DELIVERABLE D3.9

CONTRACT N°	TIP4-CT-2005-516420
PROJECT N°	FP6-516420
ACRONYM	QCITY
TITLE	Quiet City Transport
Subproject 3	Vehicle/infrastructure interface related noise
Work Package 3.2	Quantification of effect of mitigation measures for wheel/rail noise
	Report on mitigation measures for rails: conceptual designs

Written by	Didrik Thijssen	CDM
	Patrick Carels	CDM
	Odd Sylwan, Patrik Ragnarsson	ACL
	Patrick Vanhonacker	APT
	Geert Desanghere	AKR
	Dirk Rotthier	FDP
	Marie-Noëlle Van Leeuw	FDP
	Iossif Argiropoulos	TRAM
	Constantinos Vogiatzis	TTE
	Brantner Hans Peter	VAS

Date of issue of this report	February 16, 2006

PROJECT CO-ORDINATOR	Acoustic Control	ACL	SE
PARTNERS	Accon	ACC	DE
	Akron	AKR	BE
	Amec Spie Rail	AMEC	FR
	Alfa Products & Technologies	APT	BE
	Banverket	BAN	SE
	Composite Damping Material	CDM	BE
	Havenbedrijf Oostende	HOOS	BE
	Frateur de Pourcq	FDP	BE
	Goodyear	GOOD	LU
	Head Acoustics	HAC	SE
	Heijmans Infra	HEIJ	BE
	Royal Institute of Technology	KTH	SE
	Vlaamse Vervoersmaatschappij DE LIJN	LIJN	BE
	Lucchini Sidermeccanica	LUC	IT
	NCC Roads	NCC	SE
	Stockholm Environmental & Health Administration	SEA	SE
	Société des Transports Intercommunaux de Bruxelles	STIB	BE
	Netherlands Organisation for Applied Scientific Research	TNO	NL
	Trafikkontoret Göteborg	TRAF	SE
	Tram SA	TRAM	GR
	TT&E Consultants	TTE	GR
	University of Cambridge	UCAM	UK
	University of Thessaly	UTH	GR
	Voestalpine Schienen	VAS	AU
	Zbloc Norden	ZBN	SE
	Union of European Railway Industries	UNIFE	BE

PROJECT START DATE	February 1, 2005
DURATION	48 months

Project funded by the European Community under the
SIXTH FRAMEWORK PROGRAMME
PRIORITY 6
Sustainable development, global change & ecosystems

This deliverable has been quality checked and approved by QCITY Coordinator
Nils-Åke Nilsson

TABLE OF CONTENTS

0 EXECUTIVE SUMMARY

0.1 OBJECTIVE OF THE DELIVERABLE

This report presents the multiple concepts for mitigation of rail noise, which were identified or developed by the QCITY WP3.2.2 partners. The report describes the solutions with their technical performance and user cost.

The report appears 12 months after the start of the QCITY project. It is a first milestone in a process, which will consecutively lead to a ranking and finally the realisation of test applications of the most effective mitigation measures.

QCITY aims to constitute a practical toolbox with effective trouble shooting solutions for city acoustic hot spots with optimal airborne noise reduction performance and price.

A state of the art document on noise mitigation at wheel/rail level has been established (chapter 10), which has been transferred to SP1 and SP6 for use in the Action Plans.

0.2 STRATEGY AND DESCRIPTION OF THE TECHNIQUES USED

Based on specific experience and know how, the various solution domains were allocated to the partners. After analysing the state of the art new, concepts were screened for effectiveness and costs, keeping in mind a short-term, relatively low cost and general application perspective.

The following domains were identified, investigated and documented:

Special track work

Impact noise at turnouts can be very disturbing. The airborne noise increases with on average 10 dB(A) in comparison with tangent rolling noise at the same speed.

Within the EC Turnouts research project, solutions have been designed and some already tested to reduce the noise emission during turnout passage.

The two major European groups manufacturing turnouts for urban rail have now solutions available to limit the noise increase coming from turnouts to very low values, e.g. 1 dB(A). Information on more solutions are under design and test within the Turnouts program and will be transferred to the QCITY project as soon as they become available.

It is not necessary to spend more resources to this topic within the QCITY project.

Tuned rail dampers

A solution to reduce noise radiation of rails is the application of tuned rail dampers. The application of rail dampers will attenuate the rail vibrations and reduce noise radiation from rails, sleepers and to lesser extend also of the wheel.

The effect on noise reduction of tuned dampers is however mitigated. Noise reduction of 2 dB(A) seems to be a reasonable achievable value but with a large spread. For planning purposes, a cost of 250 €/lm-rail is commonly used as a value for the system and its installation. The combination of wheel mitigation measures with tuned dampers can possibly give more predictable and promising perspectives. An ACL report on rail damping combined with wheel damping investigates this matter.

Embedded rails

As discussed in WP3.1, embedded rail systems reduce the global rail emission by 3 to 4 dB(A) in comparison with discrete direct fixation systems on a concrete slab.

In comparison with open ballasted track, the embedded rail systems are still 1 to 3 dB(A) noisier since they have, in general, hard road surfaces between and next to the rails, giving yield to reflections and almost no absorption.

In order to get optimal embedded rail systems (from a noise emission point of view), the embedded rail systems have to be combined with a road surface with high acoustic absorption.

This development will yield a noise reduction of about 3 dB(a) in comparison with open ballasted track. It is the objective to develop such a road surface (between and next to the rails), which is highly absorptive, maintenance free and able to carry road traffic.

Broadband rail damping

The different broad band rail dampers (two of them are based on tuned damping, although showing broadband characteristics) and their effect on the roar noise emission are studied on straight track only. Different rail profiles are briefly presented and an estimation of the noise reduction and the cost is given for each solution. The estimation is made for rails resting on soft to medium soft rail pads by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness.

Predictions are also made for a combination of the rail solution and one selected wheel damper (‘shark’s fin’), indicating the noise reduction anticipated by combining wheel and rail damping.

All solutions based on constrained layer damping (CLD) are in a design phase, as they represent new rail designs as well as new innovative concepts. Therefore, a lot of practical issues are still to be solved.

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	Track Cost [€/m]	Limitations/ Critical Matters
CLD-Rail Type 1	5	2	200	Rail/Visco stress
CLD-Rail Type 2	6	2	300	Visco stress
CLD-Rail Type 3	7	3	1000	Prize/Size

Corus rail damper	6	3	500	-
CDM-ABSO-RAIL	6	3	500	-

Table 1

Additional cost for wheel dampers ('shark's fin') is about 500 €/wheel.

All reductions are fairly equal. The predicted total noise reductions for combined wheel and rail measures are **5 – 7 dB(A)**, if the contribution of other noise sources than the wheel/rail-system are ***not*** too significant (e.g. traction gears). Thus, a combination of wheel and rail damping might have a good effect on the rolling noise.

When extra costs for the broad band rail dampers are considered, the evaluated measures can be ranked as follows (wheel and rail measures):

1. CLD-Rail Type 1 or CLD-Rail Type 2 (80-130 €/m/dB(A));
2. Corus rail damper or CDM-ABSO-RAIL (160-170 €/m/dB(A));
3. CLD-Rail Type 3 (320 €/m/dB(A)).

Structure born noise of steel bridges

The total sound pressure produced by trains on steel bridges can be 5 to 15 dB higher than on land based ballasted track with timber sleepers. Each existing steel railway bridge is specific and will require a noise mitigation approach considering all of the bridge infrastructural elements. The recommendations focus on solutions regarding the track of existing steel bridges only.

Effective track mitigation measures for bridge structure born noise of existing steel bridges are:

- reduced rail support stiffness;
- resilient rail fixation;
- increased mass of track support;
- low rail roughness (regular grinding);
- move of special track work (rail discontinuities) from the bridge to the mainland.

These measures allow to obtain good technical (in general reductions of about 8 dB) and economical results for existing steel bridges.

Special rail profiles

With the new designed rail profile VA71b it is possible to decrease the noise emission for louder car classes up to 2 dB(A). In combination with higher rail pad stiffness, it is possible to decrease the noise emission even further. Noise reduction up to 4 - 4.5 dB(A) are possible.

All solutions are in a design phase, as they represent new rail designs as well as new innovative concepts. Therefore, a lot of practical issues are still to be solved.

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	Extra Cost [€/m]	Limitations/ Critical Matters
Reduced stiffness	8	6	100	Manufac/Stress
Saddle profile rail	7	5	1000	Prize/Stress
VA71b + Stiff pads	Not estimated	6	[1]20	Stiff pads, wear

Table 2

When extra costs for the special rail profiles are considered, the evaluated measures can be ranked as follows:

1. VA71b + Stiff pads (3.6 €/m/dB(A));
2. Reduced stiffness (18 €/m/dB(A));
3. Saddle profile rail (200 €/m/dB(A)).

Anti corrugation systems

Corrugation in curves can be strongly reduced when installing new track or renewing existing track by avoiding low damped track resonances, which correspond with corrugation wavelengths between 2 cm and 20 cm. This can be done by using high damped embedded tracks or by using very resilient track in curves (vertical dynamic stiffness typically lower than 10 kN/mm/m rail).

Furthermore it is mandatory to impose, control and enforce small diameter differences on the wheels of the same wheelset (typically lower than 1 mm).

For existing systems, the best solution is rail grinding, which is very effective in reducing noise (up to 10 dB(A)) and controlling corrugation. Cost is estimated at 4000 €/km track. But rail grinding treats corrugation symptoms, it does not eliminate its cause.

It has to be noted that in case all resonance excitation (track and wheelset) is avoided during curve negotiation, rail wear will still occur due to the friction forces. This rail wear (quasi-random patterns) can be reduced using harder rail contact areas. The use of friction modifiers is also promising. Special attention has to be paid to harder rail and lubrication, which can lead to increase of rolling contact fatigue.

Anti squeal

This chapter summarises the results of the research carried out under the EC contract BRPR-CT97-0477, "*Squeal noise reduction in urban transport by rail treatment*" and the results of recent findings about mitigation methods for squeal noise.

A mathematical model for squeal noise calculations is presented. In the scope of the QCITY project, this model is integrated within the SYSTUS finite element program and

[1] Extra costs for maintenance is not included (e.g. more frequent inspections and rail grinding).

applied to some case studies where anti-squeal measures have been applied. The model and associated software have been validated in this way. Squeal mitigation measures have been identified.

Emphasis is put on the validation of two mitigation measures against squeal noise: top of rail friction modifiers and laterally resilient rail fasteners with low damping.

For friction modifiers, a low friction coefficient is required in order for the friction modifiers to be effective. This is the major challenge in this development, together with a uniform application of the friction modifier product on the rail surface.

The costs of a rail treatment against squeal noise are budgeted as follows, considering a 100 m long curve:

- top of rail friction modifier: installation and equipment costs of unit with 6 nozzles: ± €30 000 (plus consumption of friction modifier product, estimated at €150/month);
- laterally resilient rail fasteners: cost of 320 rail fasteners: €32 000 (extra material cost in comparison with classical rail fastener).

0.3 PARTNERS INVOLVED AND THEIR CONTRIBUTION

	Topics WP3.2.2	Partners
1.	Special track work (crossings, turnouts ...)	LIJN + FDP + APT
2.	Tuned rail dampers (open track)	CDM
3.	Embedded rails	FDP + CDM + APT
4.	Broadband rail damping combined whit wheel damping	ACL
5.	Structure born noise of steel bridges	TRAM + TTE + CDM
6.	Special rail profile (Saddle rail, hollow rail, low profile rail)	VAS + ACL
7.	Anti corrugation systems	AMEC + FDP
8.	Anti squeal	APT

0.4 CONCLUSIONS

The partners involved in the WP3.2.2 concerning noise mitigation measures for rail have identified, analysed qualitatively and quantitatively a number of practical solutions to reduce track noise. Most solutions are an evolution of the state of the art. Others are more innovative but still need some development for a quick practical use. Performance and cost of the mitigation measures vary. This information will in an ulterior phase of the project be used to establish an effectiveness ranking of the measures.

0.5 RELATION WITH THE OTHER DELIVERABLES

The WP3.2.2 mitigation solutions as presented in this D3.9 deliverable have a relation with:

- WP3.1 provided input WP3.2

- WP3.2.1. - D3.8 on mitigation measures for wheel. In some cases rail mitigation measures will be more effective when used in combination with wheel measures and vice versa.
- WP3.2 provides input to the following WP:

  WP5.1 – Antwerp - LIJN
  WP5.2 – Athens - TRAM
  WP5.3 – Brussels – STIB
  WP5.7 – Ostend - HOOS
  WP5.9 – Stockholm – BAN

1 SPECIAL TRACKWORK (CROSSINGS AND TURNOUTS)

1.1 INTRODUCTION

In the sixth Frame Project FP-6-505592 'Turnouts' which started up in 2003, new concepts for turnouts in urban rail transit infrastructure are studied.

The objective of the 'Turnouts' project is to improve the vehicle-track interaction in the turnout system for urban rail transit, and therefore to improve their efficiency, enhance their safety levels, reduce the required maintenance. An important technical objective is to reduce the emitted noise in comparison with classical turnout systems by at least 6 dB(A).

This report transfers knowledge gained in the Turnouts project to the QCITY project.

1.2 EXISTING SYSTEMS – STATE OF THE ART

If trains never needed to pull into or out of sidings, enter or leave track, cross other track, or cross over to another track, special track work like turnouts would be unnecessary. But trains do all of those things, and in the process, hammer away at frogs, contributing to the short service life of one of the maintenance of wayside's most expensive single items.

That's why this project's resources are focused on a challenging program to improve the efficiency, performance and safety of turnouts.

Since the magnitude and the nature of dynamic forces within the track/train interaction is directly related to the material endurance, the design philosophy of advanced design turnouts strives to reduce dynamic impact.

The QCITY as well as the TURNOUTS project deal with low axle load and low speed systems (urban rail transit).

The problems with turnouts in urban rail networks are specific and the cures are different from those applicable to turnouts in high speed or normal passenger and freight railways.

1.3 TURNOUTS: GENERAL CONCEPTS

We use the word 'turnout' to describe the junctions in trackwork where lines diverge or converge so as to avoid the terms 'points' (UK) or 'switches' (US), both of which can be confusing. In the railway 'trade', turnouts are referred to as 'switch and crossing work'.

The moving part of the turnout is the switch 'blade' or 'point' or 'tongue', one for each route. The two blades are fixed to each other by a coupling bar to ensure that when one is against its stock rail, the other is fully clear providing room for the wheel flange to pass through cleanly. In both sides of the crossing area, wing and check rails are provided to provide guidance to the wheelsets through the crossing.

The crossing or frog can be cast or fabricated.

The two main types of frogs are:

- cast manganese frogs in two types, i.e. leg type cast manganese frog or solid frog and rail bound crossing;
- welded frogs.

Cast Manganese Frogs (see figures 1 & 2)

Rail bound manganese frogs are used at many transit systems, and serve as the baseline for the purpose of discussing noise control. They include a fixed gap and conventional rail wrapped around a manganese centre. This places the wear-resisting element at the discontinuities of the frog, which reduces wear, but generates the impact noise when wheels traverse the fixed gap.

Figure 1

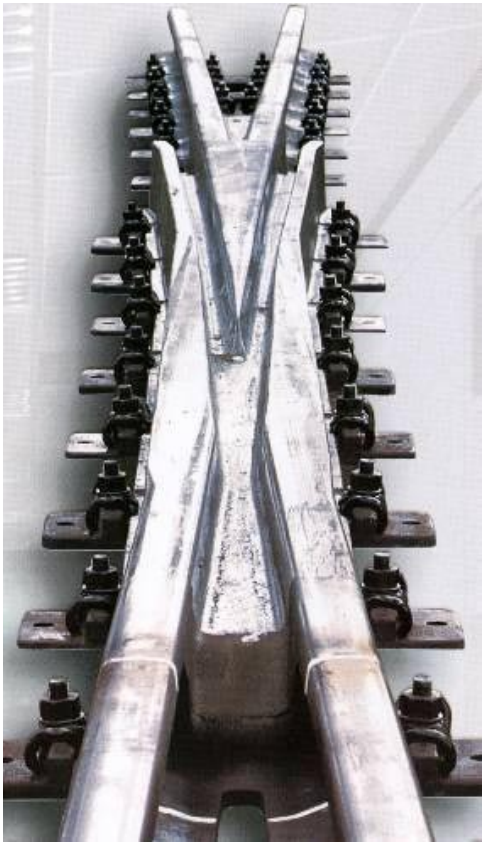


Figure 2

Welded Frogs (see figure 3)

Welded frogs are fixed gap frogs constructed from standard rolled rail. The welded frog holds together two tapered rails with a continuous longitudinal weld.

Figure 3

Impact noise generation occurs as the wheel traverses the frog. Impact noise from special trackwork is very noticeable to wayside receivers and transit passengers, with A-weighted maximum noise levels roughly 10 dB (A) (fast sound level meter response) greater than levels at tangent track with ground rail (see results obtained in WP3.1). Special trackwork noise will generally be greater than the one emitted by rail joints, due to the significantly larger gap that must be traversed.

1.4 TURNOUT: SPECIAL CONCEPTS

Moveable Point Frogs

Moveable point frogs, also known as swing nose frogs, are perhaps the most effective at eliminating the impact noise associated with fixed gap frogs. Modern moveable point frogs have been developed for high-speed railways. The gap of the frog is eliminated by laterally moving the nose of the frog in a direction corresponding to the direction of train travel. The moveable point frog generally requires additional signalling,

Flange Bearing Frogs

Flange bearing frogs provide support to the wheel flange while traversing the frog gap in embedded track. A properly installed frog supports the flange, maintains the wheel height through the frog, and reduces the impact forces associated with the wheel traversing the gap. The depth of the flange support below the top of rail is critical in providing a smooth transition through the gap. If this support is too high or too low, the transition is not smooth and the impact noise is not eliminated.

Therefore, flange bearing frogs do not necessarily eliminate impact noise and vibration, and their effectiveness depends on the degree of frog and flange wear.

Welding & Grinding

Maintenance of frogs by welding and grinding the frog point results in lower impact forces, and thus lower impact noise levels and wheel and frog wear.

1.5 NOISE MITIGATION

Noise mitigation goes in parallel with reducing impact forces. In his way, also a reduction of bogie shock and vibration may be expected, which may lead to reduced bogie maintenance.

There is a list of possible mitigation measures of different nature, which are studied within the Turnouts project. These measures are based on know how and expertise of the participating partners:

- Elastic mounting of turnout.
- Adding mechanical damping to the turnout.
- Coating of turnout components with insulating and damping material.
- Welded switches with low rail profile whereby the half switches are embedded in a polymer component (1), or whereby the half switches are prefabricated,

embedded in a polymer component and integrated in a concrete structure (2) with complete suppression of bearing plates, attachments, coupling bars and sleepers.

- New rail types in grooved or vignol rail.
- Integration of complete turnout in concrete structure.
- New materials including austenitic manganese steel, low carbon austenitic manganese steel, bainitic steel.
- Longitudinal profile correction (grinding, recharging or surfacing) in such a way that a new or an existing turnout yields a longitudinal profile which matches the ideal running profile. The actual running profile is to be determined by running a measurement lorry over the turnout. This measurement lorry has a contact profile which exhibits the same vertical displacement as the wheel of a train running over the turnout. The deviation of the measured actual running profile from the ideal profile is then corrected by railhead surfacing and successive grinding of the rail (frog section).
- Reducing of production tolerances, enhancing production processes.
- Improved wheel/rail contact by redesigning running surface cross section and by rail reprofiling.
- Maintenance optimisation of used wheel treads.
- Frog for tramway with moveable point.
- Switch point design with head hardened vignol rails for embedded tram turnouts.
- Use and optimisation of longitudinally ramped flangeway corners (turnouts with high angle crossing).
- New welding techniques for joining rail and frog legs.
- New types of road-pavement (concrete, asphalt, ...).
- Combination of new track-treatment and road-pavement.

The partners including the two major European suppliers of turnouts (Jez belonging to the VAE group and Cogifer belonging to the Vossloh group) have pooled their knowledge into the design of different solutions based upon the above described concepts. Some designs have resulted into prototypes, which are under test. It can already be concluded that turnout solutions are available which meet the objectives of the turnouts project.

Cogifer has designed and installed a turnout which in completely embedded in a concrete slab (without rail fasteners) into the STIB tram network in Brussels. The resulting noise levels measured at 7.5 m from the track centre line during tram straight passage over the turnout are only 1dB(A) higher at the frog section in comparison with a tangent section at the same vehicle speed (see figure 4 & 5).

Figure 4

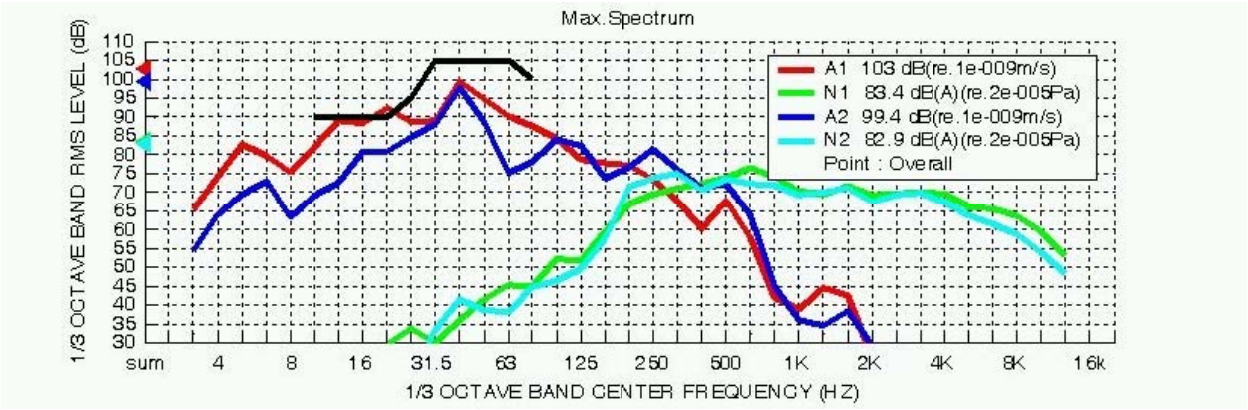


Figure 5

N1 is noise spectrum during frog passage, N2 is noise spectrum on tangent track

Jez has designed a turnout with moveable point frog for tram. This prototype is to be installed in the tram network of De Lijn in Antwerp. Noise measurement results will be available within 6 months.

An elastically mounted turnout on ballast (with ballast mat and elastomer undersleeper pads) has been installed in the network of STIB (Brussels) using a standard turnout. Noise and vibration reduction of in average 8 dB has been measured, meeting the goals of the turnouts project (see figures 3.3 – 3.5).

Figure 6

Figure 7

Figure 8

1.6 CONCLUSION

Impact noise at turnouts can be very disturbing. The airborne noise is increase with on average 10 dB(A) in comparison with tangent rolling noise at the same speed.

Within the EC Turnouts research project, solutions have been designed and some already tested to reduce the noise emission during turnout passage.

The two major European groups manufacturing turnouts for urban rail have now solutions available to limit the noise increase coming from turnouts to very low values, e.g. 1 dB(A). Information on more solutions under design and under test within the Turnouts program will be transferred to the QCITY project as soon as they become available.

It is not necessary to spend resources to this topic within the QCITY project.

2 TUNED RAIL DAMPERS

2.1 INTRODUCTION

Railway induced vibrations are a growing matter of environmental concern. In general about 20 % of the population is in some way affected by rail noise.

Between 40 km/h and 250 km/h, rolling noise is the dominant train noise source [2.1]. Rolling noise emanates from the roughness of the wheel and the rail. Wheel, rail and sleepers vibrate and radiate noise. QCITY WP3.2 focuses on noise mitigation measures for the rail.

A solution to reduce noise radiation of rails is the application of tuned raildampers. Raildampers are preformed elements made of elastic material containing steel strips or plates. The dampers are placed against the sides of the rails, dampening the vibration of the rails when a train rides over them. The application of rail dampers will attenuate the rail vibrations and reduce noise radiation from rails, sleepers and to lesser extend also of the wheel.

By designing the dampers as a mass spring-system, their characteristics can be adapted to the dynamic properties of the rails. Since the late nineties a lot of experience has been gained with these kinds of dampers.

Figure 9 Examples of the application of tuned raildampers

2.2 STATE OF THE ART

2.2.1 Principle

The concept of tuned rail dampers is based upon a mass-damper vibrating at the pinned-pinned frequencies of the rail (preferably first p-p around 800-1000 Hz.)

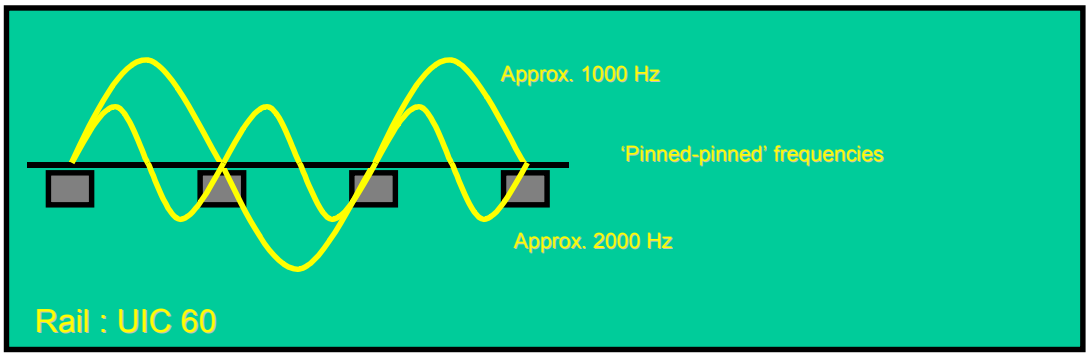


Figure 10 Pinned-pinned rail eigenfrequencies

This spring-mass system acts as a dynamic vibration absorber. To work properly, the mass and the stiffness must be chosen in such a way that the eigenfrequency of the damper is the same as the vibration frequency of the rail [2.3]. The damper has to be placed where movement of the rail is maximal.

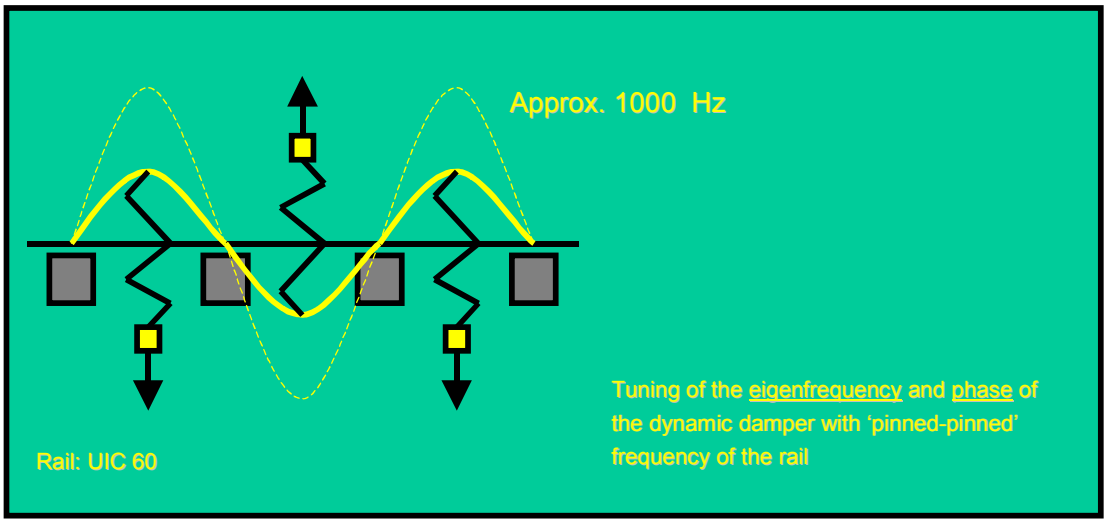


Figure 11 Rails with tuned rail dampers

2.2.2 Designs on the market

Over the years several test projects with the following tuned dampers have been realised:

Constrained damping layer system with steel plate in rail web (Stegdämpfer - Vossloh)

Figure 12 Vossloh rail damper

Continuous double tuned rail damper glued in rail web (Silenttrack – Corus and Edilon)

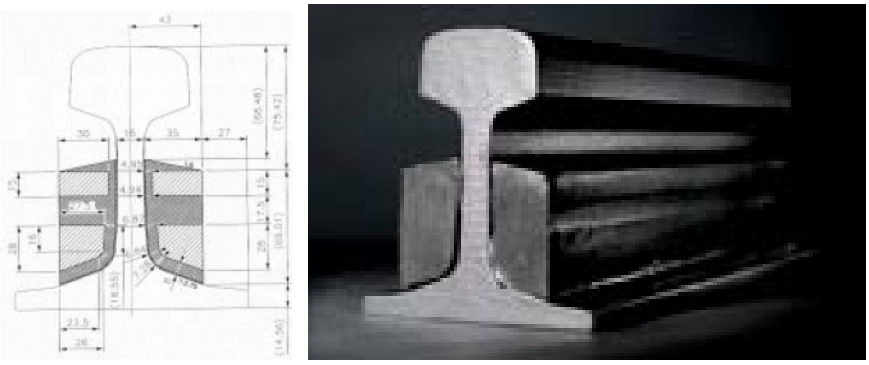


Figure 13 Corus rail damper

Light weight gun-shot tuned rail damper with bending plate (ABSORAIL-CDM)

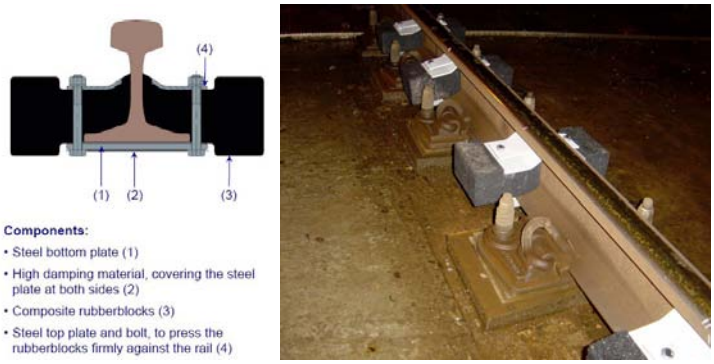


Figure 14 CDM rail damper

Quiet-stone tuned rail damper (Sound Absorption U.K. Ltd)

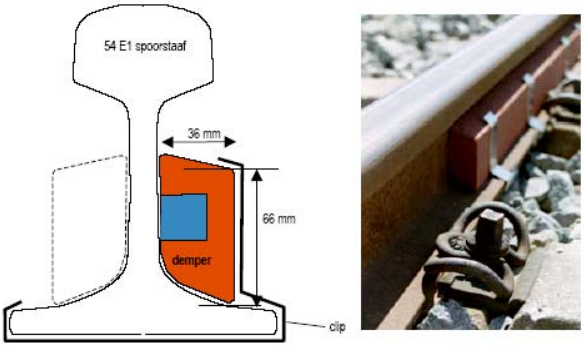


Figure 15 Quiet-stone rail damper

Multidirectional tuned rail damper (lateral absorber shoe) (Schrey&Veit)

Figure 16 Schrey&Veit rail damper

2.2.3 Performance & cost

The interest for tuned rail dampers grew in the late 1990's as a possible alternative to noise screens. This new mitigation measure seemed suitable to reduce noise along trackside when only small gains were to be pursued.

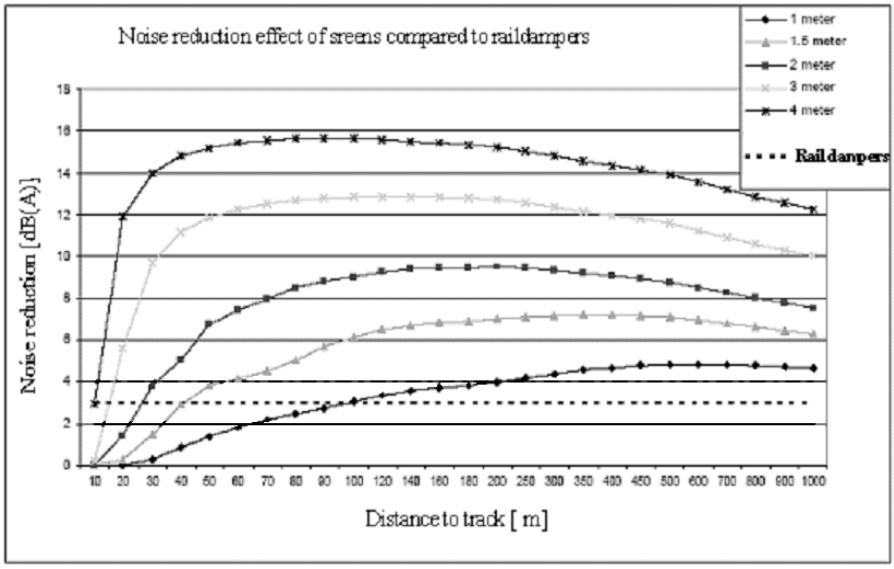


Figure 17

Noise reduction of rail dampers compared to noise screens [2.2]

The noise mitigation effect of rail dampers is positively correlated with the increase of rail roughness [2.3], [2.4]. The rougher the rail the higher the noise reduction effect of raildampers. The rail material and type of rolling stock have however no influence on the effectiveness of the dampers [2.5].

Measurements show that the track decay rate significantly improves when dampers are installed.

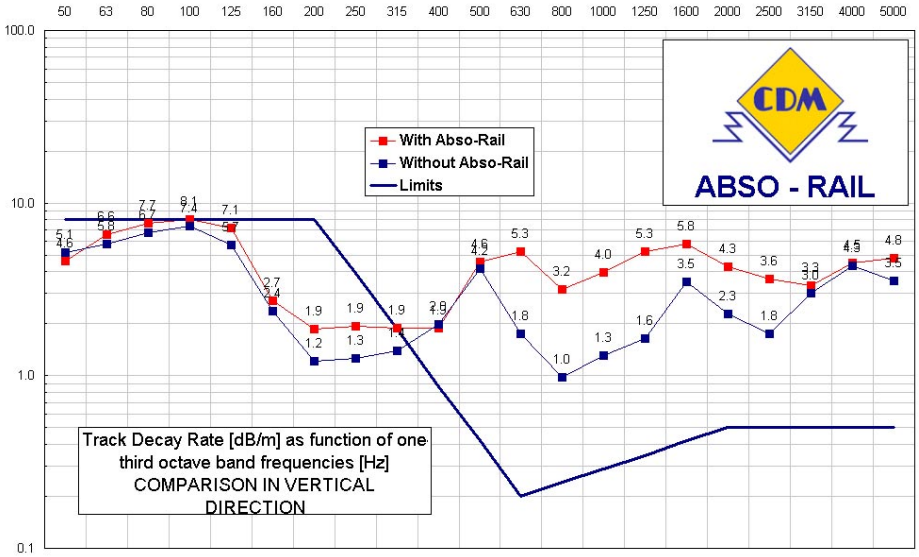


Figure 18

Example of CDM track decay measurements in Antwerp station

The effect of tuned rail dampers on noise reduction is however mitigated. A noise reduction gain of 2 dB(A) seems to be a reasonably achievable value but with large spread, resulting only in a positive effect when looking at Leq-day.

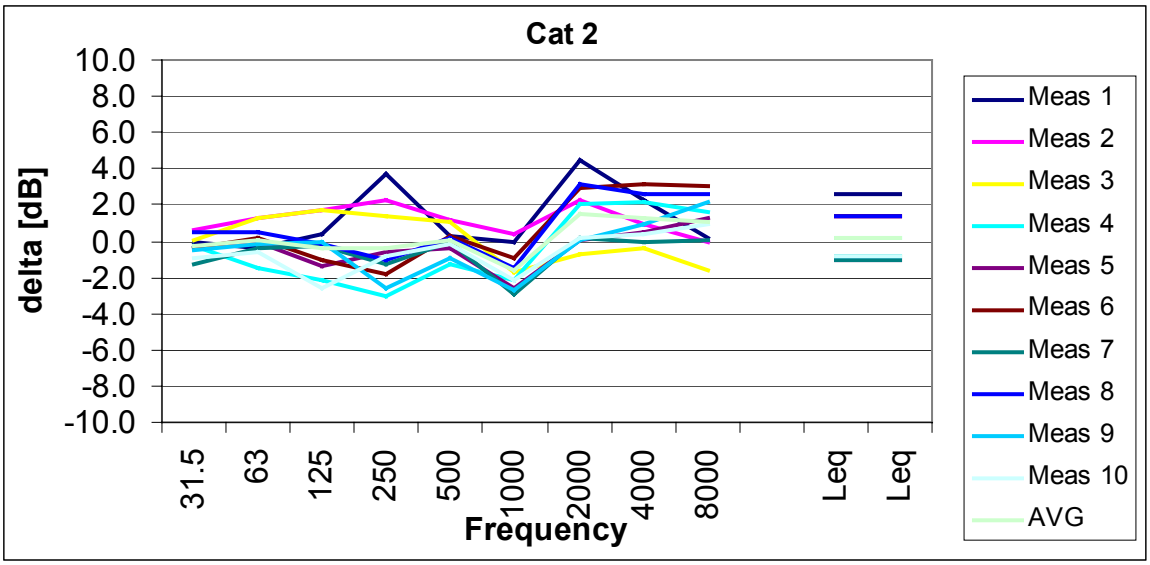


Figure 19

Example of Leq measurements, NS Rotterdam stadion

The 5th Framework Program STAIRRS [2.6] recognised rail dampers as a solution to gain approx. 2 to 3 dB(A) overall noise. Extensive research in US (Pueblo – Test Center) has shown however mitigated conclusions and small interest in using this type of equipment (results < 2 dB(A)). The effectiveness of tuned rail dampers in relation with a mixed rolling stock is also questioned.

Cost for the system and installation is function of its weight and can vary from 150 €/lm-rail up to 600 €/lm-rail. As an indicative value for planning purposes, a cost of 250 €/lm-rail is commonly used.

The STAIRRS Program also indicates the effectiveness of rail dampers in comparison to some other mitigation measures. Figure 12 shows the cost of tuned rail dampers and their impact on the reduction of the number of people exposed to a noise level >60 dB.

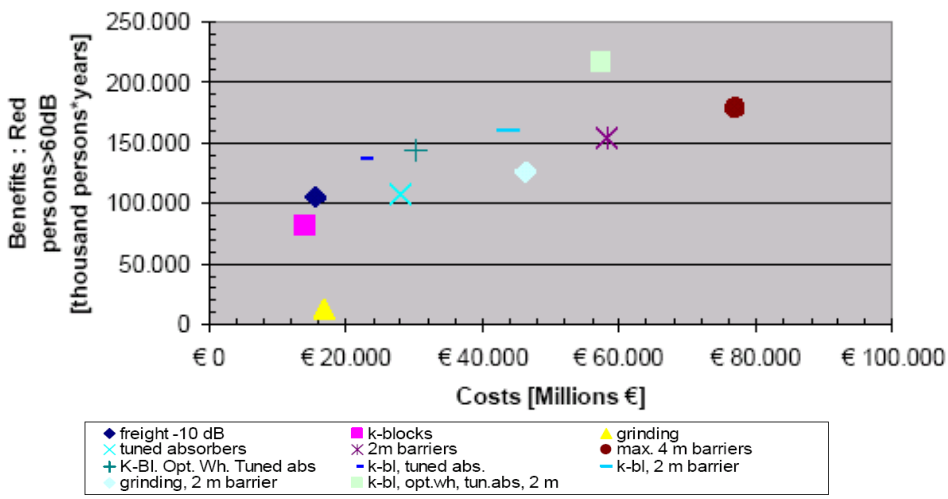


Figure 20

Cost and effect of mitigation measures when applied on a European scale (21 countries)

2.3 NEW CONCEPTS

From the point of view of QCITY, aiming at achieving noise mitigation with impacts >5 dB(A), tuned rail damper state of the art does not allow to take a clear position.

No other new tuned rail damper concepts have come up so far.

A combination of wheel mitigation measures with tuned dampers can possibly give more predictable and promising perspectives. Comparative tests should be conducted to evaluate this option. This testing is part of the ACL report on the combined use of rail dampers and wheel mitigation measures.

2.4 CONCLUSIONS

A solution to reduce noise radiation of rails is the application of tuned rail dampers. The application of rail dampers will attenuate the rail vibrations and reduce noise radiation from rails, sleepers and to lesser extend also of the wheel.

The effect on noise reduction of tuned dampers is however mitigated:

- Europe recognises rail dampers as solution with a potential of 2 to 3 dB(A);
- on the other hand US R&D has no interest in this technology;
- rail dampers are OK to increase the track decay rate;
- noise reduction of 2 dB(A) seems to be a reasonable achievable value but with a large spread.

For planning purposes a cost of 250 €/lm-rail is commonly used as an indicative value for the system and its installation A combination of wheel mitigation measures with tuned dampers can possibly give more predictable and promising perspectives. The ACL report on rail damping combined whit wheel damping investigates this matter.

The decision of possible use of tuned rail dampers as an effective mitigation solution will be taken at the QCITY M18 meeting.

2.5 REFERENCES

[2.1] ir. Ph. H van den Dool, Holland Railconsult, Postbus 2588, 3500 Utrecht, Nederland Raildempers in de praktijk, page 1, 2004.

[2.2] Gilles Janssen en Peter van Tol, Pro Rail. ProRail-eindrapport slijpen en dempen Noise Innovation Programme, 20/05/2005 http://www.innovatieprogrammageluid.nl/

[2.3] J. Maes*, H. Sol*, P. Carels***, P. Guillaume** Vrije Universiteit Brussel, Pleinlaan 2, B-1050 Brussel
* Dept. Mechanics of Materials and Constructions
** Dept. Mechanical Engineering
*** CDM NV Reutenbeek 9-11, B-3090 Overijse Dynamic damping at pinned-pinned frequencies

[2.4] Udo Lenz Schallimmissionen, Lärmminderung durch kontinuierliche Schienenlagerung, Eisenbahningenieur (56) 12/2005 – page 15 -16

[2.5] Jan Willem Lammers, State of the art: Rail dampers, Noise innovation program (IPG) - Rail project 2.2.1 - 3/2005.

[2.6] Aude Lenders, 5FP STAIRRS Program results - November 2002, Cesse –ULB, http://www.ulb.ac.be/ceese/nouveau%20site%20ceese/documents/stairrs_presentation.pdf

3 EMBEDDED RAIL SYSTEMS

3.1 INTRODUCTION

In this report, only tram track systems belonging to the embedded rail system family are considered which are sharing the right-of-way with road vehicles (cars, busses, trucks, …). This is the most common situation in an urban tram network.

First existing systems are discussed. Then the acoustic behaviour of the embedded rail system is discussed and conclusions are drawn to come up with embedded rail systems which are up to 3 dB(A) less noisy (rolling noise in tangent track) than open ballasted track systems.

3.2 EXISTING SYSTEMS – STATE OF THE ART

Types of ballastless tram infrastructure with shared right-of-way (tram-road)						
Track Family		Discontinuous supported rail	Continuous supported rail			
Fastening		Mechanical	Mechanical	Mechanical	Fastenerless	
Name		Slabtrack with sleepers or direct fixation	Tiebar Slabtrack	Hybrid	Poured embedded track	Precoated embedded track
Some pictures of sites for the different systems (pictures by courtesy of CDM)		type 1 	type 2 	type 3 	type 4 	type 5 
Other names - Trademarks (not limited)		Many variations exist, all depending on the type of sleeper block - attention: sleeper may also be direct fastener plate CDM-DPHI-E FDP - DS-ISO-RAIL	Thyssen-Krupp system CDM-Comfotrack	FDP hybrid system - new lines: poured in concrete on jobsite - existing lines: prefabricated concrete slab	Edilon - Corkelast (NL) CDM-Porotrack (BE) Polyplan (DE) ALH (UK)	CDM-Prefarail "jacketed" track
Track elements	for the fastening	main railway line mechanical fasteners (Nabla, Pandrol, Vossloh...) - every 600/750 mm.	main railway line mechanical fasteners each 1000/3000 mm.	main railway line mechanical fasteners each 900 mm.	by a liquid poured elastomer product (PUR) in the reservation in the concrete slab, that polymerises in situ.	by the "jacket" shape and friction between jacket and slab
	to get the proper elasticity	by railpad - elasticity is limited by the deflection being able to be taken by fastener and gauge widening.	by railstrip - elasticity is limited by the deflection being able to be taken by fastener and gauge widening.	by railpad and by elasticity of poured elastomer	by resilient rail strip and elasticity of PUR-elastomer	by adapted resilient rail strip integrated in the "jacket" (overall stiffness comes partly from "jacket shape" and supporting resilient strip stiffness)
	to protect the embedded rail from environment	by "webchambers" adapted to the type of rail and sleeper - PE-foam or rubber, a finishing longitudinal rail joint is used in general	by "webchambers" adapted to the type of rail and sleeper - PE-foam or rubber, a finishing longitudinal rail joint is used in general	the poured elastomer serves this purpose, a longitudinal rail joint is used	the PUR elastomer serves this purpose	the elastomer "jacket" serves this purpose

Types of ballastless tram infrastructure with shared right-of-way (tram-road)

Track Family			Discontinuous supported rail	Continuous supported rail			
Fastening			Mechanical	Mechanical	Mechanical	Fastenerless	
Name			Slabtrack with sleepers or direct fixation	Tiebar Slabtrack	Hybrid	Poured embedded track	Precoated embedded track
Technical Description	Installation	Bottom up	the "frames" (=sleepers with rail mounted) are mostly aligned by use of integrated rods in sleeper blocks - TOR = consequence of sleeper or direct fastener plate alignment)	the "frames" (rail tied by tiebars) are mostly aligned by shimming the tiebars - TOR = consequence of tiebar alignment!)	The rail alignment starts from the bottom of the rail by means of shims	the rail alignment starts mostly from the bottom of the channel by means of shims .- TOR = consequence!	-
		Top down	is possible, by use of jigs holding the rail head during the concreteing	is possible, by use of jigs holding the railhead during the pouring of the non-shrinkable grout	possible with jigs	is possible with jigs holding the railhead,... but non practical	TOR is fixed by the GSF-jig holding the whole rail body,...the rest = consequence!
	Rail support		discontinuous (attention: the filler material in between the rail-supports plays an important role in the overall stiffness, if not properly executed - danger for "stiff" points!)	continuous - attention: around the fasteners the imposed toe toad may exercise a local increase in the stiffness of the railstrip - non homogeneous elasticity distribution!)	continuous, stiff points are generated at rail fixation points	continuous - attention: stiff points may also be generated at places where shimming plates are installed (inhomogeneous elasticity along the rail)	continuous with homogeneous elasticity!
	Range of track stiffness	MN/m/ml-rail	> 45	> 25	> 15	> 25	> 15
Typical Remarks			fairly thick track body (see next point)	tiebars with isolation are embedded close to road surface = danger for fissuration in the road finish.	the large longitudinal rail joint allows removal of the rail without breaking the road surface but this rail joint needs special care (components and installation)	the installation may be affected by climatic conditions (extreme temperature, humidity (rain, snow etc.) - also in some countries remarks on H&S (health and safety) because of chemical composition.	particular attention is required during pre-coating operation (sticking the "jackets" to the rail) - need for skilled work force to avoid joints and proper glueing.
Typical overall track body thickness - rail Ri60 (180 mm)							
	foundation concrete		100	100	100	100	100
	track fixation concrete		300	200	300	300	300
	finish		200	200			
	total (mm)		600	500	400	400	400
Minimum amount of construction phases above the lean foundation concrete (assuming concrete as road finish)		1	to install "frames" (=rail on sleepers) & track alignment	to pour support concrete	to install the concrete slabs and the premounted rail	to install the concrete slab with slipform paver - channels to receive rails are left in the concrete during the slipforming process	to pre-coat the rail with jackets
		2	to pour track fixation concrete	to install frames (=rails tied by tiebars) + track alignment	to install the rail and pour the second phase of concrete	to install rail strip, rail and filler elements (concrete, rubber blocks or plastic tube holders) in the channels	to install pre-coated rail - alignment with GSF-jigs
		3	to equip rail with webchambers and isolating elements	to pour the non shrinkable grout under the rail foot to fix the level	to pour the elastomers and to install the longitudinal rail joint	to align the track (shims under the rail and conical elements on the 2 sides) and to pour the PUR elastomeric material	to pour the concrete
		4	to install the concrete finish	to equip the rails and teibars with webchambers & isolating elements	XXXXXXXXXXXXXXXXXXXXXX	XXXXXXXXXXXXXXXXXXXXXX	XXXXXXXXXXXXXXXXXXXXXX
		5	XXXXXXXXXXXXXXXXXXXXXX	to install the concrete finish	XXXXXXXXXXXXXXXXXXXXXX	XXXXXXXXXXXXXXXXXXXXXX	XXXXXXXXXXXXXXXXXXXXXX

Types of ballastless tram infrastructure with shared right-of-way (tram-road)

Track Family			Discontinuous supported rail	Continuous supported rail			
Fastening			Mechanical	Mechanical	Mechanical	Fastenerless	
Name			Slabtrack with sleepers or direct fixation	Tiebar Slabtrack	Hybrid	Poured embedded track	Precoated embedded track
Installation	Limitations due to climatic conditions		no	no (but pay attention to the non-shrinkable grout!)	attention: liquid elastomere doesn't withstand temperatures under 5°C	attention - liquid elastomere doesn't withstand rain/cold	no
	Typical installation advance speed	lmst/day	< 54	< 36	< 144	< 144	< 144
Contractor's Cost Budget (€/lmst)	Supplies	Track elements	150	180	170	300	170
		Concrete	160	160	120	160	120
		60kg Rail	100	100	100	100	100
	Installation		180	250	150	280	150
	Total €/lm single track)		**590**	**690**	**540**	**840**	**540**
RAMS - Evaluation	Reliability		OK - system based on main railway line experience and proven record. Attention: longitudinal rail joints	OK- system has proven record of reliability. Attention: longitudinal rail joints	Attention: longitudinal rail joints	OK - system has proven record of reliability. Attention: special care for gauge widening in case of highly resilient systems	OK - system has proven record of reliability. Attention: special care for gauge widening in case of highly resilient systems
	Availability		OK - track remains available at all times within normal operation hours - Attention: renovation of rail in case of highly corrugated rail is possible within nightly time window outside operation. Attention: road finish may be the problem to finish in time!	OK- track remains available at all times within normal operation hours - Potential hazard = cracks in the road surface finish and consequential damage where tiebars are close to surface	OK- track remains available at all times within normal operation hours.	OK- track remains available at all times within normal operation hours. Potential hazard: polymerisation time of the elastomer to make track available in time.	OK- track remains available at all times within normal operation hours.
	Maintainability		OK - system is maintainable during night time window (rail = removable and replaceable,...attention for road finish renovation!)	OK - system is maintainable during night time window (rail = removable and replaceable,...attention for road finish renovation!)	OK - system is maintainable (rail = removable and replaceable without need for road finish renovation, but if welding in existing channel = electric welding).	OK - system is maintainable (rail = removable and replaceable without need for road finish renovation, but if welding in existing channel = electric welding). Longer maintenance window required than types 1, 2 & 3	OK - system is maintainable (rail = removable and replaceable without need for road finish renovation, but if welding in existing channel = electric welding). Longer maintenance window required than types 1, 2 & 3
	Safety		OK - system based on main railway line technology.	OK - system is safe	OK - system is safe	OK - system is safe. Attention: in some countries special conditions for H&S	OK - system is safe
EXAMPLES			Most of tramway networks in France, Italy and Holland	Most of tramway networks in Germany and Roma (IT)	Antwerp tramway (BE)	Most of trams in UK	Tram Brussels (BE) and new lines in Ghent (BE)
			Euskotren - Bilbao (ES)	Tram Barcelona (ES)		Tram Valencia (ES))	Tram Athens (GR), Metro Ligero Madrid (ES)
			Metro Tenerife (ES)	STCP (PT)		Krakow (PL)	Ext.T1 & TMS- RATP (FR)
			Tram Brussels (BE)	Tram Antwerp (BE)			
			Tram Antwerp (BE)				

3.3 NOISE MITIGATION

Only type 4 is a classical embedded rail system: continuously supported, fastenerless, poured on site.

Recommendations and guidelines for embedded rail systems are drawn up by CROW.

CROW is the Dutch national information and technology centre for transport and infrastructure. This non-profit organisation allows central and provincial government, local councils, building trade organisations, consultancies, public transport organisation and educational institutes to work together in pursuit of their common interests through the design, construction and management of roads and other traffic and transport facilities.

CROW
Galvanistraat 1, NL – 6716 AE Ede
P.O.Box 37, NL – 6710 BA Ede, the Netherlands
Tel. +31 318 69 53 00
Fax. +31 318 62 11 12
E-mail crow@crow.nl
web site www.crow.nl

These "Recommendations And Guidelines for Embedded Rail Systems", RAGERS constitute a first step towards a clear, unified standard (or draft standard) for embedded rail systems in the Netherlands and, possibly, in the European Union. It is based on a survey of existing knowledge and experience, and discusses all the requirements that embedded rail systems could be required to meet. RAGERS is the result of projects carried out for Dutch rail infrastructure managers, plus the knowledge and experience of contractors, manufacturers, engineering firms and consultants.

This is both a reference work and a design handbook, covering all types of railway operation on embedded rail systems. The document addresses as many as possible of the issues that arise when writing functional requirements specification (FRS). These include design, construction and maintenance, mechanical requirements (elasticity, strength and dynamics), disruption/nuisance and use for as wide a range of applications as possible.

RAGERS consists of three parts, A, B & C, each with its own specific purpose:

Part A is a reference work, covering the design, construction and maintenance of embedded rail track. This part reflects practical experience in the use of unballasted track in general and embedded rail in particular.

Part B deals with test and calculation methods for embedded rail systems. It is important that the methods used to measure and calculate the behaviour of the track system be consistent and that the results lie within certain bounds, if one is to control such characteristics as durability, resistance to temperature change, noise, vibration, strength and stiffness. This part sets out such methods of determining and verifying the behaviour of a track design.

Part C is a FRS drafter's handbook. The aim of this part is to enable the person drafting an FRS to make best possible use of the information in RAGERS and all relevant information derived from practical experience. A ten-step plan for the drafting of an FRS is proposed and explained, and a number of examples are given. The plan consists of a generic part and a specific part. The generic part covers application, utilisation and conditions. This is separate from the specific design, which covers the dimensions of the embedded rail structure and the way it is constructed.

Following the ten-step plan ensures that one has made maximum use of RAGERS and has produced an FRS that reflects the state of the art with regard to embedded rail.

Types 3 and 5 (§2) belong also to the family of embedded rail systems.
Type 3 has some discrete fixations (hybrid system) and type 5 is not poured on site but prefabricated.

These three systems behave in the same way from a noise emission point of view: the reflecting rail surface is limited and the rail itself is highly damped (no low damped rail modes).

Types 1 and 2 normally do not behave as types 3, 4 & 5 since, in general, the railweb filling blocks are not firmly glued or connected to the rail.

3.4 CONCLUSION

As discussed in WP3.1, embedded rail systems (types 3, 4 & 5) reduce the global rail emission by 3 to 4 dB(A) in comparison with discrete direct fixation systems on a concrete slab.

In comparison with open ballasted track, the embedded rail systems are still 1 to 3 dB(A) noisier since they have, in general, hard road surfaces between and next to the rails, giving yield to reflections and almost no absorption.

In order to get optimal embedded rail systems (from a noise emission point of view), the embedded rail systems have to be combined with a road surface with high acoustic absorption.

This development will yield a noise reduction of about 3 dB(a) in comparison with open ballasted track. It is the objective to develop such a road surface (between and next to the rails), which is highly absorptive, maintenance free and able to carry road traffic.

As part of the QCITY project, the 3 dB(A) gain is to be obtained for rolling noise on tangent track, all other conditions being equal (vehicle type, speed, ...).

This development is proposed to be made by partners Frateur de Pourcq (FDP), Heijmans (HEIJ) and De Lijn (LIJN): prototype development from month 12 to month 18 (as part of WP3.2) and is proposed to be validated in the network of De Lijn (combined bus/tram track) as part of SP5.

4 BROAD BAND RAIL DAMPERS IN COMBINATION WITH A WHEEL DAMPER

4.1 SUMMARY

This document summarises the different broad band rail dampers (two of them are based on tuned damping, although showing broadband characteristics) and their effect on the roar noise emission on straight track only. The different rail profiles are briefly presented and an estimation of the noise reduction and the cost is given for each solution. The estimation is made for rails resting on soft to medium soft rail pads by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness.

Predictions are also made for a combination of the rail solution and one selected wheel damper ('shark's fin'), indicating the noise reduction anticipated by combining wheel and rail damping.

All solutions based on constrained layer damping (CLD) are in a design phase, as they represent new rail designs as well as new innovative concepts. Therefore, a lot of practical issues are still to be solved.

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	Track Cost [€/m]	Limitations/ Critical Matters
CLD-Rail Type 1	5	2	200	Rail/Visco stress
CLD-Rail Type 2	6	2	300	Visco stress
CLD-Rail Type 3	7	3	1000	Prize/Size
Corus rail damper	6	3	500	-
CDM-ABSO-RAIL	6	3	500	-

Table 3

Additional cost for wheel dampers ('shark's fin') is about 500 €/wheel.

All reductions are fairly equal. The predicted total noise reductions for combined wheel and rail measures are **5 – 7 dB(A)**, if the contribution of other noise sources than the wheel/rail-system are ***not*** too significant (e.g. traction gears). Thus, a combination of wheel and rail damping might have a good effect on the rolling noise.

When extra costs for the broad band rail dampers are considered, the evaluated measures can be ranked as follows (wheel and rail measures):

1. CLD-Rail Type 1 <u>or</u> CLD-Rail Type 2 (80-130 €/m/dB(A));
2. Corus rail damper <u>or</u> CDM-ABSO-RAIL (160-170 €/m/dB(A));
3. CLD-Rail Type 3 (320 €/m/dB(A).

4.2 INTRODUCTION

This document summarises the different broad band rail dampers (two of them are based on tuned damping, although showing broadband characteristics) and their effect on the roar noise emission on straight track only. The different dampers are briefly presented and an estimation of the noise reduction and the cost is given for each solution. The estimation is made for rails resting on soft to medium soft rail pads by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness.

Predictions are also made for a combination of the rail solution and one selected wheel damper ('shark's fin' damper tuned below the fundamental wheel resonance frequency but with broadband characteristics for all significant wheel resonance frequencies), although wheel dampers are not part of WP3. This is done to indicate the noise reduction effects of combining wheel and rail damping, which is of particular interest when rail damping only is not sufficient.

The solutions presented in section 4.1 (constrained layer damping of UIC60 rail; four examples) represent new rail designs as well as new innovative concepts. These solutions are in a design phase. Therefore, a lot of practical issues are still to be solved.

4.3 THEORETICAL OUTLINE

The different rail damping projections have been studied and evaluated by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness. It is assumed that lateral rail vibrations and the track infrastructure (i.e. sleepers and ballast) are not contributing significantly to the noise level of the wheel/rail system, which is sufficiently true for rails resting on soft to medium soft rail pads [4.1] (the mathematical model is not applicable for fairly stiff rail pads, but this is not a big problem as rail dampers are much less effective for rails resting on stiff pads).

The rail impedance is described by an analytical expression of a damped slender beam[2] resting on a continuous and damped rail pad system. The vertical wheel impedance is simply described as a big solid mass [4.2] for studied frequencies between 500 – 2500 Hz. Furthermore, it is assumed that vertical wheel vibrations are effectively coupled to lateral and resonant wheel vibrations by the inclined/curved web and that these vibrations are dominating the noise level contribution of the wheel. The wheel rail contact spring is modelled as a discrete and vertical spring element mounted directly to the wheel and rail contact areas.

The effective radiation area of rail (radiation length x radiation area per meter length of rail x radiation coefficient[3]) is used as a parameter reflecting the total sound power

---

2 Corrected for shear deformation of rail web.

3 Radiation coefficient ≈ 1for frequencies greater than 500 Hz.

radiated by the rail. This parameter is directly related to the decay rate of rail in dB/m, which in turn is proportional to the apparent loss factor[4] of rail.

The wheel/rail system is excited by a "moving roughness strip" located in the middle of this wheel/rail contact spring (i.e. split spring element). Thus the wheel/rail roughness induced vibrations tends to propagate more down to the rail, if the rail impedance is lower than the wheel impedance including contact stiffness and vice versa.

There is a "cut off" frequency at about 800 - 1000 Hz caused by the contact stiffness of a typical wheel/rail system (1.1E+9 N/m). Above that frequency range, there is a significant isolation effect for vibrations propagating down to the rail. For lower frequencies, there is no such isolation effect. Further decrease of the wheel/rail contact stiffness will result in corresponding decrease of the dynamic contact forces for higher frequencies, which in turn will result in almost ***equally*** decreased wheel and rail sound level contributions. This observation is very important, as there is just little or no reduction effect at all on the noise contribution of wheel for most other rail measures evaluated.

Vertical decay rates (apparent loss factor, bending stiffness, mass per length of rail), mechanical point impedance (bending stiffness, mass per length of rail) and wheel/rail contact stiffness are often sufficient input data for approximate prediction and ranking of total sound level reductions caused by different rail measures for e.g. UIC60 rails resting on soft to medium stiff pads for the most significant frequencies (500 – 2500 Hz).

4.4 STUDIED SOLUTIONS

4.4.1 Constrained layer damping of UIC60 rail

One studied method to create a broad band damping of the rail is to introduce a constrained layer damping (CLD). In this section, three different design ideas (Type 1, 2 and 3) have been evaluated. However, maximum stress levels of the damping layers might be critical and will therefore be investigated during next 6M period.

4.4.2 Split UIC60 rail foot (Type 1)

The primary purpose of implementing constrained layer damping of rail according to Figure 21 is to reduce the noise contribution of the rail significantly. This is achieved by increasing the apparent loss factor, which in turn also increases the corresponding decay rates in dB/m. A secondary effect is to reduce the noise contribution of the wheel caused by a fairly small decrease in the point impedance of the rail. The lower point impedance is a result of the reduced dynamic bending stiffness of a split rail. Due to the reduced bending stiffness, the maximum stress level in the rail foot will be significantly increased for a passing train.

---

[4] Including stop band effects of the periodic rail-pad-sleeper structure.

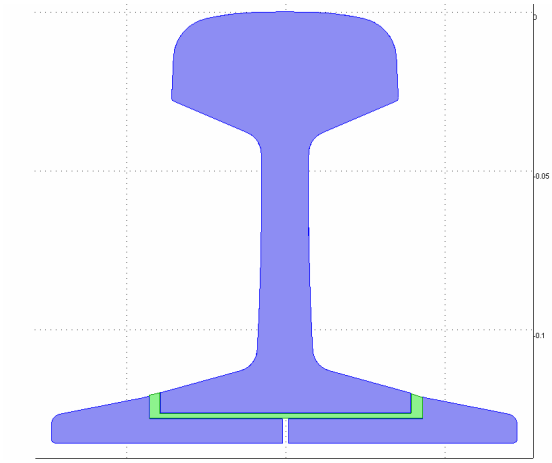


Figure 21 Viscoelastic damping layers (green) implemented in a split UIC60 rail foot with no increase of weight

4.4.3 Narrow UIC60 rail foot with constraining beams or Modified saddle profile rail (Type 2)

The primary purpose of implementing constrained layer damping of rail according to Figure 22 and 23 is to reduce the noise contribution of the rail significantly. This is achieved by increasing the apparent loss factor, which in turn increases the corresponding decay rates in dB/m. However, the noise contribution of the wheel will now be slightly increased due to a fairly small increase in the point impedance of rail. The increased point impedance is the result of an added mass of typically 10 – 20 kg/m combined with a slightly reduced dynamic bending stiffness of the rail. Thus, the maximum stress level in rail for a passing train will be increased too, but less increased than for the damped rail according to figure 21.

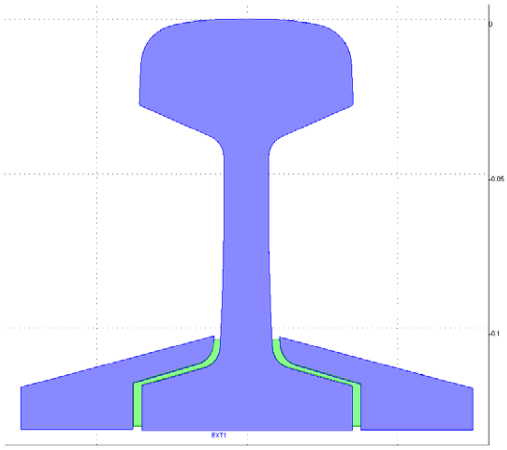


Figure 22 Viscoelastic damping layers (green) and constraining beams added to rail foot with typically 20 kg/m increase of weight

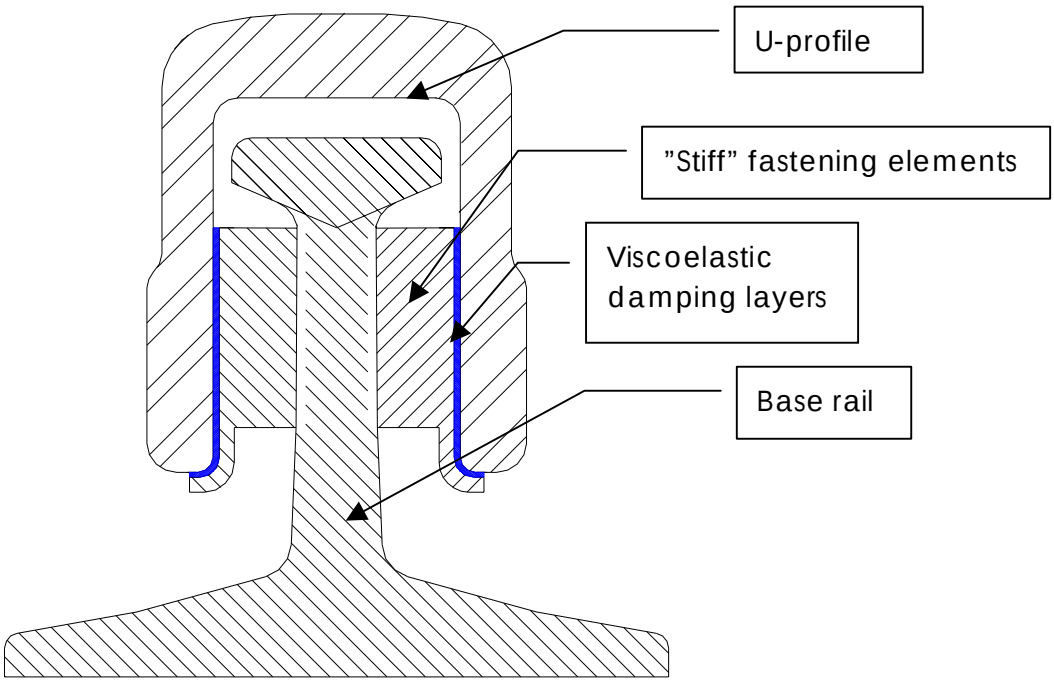


Figure 23

Viscoelastic damping layers (vertical, thick blue lines) implemented in a modified saddle profile rail with typically 10 – 15 kg/m increase of weight. All rubber elements are replaced by "stiff" fastening elements (e.g. steel)

4.4.4 UIC60 rail embedded in a box (Type 3)

The primary purpose of implementing constrained layer damping of rail according to Figure 24 is to reduce the noise contribution of the rail significantly. This is achieved by increasing the apparent loss factor, which in turn increases the corresponding decay rates in dB/m. However, the noise contribution of the wheel will now be significantly increased due to a fairly big increase in the point impedance of rail. This increase is the result of an added mass of typically 120 kg/m combined with a significantly increased dynamic bending stiffness of the rail (including core and steel box). Due to the increased bending stiffness, the maximum stress level in the rail foot will be significantly decreased for a passing train.

Figure 24 shows an example of a cross section of a constrained layer damped UIC60 rail for ballasted tracks. The core (green) is made of epoxy or any other suitable material with a Young's modulus of about 5E+9 N/m^2. The thin viscoelastic damping layer is implemented between rail and core (not indicated in Figure 24).

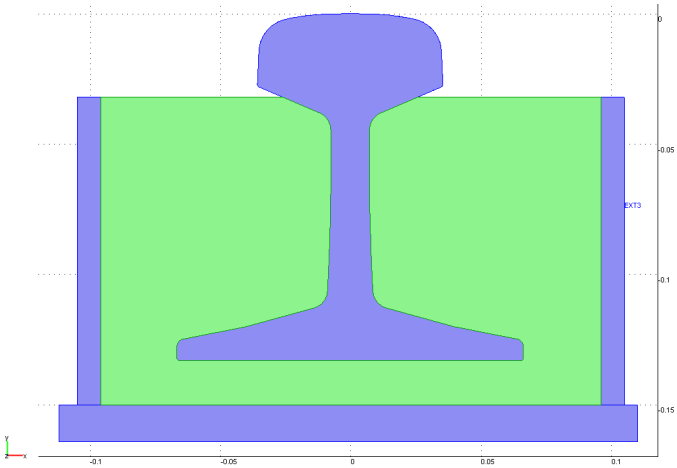


Figure 24 Viscoelastic damping layers (not indicated) between the UIC60 rail and core (green), which is contained in a steel box. This will result in typically 120 kg/m increase of weight or a total weight of 180 kg/m

4.4.5 Corus rail dampers

The Corus rail damper is a commercially available product based on tuned damping, (although the decay rates of rail are significantly increased within a wide frequency range up to about 5000 Hz [4.5]). The primary purpose of implementing tuned damping of rail according to Figure 25 is to reduce the noise contribution of the rail significantly. This is achieved by increasing the apparent loss factor, which in turn increases the corresponding decay rates in dB/m.

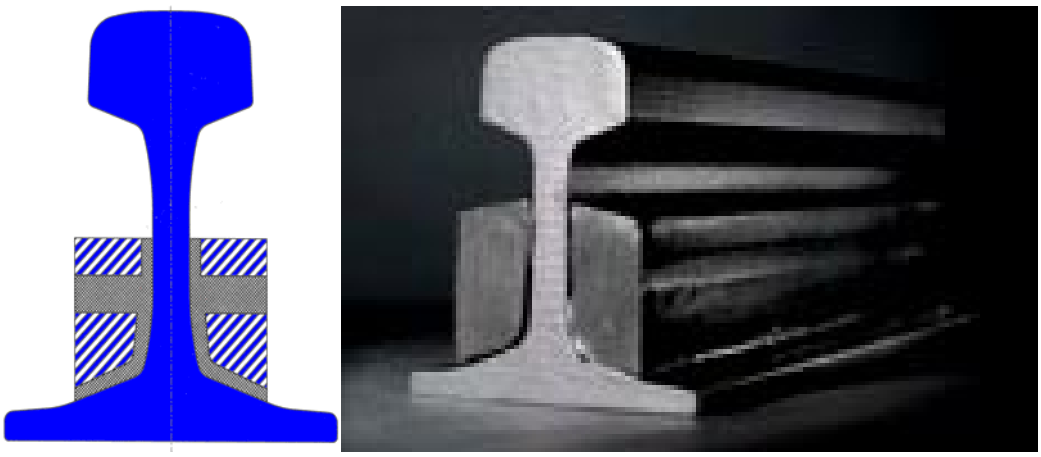


Figure 25 Cross section and photo of a Corus damper (rubber is grey in the left picture) mounted to foot/web of rail

4.4.6 CDM-ABSO-RAIL dampers

The CDM-ABSO-RAIL damper is a commercially available product also based on tuned damping (although the decay rates of rail are significantly increased within a wide frequency range up to about 5000 Hz [4.6]). The primary purpose of implementing tuned damping of rail according to Figure 26 is to reduce the noise contribution of the rail. This is achieved by significantly increasing the apparent loss factor, which in turn increases the corresponding decay rates in dB/m.

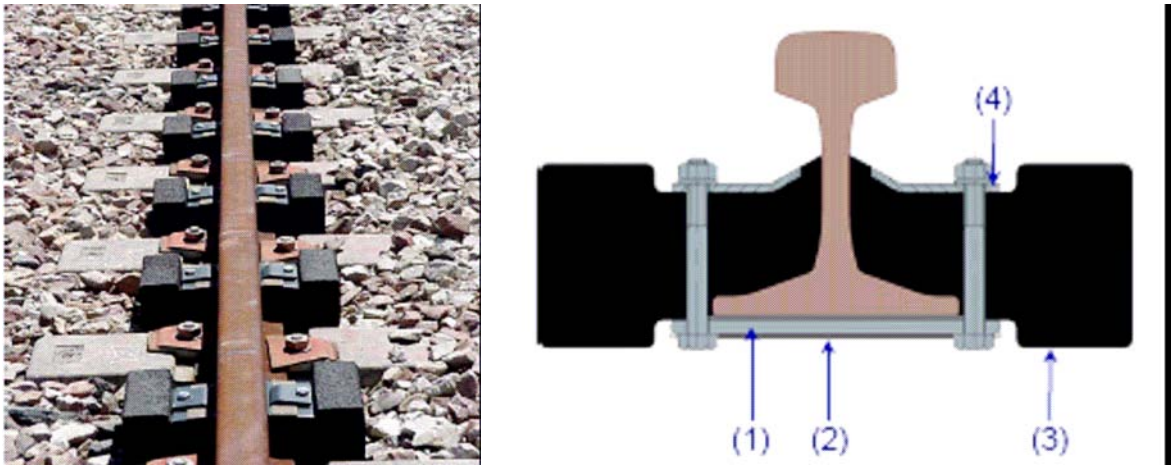


Figure 26 Cross section and photo of a CDM-ABSO-RAIL damper (rubber black) mounted to foot/web of rail.

4.5 PREDICTED SOUND LEVEL REDUCTIONS

In this chapter predicted noise reductions of the evaluated broad band rail dampers are presented alone and in combination with wheel dampers for a speed of 80 kph.

These reductions are based on reference data derived from pass-by noise measurements in Tjörnarp [4.3] (in the southern part of Sweden) for different trains and speeds and from the decay rates of the corresponding UIC60 rails [4.4].

According to these reference measurements, there is only ***one dominating octave band*** in the measured bass-by noise levels (TEL in dB(A)). Therefore, the predicted total noise reduction (dB(A)) of the wheel/rail-system at 80 kph (due to the damped rail profile) is considered to be as large as the corresponding noise reduction (dB) of the dominating octave band (here 800 Hz). This is often fairly true, but in some cases, there is a significant difference (positive or negative).

Furthermore, the wheel/rail-system is considered to be the dominating noise source (at least before rail damping). This is often true for modern electric trains running with lower speeds up to about 80 kph, but it is ***not*** true for e.g. metro cars with self ventilated motors. For higher speeds, the noise contribution of the traction gears tends to be more significant and in some cases even dominating. Then, of course, the effects of rail dampers will be less than predicted.

4.5.1 Constrained layer damping of UIC60 rail

Constrained layer damping of an UIC60 reference rail in Tjörnarp has been evaluated regarding prediction of sound level reductions for the following three projections:

1. Viscoelastic damping layers implemented in a split UIC60 rail foot with no increase of weight.
2. Viscoelastic damping layers and constraining beams added to rail foot with 20 kg/m increase of weight (Figure 21) ***or*** saddle profile on a UIC60 rail with the soft rubber elements replaced by viscoelastic damping material (Figure 23).
3. Viscoelastic damping layers and constraining core/steel box added to rail foot/web with 120 kg/m increase of weight (Figure 24).

4.5.2 Split UIC60 rail foot (Type 1)

Sound level reductions (-) in dB(A) for ***Type 1*** rail dampers at 80 kph (800 Hz):

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	-0.6	-3.0	-1.8
Wheel+Rail	-6.6	-5.0	-5.6

Table 4

The first row is sound level decrease caused by damped wheels ('shark's fin'). The second row is sound level decrease caused by rail damping. The third row is sound level decrease caused by a combination of both rail and wheel damping. A correction for shear deformation of rail web (-0.1 dB(A)) is included for the second and third row numbers.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, then the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total sound reduction of about **-5.3 dB(A)** (i.e. the average of -5.0 and -5.6) can be anticipated by combining rail dampers type 1 with wheel dampers ('shark's fin'). The sound reduction for rail damping type 1 only is **-2.4 dB(A)** (i.e. average of -1.8 and -3.0).

Changes of significant parameters for ***Type 1*** rail dampers:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+117	-1.24[5] → -2.69 dB/m
Apparent rail loss factor:	+100	0.15 → 0.30 for [6]β_2 = 1.0
Mass per m length of rail:	0	60 kg/m
Dyn. bending stiffness[7] of rail:	-30	63E+5 → 44.5E+5 Nm^2
Rail/pad frequency[8]:	0	300 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m

Table 5

[5] Measured for UIC60 rail in Tjörnarp [4].

[6] The quantity β_2 is loss factor of the damping layer.

[7] The dynamic bending stiffness change of rail is -30 % and the static bending stiffness change is -50 %.

[8] Second order rail-pad-sleeper-ballast resonance frequency (≈ rail to pad resonance frequency).

4.5.3 Narrow UIC60 rail foot with constraining beams or Modified saddle profile rail (Type 2)

Sound level reductions (-) in dB(A) for ***Type 2***[9] rail dampers at 80 kph (800 Hz):

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	1.0	-3.6	-1.0
Wheel+Rail	-5.0	-5.6	-5.3

Table 6

A correction for shear deformation of rail web (+0.4 dB(A)) is included for the second and third row numbers.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, then the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant.

Thus, a total sound reduction of about **-5.5 dB(A)** (i.e. the average of -5.3 and -5.6) can be anticipated by combining rail dampers type 2 with wheel dampers ('shark's fin'). The sound reduction for rail damping type 2 only is **-2.3 dB(A)** (i.e. the average of -1.0 and -3.6).

Changes of significant parameters for ***Type 2*** rail dampers:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+111	-1.24 → -2.62 dB/m
Apparent rail loss factor:	+100	0.15 → 0.30 for β_2 = 1.0
Mass per m length of rail:	+33	60 → 80 kg/m
Dyn. bending stiffness of rail:	0	63E+5 Nm^2
Rail/pad frequency:	-13	300 → 260 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m

Table 7

[9] Predicted sound level reductions are approximately equal for Type 2 rail dampers, thus for "Narrow UIC60 rail foot with constraining beams" according to Figure 4.1.2 and for "Modified saddle profile rail" according to Figure 4.1.3.

4.5.4 UIC60 rail embedded in a box (Type 3)

Sound level reductions (-) in dB(A) for constrained layer damping at 80 kph (800 Hz):

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	+2.2	-5.6	-0.6
Wheel+Rail	-3.8	-7.6	-5.6

Table 8

A correction for shear deformation of rail web (+2.6 dB(A)) is included for the second and third row numbers.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, then the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total sound reduction of about **-6.6 dB(A)** (i.e. the average of -5.6 and -7.6) can be anticipated by combining a UIC60 rail embedded in a box with wheel dampers ('shark's fin'). The sound reduction for CDM-ABSO-RAIL dampers only is **-3.1 dB(A)** (i.e. the average of -0.6 and -5.6).

Changes of significant parameters:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+7	-1.24 → -1.33 dB/m
Apparent rail loss factor:	0	0.15 → 0.15
Mass per m length of rail:	+200	60 → 180 kg/m
Dyn. bending stiffness of rail:	154	63E+5 → 160E+5 Nm^2
Rail/pad frequency:	-50	300 → 150 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m

Table 9

4.5.5 Corus rail dampers

Corus rail dampers, mounted on both sides of the web of a UIC60 reference rail in Tjörnarp, have been evaluated regarding prediction of sound level reductions.

Sound level reductions (-) in dB(A) for Corus rail dampers at 80 kph (800 Hz):

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	0	-4.3	-2.0
Wheel+Rail	-6.0	-6.3	-6.2

Table 10

No correction for shear deformation of rail web is needed.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, then the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total sound reduction of about **-6.3 dB(A)** can be anticipated by combining Corus rail dampers with wheel dampers ('shark's fin'). The sound reduction for Corus rail dampers only is **-3.2 dB(A)** (i.e. the average of -2.0 and -4.3).

Changes of significant parameters for Corus Rail Damper:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+240	-1.24 → -4.21[10] dB/m
Apparent rail loss factor:	+240	0.15 → 0.51
Mass per m length of rail:	+33	60 → 80 kg/m
Dyn. bending stiffness of rail:	0	63E+5 Nm^2
Rail/pad frequency:	0	300 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m

Table 11

[10] Based on decay rates presented in [5].

4.5.6 CDM-ABSO-RAIL dampers

CDM-ABSO-RAIL dampers, mounted on both sides of the web of a UIC60 reference rail in Tjörnarp, have been evaluated regarding prediction of sound level reductions.

Sound level reductions (-) in dB(A) for CDM-ABSO-RAIL dampers at 80 kph (800 Hz):

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	0	-3.9	-1.8
Wheel+Rail	-6.0	-5.9	-6.0

Table 12

No correction for shear deformation of rail web is needed.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, then the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total sound reduction of about **-6.0 dB(A)** can be anticipated by combining CDM-ABSO-RAIL dampers with wheel dampers ('shark's fin'). The sound reduction for CDM-ABSO-RAIL dampers only is **-2.9 dB(A)** (i.e. the average of -1.8 and -3.9).

Changes of significant parameters:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+194	-1.24 → -3.64[11] dB/m
Apparent rail loss factor:	+194	0.15 → 0.44
Mass per m length of rail:	+25	60 → 75 kg/m
Dyn. bending stiffness of rail:	0	63E+5 Nm^2
Rail/pad frequency:	0	300 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m

Table 13

[11] Based on decay rates presented in [6].

4.6 COMMENTS

It should be noticed that the main quantities governing the noise contribution of the rail is the apparent loss factor, the corresponding decay rate and to some extent also the mechanical point impedance of the rail. The decay rate is proportional to the apparent loss factor of the rail divided by the wave length, which in turn (slender beam) is proportional to (bending stiffness/mass per length of rail)$^{1/4}$. Obviously, minor changes in bending stiffness and/or mass per length of the rail are ***not*** significantly influencing the decay rate and the corresponding noise reduction of the rail. Therefore, the increase of the decay rate must primarily be achieved by increasing the apparent loss factor of the rail (i.e. rail loss factor and/or pad stiffness).

The distribution of the wheel/rail-roughness vibrations to the wheel and to the rail depends on the mechanical point impedance of the rail and the wheel/rail contact stiffness. The mechanical point impedance of the rail (slender beam) is proportional to (bending stiffness)$^{1/4}$ x (mass per length of rail)$^{3/4}$. For example, an increase of the mechanical point impedance by increasing the bending stiffness and mass per length of rail will significantly reduce the noise of an ordinary rail above the 'cut off' frequency (due to the contact stiffness) typically at 800 – 1000 Hz, but the effect on the wheel is negligible. Below the 'cut off' frequency, but above 500 Hz, there is instead a significant noise reduction of the wheel but no reduction of the rail. Below about 500 Hz the coupling effects of the wheel/rail system to the track infrastructure are getting more and more complex. Therefore, general "rules" can not be defined for that frequency range. However, frequencies below the 'cut off' frequency are often less important for the pass-by noise.

4.7 CONCLUSIONS AND COSTS

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	Track Cost [€/m]	Limitations/ Critical Matters
CLD-Rail Type 1	5.3	2.4	200	Rail/Visco stress
CLD-Rail Type 2	5.5	2.3	300	Visco stress
CLD-Rail Type 3	6.6	3.1	1000	Prize/Size
Corus rail damper	6.3	3.2	500	-
CDM-ABSO-RAIL	6.0	2.9	500	-

Table 14

Additional cost for wheel dampers ('shark's fin') is about 500 €/wheel.

It can be concluded that all reductions are fairly equal. The noise reduction anticipated for rail measures is typically 2 – 3 dB(A) only. However, the predicted total noise reductions for combined wheel and rail measures are typically **5 – 7 dB(A)**, if the contribution of other noise sources than the wheel/rail-system are ***not*** too significant (e.g. traction gears). This combination effect is more promising and of great interest for this project. Thus, a combination of wheel and rail damping might have a good effect on the rolling noise.

When extra costs for the broad band rail dampers are considered, the evaluated measures can be ranked as follows (wheel and rail measures):

1. CLD-Rail Type 1 ***or*** CLD-Rail Type 2 (80-130 €/m/dB(A));
2. Corus rail damper ***or*** CDM-ABSO-RAIL (160-170 €/m/dB(A));
3. CLD-Rail Type 3 (320 €/m/dB(A).

4.8 REFERENCES

[4.1] V. V. Krylov et al, Noise and vibration from high-speed trains, MPG Books, London, 2001

[4.2] A. Nordborg, Vertical rail vibrations: Noise and structure-borne sound generation, Doctorial Thesis RITA-FKT Report 9535, The Marcus Wallenberg Laboratory for Sound and Vibration Research. Kungliga Tekniska Högskolan, Stockholm, 1995

[4.3] H. Malker, H. Samuelsson, N. Renard, The Tjörnarp site for railway noise studies - characterisation of original status. Sound and vibration measurements during train passages, QCITY, subtask 3.1, Stockholm, 2005

[4.4] N. Renard, J. Nielsen, Definition of a reference system for freight trains to compare against in quantification analyses, QCITY, subtask 3.1.1, Stockholm, 2005

[4.5] C.J.C. Jones ISVR, D. Farrington & D. Benton Corus Rail, Reduction of Railway Noise at Source – Practical and Cost Effective Methods for Different Track Forms, Switch to Delft 2004

[4.6] CDM-ABSO-RAIL, Rail Noise Absorber data sheet from CDM, Rev 09/09/2003

[4.7] DRAFT prEN ISO 3095, Railway applications – Acoustics – Measurement of noise emitted by rail bound vehicles (ISO/DIS 3095:2001), Draft international standard.

5 STRUCTURE BORN NOISE (STEEL BRIDGES)

5.1 INTRODUCTION

Although during the last decennia many new railway bridges have been made of concrete, the European railway grid still contains a lot of steel bridges. In France for instance there are about 900 and every one of them is a source of noise [5.1]. This is of course especially a problem in urban areas where noise is an annoyance to people living near a bridge.

The QCITY WP3.2 focuses on mitigation measures for wheel and rail. Structure born noise of steel bridges is however also taken in to account because of its predominant nature and strong relation with wheel and rail noise mitigation.

On a normal ballasted track, sound is mainly radiated by the rails, the sleepers and the wheels of the train. It is the rolling noise of the train, caused by roughness of the wheel and rail contact surface. On a steel bridge, the train also induces vibration of the structure of the bridge itself. These structure born vibrations radiate noise, which is generally louder than the rolling noise of the train itself. A train passing on a steel bridge will therefore have a more important environmental impact than when running on land based track.

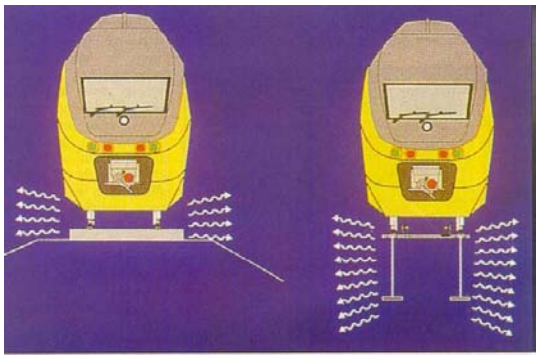


Figure 27 Ballasted track and bridge track noise radiation

Specific mitigation measures for structure born noise of steel bridges have been developed. They allow obtaining good results as well for existing as new steel bridges.

Figure 28 Example of mitigation measures on a steel bridge

WP3.2.2 of the QCITY project focuses on mitigation measures for rails. The European Sustainable Bridge Project is already taking care of matters concerning the bridge structure. The recommendations of this report focus therefore on track solutions (e.g. support stiffness, special track work) for existing steel bridges only.

5.2 STATE OF THE ART

On normal ballasted track, sound is radiated mainly by the rails, the sleepers and the wheels of the train. This is called rolling noise. On a steel bridge the overall noise is louder than on normal track. In addition to the rolling noise the bridge itself vibrates and radiates structure borne noise consisting mainly of frequencies lower than 200 Hz. The total sound pressure produced by trains on steel bridges can be 5 to 15 dB higher than on ballasted track with timber sleepers [5.2].

To reduce structural noise when a train passes over a steel bridge one can:

a. prevent excitation of the bridge structure by:
   1. reducing rail vibrations (lower rail contact roughness, avoid rail discontinuities)
   2. isolating the bridge structure from rail vibrations (ad vibration control measures in rail fixation, rail support or track support)
   3. making the bridge structure less sensitive for excitation by the rail vibrations. (intelligent bridge design)

b. dissipate residual bridge structure vibration by damping (ad additional energy absorption devices in or on the bridge structure). This can be considered when bridge noise still dominates rolling noise even after application of the preventive measures sub a.

The following cases show the application of various mitigation measures for existing and new steel bridges that have led to satisfying results.

5.2.1 South railway bridge of Budapest [5.3]

The 451 m long, double-track riveted steel South Railway bridge is a major link between West and East Hungary, carrying heavy rail traffic day and night, 7 days a week. The Hungarian Railway decided to take serious noise control measures on the bridge because of the recent construction of some cultural buildings in the close vicinity of the bridge. Apart from the wooden sleepers, the bridge consisted of steel elements only, without any vibration isolation or damping.

Measurements

Preliminary noise measurements and sound field mapping calculations revealed that the noise of the bridge exceeded the overall environmental noise level by more than 10 dB(A) in the case of train pass-bys.

In order to reveal the noise generation and radiation mechanisms in more detail, extensive vibration and noise measurements were conducted. The vibration of all important bridge elements: rail, baseplate, wooden sleeper, main beams, girder spars and walking plates were measured and, simultaneously, near-field and far-field noise recorded.

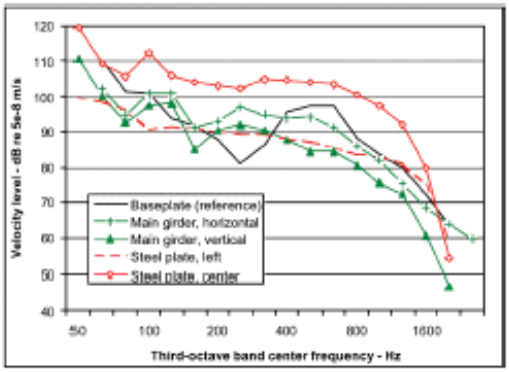


Figure 29 Comparison of vibration spectra and the farfield noise

Considering that the conclusions to be drawn from the investigations had serious financial implications, the spectrum analyses were complemented by finite element simulation of the bridge structure as well.

The end result had a very clear interpretation: half of the energy was radiated by the main girders of the bridge and another half by the walking plates. This in turn implied that neither the vibration reduction of the girders, nor the essential improvement of the walking plates was sufficient to solve the problem entirely, but instead, both measures were necessary in order to achieve satisfactory noise control.

5.2.1.1 Noise and vibration control measures

The design of vibration control of the rail fastening was based on a combination of a level 1 highly resilient flapped rail pad and a level 2 underbaseplate-pad. These elastic elements are manufactured from natural rubber, corkelastomer composites and kevlar fibres and aim at reducing the vibration energy entering the bridge structure.

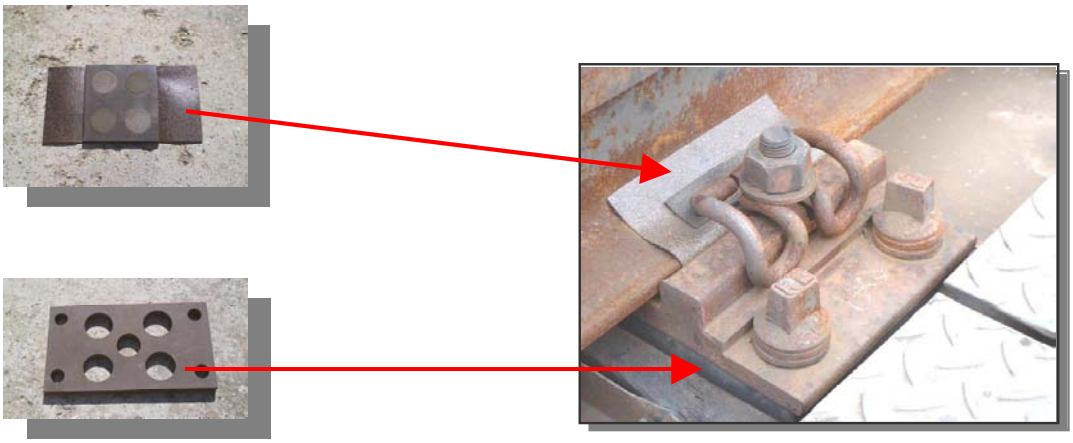


Figure 30 Rail fastening with CDM flapped rail pad & under base plate pad

As a consequence, the dynamic forces generated by the wheel/rail interaction put the rail into increased motion and therefore the rail can easily become an important airborne noise source. This rail reradiated noise component is to be reduced by a special tuned absorber fixed to the rail web.

Figure 31 Tuned rail damper as part of the CDM FERPONT system

The noise radiation from the walking plates was treated by exchanging the steel plates for plates of a damped plastic composite material. This system was specially developed for general bridge applications, and its damping was optimised for the given case.

Figure 32 Installation of damped walking plates of composite material

5.2.2.1 Results

Figure 33 shows the comparison of noise spectra, measured before and after the conversion works on the balcony of the new National Theatre, at approximately 150 m distance from the bridge. Depending on the speed, load and other parameters of the trains, both the vibration and the noise level reduction ranges from 5 to 8 dB. In other words, this means that the train traffic along the bridge does no longer stand out from the general background noise of the area.

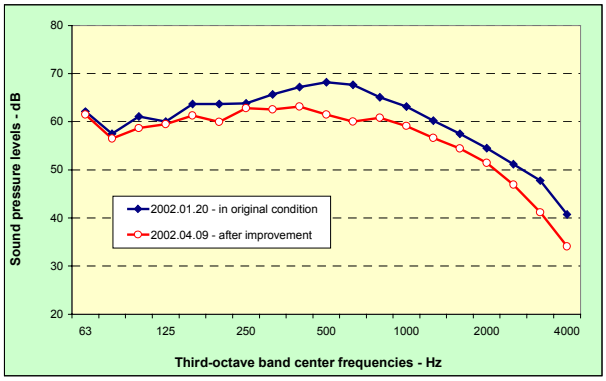


Figure 33 Comparison of maximum noise spectra, measured for the pass-by of a passenger train

Within 7 months, some 8 dB noise reduction had been realised for a budget of approximately 1.2 M EUR. Measuring over time showed however, the importance of regular rail grinding to keep the noise & vibration control measures efficient. **Vienna Wasserpark Bridge [5.4]**

The 82 meter long steel Vienna based bridge has two tracks and dates back to a period where noise control was not an issue. It supports a daily traffic of about 350 passenger trains and 20 fright trains and is a nuisance for the many visitors of the nearby water park of the "Alten Donau".

Figure 34 Wasserpark Bridge

A classic bridge renovation trough replacement by more silent concrete structures was however out of budget. The problem had nevertheless to be solved and should serve as an example of successful and economical noise mitigation for the numerous other steel bridges in Vienna.

The idea to apply vibration dampers was adopted. The train induced vibrational energy should be absorbed allowing the steel bridge to radiate less noise.

The applied special vibration dampers consist of a sandwich like assembly of metal plates and rubber sheets. They were mounted in places where the amplitude of vibration is maximal for instance on and under the rail and also on the structural elements of the bridge.

Figure 35 Example of S&V rail damper

Two other important measures to prevent the excitation of the bridge were taken as well:

1. replacement of the old and very worn rails by new smooth ones;
2. relocation on the main land of the existing compensation device for temperature induced rail length variations. This device was located on the bridge and proved to have been a serious cause of vibrational excitation at each train passage.

Measurements were performed at close range as well as at 150 m of the bridge. The combination of mitigation measures allowed a total noise reduction of about 20 dB, which exceeded all expectations. New rails and the relocation of the rail length compensation device accounted for a 10 dB reduction, the bridge and rail dampers for about 6 dB.

The overall cost for this project of some 0.5 M Euro was well below the budget of the other bridge renovation options.

5.2.3 Nieuwe vaart Bridge [5.2]

When new railway bridges are build concrete is often preferred because of the bad noise emission reputation of steel bridges. Dutch researchers challenged this fact and adopted the idea to develop a silent steel bridge. Their new steel bridge concept should show no noise increase compared to ballasted track with timber sleepers.

They investigated the potential of damping of the rail, damping of the bridge structure, the principle of spring mass damping constructions, different girder shapes and composite girders, local stiffeners and different plate thickness combinations.

To prevent the excitation of the bridge structure the following measures were implemented in the final bridge design:

1. rail damping by embedding the rail with pourable cork-rubber in a gutter (see figure 36);
2. vibration isolation between the rails and the bridge by inserting a resilient material under the rail foot (see figure 36).

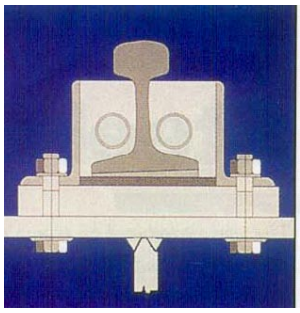


Figure 36 Embedded rail in gutter

3. Intelligent design of the bridge structure. Increased plate thickness, differences between plate thicknesses, prevention of parasite bridge deformations by placing each rail directly above the side girders of the bridge (see figure 37).

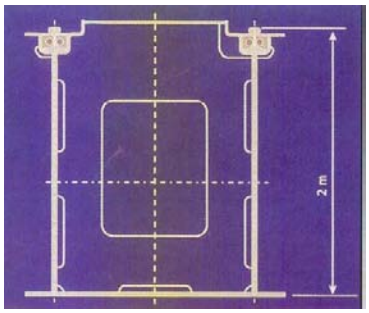


Figure 37 Integrated bridge design with direct rail support

In order to reduce the residual vibration level of the bridge structure the application of sandwich damping (energy absorption in a visco-elastic layer placed between the construction and an additional thin metal sheet) within the box girder was considered.

Vibration measurements showed however that the vibration level of the bridge had already become very low and that the structure born bridge noise was not dominant any more. The further reduction of the bridge vibration would not significantly reduce total noise level and sandwich damping was therefore not applied.

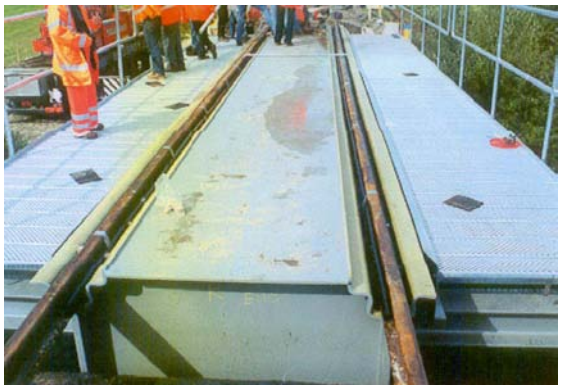
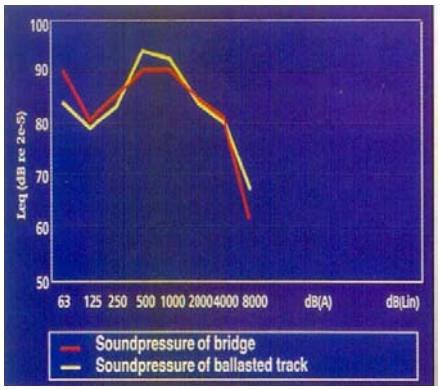
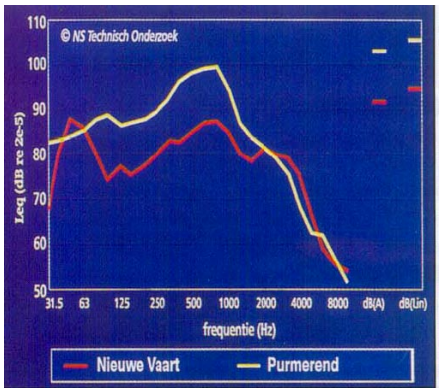


Figure 38 Nieuwe vaart bridge

The results of the noise level measurements were better than expected. The new silent bridge concept withstood the comparison with normal ballasted track and showed a noise reduction of 8 dB(A) in comparison with a reference bridge.

Figure 39





Noise measurement comparisons of silent bridge with ballast track and reference bridge

The new and simpler bridge design compensates for the somewhat increased material costs. The silent bridge can in any case compete on cost with traditional steel railway bridges.

5.2.4 Bridge over river Mur in Austria [5.5]

Close to Leoben the Austrian the railways company ÖBB uses a set of two parallel bridges to cross the river Mur. The bridge for track 1 dates from 1939, the other bridge for track 2 was build in 1969. This part of the rail network between Vienna and Italy supports a daily traffic of about 280 trains, which can run up to 140 km/h.

The bridge for track 2 consisted of a steel armature construction with a length of 73.8 meter directly supporting an open track without ballast. The very high noise emissions of 112 dB and need for an upgrade of this bridge to a higher load class triggered the modification project.

The following options were reviewed:

- lateral noise screens on the bridge deck;
- steel reinforced concrete slab on top of the bridge to support the ballasted track;
- steel reinforced concrete slab on top of the bridge with an integrated embedded Edilon Corkelast® rail system;
- replacement of the old bridge by a new Silent Steel Bridge®;
- replacement of the old bridge by a new concrete bridge.

Budget and technical considerations resulted in the following implementation:

- reinforced concrete slab on top of the bridge (Increased mass and stiffness of bridge deck for decreased vibration sensitivity. Downward acoustic closing of bridge deck);
- embedded Edilon Corkelast® rail system (vibro-acoustic isolation);
- reinforcement of the bridge armature by welding of extra steel parts (for higher load class and support of concrete slab construction);
- 2-meter highly absorbing noise screen on full length of the north side of the bridge;
- noise absorbent material on bridge deck surface.

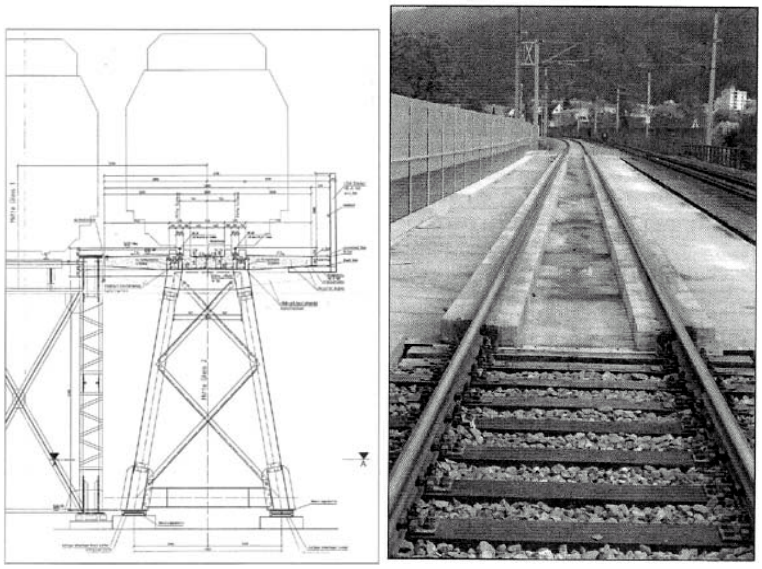


Figure 40 Cross section and general view of the renovated bridge

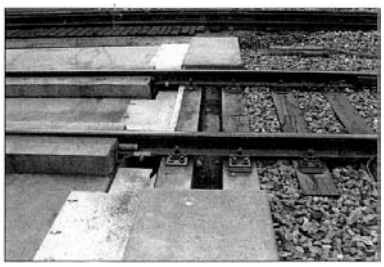


Figure 41 Connection of embedded bridge track to ballasted land track

Measurements with microphones at different locations close to the bridge showed a noise reduction of up to 19 dB. This represents a division by four of the subjective noise perception. Tests were done with a freight train (locomotive with four wagons), rail roughness did not change during the measurement campaign.

v [km/h]	MP-1a $\Delta L_{A,max}$ [dB]	MP-1b $\Delta L_{A,max}$ [dB]	MP-1c $\Delta L_{A,max}$ [dB]	MP-2 $\Delta L_{A,max}$ [dB]	MP-3 $\Delta L_{A,max}$ [dB]	MP-4 $\Delta L_{A,max}$ [dB]
60	-14	-18	-17	-5	-5	-7
80	-14	-19	-19	-4	-4	-6
90	-13	-19	-19	-5	-4	-7

Figure 42 Differences in noise peaks at 6 measuring points for train speeds of 60, 80 and 90 km/h. Measuring points MP-1 a/b/c very close to the bridge, MP-3 and MP-4 in the immediate surrounding of the bridge and MP-2 at 450 m from the bridge

Application of special noise absorbent material (wood/concrete panels) on the bridge deck surface proved not to be significant for noise mitigation.

Replacement of the bridge would have cost some 3.0 million Euro. The refurbishment of the bridge had just cost about one third of this amount.

5.3 NEW CONCEPTS

Some new interesting active and passive bridge structure damping devices have appeared. They do however not operate on the track level and are thus out of scope of this report.

No new structure born noise mitigation concepts regarding only the bridge track are available for the moment.

5.4 CONCLUSIONS

The total sound pressure produced by trains on steel bridges can be 5 to 15 dB higher than on land based ballasted track with timber sleepers.

Each existing steel railway bridge is specific and will require a noise mitigation approach considering all of the bridge infrastructural elements.

WP3.2.2 of the QCITY project focuses on mitigation measures for rails. The European Sustainable Bridge Project is already taking care of matters concerning the bridge structure. The recommendations of this report focus therefore on solutions regarding the track of existing steel bridges only.

To reduce structural noise when a train passes over a steel bridge one can prevent the excitation of the bridge structure by:

1. reducing rail vibrations (lower rail contact roughness, avoid rail discontinuities);
2. isolating the bridge structure from rail vibrations (ad vibration control measures in rail fixation, rail support or track support).

Effective track mitigation measures for bridge structure born noise of existing steel bridges are:

- reduced rail support stiffness;
- resilient rail fixation;
- increased mass of track support;
- low rail roughness (regular grinding);
- move of special track work (rail discontinuities) from the bridge to the mainland.

These measures allow to obtain good technical (in general reductions of about 8 dB) and economical results for existing steel bridges.

No new structure born noise mitigation concepts regarding only the bridge track are available for the moment.

5.5 REFERENCES

[5.1] Reducing noise from steel Bridges, Innovative solutions for noise reduction, Franck Poisson, SNCF - 2003.

[5.2] Dutch group cuts steel bridge noise, Jelte Bos, Holland Rail consult, International Railway Journal, September 1997.

[5.3] Noise and vibration control of the South Railway Bridge of Budapest – P. Carels, World Congress on Railway Research, Edinburgh, October 2003.

[5.4] Dämpfung des Körperschallpegels im Brückerkörper, S&V, Wien, December 2000.

[5.5] Ing. Herwig Riegler, Martin Hanisch, Brückensanierung, Lärmtechnische Sanierung der Murbrücke in Leoben bei den ÖBB. EI - Eisenbahningenieur (56) p 24, 7/2005.

6 MITIGATION MEASURES FOR RAIL: NEW RAIL PROFILE, INFLUENCE OF RAIL PAD STIFFNESS

6.1 INTRODUCTION

This document comprise the influence of a new developed rail profile geometry on the rail noise emission and the influence of different rail pad stiffness to the overall noise emission.

By using numerical analysis methods (done by the company AVL under contract from VAS) voestalpine Schienen GmbH optimised, the rail profile UIC60 regarding its acoustic behaviour. As a result, a new rail profile VA71b with optimised characteristics was designed (profile drawings UIC60, VA71b see Figure 43). The profile VA71b has a higher vertical and horizontal bending stiffness and a higher weight per meter than the UIC60 (see **TAB 1**). Both properties improved the acoustic behaviour (lower noise emission). In the following report, we will talk about the development of this profile and field measurements done in the track of the Austrian railways ÖBB.

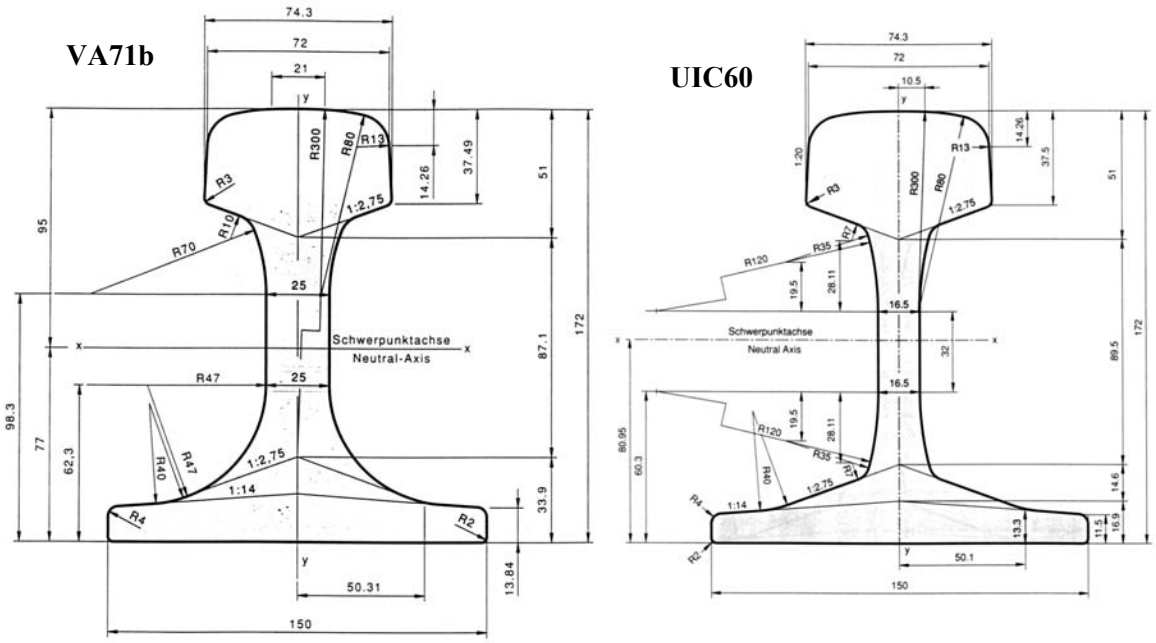


Figure 43 Comparison VA71b - UIC60 rail profile

	moment of inertia I_x [cm^4]	bending stiffness [Nm^2]	mass per meter [kg/m]
UIC60	3055.0	6.4*10^6	60.34
VA71b	3181.4	6.9*10^6	71.44

Table 15 Characteristic values for the profiles UIC60 and VA71b

6.2 DEVELOPMENT OF THE VA71B PROFILE

The company AVL did numerical analysis under contract of VAS to improve the noise behaviour of the standard rail UIC60. The following recommendations were given by AVL to improve the noise behaviour this profile.

The rail profile UIC60 should have:

- a thicker web;
- increased food height;
- bigger radii from the food region to web.

Because of this recommendations the new profile VA71b was developed.

After the development, the rail profile was tested in track tests in two different ways and in both cases compared to standard rails profiles.

Method 1: impulse hammer method

The VA71b is stiffer than the UIC60 rail profile, so its transfer-mobility (sensitivity against vibration stimulation - transfer function, see Figure 44) is lower.

For rails characteristic heights in noise emission at 1500-2000 Hz were transferred to higher frequencies (2300-2600 Hz) at the same time the absolute value of the maximums were reduced (see Figure 45).

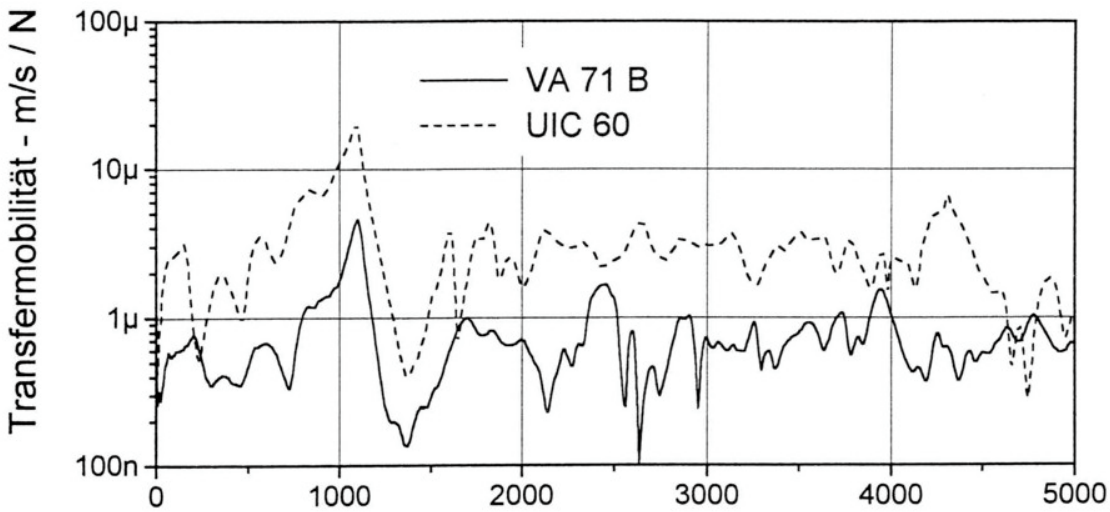


Figure 44

Transfer function, comparison profile VA71b, UIC60

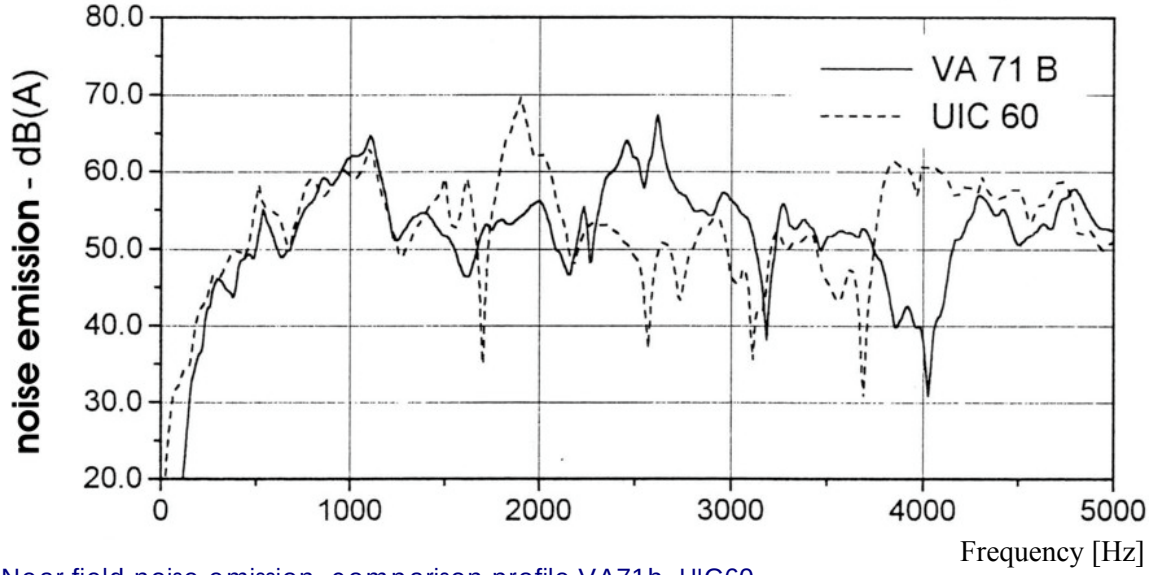


Figure 45

Near field noise emission, comparison profile VA71b, UIC60

Method 2: train pass by measurements

Here the stimulation takes place at lower frequencies compared to the hammer impulse. This means the stimulation amplitudes at higher frequencies are lower. Because of that the increased emission (at about 2500 Hz) of the VA71b is not of importance in this case.

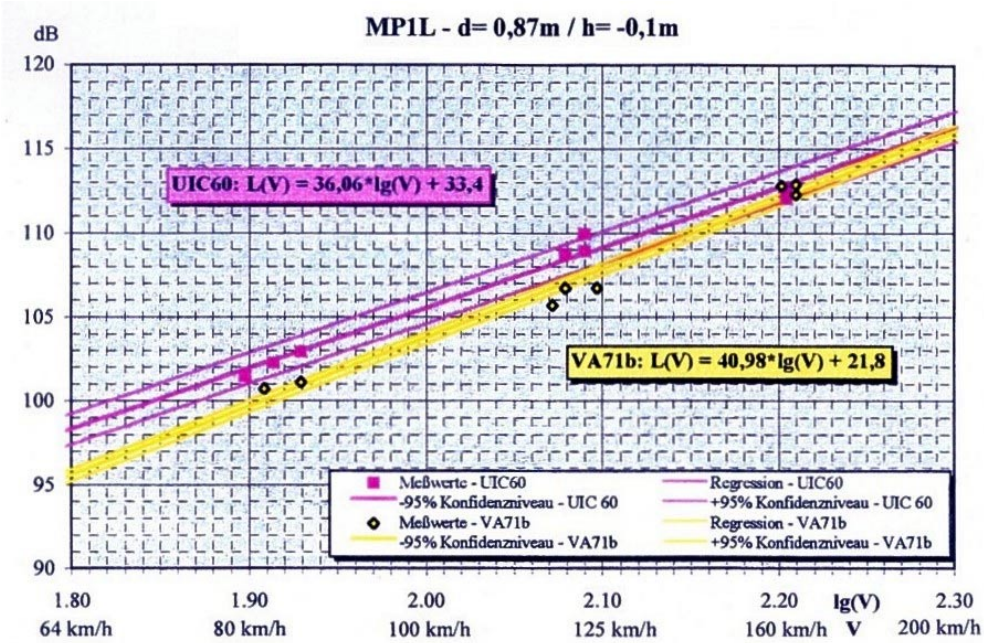
At 1500 Hz the measured emission values for the VA71b are 10-20 dB lower then the values of the UIC60.

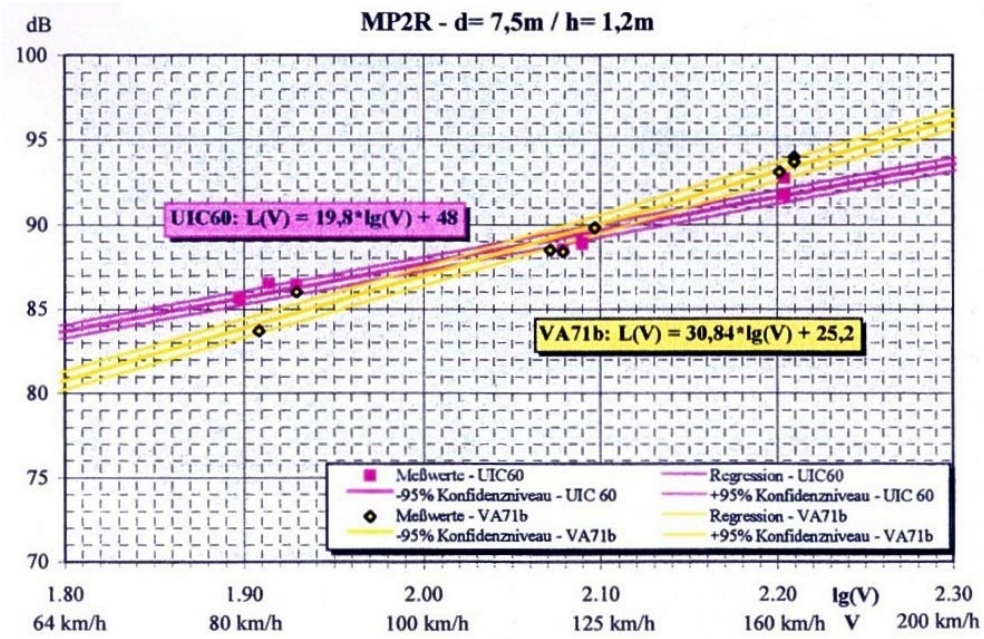
The difference at the important frequency of about 800 Hz was negligible.

The analyses of the measurements can be summarised as follows:

- the VA71b rail shows a tendential lower noise emission compared to the UIC60 rail;
- the advantage regarding the sound emission of the VA71b rail compared to the UIC60 rail is higher in the linear evaluation than in the A-weighted evaluation, decreases with the distance of the measuring point from the rail track and also decreases with increasing speed of the train and becomes negative at a train speed of approximately 160 km/h;
- the advantage regarding the sound emission is higher for louder car classes and is in this case highest for speeds of approximately 80 km/h at a level of at least 2 dB.

Figure 46





Dependence of the noise emission on the train speed for the UIC60 and VA71b rail for two different measuring points in different distances of the track. MP1 with a distance of 0,87m and MP2 with a distance of 7,5m from the track [6.1]

Because of these positive but quantitative not convincing results a further step were set to optimise the system further. Because of analyses done by TNO (see SP 3.1.), it is known that higher pad stiffness lead to better results (see Figure 47).

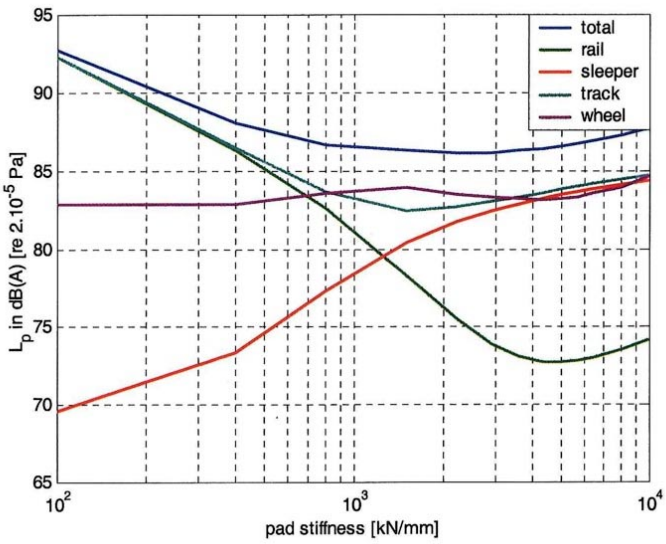


Figure 47

Total sound pressure levels at 7.5 m for the VA71b rail type for a train speed of 80 km/h as a function of the pad stiffness [6.2]

Because of this calculations a new test track at the Austrian railways ÖBB was built. There it was possible to compare different superstructures together. In two measurement series, the following substructures were compared.

- UIC54 on wooden sleepers (*Track_1*);
- VA71b on concrete sleepers with rail pads with a nominal static stiffness of 85 kN/mm (*Track_2*);
- VA71b on concrete sleepers with rail pads with a nominal static stiffness of 700-1000 kN/mm (*Track_3*).

Series 1: comparison Track_1 - Track_2 [results see Table 16]

Series 2: comparison Track_2 - Track_3 [results see Table 17]

Before each series both rails were grinded and the rail roughness was measured (Figure 48, 49).

A-weighted noise emission for 80km/h (d=2,35m/h=0,5m) $L_{pb,80}$ [dB(A)]		confidenze region -95%	-90%	mean value	confidenze region +90%	+95%
grey iron block braked wheels	Track_1	102,3	102,5	102,9	103,3	103,4
	Track_2	100,2	100,3	100,8	101,3	101,4
		differenz		-2,1 dB(A)	decrease	
disk braked wheels	Track_1	87,5	87,7	88,3	88,8	89,0
	Track_2	89,2	89,3	89,8	90,3	90,5
		differenz		+1,6 dB(A)	increase	

Table 16

Comparison between *Track_1* and *Track_2* [1]. The increase for the less louder cars is not of importance to the overall noise emission.

If we use this values to calculate the noise emission under ÖAL 30 (Austrian standard for noise calculations) this results in a noise reduction by 1.8 dB(A) on day and 2.0 dB(A) at night.

Table 17

A-weighted noise emission for 80km/h (d=2,35m/h=0,5m) $L_{A,m,80}$ [dB]		minimum	mean value	maximum
grey iron block breaked wheels	Track_3	*99,2*	*102,1*	*104,7*
	Track_2	*105,0*	*106,4*	*108,8*
		differenz	*-4,3 dB(A)*	decrease

Comparison between *Track_2* and *Track_3* for block braked trains [6.1]

This leads to the following additional conclusion:

- in combination with stiffer rail pads the noise reduction reaches a value of at least 4 dB (A)

Comments to the rail roughness:

For measurement series 1, the roughness of both rails is to high for smaller wave lengths in accordance to the prEN ISO 3095 standard (Figure 48). The VA71b (*Track_2*) rail is rougher for this wave length so the measured noise emissions are on the conservative side. For measurement series 2, the roughness for *Track_2* and *Track_3* is quite similar (Figure 49).

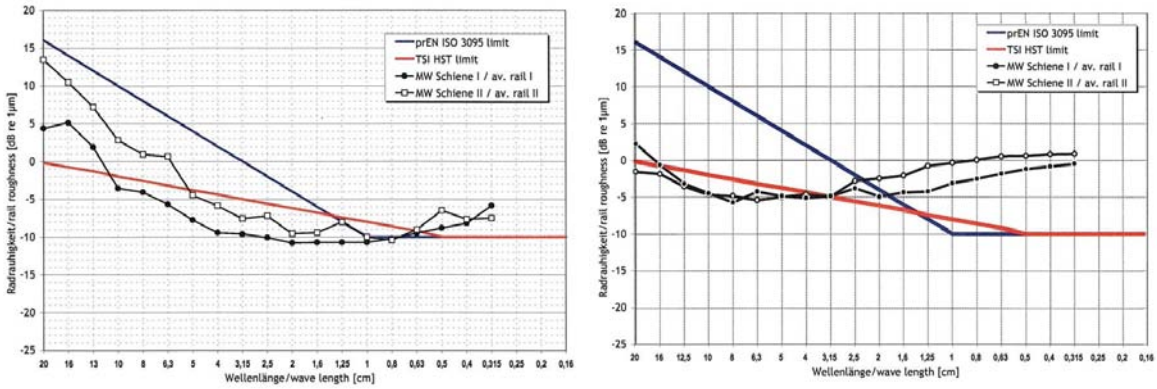


Figure 48

Roughness of Track_1 and Track_2 for measurement series 1 [6.1]

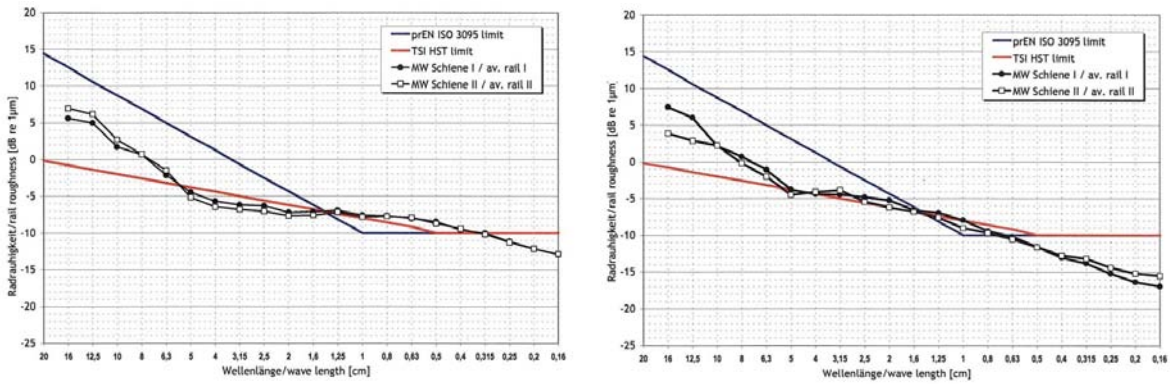


Figure 49

Roughness of Track_2 and Track_3 for measurement series 2 [6.1]

6.3 COST COMPARISON

The VA71b rail can be used like all other common rail profiles. No additional components are necessary to use this profile. So no extra costs will accumulate. The weight of the profile is approximately 20% higher than the weight of the common UIC60 rail and so also is its price.

The welding of this profile is not more complicated than welding of other profiles.

The only thing which have to keep in mind is that profile specific parts (for example fish-plates, clamping for train protective devices) have to be modified.

6.4 CONCLUSION

With the new designed rail profile VA71b - which can be used in open track - it is possible to decrease the noise emission for louder car classes up to 2 dB(A).

In combination with higher rail pad stiffness, it is possible to decrease the noise emission even further. Noise reduction up to 4 - 4.5 dB(A) are possible.

It has to be noted that compared to a third VA71b noise test track our measured noise emission values are up to 5db(A) higher. The differences between our discussed test tracks and the low noise test track are the following:

- roughness of the in this report discussed test tracks is a little bit high.
- the sleepers of our test track have plastic pads, theoretically this pads lead to an higher vibration of the sleepers and so also to an increased noise emission of the system.

These points have to be investigated in detail in new test tracks.

6.5 REFERENCES

The measurements and calculations in this report were done by

[6.1] psiA-Consult (Umweltforschung und Engineering GmbH, Manfred T.Kalivoda) and

[6.2] TNO TPD both in contract to VAS.

7 SPECIAL RAIL PROFILES

7.1 SUMMARY

This document summarises the different special rail profiles and their effect on the roar noise emission on straight track only. The different rail profiles are briefly presented and an estimation of the noise reduction and the cost is given for each solution. The estimation is made for rails resting on soft to medium soft rail pads by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness.

Predictions are also made for a combination of the rail solution and one selected wheel damper ('shark's fin'), indicating the noise reduction anticipated by combining wheel and rail damping.

All solutions are in a design phase, as they represent new rail designs as well as new innovative concepts. Therefore, a lot of practical issues are still to be solved.

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	ExtraCost [€/m]	Limitations/ Critical Matters
Reduced stiffness	8	6	100	Manufac/Stress
Saddle profile rail	7	5	1000	Prize/Stress
VA71b + Stiff pads	Not estimated	6	[12]20	Stiff pads, wear

Table 18

Additional cost for wheel dampers ('shark's fin') is about 500 €/wheel.

All reductions are fairly high and equal. The predicted total noise reductions for combined wheel and rail measures are 7 – 8 dB(A), if the contribution of other noise sources than the wheel/rail-system are ***not*** significant (e.g. traction gears). Thus, a combination of wheel and rail damping might have a very good effect on the rolling noise, but the noise reduction for rail measures only is typically as much as **5 dB(A)**.

When extra costs for the special rail profiles are considered, the evaluated measures can be ranked as follows (rail measures only):

1. VA71b + Stiff pads (3.6 €/m/dB(A))
2. Reduced stiffness (18 €/m/dB(A))
3. Saddle profile rail (200 €/m/dB(A))

[12] Extra costs for maintenance is not included (e.g. more frequent inspections and rail grinding).

7.2 INTRODUCTION

This document summarises the different special rail profiles and their effect on the roar noise emission on straight track only. The different rail profiles are briefly presented and an estimation of the noise reduction and the cost is given for each solution. The estimation is made for rails resting on soft to medium soft rail pads by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness (except for 'VA71b combined with stiffer rail pads', which has been evaluated by VAS).

Predictions are also made for a combination of the rail solution and one selected wheel damper ('shark's fin' damper tuned below the fundamental wheel resonance frequency but with broadband characteristics for all significant wheel resonance frequencies), although wheel dampers are not part of WP3. This is done to indicate the noise reduction effects of combining wheel and rail damping, which is of particular interest when rail damping only is not sufficient.

All solutions are in a design phase, as they represent new rail designs as well as new innovative concepts. Therefore, a lot of practical issues are still to be solved.

7.2.1 Theoretical outline

The different rail damping projections have been studied and evaluated by using a mathematical model for vertical direction of wheel/rail system mainly based on decay rates, mechanical impedance data of rail as well as wheel/rail contact stiffness. It is assumed that lateral rail vibrations and the track infrastructure (i.e. sleepers and ballast) are not contributing significantly to the noise level of the wheel/rail system, which is sufficiently true for rails resting on soft to medium soft rail pads [7.1] (the mathematical model is not applicable for fairly stiff rail pads, but this is not a big problem as rail dampers are much less effective for rails resting on stiff pads).

The rail impedance is described by an analytical expression of a damped slender beam[13] resting on a continuous and damped rail pad system. The vertical wheel impedance is simply described as a big solid mass [7.2] for studied frequencies between 500 – 2500 Hz. Furthermore, it is assumed that vertical wheel vibrations are effectively coupled to lateral and resonant wheel vibrations by the inclined/curved web and that these vibrations are dominating the noise level contribution of the wheel. The wheel rail contact spring is modelled as a discrete and vertical spring element mounted directly to the wheel and rail contact areas.

The effective radiation area of rail (radiation length x radiation area per meter length of rail x radiation coefficient[14]) is used as a parameter reflecting the total sound power

[13] Corrected for shear deformation of rail web.

[14] Radiation coefficient ≈ 1for frequencies greater than 500 Hz.

radiated by the rail. This parameter is directly related to the decay rate of rail in dB/m, which in turn is proportional to the apparent loss factor[15] of rail.

The wheel/rail system is excited by a "moving roughness strip" located in the middle of this wheel/rail contact spring (i.e. split spring element). Thus the wheel/rail roughness induced vibrations tends to propagate more down to the rail, if the rail impedance is lower than the wheel impedance including contact stiffness and vice versa.

There is a "cut off" frequency at about 800 - 1000 Hz caused by the contact stiffness of a typical wheel/rail system (1.1E+9 N/m). Above that frequency range, there is a significant isolation effect for vibrations propagating down to the rail. For lower frequencies, there is no such isolation effect. Further decrease of the wheel/rail contact stiffness will result in corresponding decrease of the dynamic contact forces for higher frequencies, which in turn will result in almost equally decreased wheel and rail sound level contributions. This observation is very important, as there is just little or no reduction effect at all on the noise contribution of wheel for most other rail measures evaluated.

Vertical decay rates (apparent loss factor, bending stiffness, mass per length of rail), mechanical point impedance (bending stiffness, mass per length of rail) and wheel/rail contact stiffness are often sufficient input data for approximate prediction and ranking of total sound level reductions caused by different rail measures for e.g. UIC60 rails resting on soft to medium stiff pads for the most significant frequencies (500 – 2500 Hz).

[15] Including stop band effects of the periodic rail-pad-sleeper structure.

7.3 DESCRIPTION OF STUDIED SOLUTIONS

7.3.1 Rail profile with reduced contact stiffness

The idea behind the rail profile with reduced contact stiffness is to reduce the exciting forces of the wheel-rail system. This rail profile is still in a design phase and all issues regarding production and strength requirements are not completely investigated. Figure 50 below shows an example of a basic design of a rail profile with reduced contact stiffness.

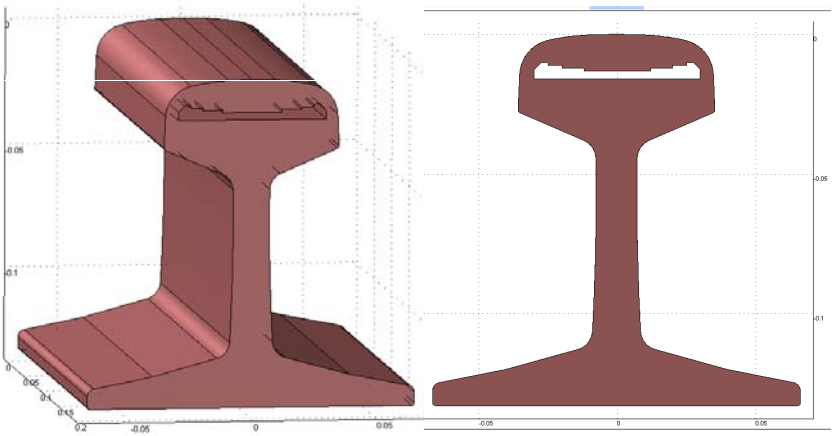


Figure 50 Conceptual design of the rail profile with reduced contact stiffness.

The design goal of the rail profile with reduced contact stiffness has been to decrease the rail contact stiffness by a factor 3 - 4 and still maintaining the strength requirements (i.e. keeping the stress levels under the endurance limit). The rail profile in Figure 50 has the following characteristics:

Rail contact stiffness: Reduced by a factor 3.23
Maximum stress level: 226 MPa (von Mises, see Figure 51)
Maximum rail head contact area compression: 0.073 mm (50 kN)
Dimensions of 'bridge': 60 mm wide
17 mm thick on the top
Approximately 5 mm thick at the edges

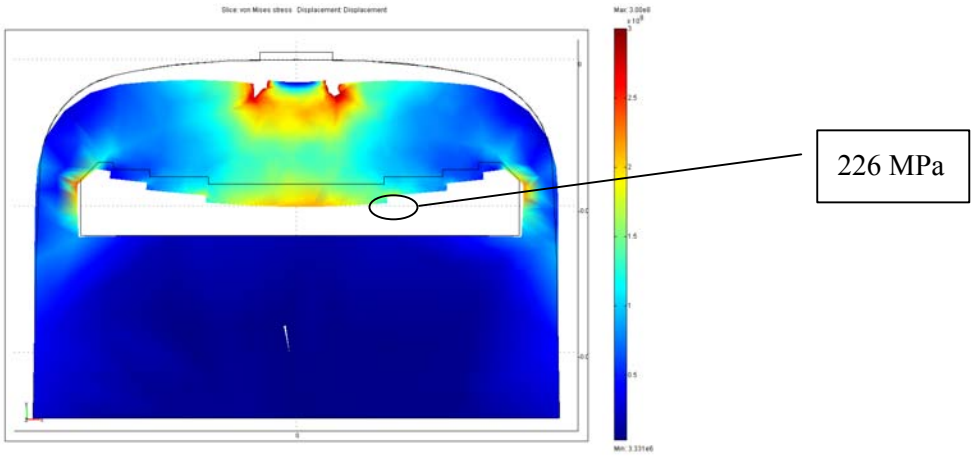


Figure 51 Strength analysis of the rail profile according to Figure 50

7.3.2 Saddle profile rail

The saddle profile rail was developed in the FP5 Growth Corrugation project as a measure for reducing corrugation. In QCITY the sound reducing effect of the saddle profile rail is studied. Figure 52 shows the basic design of the saddle profile rail.

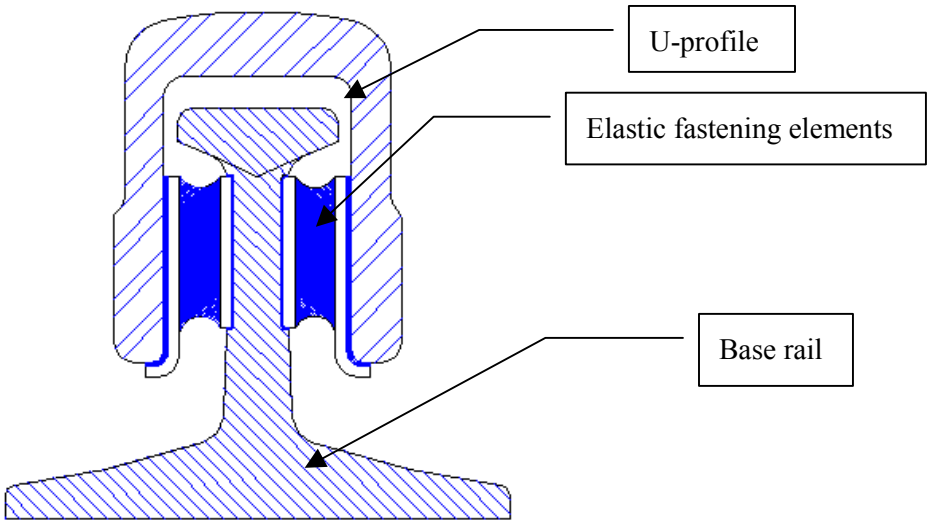


Figure 52 Saddle profile rail.

The add-on rail profile was optimised to give a low vertical resonance frequency when loaded with the train axle load during a train passage. The maximal vertical deflection of the U-profile during a train passage was calculated to be between 4-5 mm.

In the QCITY project, the main objective for the saddle profile rail is to evaluate the effect on squeal noise, but rolling noise will be addressed as well. The saddle profile rail is 'weak' in lateral direction. This will probably have a positive effect on squeal noise. Since the U-profile of the saddle profile rail has very low internal losses, mainly due to the continuous support, the decay rate will be very low. This problem will have to be resolved in order to achieve a significant noise reduction. In the QCITY project, different ways to increase the loss factor will be studied during the next 6M.

Thus, the primary purpose of implementing constrained layer damping of the U-profile is to compensate for the expected decrease in apparent loss factor, when the point impedance of rail is significantly increased.

7.3.3 New profile rail (VA71b) and stiff pads

The basic idea behind this new rail concept from VAS is to combine a sound optimised rail profile (VA71b) with sound optimised (stiff) rail pads [7.5], [7.6].

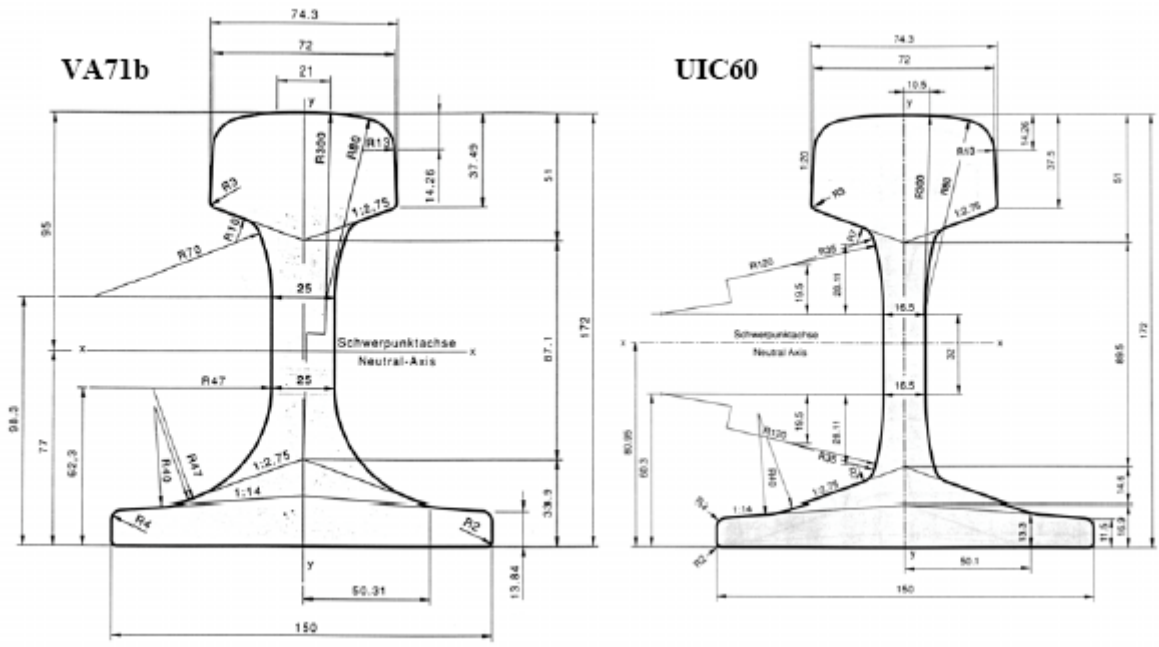


Figure 53

The new profile rail (VA71b) and a UIC60 rail.

7.4 PREDICTED SOUND LEVEL REDUCTIONS

In this chapter predicted noise reductions of the evaluated special rail profiles are presented alone and in combination with wheel dampers for a speed of 80 kph.

The reductions are based on reference data derived from pass-by noise measurements in Tjörnarp [7.3] (in the southern part of Sweden) for different trains and speeds and from the decay rates of the corresponding UIC60 rails [7.4].

According to these reference measurements, there is only ***one dominating octave band*** in the measured bass-by noise levels (TEL in dB(A)). Therefore, the predicted total noise reduction (dB(A)) of the wheel/rail-system at 80 kph (due to the special rail profile) is considered to be as large as the corresponding noise reduction (dB) of the dominating octave band (here 800 Hz). This is often fairly true, but in some cases, there is a significant difference (positive or negative).

Furthermore, the wheel/rail-system is considered to be the dominating noise source (at least before rail damping). This is often true for modern electric trains running with lower speeds up to about 80 kph, but it is **not** true for e.g. metro cars with self ventilated motors. For higher speeds, the noise contribution of the traction gears tends to be more significant and in some cases even dominating. Then, of course, the effects of special rail profiles will be less than predicted.

7.4.1 Rail profile with reduced contact stiffness

The rail profile with reduced contact stiffness has been evaluated regarding prediction of sound level reductions in relation to the UIC60 reference rail in Tjörnarp.

Sound level reductions (-) in dB(A) for this rail profile at 80 kph:

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	-5.7	-5.7	-5.7
Wheel+Rail	-11.7	-7.7	-9.0

Table 19

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least 80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total noise reduction of about **-8.4 dB(A)** (i.e. the average of -7.7 and -9.0) can be anticipated by combining this rail profile with damped wheels ('shark's fin'). The sound reduction for the rail profile with reduced contact stiffness only is **-5.7 dB(A)**.

Note that the rail profile with reduced contact stiffness is the only one of the evaluated rail profiles, which will reduce equally much the rail and wheel noise contributions and

where the predicted wheel and rail noise reduction is as much as -5.7 dB(A) combined with ordinary undamped wheels. That is as good as most other evaluated rail profiles combined with damped wheels ('shark's fin').

Changes of significant parameters:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	0	-1.24[16] dB/m
Apparent rail loss factor:	0	0.15
Mass per m length of rail:	0	60 kg/m
Dyn. bending stiffness of rail:	0	63E+5 Nm^2
Rail/pad frequency:	0	300 Hz
Wheel/rail contact stiffness:	-50	1.1E+9 → 0.55E+9 N/m
a) Rail contact stiffness:	-67	2.2E+9 → 0.73E+9 N/m
b) Wheel contact stiffness:	0	2.2E+9 N/m

Table 20

7.4.2 Saddle profile rail

A damped version of the saddle profile rail, based on the UIC60 reference rail in Tjörnarp, have been evaluated regarding prediction of sound level reductions.

There is no noise reduction expected for the ordinary undamped saddle profile rail, if no extra damping is added to the U-profile. This is because the apparent loss factor of the continuously supported U-profile will be very small in comparison to the UIC60 reference rail. The main reason is that the continuously supported U-profile does not show any stop band effects at all. A periodic structure like the UIC60 reference rail shows clear stop band effects which also increases the apparent loss factor.

Sound level reductions (-) in dB(A) for the damped saddle profile rail at 80 kph:

Damped comp.	Wheel reduction	Rail reduction	Wheel+Rail red.
Wheel	-6.0	-2.0	-3.3
Rail	-10.8	-4.1	-5.8
Wheel+Rail	-16.8	-6.1	-8.2

Table 21

A correction for shear deformation of rail web (+0.6 dB(A)) is included for the second and third row numbers.

It is assumed that the A-weighted sound power level of the rail exceeds the sound power level of the wheel by 1 – 2 dB(A). Sometimes and for speeds up to at least

[16] Measured for UIC60 rail in Tjörnarp [4].

80 kph, this difference might be significantly greater than 1 – 2 dB(A). If the difference would be much greater, the total sound reduction will be completely dominated by the sound reduction of the rail. Hence the mid column will be more relevant. Thus, a total noise reduction of about **-7.2 dB(A)** (i.e. the average of -6.1 and -8.2) can be anticipated by combining the saddle profile rail with wheel dampers ('shark's fin'). The sound reduction for the damped the saddle profile rail only is **-5.0 dB(A)** (i.e. the average of -4.1 and -5.8).

Changes of significant parameters:

Parameter	Change (%)	Before → After
Decay rate of rail at 800 Hz:	+64	-1.24 → -2.03 dB/m
Apparent rail loss factor:	0	0.15 → 0.15 for β_2 = 1.0
Mass per m length of rail:	-58	60 → 25 kg/m (U-profile)
Dyn. bending stiffness of rail:	-93.5	63E+5 → 4.09E+5 Nm^2
Rail/pad frequency:	-67	300 → 100 Hz
Wheel/rail contact stiffness:	0	1.1E+9 N/m → less?

Table 22

7.4.3 New profile rail (VA71b) and stiff pads

The sound optimised profile rail (VA71b) combined with sound optimised stiff pads has been evaluated by VAS and is presented in [7.5], [7.6].

The indicated total sound reduction of using the new profile rail (VA71B) combined with stiff pads (static stiffness 700 – 1000 MN/m) is about -4 to -4.5 dB(A) at 80 kph according to field measurements [7.5] and **-5.5 dB(A)** (the average of -5 and -6 dB(A)) when based on TWINS calculations [7.6], [7.7].

7.5 COMMENTS

It should be noticed that the main quantities governing the noise contribution of the rail is the apparent loss factor, the corresponding decay rate and to some extent also the mechanical point impedance of the rail. The decay rate is proportional to the apparent loss factor of the rail divided by the wave length, which in turn (slender beam) is proportional to (bending stiffness/mass per length of rail)$^{1/4}$. Obviously, minor changes in bending stiffness and/or mass per length of the rail are ***not*** significantly influencing the decay rate and the corresponding noise reduction of the rail. Therefore, the increase of the decay rate must primarily be achieved by increasing the apparent loss factor of the rail (i.e. rail loss factor and/or pad stiffness).

The distribution of the wheel/rail-roughness vibrations to the wheel and to the rail depends on the mechanical point impedance of the rail and the wheel/rail contact stiffness. The mechanical point impedance of the rail (slender beam) is proportional to (bending stiffness)$^{1/4}$ x (mass per length of rail)$^{3/4}$. For example, an increase of the mechanical point impedance by increasing the bending stiffness and mass per length of rail will significantly reduce the noise of an ordinary rail above the 'cut off' frequency (due to the contact stiffness) typically at 800 – 1000 Hz, but the effect on the wheel is negligible. Below the 'cut off' frequency, but above 500 Hz, there is instead a significant noise reduction of the wheel but no reduction of the rail. Below about 500 Hz the coupling effects of the wheel/rail system to the track infrastructure are getting more and more complex. Therefore, general "rules" can not be defined for that frequency range. However, frequencies below the 'cut off' frequency are often less important for the pass-by noise.

7.6 CONCLUSIONS AND COSTS

The predicted total noise reductions of the evaluated solutions at 80 kph are:

Type of Rail Measure	Noise reduction of Wheel+Rail Measures [dB(A)]	Noise reduction of Rail Measures [dB(A)]	Extra Cost [€/m]	Limitations/ Critical Matters
Reduced stiffness	8.4	5.7	100	Manufac/Stress
Saddle profile rail	7.2	5.0	1000	Prize/Stress
VA71b + Stiff pads	Not estimated	5.5	[17]20	Stiff pads, wear

Table 23

Additional cost for wheel dampers ('shark's fin') is about 500 €/wheel.

It can be concluded that all reductions are fairly high and equal. The noise reduction anticipated for rail measures only is typically as much as **5 dB(A)**. The predicted total noise reductions for combined wheel and rail measures are 7 – 8 dB(A), if the contribution of other noise sources than the wheel/rail-system are ***not*** significant (e.g. traction gears). That is very promising for this project. Thus, a combination of wheel and rail damping might have a very good effect on the rolling noise, but the noise reduction anticipated for rail measures only are also good.

When extra costs for the special rail profiles are considered, the evaluated measures can be ranked as follows (rail measures only):

1. VA71b + Stiff pads (3.6 €/m/dB(A))
2. Reduced stiffness (18 €/m/dB(A))
3. Saddle profile rail (200 €/m/dB(A))

The new profile rail (VA71b) combined with stiffer rail pads has been evaluated by VAS and is presented in two technical reports [5, 6].

[17] Extra costs for maintenance is not included (e.g. more frequent inspections and rail grinding).

7.7 REFERENCES

[7.1] V. V. Krylov et al, Noise and vibration from high-speed trains. MPG Books, London, 2001

[7.2] Nordborg, Vertical rail vibrations: Noise and structure-borne sound generation. Doctorial Thesis RITA-FKT Report 9535, The Marcus Wallenberg Laboratory for Sound and Vibration Research. Kungliga Tekniska Högskolan, Stockholm, 1995

[7.3] H. Malker, H. Samuelsson, N. Renard, The Tjörnarp site for railway noise studies - characterisation of original status. Sound and vibration measurements during train passages. QCITY, subtask 3.1, Stockholm, 2005

[7.4] N. Renard, J. Nielsen, Definition of a reference system for freight trains to compare against in quantification analyses. QCITY, subtask 3.1.1, Stockholm, 2005

[7.5] H. P. Brantner, Evaluation of effect of wheel & rail quality, geometry and material properties. QCITY, subtask 3.1, 2005

[7.6] H. P. Brantner, Mitigation measures for rail: new rail profile, influence of rail pad stiffness. QCITY, subtask 3.2, 2005

[7.7] DRAFT prEN ISO 3095, Railway applications – Acoustics – Measurement of noise emitted by rail bound vehicles (ISO/DIS 3095:2001). Draft international standard.

8 SYSTEMS WHICH REDUCE RAIL CORRUGATION

8.1 INTRODUCTION

The objective of this technical report is to summarise the state-of-the-art in understanding rail corrugation phenomena. ACL, AMEC, FDP, LUC and STIB, partners of QCITY project, have also been involved for the last three years in the CORRUGATION EC funded research project. This report is based on:

- the main results of CORRUGATION project (project end is planned at 06/2006);
- up-to-date literature survey, such as proceedings of Rail Transit 04 (Advanced Rail Management Corporation).

8.2 CORRUGATION FACTOÏDS IN TRANSIT

Some generally accepted factoïds about corrugation are listed here below:

- about 40% of all transit track is prone to corrugation development;
- noise generated by corrugation can exceed normal rolling noise by 10 dB(A);
- generally, all track geometry (mostly curves but also tangents, braking and acceleration sections) are equally prone to corrugation development;
- there is an exception to the previous factoïd: embedded tracks seem to be significantly less prone to corrugation, as shown here below:

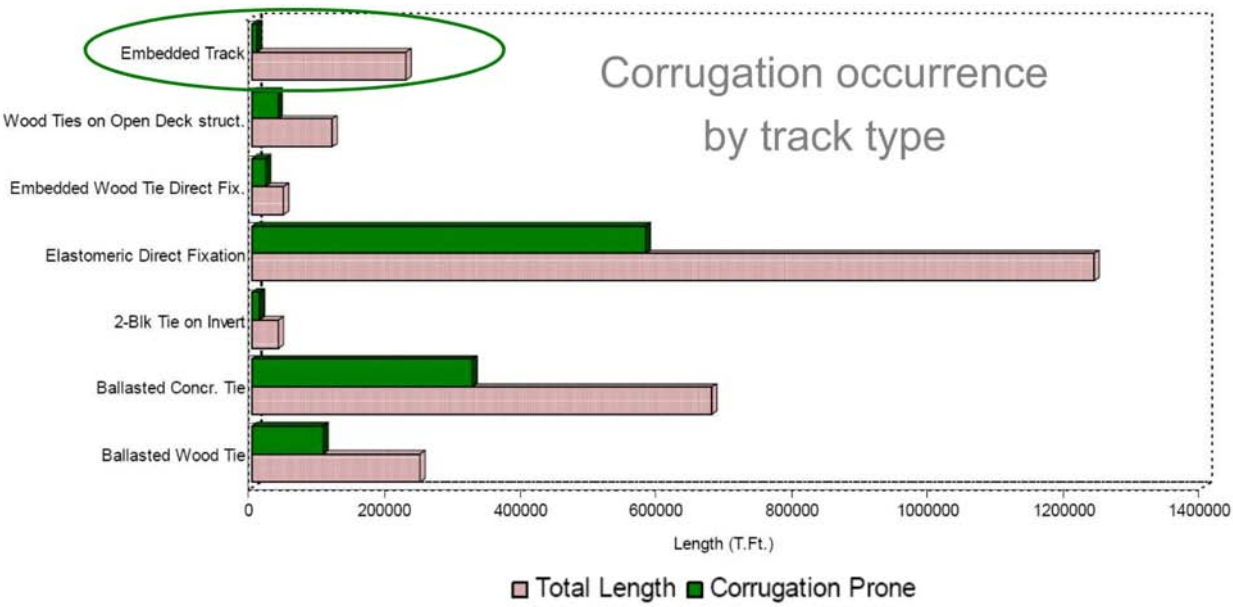


Figure 54

Comparison of different track types vs. corrugation development

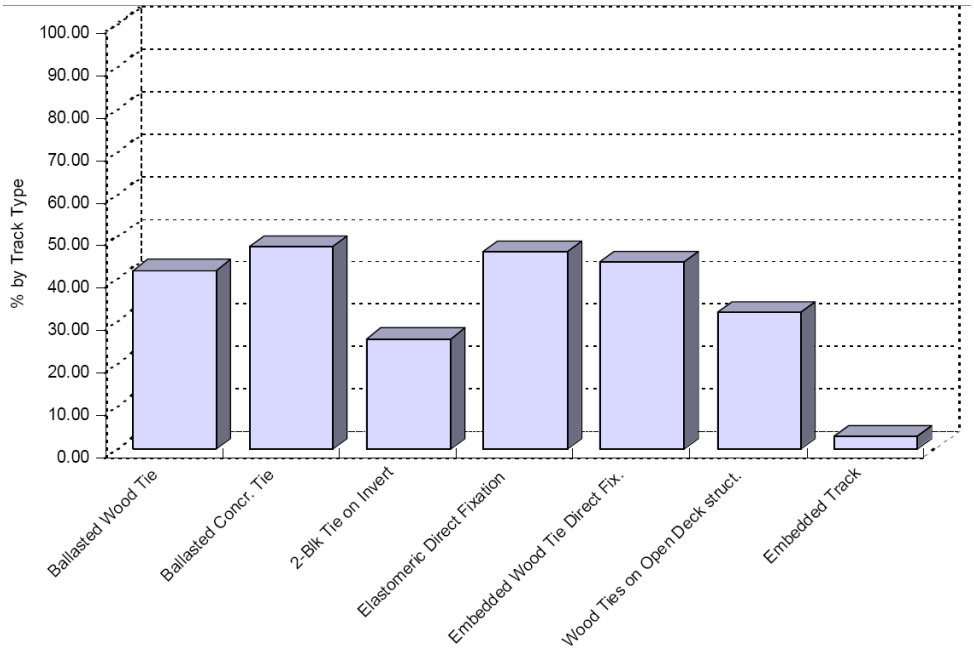


Figure 55

Percent of total track type length prone to corrugation

- corrugation increases with negative superelevation unbalance (car riding toward the low rail in curves – see here below):

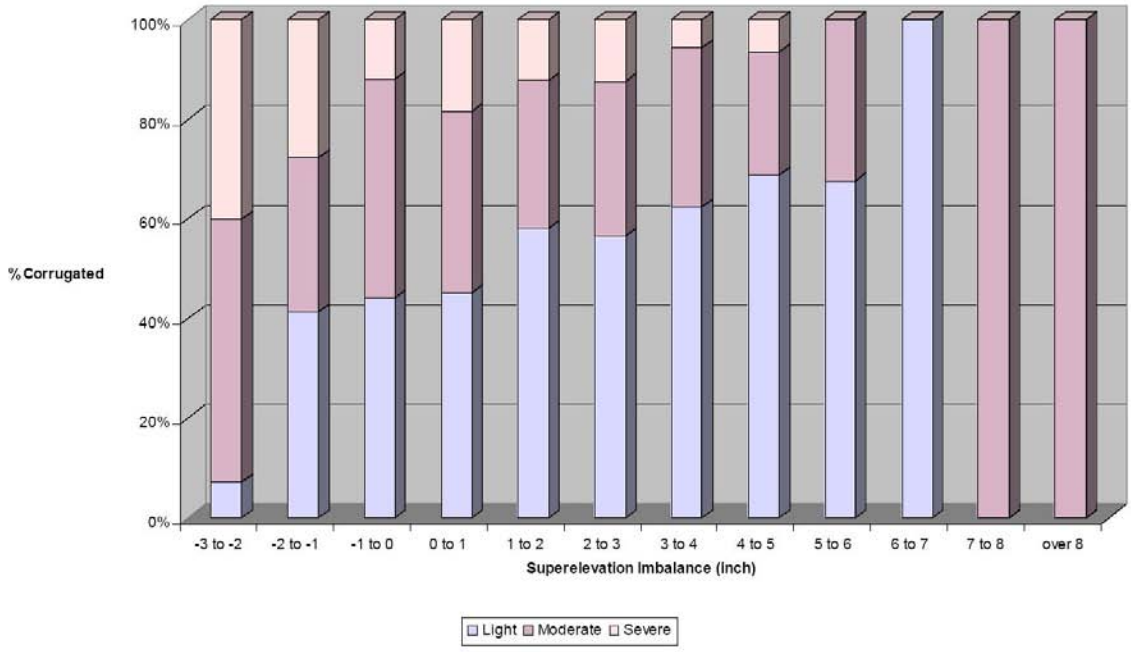


Figure 56

- standard carbon rail is twice as likely to develop corrugation compared to alloyed or heat treated rail, and this corrugation will have a tendency to be more severe.

8.3 RAIL CORRUGATION: GENERAL PRINCIPLES

Based on these factoïds and on preliminary results of CORRUGATION project, it can be stated that the passage of a bogie in small curve leads to important dynamic forces between wheel and rail in all directions:

- tangent;
- radial;
- vertical.

These forces are due to:

- important friction between wheel and rail due to curving (tangential and radial friction forces);
- rail and wheel imperfection (including welds).

The amplitude of these forces is potentially increased by:

- flange friction at high rail;
- misalignment of wheelsets in bogie;
- rail support stiffness variations;
- important slip between wheel and rail due to difference in wheel diameter between wheels of same wheelset;
- existing corrugation;
- poor bogie steering.

These forces are modulated by resonance frequencies of track and wheelset:

- P2 low frequency resonance (non suspended mass bouncing on track spring): wheel/rail bending mode;
- axle torsion mode (especially 2nd order torsion);
- sleeper resonance (e.g. Stedef block (mass) bouncing on rail pad spring);
- axle bending/wheel lateral mode;
- …

This leads to rail corrugation in two different forms:

1. excessive corrugation
2. wear corrugation

8.3.1 Excessive corrugation

The curving forces generate plastic flow of rail, plastic bending of rail or contact fatigue with important corrugation pattern on inner and outer rail.

This type of corrugation is associated with:

- P2 resonance excitation: rather stiff track with variations of rail support stiffness leading to low frequency excitation and a corresponding pronounced anti-resonance as measured on the railhead in vertical direction, which determines the wavelength of the corrugation;
- in combination with:
  - very high axle loads (case of plastic flow);
  - and/or use of low inertia rail (case of plastic bending);
  - and/or important amplification of curving forces due to one of the following factors:
    - misalignment of wheelsets in bogie;
    - important difference in wheel diameters for wheels of the same wheelset;
    - very poor bogie steering.

Excessive corrugation can grow up to 1 mm in 6 weeks of time. Such a corrugation was for instance observed in Skytram Vancouver and STIB metro. Here below is shown the evolution of corrugation as it was measured in STIB's network in 2002.

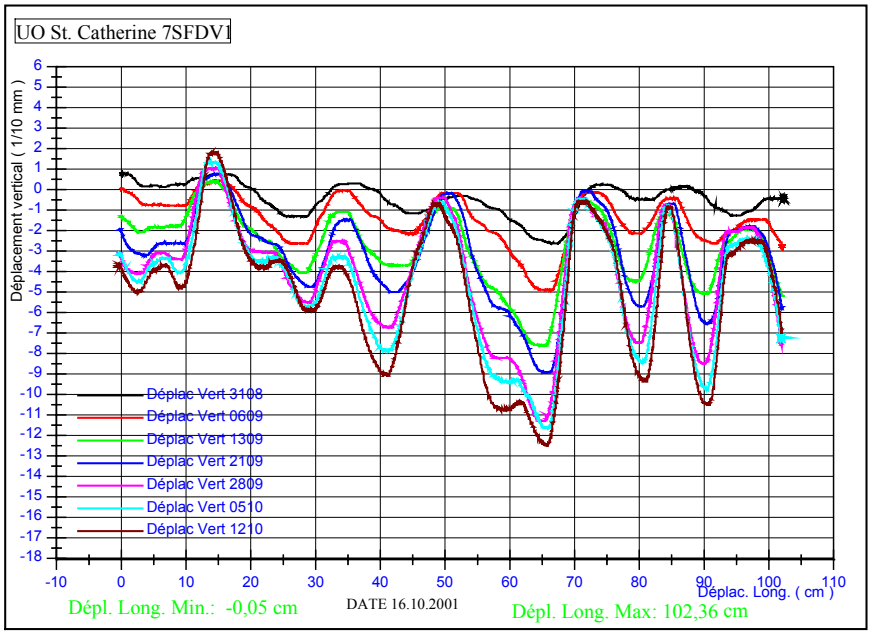


Figure 57 Measured corrugation growth in STIB's network (2002 situation)

Excessive corrugation can only be cured by eliminating the cause, e.g.:

- force amplification factor: STIB reduced tolerance on diameters of wheels on same wheelset to solve the problem, obtaining dramatic improvement;
- increase of the rail inertia when too low.

Excessive corrugation can be reduced:

- by increase of the allowed yield stress (use of head hardened rail, but this solution increases the risk of rolling contact fatigue)
- by increasing the shakedown limit (e.g. rail lubrication, but this has a negative influence on crack growth).

8.3.2 Wear corrugation

The friction forces create a wear pattern mainly on the lower rail (highest axle load) with a wavelength corresponding with one of the track/wheelset resonances:

- stiff track can lead to long wavelengths, corresponding with:
  - P2 resonance, giving longitudinal wear;
  - or with wheelset bending resonance, leading to transversal wear;
- soft track without sleepers (soft direct rail fixation systems, soft continuously supported or embedded track) can lead to short wavelengths, corresponding with torsion axle resonances and giving longitudinal wear;
- soft track with booted sleepers and resilient under sleeper pads (e.g. Stedef track) can lead to short wavelengths, corresponding with sleeper/rail pad resonance and leading to lateral wear;
- track with guard rail can lead to short wavelength corresponding with lateral wheelset resonance and giving lateral wear.

Under normal conditions, corrugation amplitude growth is moderate: amplitude of 0.05 mm in 1 year time, but amplitude grows exponentially!

Rail grinding is a preventive measure to control wear corrugation growth and to remove corrugation.

Wear corrugation on stiff track is more aggressive than wear corrugation on soft track without sleepers.

Wear corrugation on soft track corresponding with axle torsion resonances can be reduced by using high rail gauge face lubrication. Also the use of wheelset torsion damped vibration absorbers (patented by Bombardier Corporation) is promising and has to be demonstrated.

Wear corrugation on booted track can be very aggressive, especially in small curves and with high axle loads (e.g. RER track in Paris). It can be reduced by using high damped resilient rail pads and/or by using high rail gauge face lubrication.

The following measures reduce wear rate of all types of wear corrugation:

- use of heat hardened rail: HHR instead of e.g. 900A grade rail will decrease the wear rate with a factor of at least two
- improve bogie steering: using yaw dampers/springs with reduced stiffness, use of gauge narrowing in curves
- use of resilient wheels (for new wheels only)
- use of different vehicles passing at different speeds
- increase of curve radius (only possible for new projects).

The following measures have a high potential for reducing wear corrugation rate:

- use of railhead friction modifiers: the friction modifier reduces the friction forces (up to factor 2) and reduces stick-slip but does not change the vertical pressure in the wheel/rail contact. The effect of the presence of steel particles from wheel and rail and of friction modifier particles in the third body layer on the rail wear rate is a topic of further research
- use of embedded rail systems with high damping and low impedance (flat rail impedance curve).

Results obtained with the most promising mitigation measures are detailed here below.

8.4 PROMISING MITIGATION MEASURES

Some of the most promising mitigation measures are detailed here after, together with preliminary results illustrating their potential for corrugation growth reduction.

8.4.1 Continuously supported embedded tracks

In the CORRUGATION project, a continuously supported embedded track system has been tested, showing interesting results.

8.4.2 Description of the system (=standard ML-embedded system)

The system is schematically shown here below:

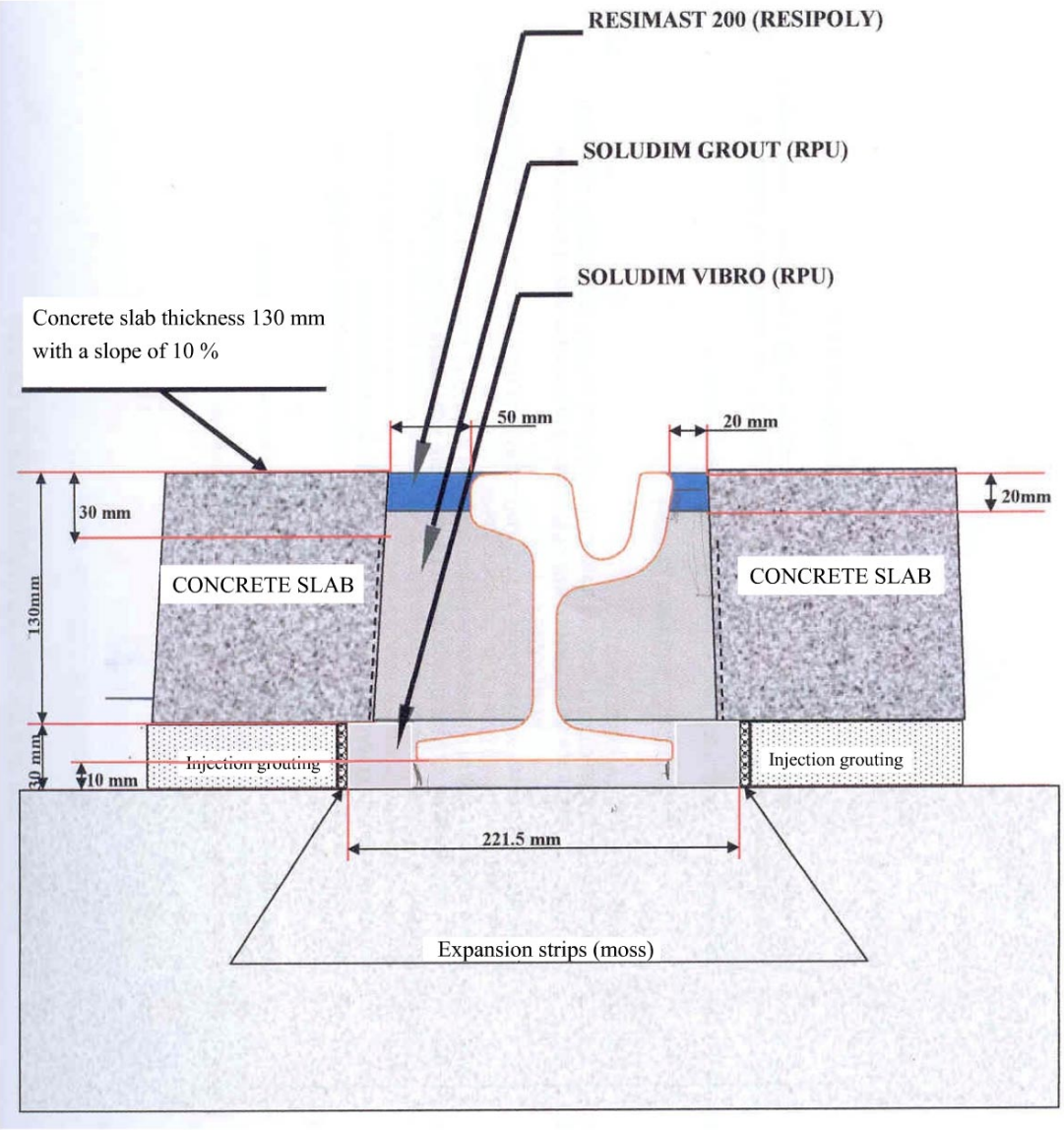


Figure 58

It is further called ML (multi layer) embedded system. The embedded rail is put inside a gutter in the surrounding concrete slab. Rail fastening mainly consists of three components:

- at the rail foot, a rather soft polymer, enabling downward and upward movements of the rail thus providing an as low as possible vertical stiffness to the rail;
- around the rail web, a stiffer polymer, in order to assure sufficient lateral and rotational stiffnesses to the rail;
- at the height of the rail head, a polymeric joint, perfectly adherent to the rail and the surrounding concrete.

The rail surfaces are coated with a primer, in order to assure an excellent adherence between the rail and surrounding polymers. The interior faces of the gutter in the concrete are partially coated with a grease or another lubricant in order to assure a free vertical motion of the rail and surrounding stiffer polymer. These faces are also manufactured with vertical ribs in order to prevent longitudinal displacement of the rail.

In order to prevent also excessive upward displacement of the rail and its embedding, the internal faces of the gutter are slightly inclined toward the rail head.

8.4.3 First installation

The ML-system was first tested on RATP tramway line T1 in Bobigny. The aim of this installation is to test the feasibility of the proposed design. Available space between rail foot and subconcrete is not constant. It is thus not possible to have a uniform stiffness under the rail.

A section of T1 line was renewed using this system, while another section, equipped with a conventional embedded system, is located just beside. Both tracks are continuously supported.

In the conventional embedded fixation system, only one product (polyurethane) is poured around and under the rail, while there are three different products in the new system.

Measurements after installation

The following measurements were carried out:

- vibration measurements along the tracks during tram passages;
- rail displacements in five sections;
- dynamic impedance of the rail in 18 sections.

leading to the following observations:

- the ML system had similar static displacements and generated vibration levels than classical embedded track, i.e. it was a rather stiff track. In order to have a more resilient track, it is possible to divide its static stiffness with at least a factor 5 by selecting a softer lower polymer layer.

- a big difference has been observed between direct transfer functions measured on top of the rail for the ML system and the conventional system:
  - as shown here below, ML track gives rather flat direct transfer functions, without any pronounced resonance or anti-resonance (up to 800 Hz). It is a highly damped fixation system.

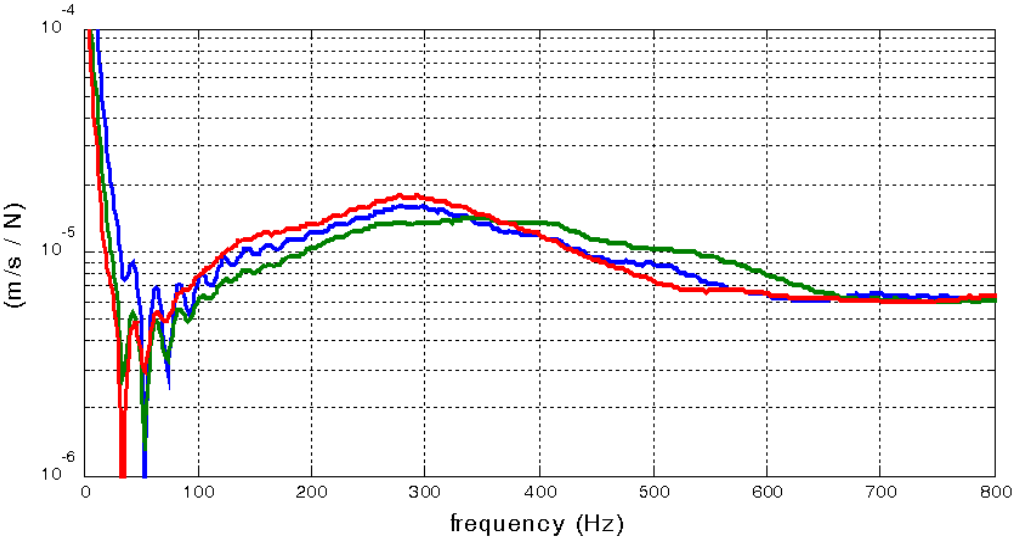


Figure 59

Measured direct transfer function on top of the rail for ML embedded track

  - classical embedded track gives transfer functions with an important peak around 250-300 Hz and an anti-resonance around 350 Hz. It is less damped than the ML system (factor of 2 to 3 between both). It is more subject to creation of corrugation with wavelengths of 3 to 4 cm.

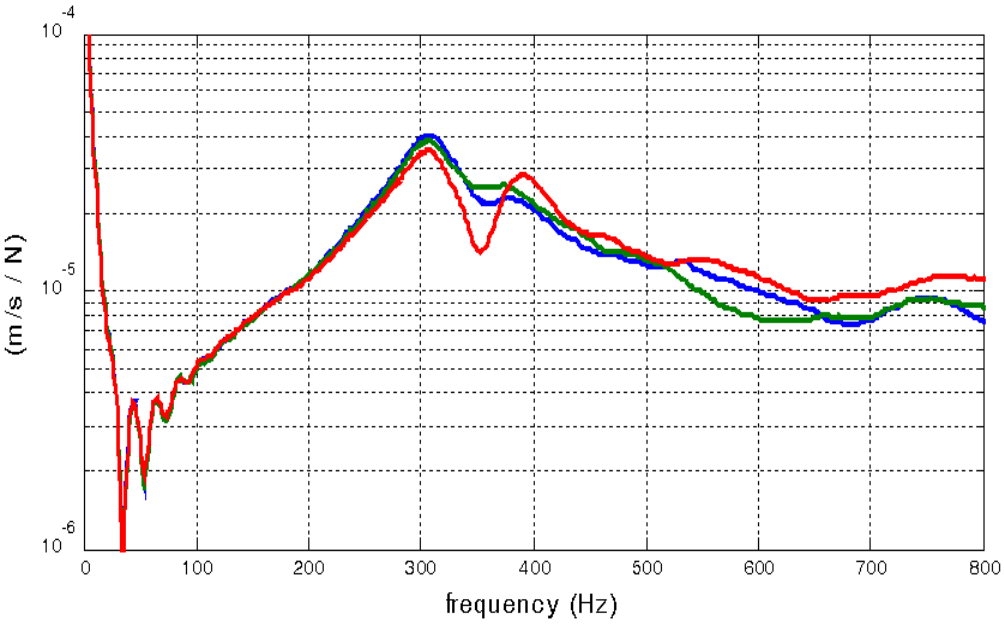


Figure 60

Measured direct transfer function on top of the rail for classical embedded track

Corrugation follow-up 30 months after installation

Though the installed ML embedded system was rather stiff, it can be seen, 30 months after installation, that visible corrugation appeared on adjacent classical embedded track and not on the ML embedded system:

Figure 61a Classical embedded track,
30 months after installation

Figure 61b ML track,
30 months after installation

Figure 62 Yellow line = transition between ML and classical embedded track (30 months after installation)

Corrugation clearly starts at the junction between ML track and classical embedded track. This is probably due to the fact that the classical system is less damped and has more pronounced resonances and anti-resonances.

8.4.4 Resilient ML system

Based on results obtained with the first design (see §8.4.1 and 8.4.2 here above), a resilient version has been designed in order to achieve a higher rail deflection under wheel load.

Compared with the design implemented on Bobigny tramway, the main modifications are:

- a different resilient material is used, composed of closed cells to work in a confined space;
- the height of the first resilient layer is carefully controlled to maintain constant resilient property.

The new design is shown here below:

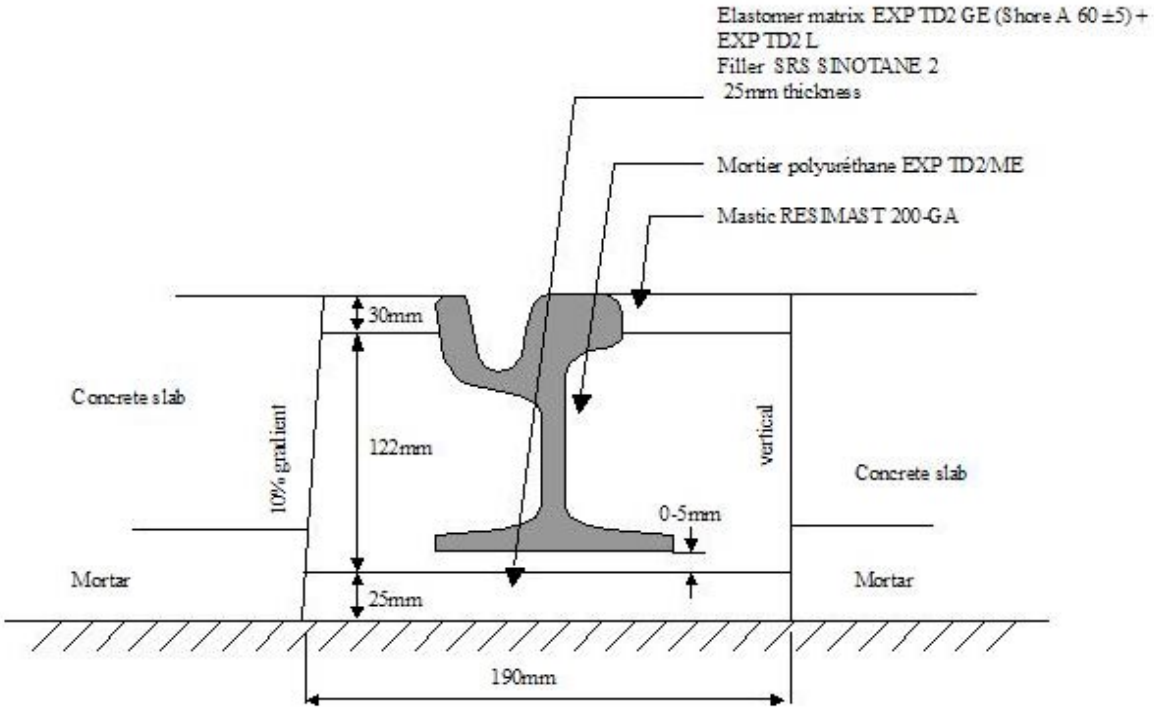


Figure 63

A cross-section in a prototype of this fixation system is shown here below:

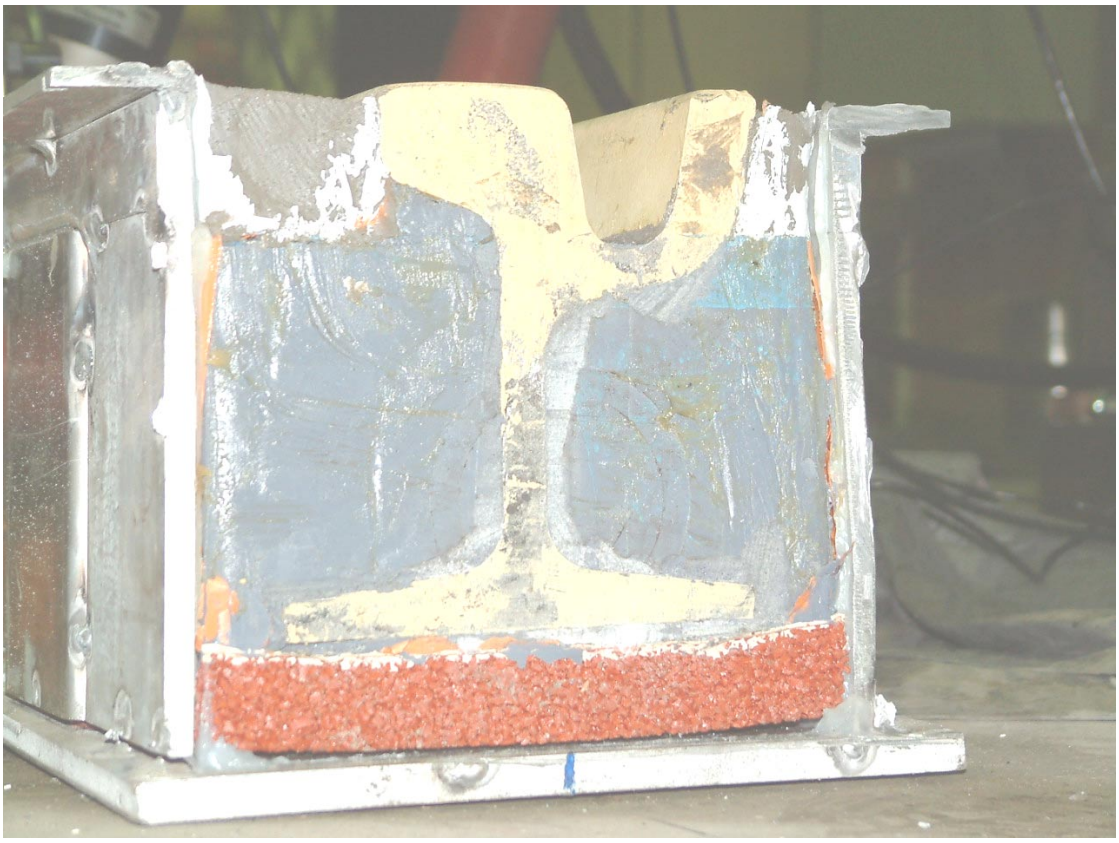


Figure 64

The rail can be embedded on site or can be installed in prefabricated beams in the factory (with an option to use prefabricated elastomer strips and jackets).

The working principle of this system is to avoid all track resonances in the frequency range, which can give rise to corrugation wavelengths between 2 cm and 20 cm.

This system is going to be installed in a test track in January 2006. Corrugation follow-up will enable to validate this solution.

8.5 REDUCED GEOMETRICAL TOLERANCES ON WHEEL DIAMETERS

Reducing the geometrical tolerances on the wheel diameters gave dramatic improvement in terms of corrugation at STIB. New tolerance values are given here below:

M.I.V.B. METRO DELTA	CONVENTIES CONV 00-N-01.	
Metro 01/02/2002	IMMOBILISATIECRITERIA WIELEN	Geldig tot 31/12/2003

DIAMETERVERSCHIL	
2→ 2,5 MM DIAMETERVERSCHIL OP DEZELFDE AS	AFDRAAIEN BINNEN 7 KALENDERDAGEN
2,5→ 3 MM DIAMETERVERSCHIL OP ASSEN VAN HETZELFDE DRAAISTEL	AFDRAAIEN BINNEN 7 KALENDERDAGEN
> 3MM DIAMETERVERSCHIL OP ASSEN VAN HETZELFDE DRAAISTEL	AFDRAAIEN BINNEN 4 KALENDERDAGEN
MAXIMUM DIAMETERVERSCHIL TUSSEN DRAAISTELLEN VAN DEZELFDE KAST:	
M1 → M2 14 MM	
M3 → M5 14 MM	
MAXIMUM DIAMETERVERSCHIL TUSSEN KASTEN	
M1 → M5 60 MM	
MINIMUM WIELDIAMETER	
761 =0 /-1 MM	OPMAAK AVERIJBLAD GEBLOKKEERD BINNEN DE 7 KALENDERDAGEN
QR	
8 > QR > 6,5 MM	VERWITTIGEN DRAAIBANK
< 6,5 MM	GEBLOKKEERD
HOOGTE WIELKRAAG	
35→ 36 MM	AFDRAAIEN BINNEN DE 7 KALENDERDAGEN
> 36 MM	GEBLOKKEERD
VALSE WIELKRAAG	
MEETMETHODE NOG VAST TE LEGGEN	
DIKTE WIELKRAAG	
< 22 MM	AFDRAAIEN BINNEN DE 7 KALENDERDAGEN
OVALISATIE	
≥ 120 dB	MELDING DOOR S & P
≥ 130 Db	GEBLOKKEERD
PLATTE KANTEN	
PLATTE KANT LENGTE OP OMTREK < 20 MM	SLIJPEN BINNEN DE 4 KALENDERDAGEN
PLATTE KANT LENGTE OP OMTREK > 20 MM	GEBLOKKEERD VOOR DRAAIEN
TRILLINGEN/ PLATTE KANTEN /KLACHTEN BESTUURDERS	
MELDING OF KLACHT	AAN ELKE MELDING OF KLACHT WORDT GEVOLG GEGEVEN

Opgesteld door :	Gevalideerd door :	Goedgekeurd door :
KZM Delta	De Brauwer Paul Petitjean Bernard De Ridder Francois Mosselmans Pierre	Goeman Marcel Vercauteren Joël

24-6-03K:\Verhaegen\Word\WIELEN\wielparameters_nota_NL.doc 1.

Figure 65

In the CORRUGATION project, ULB tried to validate/optimise the geometrical tolerance values in order to reduce corrugation growth.

8.5.1 Theoretical considerations

The wear of bogie wheels is limited by geometrical tolerances defined by the maintenance department of metro networks. As long as the design of the profile does not reach certain limits, the security of the passengers remains insured. However, these restrictions do not give any information on how the change in wheel profile influences rail wear. For instance, can a worn wheel profile give birth to rail corrugation faster than a new (smooth) one? Will the wavelength of corrugation still be unchanged? Answers to these questions were given by a multi-body dynamics model of the vehicle-track system, used to perform a parametric study of the profile influence.

Figure 66a (resp. 66b) shows a new (resp. old) wheel profile from STIB network. On these figures, all lines linking the wheel and the rail depict possible contact points for various relative displacements between the wheel and the rail. These lines clearly show that in some conditions (at least) two contact points can exist between the wheel and the rail. As the wheel profile becomes older, wheel and rail profiles become more and more conformal, showing a smaller gap between the two contact points.

The test case considered is a track section from the STIB metro network, between stations Delta and Beaulieu.

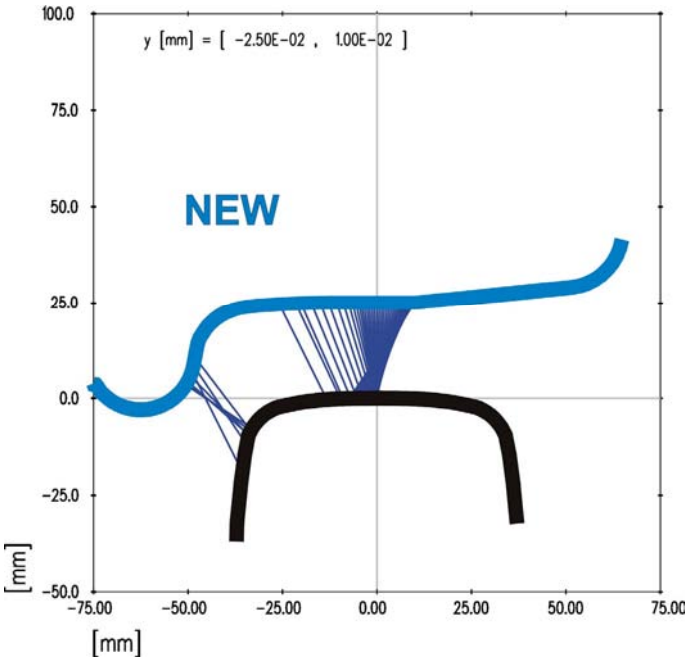


Figure 66a New wheel profile

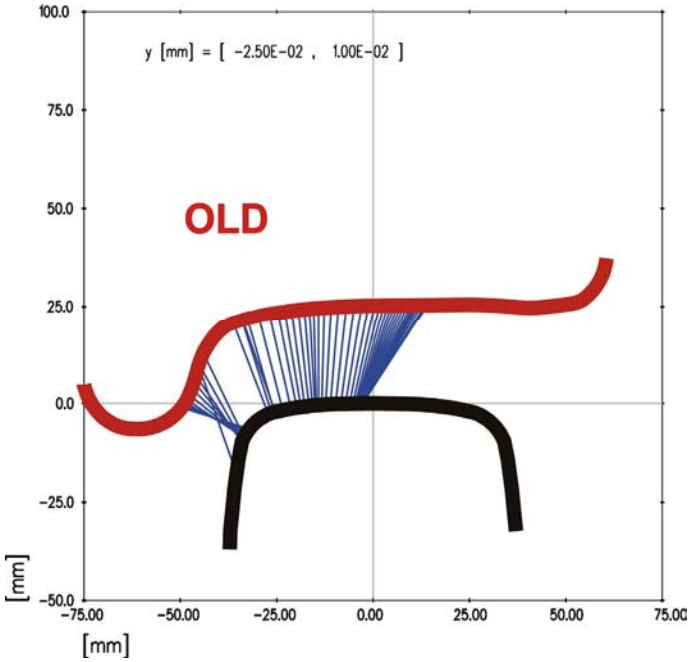


Figure 66b Old wheel profile

8.5.2 Parametric study

ULB carried out calculations under different assumptions, i.e.:

- different diameters for both wheels of one wheelset;
- different diameters for wheels of both wheelsets of one bogie;
- different diameters for wheels of two bogies of one vehicle;

leading to the following conclusions and recommendations:

- there is no noticeable difference between new and worn profiles, in terms of corrugation growth;
- stricter tolerance for the wheel radius could enable to save energy and decrease the global dissipation of energy in the contact patch;
- apply stricter tolerances for diameter differences of wheels on the same wheel set (especially front wheel set) was found very efficient in reducing corrugation growth.

8.6 RESILIENT DISCRETE RAIL FIXATION SYSTEMS FOR METRO

8.6.1 General principle: APT-BF fastener

Numerical simulations carried out within the CORRUGATION project have shown that it was necessary to develop rail fastening systems with a vertical dynamic stiffness lower than 10 kN/mm in order to avoid corrugation (to avoid track resonances corresponding with the 2 to 20 cm corrugation wavelength range). The development is based on an existing pre-loaded rail fastener:

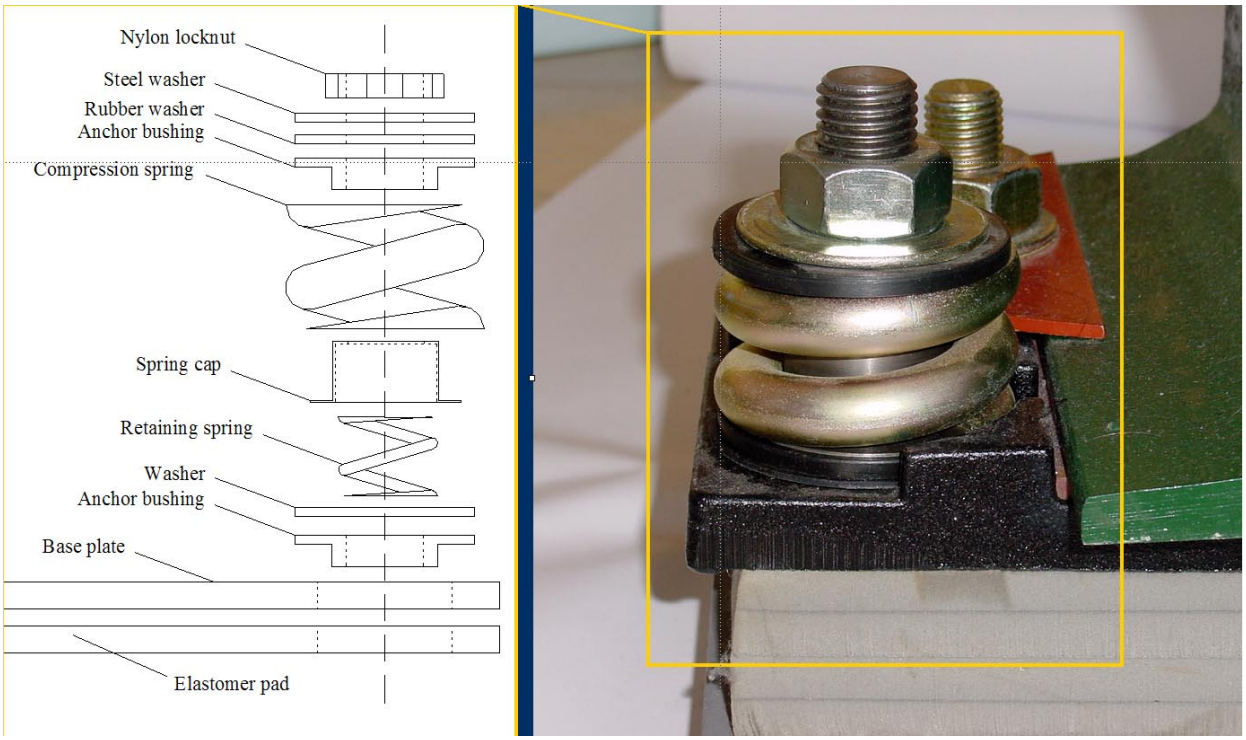


Figure 67

The existing standard pre-loaded rail fastener can be further improved by eliminating the stiffness of the compression springs from the global fastener stiffness during vehicle passage:

- during wheel passage, a residual load from the compression springs in parallel with the wheel load remains for standard pre-loaded fasteners
- since the springs remain in solid contact with both the base plate and the nut on the anchor bolt, they add to the dynamic stiffness of the fastener assembly
- de-coupling & complete unloading of the pre-loading springs is possible by:
  - including a thin spring with a low stiffness, but a higher free length, inserted inside the compression spring
  - the heavier compression spring generates the pre-load and uncouples during wheel passage
  - the smaller spring holds the assembly together during wheel passage; this mechanism further lowers the dynamic stiffness of the whole system.

8.6.2 Influence on the corrugation growth

Test tracks are equipped with this system in Stockholm (SLI), Brussels (STIB) and Milano (MM). Corrugation follow-up is still in progress. Preliminary results, six months after installation, show a favourable evolution, as detailed here below.

STIB test track

Rail roughness measured 8 months after grinding on the old track type is given in figure 4.11a, while figure 4.11b shows rail roughness measured at the same location 6 months after grinding on the new track with very resilient discrete rail fixations. Corrugation 6 months after grinding is significantly lower for modified track than 8 months after grinding for the old track. Of course, this trend will have to be confirmed with longer term follow-up.

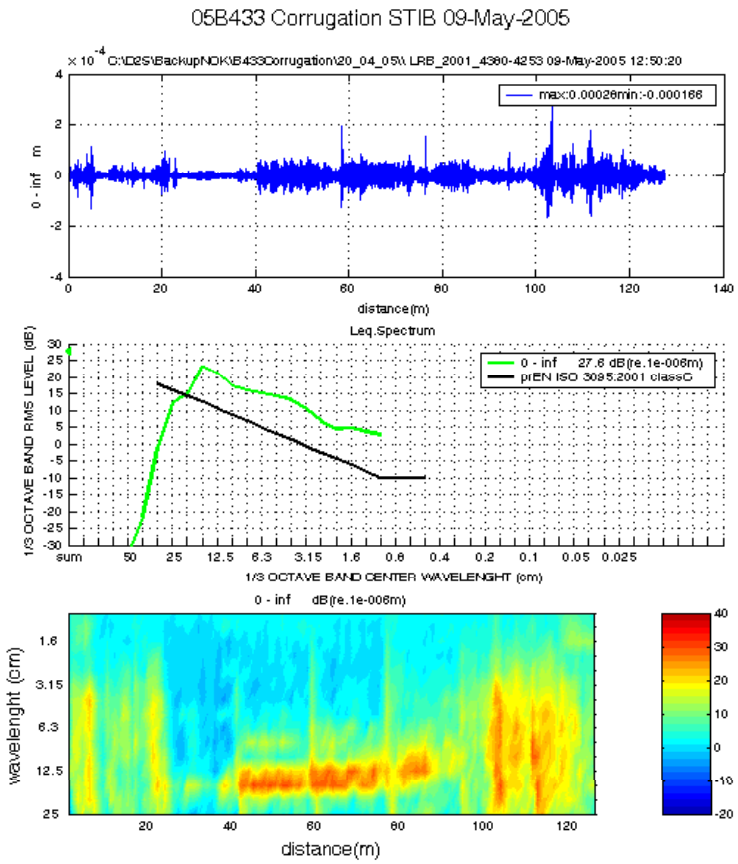


Figure 68a Rail roughness measured 8 months after grinding (old track)

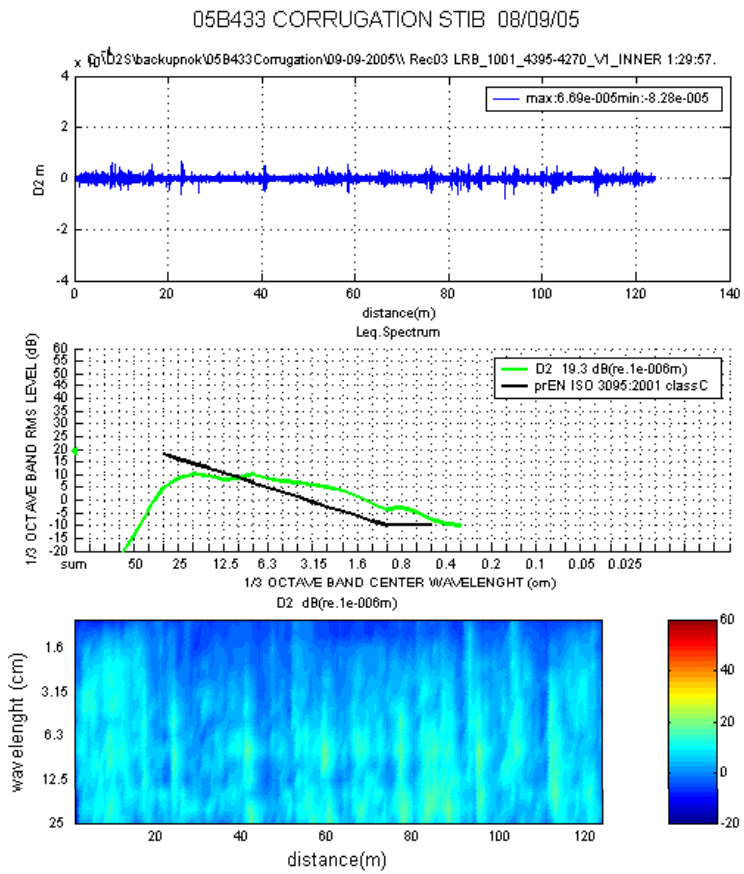


Figure 68b Rail roughness measured 6 months after grinding (modified track)

SLI test track

Rail roughness measured 6 months after grinding on the very resilient track is given in figure 4.12 for the middle of the rail (figure 4.12a) and outer side of the rail (figure 4.12b).

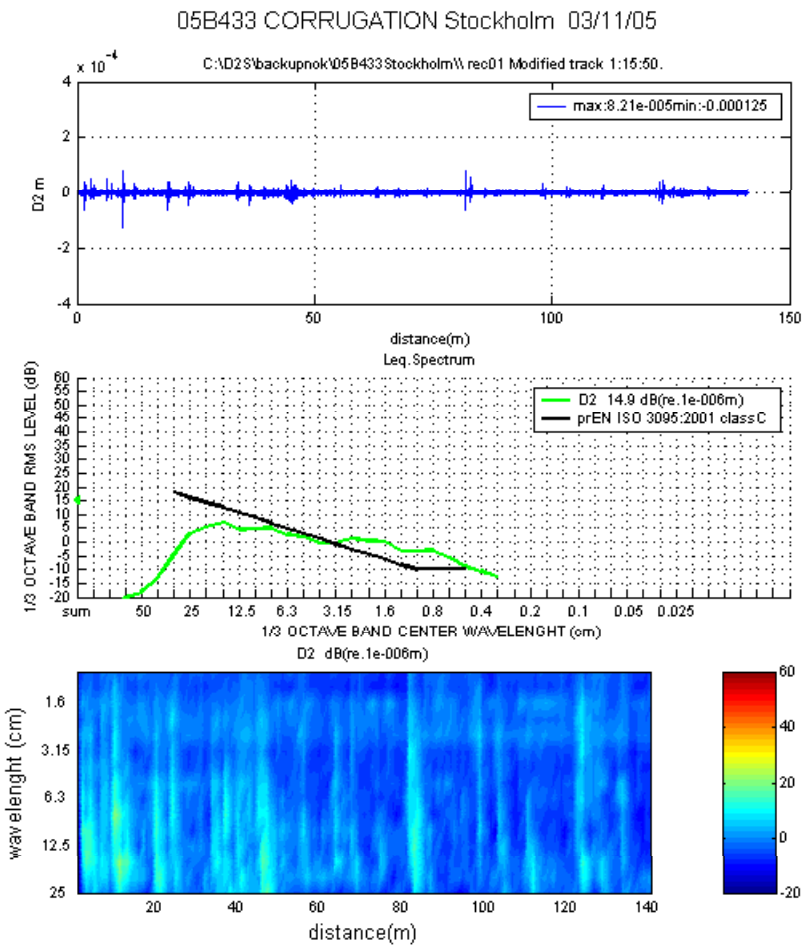


Figure 69a Rail roughness measured 6 months after grinding (middle of the rail - very resilient track)

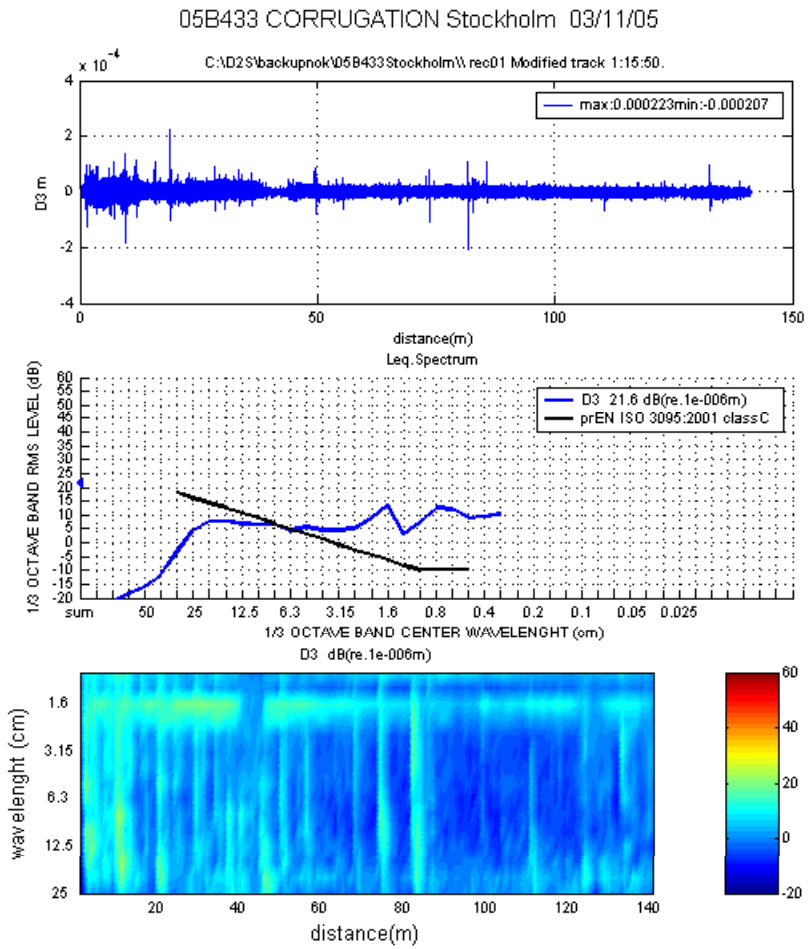


Figure 69b Rail roughness measured 6 months after grinding (outer side of the rail - very resilient track)

A detailed analysis of the measurement results on SLI test track further shows that:

- on the very resilient track (which was recently grinded (April 2005), the sensor D3 (outer side of the rail) measures clearly the wavelength of 1.6 cm (grinding profile);
- rail joints are clearly visible in the RMS global levels and colour plots;
- no corrugation can be found on the very resilient track:

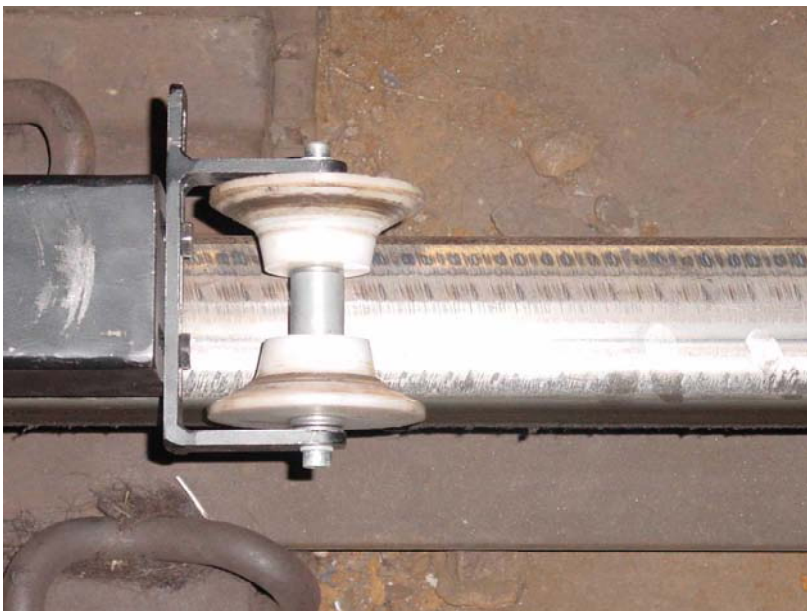


Figure 70 Modified track

- 15 m behind the very resilient track in an inner curve rail with standard track, corrugation profiles were visually identifiable; two wave lengths are measured: 6 cm and 16 cm:

Figure 71 Standard track type, 15 m behind the very resilient zone

8.7 SADDLE PROFILE RAILS

8.7.1 General principle

The Saddle profile rail is another solution designed by Acoustic Control in the CORRUGATION project in order to reduce corrugation growth. The general concepts are highlighted here below:

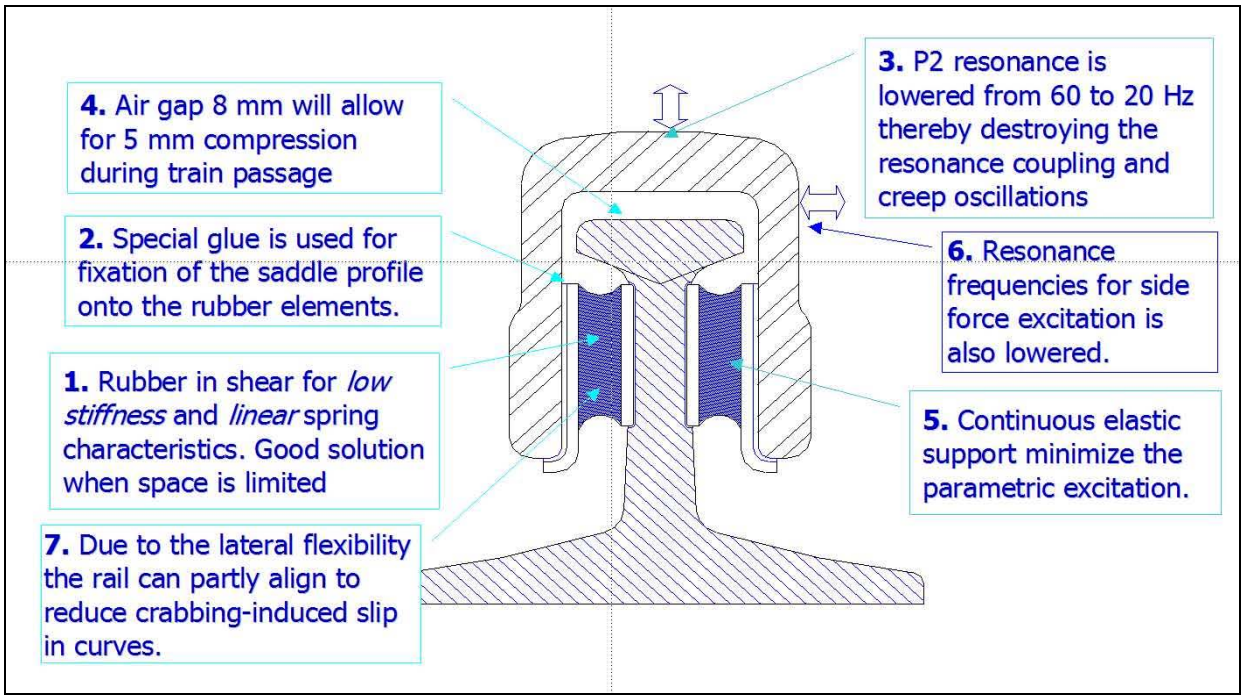


Figure 72 Saddle profile rail: general concepts

It is installed in a depot track (Högdalen site) in SLI (Stockholm) network. The main corrugation mechanism at the Högdalen site was found to be the longitudinal creep at constant speed or acceleration, modulated by the P2-mode which couples to a torsional mode of the wheel-set all at about 60 Hz. By a substantial increase of the vertical compliance, the P2 mode is moved to a much lower frequency, which destroys the modal (vertical/torsional) coupling controlling the creep modulation.

8.7.2 Corrugation follow-up

Seven months after installation, no corrugation has appeared on the saddle profile rail. But no substantial corrugation appeared on the adjacent standard track, so it is too early to draw final conclusions, corrugation follow-up will be continued in the coming months.

Figure 73 Saddle profile rail 7 months after grinding

Figure 74 Standard SJ43 rail 7 months after grinding

8.8 CONCLUSIONS

All the mitigation measures proposed here above can be applied for new systems in order to avoid excessive corrugation.

For existing systems, most of them can no more be applied. The unique solution is then rail grinding, which is definitely very effective in reducing noise (up to 10 dB(A)) and controlling corrugation. Cost is estimated at 4000 €/km track. But rail grinding treats corrugation symptoms, it does not eliminate its cause.

It has to be noted that in case all resonance excitation (track and wheelset) is avoided during curve negotiation, rail wear will still occur due to the friction forces. This rail wear (quasi random patterns) can be reduced using harder rail contact areas. The use of friction modifiers is also promising.

Special attention has to be paid to harder rail and lubrication, which can lead to increase of rolling contact fatigue.

Our advice to network infra managers installing new track or renewing existing track is to avoid low damped track resonances, which correspond with corrugation wavelengths between 2 cm and 20 cm in curves. This can be done by using high damped embedded tracks or by using very resilient track in curves (vertical dynamic stiffness typically lower than 10 kN/mm/m rail).

Our advice to rolling stock maintenance managers is to impose, control and enforce small diameter differences on the wheels of the same wheelset (typically lower than 1 mm).

9 SQUEAL NOISE MITIGATION IN URBAN RAIL SURFACE TRANSPORT BY RAIL TREATMENT

9.1 INTRODUCTION

This report summarises the results of the research carried out under the EC contract BRPR-CT97-0477, "*Squeal noise reduction in urban transport by rail treatment*" and the results of recent findings about mitigation methods for squeal noise.

A mathematical model for squeal noise calculations is presented. In the scope of the QCITY project, this model is integrated within the SYSTUS finite element program and applied to some case studies where anti-squeal measures have been applied. The model and associated software have been validated in this way. Squeal mitigation measures have been identified.

Almost any tram system has curved track sections in which squeal is generated. Squeal is normally most predominant in tight curves. Curved track sections where squeal is a problem normally cover a total track length of a couple of hundred metres in a normal tram system of many kilometres of track. The most cost-effective way of solving squeal noise problems is thus to treat only the track section causing the squeal rather than to treat all wheels on the entire fleet of vehicles. The aim is therefore to find measures applied onto the rail, which can reduce the squeal noise generation.

What do European experts and programs say about SQUEAL noise?

1. Working group Railway Noise, Michael Dittrich – TNO, Workshop Railway Noise Abatement in Europe, October 29, 2003

Recent development:

- new lines, high speed lines, freight at night, more light rail systems;
- more noise sensitive areas, more people affected;
- local problems with curve squeal, shunting, bridges.

2. European Rail Research Advisory Council (ERRAC), Pierre-Etienne Gautier – SNCF, Noise Research Strategies for a Quieter Europe, October 19, 2004

- other/specific sources need deeper knowledge;
  curve squeal (although investigated for trams, and by UIC).

3. Harmonoise IST-2000-28419, Work Package 1.2 Rail Sources, report HAR12TR-020118-SNCF10, revision 04 dd. 05-08-02, p. 28/78

"Curve squeal noise:

Dutch research [De Jong, Opmeer & Miedema, 1994] has pointed out that 7% of the group of severely annoyed people, are annoyed from noise in curves. However, curve squeal is not included in the most noise calculation schemes. However, it is included in the German scheme Schall03. In this guideline it is taken

into account by a radius dependent correction value: 3 dB(A) for radii between 300 and 500 m and 8 dB(A) for radii smaller than 300 m."

4. Keele University (UK), Department of Mathematics, Maria Heckl

"Curve squeal can be eliminated if the unstable amplitude growth can be prevented. This has been achieved in practice by increasing the wheel damping (equivalent to increasing the modal loss factors in our model). It has also been achieved by applying lubrication to change the properties of the wheel/rail interface in such a way that the slope of the increasing section of the friction characteristics is lessened (equivalent to changing the friction characteristic in our model in such a way that the gradient of the slip section is reduced). Curve squeal can be reduced in intensity by reducing the train speed and by increasing the curve radius; both measures reduce the crabbing speed which, according to our model, determines the velocity maximum of the limit cycle."

9.2 TRAM SQUEAL NOISE MODEL INCLUDING A SUITABLE MODEL FOR THE ROLL-SLIP EXCITATION OF THE WHEEL AND RAIL

9.2.1 Squeal generation

There are two possible mechanisms for wheel squeal:

- longitudinal stick-slip due to the different translation velocities between two wheels on a rigid axle;
- crabbing of the wheel across the top of the rail: lateral stick-slip (rolling slip).

Due to the finite length of the 2-axles truck and the radius of curvature of the rail, both axles cannot lie upon curve radius, as shown on figure 75. This figure shows the geometrical relation between the creep angle ξ, the wheel base l and the curve radius R. Under actual conditions, however, the leading axle of the truck rides toward the outside of the curve, while the trailing axle travels between the two rails, i.e. a reduction of the creep angle at the trailing axle, but an increase of the creep angle at the leading axle, as shown on figure 76.

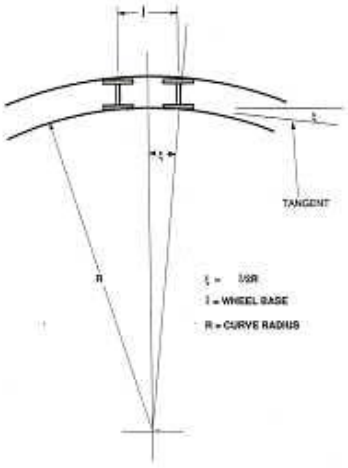


Figure 75

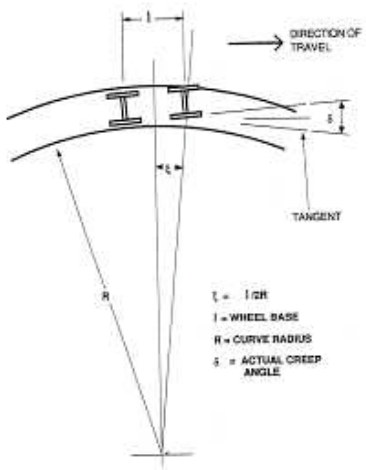


Figure 76
Truck crabbing under actual conditions

The SQUEAL project has demonstrated that in metro systems with solid wheels both mechanisms can generate squeal whilst in tram systems with resilient wheels, the lateral roll-slip mechanism is the most likely mechanism for squeal. Since we are considering tram systems in this section, we will further concentrate on mitigation methods, which reduce lateral roll-slip.

The radiated sound power by the wheel during squeal can be calculated as:

$$L_W = 10\log\frac{\phi\rho c A u_0^2}{10^{-12}} \qquad 1$$

where ϕ is the radiation efficiency
ρc is the air impedance
A is the wheel radiating area
u_0 is the wheel vibration velocity

As a first approximation, and for a track in open air without major screen effect, the squeal sound pressure level can be calculated by:

- SPL = L_w –10 log ($2\pi d^2$)for reflecting ground
- SPL = L_w –10 log ($4\pi d^2$) for absorbing ground

9.2.2 Lumped parameter model with non-linear friction element

Justification of the model

Wheel squeal originates from frictional instability in curves between the wheel and rail. Stick-slip oscillations (more accurately referred to as roll-slip) are amplified by the wheel web. The accepted model for tram systems with resilient wheels involves Top Of Rail (TOR) frictional instability under lateral creep conditions leading to excitation of out of plane wheel bending oscillations. These are radiated and heard as squeal. The starting point for squeal is lateral creep forces that occur as a bogie goes through a curve and the wheel / rail contact patch becomes saturated with slip (creep saturation). A critical component in all the modelling work is the requirement that beyond the point of creep saturation, further increases in creep levels lead to lower coefficient of friction. This is known as negative friction, referring to the slope of the friction creep curve at saturated creep conditions. In more general tribological terms, this would be equated to changes in sliding velocity. This leads to roll-slip oscillations between the wheel and the rail which are amplified in the wheel.

The squeal noise generation is thus a non-linear process. It is therefore necessary to establish a mathematical model that will incorporate the non-linearity of the process.

Short description of the model

The lumped parameter model includes two damped single-degree-of-freedom systems (representing wheel and rail) on each side of a non-linear friction element. The system is driven to self-sustained vibrations by pulling the wheel end of the system with a constant velocity similar to the constant crabbing velocity occurring when a two-axle bogie with fixed axles is passing through a curved track. The model, shown in summarised form in figure 2.2.1, includes one mode of the wheel (shown to the left of the friction element) and one mode of the rail (shown to the right of the friction element).

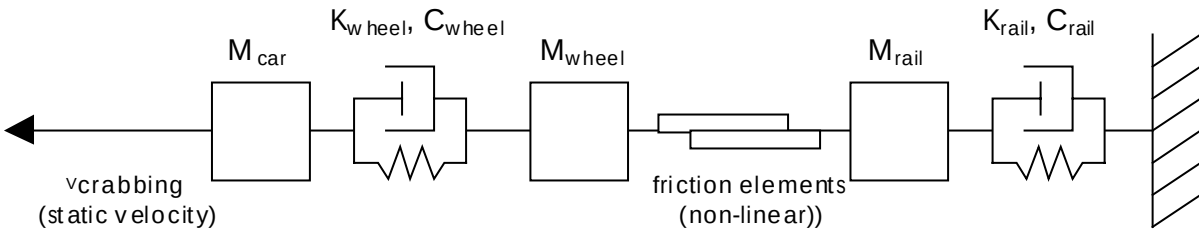


Figure 77

Conceptual sketch of the lumped parameter model for prediction of wheel/rail squeal noise

A typical non-linear friction characteristics of the contact between the wheel and the rail is stored as a (numerical) function in a finite element model and is based on measurements of actual wheel/rail friction, see figure 77.

A standard wheel/rail configuration with data according to Figure 78 leads to a predicted vibration velocity in the wheel according to figure 79.

As can be seen in figure 75, the system is squealing nicely at a frequency determined by the resonance (mode) in the wheel. This model was used to study the effect of parameter modifications.

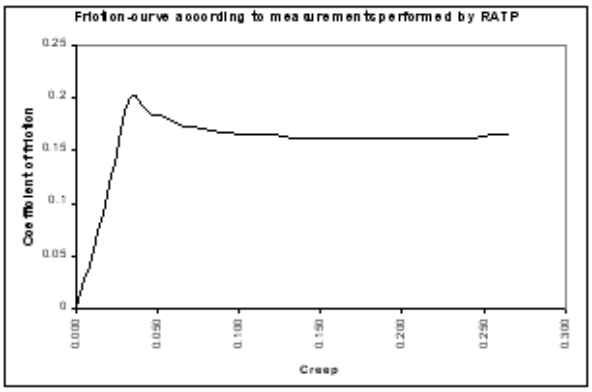


Figure 76 Friction curve measured by RATP and used in numerical form in the computer model

Parameter	Value
Loss factor, wheel	0.004
Loss factor, rail	0.035
Resonance frequency, wheel	670 Hz
Resonance frequency, rail	400 Hz
Dynamic mass, wheel	80 kg
Dynamic mass, rail	125 kg
Vehicle velocity	7 m/s
Axle distance	2 m
Curve radius	100 m

Figure 78 Data for the standard parameters case.

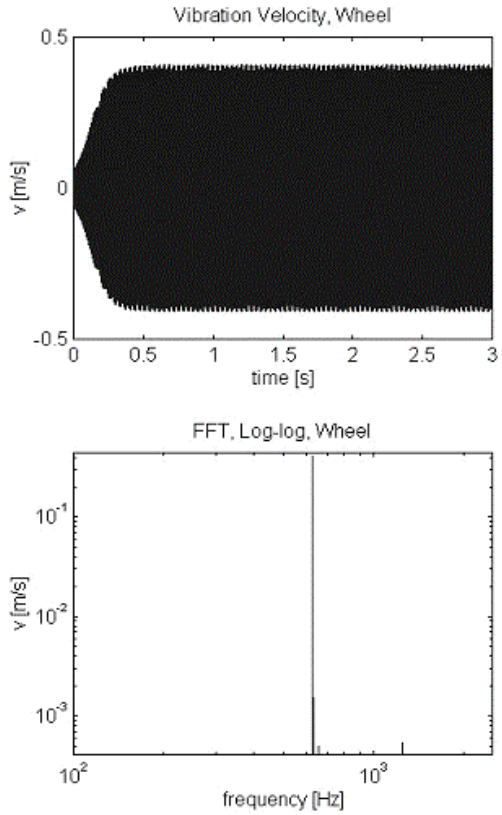


Figure 79 Wheel disc vibration velocity during squeal for a standard wheel/rail configuration shown as time history and frequency spectrum

9.2.3 Parametric study

A special non-linear friction element has been introduced within the SYSTUS finite element software in order to be able to carry out a parametric study.

Hereafter, the main results of this parametric study are given.

Base case

The base case is based on actual STIB data. A vehicle is considered with a 2 m axle spacing, velocity of 7 m/s and 100 m radius curve. The wheel squeal mode is represented by following equivalent one degree of freedom parameters: m = 24 kg; k = $2.85*10^8$N/m and c = 330 Ns/m (550 Hz mode). The lateral rail dynamic behaviour is represented by following equivalent parameters: m = 10 kg; k = $63*10^6$N/m; c = 10 000 NS/m.

The friction-creep curve as shown in figure 80 is considered. Carrying out a time domain analysis considering constant crabbing velocity yields wheel and rail vibrations. Figure 81 shows clearly the wheel squealing (top picture) and the rail vibration (bottom picture) for this base case (left pictures).

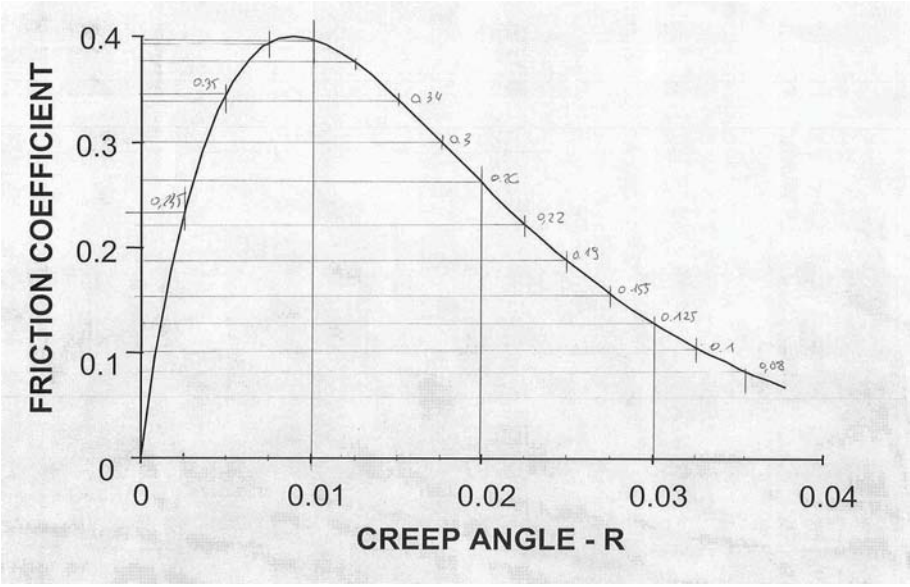


Figure 80 Friction-creep curve

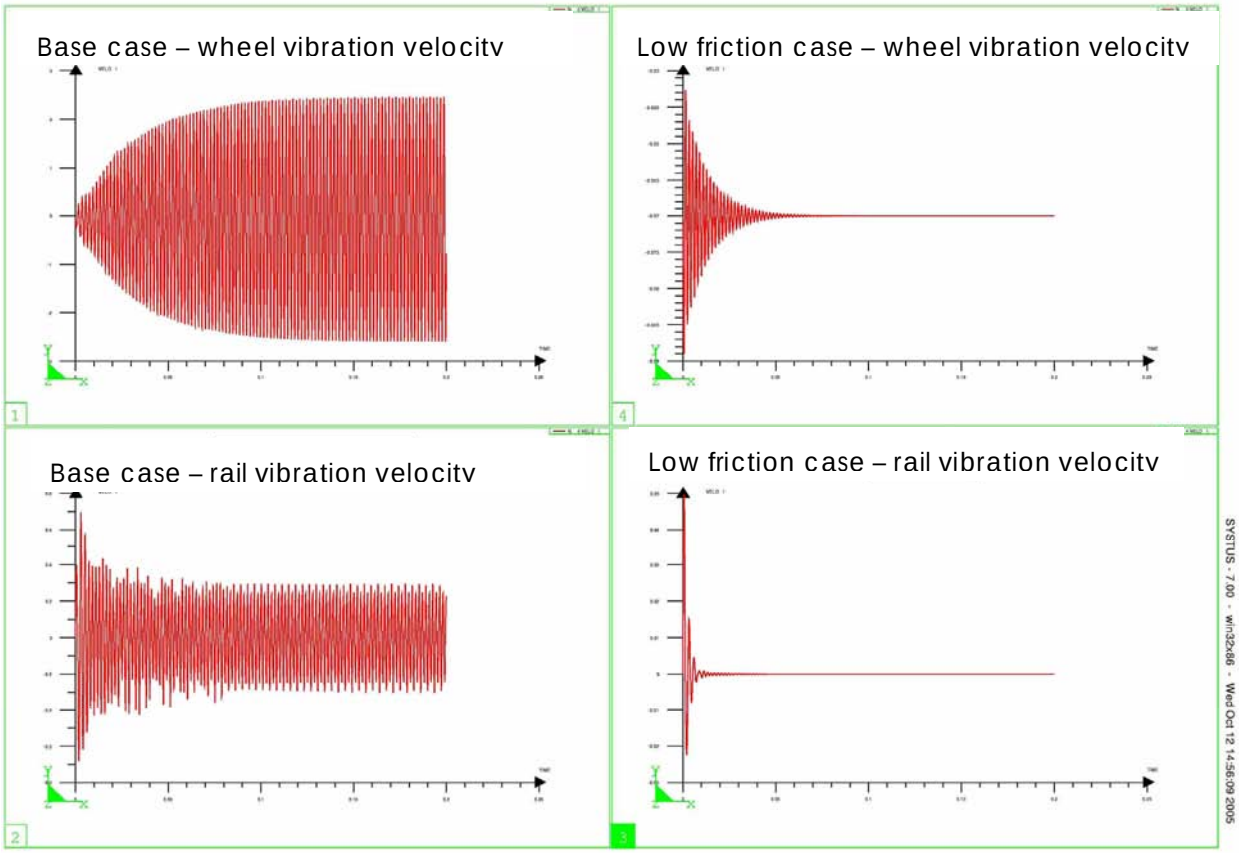


Figure 81

Low friction

Carrying out the calculation with a reduced friction curve (friction-creep curve amplitude divided to a factor of 3 e.g.) yields results given in the right part of figure 80: wheel and rail vibrations are reduced significantly and no squeal is occurring.

Laterally resilient rail

Carrying out the squeal calculation with a laterally resilient low damped rail system yields results given in figure 82: the wheel is squealing during 0.4 s but is then damped out; the rail is squealing at its low resonance frequency (not audible).

Following equivalent rail parameters have been considered for the lateral rail dynamics: m = 120 kg; k = $14*10^6$N/M; c = 250 Ns/m.

It is observed that as soon as the rail squeal vibration velocity reaches the wheel squeal vibration velocity, the wheel vibration is damped out to a very small value: no wheel squeal is occurring any more.

This model explains what is happening in a curve in the Antwerp tramway system (keerlus Mechelen): with laterally resilient rail fasteners squeal is minimal (audible during very short periods and then disappearing). In another identical curve with classical ballasted track (keerlus Zwijndrecht), squeal is present and continuous at a much higher amplitude, see figures 83 & 84 (same vehicles at same speeds, same curve radii, same rails).

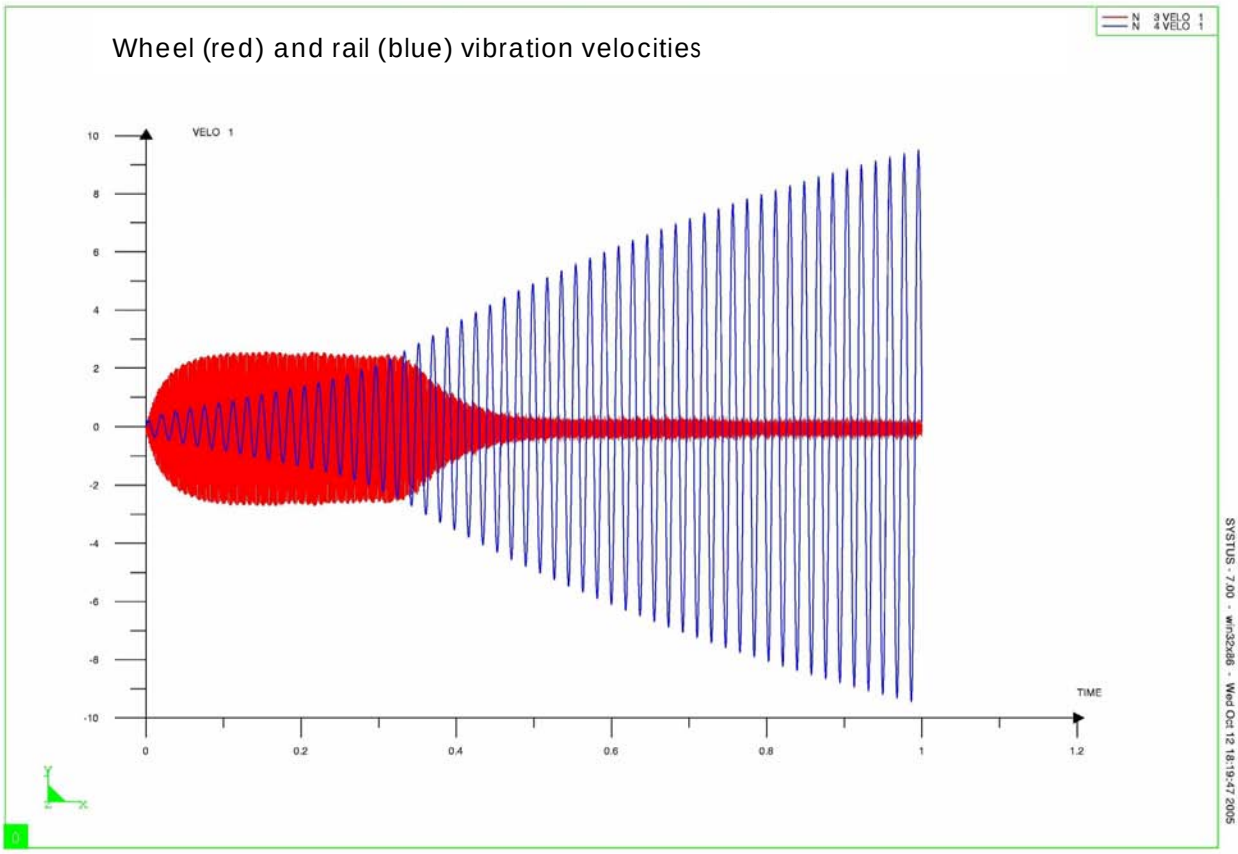


Figure 82

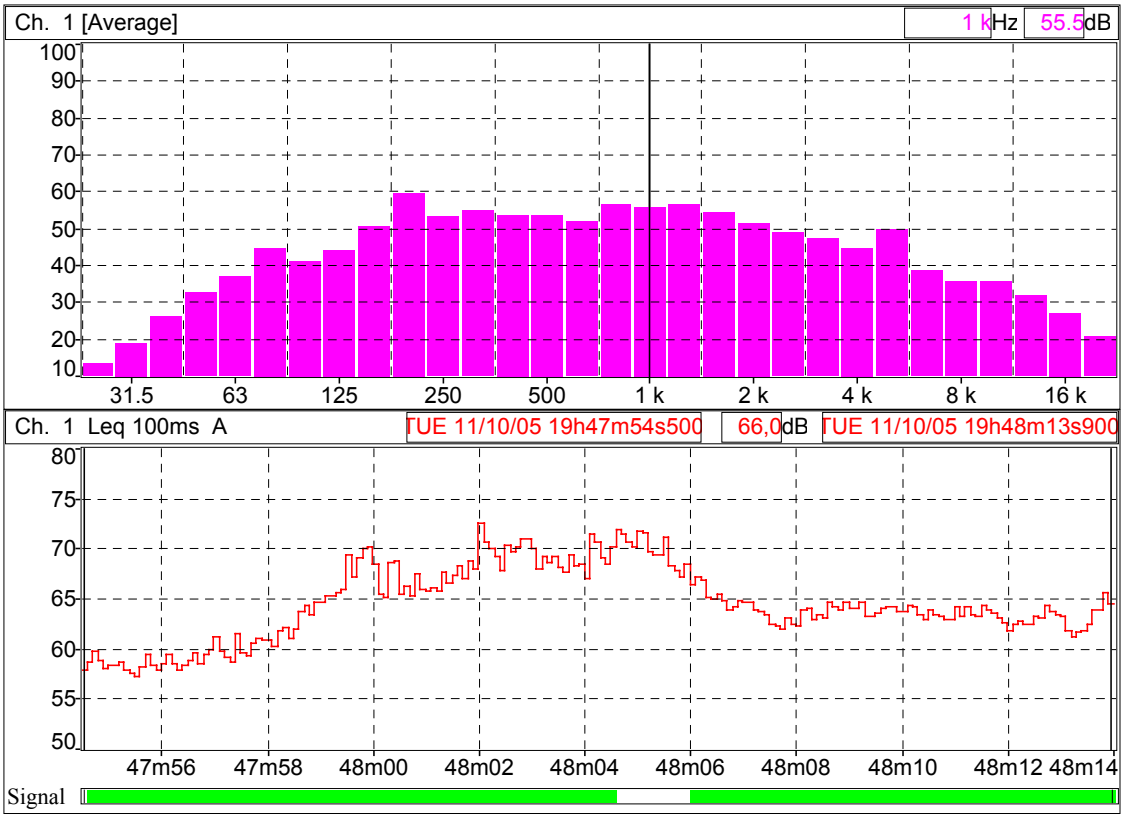


Figure 83

Noise spectrum and L_{Aeq} values during curve negotiation in Melsele (low damped tram track with high lateral resiliency)

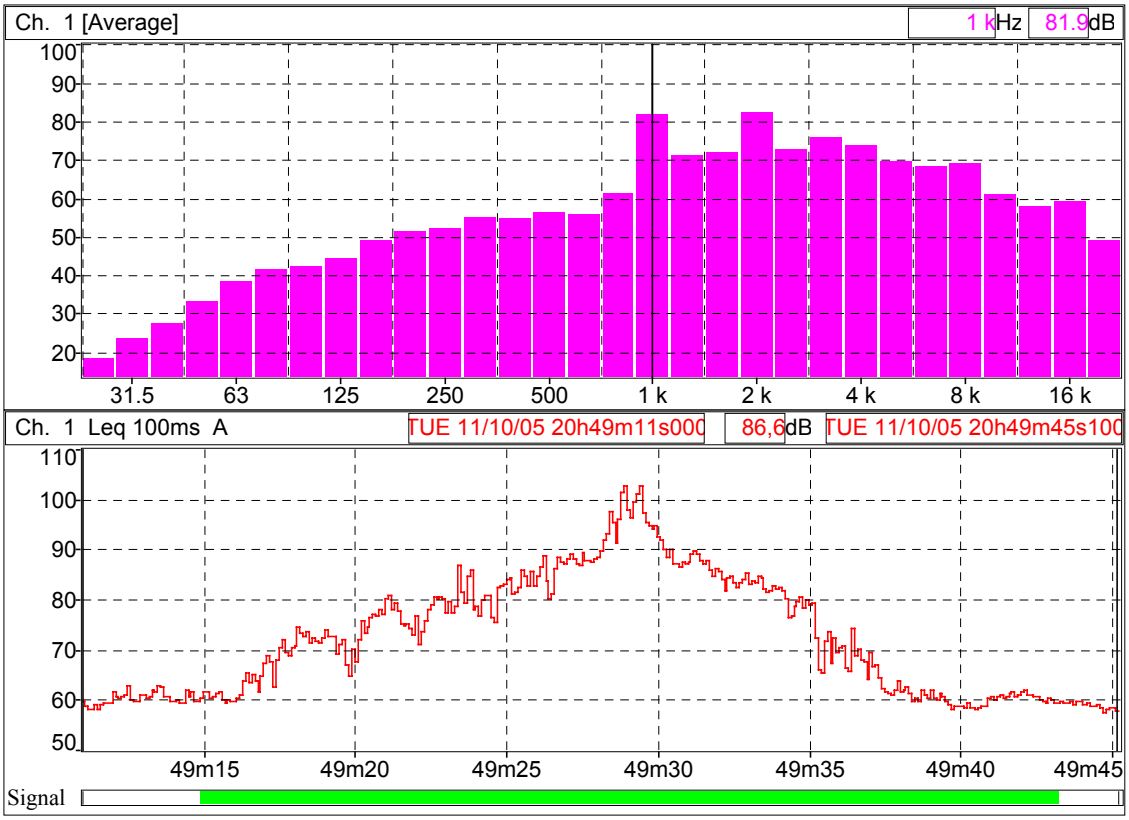


Figure 84

Noise spectrum and L_{Aeq} values during curve negotiation in Zwijndrecht (classical ballasted tram track)

Conclusions

The conclusions from the parametric study are as follows:

1. result 1: control of wheel/rail friction behaviour can avoid/reduce squeal noise. Squeal will be reduced with a lower coefficient of friction. Squeal will be avoided with a friction curve which exhibits no negative slope (positive friction curve);
2. result 2: reduction of the rail mechanical impedance at squeal frequency to a value below the wheel mechanical impedance will reduce squeal noise as long as the rail damping is low enough;

9.3 MITIGATION MEASURES

Mitigation measures according to the principles discussed in point 10.2.3 above are:

9.3.1 Rail Hardsurfacing (according to result 1)

A method for reducing friction by applying a bonded film coating to the rail was shown effective, but only for a very short time. This was demonstrated in the EC INFRA research project and in the US TCRP project (D-7). Data suggest that although the reduction in friction was limited, squeal noise was reduced significantly immediately after application. However, the coating used here did not provide a sufficiently robust modification of the surface to affect all trains for an extended period.

Rail hardsurfacing will therefore no longer be considered in this project.

9.3.2 Top of rail friction modifiers (according to result 1)

A top of rail friction modifier is a water-based liquid material. After water removal, a thin dry film remains that provides an intermediate coefficient of friction in the range 0.30-0.35. This friction level does not compromise braking or traction. The differences between a true friction modifier and a lubricant have been reviewed [Appendix A]. The friction modifier is a suspension of engineered solids that provide the required frictional properties.

The positive friction characteristics of this friction modifier thin film have been established in different rig studies. The frictional properties of the thin film are the net result of the friction modifier and the other Third Body components. The friction modifier is designed for optimal interaction with the iron oxide wear components that dominate the Third Body under normal conditions. The film characteristics are developed to provide the appropriate durability so the film lasts for as many axles as possible.

The friction modifier can be applied by using a trackside applicator.

Top of Rail friction modifiers will be further considered in this project. A low friction coefficient is required in order for the friction modifiers to be effective. This is the major challenge in this development, together with a uniform application of the friction modifier product on the rail surface.

9.3.3 Saddle profile (according to result 2)

What is a saddle profile rail?

A saddle profile rail is a rail concept where the top of the railhead is arranged in form of a U-shaped profile (called Saddle Profile, figure 84) where the wheel will run on the top of the saddle profile. The saddle profile is in turn uncoupled from the rest of the rail structure by a thin rubber mat.

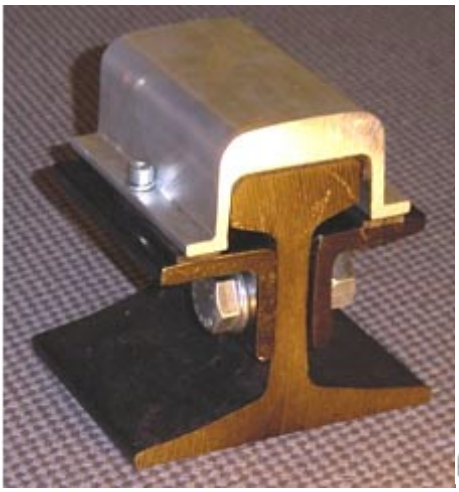


Figure 85

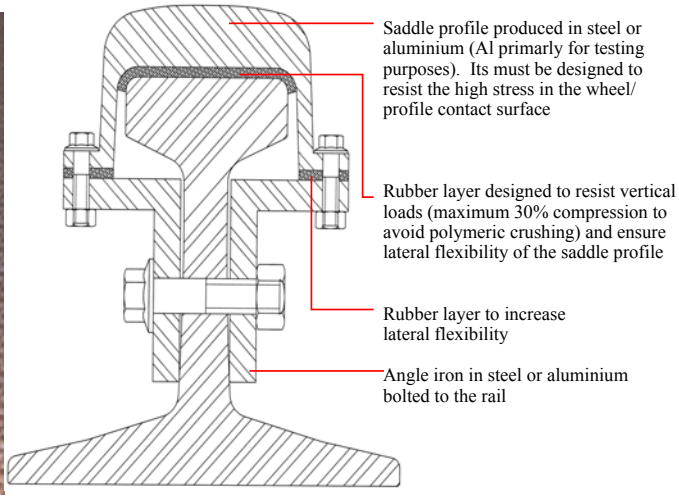


Figure 86 Saddle profile for squeal noise reduction mounted on top of the railhead. Conceptual sketch - cross section.

Reduced rail impedance – selecting the most feasible concept

For any efficient treatment for squeal noise reduction aimed to be applied onto the rail, the rail impedance must be decreased so that it is lower than the wheel impedance. Decreasing the dynamic mass of the rail is one way of achieving this since the dominating part of the rail impedance at squeal frequencies is the mass reactance. One way of achieving reducing the rail dynamic mass and thus also the rail impedance at squeal frequencies is to uncouple part of the railhead from the rest of the rail structure. Achieving such lowering of the dynamic mass and impedance of the rail is possible by adding a U-shaped profile onto the railhead, which is elastically uncoupled from the rest of the rail: "Saddle Profile Rail". A sketch of the Saddle Profile Rail mounted on top of the railhead is presented in figure 86 as a cross section.

Squeal sound measurements before and after installation of the saddle profile rail

The saddle profile was installed at a test site in Stockholm (SL network) in October 2000. Measurements of squeal sound levels were performed.

Figure 87 presents the squeal sound pressure levels during train passage.

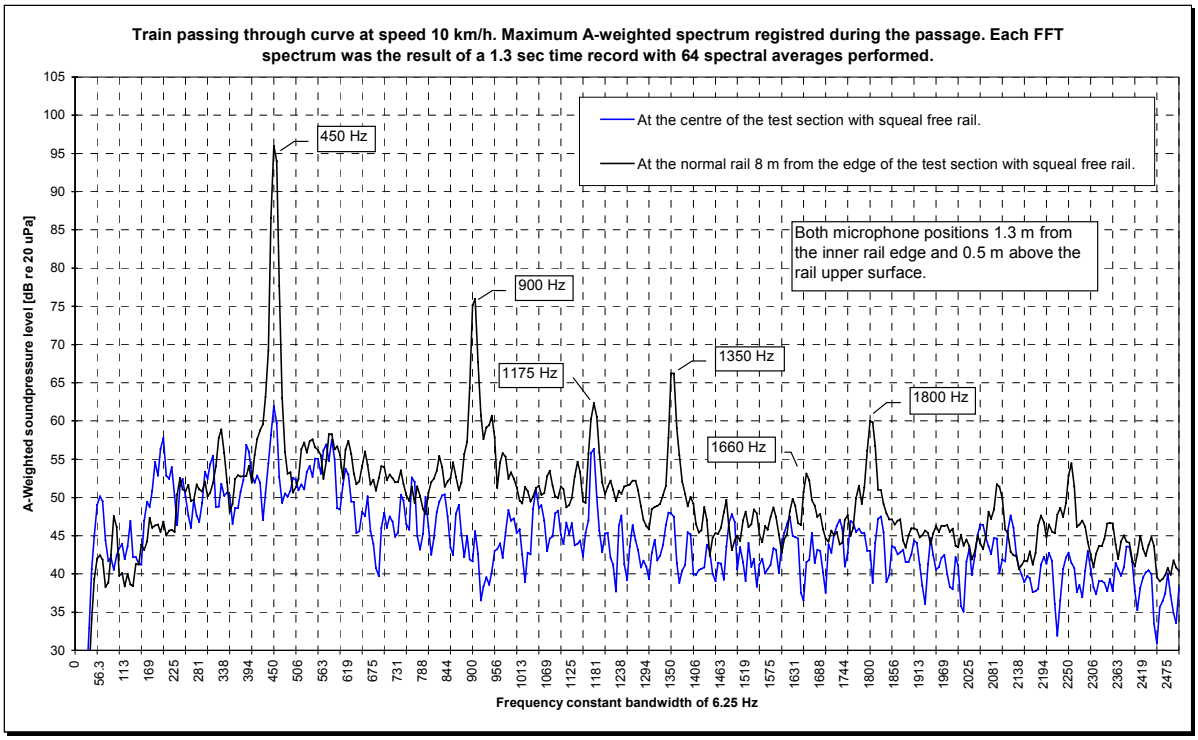


Figure 87

The two microphone positions are registered simultaneously during a train passage. FFT were performed on the signals at time intervals of 1.3 s, which produced 64 spectral averages. The frequency resolution is 6.25 Hz. The maximum spectrum produced during the whole passage for each microphone position is presented.

The reduction in squeal sound pressure level is about 35 – 37 dB for the fundamental squeal frequency at 450 Hz and as much as 25 - 30 dB is achieved for a squeal frequency at 900 Hz. This means that the squeal sound is almost totally eliminated.

Although the performance of the saddle profile has been demonstrated, this method will not be considered further in the project due to the complexity of installing this profile on girder rail (required for tram application).

9.3.4 Laterally resilient rail fasteners (according to result 2)

A low rail impedance in lateral direction (with low modal damping of first modes) can be obtained with a special fastener concept yielding dynamic stiffness below 10 kN/mm. This results in a solution which is applicable for tramways and which will be further considered in the project. A typical rail fastener is shown in figure 88.

Figure 88

9.4 CONCLUSIONS

Emphasis will be on the validation of two mitigation measures against squeal noise: top of rail friction modifiers and laterally resilient rail fasteners with low damping.

For friction modifiers, a low friction coefficient is required in order for the friction modifiers to be effective. This is the major challenge in this development, together with a uniform application of the friction modifier product on the rail surface.

None of the above described mitigation methods will avoid possible contact between wheel flange and the outer railhead during curving. It is anticipated that this phenomenon (flange rubbing) will not be a sufficient source mechanism to generate squeal noise.

The costs of a rail treatment against squeal noise are budgeted as follows, considering a 100 m long curve:

- top of rail friction modifier: installation and equipment costs of unit with 6 nozzles: ± €30 000 (plus consumption of friction modifier product, estimated at €150/month);
- laterally resilient rail fasteners: cost of 320 rail fasteners: €32 000 (extra material cost in comparison with classical rail fastener).

10 QUALITATIVE ASPECTS OF NOISE MITIGATION AT WHEEL/RAIL LEVEL – STATE OF THE ART

10.1 INTRODUCTION

This document is a qualitative guide for identifying and evaluating noise control treatments with respect to noise reduction effectiveness at wheel and rail level and cost. Treatments are segregated according to tangent track, curved track, and special trackwork noise control. In each of these categories, treatments are presented for bogies (wheels) and trackwork.

The various categories of wheel/rail noise include:

1. rolling noise at tangent track,
2. curving noise, and
3. special trackwork noise.

Rolling noise at tangent and moderately curved track without squeal is most representative of conditions used for qualification testing of transit vehicles. Normal rolling noise with smooth rails and trued wheels, excessive rolling noise due to excessive random rail and wheel roughness, impact noise due to rail and wheel imperfections and joints, and noise due to short-pitch rail corrugation are normally associated with tangent track noise.

Curving noise primarily involves wheel squeal and, perhaps, wheel howl, in addition to rolling noise. The discussions concerning tangent track noise should be referred to when dealing with rolling noise at curved track. Noise from special trackwork includes impact noise generated by wheels traversing gaps in trackwork components, specifically switch frogs and crossovers and the associated treatments include special trackwork components that minimise this type of noise.

Any treatment selected for noise control should be carefully reviewed by the transit system engineering staff for cost, practicality, and safety.

Representative, order of magnitude, costs are provided for a comparison of various noise control treatments. These costs are listed to aid the selection process, and the user should verify costs with suppliers and contractors before selecting or rejecting a treatment.

This document summarises work done mainly at TCRB (TCRB, Report 23, Wheel/Rail Noise Control Manual, James T. Nelson, National Academy Press, Washington, D.C., 1997).

10.2 TANGENT TRACK NOISE

Tangent track noise includes:

1. normal rolling noise,
2. excessive rolling noise,
3. impact noise, and
4. corrugated rail noise.

The selection of a noise control treatment depends on the type of noise. For example, damped wheels are effective in controlling squeal, but have historically produced little reduction of tangent track rolling noise at transit systems. As another example, normal rolling noise at tangent track would not be reduced significantly by rail grinding or wheel truing, because the rails and wheels would already be in good condition (though some minor noise reduction might still be expected). Therefore, the user must identify the type of noise before deciding on a treatment scenario.

10.2.1 Normal Rolling Noise

Normal rolling noise occurs for smooth ground rail with optimum rail and wheel profiles. The rail will appear smooth, free of spalls, pits, shelling, and corrugation. The contact patch width will be uniform in width, without plastic flow at the edges, and will be about 12 to 18 mm. A straight edge placed longitudinally along the rail running surface will indicate a uniform rail height profile when backlit. There should be a minimum of flange contact with the gauge face, and there should be no two-point contact wear patterns on the railhead. The contact wear strip should ideally be centred on the railhead, over the stem, or, at most, centred 12 mm to either side of the rail centre if variation of contact location is desired to avoid tread rutting. The ball of the rail should be radiused, and the wear pattern should not encounter the edge of the radiused portion of the ball. The wheel tread will be smooth, without pits, spalls, polygonisation, or other imperfections, and the tread profile will not be significantly worn.

The passby noise will appear to be uniform from one vehicle to the next, and not exhibit harsh pure tones, or "roar" due to corrugation or other imperfections. The passby noise level signature will vary smoothly with time, each wheel set contributing a similar amount of noise energy. In contrast, a single flatted wheel will produce as much as 7 to 10 dB higher noise level in the open on at-grade or aerial structure track than each of the remaining wheels, and will be clearly identifiable during passage. Rail grinding and wheel truing will not greatly reduce normal rolling noise levels, as the running condition of the rail and wheels should already be good. There may be some optimisation of railhead contour and wheel profile that may reduce noise, but this will likely be limited to a few decibels or less.

Table 24 lists various options available for controlling normal rolling noise. Expected noise reductions, costs, and site specific limitations are also listed. The options available for noise control are not extensive.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	resilient wheels	1 to 2	2 400,00 to 3 000,00 per wheel	May not be appropriate for tread braked systems.
trackwork	trackbed absorption	5	100,00 per m²	Ineffective for ballasted track.
	rail vibration absorbers	1 to 2	500,00 per m	Requires clearance between rail and invert. Concern over failing from aerial structures.

Table 24 Noise control options for normal rolling noise at tangent track

10.2.2 Bogie Treatments

The bogie (wheel) options available for controlling normal rolling noise are limited primarily to resilient wheels though the latter's noise reduction is limited to 1 to 2 dB. Damped wheels are not considered to be effective, because the maximum A-weighted noise reduction observed for typical transit application has been about 0 to 1 dB(A). Similarly, although on-board dry stick lubricants have been promoted by various manufacturers as an effective reducer of rolling noise, data collected do not demonstrate this, and thus are not included here.

Resilient Wheels

Resilient wheels may provide about 1 to 2 dB reduction of wayside and car interior noise. However, their use would likely be driven by the need to control wheel squeal at curves, for which they are most effective; resilient wheels should not, as a rule, be used solely to control rolling noise. Another benefit of resilient wheels is reduced bogie shock loading. Costs for resilient wheels vary by manufacturer and by application, and range between € 2 000,00 and 3 000,00 per wheel, considerably higher than the cost of about € 700,00 for solid steel wheels.

10.2.3 Trackwork Treatments

Trackwork treatments include sound absorption at the track level, perhaps between the rails, rail vibration absorbers, and any other measure for which the track maintenance department would be responsible for. Each of these is briefly discussed below.

Trackbed Absorption

Trackbed absorption is effective for direct fixation track with concrete inverts or slabs, such as at concrete aerial structures. Noise levels at ballast and sleeper track are normally 4 to 5 dB lower than at aerial structure or concrete slab track with direct fixation fasteners, ostensibly because of the sound absorption provided by the ballast, and additional trackbed sound absorption would be ineffective at ballast-and-sleeper track. There may be substantial maintenance problems associated with sound absorption treatments positioned beneath the train in exposed situations. Such

problems may involve ability to inspect and maintain track components. Debris may accumulate readily beneath the absorption, making cleaning of the invert difficult. The treatment would be effective for station platform areas, or areas where debris would not accumulate. The absorption must be protected from tunnel washing machines or other maintenance equipment, which might otherwise damage the treatment.

Candidate treatments include:

1. encased glass fibreboard protected by perforated sheet metal or fibre reinforced panels,
2. spray-on cementitious sound absorption, or
3. ballast.

In the case of ballast, electrical insulation may be compromised if the ballast extends to the top plate of the fastener. A variant of trackbed absorption is under-platform absorption for stations. There is usually a recess under the platform edge and underplatform absorption placed against the far wall of this recess and the underside of the platform overhang provides a particularly effective means for controlling station platform noise levels in subways.

Rail Vibration Absorbers

Rail vibration absorbers give a noise reduction of max. 1 to 2 dB(A). Vibration absorbers are spring-mass systems with damping incorporated into the spring to absorb and dissipate vibration energy. They are attached to the rail with clamps, without contacting the invert or ballast. Vibration absorbers may be tuned by the absorber manufacturer to optimise dissipation of rail vibration energy into heat over a particular range of frequencies, and may be particularly desirable at locations where a sound barrier would be impractical and the needed noise reduction is on the order of a few decibels. The unit cost for rail vibration absorbers is expected to be on the order of € 100,00 per absorber. Recent extensive work in the U.S. has shown the small interest of using rail vibration absorbers to reduce noise.

Resilient Rail Fasteners

Resilient fasteners are not normally considered as a treatment for wheel/rail noise. They are designed to reduce low frequency groundborne or structureborne noise above about 30 Hz, and can be effective in reducing wayside noise radiated from steel elevated structures or aerial structures with steel box girders. Included in this category are the twin booted sleeper systems. Resilient fasteners with elastomer springs have been proposed for reducing wheel/rail noise radiation by utilising the damping properties of the elastomer. In general, very resilient rail fasteners can generate up to 3 dB(A) more rolling noise.

10.2.4 Excessive Rolling Noise

Excessive rolling noise due to random roughness is caused by rough rails and/or wheels, but the rail is without identifiable rail corrugation, joints, or other large imperfections in the running surface. Excessive rolling noise would normally arise after a period of no rail grinding or wheel truing, and rail condition should be visibly deteriorated with pits, fatigue cracking, gauge corner metal plastic flow, etc. Excessive rolling noise may occur without obvious visible defects, resulting simply from high amplitude random roughness, flat railhead, improper cant, rail grinding pattern, etc. Excessive rolling noise may also exist in spite of rail grinding, where the rail grinding is minimal, or does not provide a smooth, uniform contact wear pattern edge definition.

Candidate treatments for controlling excessive rolling noise are listed in Table 25. These treatments are further discussed below with respect to bogie and trackwork. All of the treatments for normal rolling noise are applicable to excessive rolling noise, though treatments for normal rolling noise should normally not be applied unless needed after the treatments identified in Table 25 are considered. The treatments listed in Table 2 are effective specifically against the wheel and rail conditions, which produce excessive noise, and should be explored before resorting to treatments for normal rolling noise.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	wheel truing (in combination with rail grinding)	7 to 10	60,00 per wheelset	-
	treatments listed for normal rolling noise			
trackwork	rail grinding (in combination with wheel truing)	7 to 10	20,00 per m per year	-
	treatments listed for normal rolling noise			

Table 25 Noise control treatments for excessive rolling noise without corrugation

10.2.5 Trackwork Treatments

The principal trackwork treatment for excessive rolling noise is rail grinding. The trackwork treatments indicated for treating normal rolling noise are also applicable, but should be considered only after grinding the rail.

Rail Grinding

Rail grinding in combination with wheel truing is the most effective means of controlling noise due to excessive rail roughness. There is disagreement in the literature concerning the best profile for obtaining the least noise. Widening the contact patch has been conjectured to reduce net contact dynamic forces by averaging the rail and wheel roughness over the contact patch area. On the other hand, increased rail conformity increases spin-creep, possibly contributing to rail corrugation, wear, and, thus noise. At present, "squaring up" the contact patch appears to be a reasonable approach to

control wayside noise, subject to further study. Increased conformity due to wheel tread wear can be controlled with wheel truing. If wheel truing is insufficiently frequent such that concavity is prevalent in the tread profile, there may be other problems besides spin creep, such as poor ride quality and steering. In all cases, the track should be ground and the wheels trued to prevent two-point contact on the running surface of the rail. Rail grinding may actually increase wheel/rail noise if grinding introduces a periodic grinding pattern in the rail head, a condition which should be avoided, because there is no guarantee that the pattern will be entirely worn away with time. Visual evidence of the grinding pattern may disappear with time, but undulation and residual hardness variation may persist.

Grinding equipment must be selected and maintained in good condition to avoid tool chatter and debris accumulation in the grinding wheels to avoid excessively deep grinding patterns. The wavelength or pitch of any grinding pattern should be reduced to less than the contact patch longitudinal dimension by reducing grinding train speed. Patterns in narrow (2 to 3 mm wide) grinding facets, produced by multiple stone grinders and/or multiple passes to shape the head, will necessarily be averaged over the contact width, thus reducing noise. Wide grinding facets, on the other hand, would prevent such averaging.

Site specific limitations for rail grinding include:

1. lack of clearance in tunnels for the grinding machine,
2. lack of track access, due to conflict with revenue operation, and
3. inability to grind certain kinds of track, such as embedded curves.

10.2.6 Bogie Treatments

Wheel truing is the principal on-board treatment for controlling excessive rolling noise, and should be considered before resorting to treatments identified for normal rolling noise.

Wheel Truing

Wheel truing in combination with rail grinding is the most effective on-board treatment for controlling excessive rolling noise due to wheel roughness. Wheel truing may be considered as a necessary part of a vehicle maintenance program. Conversely, systems, which do not have effective wheel truing programs, probably experience abnormally high rolling noise. The present information indicates that there is little difference between the type of wheel truing machine used, e.g. milling or lathe, and the resulting noise level. Noise reductions on the order 7 to 10 dB may be expected for initially rough wheels if the rail is also ground, though actual noise reductions will depend on the state of roughness of both the rails and wheels. Wheel truing without an effective rail-grinding program may not achieve the lowest noise levels possible, or vice versa, because a rough rail may partially or completely mask the noise control benefits of wheel truing. However, even without rail grinding, wheel truing should yield significant noise reductions and should be performed. There are no apparent site-specific limitations for wheel truing, other than availability of facilities.

The cost of wheel truing includes the cost of the wheel truing machine, maintenance, materials such as cutting tools and labour. The cost of a typical truing machine is on the order of € 1 000 000,00 with the milling type of machine being the less expensive and less accurate.

10.2.7 Impact Noise

Impact noise may be the most significant source of noise at transit systems where rail grinding and wheel truing are not performed or are performed on an infrequent basis.

The causes of impact noise include chips, spalls, burns, rail joints, and simply excessive curvature of the rail surface in the longitudinal direction. Presented below are remedies for impact noise. As with excessive rolling noise, the most effective treatments are rail grinding and wheel truing. The noise control provisions appropriate for impact noise are presented in Table 26.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	wheel truing	7 to 10	60,00 per wheelset	Single most important treatment, because wheel flats are most significant cause of impact noise.
	slip-side control	7 to 10	5 000,00 to 10 000,00 per vehicle	Reduces flat occurrence by about 50%, and thus reduces wheel truing costs proportionally.
trackwork	rail grinding	7 to 10	4 000,00 per km track	Must be done in conjunction with wheel truing.
	defect welding & grinding	0 to 3	200,00 per defect	Noise reductions depend on number of defects. Costs are subject or local labour rates and field conditions.
	joint maintenance	2 to 3	200,00 to 400,00 per joint	Primarily relevant to older transit systems with steel elevated structures.
	field welding of joints	5	600,00 per joint	Ancillary cost benefits in reduced maintenance.
	eliminate rail support looseness	5	250,00 per m	Achieved with resilient direct fixation fasteners or concrete ties with spring clips. Primarily relevant to steel elevated aerial structures.
wayside	treatments for normal and excessive rolling noise			

Table 26 Treatments for impact noise du to rail defects

10.2.8 Trackwork Treatments

Treatments for impact noise include rail grinding, defect welding and grinding, joint maintenance, field welding of joints, and elimination of loose track supports. These should be considered before resorting to trackwork treatments identified for normal rolling noise.

Rail Grinding

Rail grinding is by far the most effective means for controlling impact noise due to rail defects. The rail-grinding machine should be able to control rail height uniformity over a length of about 2 m to eliminate impact noise due to excessive rail height curvature in the longitudinal direction.

The grinder should remove enough metal to eliminate fatigue cracks, pits, spalls, chips, and burns. Over grinding may be desirable to reduce stress concentrations and hardness variation. After an initial deep grind to remove defects, the railhead should be recontoured to maintain proper contact patch width and location. After this process is complete, the rail should be regularly ground to maintain a smooth running surface.

Defect Welding and Grinding

This procedure involves deposition of weldment, and grinding the weldment to achieve a smooth running surface. Field welding costs are estimated to be € 200,00 per defect. Site specific limitations may involve weldability of alloy steels, and track access. The rails should be inspected prior to treatment to determine if other, more serious defects exist, such as fatigue fracture of the body of the rail, in which case rail replacement may be the best choice. Chips and spalls should be ground out without welding by moving the contact strip and cleaning up the defect. If the chips and spalls are not too deep, they may be taken out by a thorough grinding of the rail with a rail grinding train as discussed above.

Joint Maintenance

Impact noise is generated at rail joint gaps and elevation discontinuities. Large joint gaps create more noise than short joint gaps. Misalignment of the running surface elevation will result in impact noise. Further, the ends of the rail may require weldment deposition and grinding to repair end-batter. Thus, joint maintenance includes tightening rail joints to remove or reduce gaps, aligning running surface elevations, and repairing battered ends.

Field Welding of Joints

Field welding of joints, or replacement of the rail with continuous welded rail, eliminates impact noise at joints, and results in an overall reduction of maintenance effort.

Field welding, or use of continuous welded rail, may not be practical on aerial structures, where thermal expansion and contraction may place high loads on aerial

structure components. Examples include the NYCTA steel elevated structures, some of which may be over 100 years old. Modern reinforced concrete aerial structures are normally capable of carrying continuous welded rail, though provisions are made in fastener design to accommodate thermal expansion and contraction.

Eliminate rail support looseness

Elastomer fasteners add damping to the rail, which helps to reduce the effective noise radiating length of the rail, and thus noise. Resilient fasteners also may help to reduce noise radiated by steel box girders supporting aerial structures, or from older steel elevated structures. Very resilient fasteners with very low lateral compliance may be less desirable since they allow a greater radiation length of the rail. Costs for resilient fasteners are in the range of € 80,00 to 100,00 per fastener. This solution has reduced the noise radiated by steel aerial bridges at RATP metro by more than 5 dB(A): Pont du Nord, Pont de l'Est.

10.2.9 Bogie Treatments

Wheel truing and slip-slide control are the principal on-board treatments available for controlling impact noise, and these should be considered prior to employing treatments identified for normal rolling noise.

Wheel Truing

As with rail grinding, wheel truing is the most effective means for controlling impact noise produced by wheel flats, chips, and spalls, or other defects in the tread running surface. Noise reductions on the order of 10 dB may be expected where no truing had been performed before, provided that the rail is sufficiently ground and maintained. There is also an improvement in the qualitative perception of the noise. After truing, and assuming the rail is smoothly ground with no defects or joints, the wheel/rail noise should have the sound of a smooth running bearing, without harshness or audible impacts. In fact, there may be some difficulty distinguishing between wheel/rail noise and propulsion system noise.

Slip-Slide Control

Slip-slide control, standard on most new transit vehicles, is an electro-mechanical servo-controlled system, which limits wheel slip during acceleration and sliding during braking, and reduces the occurrence of wheel flats and burns. Braking pressures and motor torques are modulated to equilibrate wheel set rotational velocities. Wheel flats will still occur with a slip-slide system, thus the need for wheel truing is not eliminated. However, the truing interval can be lengthened, and lengthening the truing interval reduces truing costs. Roughly, a 50% reduction of wheel flat occurrence may be expected under normal conditions, which would translate into a 50% reduction of wheel truing periodic costs. Further, slip-slide control improves traction and braking during wet weather, providing ancillary benefits in addition to noise control which might justify its cost regardless of noise reduction benefits. The cost of slip-slide control is difficult to determine, since many vehicles come standard with such control, but should range between € 5 000,00 and 10 000,00 per vehicle. The cost should be balanced against the savings in reduced wheel truing and extended wheel life.

10.2.10 Corrugated Rail Noise

Noise due to rail corrugation is, perhaps, the most objectionable of the types of wheel/rail noise occurring at tangent or moderately curved track, and also one of the most difficult to control. The harsh tonal character of corrugation noise makes it one of the most easily heard and identifiable types of community noise, often affecting large areas.

Rail corrugation noise can be painful to transit system patrons, not to mention interfere with conversation, and many complaints concerning excessive noise from rail transit systems are directly related to rail corrugation. Descriptive terms for noise due to rail corrugation are “roaring rail” or "wheel/rail howl”. Roaring rail or wheel/rail howl at severely corrugated track may be a special type of periodic impact noise due to loss of contact between the wheel and rail. An inspection of the theory of impact generated noise for smooth rail undulations reveals that typical corrugation amplitudes are sufficient to produce contact separation.

The noise control provisions appropriate for corrugated rail noise, are presented in Table 27. Again, rail grinding is the most effective method of treating the symptoms of rail corrugation, but may not remove the conditions, which lead to or promote corrugation. Many other treatments are indicated for which there is insufficient information concerning effectiveness, but are included for consideration and further evaluation by the user. A link has been established with the EC Research Project CORRUGATION.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	wheel truing	NA	60,00 per wheelset	Believed to reduce spin-slip corrugation
	friction modifier	NA	1 400,00 per vehicle per year	Believed to be effective but data inconclusive. Costs based on two axles treated per vehicle.
	damped wheels	NA	500,00 per wheel	Effectiveness unknown, but should be effective to extent that wheel resonances influence corrugation. Little direct reduction of wheel/rail noise.
trackwork	aggressive rail grinding	7 to 10	4 000,00 per km track	Definitely very effective in reducing noise and controlling corrugation. Costs vary substantially.
	reduced rail support stiffness	NA	0,00	Believed to be effective in reducing corrugation rate, but would not reduce noise directly.
	hardfacing	7 to 10	50,00 per rail per m	Controls corrugation by providing hard running surface

Table 27 Treatments for noise due to rail corrugation

10.2.11 Trackwork Treatments

Treatments for rail corrugation are limited primarily to aggressive rail grinding. A second direct treatment is hardfacing. Additional trackwork design provisions, subject to field evaluation, which might be effective in controlling rail corrugation, include reduced track support stiffness, reduced rail support separation or pitch, stiffened top plate, and vibration absorbers.

Aggressive Rail Grinding

The most effective approach to controlling recurrent or chronic rail corrugation is to grind the rail running surface, using an aggressive rail grinding program optimised to minimise long-term material loss and cost. Assuming that the corrugation growth rate is exponential, at least during the early stages of corrugation where the amplitude is not sufficient to produce contact patch separation, and that sufficient metal is removed to eliminate corrugation without over-grinding, an optimum grinding interval can be approximated as the time interval required for corrugation to grow by about 170% (based on an exponential growth rate). Both longer and shorter grinding intervals will result in higher rates of metal removal, and thus reduced rail life. A longer grinding interval will increase noise exposure; a shorter grinding interval will maintain lower levels of noise, but increase grinding costs. The corrugation growth time may not be sufficient for significant corrugation to appear after rail grinding, and once corrugation amplitudes develop to a point that corrugation noise is audible, the corrugation growth may already exceed the above criterion. In this regard, the rail should be ground sufficiently often to avoid visible corrugation growth in excess of random rail roughness, and avoid audible corrugation noise. The optimum grinding interval is thus difficult to define, and some experimentation and careful monitoring of growth rates may be required.

Costs for rail grinding include the capital cost of the rail grinder, fuel and grinding materials, and personnel. Dust collection equipment in the form of a vehicle mounted vacuum cleaner may also be needed, producing an additional cost.

Hardfacing

Hardfacing with a very hard rail head inlay has been incorporated at European transit systems to control rail corrugation, especially at light rail or streetcar systems. The effectiveness of the treatment in controlling rail corrugation at light and heavy rail transit systems has not been demonstrated. The cost for hardfacing rail is estimated to be about € 100,00 per m of rail.

Alloy and Hardened Rail

Alloy rail, such as chromium vanadium have been considered for controlling corrugation at curves in heavy freight railroads. The use at transit systems for controlling short-pitch corrugation has not been determined. Alloy rail with greater hardness and wear characteristics might be considered to be less prone to corrugation than standard carbon steel rail, though exactly the opposite has been observed. There may be a reduction of weldability with hardened or alloy steels. Alloy and hardened rail should

not be considered for rail corrugation control without careful evaluation and consultation with a metallurgist.

Track Support Stiffness

Rail corrugation growth appears to be most prevalent at stiff direction fixation track where the rail support modulus is in excess of perhaps 70 kN/mm per m of rail, though no clear quantitative relation has been identified between track stiffness and rail corrugation growth. Many other factors must be considered. The anecdotal evidence suggests that a stiffness reduction may be beneficial in reducing corrugation growth rates. With stiff track supports providing a dynamic rail support modulus of about 70 to 100 kN/mm per m of rail, the rail-on-fastener resonance frequency is on the order of 175 to 200 Hz. A fastener consisting of a steel base plate and thin elastomer pad is very stiff, on the order of 200 kN/mm, giving a rail support modulus of 300 to 400 kN/mm per m of rail. Under these conditions, the vertical rail on fastener resonance frequency is on the order of 350 Hz or higher, at the lower end of the range of corrugation frequencies, suggesting a possible interaction between corrugation and vertical rail resonance. A benefit of soft rail supports is that the rail is less affected by the fastener pitch, which controls the pinned-pinned resonance frequency of the rail. Another benefit is that at corrugation frequencies, soft fasteners decouple the rail from the concrete invert, so that the rail is essentially an infinite beam, and thus has an input mechanical impedance with equal real and imaginary parts. (The real part is due to energy transmission along the rail.)

Thus, there is a natural damping mechanism that may reduce local resonances in the track and wheels at corrugation frequencies, a mechanism that is negated if the rail support modulus is too high. Although low stiffness fasteners appear to be attractive, the dynamic interaction between the rail and wheel is very complex, and a careful analysis, testing, and evaluation should be conducted before committing to a wholesale replacement of fasteners to control rail corrugation.

Track support stiffness may also be altered by selecting a special rail profile.

10.2.12 Bogie Treatments

On board treatments, do not directly reduce noise due to rail corrugation significantly, but may help to control rail corrugation rates. The on-board treatments are offered below for consideration, but are subject to testing and careful analysis; they are not proven approaches to corrugation control. However, to the extent that corrugation is intimately related to vibration and dynamic interaction, the options can be expected to influence the corrugation process.

Wheel Profile and Dimensions

The profile of the wheel's running surface directly affects interaction between the railhead and wheel. High wheel/rail conformity has been identified as a contributing factor in rail corrugation due to increased spin-slip. It is important to have wheels with a min. difference in wheel diameter on the same axle in order to reduce stick slip.

Reducing the maintenance tolerance on the diameter of two wheels of the same axle at STIB has reduced drastically the rail corrugation initiation and growth.

Dry-Stick Friction Modifiers

On-board lubrication of the tread with dry-stick friction modifiers to reduce or eliminate negative damping associated with stick-slip or roll-slip of the wheel/rail contact has been claimed to reduce corrugation growth rates. Only anecdotal information has been obtained to indicate that dry-stick lubrication reduces the need for grinding.

Damped Wheels

Damping of the wheel to reduce its response at corrugation frequencies is particularly attractive, though no data have been obtained indicating a reduction of corrugation rate with wheel damping. In any case, addition of damping to wheels should not increase the rail corrugation rate. A careful evaluation of damping should be conducted before employing damped wheels strictly for corrugation control.

10.3 CURVING NOISE CONTROL

As discussed previously, wheel/rail noise at curved track may differ considerably from that at tangent track, and may include a combination of normal and excessive rolling noise, impact noise, noise due to corrugation, wheel squeal due to stick-slip oscillation, and wheel howl. Wheel squeal is the most common form of curving noise, caused by stick-slip oscillation during lateral slip of the tread over the railhead, and may be excruciating to patrons or pedestrians. Wheel howl at curves may be related to oscillation at the wheel's lateral resonance on the axle, caused by lateral slip during curving. At short radius curves where train speeds may be limited to 20 km/h, rolling noise may be insignificant relative to wheel squeal. At curved track, normal rolling noise, excessive rolling noise due to roughness and corrugation, and impact noise due to rail defects and undulation are similar to those at tangent track, and the user is referred to the section on tangent track for discussion of noise not directly related to curving. Thus, the discussion of curving noise control presented below focuses on wheel squeal and wheel rail howl.

A link has been established with the EC Research Project SQUEAL.

Wheel Squeal

Treatments for controlling wheel squeal are listed in Table 28 with respect to bogie and trackwork.

The results of the survey of transit systems and inspection of track reveal a combination of factors which, taken together, appear to control or eliminate wheel squeal at embedded track. These are:

1. use of resilient wheels;
2. resiliently supported track;
3. gauge widening at curves;
4. wider wheel gauge than standard;
5. minimum curve radius of 30 m;
6. rail web filling blocks against the rails at both sides;
7. on-board dry stick low coefficient of friction flange lubrication;
8. on-board dry stick friction modifier applied to wheel tread.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	resilient wheels	10 to 20	2 000,00 to 2 300,00 per wheel	Well demonstrated to be effective.
	constrained layer damped wheels	5 to 15	500,00 to 1 200,00 per wheel	Effective.
	ring damped wheels	5 to 10	30,00 to 50,00 per wheel	
	wheel vibration absorbers	5 to 15	500,00 to 700,00 per wheel	Demonstrated effectiveness in trials.
	steerable bogies	elimination at large radius curves	unknown	
	onboard friction modifier	possible elimination	1 400,00 per vehicle per year	Limited effectiveness below 30 mradii.
trackwork	petroleum lubrication	partially eliminates squeal	10 000,00 to 40 000,00	Can lubricate flange only, so that effectiveness is limited.
	water spray lubrication	eliminates squeal	10 000,00 to 40 000,00 per track curve	Not practical in freezing weather, though antifreeze can be used with water.
	maintain constant gauge or gauge narrowing in curves	reduces truck crabbing & potential for squeal	0,00	Gauge narrowing has been correlated with squeal elimination in conjunction with resilient wheels, HPF friction modifier, and elastomer rail embedments.
	rail vibration dampers	unknown	100,00 per m	Reported to be effective
	rail inlay (anti-squeal)	reduces squeal		Reduced or eliminated squeal for several months.

Table 28 Wheel squeal noise control treatments

10.3.1 Bogie Treatments

Onboard treatments for controlling wheel squeal include resilient wheels, constrained layer damping, ring dampers, vibration absorbers, dry-stick lubrication, oil spray lubrication, longitudinal primary suspension compliance, and steerable bogies. Of these, resilient wheels are the most popular and are almost universally used for light rail transit, though squeal is not eliminated at short radius curves under 30 m.

Resilient Wheels

Resilient wheels incorporate elastomer springs between the tire and wheel rim to provide compliance between these components. Examples of resilient wheels include the Bochum 54 and 84 wheels, the SAB wheel, and the older PCC wheel. Resilient wheels are fitted with most light rail transit vehicles, and are enjoying widespread popularity. They are effective in reducing or eliminating wheel squeal at curves of radii greater than about 30 m, though squeal may still occur. Vibration dampers may be fitted to the tread of resilient wheels to further improve their squeal reducing characteristics. Regardless of the configuration, resilient wheels are expected to greatly reduce if not eliminate wheel squeal occurrence and energy.

The cost of a typical resilient wheel may range between € 2 400,00 and 3 000,00, which can be compared with a cost of roughly € 700,00 for a standard solid steel wheel.

Constrained Layer Damped Wheels

Constrained layer damping consists of visco-elastic material constrained between the wheel web and a metal plate or ring. Constrained layer damped wheels have been shown to be at least partially effective in reducing squeal. Although damped wheels tend to be less effective than resilient wheels, they may have an advantage over resilient wheels in that monobloc or composite steel and aluminium wheels would not have any difficulty with dynamic alignment, and can withstand high cyclic loading without heat build-up. A disadvantage of constrained layer dampers is that they add weight to the wheel, though this may be compensated partially by weight losses due to machining.

Wheel Vibration Absorbers

Wheel vibration absorbers are tuned and damped spring-mass mechanical systems that are attached to the tire to absorb vibration energy at wheel modal frequencies.

The absorbers can be effective over a broad frequency range, though they are most effective if tuned to the modal resonance frequencies of the wheels. A disadvantage is that, like the constrained layer damper, they add weight to the wheel.

Dry Stick Friction Modifier

Dry stick friction modifiers are applied to the tread running surfaces of one or two axles sets of each vehicle to enhance adhesion and flatten the friction-versus-creep-curve of the wheel and rail running surfaces, thus reducing negative damping and squeal. Dry

stick lubrication also includes low coefficient of friction flange lubricant applied to the flange throat to reduce friction and flange wear, and may help with curving of the truck and reduce lateral slip. Several light rail transit systems are now using on-board friction modifiers and flange lubricant on a regular basis.

Steerable Bogies

Steerable bogies have been developed for reducing wheel squeal by allowing the axles to align themselves parallel with the curve radius at the point of wheel contact.

Although steerable bogies may eliminate axle crabbing, longitudinal slip must still occur unless compensated with rolling radius differential.

10.3.2 Trackwork Treatments

Trackwork treatments include restraining rails, flange lubrication, asymmetrical rail profile grinding, gauge widening, rail head inlays, hardfacing, rail vibration dampers, and frictionless rail. Of these, flange lubrication is the most common trackwork treatment for controlling wheel squeal. Rail inlays are effective in reducing squeal, but are limited in life. The remaining treatments have been tried at various systems with varying degrees of success, and care should be used in selecting these for incorporation in track as noise control provisions.

Flange Lubrication

Flange lubrication with automatic grease lubricators is employed by various systems to control wheel squeal, based on the theory that flange contact with the rail is the principal cause of squeal. However, flange lubrication also may involve migration of lubricant to the rail running surface, and this limited and inadvertent lubrication may, in fact, be the principal cause of wheel squeal reduction. Excessive migration of lubricant to the rail running surfaces will result in loss of adhesion and braking performance, and is therefore to be avoided. For this reason, flange lubrication may not be employed by many systems.

Water Spray Lubrication

Water sprays are used at some systems to control wheel squeal. The advantage of water spray over petroleum lubrication is that the water evaporates quickly, and thus traction is reduced for only a short period of time and for a short distance. Also, water spray systems should pollute soils much less than petroleum systems. Water sprays can not be used during sub-freezing temperatures, due to build-up of ice.

Rail Head Inlay

Treating the rail running surface with an alloy inlay in the rail head can reduce wheel squeal due to lateral stick-slip across the rail running surface by modifying the friction-creep curve.

No data have been obtained concerning noise reduction effectiveness.

Hardfacing

Hardfacing with a very hard inlay has been used with mixed results, though the treatment appears to be most desirable for controlling wear at curves. The use of hardfacing inlay at the flange contact face may not produce a reduction of squeal, because, as theory suggests, the squeal is likely due to lateral stick-slip of the wheel tread across the top of the rail. Still, the dissimilar metallurgical properties of the inlay and wheel tread material may modify the friction-creep behaviour to have some benefit.

Rail Vibration Dampers

Rail vibration dampers have been advertised to reduce wheel squeal. If so, this would be contrary to expectations based on theoretical grounds, due to low participation of the rail in the squeal process relative to the wheel.

Gauge Widening

Gauge widening is reputed to reduce wheel squeal by increasing the rolling radius differential or by reducing flange rubbing. However, gauge widening promotes bogie crabbing, increasing the angle-of-attack, or creep angle, of the wheel relative to the rail, thus increasing lateral creep and squeal. Thus, gauge widening should not be expected to reduce squeal unless a rolling radius differential can be effected. Further, high crab angle allows flange contact and wear at the high and low rail gauge faces, evidenced by a sharp edge between the gauge face and rail running surface.

Curvature Design

Limiting track radius of curvature to greater than 45 m for vehicles with resilient wheels and 200 m for vehicles with the solid wheels with conical treads is probably the most practical wheel squeal noise control provision for new track construction. These appear to be practical limits below which wheel squeal can be expected. In the case of light rail systems where tight curvature is required to negotiate intersections, 30 m should be a limiting radius, though some limited squeal may be expected. Gauge widening should be avoided if possible. Although large radii are recommended for noise control, the benefits in reduced flange and rail wear should be obvious.

10.4 SPECIAL TRACKWORK

Special trackwork includes turnouts, crossovers, and switches, which may cause particularly intrusive impact noise. Noise from special trackwork is usually associated with switch frogs and crossover diamonds. Noise control measures are thus directed to reducing or eliminating the frog gap.

A link has been established with the EC Research Project TURNOUTS.

10.4.1 Special Trackwork Designs for Noise Control

Noise control methods, which may be applied to special trackwork, are listed in Table 29, and include moveable point frogs, spring frogs, and embedded track flange bearing frogs. Reduction of impact noise is the result of reduction of impact forces, so a reduction of truck shock and vibration may be expected, which may lead to reduced bogie maintenance.

location	treatment option	noise reduction [dB(A)]	cost [€]	site specific limitation
bogie	resilient wheels	3	2 400,00 to 3 000,00 per wheel	
trackwork	moveable point frogs	7	120 000,00 per turnout	Requires additional signalling.
	spring frogs	3	0,00	Standard design for embedded girder rail track.
	floating slab cut	5	unknown	Reduces floating slab rumble in tunnels.

Table 29 Special trackwork noise control treatment selection

Moveable Point Frogs

Moveable point frogs are most suited to high-speed turnouts. They are used in the railroad industry primarily as a means of reducing frog and wheel wear under high load environments. They are particularly effective for noise control because they virtually eliminate the gap associated with normal rail bound manganese frogs. Switches with moveable point frogs require additional signal and control circuitry compared with those with standard frogs, and, as a result, the total cost of a turnout with moveable point frogs is about € 200 000,00 or roughly twice that of turnouts with standard frogs.

Spring Frogs

Spring frogs are suitable for low speed turnouts and crossovers where use is occasional or emergency in nature, since ancillary noise may be produced by the spring frog for other than tangent running trains. Where frequent use of the turnout is expected, the spring frog may not be appropriate due to secondary noise related to spring frog actuation. The cost of a spring frog is roughly € 12 000,00 or twice the cost of a

standard frog. The incremental cost of a turnout with spring frogs relative to the cost of a turnout with standard frogs is thus relatively small. Spring frogs are not appropriate for speeds greater than about 30 km/h.

Flange Bearing Frogs

Flange bearing frogs are used in embedded track, and support the wheel flange as the tread traverses the frog gap. Flange bearing frogs do not necessarily eliminate impact noise and vibration, and their effectiveness depends on the degree of frog and flange wear. Flange bearing frogs are not used in high-speed sections of non-embedded track, and are therefore not normally considered as a noise control treatment.

Welding and Grinding

Maintenance of frogs by welding and grinding the frog point results in lower impact forces, and thus lower impact noise levels and wheel and frog wear.

Floating Slab Cuts

Substantial rumble noise is generated by special trackwork located on large continuous floating slabs. The floating slab acts as a sounding board, and is particularly efficient in radiating noise. An effective method to control this type of noise is to cut the slab between the tracks.

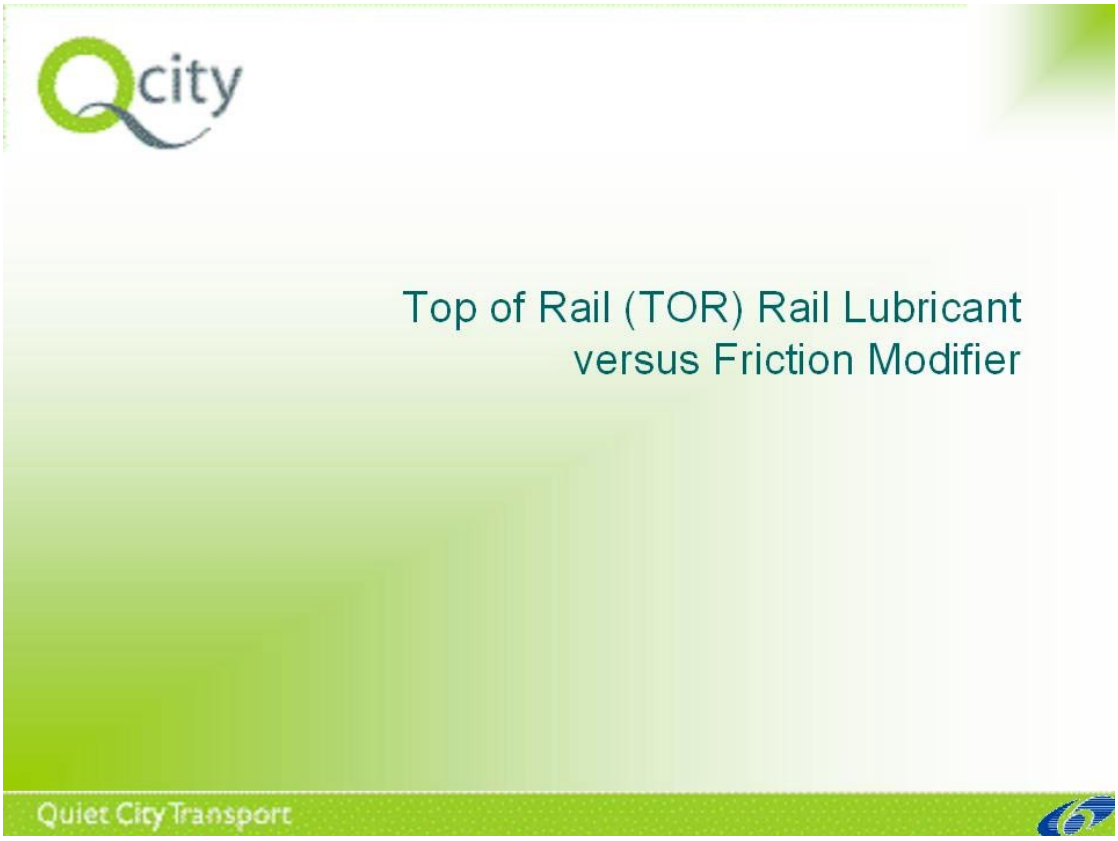
10.4.2 Bogie Treatments

There are few on-board treatments available for reducing special trackwork noise. However, some of the treatments identified for controlling rolling noise may be beneficial in reducing impact noise as well

Resilient Wheels

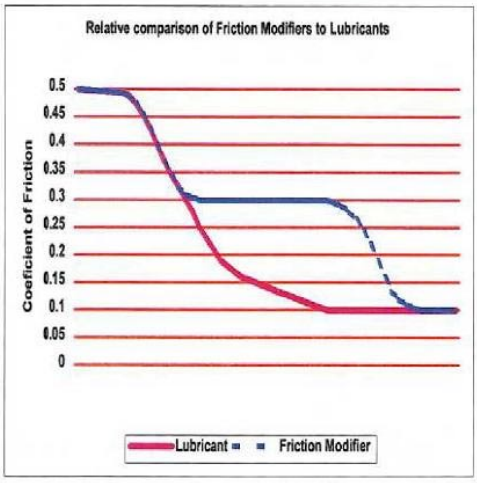
Resilient wheels reduce shock and vibration transmission into the truck and car body, and thus may be expected to reduce vehicle interior impact noise at certain frequencies, though the A-weighted noise reduction may be slight. No data have been collected indicating the possible car interior noise reduction obtained with resilient wheels at special trackwork.

APPENDIX A – LUBRICANT VERSUS FRICTION MODIFIER



What is a Friction Modifier ?

Friction modification products are designed to provide one friction level over a range of material thickness and/or hold friction over a specific range of wheel/rail creepage.

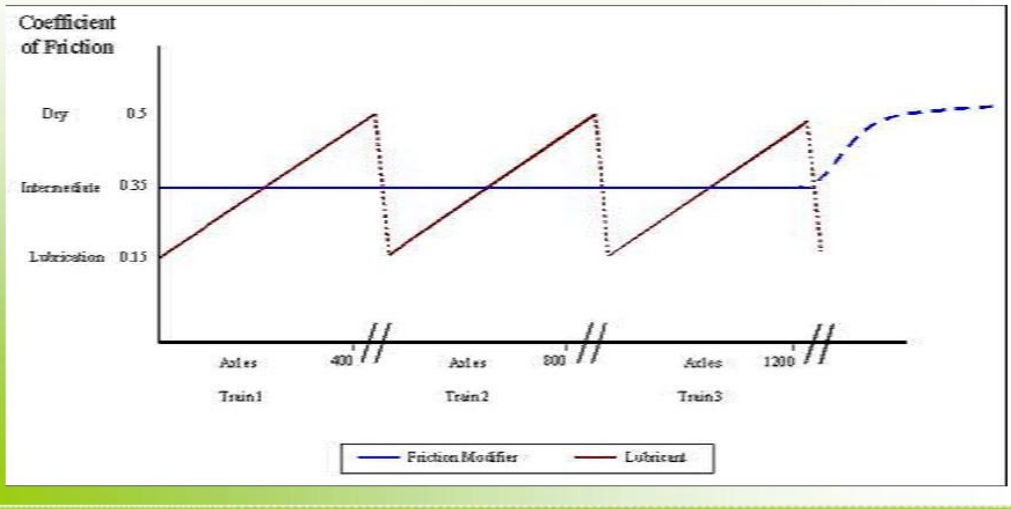


Note: Non-dimensional increase of friction modifier/lubricant thickness/amount left to right.
Figure 1. Conceptual Performance of a Lubricant and a Friction Modifier

Quiet City Transport

TOR lubricant vs. friction modifier

TOR lubricant provides continuously varying friction between 0.15 and 0.50 rather than at a controlled 0.30 - 0.35 range

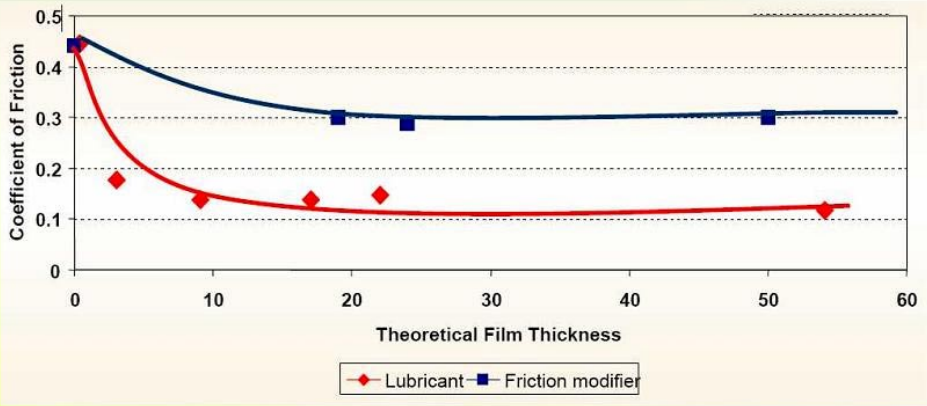


Quiet City Transport

TOR lubricant vs. friction modifier

Precise control is not possible to manage friction at an intermediate level with a TOR lubricant

Conversely, a true friction modifier has a broader application rate range to achieve intermediate friction control



Quiet City Transport

TOR lubricant vs. friction modifier

Significant differences in application philosophy:

LUBRICANT	*FRICTION MODIFIER*
PROCESS based friction control	***PRODUCT*** based friction control
Mechanism calls for oxidation of lubricant resulting in low initial friction readings which subsequently climb as the product is consumed	Engineered (inorganic) solids composite introduced to wheel rail interface provides intermediate coefficient of friction
Inconsistent lateral force reduction	Consistent lateral force reduction
Sophisticated system required to control and vary application (tonnage, train length, curvature, temperature) to assure consumption	Simple control system required as product does not need to be consumed. Only two application rates are required: tangent & curve

Quiet City Transport