmetric eigenmode in wheelset bending on the combined receptance is small.

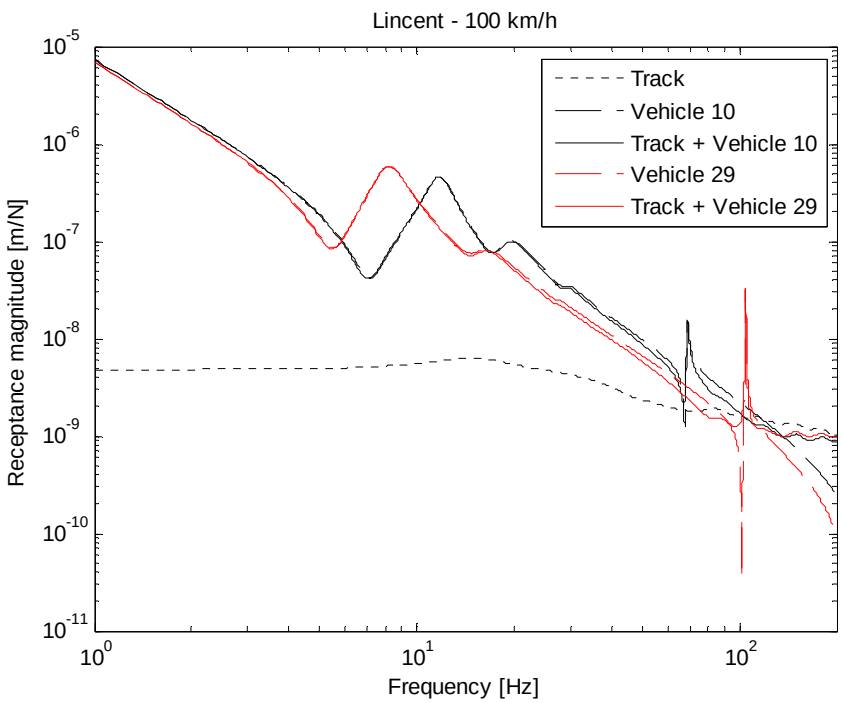
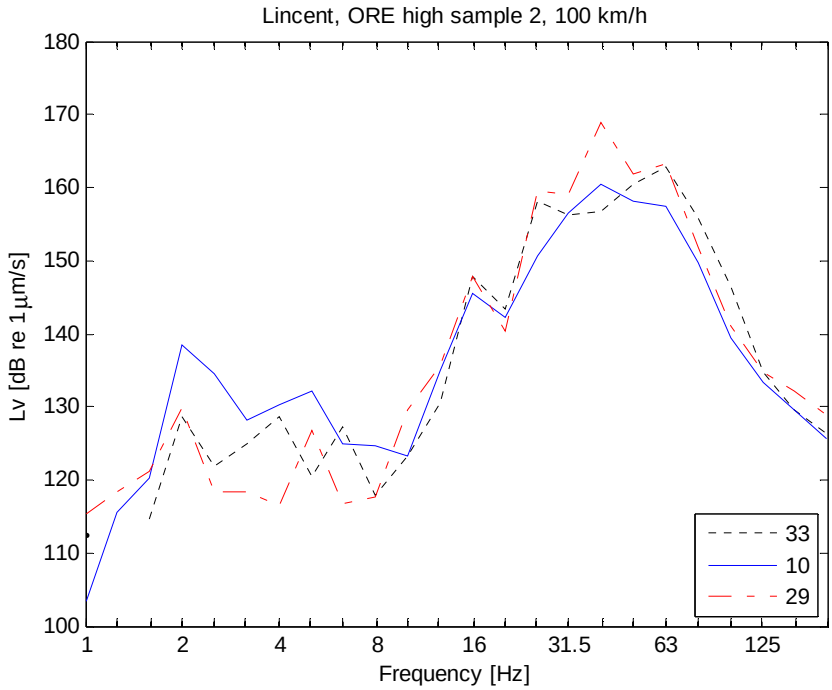
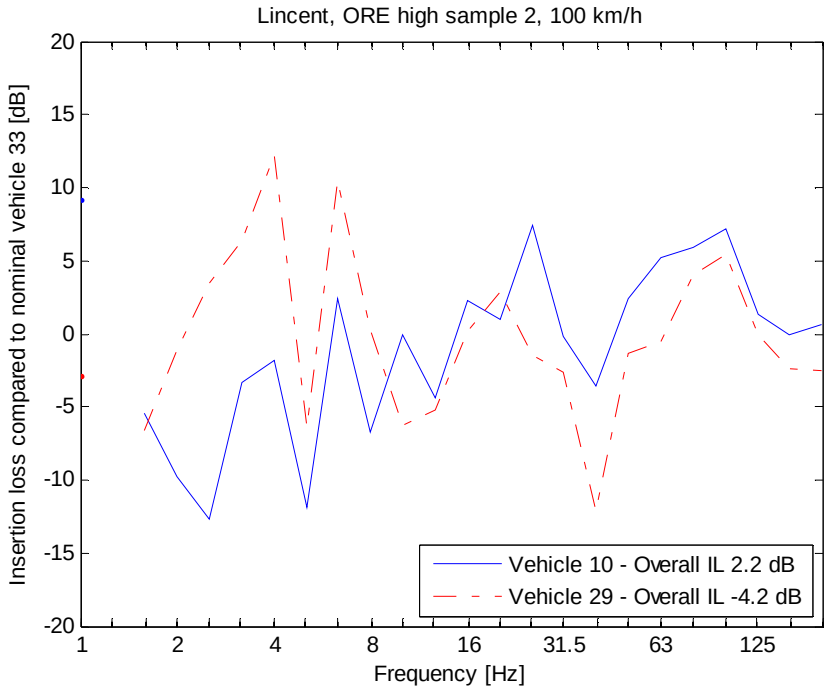


Figure 5.4. Magnitude of combined vehicle and track receptance for two vehicle parameter combinations. The combined receptance is dominated by the vehicle receptance at frequencies up to about 80 Hz. Track characteristics according to Lincent reference model. From [7]



(a)



(b)

Figure 5.5. (a) Level of RMS free field ground-borne vibration at 8 m from track evaluated in one third octave bands, nominal vehicle parameters (33), studied vehicle model with combination of parameters leading to lowest vibration (10), studied vehicle model with combination of parameters leading to highest vibration (29). (b) Insertion loss compared to nominal vehicle. Vehicle speed 100 km/h. Soil conditions: Lincent. Track irregularity profile based on spectrum ORE B176 high. From [7]

5.2 DRIVE CONCEPTS AND SUSPENSION PROPERTIES

5.2.1 Drive concepts

The influence of vehicle unsprung mass on the generation of ground-borne vibration has been studied in RIVAS, see [5,7,8].

The unsprung mass primarily consists of the wheelset mass and part of the mass of the drive system. The choice of drive concept and drive suspension concept will therefore influence the unsprung mass. The potential for reduction of wheelset mass was discussed in Section 5.1.

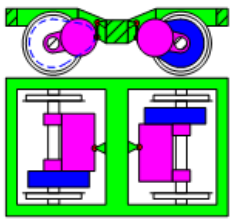
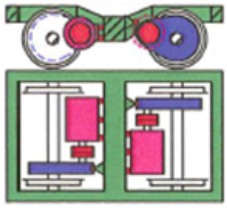
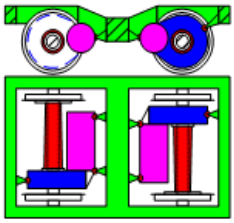
The mechanical drive is defined as the traction motor, the gearbox and any mechanical elements designed to compensate for the relative motion between the wheelset axle and the bogie frame. This section gives an overview of the most commonly used concepts for mechanical drive suspension on passenger trains with powered bogies. For further information, see [7].

Nose suspended drives are riding the wheelset axle via two bearings, see Table 5.4. The third suspension point connects the traction motor flexibly to the centre of the middle bogie beam. The gearbox mass and most of the traction motor mass are unsprung, which limits the maximum vehicle speed for this design to 100 – 140 km/h. The traction motor tends to be large to achieve the required torque at vehicle start-up, thus contributing to a high total drive mass. The nose-suspended drives are of a simple and proven design. The limited number of rubber mountings will ensure a low life cycle cost.

The semi-suspended drive, sometimes also called partly-suspended, features a gearbox riding on the wheelset axle and a traction motor mounted on the bogie middle beam. The motor mass is hence completely sprung by the primary suspension, see Table 5.4. This leads to a low unsprung mass of the bogie which is suited for high-speed applications. The semi-suspended drives are of simple and proven designs. Semi-suspended drives are not appropriate for bogies with inboard bearings as such bogies offer less space inside the bogie frame.

In a *fully suspended mechanical drive*, the traction motor and the gearbox are forming one unit which is completely sprung by the primary suspension, see Table 5.4. The unsprung mass of the bogie is therefore limited to the wheelset mass which is desired in high-speed applications. The relative movements between wheelset and drive are compensated by a hollow shaft. The initial cost of a fully-suspended drive is usually high due to the need of a two-stage gearbox and a hollow shaft coupling. The life cycle costs can be high due to the coupling via rubber bushings.

Table 5.4. Overview of the three most commonly used drive suspension concepts

Principle			
Name	Nose-suspended	Semi-suspended	Fully-suspended
Mounting characteristics	Motor riding the axle with bearing. Third connection point on the bogie middle beam	Motor suspended on the bogie middle beam. Gear riding the axle. Coupling connecting both	Drive fully supported on the bogie frame. Hollow shaft compensating the relative movements
Typical application	Locomotives	Locomotives operating at higher speeds, passenger trains with distributed power	High-speed trains
Unsprung part of total drive mass	85 %	20 %	5 %

5.2.2 Suspension properties

Axle guidance

The design of the axle guidance in a bogie is usually determined by vehicle dynamics, aiming at a best compromise (in case of a conventional design) between running stability and curving ability. The axle guidance stiffness is determined by the effective axle guidance itself (taking the kinematics into account, i.e. the longitudinal stiffness of the axle guide bush multiplied by the squared ratio between the lateral spacing of the bushes and the track width) and by the effective longitudinal stiffness of the primary suspension springs. Unconventional active or passive concepts of axle guidance include

- Hydraulic axle guide bushes with frequency-dependent longitudinal stiffness for passenger coaches
- Cross-coupling of the wheelsets in a bogie (e.g. SBB Re460, see third example of running gear in Figure 5.6)
- Steered wheelsets via a mechanical rod transmission of the car body versus the bogie steering angle (e.g. SBB ICN, see fourth example of running gear in Figure 5.6)
- Steered individual wheelsets or steered independent wheels (e.g. Talgo Pendular)
- Actively controlled steering of wheelset or independent wheels
- Active steering of the bogie frame (active yaw damper)

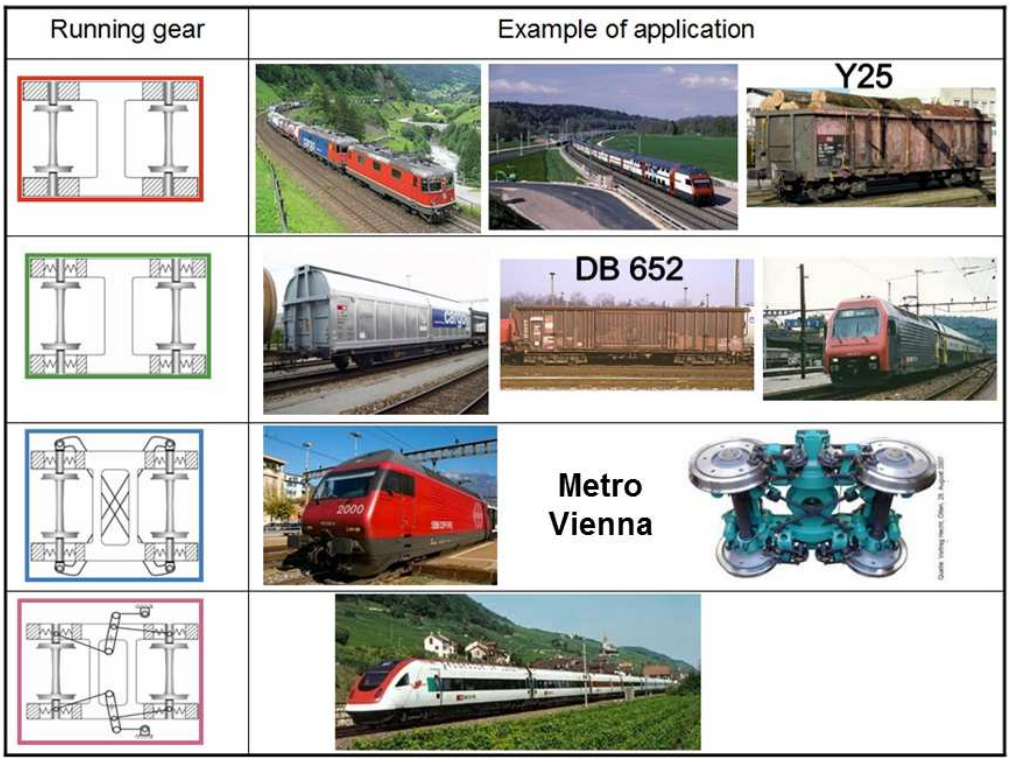


Figure 5.6. Examples of bogie design and applications

The use of soft (conventional design) axle guidance in small radius curves, where there is insufficient difference in wheel rolling radius in a wheelset, may lead to wheel OOR, and possibly also to growth of rail roughness at the corresponding wavelength if the traffic is uniform (several vehicles of the same type operating on the track), e.g. freight wagons or LRVs (Light Rail Vehicles). For a given combination of wheel load and wheel/rail friction coefficient, a softer axle guidance is more critical for this rail roughness generation mechanism because of the higher energy released during the slip phase. The corresponding maximum force in the axle guidance (reached at the end of the stick phase in the wheel/rail contact) is proportional to the axle guidance stiffness and the deflection of the suspension, while the stored energy released during the short slip phase is proportional to the axle guidance stiffness and the square of the deflection. Accordingly for given axle load and wheel/rail friction characteristics, the abrasive energy in the wheel/rail contacts is proportional to the inverse of the effective axle guidance stiffness.

Bogies with cross-coupled wheelsets are not only more complicated and expensive but also more prone to cause hollow wheel wear (flange growth) unless the vehicles are operated on track with a high portion of small radius curves, e.g. SBB transalpine operation. Thus, it is important to select an axle guidance design that is appropriate for the network it will be operated on (distribution of tangent track and curve radii).

Vertical suspension

The vertical stiffness and damping of the primary suspension are designed to fulfil requirements on vehicle dynamics (especially safety against derailment, roll coefficient), structural integrity and space. The bogie bounce frequency might have a small impact on ground-borne vibration, e.g. if coinciding with the sleeper passing frequency at low speeds. The secondary suspension level implicates frequencies lower than those dominating the ground-borne vibration. It is not anticipated that the vertical suspension characteristics (primary and secondary levels) have a significant impact on the generation of wheel OOR.

For the Y25 bogie, inner and outer coil springs in the primary suspension without any viscous dampers are typical. For the damping, a so-called Lenoir link is installed which provides a normal force onto dry friction surfaces with respect to the vertical spring force. This kind of friction damper can lead to sticking and sliding effects, especially for a partially loaded wagon [13]. A malfunctioning primary suspension of a trailer bogie could lead to a higher unsprung mass. If the suspension is locked, parts of the bogie mass will also be unsprung. This induces higher vertical forces in the wheel/rail contact and higher excitation of ground-borne vibration. In contrast, leaf springs and hydraulic dampers (as for example integrated in the Axiom TF 25E bogie) lead to a smoother vertical characteristic of the spring and are therefore better suited for low vibration excitation.

5.2.3 Technology assessment

The vehicle design in terms of the drive concept and the suspension properties may have an influence on the generation and growth of wheel OOR. Since wheel OOR is one of two key vehicle factors leading to the generation of ground-borne vibration, a technology assessment should consider the design changes on bogie and/or on vehicle level which potentially could reduce the growth of OOR. The assessment should include

- *Impact on OOR*
  - Systems which may influence OOR initiation and growth
  - Interaction with mechanisms driving OOR growth
  - Potential design modifications which could contribute to reduction of OOR growth
- Conflicts between various design targets
  - Noise & vibration
  - Structural integrity, vehicle dynamics, aero-/thermo-dynamics, weight, space, etc.
- Cost (manufacturing and LCC) of any identified vehicle design improvement mitigating OOR

5.2.4 Cost benefit analysis

Experience from specific vehicle operations gives some indication of the possible benefits in reduced growth of wheel OOR, probably resulting from a more curve friendly axle guidance, see Section 5.2.5.

The additional cost for a locomotive with a curve friendly axle guidance concept, which simultaneously does not affect running stability, is difficult to quantify.

Passive solutions, such as cross-coupled wheelsets, complicate maintenance and increase the risk of generating hollow wheel wear, especially if operated on tracks with few small radius curves. Active axle guidance on the other hand adds a different type of complexity.

Apart from the benefit of lower ground-borne vibration, reduced OOR growth can obviously also lead to reduced life cycle cost due to considerably increased wheel reprofiling intervals.

5.2.5 Discussion – locomotives

Long term investigations and research projects on OOR growth have previously been carried out in Germany and Switzerland, involving DB and SBB. These projects did not lead to any clear conclusions regarding the mechanisms driving the growth of wheel OOR, and thus in many cases the causes for OOR growth remain unknown.

The vibration measurements on Swiss track performed by SBB and evaluated in RIVAS have shown significant ground-borne vibration generated by freight locomotives of the types Re420, Re474 and Re482/484. The vibration spectra are dominated by two excitation

mechanisms: the parametric excitation due to sleeper passing (for all vehicles) and the wheel OOR (large spread between different wheels). In contrast, the Re460 passenger locomotives do not generate severe ground-borne vibration related to wheel OOR while vibration generated at sleeper passing frequency are at levels comparable to the other locomotive types.

It appears that even when operated in Swiss transalpine freight traffic, the Re460 locomotives (with sinter block tread brakes, fully suspended drive and curve friendly cross-coupled wheelsets) reach very long wheel reprofiling intervals. This is only possible with very slow or no OOR growth. In similar operation, the older Re420 locomotives (with cast iron block tread brakes, partly suspended drive and conventional axle guidance) seem to have much faster wheel polygonalisation growth of mainly order 14 (fourteen). This is probably also the reason for the comparably short reprofiling intervals.

One fundamental open question is what is the cause of the OOR growth? Also the newer freight locomotives currently used in Swiss transalpine freight traffic, Re474, Re484 and Re482 (with disc brakes, nose suspended drive and conventional axle guidance with longitudinal dynamic stiffness 53 kN/mm), generate a wide range of ground-borne vibration levels at frequencies corresponding to a wavelength of about 270 mm. OOR measurements on some of these locomotive wheels confirm amplitudes and wavelengths with a wide spread of OOR levels of predominantly the order 14, see Figure 5.7.

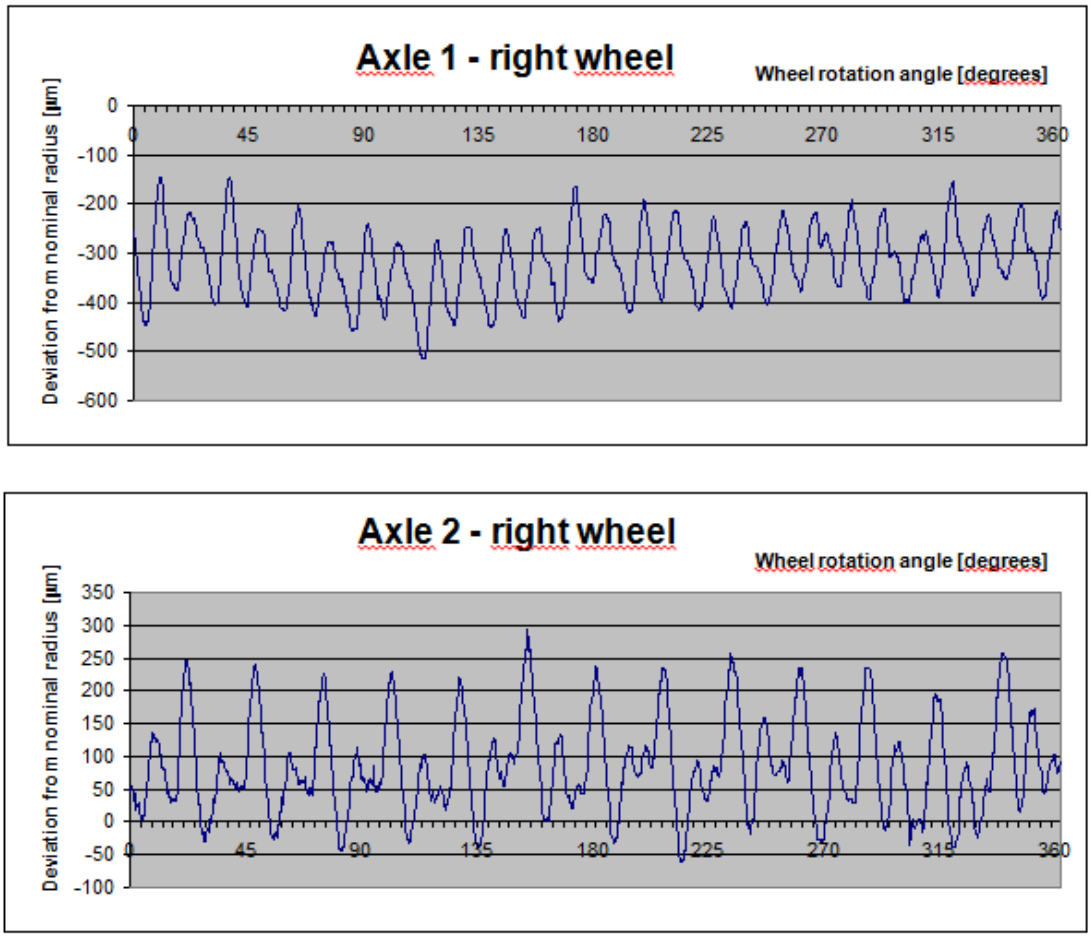


Figure 5.7. OOR measurement on SBB freight locomotive SBB Re484 with dominating wavelengths 135 and 270 mm. The nominal diameter of a new wheel is 1250 mm

5.2.6 Discussion – freight wagons

The focus of the extensive field test performed in RIVAS was to study the influences of wheel condition and unsprung mass. However, due to the bilinear stiffness characteristics of the Y25 coil springs (inner and outer springs) and the different axle loads in the test train, the influence of primary suspension was also included in the measurement campaign. The results do not show a significant effect in the expected frequency range (< 10 Hz). A possible reason for this is the evaluation process where the frequency analysis had to be performed based on short time windows, which does not allow for a proper evaluation at low frequencies.

Also according to numerical simulations, the influence of suspension stiffness on induced ground-borne vibration is low and restricted to one or two one third octave bands [11,14].

The use of innovative, modern freight bogies like the LEILA bogie (radial steering, disc brakes, hydraulic dampers), especially designed for air-borne noise reduction, have not yet been established in the highly competitive freight traffic market. As long as using low noise emission freight bogies generate no or only few financial benefits, the incentives to use these bogies will not be sufficient. Because low vibration bogies have similar properties as track friendly bogies (low unsprung mass, radial steering or shorter axle distance, low static axle load), there is a chance to establish such bogies where they benefit from lower track access charges, see e.g. in the UK with the focus to reduce LCC of infrastructure. For the Desiro City train used in the UK, a track friendly bogie SF 7000 is being used [15]. Such bogies with the before mentioned properties, have the potential to reduce ground-borne vibration.

5.3 WHEELSET DISTANCE

The wheelset distance could influence the induced ground-borne vibration in several ways, directly or indirectly. A shorter distance between the wheelsets improves the bogie curving ability and leads to less wheel wear [15]. Wheel wear due to wheel/rail interaction has been pointed out as one important cause of wheel OOR and hence less wear in curves is expected to reduce the growth of OOR.

Out-of-round wheels have a significant influence on the generation of ground-borne vibration [11], and hence an indirect effect of wheelset distance on vibration level is expected. A direct effect of wheelset distance is seen through its influence on the frequency with which individual wheelsets are passing a given position along the track (the axle passing frequency). Similarly the distance between the bogies in one wagon/coach will influence the bogie passing frequency. Both of these frequencies, depending on vehicle speed, can often be identified in the vibration spectrum measured on the rail or on the ground at short distances from the track. At longer distances from track, these characteristic excitation frequencies tend to be less clear due to the strong influence of the ground vibration response. Figure 5.8 shows measured spectra of rail vibration due to train pass-bys at different vehicle speeds. The frequencies corresponding to the pass-by of individual axles and bogies have been traced to illustrate how the frequencies are influenced by a change in speed. The excitation due to the discrete sleeper supports (the sleeper passing frequency) has also been identified and is in the same fashion determined by vehicle speed and the distance between the sleepers.

The influence of vehicle geometry and vehicle speed has also been shown in numerical simulations [5,7,14]. For a given vehicle speed, the excitation is shifted to a higher axle passing frequency if the wheelset distance is shorter and in the opposite way to a lower axle passing frequency if the wheelset distance is longer. This means that the vibration level will increase respectively decrease in two adjacent frequency bands. On the rail and close to the track this redistribution of vibration energy from one frequency band to another could be rather large as indicated by Figure 5.8. However, according to simulations, at longer distance from track the influence is in the range of 1 dB in the affected frequency bands and insignificant on the overall vibration level.

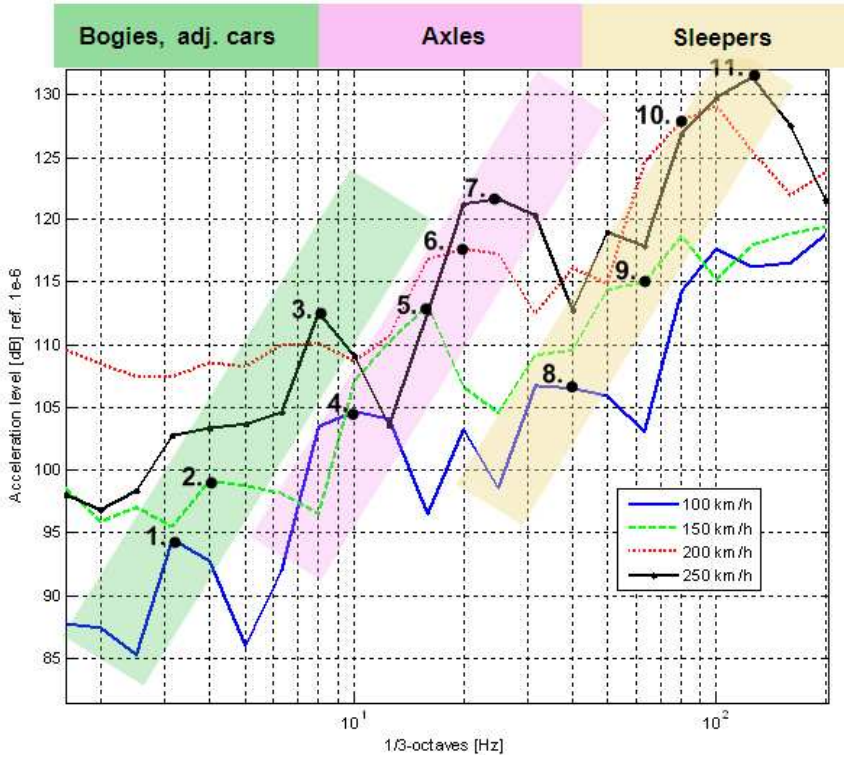


Figure 5.8. Rail head vibration spectrum of vehicle pass-bys at different speeds. Peaks 1, 2 and 3 correspond to the pass-by of neighbouring bogies in two adjacent cars, peaks 4 – 7 correspond to the pass-by of individual axles within the same bogie while peaks 8 – 11 correspond to the excitation due to discrete sleeper supports (sleeper distance 0.65 m). From [16]

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5.4 TRACTION CONTROL AND BRAKE SYSTEM

5.4.1 Traction control

Freight locomotives must be able to generate high traction forces also under conditions with low wheel/rail friction. This inevitably leads to high longitudinal creepages in the wheel/rail contacts. In order to generate a high traction bound, an increased wheel tread roughness level is often aimed for and achieved through tread braking or (on locomotives without tread brakes) by applying high traction forces which generate high longitudinal creepage. It is possible that operation with high longitudinal creepages accelerates the growth of wheel OOR. Independently of the means to obtain the required wheel roughness, levels should be low at wavelengths relevant for ground-borne vibration.

5.4.2 Brake system

While tread brakes with cast iron blocks are known to generate high wheel roughness levels at short wavelengths inducing rolling noise, their effect on OOR growth at longer wavelengths is less evident. The influence of so-called Sgns freight wagons with cast iron or composite brake blocks on ground-borne vibration has been measured at Cadenazzo in Switzerland. The ground-borne vibration was almost equivalent up to 200 Hz (at speed 65 - 70 km/h). Above 200 Hz, there is a positive effect of the composite brake blocks due to the smoother wheel tread, see Figure 5.9.

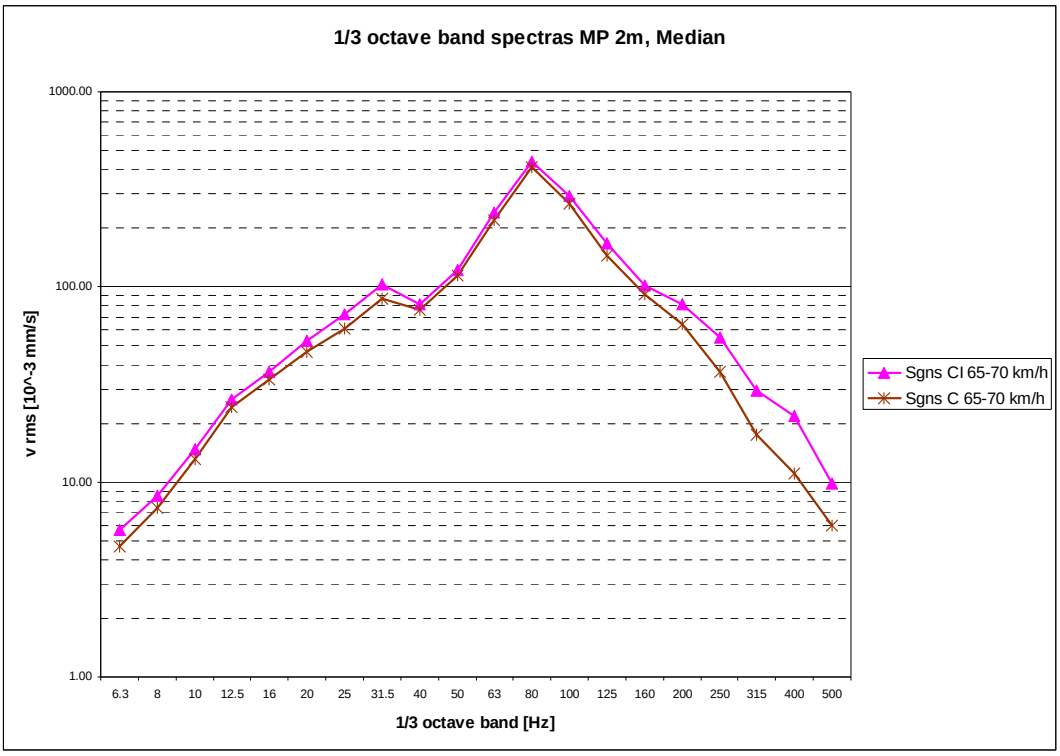


Figure 5.9. Comparison of one third octave band spectra (median) of ground-borne vibration measured at 2 m for Sgns container wagons with cast iron or composite brake blocks

The abrasion of the wheel tread by tread brakes with brake blocks could even be imagined to reduce OOR growth by reducing initial RCF crack formation and growth. A further positive effect of brake blocks is that they do not increase the unsprung mass as is the case with disc brakes.

Insufficient brake control can generate wheel flats. One typical example observed in an extensive field test on the SBB network is the vibration induced by inter-car bogies in the middle of the NINA regional train. These bogies have insufficient wheel-slide protection and wheel flats are common. In Figure 5.10, it is assumed that the peak at 80 Hz can be explained by the P2 resonance being excited by the impacts of wheel flats.

Also on the SBB Re420 locomotive of the test train, despite the use of wheel-slide protection, a wheel flat was detected already after one month of operation after wheel reprofiling, see Figure 5.11. Freight wagons generally do not have wheel-slide protection and wheel flats are often observed.

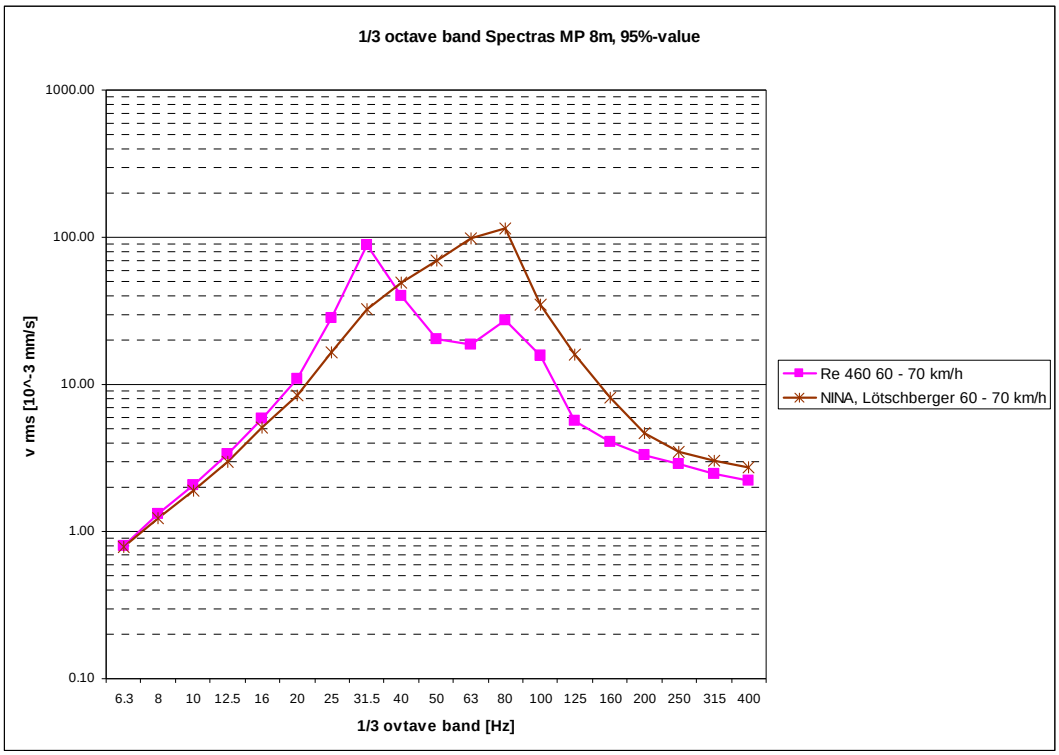


Figure 5.10. Comparison of one third octave band spectra (95 % value) of ground-borne vibration measured at 8 m for NINA regional trains and SBB Re460 locomotives. Based on measurement of 227 NINA bogies and 156 Re460 bogies

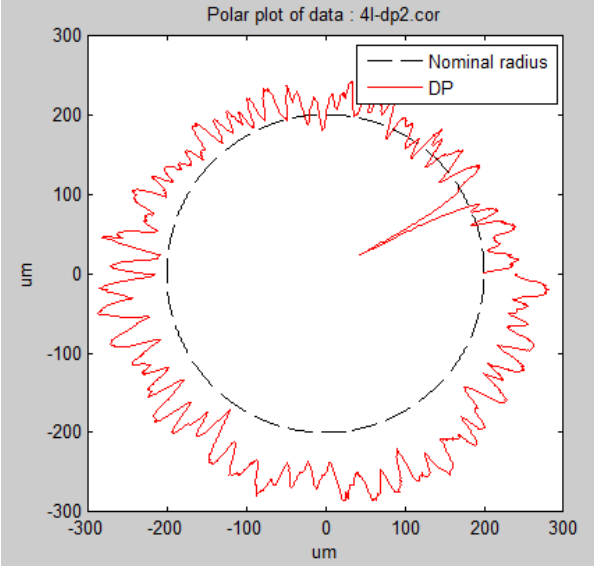


Figure 5.11. Polar plot of measured wheel OOR on the SBB test train Re420 locomotive. DP illustrates the result from measurement at the centre of the running band

5.4.3 Cost benefit analysis

The brake system of freight wagons is in most cases based on tread brakes because this is the most economic (LCC) solution. Compared to the more expensive disc brakes, there are also some advantages in terms of reducing ground-borne vibration due to

- Lower unsprung mass (in the order of 250 kg)
- Reducing RCF crack initiation and growth

Disadvantages of brake blocks are higher wheel wear and thermal impact on the wheel tread, especially when using composite brake blocks. At high speeds, the braking power of brake blocks is not sufficient. This means that for passenger coaches usually disc brakes are necessary.

The influence of changing the brake block material on induced ground-borne vibration, e.g. replacing cast iron blocks with composite blocks, is rather small and for usual speeds of freight trains only above 200 Hz. The cost of this retrofit (executed in Switzerland for noise reduction) is around 5 000 € per wagon.

Older freight locomotives in Switzerland with cast iron brake blocks (SBB Re 420/620) have not been retrofitted with composite brake blocks. On one hand, the generated level of wheel roughness allows for application of high traction forces, which is crucial for a freight operator to be competitive. On the other hand, this means no noise reduction compared to the retrofitted freight wagons behind the locomotive.

Wheel slide protection is a means to reduce the generation of wheel flats, which have a certain influence on ground-borne vibration in the near field. For locomotives and passenger coaches, it is widely used but not for freight vehicles. Since wheel slide protection on freight vehicles is rarely used, it is difficult to obtain information about costs. The cost for wheel slide protection is estimated at around 4 000 € per wagon.

6. VIBRATION MITIGATION BY VEHICLE MAINTENANCE

6.1 RELATIVE INFLUENCE OF UNSPRUNG MASS AND WHEEL TREAD CONDITION

The unsprung mass and the conditions of the wheel treads have strong influences on the ground-borne vibration generated by a vehicle [5,11]. The wheel tread condition in terms of level of wheel OOR and the presence of wheel flats may vary substantially among individual vehicles in a fleet and will depend on vehicle and brake design, operating conditions as well as the maintenance schedule applied to the vehicle. Figure 6.1 shows measured OOR order spectra for several wheels in both freight and passenger vehicles [17]. It is clear that both types of vehicle show a large spread in OOR level. The highest levels of OOR are commonly observed for the lowest wheel orders (such as order 1 = eccentricity, order 2 = ovality, and order 3 = third order polygon), which are especially important at the low frequencies important for ground vibration.

The significant influence of wheel OOR on vibration emission has been shown by using the measured data in Figure 6.1 as input to numerical predictions of ground-borne vibration. The influence of OOR has been studied in parallel to design changes in vehicle properties, level of vertical track irregularities and soil conditions. For a severe form of OOR (one of the worst freight wheels in Figure 6.1), Figure 6.2 illustrates the influence of four different vehicle designs on ground vibration. The vehicle design changes include two levels of unsprung mass and two levels of primary suspension stiffness. In the studied example, a doubling of the unsprung mass leads to an increase in the overall vibration level which is in the same order of magnitude as the increase caused by wheels with severe OOR compared to wheels in good condition. In cases where the OOR is particularly high for a specific order, the vibration level in a single one third octave band may increase up to 15 dB, see Figure 6.2 and the influence of severe wheel eccentricity (order 1) at 10 Hz.

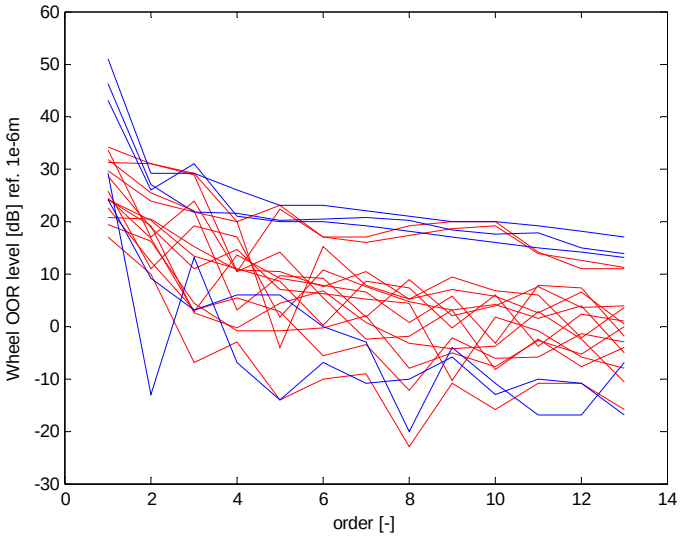


Figure 6.1. OOR measured on freight vehicles (blue lines) and passenger vehicles (red lines)

Similar results have been achieved by studying the influence of unsprung mass and OOR in field tests [8]. It was found that the lower unsprung mass of a freight wagon compared to the unsprung mass of a locomotive leads to a reduction in vibration level which is in the same order of magnitude as the difference between wheels in good or poor condition.

Note that a reduction of unsprung mass by 50 %, as considered in Figure 6.2, should be considered a large reduction which in most cases would be difficult to achieve. Such a reduction is more representative for the difference between a locomotive and a wagon/coach. The 6 dB reduction (see the grey legend) resulting from this design change should therefore be considered an upper limit. Also the OOR considered in the simulations is a worst case and the average reduction achieved by removing OOR is therefore expected to be less than the 5 dB seen here. However the possible reductions by reducing the unsprung mass and by removing OOR are in the same order of magnitude. This implies that if a vehicle is designed with a low unsprung mass but is not properly maintained, the positive effect of the low unsprung mass on the vibration level may be completely lost due to high levels of OOR.

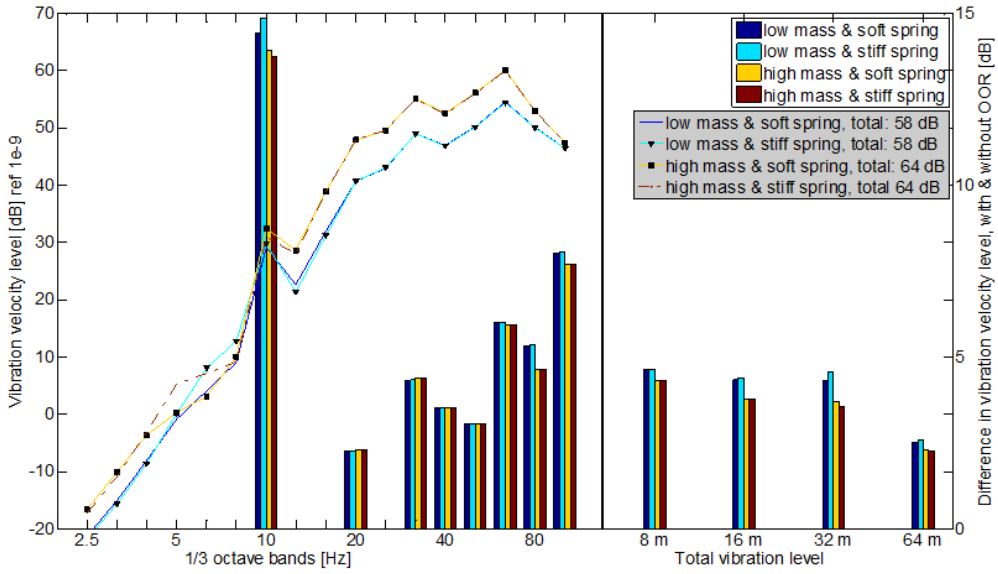


Figure 6.2. Vibration spectra at 8 m from track centre (read scale on left ordinate) with total levels indicated in the grey-boxed legend. Wheel/rail excitation by the ISO 3095 track irregularity spectrum (track in good condition) and OOR corresponding to a freight wheel in poor condition. Four different vehicle designs with high or low level of unsprung mass (2850 kg, 1425 kg), and stiff or soft primary suspension (7 kN, 4 kN). Soil conditions according to Lincent and vehicle speed 100 km/h. The left bar diagram (read scale on right ordinate) shows the increase in vibration level at 8 m caused by the OOR compared to a round wheel, evaluated in 1/3 octave bands. The right bar diagram (read scale on right ordinate) shows the corresponding increase in vibration level caused by the OOR on the total level at 8, 16, 32 and 64 m from track

A further assessment of measured ground-borne vibration levels show that overall mean vibration level and the spread from one bogie to the next is significantly dependent on the type of locomotive, as illustrated in Figure 6.3. Compared to the passenger locomotive Re460, the freight locomotives (TRAXX, Re420 and Re620) induce higher mean vibration levels. The significant scatter between freight locomotives of the same type, as indicated by the large difference between the mean value and the 95 % value, is mainly explained by the difference in wheel tread conditions within a given fleet of locomotives. Different types of wheel OOR are discussed in Appendix A. The influence of unsprung mass on ground-borne vibration level was difficult to evaluate from this set of measurement data.

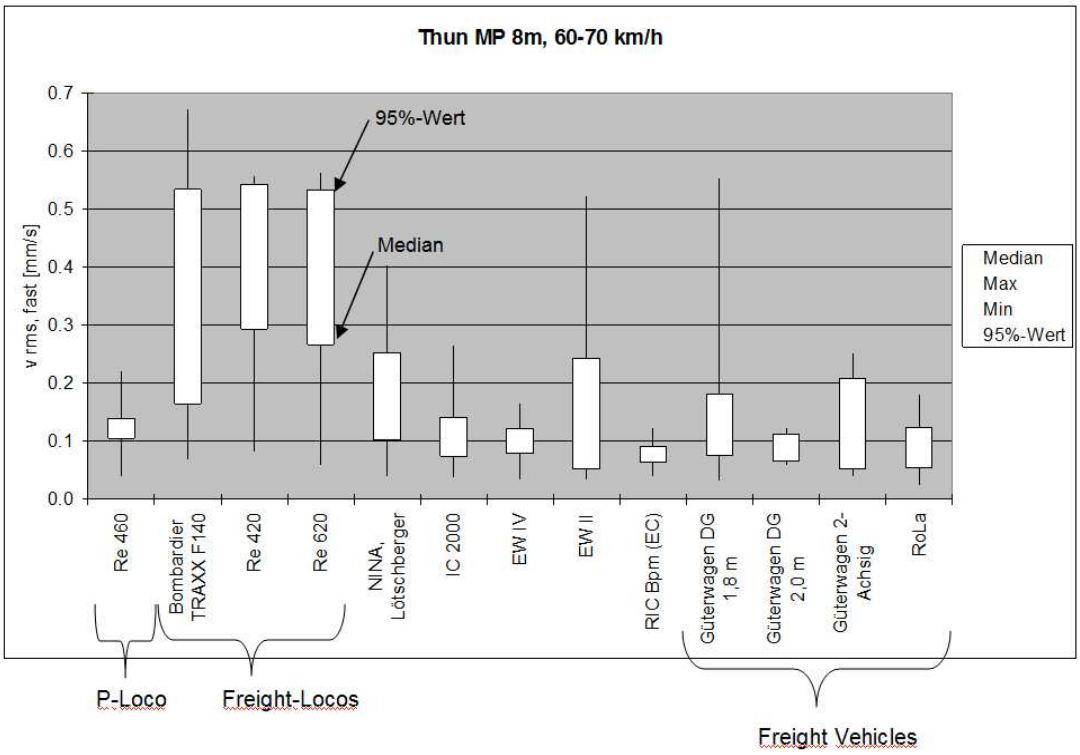
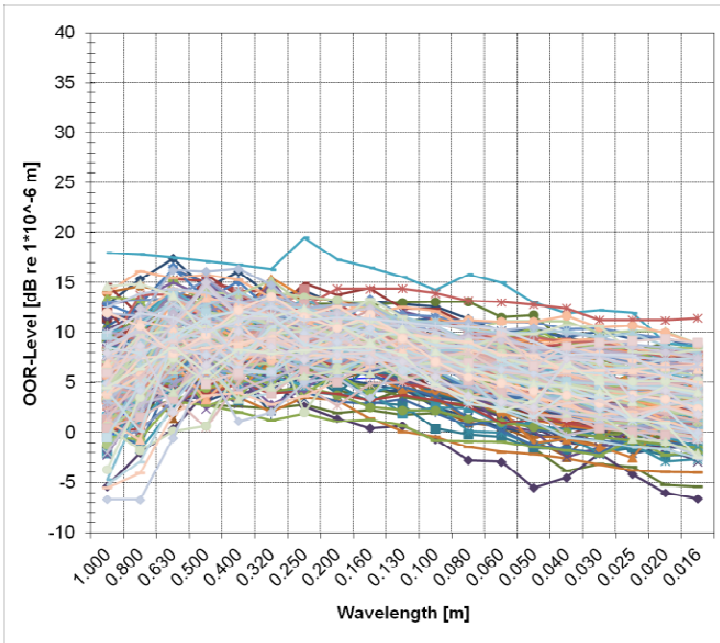


Figure 6.3. Comparison of measured ground vibration at 8 m from track for different types of bogie. From [12]

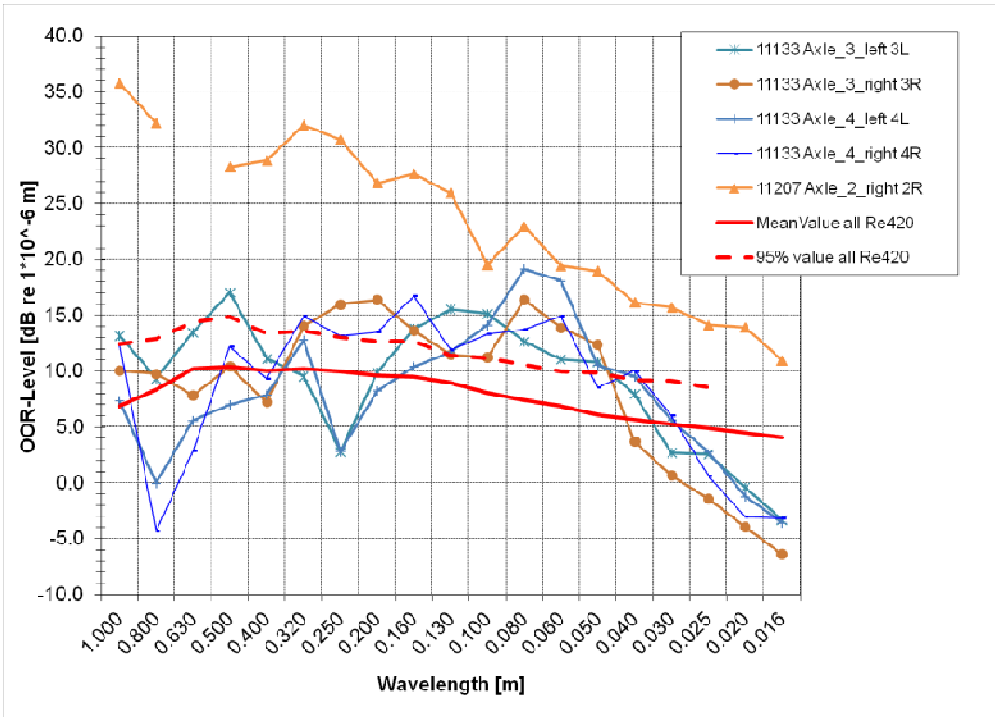
The significant scatter in OOR on Re420 wheels, as measured in a SBB workshop, is illustrated in Figure 6.4(a). The mean value of these measurements is marked by the red curve in Figure 6.4(b). For comparison, the measured OOR of one of the wheels on one of the locomotives (11207) used in the RIVAS field test at Dottikon, Switzerland, is shown as the orange curve. It is clear that this wheel is extreme in terms of OOR (up to 20 dB above the mean curve), and that it would be an efficient ground-borne vibration mitigation measure if such wheels are detected at an early stage and maintained (turned).

The potential insertion loss if the Re420 wheels could be maintained to a level below the ISO3095 rail roughness spectrum has been calculated, see Figure 6.5. It is assumed that a reduction in combined wheel/track irregularity level corresponds to the same level of reduction in ground-borne vibration. Based on the scatter of measured wheel OOR on Re420 and taking the 95 % percentile level as a representative spectrum, it is concluded that early detection in combination with proper maintenance actions could reduce ground-borne

vibration levels by about 10 dB. If rail roughness levels (track irregularities) are higher than the ISO3095 spectrum, the potential reduction will be lower.

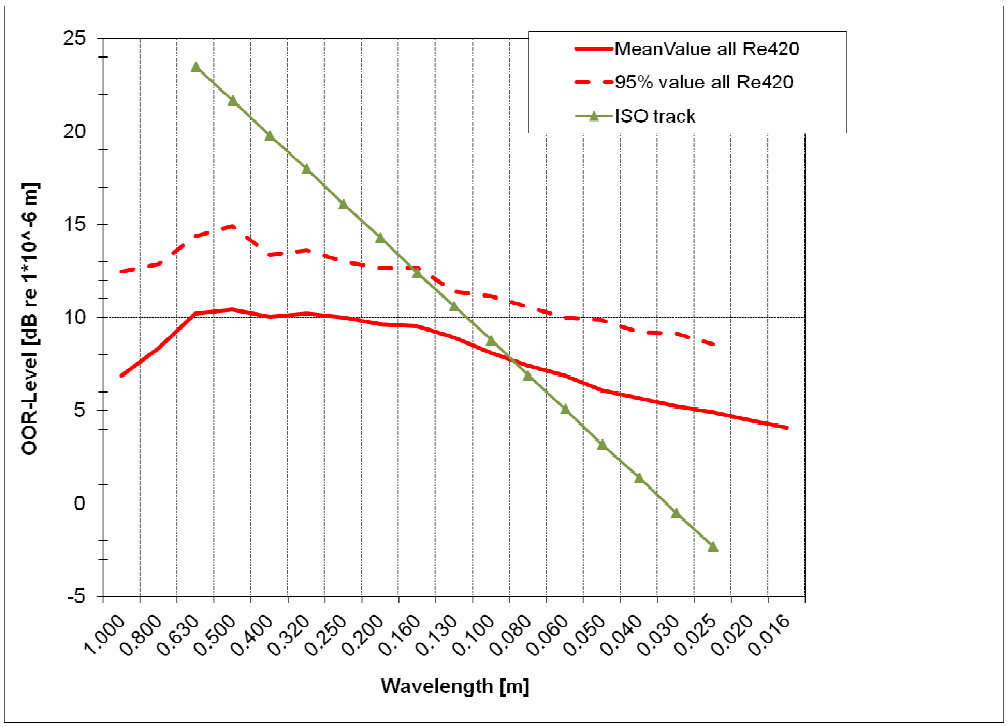


(a)

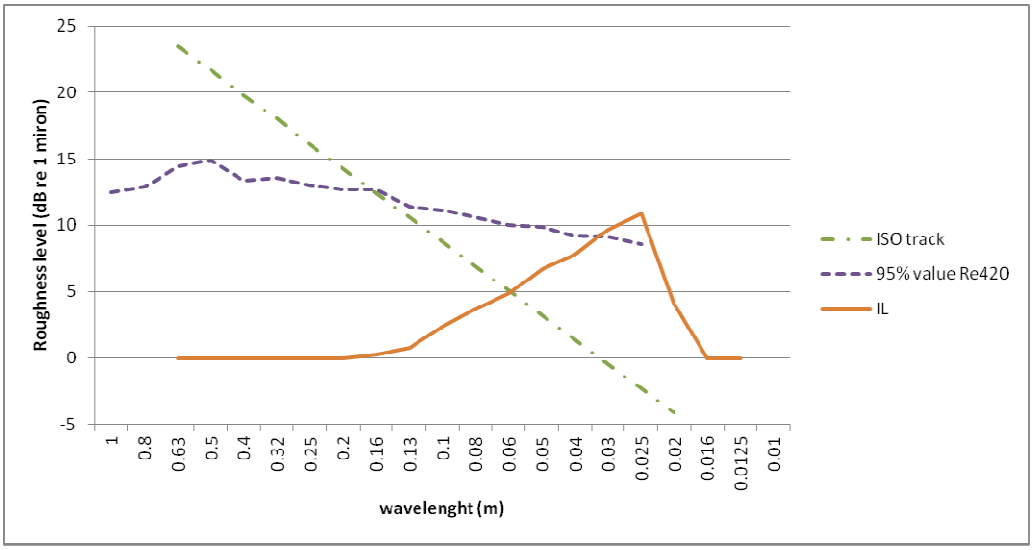


(b)

Figure 6.4. (a) OOR of Re420 wheels (database based on measurements in SBB workshop). (b) OOR of wheels measured on two Re420 test locomotives (11133 and 11207) in field test at Dottikon (not same measurement equipment as in SBB workshop). Mean and 95 % percentile of wheel OOR spectrum based on SBB Re420 database (red curves)



(a)



(b)

Figure 6.5. (a) Mean and 95 % percentile OOR level spectra measured on Re420 wheels (SBB workshop) and ISO3095 rail roughness spectrum. (b) Estimated insertion loss (IL) spectrum (orange) when reducing wheel OOR from 95% level spectrum (blue) to below ISO rail roughness spectrum (green)

6.2 WHEEL MAINTENANCE

Mitigation measures based on wheel tread maintenance to reduce ground-borne vibration excitation were analysed and tested in RIVAS D2.7 [10] and D2.9 [18]. To define the need for maintenance, the interaction between mechanisms determining OOR growth and product qualities needs to be better understood, see Figure 6.6.

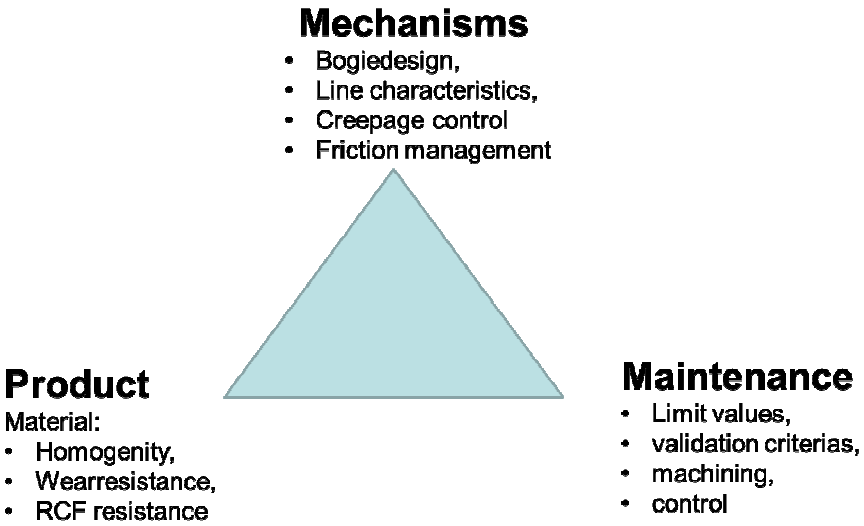


Figure 6.6. Interaction between product quality, mechanisms of OOR and maintenance

6.2.1 Maintenance methods and intervals

EN 15313 [19] defines maintenance requirements for secure interoperability of wheelsets. The standard is primarily concerned with organisational aspects and the management of wheelset maintenance. It defines the geometrical limits for safe wheel/rail (wheelset/track) interaction. Based on the application of GM/RT2466 [20], it is common practice for train operators to turn the wheels at short enough intervals to avoid either crack length or cavity length limits being reached.

Depending on application, the wheelset consists of different components. There is a difference if the wheelset is used in a freight wagon, in a coach or in a driven bogie. If additional elements are used (equipment for steering the wheelsets, lubrication, etc.) for the reduction of wear of wheel and rail, these must be considered in the wheelset maintenance. In practice, wheelsets of high-speed trains require regular attention on a wheel lathe to remove tread defects before they reach more than 0.5 mm in depth. On other train types, OOR up to 2.5 mm are removed by the wheel lathe.

A methodology for establishing a reprofiling strategy includes a two-stage process. Based on data gathered as part of normal wheelset maintenance activities, the first step involves knowledge of frequency of occurrence of tread defects and wear rate statistics. The tread defect and wear data are then used as input to a probabilistic computer simulation specifically designed to explore the impact of different wheel lathe operating strategies on wheelset maintenance costs. This simulation has to investigate the effect of a number of different

aspects of a wheelset maintenance strategy, such as re-profiling policy, parity rules and planned turns. Another approach is based on systematic preventive maintenance. Instead of applying a condition based maintenance approach with the scope to maximise reprofiling intervals (cutting depth of 6 to 7 mm), the wheels are reprofiled more frequently (e.g. after each 70 000 km) with a cutting depth of around 1 mm. As a consequence, wheelset overhaul (life) can be extended significantly due to this “reprofiling philosophy”.

6.2.2 Detection

Different condition monitoring systems are used to detect out-of-round wheels. Most condition monitoring systems are focused on the wheels and bogies since these are the vehicle components that have the largest impact on the performance and are also the major cost drivers in maintenance. There are track based detection systems and workshop based detection systems. Track based detection systems are installed in lines and are working without any speed restriction. By using wheel impact load detectors, structural health monitoring trends can be observed based on wheel impact data indicating the actual condition of the wheels. Track based detection systems in long-time commercial use are for example DAfuR in Germany and GOTCHA in Netherlands.

Workshop based detection systems allow for the detection of different wheel/wheelset data (cracks, wheel profiles, OOR, wheel diameter, wheel tread defects) but these are situated in the vicinity of a workshop. The monitoring requires reduced train speed or stand still. A sophisticated workshop based monitoring system is for example ARGUS.

It is important to share data from the detection systems with the rolling stock owner. A direct data transfer allows the vehicle owner to take immediate remedial actions. If different levels of alarm are implemented, it allows the vehicle owner to shift from corrective maintenance to conditional maintenance.

6.2.3 Machining

Clamping the wheel by a three-jaw chuck during reprofiling might lead to the generation of an initial periodic OOR with order 3. If measurements of initial OOR after reprofiling show high amplitudes, the fixation of the wheelsets or the precision of the wheel lathe have to be checked.

It is important that the reprofiling depth is sufficient such that also faults below the tread surface are removed which otherwise could promote fast OOR growth.

It seems that machining tools based on grinding (not turning), as sometimes used for trams, are not suitable to remove wheel OOR because the grinding depth is not sufficient to remove all sub-surface defects.

Ground wheel lathes are capable of adequately removing OOR (including ovality) [21].

6.2.4 Technology assessment and LCC for maintenance

Technology assessment is based on a failure mode effect analysis (FMEA) where the different tread failures are treated in a systematic way. Based on this analysis, mitigation measures are defined. These measures can for example be related to design, system maintenance, workshop based monitoring, track based monitoring and in the area of improved wheel material properties.

Faults on the running surface of wheel treads can lead to damage of components in the unsprung part of the bogie (springs, connection rods, earthing brush, speed sensors, etc), and to damage of track components (rails, sleepers, ballast). Wheel tread defects also affect the environment in terms of noise, vibration, etc. To avoid damage to critical vehicle and track components, additional costs have to be accounted for in the wheelset maintenance. The resulting savings are difficult to quantify but can be estimated based on experience, test results, theoretical investigations, etc. As long as the costs of the overall system are not known, the LCC has to be established in a pragmatic way by considering durability diagrams as a basis for the calculation. Based on these durability diagrams it can be verified if the reprofiling philosophy is correct or if it needs to be modified. The LCC is the sum of costs for reprofiling, wheel replacement including cost for material, immobilisation of vehicles, and transportation of the vehicles to the different workshops (wheel lathe, overhaul).

The best approach to improve a wheel maintenance plan is to investigate the causes for failures and its effects and have in mind LCC over a reasonable lifetime (e.g. wheelset life). Preventive reprofiling, according to train and track specifications, can then be a mitigation solution to considerably reduce vibrations but it has to be checked if other solutions could be more cost-effective.

6.3 VEHICLE MAINTENANCE

Accurate function of the axle guidance, the wheel-slide protection system, springs, dampers and other suspension elements are important primarily for vehicle dynamics, safety against derailment, running stability etc. The related maintenance practice and the monitoring methods are therefore well established.

A maintenance guideline for freight wagons is published by the VPI (Vereinigung der Privatgüterwagen-Interessenten). Specifications of freight wagons usually refer to the VPI regarding maintenance. With the delivery of a wagon, the manufacturer has to provide a maintenance plan for all important components of his wagon.

For ground-borne vibration, apart from wheelset maintenance, the following components of freight wagons have to be kept in good condition:

- Brake blocks: periodic control of function, wear and defects.
- Axle box bearing: periodic control of bearing lubrication, visual control of removed axle box bearing.
- Damper: periodic control of loss of oil, recording of characteristic diagram.
- Primary suspension: periodic control of displacement, visual control of anticorrosive coating, cracks, notches.

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8. APPENDIX A: WHEEL OUT-OF-ROUNDNESS

It was concluded in RIVAS D5.1 [4] and D5.2 [5] that wheel OOR is a key parameter in the generation of ground-borne vibration. Out-of-round wheels can also have a detrimental influence on track and vehicle components, contributing to increased risks of rail breaks, sleeper cracking, high-cycle fatigue of wheels and axles, and bearing damage [22].

Literature reviews and a classification of wheel OOR are presented in [22-24].

Examples of wheel OOR are local tread damage such as wheel flats causing severe repeated impact loads, and polygonal wheels containing a periodic deviation from the nominal wheel radius that is dominated by a few wavelengths (orders, harmonics) around the wheel circumference. A polygonal wheel leads to increased components of the dynamic vertical wheel–rail contact force at certain excitation frequencies that are determined by train speed and the irregularity wavelengths, whereas wheel flats generate impact forces with significant contributions in a wide frequency range. Impact noise and rolling noise are other consequences of wheel OOR.

There are several mechanisms or issues that may result in out-of-round wheels. Examples are irregular wear around the wheel circumference, brake system failures, vehicle curving in small radius curves, wheel machining issues, misaligned axle bore holes, surface or subsurface initiated fatigue cracking, local variations in material microstructure, plastic deformation and build-up of brake block material on the wheel tread [22]. Two important types of wheel OOR are briefly discussed below.

8.1 DISCRETE WHEEL TREAD DEFECT - WHEEL FLAT

A discrete wheel tread defect is a deviation from the nominal wheel radius on a small section of the wheel tread that for each wheel revolution may generate an impact load in the wheel–rail contact. One common discrete tread defect, the wheel flat (see Figure 8.1), is developed due to unintentional sliding (without rolling) of the wheel along the rail. The reason for the sliding may be that the brakes are poorly adjusted, frozen or defective, or that the braking force is too high in relation to the available wheel/rail friction [25]. Contaminations on the rail surface, such as leaves, grease, frost and snow aggravate the problem. As a consequence, part of the wheel tread is worn off and locally the wheel temperature is raised significantly due to the dissipated friction energy. When the wheel starts rolling again, this is followed by a rapid cooling due to conduction into the large steel volume surrounding the flat. This may lead to material phase transformation (formation of martensite) and residual stresses. The residual stresses are predominantly compressive in the martensitic region and tensile in the region surrounding the martensite [22].

The initial flat with sharp edges will soon be transformed into a longer flat with rounded edges because of wear and plastic deformation of the wheel material at subsequent impacts with the rail [26]. Further, if martensite is formed, cracks will initiate and propagate in the brittle material due to the rolling contact loading and the repeated impacts. Due to the tensile residual stresses in the surrounding material, cracks may grow to considerable depths and relatively large parts of the wheel tread may detach.

Other reasons for discrete wheel defects may be local deviations from the nominal material properties, plastic deformation, and tread metal build-up where wheel, brake block and rail debris are welded to the wheel tread due to heavy tread braking [27].

The influence of wheel flats on ground-borne vibration is discussed in more detail in RIVAS D2.2 [28].

Figure 8.1 Wheel flat. From [29]

8.2 WHEEL POLYGONALISATION

Polygonal wheels exhibit a periodic radial tread irregularity from the mean wheel radius [17]. Often the wheel out-of-roundness is dominated by a few waves that correspond to 1 – 5 wavelengths (orders, harmonics) around the wheel circumference.

The shape of the out-of-roundness can be classified by conducting a Fourier analysis of the measured radial tread irregularity and plotting the result in an order spectrum to determine the contributions of different harmonics. The corresponding wavelengths λ [m] are defined by

$$\lambda = \frac{2\pi R}{\theta}, \quad \theta = 1, 2, 3, \ldots, \qquad [1]$$

where θ is the order and R is the wheel radius. The case $\theta = 1$ corresponds to an eccentricity which is generally caused by misaligned axle bore holes or by a misalignment in the fixation of the wheel during profiling. The case $\theta = 2$ corresponds to a wheel ovality. The first three orders are illustrated in Figure 8.2. Normally, several OOR orders exist simultaneously.

Based on measurements of the radial tread irregularity on 22 wheels from Y25 bogies, the mean, maximum and minimum order spectra are illustrated in Figure 8.3 [17]. It is observed that the irregularity level of the first order (eccentricity) is typically in the order of 40 dB rel 1 μm. The irregularity level is generally decreasing with the order number except for a local

maximum in the spectrum at order 3. According to the measurements summarised in Figure 8.3, the irregularity level for orders higher than 3 is seldom higher than 20 dB rel 1 µm.

Wheels with a periodic out-of-roundness may be caused by a system interaction issue or by a wheel machining issue [30].

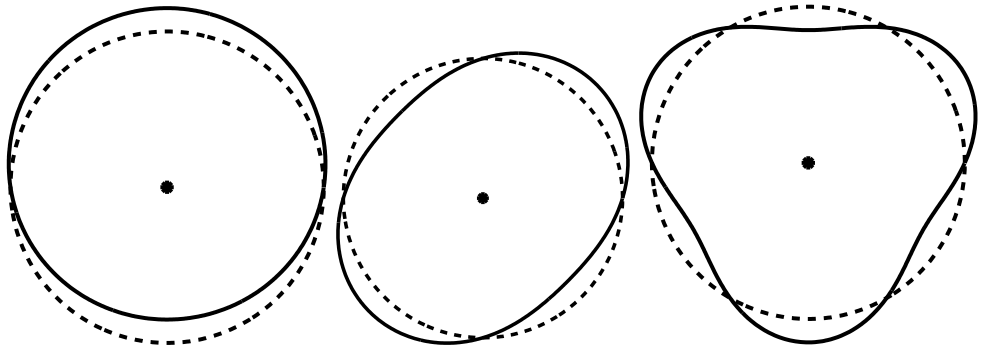

Figure 8.2 Examples of different orders of wheel out-of-roundness: (a) first out-of-roundness order (θ = 1, eccentricity), (b) second out-of-roundness order (θ = 2, ovality), and (c) third out-of-roundness order (θ = 3). From [17]

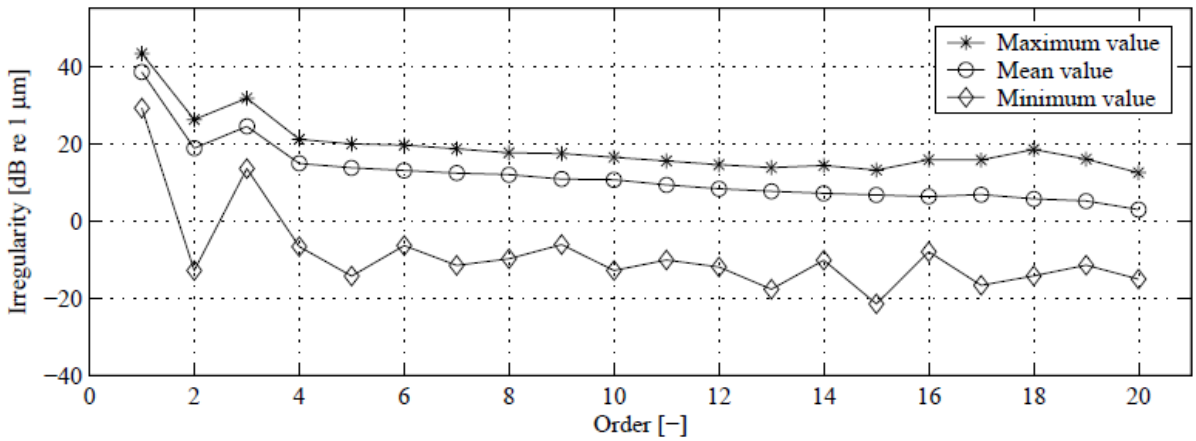

Figure 8.3. Order spectra based on measurements of radial deviation from mean wheel radius for 22 wheels in Y25 bogies. From [17]