STARDAMP

Standardisation of damping technologies for the reduction of rolling noise

Final Report

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List of abbreviations:

DR:	Decay rate of the wave field in a freely suspended rail
EMA:	Experimental modal analysis
FE:	Finite element
FRF:	Frequency response function
GUI:	Graphical user interface
SEMA:	Simplified experimental modal analysis
TDR:	Track decay rate – decay rate of the wave field in a real track

I. Summary

The central target of STARDAMP was to provide all stakeholders in the field of damping technologies for rail and wheel with a comprehensive methodology to predict and assess the performance of rail and wheel dampers for a wide variety of applications. A combination of laboratory testing procedures and computer simulation shall replace field tests as far as possible.

The STARDAMP consortium brought together leading manufacturers of rail and wheel dampers, the end-users of damping technologies (railway operators, suppliers of rolling stock and bogies/wheel sets), and research institutes and engineering companies with excellent reputation in the field of acoustic simulation and acoustic measurements. French partners are Alstom Transportation, Valdunes SAS, SNCF, Tata Steel and Vibratec. German partners are Deutsche Bahn AG, Gutehoffnungshütte Radsatz GmbH, Schrey & Veit GmbH and TU Berlin. The Institute of Sound and Vibration Research (ISVR, Univ. of Southampton) was included in the consortium as subcontractor.

As a first step, the scope and requirements for the STARDAMP tool were defined. Application of the STARDAMP assessment procedure shall not be limited to products already available on the market but it shall be applicable also to future products. In the case of rail dampers, it was necessary to define precisely, what a “rail damper” is, as this term is commonly used also for products with completely different mechanisms. The STARDAMP methodology shall be applicable to devices, which substantially increase the track decay rate when attached to the rail. It furthermore shall cover a wide variety of applications including ballasted track and slab track, all common rail types, wide range of rail pads stiffness and all kinds of traffic (high-speed, regional trains, freight). The speed range shall be 50 km/h – 320 km/h. Software and laboratory test methods shall be user-friendly and easy-to-handle as far as possible given the complexity of the rail-wheel system.

Starting from the requirements, the STARDAMP method has been developed going through the following steps:

- Definition of physical quantities to be measured in laboratory tests
- Elaboration of measurement protocols
- Validation of laboratory tests in round-robin tests separately for wheel dampers and for rail dampers
- Development of the STARDAMP software
- Validation of the full STARDAMP method
- Development of guidelines for testing the mechanical integrity and durability of dampers
- Definition of a measurement protocol for efficient field testing
- User-friendly documentation

The method developed for predicting the acoustic performance of rail dampers is based on the assumption that the track decay rate (TDR) of a damped track can be predicted by summing up the measured TDR of a real track and the decay rate measured on a short

section of rail equipped with dampers and freely suspended. Measurement protocols for the “short rail method” have been defined. A round robin test was organized, where different partners tested individually the method in their laboratory with equivalent dampers.

For assessing the acoustic performance of wheel dampers a method has been developed in STARDAMP, where a modal analysis of a bare wheel based on finite-elements calculations is supplemented by an experimental modal analysis of the wheel with damper. The latter provides the damping ratios of the eigenmodes, which are hardly predictable in FE simulations. For easy input of modal parameters into the STARDAMP software a dedicated ‘mp-editor’ has been developed. As in the case of the rail dampers the full procedure has been tested in a round robin test, where wheel sets have been circulated among partners.

As dampers in service are subjected to a number of environmental conditions that cause ageing of their components and might thus decrease the effectiveness of the noise reduction, test procedures for rail dampers and wheel dampers to simulate the long term impact of environmental conditions (ageing) and of varying load cycles caused by the passing of trains (mechanical integrity) have been developed.

The STARDAMP software is documented in manuals and guidelines for the application of the full STARDAMP method have been written.

Close cooperation has been established with the Swiss railways SBB. An extensive measurement campaign has been performed by SBB within the frame of a national project in order to gather information on the network-wide track decay rates with the final target to estimate the potential effect of rail dampers.

II. Detailed description of the project and the obtained results

Content

1. Introduction

This document summarizes the results obtained during the course of the Deufrako research project STARDAMP, which has been co-financed by BMWi, PREDIT, DGCIS, OSEO, and Région Nord Pas de Calais. It gives an overview of the targets and the organization of the project and key results are presented. The full documentation of the project results, which includes measurement protocols, measurement reports, software manuals and guidelines is contained in 29 reports, which are listed in Section 12. The STARDAMP reports are referenced by [SRnn] in order to distinguish STARDAMP results from other work, referenced in this report.

2. Motivation

Rolling noise is the main source of railway noise in the velocity range from 30km/h to 300km/h. Roughness of the surfaces of rail and wheel cause time-dependent contact forces in the rail-wheel contact, which excite vibrations of rail, wheel and sleeper, leading to the radiation of sound waves.

Dampers for rail and wheel have been developed by several manufacturers, partially within the framework of R&D projects both on national level in France and in Germany and on EU level (e.g. Silent Freight, Silent track, SILENCE, LZarG). Examples for different damper types can be seen in Fig. 1. They have proven to be an efficient way of reducing rolling noise from rail traffic.

The development continues, new products come to the market, already established products will be optimized for special applications (e.g. rail dampers for tracks with stiff/soft rail pad, curves, slab track, steel bridges, ...). Wheel dampers for different types of rolling stock are under development. In particular there is a high demand for wheel dampers for freight wagons with block brakes, which can withstand the high temperatures the wheel is subjected to during the braking process.

Fig. 1: *Examples for dampers for rail and wheel, which have been developed by partners of STARDAMP and are already available on the market.*

Prior to the STARDAMP project, the situation was characterized by the lack of generally agreed physical quantities and well-defined measurement protocols for the assessment of the performance of a damper. The consequence was that the performances of different products were hardly comparable and an objective evaluation procedure for their mitigation effect was not available. The assessment usually had to be carried out as field test with real trains on real tracks, making such tests extremely **costly and time consuming**. Especially to the need of applying for temporarily approval for the use of prototypes in real trains/on real tracks required both effort and time. This lack of simple and reliable assessment methods considerably slows down the process of technological innovation and hinders the optimum application by end-users.

Another issue that could impede the development and usage of dampers is the ambiguous situation concerning legislation. In Germany wheel dampers are included in the noise legislation with an overall reduction of 4 dB, whereas rail dampers are still in the process of being included in the national noise legislation.

In France wheel dampers are still neither in the process of testing under real conditions nor in the process of a first homologation.

Rail dampers have been homologated in France in 2004 with the Infrastructure Manager RFF. Two types of rail dampers are homologated and can be applied on a track. A complete but complex procedure of evaluating the effect of dampers for a specific track has been developed and proposed. This homologation has required five years before it was accepted. Beyond that the few number of applications since 2005 shows that it needs a more comprehensive methodology to predict and assess the performance of rail and wheel dampers for a wide variety of applications.

From an economic point of view, the added value of the STARDAMP project can be quantified for two standard situations:

1. Today, the efficiency of wheel dampers is measured in field tests comparing the noise emitted by standard wagons with the noise emitted by the wagons fitted with wheel dampers. Such a test is expensive and the minimum duration including obtaining all the necessary approvals is about 6 months. The STARDAMP methodology will decrease costs and time considerably.

2. When rolling stock operators buy new rolling stock or modify existing vehicles, this must comply with the TSI specifications which limit the noise at source. The compliance with these more and more stringent limits needs to put in place, by the train manufacturer, solutions for noise reduction like wheel dampers. Reducing the costs of these systems and their standardization is a real challenge. The solution of wheel dampers, easy to implement on existing rolling stock, is particulary interesting. Therefore, SNCF is directly confronted with the choice and approval of a product for which the characteristics and the performances differ between suppliers. Otherwise, this product must be compatible and approved throughout the European rail system.

The central target of STARDAMP was to provide a comprehensive methodology for prediction and assessment of the performance of rail and wheel dampers that could be applied by all stakeholders in the field of damping technologies for rail and wheel. The key part of the STARDAMP method shall consist in a combination of laboratory tests and computer simulation based on the TWINS software. For predicting rolling noise, the “Track Wheel Interaction Noise Software” TWINS is a powerful tool for predicting rolling noise. It enables the simulation of the noise radiation from wheel and track and the assessment of the influence of single parameters. However, successful use of TWINS requires long experience and a strong scientific background of the user. This is usually not available in the target group for the STARDAMP results. Therefore an adaptation (‘down-sizing’) of TWINS to the specific needs of the STARDAMP method and to the needs of the potential end-users had to be done within the STARDAMP project.

3. Project Organization

3.1 Project structure

This section gives an overview of the project structure and the key topics of the work packages (WP).

STARDAMP was organized in three work packages (WP) and ten tasks (see Fig. 2). The overall coordination was performed jointly by SNCF and DB.

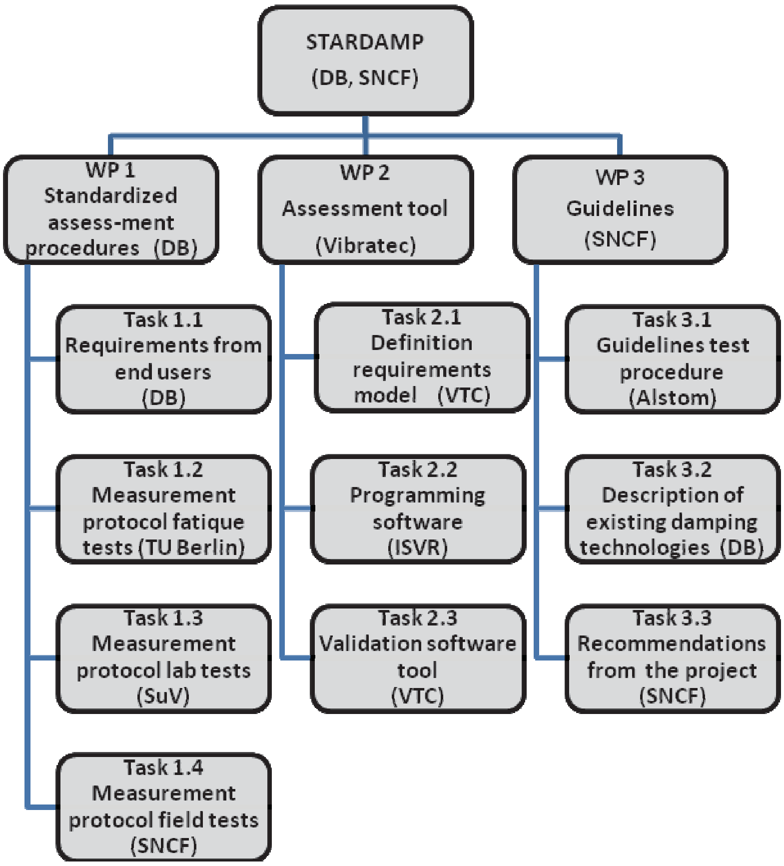


Fig. 2: *Project structure of STARDAMP. Work package leader and task leader are listed in brackets.*

In **WP1** general procedures for the generation of those quantities, which are necessary for the objective assessment of a damper, were developed. This was done on the basis of requirements specified by the potential end users. The different configurations (wheels, track types etc.) to be covered by the STARDAMP method and tool were specified. Measurement protocols for laboratory tests and field tests were elaborated.

The central part of the project was **WP2**, where software tools have been developed to calculate the noise reduction obtained by a damping system for a particular scenario with standard measurement data (wheel modal damping, track decay rates etc.) as input.

The STARDAMP approach was to use the TWINS model as basis and to downsize it according to the specified requirements defined in WP1. This also included a user-friendly interface, which allows using the software without the necessity of learning TWINS. An extensive validation of the software was performed.

The results of **WP3** provided the direct link from the project results to the end-users. Guidelines describing the STARDAMP methodology for the assessment of a damping system were written.

From these works, two main outputs are delivered :

- the STARDAMP software with user guide which will be distributed by Vibratec

- the public guidelines to assess a damping system (acoustic and reliability assessments)

The STARDAMP project started in October 2010 with a kick-off meeting in Oberhausen, Germany on 9th of November 2010. It was organized by the project partner GHH Radsatz. The German part of STARDAMP was intended to finish on 30th Sept 2012. In June 2012 it was decided by the German partners in the STARDAMP consortium to ask for an extension until 31st of December 2012, which was approved by the Bundesministerium für Wirtschaft and TÜV Rheinland.

In a similar way, the French part was intended to finish on 30th June 2012. In May 2012, the French partners in the STARDAMP consortium had the agreement from the French funders to extend the project until 30th of April 2013.

The following meetings were organized on project level:

1st project meeting:	9th November 2010 in Oberhausen (hosted by GHH Radsatz)
2nd project meeting:	25th / 26th January 2011 in Lyon (hosted by Vibratec)
3nd project meeting:	7th / 8th June 2011 in Frankfurt (hosted by DB)
4nd project meeting:	15th/16th November 2011 in Paris (hosted by SNCF)
5nd project meeting:	6th March 2012 in Oberhausen (hosted by GHH Radsatz)
6nd project meeting:	18th July 2012 in Paris (hosted by SNCF)
7nd project meeting:	17th – 19th October 2012 in Munich (hosted by DB)
8nd project meeting:	11th / 12th December 2012 in Oberhausen (hosted by GHH Radsatz)

For the dissemination of the STARDAMP results, a public workshop was organized on 18th of October 2012 in Munich, Germany (see Sec. 14).

3.2 The STARDAMP Consortium

STARDAMP is a Franco-German research project within the Deufrako framework. The project is coordinated by DB (the German coordinator of the STARDAMP project) and SNCF (the French coordinator).

The consortium brought together leading manufacturers of rail and wheel dampers with the end-users of damping technologies (railway operators, suppliers of rolling stock and bogies/wheel sets). It also included well-reputed research institutes and engineering companies in the field of acoustic simulation and acoustic measurements:

- **Alstom Transportation**
- **Deutsche Bahn AG** (DB Systemtechnik)
- **Gutehoffnungshütte Radsatz GmbH** (GHH Radsatz)
- **Valdunes SAS**
- **SNCF** (Innovation and research department)
- **Schrey und Veit**
- **Tata Steel**
- **TU Berlin** (Section Track and Railway Operations of the Berlin Institute of Technology)
- **Vibratec**

Potential end users are well represented in the consortium and ensure that the STARDAMP methodology is applicable, practice oriented and used after validation.

3.3 Funding

STARDAMP was a Franco-German research project within the DEUFRAKO framework.

The German part of STARDAMP has been co-financed by BMWi.

The French part has been co-financed by PREDIT, DGCIS, OSEO innovation, Fonds Européen de Développement Régional FEDER 2007-2013, Région Nord Pas de Calais, Conseil régional du Nord Pas de Calais. I-TRANS has labeled, supported the project and the French partners in their approach.

This funding is greatfully acknowledged by the STARDAMP consortium. It was a precondition for the successful execution of the project.

German Funding Number: FKZ 19U10012 A-D
French Funding Number: FUI 092906631

German partners are financed in the frame of the research programme 'Mobilität and Verkehrstechnologien' which is piloted by the ministry of economics and technology. French partners are financed in the frame of the 'fonds unique interministeriel (FUI)' which is piloted by the ministry of 'Redressement Productif'.

3.4 Cooperation with the University of Southampton (ISVR)

The Institute of Sound and Vibration Research (University of Southampton) was included in the STARDAMP project as subcontractor. A research and development agreement was signed among the STARDAMP consortium and ISVR. The costs for the subcontract were shared among the STARDAMP partners based on their relative contribution to the overall STARDAMP budget.

The main task of ISVR was to develop a software tool to enable predictions of the effect of wheel and rail dampers, consisting of

- The STARDAMP tool – used for rolling noise predictions to assess wheel and/or rail dampers.
- The modal parameter file editor – used to create input files for the STARDAMP tool for damped and undamped wheels from a finite element model of the wheel section.

3.5 Cooperation with SBB

Close cooperation has been established with the Swiss railways SBB. Within the frame of a national Swiss project, SBB carried out an extensive network-wide measurement campaign in order to estimate the potential effect of rail dampers. Regular exchange of information was established between STARDAMP and the coordinator of the SBB activities. STARDAMP procedures were applied and tested as far as they were available by SBB.

4. Fundamental physical phenomena relevant for the generation of rolling noise

A brief review of the main physical phenomena accounting for wheel/rail rolling noise is useful to obtain a better understanding of the physical parameters of interest for the assessment of the efficiency of wheel dampers solutions. The physical steps involved in rolling noise emission are explained in Fig. 3.

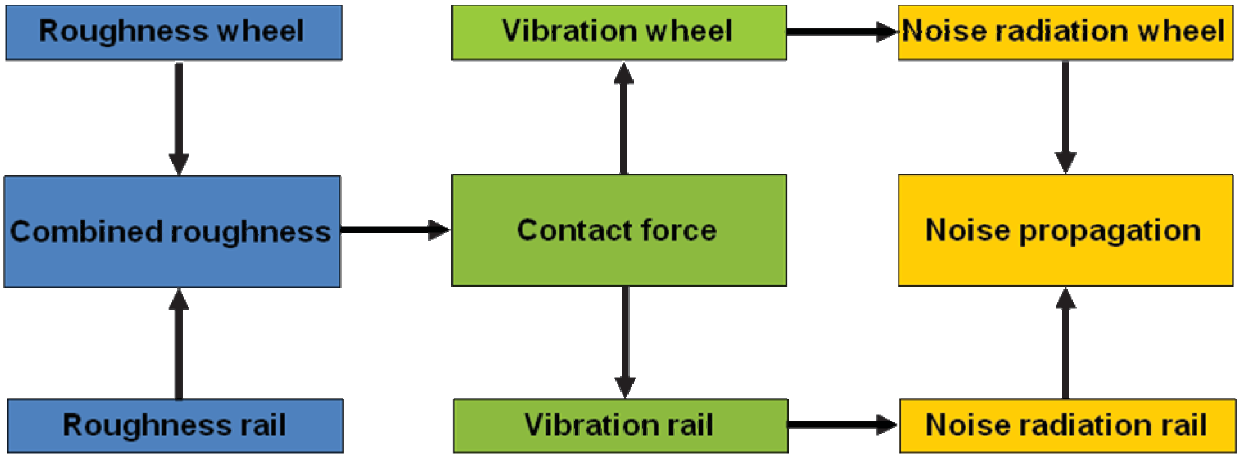


Fig. 3: *Principle of rolling noise generation*

The focus of STARDAMP is on the effect of dampers to reduce the vibration of rail and wheel.

The combined roughness of rail and wheel imposes relative displacements in the rail-wheel contact. These displacements are taken up by the rail, the wheel or by a deformation in the contact. At frequencies below about 1500 Hz, the rail has the highest receptance of these three and therefore the rail responds to the excitation and is the dominant noise source in this frequency range. Above 1500 Hz, the wheel typically exhibits a series of resonance frequencies at which the radial receptance of the wheel exceeds that of both the rail and the contact. Hence, the wheel is the dominant noise source at higher frequencies.

4.1 Decay rates as indicator for the noise radiated by the rail

When excited by the wheel-rail interaction waves of vibration start to propagate along the rail and will approximately decay exponentially with increasing distance from the excitation point.

Assuming the rail to be an infinite elastically supported beam (see Fig. 4) the radiated sound power W is given by:

$$W = A \, \sigma \, \rho_0 \, c_0 \, \langle v^2 \rangle \qquad (1)$$

Where <v^2> denotes the spatially averaged mean square surface-normal velocity of vibration, σ is the radiation ratio and $\rho_0 c_0$ the specific acoustic impedance of air. A is the surface area of the radiator. Under the assumption that the velocity field in the rail can be described by one wave (separately for the lateral and the vertical direction) decaying with a constant β, it can be shown (see [4]) that W is inversely proportional to β:

$$W \sim 1/\beta \tag{2}$$

β can be converted into the 'track decay rate' (TDR) Δ according to

$$\Delta = 8.686\,\beta \text{ dB/m} \tag{3}$$

Lateral and vertical bending waves of the rail have different coefficients β, which means that also TDRs differ between lateral and vertical rail vibration.

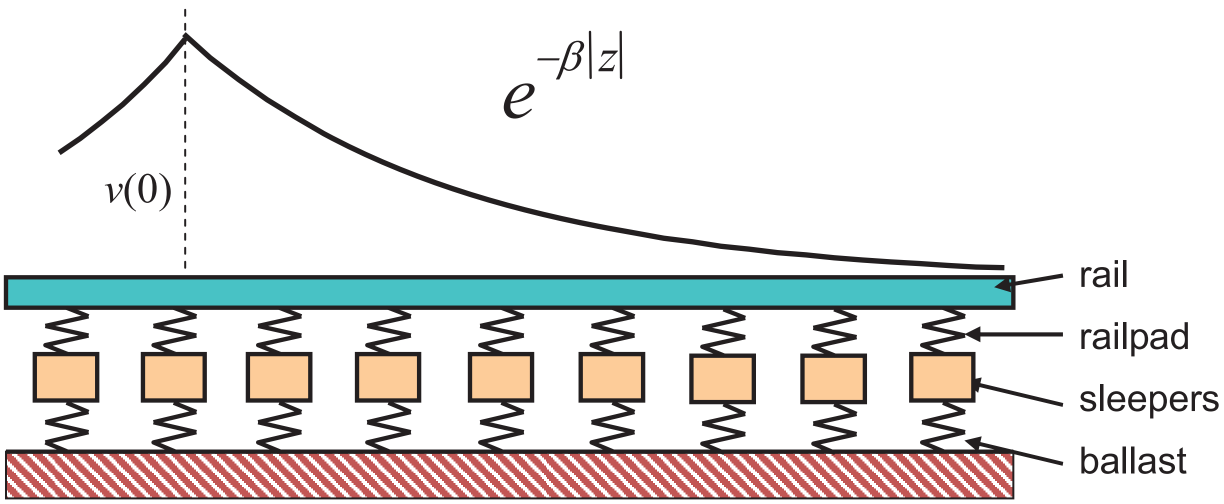


Fig. 4: *Schematic representation of the rail as elastically supported beam. Bending waves decay exponentially. The coefficient β is proportional to the track decay rate (TDR).*

When the rail is not fitted with dampers, all the forms of energy dissipation and blockage of vibration transmission.is accounted for in the track decay rate analysis. Losses occur in fact in the resilient fastening system and in terms of energy transmission to sleepers, ballast and ground. Moreover, the way in which the rail is attached to the rest of the track blocks the vibrations at low frequencies. The purpose of a damper is to increase the TDR. If the rail is highly damped, the waves decay quickly causing only a short section of rail to vibrate. The shorter the section of rail vibrating, the lower the sound emission. Consequently, by increasing the damping of the track, the rolling noise level can be reduced.

In the following, the term 'TDR' will be used for the decay rate of track with rails that are supported by a rail fastening system to sleepers (e.g. real tracks). The track can either be undamped or equipped with rail dampers. The STARDAMP method introduces a method of assessing dampers mounted to a section of freely suspended rail (see Sec. 6.2). In order to distinguish the TDR (which is strongly influenced by the properties of the fastening system) from the decay rate measured in a free rail, the later will simply be called 'decay rate' (DR) or damper-decay rate.

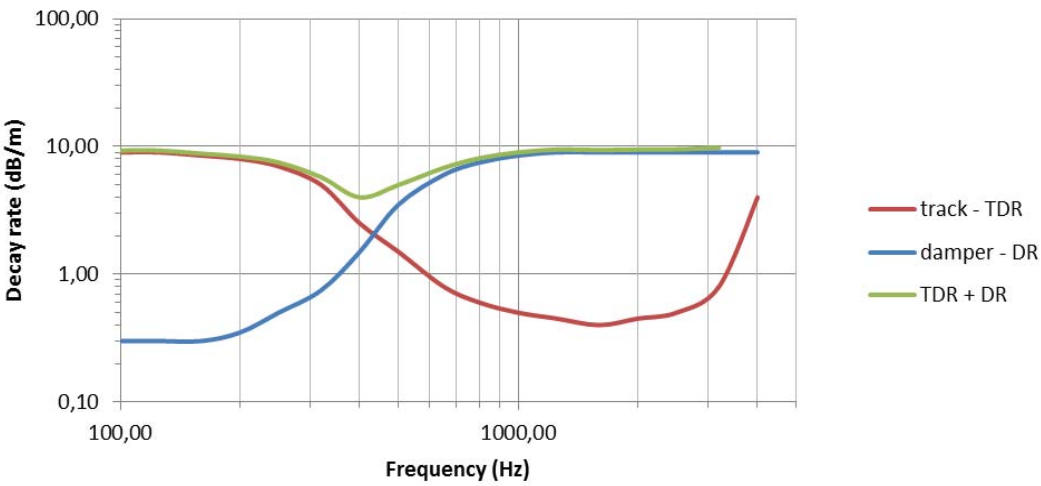


Fig. 5: *Examples for track decay rate (TDR), damper decay rate (DR) and the sum of both.*

Fig. 5 (red curve) gives an example for a typical TDR. It is high at low frequencies with values close to 10 dB/m. In the frequency range around 500 Hz, it drops to values below 1 dB/m. This is strongly dependent on the rail fastening system. The stiffer the rail fastening system is, the higher the frequency range, where this drop occurs. The blue curve is an example for the decay rate, which has been measured for a freely suspended rail with dampers attached (see Sec. 6.2). The free rail itself has only very low damping and the increase of the DR beyond 200 Hz is solely due to the additional damping imposed by the rail damper.

The blue curve reflects the specific properties of a rail damper. The STARDAMP method is based on the assumption that the TDR of a track equipped with rail dampers can be predicted by measuring the TDR of the track without damper and the DR of the damper. The TDR of the damped track is given by the sum of both:

$$\text{TDR (damped track)} = \text{TDR (undamped track)} + \text{DR(damper)} \tag{4}$$

From eq. (2) the reduction of radiated sound power of the damped track in comparison with the undamped track can be calculated. An efficient rail damper should have a DR such that the sum of TDR and DR (green curve in Fig. 5) has values close to 10 dB/m over the whole frequency range.

4.2 Vibration response of the wheel

A railway wheel is a lightly damped structure, which can be characterised by its natural modes of vibration. For an axis-symmetric wheel, these modes can be described by two identifiers (also illustrated in Fig. 6):

- The number of nodal diameters n,
- The form of the cross-section deformation:
  - o axial modes with m nodal circles, denoted (mLn)
  - o radial modes, denoted (Rn)

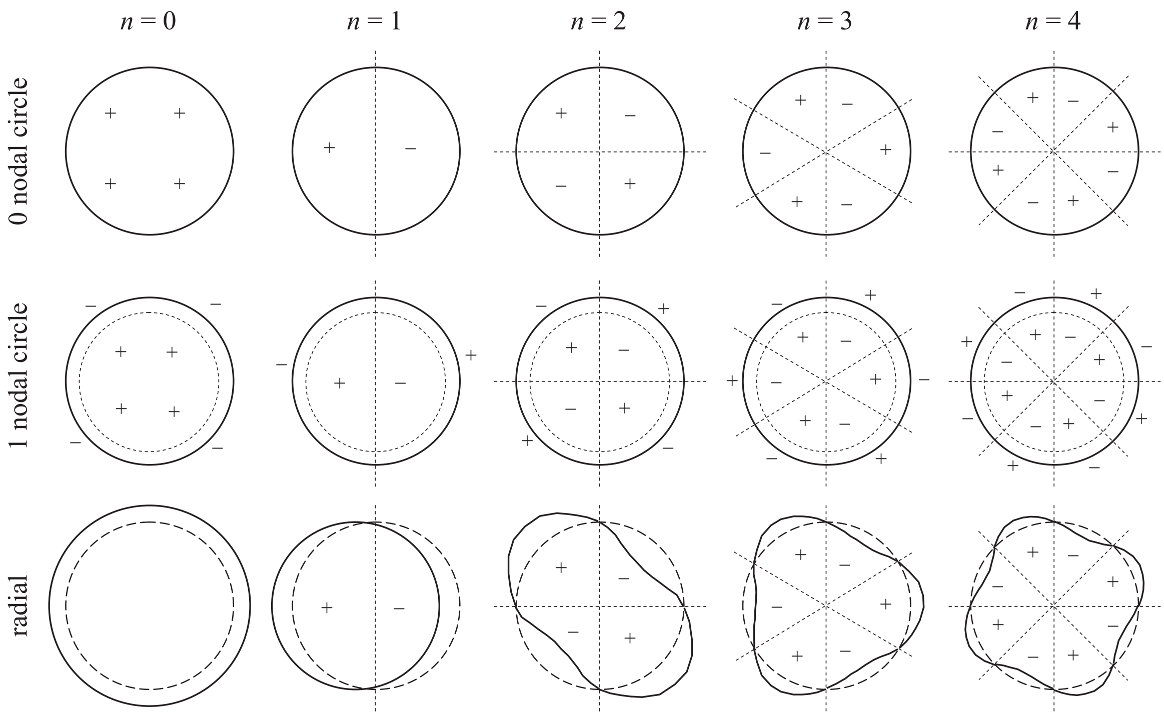


Fig. 6: *Classification of wheel modes according to their nodal circles and nodal diameters.*

The modes that generally contribute most to rolling noise are the modes of type 1Ln and Rn. Modes of type 2Ln play a minor role, in part because of their high natural frequencies. Except around these modes, the vibration response of the wheel can nearly be neglected. As a rule of thumb, the contribution of the wheel to total noise emission becomes important above the natural frequency of mode R2, which lies generally between 1 kHz and 2 kHz depending on the wheel geometry[1].

On the left hand side of Fig. 7 measured wheel receptances (displacement divided by excitation force) are plotted for a bare wheel as well as the same wheel equipped with two different dampers. The modes that have an important contribution to rolling noise are the ones leading to the receptance peaks above 1.8 kHz. The ring damper (plotted in blue) clearly reduces the wheel receptance in respect to the bare wheel, while the frequency shift of the modes remains very small. The plate damper (red curve) is a much heavier but also more effective device. Clearly, the modes initially indentified on the bare wheel become more

[1] For standard diameter freight wheels with a highly curved web (low stress design) mode R2 can be as low as 1 kHz, while wheels with a straight web and/or lower diameter lead to higher natural frequencies of the radial modes.

difficult to find for the plate damper by simply comparing both receptance plots. This is due to the very high damping introduced, but also due to a more important frequency shift of the modes.

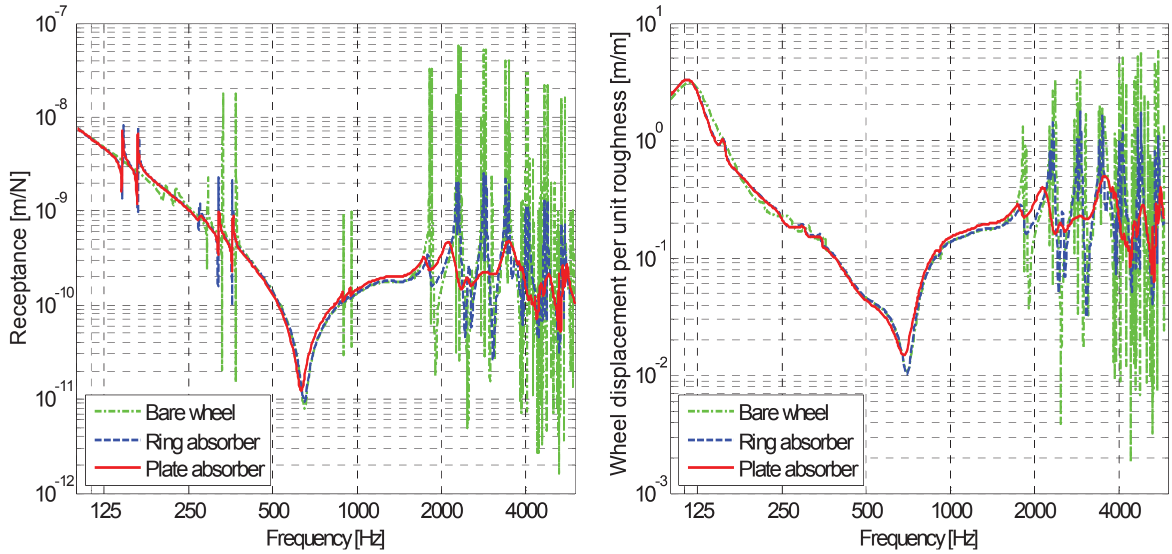


Fig. 7: *Wheel receptance (left) and wheel displacement per unit roughness (right) of a bare wheel (green curve), and the same wheel equipped with two with different kinds of damper (blue and red curve).*

Experiments show that the difference in modal damping between different monobloc wheels (without dampers) is generally very low. On the other hand, modal damping is highly dependent on the number of nodal diameters.

For wheels without dampers it can indeed be defined as a function of the number of nodal diameters. For rolling noise computations with bare wheels the measurement of modal damping is therefore not necessary but the following default values can be used:

- $\zeta_i = c/c_c = 0.001$ for $n_i = 0$ (medium damping due to coupling with extension or torsion of the axle)
- $\zeta_i = c/c_c = 0.01$ for $n_i = 1$ (high damping due to coupling with bending of the axle)
- $\zeta_i = c/c_c = 0.0001$ for $n_i \geq 2$ (low coupling)

Mode R1 represents a special case because its damping is even higher than for the remaining modes of order n=1. Indeed, mode R1 can generally not be found experimentally due to its high coupling with the axle. A damping of $\zeta_i = c/c_c = 0.2$ can be assumed for this mode.

Note that for bare wheels the additional damping due to interaction with the rail ("damping when rolling") is higher than the wheel damping itself. Therefore the use of these default values is sufficient (no need for experimental updating of the modal basis for bare wheels).

The added damping from the dampers has indeed to be compared to the "damping when rolling" and not to the free wheel damping, when conclusions about rolling noise are to be drawn. A comparison between the left hand plot and the right hand plot in Fig. 7 illustrates this behaviour: A given wheel damper may reduce the wheel receptance around the wheel natural frequencies by a factor greater than 10 (compare "Bare wheel" and "Ring damper" spectra on the left hand plot). However, calculation of the wheel displacement per unit roughness[2] (plotted on the right hand side) shows a reduction by a factor of around 2 to 3 only. The reason is that both bare wheel and “damped wheel” benefit to the same extent from the energy loss generated by the wheel-rail contact. In order to provide a net benefit, a wheel damper therefore has to provide damping which is higher than the “damping when rolling”. As detailed later in this report the STARDAMP procedure accounts for the increase of modal damping due to wheel dampers. “Dampers” that mainly act through shielding of wheel radiation are not in the scope of the project.

[2] The wheel displacement due to roughness excitation is computed using TWINS. The used wheel model is extracted from a FE calculation, updated with measured modal damping and natural frequencies. The interaction between wheel and rail is accounted for analytically in TWINS.

5. Strategy of STARDAMP

5.1 General

The aim of STARDAMP was to develop a practice-orientated methodology to predict the performance of rail and wheel dampers under realistic operating conditions by a combination of laboratory tests and computer simulation and, hence, avoid expensive and time consuming field testing.

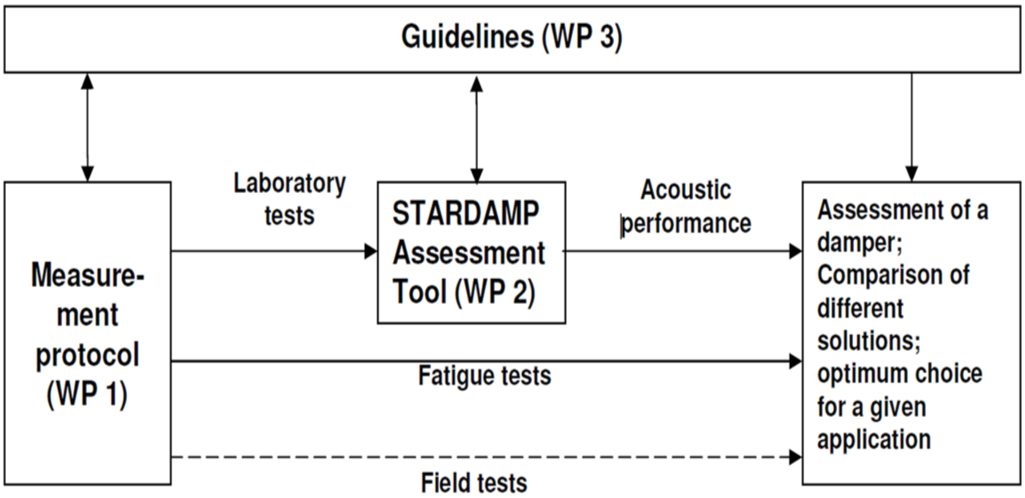


Fig. 8 : *General representation of the STARDAMP methodology*

The general procedure is outlined in Fig. 8. A set of measurements shall be performed in the laboratory according to clearly specified measurement protocols. These data shall be used in a dedicated software tool ('STARDAMP assessment tool').

In addition to the acoustic performance also durability and mechanical integrity are important issues as dampers in service are subjected to a number of environmental conditions that cause aging of their components. Mechanical integrity should also be assessed to ensure that the dampers itself and particularly their fastening systems guarantee a proper installation and functionality.

Hence, the key elements of the STARDAMP methodology are:

- Definition of physical quantities to be measured in laboratory tests
- Elaboration of measurement protocols
- Validation of laboratory tests in round-robin tests separately for wheel and rail dampers
- Development of the software tool
- Validation of the full STARDAMP method

- Development of guidelines for testing the mechanical integrity and durability of dampers
- Measurement protocol for efficient field testing
- User-friendly documentation

This rigorous methodology allows to establish the STARDAMP software tool and measurement protocols to assess the damper performance which are the key goals of this project. They will be used by each partner and then by manufacturers of dampers for rail and wheel, owners of rolling stock, railway operators, infrastructure managers, rolling stock manufacturers, planners, consultants and engineering companies for railways etc.. Software and measurement protocols will be tools for the damper developments and will allow for better knowledge on the optimum application of dampers. Considerable savings in development times and development costs may be expected.

The STARDAMP tool implements TWINS-like predictions of rolling noise in a user-friendly way. Compared with TWINS the number of options is limited in order to allow access to non-expert users. It consists of

- a simulation tool, the STARDAMP software – used to make rolling noise predictions to assess wheel or rail dampers.
- modal parameter file editor – used to create input files for the STARDAMP software for damped and undamped wheels from a finite element model of the wheel section.

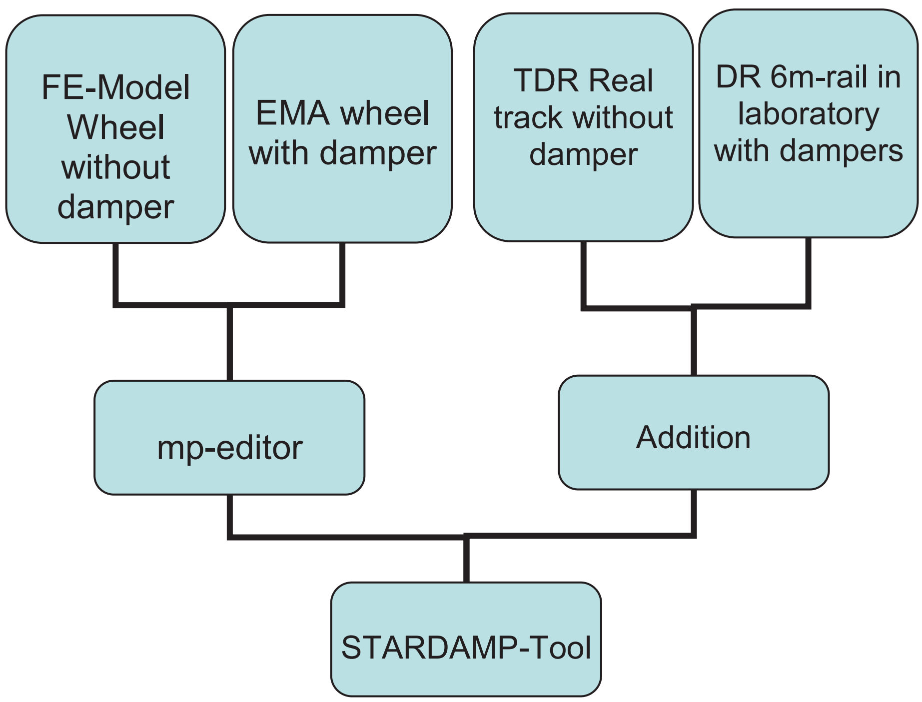


Fig. 9: *General scheme of the STARDAMP software including the necessary input data*

As shown in Fig. 9 the STARDAMP software requires as input modal parameters for the wheel and decay rates for the rail both with and without dampers. The ‘mp-editor’ provides an easy-to-handle link between the modal analysis and the STARDAMP software. It has been developed within STARDAMP and is described in detail in Sec. 5.4.

5.2 Requirements of the end users

As a first step, the scope and requirements for the STARDAMP tool were defined by the consortium. Application of the STARDAMP assessment procedure shall not be limited to products already available on the market or as prototype but it shall be applicable also to future products. In the case of rail dampers, it was necessary to better define, what a ‘rail damper’ is, as this term is commonly used also for products with completely different mechanisms. It was decided that throughout the project a ‘rail damper’ shall be defined as a device, which substantially increases the track decay rate when attached to the rail. The STARDAMP methodology shall be limited to such systems. This means that products, which e.g. reduce the noise by shielding the rail are not taken into account even though such products are often called ‘rail damper’.

The requirements were defined on the basis of a questionnaire sent out to the partners in January 2011. It was decided that the following scenarios shall be covered for the performance of rail dampers:

- Plain track without switches.
- Ballasted track and slab track
- Rail types: UIC60, UIC54, S54, S49
- Wide range of rail pads stiffness ranging from very soft pads typical for high-speed track and slab track up to very stiff rail pads like Zw 687a
- All kinds of traffic (high-speed, regional trains, freight). Each type of traffic shall be taken into account in the model by ‘pushing a button’.
- Speed range: 50 km/h – 320 km/h.

For wheel dampers it is more difficult to define general scenarios to be covered as there are many different wheel types in use and wheel dampers are usually designed and optimized for a specific wheel. The following requirements for the treatment of wheel dampers within STARDAMP were defined:

- For laboratory testing, the assembly of the test wheel and its associated damper mounted to it has to be considered
- For wheel modelling, a set of representative wheels (modal parameter file) has to be defined and the wheel modal damping has to be identified from experiments. The modal damping of the modes in the modal parameter file shall be modified accordingly.

- Wheel types: Mono-bloc wheels

Furthermore it was emphasized by the STARDAMP consortium that the user interface of the software shall be well structured and self-explanatory to avoid incorrect inputs. Sound pressure levels shall be given as TEL, LAeq,Tp , sumlevel, octave bands (63 Hz, 125 Hz, 250 Hz, 500 Hz, 1000 Hz, 2000 Hz, 4000 Hz, 8000 Hz; this is required in German legislation) and 3rd octave bands.

5.3 Description of existing damping technologies and products used in STARDAMP

Rail dampers are usually tuned absorbers where vibrational energy is dissipated into an elastomer component. Most of the rail dampers currently in use consist of a combination of elastomeric and metal materials but also designs based on pure elastomers are available on the market. In order to be effective over a wide frequency range, their internal damping must be sufficiently high. Designs including metal components – either as elements fully embedded in the elastomer or as sandwich layers – have generally higher efficiency and also cover a wider frequency range, while pure elastomer dampers are less costly.

The concepts for wheel dampers show a larger variety than those for rail dampers. Different variants were developed and tested over years: steel-shot absorbers, shrink collars, damping friction rings, constrained layer dampers on the wheel disc, foaming of the wheel disc with plastic material. For a detailed overview see ref. [13]. Damping rings and plate dampers are applied both in commuter and mainline passenger trains. Plate dampers consist of bi-layers of steel plates with a visco-elastic damping material in between. As both plates oscillate in phase, the damping material is subjected to shear stress. By increasing the number of plates the damper can be tuned to those frequencies of the wheel that have the highest contribution to the noise radiation. Wheel dampers can also be constructed as block absorbers with sandwich layers of steel elements separated by elastomeric material. Another concept for wheel dampers uses friction between two bodies. The major difference to material damping is that friction damping works at all frequencies. Friction damping can be achieved by mounting specific elements at the wheel with a frictional connection like e.g. rings or plates in friction contact

The STARDAMP procedure as outlined in the previous section is not limited to certain products. It is the intention to keep the method as general as possible to cover a wide range of products – be it existing products or future developments. Nevertheless, is was necessary to use a limited set of dampers for testing the various procedures to be developed in STARDAMP. These products were provided by the partners GHH Radsatz, Schrey und Veit, Tata Steel and Valdunes. They are described in the following.

5.3.1 Wheel damper

Four different types of wheel dampers have been used within STARDAMP. These are described in the following.

Vicon RASA RSI manufactured by Schrey & Veit

Vicon RASA RSI is a block damper for wheels constructed as a horn formed arrangement of elastomeric and metal plate layers. The dampers are tuned to the operating frequency range by the manufacturer using metal plates of different thickness and length. They reduce radial and axial vibrations of the wheel when they are excited to vibrations. They were used in connection with the BA308 wheel.

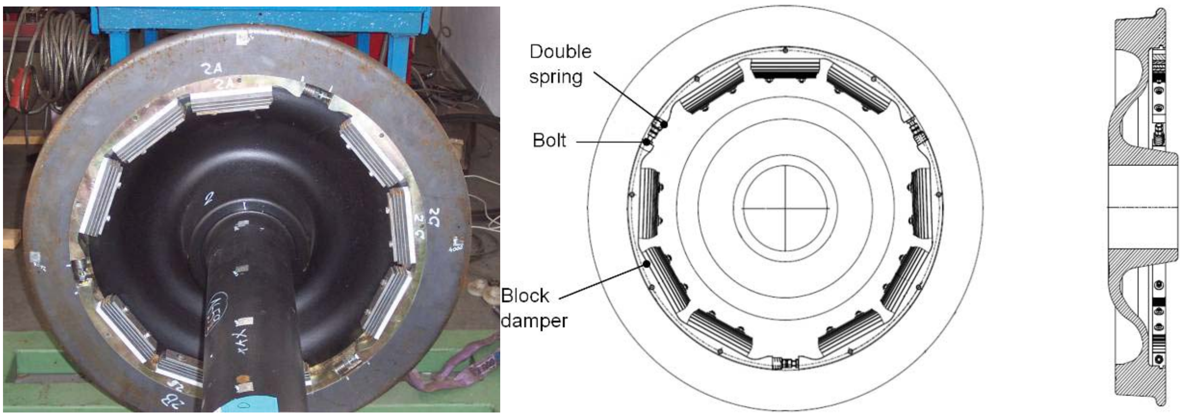


Fig. 10 : *Vicon RASA RSI damper attached to a BA308 wheel*

Plate damper manufactured by GHH Radsatz

The plate damper consists of bi-layers of slotted metal plates separated by a thin elastomeric layer. The wheel damper is tuned by the manufacturer to the desired frequency range by slotting the metal plates. Vibrations transmitted from the wheel into the plate damper lead to bending oscillations of the metal plates putting the thin elastomeric layer between them under shear stress. Large damping is possible if the elastomeric layer is very thin. The damper reduces vibrations in radial and axial direction.

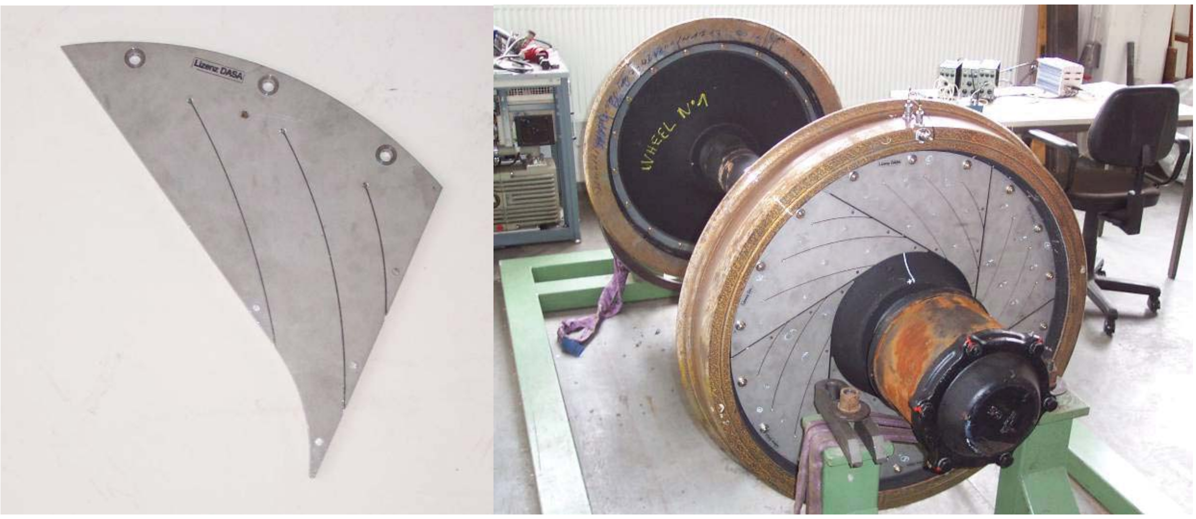


Fig. 11: *Metal plate damper*

DAAVAC damper manufactured by Pinta Enac.

The DAAVAC damper is fitted into a groove on the wheel flange. It consists of two metal rings formed of spring steel with elastomeric material between them.

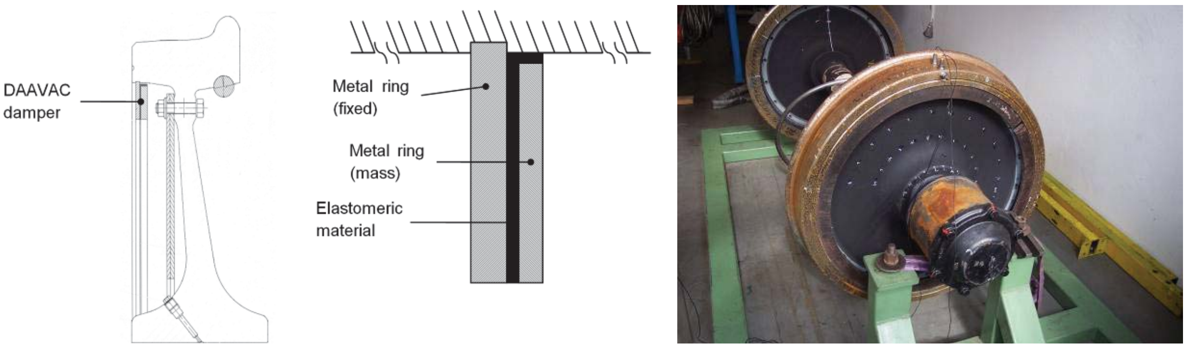


Fig. 12: *DAAVAC damper*

VLN ring damper manufactured by Valdunes.

The VLN ring damper is a friction damper. It is made of stainless steel and fitted into a coated groove on the wheel flange.

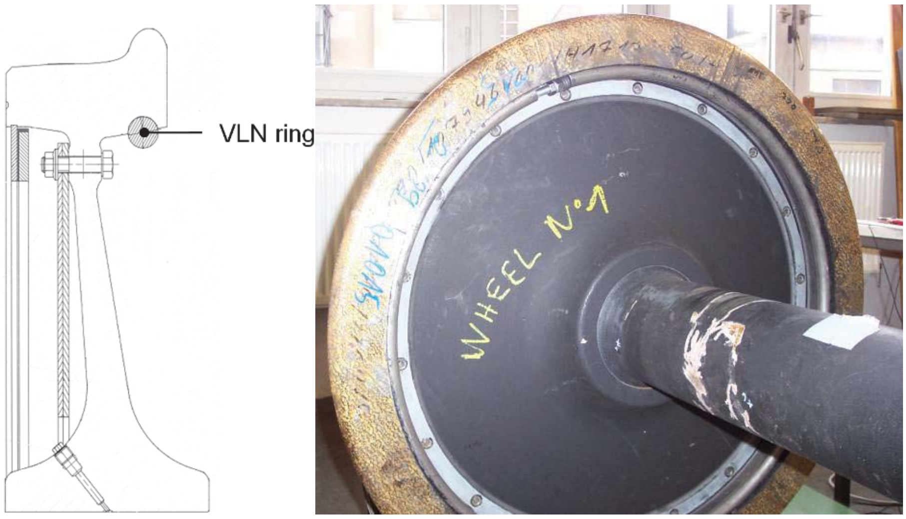


Fig. 13: *VLN ring damper*

5.3.2 Rail damper

Two different kinds of rail dampers were used within STARDAMP. They are described in the following.

Rail damper MK I and MK II manufactured by Schrey und Veit

This design is a block damper that consists of metal plates between layers of elastomeric material. The variation of the thickness of those plates and of the intermediate elastomeric layers allows tuning the damper to the desired frequency range.

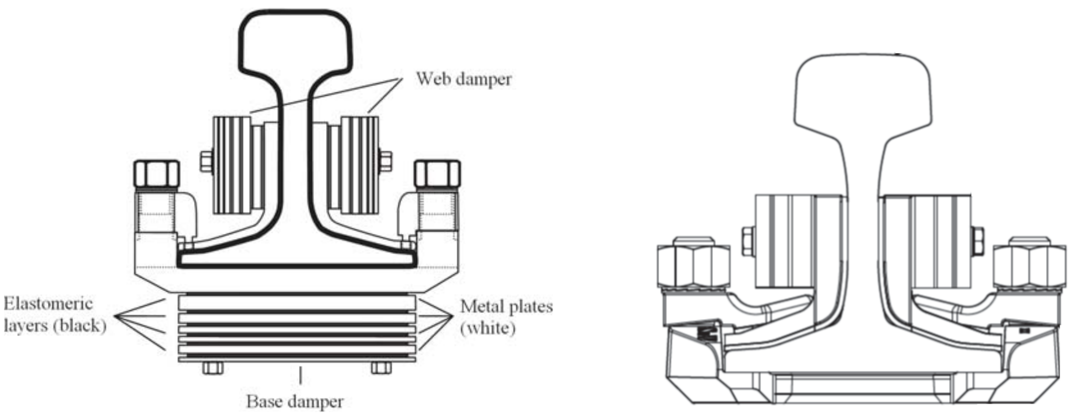


Fig. 14: *Cross section of rail damper MK I (left) and MK II (right)*

This damper is a combination of rail web dampers and a rail base damper that is friction-locked with rail clamps to the rail.

Rail damper manufactured by Tata Steel

This damper consists of metal strips that are fully embedded into elastomeric material, so the tuning is made by varying the weight of the seismic masses within the elastomeric material. These dampers come in pairs and are fastened to the rail with clips as well as glued to it with a contact paste to improve the coupling to the rail.

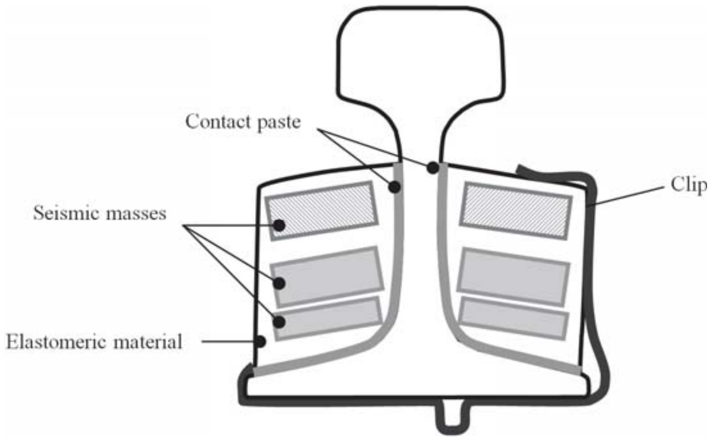


Fig. 15: *Cross section of Tata Steel rail damper*

5.3.3 Wheels

Two different kinds of wheels and wheelsets were used in STARDAMP for testing of wheel dampers.

- Single wheel BA308 and wheelset BA308 manufactured by GHH Radsatz. This wheel type is used for freight trains with a maximal wheelset load of 25t. The BA308 wheel set was developed within the LZarG project and equipped with Vicon RASA RSI dampers from Schrey&Veit. Field tests were performed for this combination in the LZarG project.
- Single wheel LK900 and wheelset LK900 manufactured by GHH Radsatz. This wheel type represents a typical design for passenger car wheels. It was developed within the STARDAMP project especially for the round robin tests and is not used in service ([SR27], [SR28]) The special design allows mounting three different kinds of dampers, each with its own interface. So the number of necessary wheelsets for the round robin test (see sec. 7.2) was reduced.

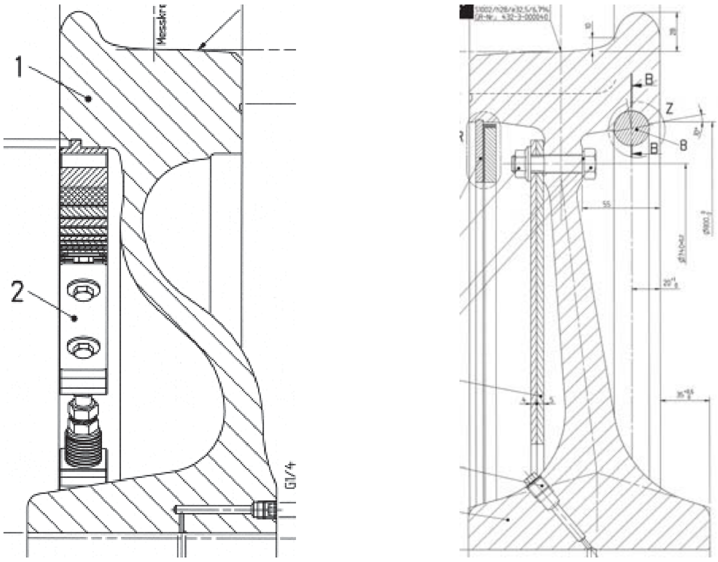


Fig. 16: *Drawings of the test wheels BA308 (left) and LK 900 (right)*

5.4 Output

Besides the full documentation of the STARDAMP results (which are in general not public), the main output from the STARDAMP project is

- the STARDAMP software, which includes a comprehensive documentation and a handbook [SR22]
- and the STARDAMP Guidelines [SR26] where the different tools and procedures developed in STARDAMP are described. For more details see Sec. 12 and 13.

6. Results for rail dampers

6.1 Round robin test for rail dampers

An objective of the STARDAMP project was to define methodologies to test rail dampers in the laboratory without the need for their installation in a track, with the attendant cost, time and variability implications. To these ends, two guideline methods were proposed to measure the decay rates of damped 'freely supported' rails [SR 9]:

(i) for long rails, by integrating decay rates derived from FRF's measured at intervals along the rail;

(ii) for short rails, at low frequency from the modal properties of the rail, and at high frequencies directly from point and transfer frequency response functions (FRFs) at either end of the rail.

Round-robin tests broadly following these guidelines were conducted by four of the STARDAMP partners (ISVR, Schrey&Veit, Tata Steel and TU Berlin) in order to validate and refine these procedures. Based on the results of the round-robin tests, a proposal for a test procedure with the short-rail method was developed [SR 10]. In the following section, a brief overview of the round-robin tests is given. The test procedure, which is finally proposed, is described in detail in Section 6.2. Detailed results for all test conditions are given in the respective round robin test report for each partner [SR 3], [SR 4], [SR5], [SR6].

6.1.1 Long rail test method

In the 'long rail' method, dampers are fitted to a freely suspended rail of length between 18 and 33 m. The analysis method is similar to that described in EN 15461:2008 [SR 1]. ISVR and Tata Steel measured the damped decay rates of dampers fitted to 'long' rails. Three types of dampers were tested using the method (Table 1).

Table 1: Long rail method: Dampers tested by each project partner and the respective test rail length.

		Project partner	
Manufacturer (type)	Report reference	ISVR	Tata Steel
Tata Steel	Tata Steel	32 m	18 m
Schrey&Veit (1)	S&VMK1	32 m	18 m
Schrey&Veit (2)	S&VMK2	32 m	

Measurements were conducted on 'freely' supported sections of UIC-60 rails, either 32 m long in the case of ISVR, or 18 m long in the case of Tata Steel. For both partners, the rails were supported on sections of hydraulic hose to achieve the 'freely' supported condition.

The 51 dampers used in the ISVR tests were attached at mid sleeper spans along the full length of the rail, except at two locations where rail welds prevented their attachment; the average spacing was 0.63 m. The dampers for the Tata Steel tests were installed at 0.6 m spacing along the full length of the rail.

Both ISVR and Tata Steel measured the vertical and lateral decay rates using a method similar to that described in EN 15461:2008 [1].

At ISVR, the rail used for the long rail tests was unclipped from a 32-m test track located within the University of Southampton Science Park. The measurements of the damped, freely supported rail were later compared to measurements on the test track. For this reason, a measurement grid was chosen that suited both measurement set-ups. The measurement grid was marked up starting at 10 sleeper spans (5.96 m) from the rail end. Measurements were made at ¼-sleeper intervals from this point up to the 16th sleeper span, then at mid-span positions 17, 18, 20, 22, 26, 30, 34, 38, 42 and 46, giving a total of 29 positions (Fig. 17).

For the Tata Steel measurements, a measurement grid with consistent 150 mm spacing between points was used.

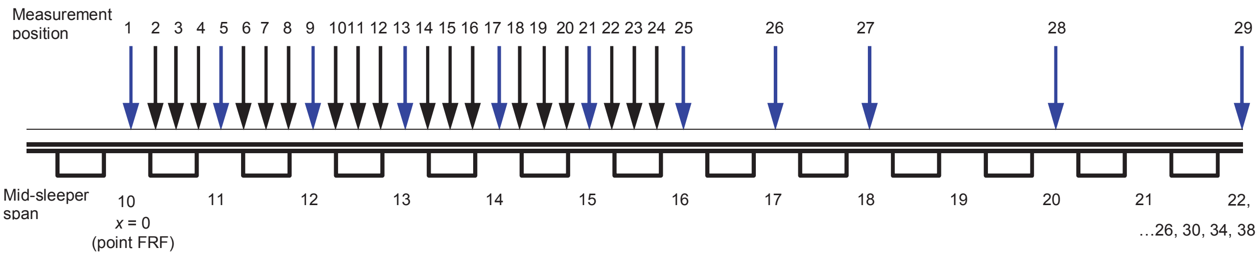


Fig. 17: *ISVR excitation positions on rail for measurement grid marked up 6.0 m from rail end*

The point FRF was derived for excitation at a position close to the accelerometer. Transfer FRFs were derived from excitation at each of the grid position. Subsequent analysis was conducted using mobility FRFs. Decay rates (DR) were calculated, in each ⅓ octave band using,

$$DR = \frac{4.343}{\sum_{x=0}^{x_{max}} \frac{|A(x_n)|^2}{|A(x_0)|^2} \Delta x_n} \qquad (5)$$

where, $A(x_0)$ and $A(x_n)$ are the point and transfer frequency-response functions (FRFs) respectively in each ⅓ octave band, x_n is the measurement position and Δx_n is the corresponding step size between adjacent points.

Additional long-rail measurements were made to give increased understanding of the method and causes of variability in the results. These are summarised in Table 2.

Table 2: Further long-rail tests conducted by project partners

Test	Partner
Effects of damper temperature	ISVR [3]
Effects of excitation position	ISVR [3]
Effects of accelerometer position	ISVR [3]
Effects of grid density	ISVR [3]
Effects of grid positions	ISVR [3]
Comparison of measured and predicted decay rates	ISVR [3]
Effects of rail pad temperature and unclipping/reclipping rail (test track)	ISVR [3]

6.1.2 Short rail test method

In the 'short rail' method, dampers are fitted to a freely suspended rail of length between 4 and 6 m. Two methods of analysis are used: the low frequency method uses modal damping values while the high frequency method relies on the attenuation between point and transfer FRF measurements.

Measurements were conducted by ISVR, Schrey&Veit and TU Berlin on 'freely supported' UIC60 rails, either 4 m or 6 m long. All partners measured decay rates using the high frequency direct method, from point and transfer responses functions (FRFs) at either end of the rail. Additionally, at low frequency ISVR determined decay rates from the modal properties of the rail. Three types of dampers were tested using the method (see Table 3).

Table 3: Short-rail method: Dampers tested by each project partner and the respective test rail lengths.

	Project partner		
Damper type	ISVR	Schrey&Veit	TU Berlin
Tata Steel	4, 6 m	6 m	6 m
S&VMK1	4, 6 m	6 m	6 m
S&VMK2	4, 6 m	6 m	

The rails were supported on soft mounts to achieve the 'freely' supported condition. ISVR and Schrey&Veit used stacks of rail pads. Schrey&Veit inserted steel interface bars between the rail and pads to reduce the contact area with the mounts. ISVR mounted the rail directly on the pads. TU Berlin mounted the rail on coil springs at either end of the rail, coupled to the rail via wooden interface blocks. All partners measured the bounce mode of the rail on the ground to be less than 30 Hz.

All partners installed ten dampers symmetrically on the 6 m rail with the central two dampers positioned 300 mm either side of the rail centre-line. All partners installed the dampers on the rail with a spacing of 600 mm. Additionally, Schrey&Veit investigated the effect of having either 8 or 9 dampers on the rail. Accelerometers were attached at either end of the rails using either beeswax or adhesive. The ISVR and Schrey&Veit accelerometers were attached 5 mm from the rail ends. The distance of the accelerometer from the end was different in the two TU Berlin tests: with the S&VMK1 dampers the accelerometers were positioned 300 mm from the rail ends; with the Tata Steel dampers this distance was reduced to 20 mm.

High-frequency measurements

For both lateral and vertical direction, a point FRF at one end of the rail and a transfer FRF to the other end of the rail was measured. The ISVR and TU Berlin measurements were conducted in a heated (20 ± 2 °C) laboratory. The temperatures during the Schrey&Veit tests were slightly lower (between 16 and 18 °C). The 'high frequency method' was used by all partners. In this method, the decay rates were determined directly from the ratio of the transfer FRF to the point FRF, each expressed in ⅓ octave bands, divided by the rail length (6 m).

Low-frequency measurements

A method also applicable to the low-frequency part of the spectrum has been proposed by ISVR. The method is applicable where distinct peaks in the FRFs could be observed, decay rates for each mode being derived from the point FRF using:

$$DR = 4.343\frac{\omega\eta}{c_g} \tag{6}$$

where the resonance frequency, ω, and the modal loss factor, η, were found using a circle fitting technique implemented in MATLAB. To determine the group velocity, c_g, first, the wave number of each mode was calculated from the order of the mode, n, and the rail length, L:

$$k_n = \left(n+\frac{1}{2}\right)\pi/L \quad \text{, for } n = 1,2,3,\ldots \tag{7}$$

Then, the dispersion curve (wave number versus modal frequency) for the rail was plotted, and finally the group velocity was estimated as the slope of the dispersion curve. The gradient at each resonance frequency was calculated as the average of the slopes on either side of this frequency, thus decay rates were not calculated for the first and last identified modes. Further details of this method can be found in [SR 10].

Additional short-rail measurements were made to give increased understanding of the method and causes of variability in the results. These are summarised in Table 4. Results of the tests are reported in [SR 8].

Table 4: Further short-rail tests conducted by project partners

Test	Partner
Low-frequency modal measurements	ISVR
4 m long rails	ISVR
Number of dampers	Schrey&Veit
Influence of measurement direction along the rail	Schrey&Veit
Influence of response (and excitation) position	TU Berlin
Influence of excitation position	Schrey&Veit
Position of soft supports	Schrey&Veit
Influence of contact with soft mounts	Schrey&Veit
Influence of temperature on decay rates	Schrey&Veit
Influence of contact paste (Tata Steel dampers)	Schrey&Veit
Repeatability	Schrey&Veit
Effect of heat ageing of dampers	TU Berlin
Measurements with rail on track components	TU Berlin

6.1.3 Comparison of decay rates between project partners

The free-rail decay rates measured by each partner for each damper type were compared with each other [SR 3], [SR 4], [SR 5], [SR 6]. ISVR defined a crude single-number measure to indicate the 'average' effect on sound pressure. This measure was applied to compare free-rail decay rates measured by each partner (Table 5). The single-number measure was defined as the mean of the difference in noise from the rail in each ⅓ octave band between 400 Hz and 5 kHz between pairs of conditions. The noise difference in each band, ΔL, was calculated from the decay rate under the two conditions DR_1 and DR_2, according to:

$$\Delta L = \left| 10 \log_{10} \frac{DR_1}{DR_2} \right| \tag{8}$$

Where decibel differences are quoted between different results they refer to ΔL. At frequencies above 400 Hz there was generally good agreement between the results – in most cases there was less than 3 dB average difference between partners (Table 5). Some of the variability at low frequencies was likely caused by measurement errors arising from the low excitation magnitudes at these frequencies resulting in low signal-to-noise ratio. In the case of the TU Berlin tests on the MK1 dampers, elevated results could be observed below 500 Hz. It seems likely that this discrepancy was caused by the fact that the accelerometer had been positioned 300 mm from the rail end, in contrast to <20 mm for the other partners. As the damper decay rates are relatively low below 400 Hz and tend to have little influence on overall track decay rates, the measured decay rates at these frequencies will generally be of less relevance.

At higher frequency, known factors will also partially account for some of the observed differences. In the case of the long-rail measurements, temperature differences between measurements have been shown to have a significant effect. In the case of the short-rail measurements, differences between the soft support conditions and temperature were likely to be the biggest sources of variability.

The results of this analysis resulted in a proposal for a revised test procedure for the assessment of rail dampers [SR 10]. This is based on the short-rail direct method and is aimed at eliminating, or at least controlling, the principal sources of variability.

Table 5: Average dB difference in free-rail decay rates measured by each partner for each damper type (calculated using Eq. (8))

Damper	Direction	Partner	Partner			
Tata Steel			ISVR (32 m)	ISVR (6 m)	Schrey&Veit (6 m)	TU Berlin (6m)
	Vertical	Tata Steel (18 m)	2.0	2.7	3.5	2.6
		ISVR (32 m)		2.1	2.2	2.1
		ISVR (6 m)			2.3	2.6
		S&V (6 m)				2.2
		TU Berlin (6m)				
	Lateral	Tata Steel (18 m)	3.2	4.9	5.6	4.6
		ISVR (32 m)		2.2	2.9	1.9
		ISVR (6 m)			1.4	1.8
		S&V (6 m)				1.2
		TU Berlin (6m)				
S&VMK1			ISVR (32 m)	ISVR (6 m)	Schrey&Veit (6 m)	TU Berlin (6m)
	Vertical	Tata Steel (18 m)	3.4	2.8	3.5	2.9
		ISVR (32 m)		2.9	2.2	4.5
		ISVR (6 m)			3.3	3.0
		S&V (6 m)				5.0
		TU Berlin (6m)				
	Lateral	Tata Steel (18 m)	2.6	2.2	3.2	2.4
		ISVR (32 m)		2.0	4.3	2.2
		ISVR (6 m)			3.3	1.5
		S&V (6 m)				4.1
		TU Berlin (6m)				
S&VMK2			ISVR (32 m)	ISVR (6 m)	Schrey&Veit (6 m)	TU Berlin (6m)
	Vertical	Tata Steel (18 m)				
		ISVR (32 m)		1.8	2.4	
		ISVR (6 m)			3.5	
		S&V (6 m)				
		TU Berlin (6m)				
	Lateral	Tata Steel (18 m)				
		ISVR (32 m)		2.3	3.4	
		ISVR (6 m)			3.2	
		S&V (6 m)				
		TU Berlin (6m)				

Fig. 18 shows an example of vertical decay rates of the Tata steel dampers determined by different partners with different methods (short rail and long rail). The corresponding results for the lateral DR are displayed in Fig. 19. All measurements including also the Schrey&Veit dampers are reported in [SR 8]. As can be seen in the Figure there was generally good agreement between all series of measurements in the vertical direction at frequencies greater than 400 Hz (compare with Table 5).

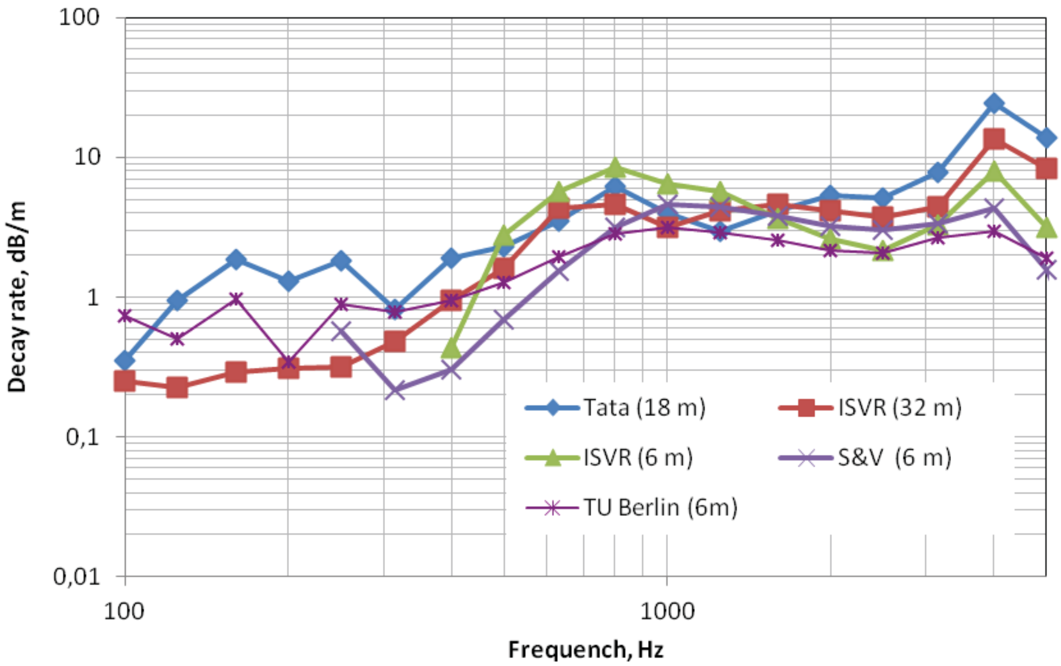


Fig. 18: *Tata Steel dampers: Comparison of vertical decay rates measured at freely suspended rails of different length by four of the project partners.*

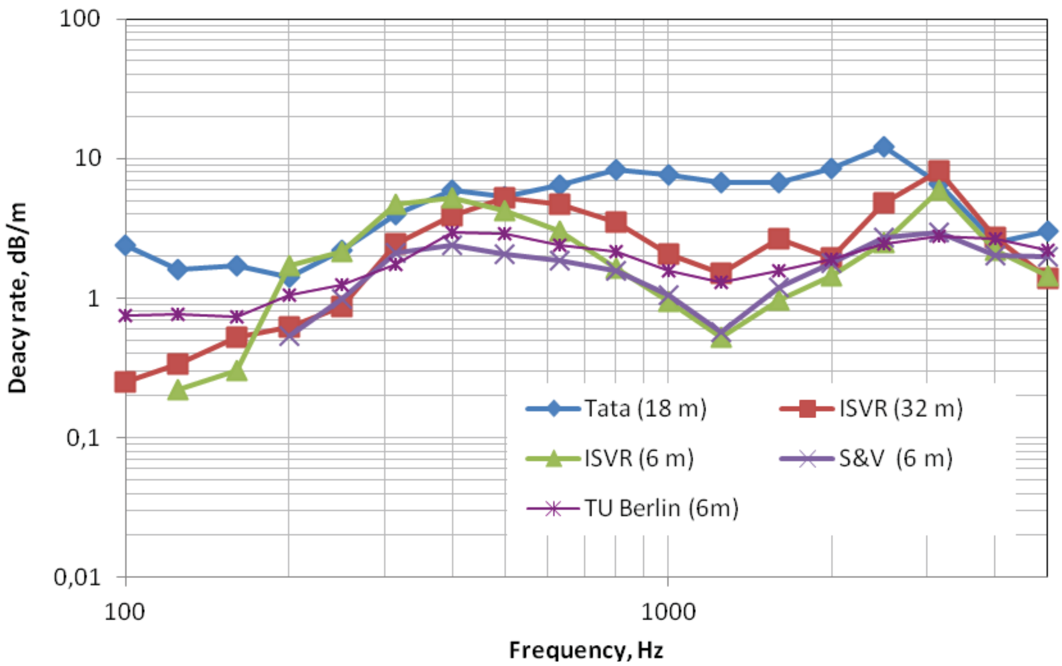


Fig. 19: *Tata Steel dampers: Comparison of lateral decay rates measured at freely suspended rails of different length by four of the project partners.*

At frequencies higher than 3 kHz, the decay rates measured using the long-rail method were somewhat higher than those measured with the short-rail method. In the lateral direction, the Tata Steel long-rail results were higher at most frequencies than the other measurement sets, particularly between 0.8 and 2.5 kHz. The reason for this difference is not known; one possibility is that there were variations in the dampers or contact paste required for fixing the Tata-dampers to the rail, that were supplied to partners.

6.1.4 Variability in long rail measurements

The ISVR single number measure (see previous Section) is used to compare average effects from variations in the long-rail test method (Table 6). Some of these variables were evaluated at the ISVR-test track and may only be applicable for undamped tracks and not for the long freely suspended rail. For comparison, the effect of introducing dampers is shown as the first entry.

Table 6: Effect of dampers and variations in long-rail test method on decay rates

		Vertical		Lateral	
Nr	**Variable**	**Comparison Conditions**	**Average effect 0.4 - 5 kHz, dB**	**Comparison conditions**	**Average effect 0.4 - 5 kHz, dB**
1	Effect of dampers on track decay rates	No dampers vs. S&VMK1 dampers	8.0	No dampers vs. S&VMK1 dampers	7.5
2	Effect of temperature on undamped track	10.7 vs. 21.0 °C (left rail)	2.4	14.8 vs. 21.0 °C (left rail)	0.9
		4.0 vs. 10.8 °C (right rail)	1.0	4.0 vs. 6.6 °C (right rail)	0.4
		4.0 vs. 21 °C (combined)	3.4	4.0 vs. 21.0 °C (combined)	1.4
3	Difference between rails (undamped)	Left (10.7 °C) vs. Right (10.6 °C)	1.9	Left (14.8 °C) vs. Right (6.6 °C)	2.2
4	Effect of temperature on MK2S&V dampers	3.6 vs. 15.0 °C	2.3	3.6 - 15.0 °C	1.0
5	Effect of temperature on Tata Steel dampers	2.6 vs.11.0 °C	2.0	2.1 vs. 14.2 °C	1.7
6	Hammer impact relative to accelerometer*, cm	(x=2, y=0) vs. (x=2, y=2)	1.6	(x=2, y=0) vs. (x=2, z=2)	0.6
7	Accelerometer position	Rail head vs. Rail foot	1.1	Gauge, Field	0.7
8	Grid position	4m vs. 8m (from rail end)	1.1	4m vs. 8m (from rail end)	0.9
9	Grid density (undamped)	1/8 to 1/2 sleeper spacing	0.1	1/8 to 1/2 sleeper spacing	0.1
10	Grid density (undamped)	1/8 to 1 sleeper spacing	0.9	1/8 to 1 sleeper spacing	0.6
11	Grid density (Tata Steel dampers)	1/8 to 1/2 sleeper spacing	0.2	1/8 to 1/2 sleeper spacing	0.2
12	Grid density (Tata Steel dampers)	1/8 to 1 sleeper spacing	1.1	1/8 to 1 sleeper spacing	0.5

The influence of the different variables is discussed in detail in [SR 8]. In the following section only the influence of temperature will be discussed due to its outstanding importance.

6.1.5 Variability in short-rail measurements

For the short-rail test method, the ISVR single-number measure (see Section 6.1.3) has also been used to compare effects from variations in method (Table 7). For comparison, the effect of introducing dampers is again shown as the first entry.

Table 7: Effect of dampers and variations in short-rail test method on decay rates

			Vertical	Lateral
Nr	**Variable**	**Comparison conditions**	**Average difference 0.4 - 5 kHz, dB**	**Average difference 0.4 - 5 kHz, dB**
1	*Effect of dampers (Short rail tests)*	*No dampers vs. S&VMK1 dampers*	*8.0*	*7.5*
2	Number of dampers	8 dampers vs. 10 dampers 9 dampers vs. 10 dampers	*0.9* *0.9*	*1.4* *1.7*
3	Measurement direction	Excitation on left vs. excitation on right S&VMK1 S&VMK2 Tata Steel	 *0.5* *0.3* *0.6*	 *0.5* *0.5* *0.3*
4	Effect of excitation position	Inside acc. vs. outside acc. S&VMK1 S&VMK2 Tata Steel dampers	 *0.5* *0.5* *0.5*	 *0.9* *0.9* *1.3*
5	Position of soft supports	Support 0 mm from end vs. support 350 mm from end	*0.6*	*0.3*
6	Effect of support area	On pads direct vs. on pads via metal bars	*2.2*	*0.8*
7	Effect of temperature	6 vs. 30 ºC S&VMK1	*0.1*	*1.0*
8	Effect of contact paste	Dampers cured for 3 days vs. 7 days Tata Steel	*1.0*	*0.5*

The influence of the various effects is also discussed in detail in [SR 8].

Effects of damper temperature

According to Table 6 and 7 the largest effect (besides the effect of the dampers) on the measured track decay rates is from variations in temperature. Changes of 3 dB and more in the estimated decay rates have been observed.

The effect of damper temperature was investigated in more detail by measuring decay rates of the damped 'freely supported' long rails at 4°C and 15°C (Variable nr 4, Table 6,). With the S&VMK2 dampers, at frequencies above 1.6 kHz the vertical and lateral decay rates decreased with increasing rail temperature, while from 250 to 500 Hz the rates increased with increasing temperature; inconsistent effects were found at other frequencies. Similar trends can be observed for the rail fitted with the Tata Steel dampers, however the frequencies at which the decay rates decreased/increased with temperature varied between the damper designs.

Changes in decay rates with changes in temperature will vary between damper designs. These effects are likely to be difficult to model or account for. For this reason controlling the test environment is likely to be a more practicable approach to achieving repeatable measurements. This supports the use of the short rail test method, being easier to utilize in a controlled environment (i.e. indoors).

The effect of damper temperature was also analysed for short rails, see Variable 7, Table 7. The decay rates were measured at 6, 18 and 30 °C with the S&VMK2 dampers fitted to the rail. Decay rates decreased between 800 Hz and 4000 Hz in the lateral direction and below 1600 Hz in the vertical direction, with increases in temperature.

These findings are consistent with the influence of damper temperature of the long-rail test results and suggest that where possible the temperature should be controlled during testing.

6.1.6 Comparison between measured and predicted decay rates

In addition to the damped free rail decay rates on the 32 m rail, ISVR also made measurements with the rail clipped in to their test track, to measure the undamped and damped track decay rates. It was hypothesized that the decay rates of a damped track could be found from summing the decay rates of a damped 'freely supported' rail and the decay rates of an undamped track. The decay rates of the damped track were predicted by adding the decay rates of the damped free rail to the decay rates of the undamped track (see Section 4.1).

Measured vertical decay rates for the S&VMK2 dampers are shown in Fig. 20; results for other dampers can be found in [SR 10]. The damped 'free' rail decay rates in both directions (dotted lines in Fig. 20 and Fig. 21) show that the dampers introduced high decay rates in the region of the broad trough in the undamped track decay rates (500 to 3150 Hz in the vertical direction and 315 to 2500 Hz in the lateral direction).

Damped track decay rates for the S&VMK2 dampers have been predicted by summing the damped 'free' rail decay rates from the long- rail measurements with those of the undamped track (Fig. 20 and Fig. 21). These show reasonable agreement with the directly measured decay rates at most frequencies - using the single number measure defined in Section 6.1.3 there was 1.0 dB average difference between the vertical decay rates and 1.1 dB average difference in the lateral decay rates. Changes in temperature between conditions (see Section 6.1.5) and transducer effects may account for some of these differences. Similar levels of consistency between measured and predicted damped track decay rates were observed with the other damper types, however there was evidence of substantial variability between measured and predicted decay rates below 300 Hz. This was less evident in the S&VMK2 results that are shown here. Transducer effects such as low signal to noise ratio of the accelerometer and thermal drift may have accounted for some of the uncertainty at low frequency. The added mass of the dampers may also have had an effect (in this frequency region they behave mainly as added mass).

As an example for the results from the 'short-rail method' Fig. 22 and Fig. 23 show a comparison of the measured and predicted damped track decay rates in vertical and lateral directions. This example is based on the ISVR short-rail measurements for the S&VMK2 damper decay rates. Agreement was reasonably good, with 2.3 dB average difference between the vertical decay rates and 1.8 dB average difference in the lateral decay rates. Differences in damper temperatures between measurements and possibly also short-rail mounting effects may account for some of these differences.

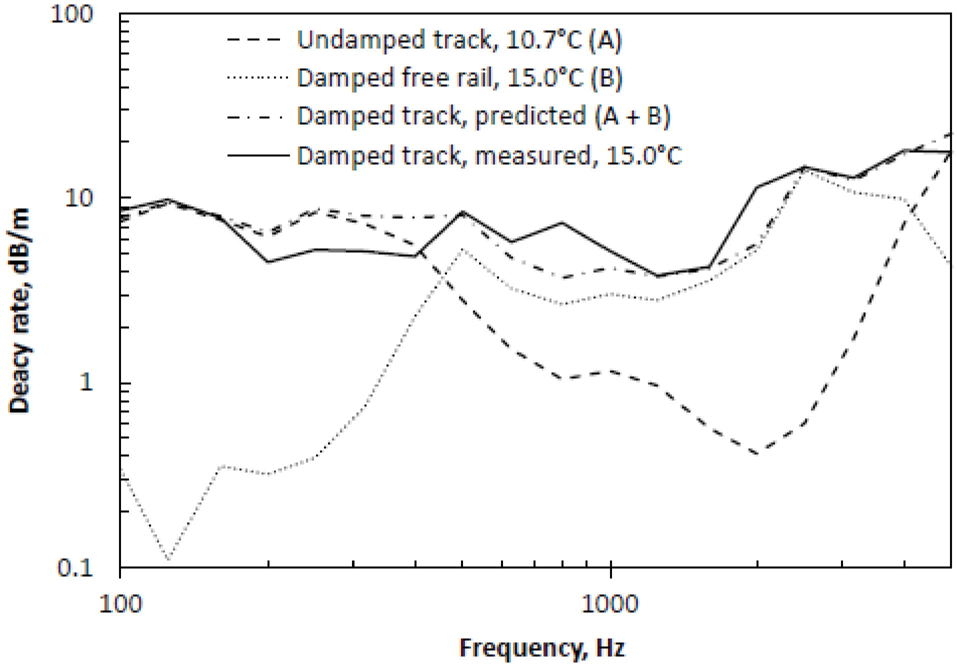


Fig. 20*: Comparison of measured and predicted vertical decay rates using the* *<u>long-rail</u>* *method to measure the decay rates of the damped free-rail (S&VMK2 dampers)*

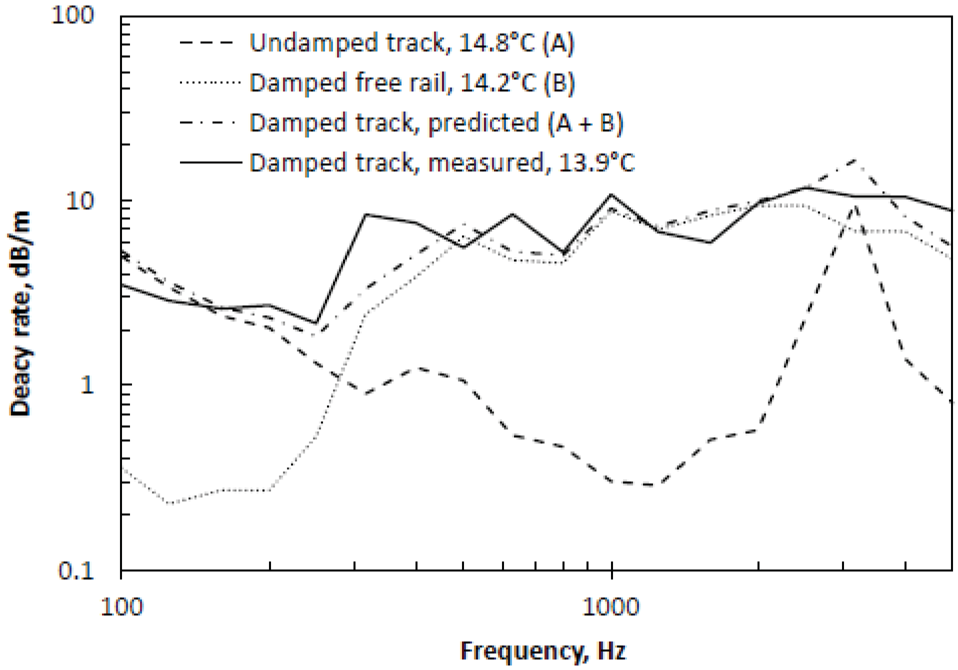


Fig. 21: *Comparison of measured and predicted lateral decay rates using the* *<u>long-rail</u>* *method to measure the decay rates of the damped free-rail (S&VMK2 dampers)*

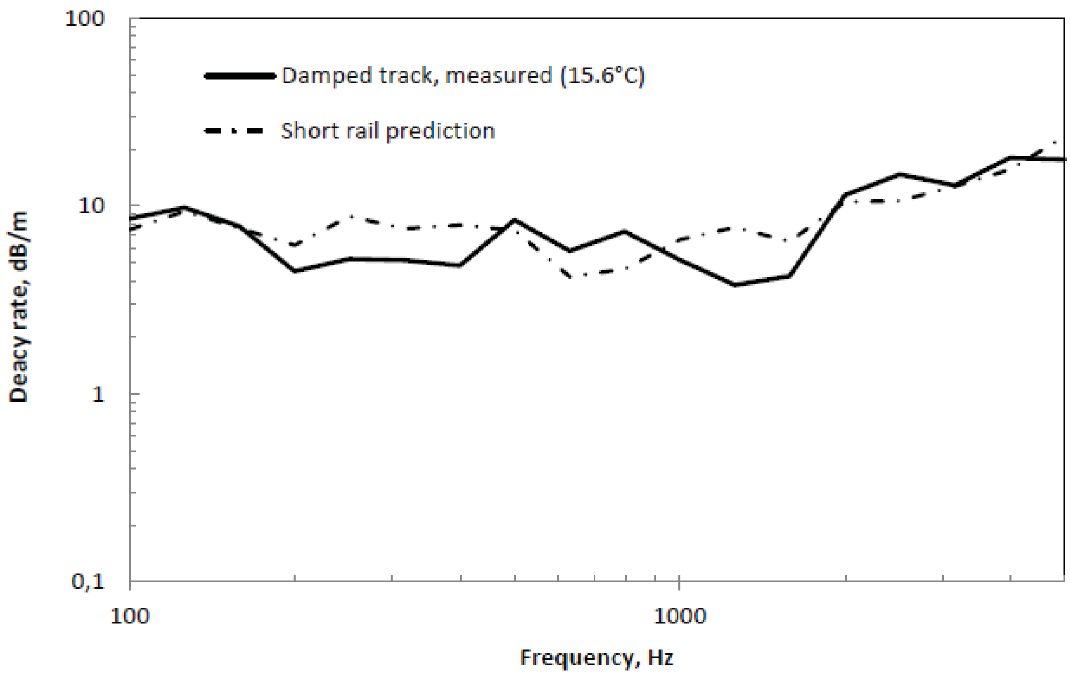


Fig. 22: *Comparison of measured and predicted vertical decay rates using the* *short-rail* *method to measure the decay rates of the damped free-rail (S&VMK2 dampers)*

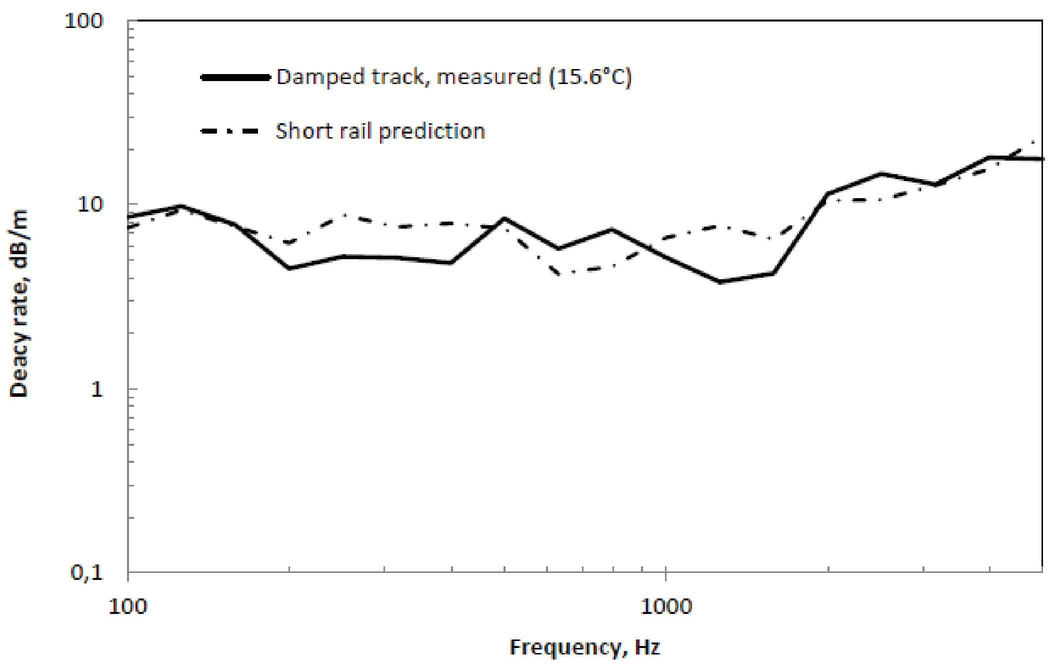


Fig. 23: *Comparison of measured and predicted lateral decay rates using the* *short-rail* *method to measure the decay rates of the damped free-rail (S&VMK2 dampers)*

6.1.7 Conclusions

Round robin tests of rail damper test methods have been conducted by four STARDAMP partners: ISVR, Tata Steel, Schrey&Veit and TU Berlin. Two different test methods have been evaluated. In the first of these methods, decay rates of freely supported long rails (18 or 32 m) were determined by integrating decay rates derived from FRF's measured at intervals along the rail. In the second of these methods, decay rates of short rails (4 or 6 m) were determined at low frequency from the modal properties of the rail, and at high frequencies directly from point and transfer response functions (FRFs) at either end of the rail.

Reasonable agreement between partners was found in decay rates measured with three different damper types. Except for a few explainable exceptions, average differences in decay rates (10 log10(DR)) between 400 Hz and 5 kHz were generally less than 3 dB between partners. Sources of variability were identified and investigated. For the long-rail method, damper temperature and measurement grid position were identified as the greatest potential sources of variability.

Using the short-rail method, the coupling with the isolation mounts, the position of the accelerometer, and damper temperature were considered the greatest sources of variability. The direct short-rail method was effectively limited to frequencies above around 300 Hz. However, as the damper decay rates are relatively low below 400 Hz and tend to have little influence on overall track decay rates, this limitation was thought unlikely to be of significance for most applications. The short-rail test method offers greater control of variability particularly with respect to controlling the damper temperature. A revised test procedure, based on the short-rail direct method, has been developed [SR 10]. This is described in the next section.

Damped track decay rates have been predicted by summing the damped 'free' rail decay rates with those of the undamped track. These showed reasonable agreement with the directly measured decay rates at most frequencies - 'average' differences were less than 1.1 dB were when using long-rail measurements, and were less than 2.3 dB when using short-rail measurements.

6.2 The '6m rail' procedure for testing rail dampers

Based on the results described in section 6.1 a laboratory test method has been defined within STARDAMP that can be used to measure the performance of rail dampers on a free rail. The purpose of such measurements is to allow a standardised assessment of rail damper performance without the need for installation of dampers in a track. Moreover the results can be applied to any track if the decay rates of the untreated track are known.

The proposed method is a laboratory-based test, in which dampers are installed on a 'short' 6 m length of rail. The method has been developed by partners in the STARDAMP project but is also intended for the use of third parties to assess the performance of rail dampers. For full documentation see [SR 10].

6.2.1 Principles

Rail dampers reduce the noise radiated by the track by increasing the track decay rate. In the test method, decay rates are determined on the basis of the ratio of the frequency response function (FRF) at the point of excitation at one end of a 'freely' supported 6 m rail to the response FRF at the other end of the rail. An instrumented hammer shall be used to excite the rail. Accelerometers shall be used to measure the responses at either end of the rail. Decay rates shall be determined for vibration in the vertical and lateral directions.

The test method is outlined below.

6.2.2 Data acquisition

A rail of length 6 m (+/- 10 mm) shall be used for the tests. This shall be of the same type that is used at the intended site for the damper installation. The rail shall be in good condition with no welding joints or significant defects.

Dampers are to be installed symmetrically over the whole length of the rail at a centre-to-centre spacing representative of the installation in the track. The damper positions shall be marked out from the centre-line of the rail. An example of installation with 0.6 m centre-to-centre damper spacing is shown in Fig. 24. Dampers shall be attached to the rail according to the manufactures guidelines.

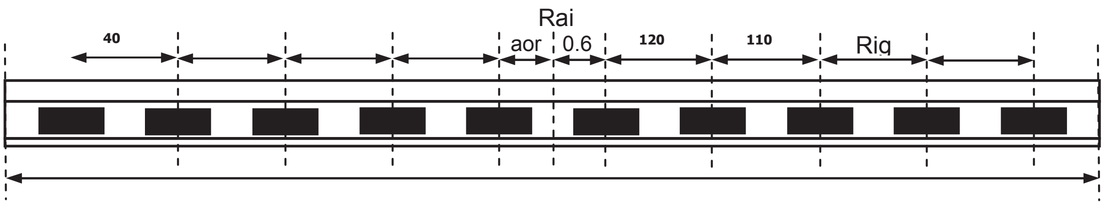


Fig. 24: *Example of damper installation with 0.6 m centre-to-centre spacing. Damper positions marked from centre-line (CL) of the rail. Dimensions in metres.*

The rail shall be 'freely suspended' at either end on a foundation that is soft enough so that the bounce mode has a natural frequency less than 30 Hz. For example, this might be achieved using stacks of rail pads or rubber mounts. Mounts with high internal damping (i.e. viscous damping ratio greater than 0.01) shall be used to minimize the influence of internal mount resonances on the results. Metal bars shall be inserted at the interface between the mounts and the rail to control the contact area with the rail. The bars shall have a cross-section of dimensions 20mm x 20mm and a length greater or equal to the rail foot width. No part of the mounts shall make direct contact with either the rail or the dampers. Likewise, the interface bars shall not make contact with the dampers or their fastenings.

The rail temperature shall be between 18 and 25°C during the tests. The air temperature as well as the rail temperature must be recorded at the start and end of each set of measurements. Rails and dampers shall be acclimatised in the test environment prior to testing

It is recommended that further measurements are made at temperatures encompassing the range expected at the intended damper installation site, particularly where there is significant deviation from those described above.

It is recommended to measure more than one sample of rail fitted with a given type of rail damper in order to check variability.

6.2.3 Tests

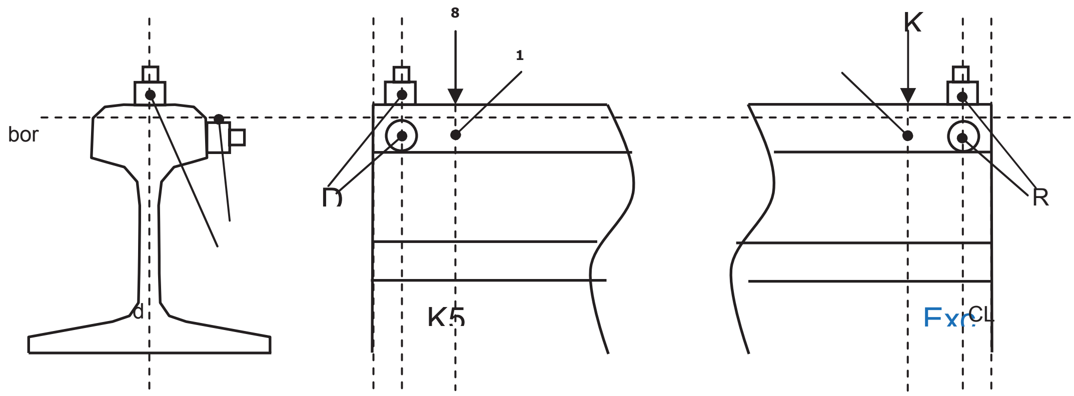
Measurements are to be taken for the vertical direction and the transverse direction.

For both measurement directions, a point frequency response function (FRF) at one end and a transfer FRF to the other end shall be measured. To check variability, measurements shall be repeated with the excitation point at either end of the rail in turn. A qualification test shall be carried out to verify that the bounce mode satisfies the requirement that it has a natural frequency less than 30 Hz (see above) for each measurement direction. A hammer with a soft tip shall be used to ensure sufficient energy is input at low frequencies.

6.2.4 Equipment

The response at either end of the rail shall be measured using accelerometers. The centres of the accelerometers shall be fixed within 10 mm of the rail ends (Fig. 25). The accelerometers shall be mounted in the vertical direction, on the centre of the rail head surface; and in the lateral direction, on the centre of the outside face of the rail head.

The accelerometer shall have a mass of less than 10 g and a diameter less than 15 mm. The accelerometer and its mounting must not have a resonance below 10 kHz. This can usually be achieved by attaching the accelerometer using a ceramic or Cyanoacrylate glue.



Key

a	accelerometer position
b	hammer impact position
c	distance from rail end to centre of accelerometer (shall be less than 10 mm)
d	distance from centre of accelerometer to hammer impact (shall be less than 30 mm)
CL	Centre lines of rail head

Fig. 25: *Position of accelerometers and excitation points on the rail*

Excitation shall be provided by an instrumented hammer containing an integral force gauge. A suitable hammer would normally have a mass of around 100-200 g and a titanium tip. It shall be demonstrated that it gives a sufficiently flat input spectrum up to at least 7 kHz (i.e. the power spectrum of the force shall be flat within 20 dB in this range). It is important to tap the rail very gently and to avoid double strikes. Care should be taken to ensure that the same point of excitation is used throughout the testing. This point shall be within 30 mm of the centre of the accelerometer and at the same position of the transverse rail profile, see Fig. 25. Data shall be recorded onto a multi-channel Fourier analyser (i.e. at least three channels).

6.2.5 Determination of FRFs

For the measurement of FRFs ISO 7626-5 shall be used [2]. The following measurement parameters shall be used:

- FRF measurement may be accelerance or mobility;
- The hammer signal shall be used as the reference to determine the transfer functions;
- An average of at least ten impacts shall be used for each FRF measurement (see [2]);
- The coherence function shall be used as an indication that the signal is not contaminated by background noise. A coherence of at least 0.8 is required apart from at isolated dips in the FRF3;
- The analysis shall be triggered by the force pulse. At least 10 ms of pre-trigger shall be used. The pre-trigger shall not exceed 15% of the analysis window;
- A rectangular window shall be used (i.e. no window). For the force signal a ‘force’ window may be used instead;
- The analysis window shall be long enough to capture the whole of the decaying response due to impact. For a damped rail it is expected that 0.25 s will be sufficient, implying a frequency resolution of 4 Hz but this shall be verified and a longer window used if necessary. A shorter time than 0.25 s shall not be used. No zero padding shall be used;
- The maximum frequency of analysis shall be at least 5.62 kHz (for the 5 kHz one-third octave band);
- The sample rate shall be at least 2.5 times the maximum frequency of interest (i.e. at least 14 kHz to obtain measurements up to the 5 kHz one-third octave band. A higher sample rate can be used if available.

A measurement of background (instrumentation) noise shall also be taken to verify that both the force and the response signal is at least 10 dB above the background noise in each one-third octave band. To achieve this, the data acquisition system shall have sufficient resolution (normally at least 16 bits; 24 bit resolution is preferable). Parallel data acquisition across channel shall be used to avoid any phase shift between the signals.

Absolute calibration of the equipment is not necessary. However, it is essential that the relative calibration between the accelerometers is verified. This can be achieved by measuring the FRF of a ‘rigid’ block of known mass: the accelerance shall be equal to the reciprocal of the mass over a suitably wide frequency range.

The analysis methods used shall be adequately documented such that they could be repeated by someone else.

[3] It is accepted that the signal may drop into the background noise at anti-resonances, especially on short sections of rail, leading to a drop in coherence. The criterion is included to ensure that the majority of the signal within a frequency band is not contaminated by noise.

6.2.6 Determination of damped free-rail decay rates

Measured FRFs shall be mobilities or accelerances. The FRFs shall be averaged into one-third octave bands.

In each one-third octave band, the ratio of the transfer FRF to the point FRF, each expressed in ⅓ octave bands, divided by the rail length (6 m) shall be used to determine the decay rate. Decay rates below 1.0 dB/m shall be regarded as unreliable and excluded from results.

6.2.7 Determination of damped track decay rates

The measured decay rates on the free rail are added to the decay rates of a track with no rail dampers in each one-third octave band to obtain the result for a track with dampers fitted. The method in EN 15461 [1] shall be used to determine the decay rates of the undamped track.

6.2.8 Test report

The following elements shall be presented in the test report:

- The precise positions of the accelerometers and excitation points;
- Description of the test rail, including type and profile;
- Description of the dampers, including types, component masses, dimensions and serial no. or other means of identification;
- Details of the number of dampers installed, their position on the rail, and the attachment procedure (e.g. fastening torques, curing times etc.).
- The temperatures of the rail and air before and after each measurement sec.
- The manufacturer, types and serial no. or other means of identification of the accelerometer, impact hammer and spectrum analyser used;
- Vertical and lateral decay rates for each condition.

The decay rates shall be presented in graphical form, in a range of one-third octave band frequencies between 100 Hz and 5 kHz. The rates shall be represented on the vertical axis, expressed in dB/m with a logarithmic scale between 0. 1 dB/m and 100 dB/m. The frequency shall be represented on the horizontal axis with equidistant divisions for each one-third octave frequency band. The frequencies considered shall be the normal frequencies corresponding to the one-third octave intervals specified in EN ISO 266 [3].

Decay rates in each one-third octave band should also be presented in tabular form.

6.3 *Measuring TDR on real tracks – best practice rules*

The STARDAMP method to predict the effect of a rail damper on a specific real track is widely based on the exact knowledge of the TDR. The standard EN 15461 [1] describes a method of how to determine the decay rate of a real track. This method is based on frequency response function measurements at several points of excitation along the rail.

A number of measurement campaigns were compared in STARDAMP. These included both pass-by and TDR measurements at various test sites around Germany in which dampers had been installed. Even though all track decay rate measurements had been performed by well-reputed engineering offices and according to the standard, there were discrepancies between theory and measurement results that could not be explained. Since TDRs measured on a real track are an important input parameter for the STARDAMP software, it was decided that the factors causing the discrepancies were to be further examined in a measurement campaign. This campaign was to be performed by DB by making a series of repeated measurements but varying the conditions, the robustness of the method should be assessed. Based on the conclusions from this measurement campaign and measurements performed by ISVR on a 32-m test track best practice rules have been defined ([SR 17]).

6.3.1 Concept of the TDR measurement campaign

The full measurement campaign and its results are described in [SR 7]. The objective of the measurement campaign was to test the reproducibility of the standard method for determining track decay rates at real undamped tracks. It included the testing of variables that may originate from different interpretations of the standard procedure and in that sense the objective was also to evaluate the practicability of the standard

The measurements were performed by the DB Systemtechnik acoustic measurement group according to the standard EN 15461 [1]. The following influencing parameters were tested:

Repeatability over a short period of time: Two complete measurement sets were performed consecutively without any changes.

Mounting and position of the accelerometer: To determine whether the position and the mounting of the accelerometer could influence the results, the complete measurement campaign was conducted with two sensors simultaneously. One was attached with a two-component adhesive (ch1) and the other one with a magnet (ch2), see Fig. 26.

Fig. 26: *Two accelerometers, Ch1 and Ch2, were used simultaneously during the whole measurement campaign to evaluate the effects of mounting and position.*

Reproducibility of results between separate days: A comparison was made between measurements performed on two separate days, at the same position and within the same temperature range (±5°C). All equipment except for the mounting plate of sensor 1 was demounted each night and set up again the following morning. The long term behaviour of the track dynamics could be investigated by comparing the measured track decay rates with ones measured during the project LZarG in year 2008 on the exact same position [8].

Changing rail temperature: The measurement campaign was conducted in a period of rather cold nights and warm days. A minimum rail temperature of ca. 19°C was measured in the mornings and because of glaring sun, the rail heated up quickly to a maximum of about 35°C in the afternoons. This variable was also tested by ISVR at the 32m test track.

Measurement operator /excitation technique: All measurement sets except for one in each direction were performed by the same operator. Both operators were trained acousticians who have performed several decay rate measurements. The first operator has many years of experience whereas the second one can still be considered a beginner.

Different starting positions: Before the measurements started, the direct FRF was measured at five positions on the right hand rail. Three of these positions were chosen to be starting positions for a complete measurement set.

Inter-rail variability: Complete measurement sets were performed on both rails with the same starting position.

Replacing of accelerometers: The accelerometer attached with a magnet was removed and repositioned at the same position. The accelerometer attached with adhesive was remounted on a new mounting plate directly next to the first position. Then the track decay rate measurement was repeated.

Different striking positions: In the lateral direction, the measurements were carried out on both sides of the rail head, with the operator striking the rail head as close to the first point as possible in the measurement direction. In the vertical direction, the direct FRF at position 0 was measured at both sides of the sensor.

Changing the measurement grid: For this test the five last measurement positions were skipped and added at the beginning of the section so that a measurement point was situated in each quarter inter-sleeper for the first 2.55 m, see Fig. 27. All other measurements were performed with the measurement grid specified in the standard.

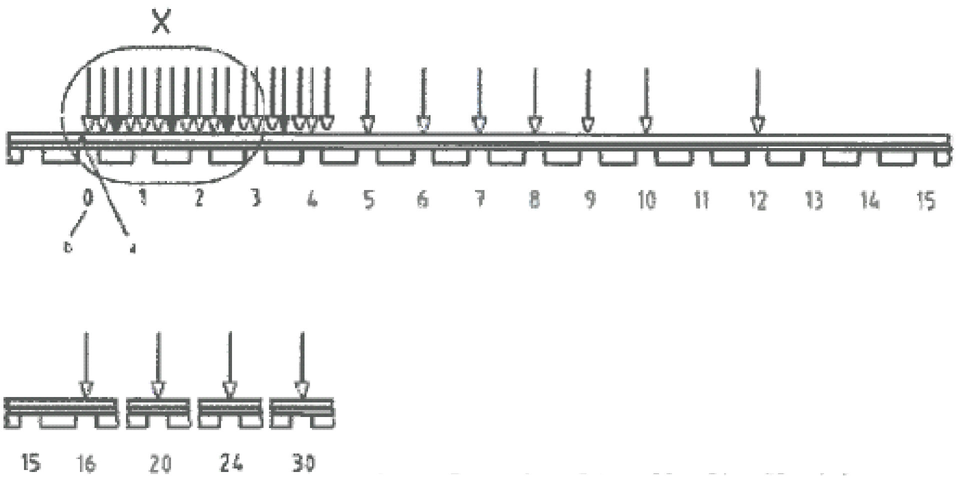


Fig. 27: *The modified measurement grid with more striking positions in the near field.*

6.3.2 Results

The test track section is located in the railway yard *München Nord* (see Fig. 28). The chosen section is a ballast track with concrete mono-sleepers at 0.6 m distance and UIC60 rail. It corresponds well with the typical track type that can be found in Germany. The track was visually in a good condition i.e. there were no loose sleepers, the surface of the rails was smooth and all fasteners looked intact. There were no indications of inhomogeneous track dynamic properties along the test site..

Fig. 28: *The test track in the railway yard „München Nord"*

Three of the influencing parameters mentioned above are described in the following section. These are the effect of the rail temperature, the position of the accelerometer and the starting position. The remaining parameters are discussed in detail in [SR 7].

The effect of rail temperature

Temperature is a factor that may influence the dynamics of a track. When the temperature is high most rail pads get softer which leads to lower decay rates. The extent of this softening depends on the pad type, its nominal stiffness, thickness and material.

It is recommended that the rail pad temperature is measured instead of the rail temperature. On a sunny day the temperatures of the rail may change much faster than the actual pad temperature giving misleading information. The effects of changing rail pad temperature was examined both by ISVR on the 32 m-test track and by DB on the real track.

Figure 29 shows a comparison of the vertical decay rate of three measurements on the real track with starting point K2 (Fig. 33). Two measurements at similar temperatures on the same day were included (2 and 3) to compare the third measurement, measurement 19, not only to the results, but also to see any points at which the difference was greater than that expected from the natural variation. The difference in rail temperature was about 5-10°C.

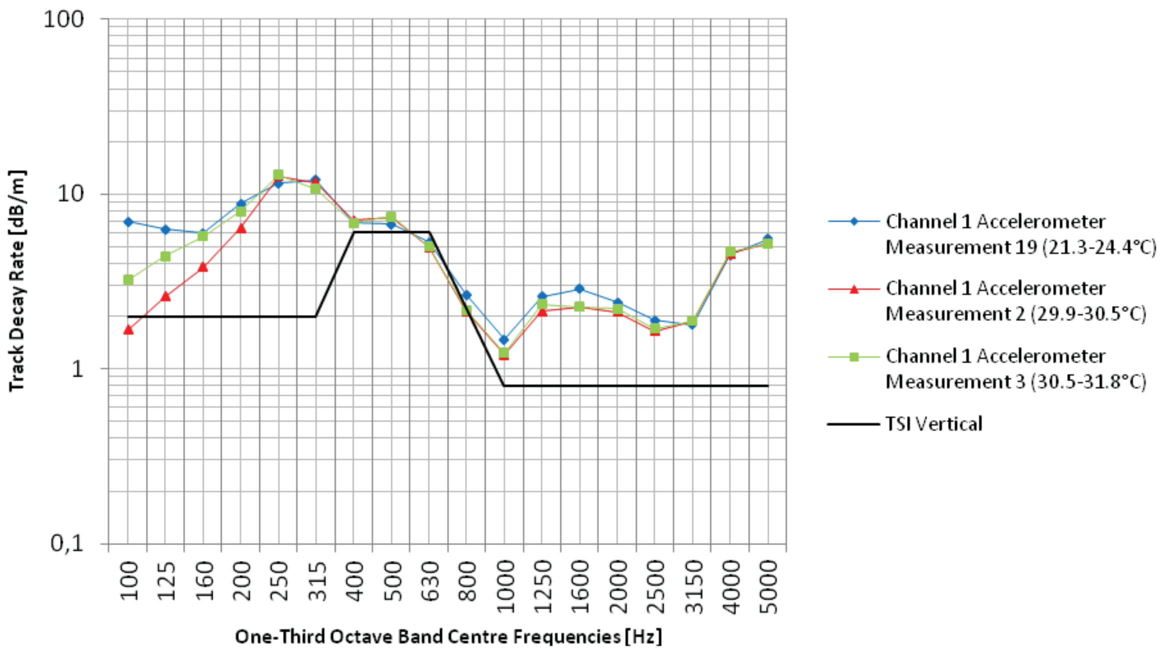


Fig. 29: *The vertical decay rate measured at starting point K2. Comparison of two repeated measurements (natural variation) with a third measurement performed at a different day and with 5-10°C temperature change*.

From the Figure, it can be seen that in this example there is a lowering of the decay rates with increasing temperature between 1000 Hz and 2500 Hz. The deviation amounts to a maximum of ca. 1 dB/m. The variations in the low frequency range are due to other reasons and can be neglected here. An influence of temperature on the lateral track decay rate could not be detected.

Also the ISVR-measurements on the 32 m test track showed a clear correlation between increasing rail pad temperature and decreasing decay rate. Figure 30 shows the temperature dependence of the decay rates of both the rails of the 32 m track in the vertical direction. With both rails, the decay rates consistently decrease between 300 Hz and 2 kHz with each increase in temperature. Temperature dependence in the lateral direction showed similar, though less pronounced, trends.

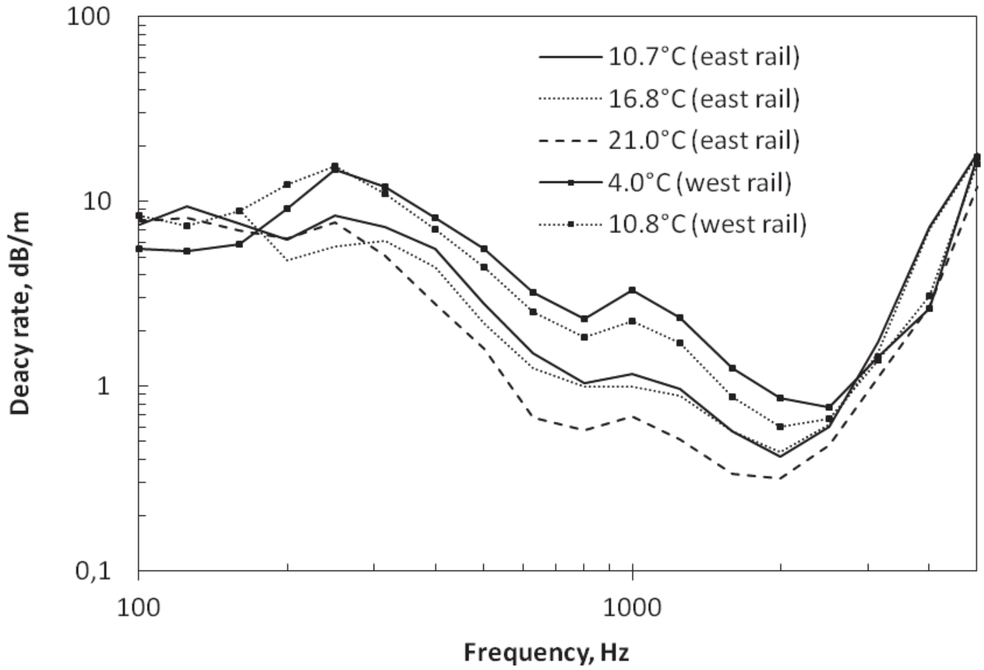


Fig. 30: *Effects of rail (pad) temperature and rail (unclipping and reclipping) on vertical decay rates of the undamped track*

Both examples give a strong indication that the rail pad temperature affects the track decay rate. It is also clear that this influence depends on the track and the measurement conditions which means that the effect will vary for different cases.

The effect of mounting and position of the accelerometer

The accelerometer shall be mounted in the vertical direction, on the longitudinal axis of the rail; and in the lateral direction, on the centre of the outside face of the rail head. According to the standard the accelerometer can be attached to the rail at two alternative positions for each direction (chapter 6.3 [1]).

The influence of the position of the accelerometer on the rail was investigated in the STARDAMP measurement of TDR on a real track. All measurements were performed with two accelerometers simultaneously. In the vertical direction sensor 1 (ch1) was mounted to the underside of the rail and sensor 2 (ch2) was fixed directly above this position at the rail head with a magnet. In the lateral direction both sensors were positioned side by side at the outer side of the rail

Fig. 31 and 32 show a comparison of the two accelerometers for both the vertical and lateral track decay rates respectively.

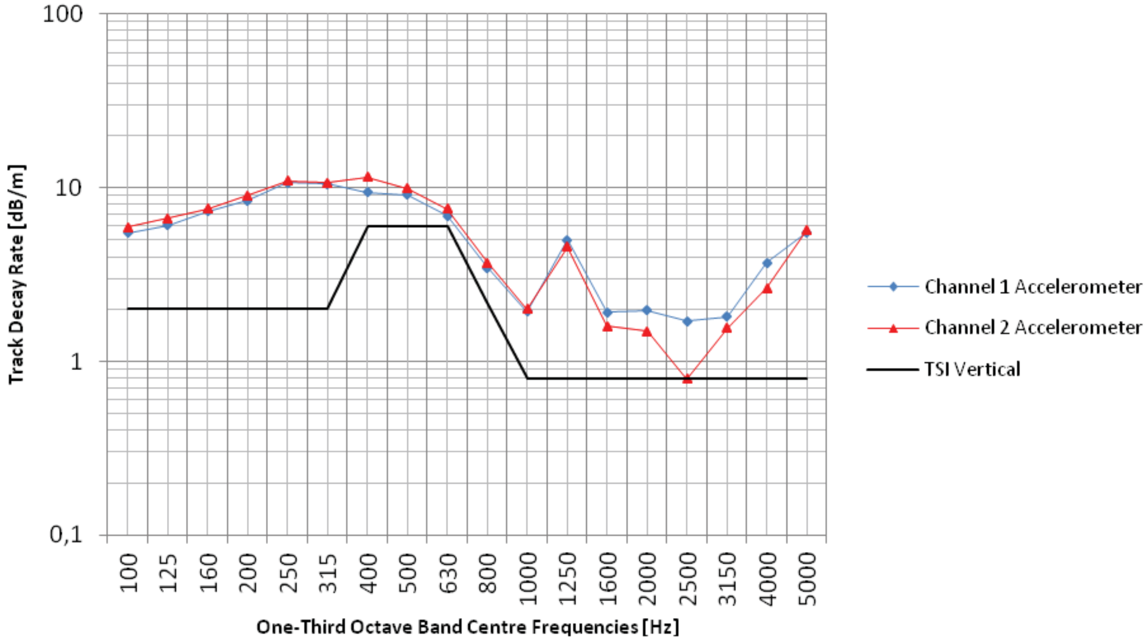


Fig. 31 : *Vertical TDR. The mean value calculated out of 9 measurement sets for a comparison between accelerometer 1 (on the foot of the rail) and accelerometer 2 (on the top of the rail) – see also Fig. 26.*

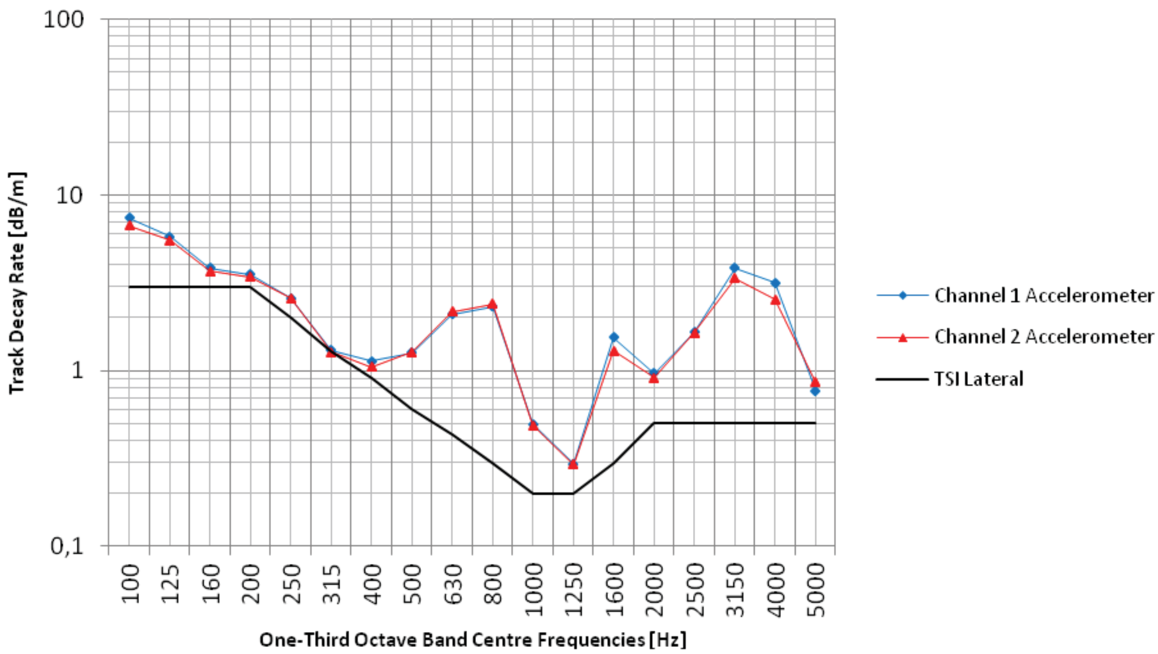


Fig. 32 : *Lateral track decay rate. The mean value calculated out of 13 measurement sets for a comparison between accelerometer 1 and accelerometer 2, positioned side by side on the rail head.*

The two curves in Fig. 31 are very similar up to the 1600 Hz band, where there is a relatively large variation up to 3150 Hz (and especially at 2500 Hz). The difference between the two sensors amounts to ca 1 dB/m in this frequency band. Such variations between the accelerometer on top and underneath the rail were seen at all four measurement positions. This leads to the conclusion that this error was reproducible in the sense that it can't have been caused by a single mistake in the mounting of the sensor. It could be observed that the discrepancies were solely due to unknown effects in the direct-FRF measurement. This generally excludes that it is caused by the measurement equipment itself or the way of mounting the sensor

The two curves for the lateral direction (Fig. 32) from Channels 1 and 2 are very close to each other, except for slight discrepancies at 1600 Hz and 3150 Hz. These results would be expected to be much more similar than the vertical results, as the accelerometers were placed adjacent to each other.

Reproducibility of results from different starting positions

Measurements carried out along the rail at three different starting positions (see Fig. 33) are compared for vertical track decay rates in Fig. 34 and in Fig. 35 for lateral track decay rates.

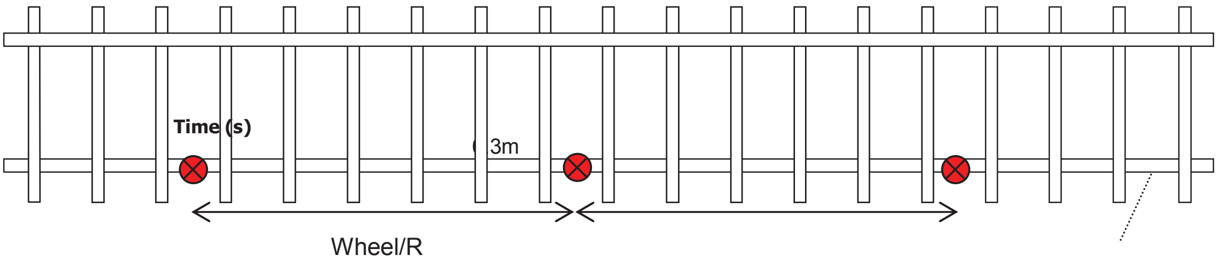


Fig. 33: *The test track with the accelerometer positions for the homogeneity test marked by red dots. Complete TDR-measurements were performed with starting point K2, K5 and K4.*

The starting position noted K2 was the reference position of this campaign at which most of the tests were carried out. The positions K4 and K5 were situated at 3.6 m distance at either side of K2. Despite being carried out on the same rail under the same conditions except for slight temperature variation (19.9-24.4°C), the vertical decay rate results show a large difference between K2 and the other two measurement positions. The results from the other tests have proven that the measured decay rate is reproducible at each of these positions. The conclusion must be drawn then that it is not the measurement conditions causing these differences but the track itself.

It is impossible at the moment to clearly identify the cause for these differences. From looking at the track no irregularities were noticed such as a pumping sleeper or damages to the clipping. It is also unlikely that another type of rail pad would have been used for such a short section of track. These results imply that the choice of starting position may have a large influence on the TDR-measurement results.

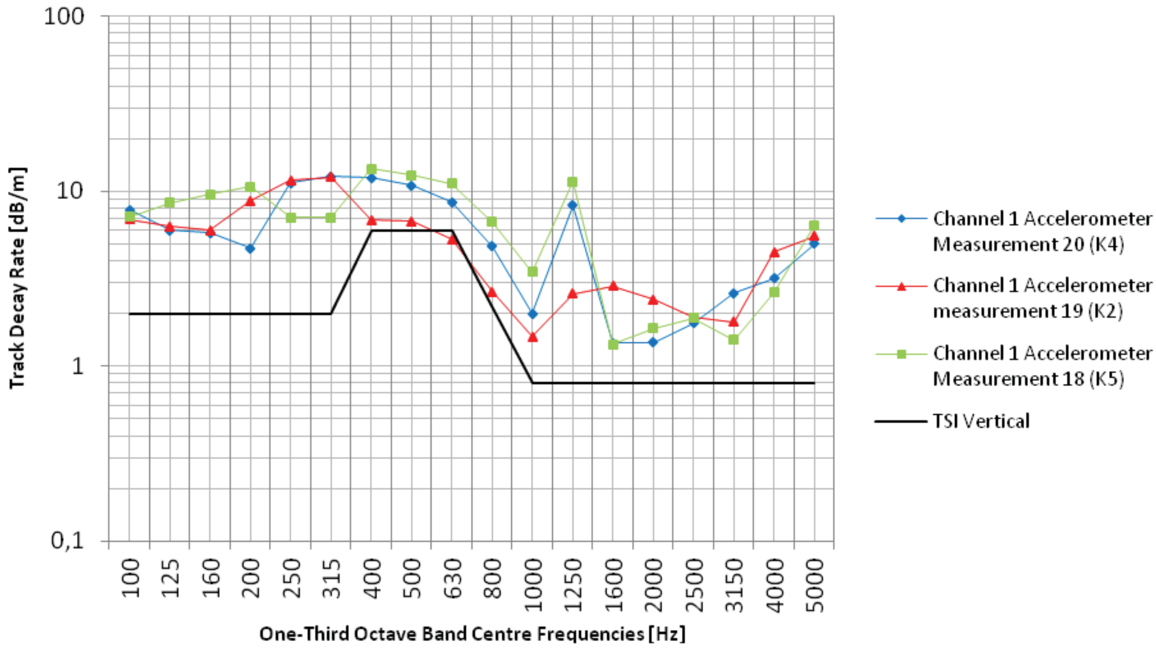


Fig. 34 : *Vertical TDR measured at three different positions along the right hand rail.*

In the lateral measurements, the variation is not as large as in the vertical measurements, see Figure 35. However, the measurements at K4 and K5 are much more similar to each other than they are to K2.

Fig. 35: *Lateral TDR measured at three different positions along the right hand rail.*

6.3.3 Conclusions

The measurement campaign has clearly demonstrated that the method described in standard EN 15461 [1] allows for variations which could affect the results and that there could be unknown sources of uncertainty that need to be investigated further.

The results showed that track decay rate variation over short periods of time and slightly longer periods of time is very small for both vertical and lateral measurement directions but over much greater periods such as several years, comparisons may become invalid. This variation is due to aging of the track parts, the wear caused by the traffic load and maintenance work such as tamping and grinding.

It is clear from most measurements that the uncertainty in measuring TDRs at low frequencies (typically below 400 Hz) is problematic and results are not reliable. This does not cause problems for the STARDAMP method as the focus for the application of rail dampers is above 500 Hz. The reasons are possibly difficulties to excite the rail with enough energy to achieve a sufficient signal to noise ratio at longer distances. The other reason might be the track on ballast resonance in this frequency range.

Mounting and position of the accelerometer particularly influences the vertical track decay rates, with a large, recurring difference between the accelerometer on the top of and the one on the foot of the rail at 2500 Hz. The difference stems from inconsistencies in the direct FRF A(0) measured on the top of the rail. It could be due to dynamic effects of the rail on the track but the cause could not be determined and it also could not be excluded that it is not specific for this track and set up. Since it only concerns the first measurement position of the measurement set it is not likely that the difference is caused by the way of mounting. Removing and replacing the accelerometers was shown to have little effect on the results. It is recommended that further investigations are made to determine the reason for the difference.

The measurements showed a small but clear influence of temperature over a range of frequencies which was consistent with the fact that warm rail pads are softer and thereby cause a lower track decay rate.

The importance of the first measurement position was shown when the striking position of lateral measurements was taken into account. The rail was struck on both the inside and outside of the rail head as well as left and right of the accelerometers on the outside and different results calculated based on these first measurements. Striking on the outside showed a change in the important middle frequencies, and striking left and right of the accelerometers showed that striking on the measurement side was ultimately more in line with other results.

Very large variations were noticed at different starting positions along the rail as well as between measurements of the left and right rail. The logical consequence would be to measure the decay rate at several positions to ensure that the measurement result correctly represents the track section of interest. On the other hand it would mean a significant increase in effort and costs and still it is not clear how to correctly deal with the results. Should a mean value be determined and from how many measurements at which positions? Which indicator would correspond better to the actual noise emission from the track at a specific track section? These questions need further investigation. One idea is to develop a TDR measurement method with a reduced number of FRF-measurement points in order to be able to assess the TDR at several positions without increasing the effort.

The final test was the effect of modifying the grid of measurement positions by adding more excitation points in the near field and excluding the points farthest away from the sensor. The results here showed good similarity but a reduction in the minimum measureable track decay rate, and hence a problem in some frequencies such as 1250 Hz in the lateral direction where the TDR was very low.

6.3.4 Recommendations and best practice rules

Based on the results of the measurement campaign and the subsequent data analysis recommendations regarding the best practice for track decay rate measurements on real tracks have been elaborated. They are collected in a separate report [SR 17]. The most relevant recommendations are:

- It should be recognized that the track decay rate can vary along a track and the measurement results from one position may not be representative for the whole track section of interest.
- According to the standard the TDR-measurement should be repeated at at least two different positions. This requirement should always be fulfilled. Further research will be required in order to give a recommendation regarding how to choose these positions and how to analyse the results.
- If TDR-measurements are to be compared to earlier TDR measured at the same track section, for instance before and after installation of dampers, the same sensor position should be chosen.
- It should be recognized that the measured track decay rate is an estimation of the track dynamics at a certain temperature.
- Measurements that are to be compared with the track decay rate measurement in question (this could be other decay rates or pass-by noise) should be performed at the same rail pad temperature. If the rail pad temperature is above 15°C a maximum deviation of 10°C should not be exceeded. If the rail pad temperature is below 15°C a maximum variation of only 5°C is recommended.
- The rail pad temperature should be measured underneath the clipping, preferably constantly during the whole measurement series.
- A suitable hammer for the excitation would normally have a mass of around 100-300 g and a titanium tip.
- The accelerometer used for the TDR-measurement shall have a mass of less than 10 g and a diameter less than 15 mm. The accelerometer and its mounting must not have a resonance below 10 kHz.
- It is recommended to perform a calibration both before and after the measurement to check the correct function of the whole measurement chain.
- The coherence function should be about 0.9 in the whole frequency range of interest at the first measurement position (direct-FRF).

- It is recommended that the operator performing the TDR-measurements has a certain level of technical skills but it can be sufficient to brief an inexperienced operator on the right technique as long as the coherence is monitored and good.
- For the measurement of the vertical decay rate it is recommended to position the accelerometer underneath the rail vertically on the centre line with an adhesive such as a ceramic or cyanoacrylate glue.
- If a position at the top of the rail is preferred a test should be performed in which results from accelerometer at both positions are compared at the specific test site to prove that no errors are caused.
- Comparisons between different TDR measurement results can only be performed if the same accelerometer position was used.
- The first excitation point should be either on x=0 cm if the accelerometer is positioned underneath the rail or at x=2 cm in the direction of the measurement grid in the case that the accelerometer is positioned on top of the rail.
- The coupling between vertical and lateral motions is very sensitive to the lateral position on the railhead. A reliable result can only be obtained if the accelerometer as well as the excitation points in the vertical direction are on the centre line of the rail.
- An average of at least 4 validated pulses is required for each FRF in the standard procedure. It is to recommend that the direct FRF is determined from an average of 10 impacts to obtain a more reliable result.
- When performing the measurement, the gain of the accelerometer should be adjusted at each excitation point. To achieve good quality measurement results the maximum possible dynamic range should be used and the system background noise should be low.
- An evaluation of the calculated decay rate compared to the actual measured FRF's versus the distance is necessary.
- If single FRF's are found to be erroneous, these should be omitted from the calculation of the TDR. It is also possible to systematically disregard values below 10 dB.

7. Results for wheel dampers

7.1 Measurement protocol wheel dampers

A measurement protocol has been developed by Vibratec within STARDAMP and applied throughout the STARDAMP project measurements. The purpose of this measurement protocol is to provide clear rules for measuring the relevant input parameters for the STARDAMP software (see Fig. 9) for assessing the acoustic performance of wheel dampers. This section gives an overview. The full procedure is reported in [SR 12].

Two measurement procedures are presented to identify the wheel modal parameters (resonance frequencies, modal loss factors and optionally mode shapes), which differ in their degree of complexity:

- The first one is the standard experimental modal analysis (EMA), which requires dedicated software for the post-processing of the measured Frequency Response Functions (FRF). This procedure is generally used in the case of non-conventional wheels (non-axis-symmetric wheel, resilient wheel…). This procedure also delivers information about the mode shapes.
- The second one is a simplified experimental procedure (termed simplified experimental modal analysis SEMA), which only needs measurements of a reduced set of FRF. It is used in the case of conventional wheels (monobloc axis-symmetric wheels), for which a calculated modal basis is also available. No direct information about mode shapes is obtained. Modal damping is estimated from FRFs using the "- 3 dB bandwidth" method.

Before describing the particularities of these two measurement procedures, several recommendations are given that apply to all laboratory measurements on wheels.

7.1.1 Principles

Wheel dampers reduce the noise radiated by the wheel by increasing the modal damping of the most influent wheel modes. Modal damping as well as a possible shift of natural frequencies due to the dampers are extracted from frequency response function (FRF) measurements. An instrumented hammer with titanium tip shall be used to excite the wheel. Accelerometers shall be used to measure the responses at different positions on the wheel.

7.1.2 Data acquisition

It is recommended to use a complete wheelset for measurements instead of a single wheel. Both wheels shall be equipped with dampers. The wheelset should be freely suspended by using elastic ropes or springs. The presence of axle boxes is no constraint for these measurements. The suspension can be fixed on the outer extremities (on the axle boxes if present) or on the axle between the wheels; however, the bouncing mode of the wheelset on its suspension should be below 30 Hz (if elastic ropes are used, a decoupling as low as 2 Hz can generally be achieved).

The ambient temperature shall be between 18 and 25°C during the tests. Tests at lower or higher temperatures may be conducted additionally if the influence of temperature on the damper behaviour is to be investigated.

7.1.3 Tests

An important issue to be controlled is the representativeness of the tested sample. Indeed, the efficiency of a wheel damper can highly depend on its mounting conditions. These do of course not only dependent on acoustical considerations but above all on security. It is important to respect the correct mounting conditions in order to obtain a wheel that is representative for operating conditions.

- Ideally, several samples should be tested to produce mean modal damping values to be used for TWINS computations. If only one wheel set is available, at least both wheels of this wheel set should be measured.
- Tests after dismantling and remounting the dampers can be compared to be sure that the mounting conditions are representative and that the dispersion is not too high.

Certain wheel dampers can exhibit nonlinear behaviour, i.e. their efficiency depends on the excitation level. It is therefore recommended to control the wheel damper behaviour at a 'high excitation level', ideally using a shaker with swept sine excitation and simultaneous control of the injected force, assuring a constant force over the entire frequency band. The 'high excitation level' can be chosen to 50 N RMS (sine) which is close to dynamic levels experienced in real exploitation. If the damper has been shown to behave linearly, the remaining testing can then be performed at lower levels and by the use of an impact hammer.

Similarly to measurements on the rail, coherence shall be recorded and controlled.

7.1.4 Equipment

Concerning equipment, the same recommendations as for rail testing apply. A multi-channel acquisition system (16 channels) is preferable if a complete modal analysis is to be performed. This permits to instrument one complete wheel section with five tri-axial accelerometers plus the channel for the force sensor.

Excitation shall be provided by an instrumented hammer containing an integral force gauge. A suitable hammer would normally have a mass of around 100-200 g and a titanium tip

The accelerometer shall have a mass of less than 10 g and a diameter less than 15 mm. The accelerometer and its mounting must not have a resonance below 10 kHz. This can usually be achieved by attaching the accelerometer using a ceramic or Cyanoacrylate glue.

7.1.5 Determination of FRFs

For the measurement of FRFs ISO 7626-5 shall be used [2]. The same recommendations as for rail testing apply, except for the following parameters:

- A rectangular window shall be used (i.e. no window) for wheels equipped with dampers. An exponentially decaying window can be used for undamped wheels in order to keep the window at a reasonable length. For shaker excitation windowing has to be used (e.g. Hanning).

- The analysis window shall be long enough to capture the whole of the decaying response due to impact. For a strongly damped wheel it is expected that 1 s will be sufficient, implying a frequency resolution of 1 Hz. For less efficient dampers an analysis window of 2 s (leading to a frequency resolution of 0.5 Hz) may be necessary. A sufficient frequency resolution is indeed necessary for the correct estimation of modal damping. The determination of modal damping for wheels without dampers will require a frequency resolution of 0.25 Hz minimum; however, the exact measurement of damping is generally not necessary because default damping values are used for bare wheels. Nevertheless, recording at least a reduced set of FRF for the undamped wheel can help identifying the wheel modes and provide validation of the FE model.

7.1.6 Experimental modal analysis procedure

An experimental modal analysis (EMA) is a procedure that permits to decompose the (measured) vibration response of an object into modes of vibration and to identify frequency, shape and damping of each mode. This has to be performed in a specific software for post-processing of the measured FRF. This procedure also requires an experienced user and its detailed description goes beyond the scope of this report.

Excitation can be provided by either an electromagnetic shaker or an impact hammer. In the case of shaker excitation, the above listed acquisition parameters (windowing, etc) have to be adapted. Hammer excitation, however, is generally sufficient if the above measurement recommendations are respected.

The procedure suggested here is based on the measurement of “direct FRFs” (in opposition to reciprocal measurements). Two excitation positions are used, sufficient for the excitation of radial and axial types. The wheel response is recorded at a more important number of points. The measurement grid has to be chosen sufficiently fine in order to identify all relevant mode shapes. An example is given in Fig. 36 which constitutes a sufficient mesh for a standard wheel.

The use of tri-axial accelerometers is preferable because the post-processing of recorded FRF becomes more straightforward. Meshing of half a wheel is sufficient because of its axis-symmetry. Only one wheel of a wheelset is meshed in detail; the opposing wheel, as well as the axle are meshed in a much coarser way. This permits to identify modes that imply the axle or both wheels simultaneously (e.g. bending modes of the axle or wheel modes that occur twice with both wheels moving in phase or out of phase). Five measurement positions per section are used (two on the tread and three on the web) and 13 sections, resulting in a total of 65 positions. In the given example three measurement points are also placed on each damper (green part of the mesh), however, this is not necessary for the determination of wheel modes and their corresponding modal damping.

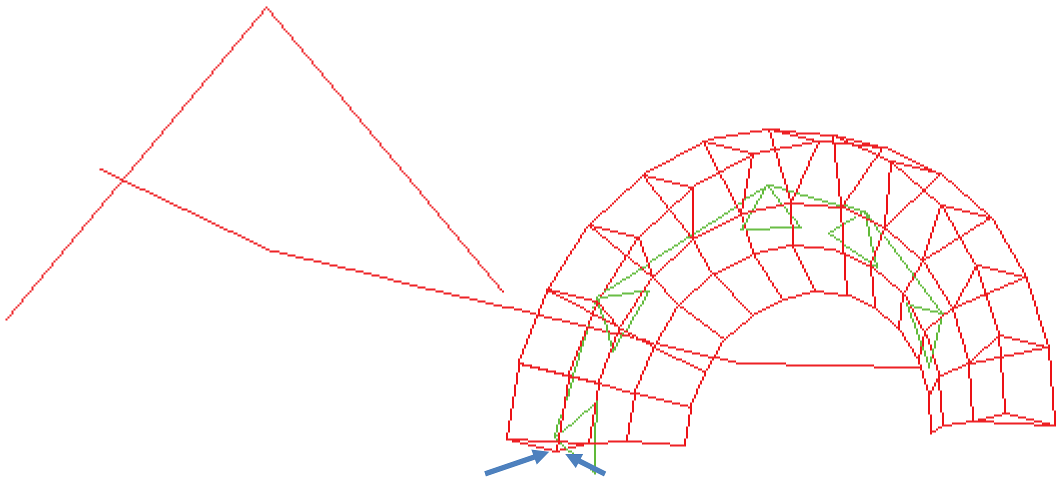


Fig. 36: *EMA mesh example*

Fig. 37: *Position of accelerometers*

Two positions for force excitation are recommended (as indicated in Fig. 36 and Fig. 37):

- 2X (radial excitation at the outer side of the tread, for *Rn* and 1-*Ln* modes),
- 1Y (axial excitation on the tread, for 0-*Ln* modes).

Fig. 37 shows that the radial excitation position is « occupied » by the accelerometer. The excitation position should therefore be shifted in circumferential direction (and not axially!).

Similarly, the tri-axial accelerometer could be fixed laterally at position “1Y”, in which case the axial excitation has to be shifted in circumferential direction.

The resulting FRF are analysed with dedicated software which allows an extraction of modal parameters (notably damping). Thanks to the experimental mesh, mode shapes can be visualised and mode types can be identified.

An EMA delivers information about wheel natural frequencies, mode shapes and damping. Measured mode shapes are generally not used for rolling noise computations, however, but assumed to be identical to the computed shapes of the undamped wheel modes. The main benefit of an EMA is the better estimation of modal damping compared to the simplified experimental modal analysis (SEMA).

7.1.7 Simplified experimental modal analysis procedure

For most wheel types, a limited measurement procedure is sufficient. It is based on the fact that the mode shapes of a wheel are known to be quite regular, and that the computed modal basis - usually very reliable - is already known. Also, the wheel is assumed to be axis-symmetrical, an assumption that is equally made in TWINS and the STARDAMP software.

In order to identify the modal parameters (natural frequencies and modal loss factors), the wheel is excited by an impact hammer, and a single accelerometer is used to measure the response. The obtained transfer functions are analysed to derive the modal parameters.

The excitation and response positions are chosen so that for a given class of modes, a distinct set of peaks can be made out on the transfer function. This is completed by other transfer function measurements, and finally this procedure allows to separate 0 - axial, 1 - axial and radial modes, and to determine if the number of nodal diameters is even or odd.

The measurement positions are presented on Fig. 39. Table 8 summarises the categories of modes that can be identified on the measured FRF, by analysing modulus and phase of the transfer functions.

With such information and the comparison with the calculated modal basis, it is possible to identify the mode shapes and the resonance frequencies of the main modes accounting for rolling noise, up to at least 3.5 kHz (Radial mode R*n* and 1-axial modes 1-L*n*). The 2-L*n* modes cannot directly be identified with this procedure, but they are of less importance for the rolling noise level. When the calculated modal frequencies are sufficiently close to the measurements, however, one might identify the 2Ln modes simply due to their frequencies.

The modal loss factor (η = 2 C/Cc, i.e. twice the critical damping ratio) is determined from the ‘half power bandwidth’ method. This method permits to estimate damping from the peaks in the frequency response, as given in the following relation:

$$\eta = 2\,\frac{c}{c_{crit}} = \frac{\Delta f\ (3dB)}{f}$$

Δf (3dB) defines the bandwidth Δf that is read 3 dB below the peak (see Fig. 38).

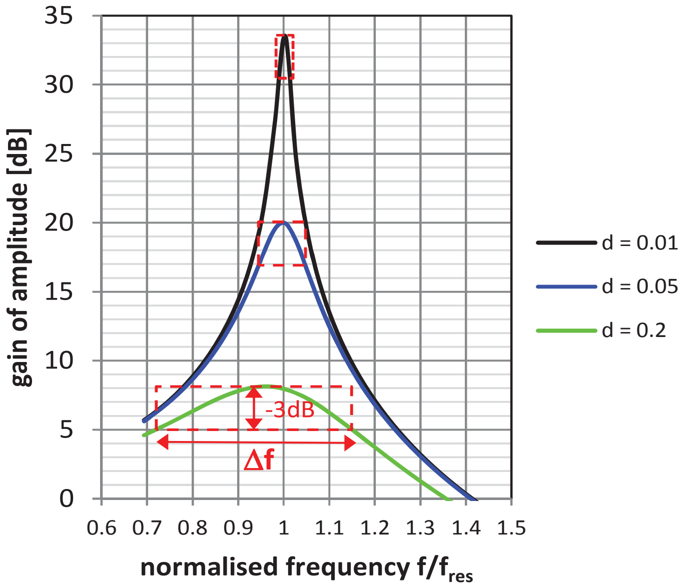


Fig. 38: *'-3dB' method for different damping ratios (η=2d)*

As outlined earlier, care must be taken on the choice of the number of frequency lines in order to measure correctly the loss factor. Remember, however, that the measurement of bare wheels' loss factors are of little interest. Nevertheless, the measurement of a bare wheel can be very useful for the identification of modes. From a damping point of view, only the measurement of the wheel equipped with dampers is important. The frequency resolution is less critical then.

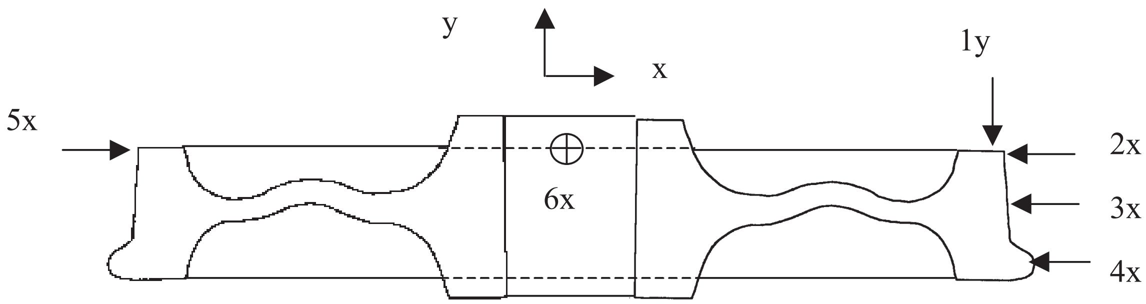


Fig. 39: *Nomenclature of excitation/ response points for the simplified modal analysis (SEMA)*

Table 8: Simplified experimental modal analysis (SEMA): measurement points and analysis procedure

FRF		**Analysis of the FRF**
Excitation position	Meas. Position	
1Y	1Y	Emergence of the 0 axial modes (0-*Ln*, *n* being the number of nodal diameters).
2X	2X	Emergence of radial modes (*Rn*) and 1 axial mode (1-*Ln*).
4X	2X	Emergence of *Rn* and 1Ln modes. To separate these modes, this FRF (4X - 2X) is compared to the FRF (2X - 2X), and the values of the phase are compared for a same resonance frequency : – if the phase is the same, points 2X and 4X move in phase : this characterises a radial mode, – if the phase shift is close to 180° C, points 2X and 4X are out of phase : this characterises a 1-axial mode.
5X Point 5X has the same position on the tread than point 2X, but on the opposite section (180° shift)	2X	Comparison of the phase for FRF (2X - 2X) and (5X - 2X) close to a resonance frequency : – same phase : the number of nodal diameters is even, – 180° phase shift : the number of nodal diameters is odd.
3X	3X	Position 3X is close to a node of vibration of 1-*Ln* modes: emergence of the *Rn* modes.
6X	2X	Comparison of the phase for FRF (2X - 2X) and (6X - 2X) close to a resonance frequency : – Allows to separate *n*=2 and 4 modes from *n*=0 modes.

7.1.8 Test report

The following elements shall be presented in the test report:

- The precise positions of the accelerometers and excitation points;
- Description of the test wheel;
- Description of the dampers, including types, component masses, dimensions and serial no. or other means of identification;
- The ambient temperature

- The manufacturer, types and serial no. or other means of identification of the accelerometer, impact hammer and spectrum analyser used;
- A table summarising the identified modes in terms of frequency and damping.
- FRF plots with receptances (or accelerances) of the damped as well as undamped wheel. Two radial point receptances (outer tread “2X” and mid tread “3X”) and one axial point receptance (“1Y”) shall be given.

7.2 Round robin test for wheel dampers

In order to validate the procedures explained in section 7.1 and to generate input data for testing the STARDAMP software tool, a round-robin test was organized. Vibratec, Schrey&Veit, Alstom, and TU Berlin performed tests on the BA308 wheel and the LK 900 wheel (both single wheels and wheel sets were used – see also section 5.3.3).

7.2.1 Set up of test rigs and measurements

Different setups were used by the partners to mount the wheels in order to carry out the modal analysis. In Fig. 40 examples of a set up for a wheel set (TU Berlin) and a single wheel (Alstom) are given.

Fig. 40: *Examples for test setup (top: wheel set – TU Berlin ; bottom : single wheel – Alstom)*

The following two tables give an overview of the measurement equipment and relevant parameters used by the partners.

Table 9: Measurement equipment used by the partners in the round robin test

	Vibratec	ATSA	TUBerlin	SuV
Accelerometers				
Type	ICP	ICP	Piezoelectric charge accelerometer	ICP
sensitivity	10 mV/g or 100 mV/g	1 mV/(m/s²) ≈ 10mV/g	31 pC/g	100 mV/g
Mass	10 g or 6 g	15 g	17 g	18 g
attachment method	Ceramic or Cyanoacrylate	Cyanoacrylate	Ceramic	strong magnetic plate
Force hammer				
Type	ICP	ICP		ICP
sensitivity	1.1 mV/N or 0.23mV/N			1.64 mV/N
Mass	160 g or 320 g	320 g		150 g

Table 10: Measurement parameters

	Vibratec	ATSA	TUBerlin	SuV
FRF #impacts	5	4	??	≥ 8
Sample rate	12.8 kHz	48 kHz	??	32.77 kHz
Bandwidth	6.4 kHz	6 kHz	6 kHz	6.4 kHz
Acquisition duration without damper with damper	1.28 – 2.0 s	10s 5s	≈ 2.6s	4s 0.25s
Pre-trigger	10 ms			5ms
Resolution	0.5- 0.78Hz	0.183 Hz	0.381 Hz	0.25Hz undamped / 4 Hz damped

7.2.2 Measured accelerances

Each partner measured accelerances for the bare wheels and the wheels/wheelsets equipped with dampers. Examples are shown in Fig 41 and Fig. 42 for the LK900 wheel. The full documentation of all results is contained in the measurement reports [SR 11], [SR 12], [SR 15], [SR 24]. The spectra in Fig 41 allow for an easy estimation of the resonance frequencies of the bare wheels. Almost identical resonance frequencies were detected by all partners for the dominant eigenmodes, whereas for weaker resonances the results may diverge. It is assumed that uncertainties in the exact location of impact point and accelerometer are the cause.

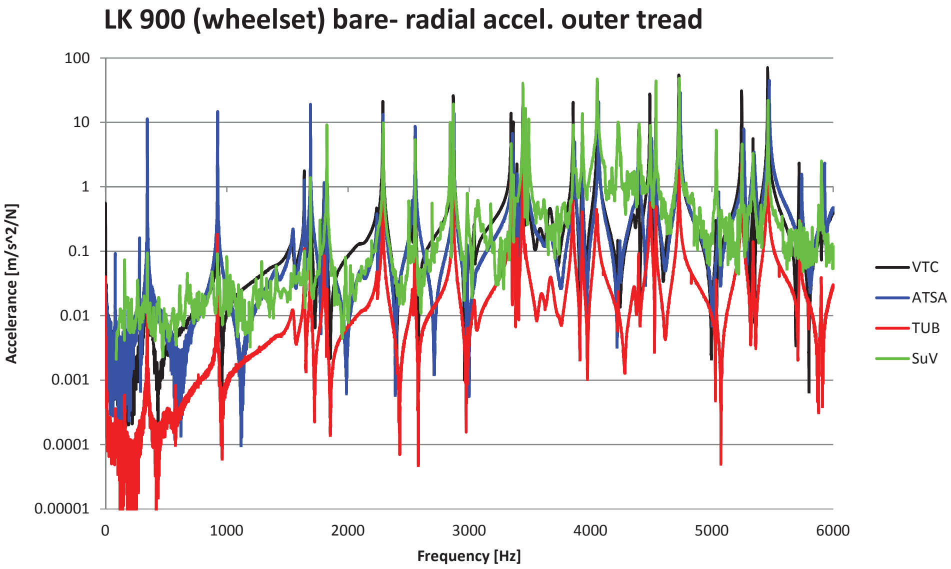


Fig. 41 : *Comparison of the radial accelerances measured by the STARDAMP partners in the round robin test for wheel dampers. Wheelset LK900 without damper.*

Fig. 42 contains the corresponding spectra for the LK900 wheel equipped with VLN ring damper. By mounting wheel dampers the resonances get broader due to the increasing damping. In addition to this it seems that the positions of the accelerometer and the driving point of the impact hammer are getting more important.

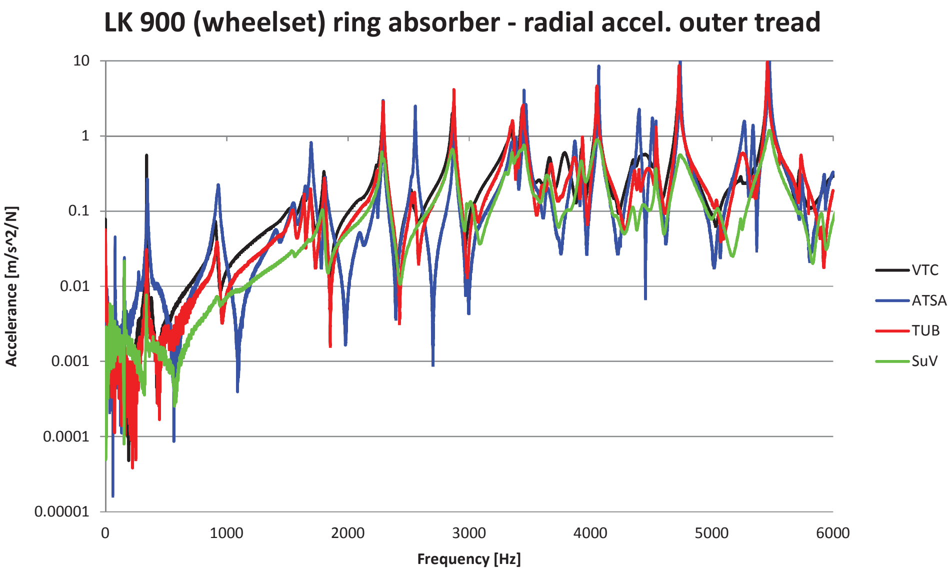


Fig. 42: *Comparison of the radial accelerances for wheelset LK900 with VLN ring damper.*

For the axial accelerances, the obtained data differ less between the different laboratories than the radial accelerances do.

7.2.3 Calculation of the damping ratios

In the section 7.1.7 the '-3 dB method' has been described for estimating damping ratios of the eigenmodes. With wheel dampers of high efficiency it can be difficult to properly define these -3dB levels (see example in Fig. 43).

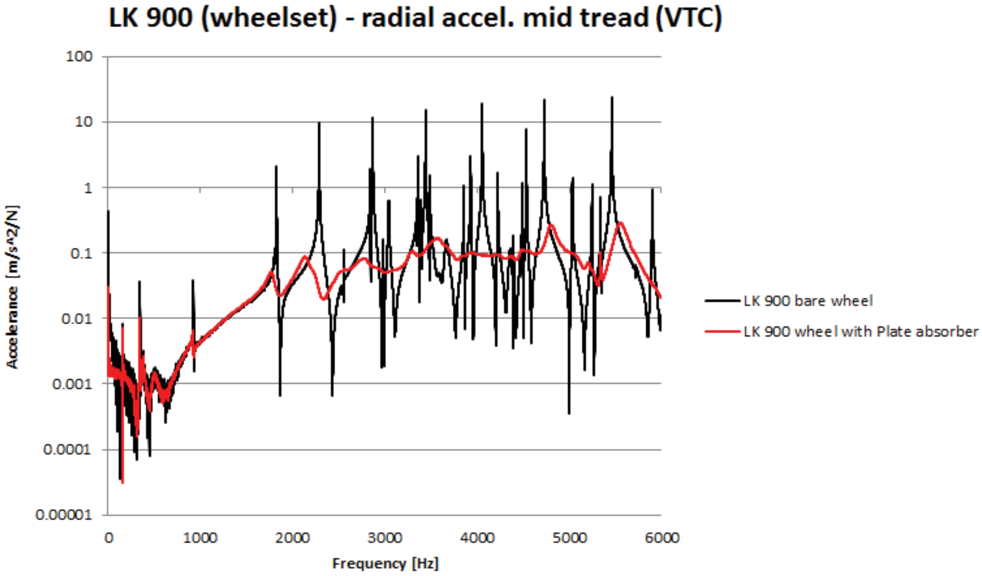


Fig. 43: *Example fo high damping (LK900 wheel with plate damper – red curve)*

The partners in the round robin tests either calculated the width Δf directly from the frequency plot of the accelerances or by a dedicated algorithm (Alstom). The following bar diagrams present examples for the Ln-modes (=axial modes) and Rn-modes (= radial modes) of the damping ratios calculated by the participants in the round robin test (wheel LK900 with ring damper and with DAAVAC damper, respectively). It should be noted that modes that are missing in the bar plots are modes that have not been identified at all (i.e. modes that are highly damped in reality !). The full results are presented in [SR 20].

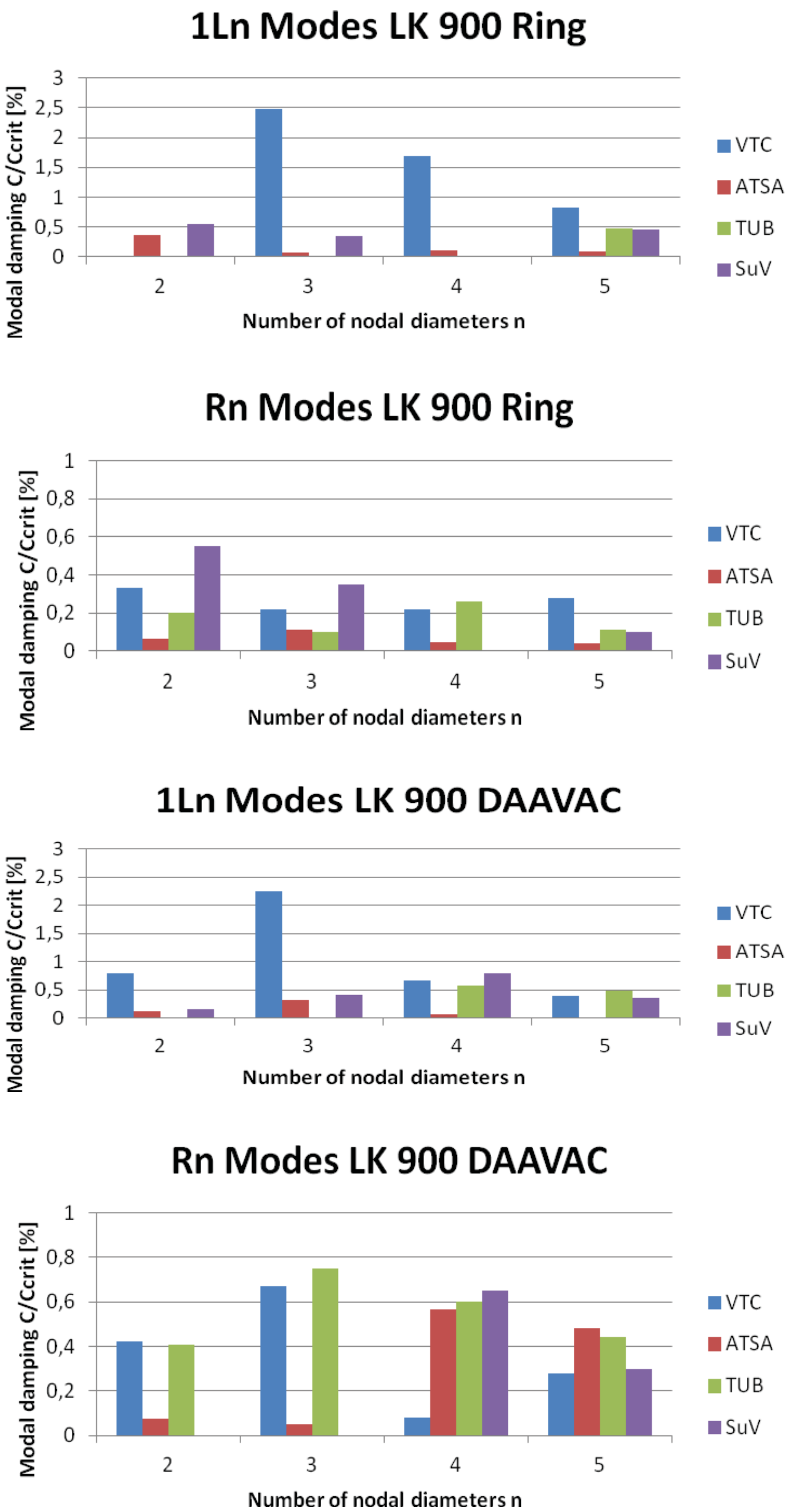


Fig. 44: *Modal damping measured by the respective partners in the round robin test for wheel dampers for Ln-modes (=axial modes) and Rn-modes (= radial modes) (wheel LK900 with ring damper and with DAAVAC damper).*

The results show that

- although SEMA is an easy to use method of analysing the damping ratios of wheel dampers it requires that measurements are carried out with high accuracy. This especially holds for the 1Ln-modes and Rn-modes.
- accelerances of highly damped wheels can be difficult to be analysed by the half power bandwidth method especially for 1Ln-modes and Rn-modes, therefore a full EMA is preferrable for highly efficient dampers.
- identified modal damping has to be adjusted by fitting calculated and directly measured receptances, which can be performed using the mp-editor (see also next section). This updating process also permits to find appropriate damping values (through an iterative process) for modes that have not been identified at all in the SEMA.

7.2.4 Application of the STARDAMP tool

In a second step of the round robin test for wheel dampers, the achievable noise reduction in pass by noise of a train was calculated by the STARDAMP software tool. Therefore, the complete procedure including modal data updating by the mp-editor has been performed by Alstom and Vibratec for the different dampers of the LK900 wheel. Rolling noise computations by the STARDAMP tool have been carried out with each of the obtained wheel modes files in order to determine the effect of the damper.

The reduction in radiated sound power is displayed in Fig. 45. On the left the reduction in sound power radiated from the wheel only is displayed and on the right side the total reduction, which also takes into account the contribution from the rail. The obtained wheel sound power reduction is quite close for the DAAVAC damper and the plate damper, leading to virtual identical overall noise reductions. The discrepancy is somewhat larger for the VLN ring damper. One possible reason is the mounting condition (clamping torque). The discrepancy in overall noise reduction is of only 0.5 dB(A) however, which is due to the dominating track contribution in the current example. As the dampers were not tuned to the wheels the obtained results do not allow a comparison of the different dampers, but only show the effect of different laboratory measurements.

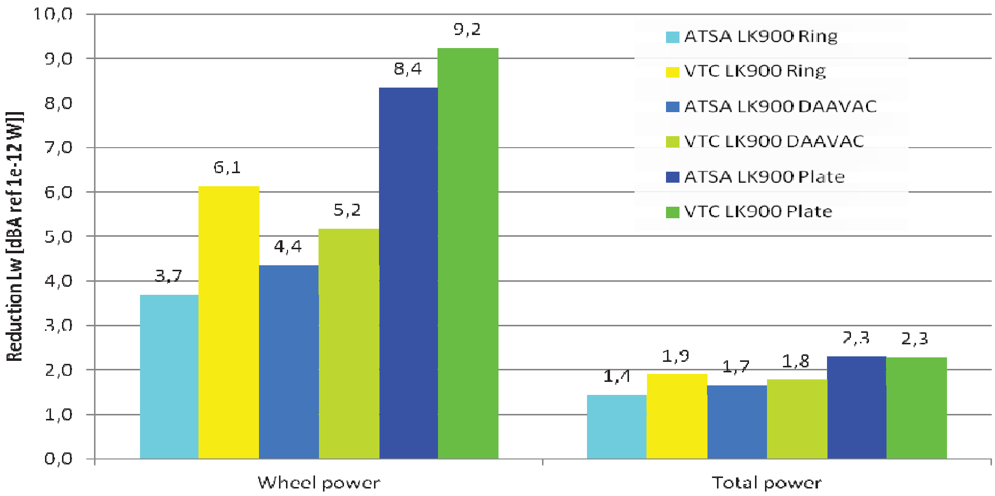


Fig. 45: *Reduction of sound power (right: wheel only; left: total wheel + track) calculated for the different dampers attached to the LK 900 wheel: comparison of results obtained by Alstom (ATSA) and Vibratec (VTC) results*.

7.2.5 Conclusion

With the simplified experimental modal analysis SEMA a method is presented which allows in combination with the STARDAMP software a prediction of the noise reduction of wheel dampers based on a reduced number of laboratory measurements.

Quantification of the damping ratios can be achieved by measuring frequency response functions at certain locations of the wheel separating the axial and radial eigenmodes, followed by an analysis of the damping ratios by the half power bandwidth method.

Although the SEMA method reduces the effort compared to a full experimental modal analysis, the measurements have to be carried out with high accuracy to achieve reliable data sets. It is strongly recommended to repeat measurements at least three times with remounted dampers to cover

- changes of the damping efficiency due to mounting effects of the wheel dampers
- influence on the measured FRF´s and subsequent calculated damping ratios due to shifted impact locations and/or locations of the accelerometer

Especially for wheel dampers with high damping ratios it is hard to detect all damped eigenmodes and to quantify the input parameters for the mp-editor. Further improvement of the accuracy with which damping ratios of strongly damped eigenmodes can be determined is recommended.

It is necessary to compare calculated receptances (i.e. calculated using the FE-model that has been updated with measured frequencies and damping) with directly measured receptances. This can conveniently be done using the mp-editor. Generally some discrepancies are observed that can be minimised by correcting the initial damping values (frequencies are most often correctly detected). This procedure is repeated until a sufficient fit is obtained (see example in Fig. 46 and Fig. 47). A full EMA gives a better 'initial guess' than a SEMA, however, the control of receptances remains necessary.

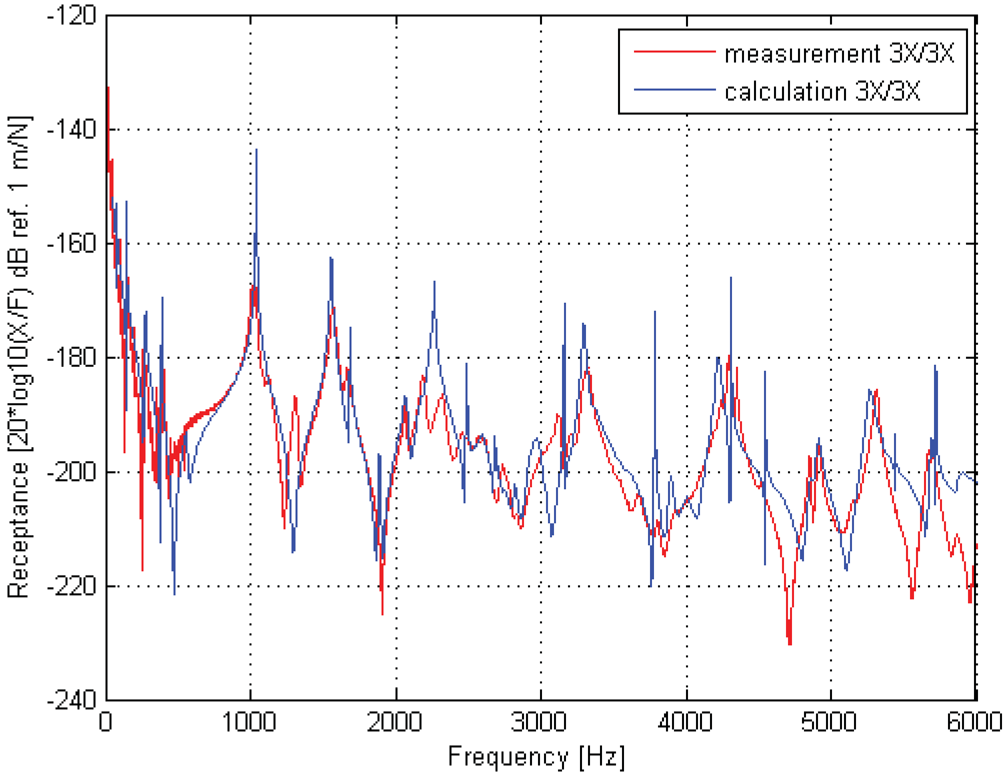


Fig. 46: *Comparison of measured and calulated radial receptance* ***before*** *correction of damping values (wheel BA 308).*

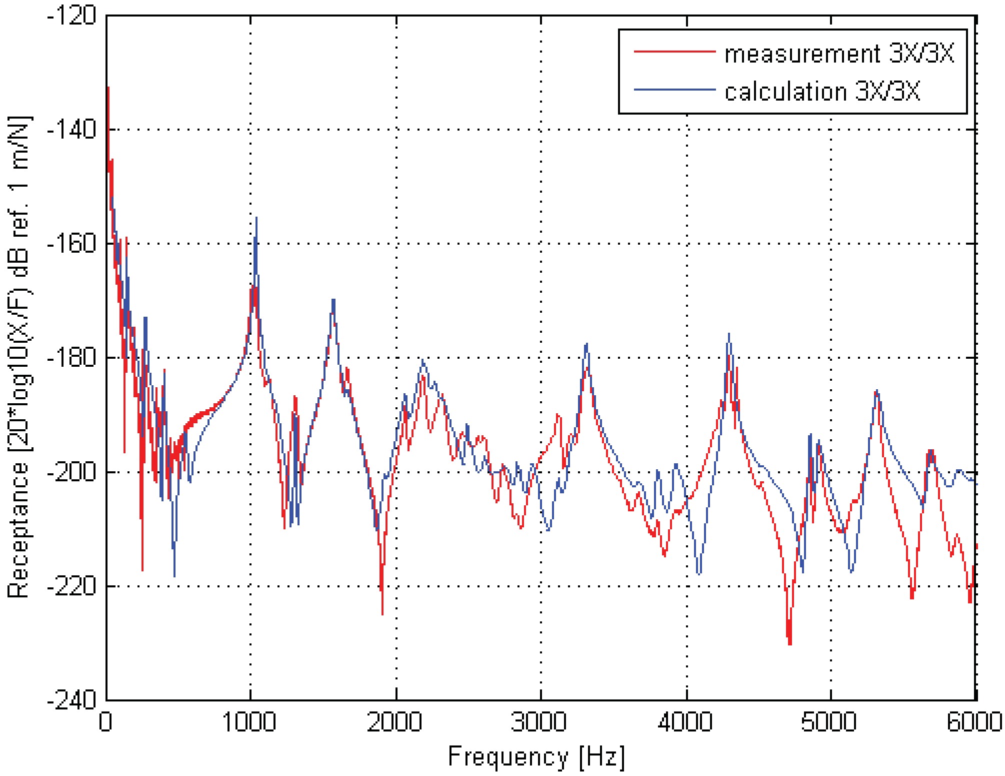


Fig. 47: *Comparison of measured and calulated radial receptance* ***after*** *correction of damping values (wheel BA 308).*

7.3 The IRSID measurement method

The IRSID method combines acoustic and vibration measurements with calculations. Though originally not included in the STARDAMP work programme, it was decided in the starting phase of STARDAMP to include it in the round robin test for wheel dampers and to explore the potential of the method. It can provide important information about the impact of the damping system on the screening effect. In this way, the IRSID method and the EMA method are complementary.

7.3.1 General procedure

The procedure to be followed is explained in [SR 16]. Measurements provide information on the wheel impedance values (more accurate for damped structures than on bare wheels). The calculation integrates the damping effect of the wheel/rail contact that cannot be considered in the rig test. The procedure takes into account the parameters and the measurements of the wheel but also those of the rail to give the value of acoustic power emission of the wheel / rail system. Rail parameters can be taken from an existing data base.

Vibration measurements are performed in direct and crossed direction. Acoustic measurements take place in a reverberant cabin with an excitation in the axial and radial direction applied by a shaker system. The vibration and acoustic measurements are made on the wheel set in free conditions, i.e. without any contact with the ground.

7.3.2 Vibration measurements

The excitation of the wheel is realized by a shaker system. The force is measured by a force sensor attached to the wheel tread or on the outside face of the wheel rim according to the direction of excitation (radial or axial direction). The acceleration is measured by accelerometers positioned on the tread and on the outside face of the wheel rim to obtain the axial and radial structure answers (Fig. 48). Four transfer functions (direct and crossed) are then obtained after measurements.

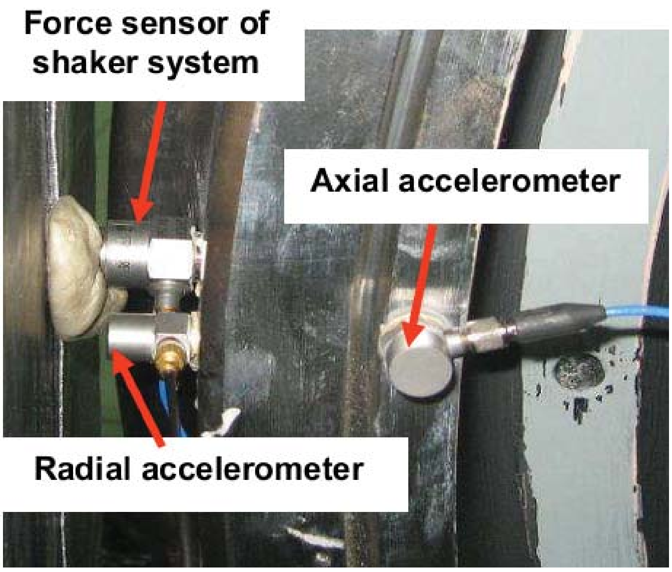


Fig. 48: *Positions of the force sensor and the accelerometers on the wheel*

7.3.3 Acoustic measurements

Acoustic measurements are helpful to see the screening effect of the damping system. The sound pressure is recorded at different locations by microphones distributed around the wheel in a reverberant cabin (Fig. 49) according to the EN ISO 3741 [7] standard.

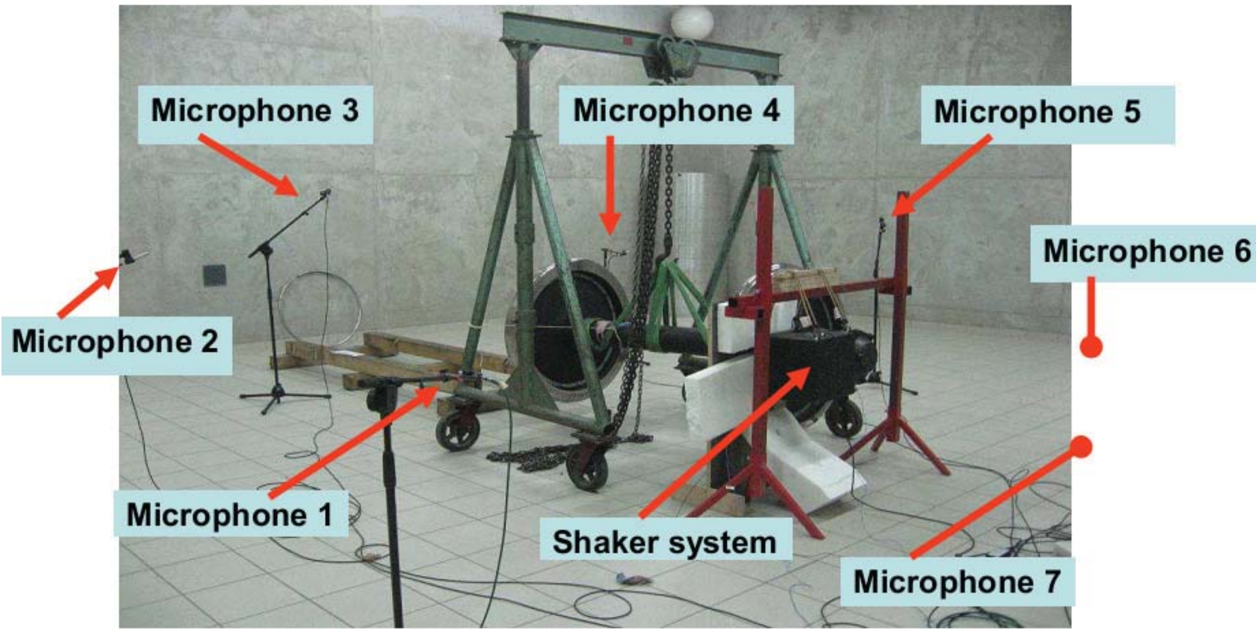


Fig. 49: *Locations of microphones around the wheel in the reverberant cabin*

7.3.4 Results obtained in STARDAMP

The IRSID method was applied within the STARDAMP round robin test to the LK900 wheel set with and without wheel damper:

- Bare LK 900 wheel as reference for the noise emission.
- LK900 wheel set VLN ring damper
- LK900 wheel set with Daavac damper
- LK900 wheel set with plate damper

Measurements were carried out by Valdunes in cooperation with the INSA National Institute of applied science, Lyon. Only a brief overview will be given here. The full documentation is contained in [SR 16]

Six measurements were performed for each configuration:

1. Axial vibratory excitation and measurement in the axial direction.
2. Radial vibratory excitation and measurement in the radial direction.
3. Axial vibratory excitation and measurement in the radial direction.
4. Radial vibratory excitation and measurement in the axial direction.
5. Sound pressure measurements at several locations around the wheel in the reverberant cabin with an excitation in the radial direction by a shaker.
6. Sound pressure measurements at several locations around the wheel in the reverberant cabin with an excitation in the axial direction with a shaker.

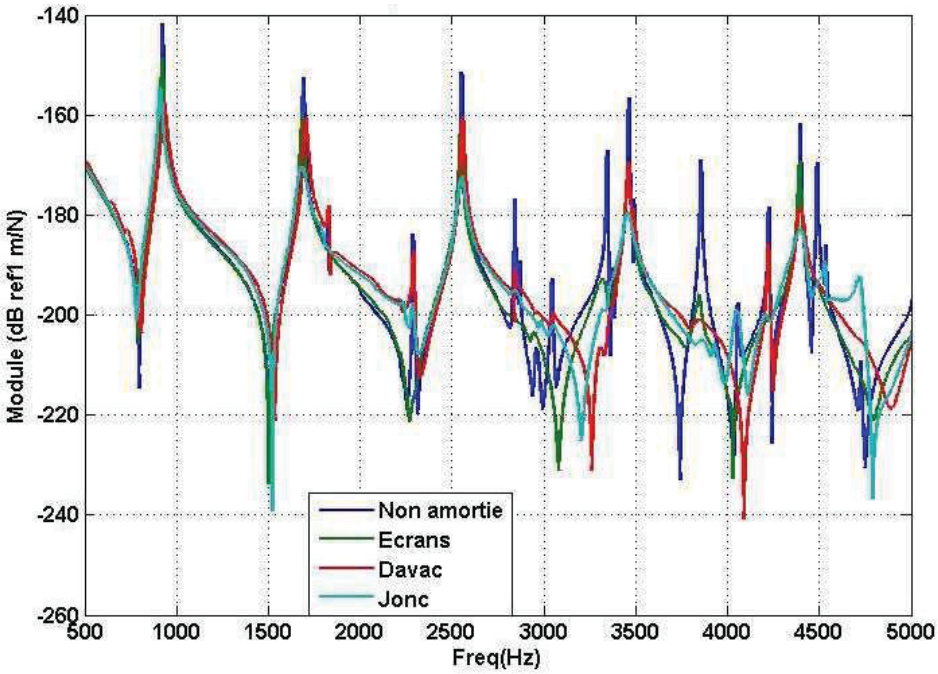


Fig. 50: *Axial /axial receptance of the damped wheel ([SR 16]). Dark blue line: bare wheel; green line: wheel with plate damper; red line: wheel with Davaac damper; light blue line: wheel with ring damper.*

Fig. 50 shows as an example measured axial/axial receptances for the bare LK900 wheel and the LK900 wheel with the three different damper types. No big difference between the the different damping systems is observed. The sound power levels of the three systems with damper is globally the same. Differences become more obvious in the one-third octave band spectra.

A global analysis according to IRSID calculation gives the following predictions for the total noise reduction obtained by a damped wheel with regard to the bare one (frequency range 500 Hz to 4000 Hz, speed of 120 km/h assuming a realistic combined wheel / rail roughness):

- Wheel with plate damper = - 3,39 dB(A)
- Wheel with Davaac damper = -2,68 dB(A)
- Wheel with VLN ring damper= -3,08 dB(A)

More details are reported in [SR 16]. As the dampers were not tuned to the wheels the obtained results do not allow a comparison between the different dampers, but results are only obtained to compare the IRSID and the STARDAMP methods.

7.4 Roller rig measurements

7.4.1. Method

A procedure for testing rail dampers, which may be considered as being between field tests and pure laboratory testing, is based on measuring the noise emission from wheels on roller test rigs. On roller test benches, the rail is replaced by a wheel ('rail-wheel') with diameter considerably larger than the diameter of the wheel ('wheel-wheel') to be tested.

Since previous investigations have demonstrated the general suitability of test rigs to quantitatively measure the noise emission from wheels, it was decided to include roller rigs in the STARDAMP project as a testing method which is in between pure field tests and the EMA + TWINS method. With respect to acoustic investigations, roller test rigs may be seen as a laboratory test method that includes elements of field tests (directly measuring the noise emission of wheels, which are excited by roughness).

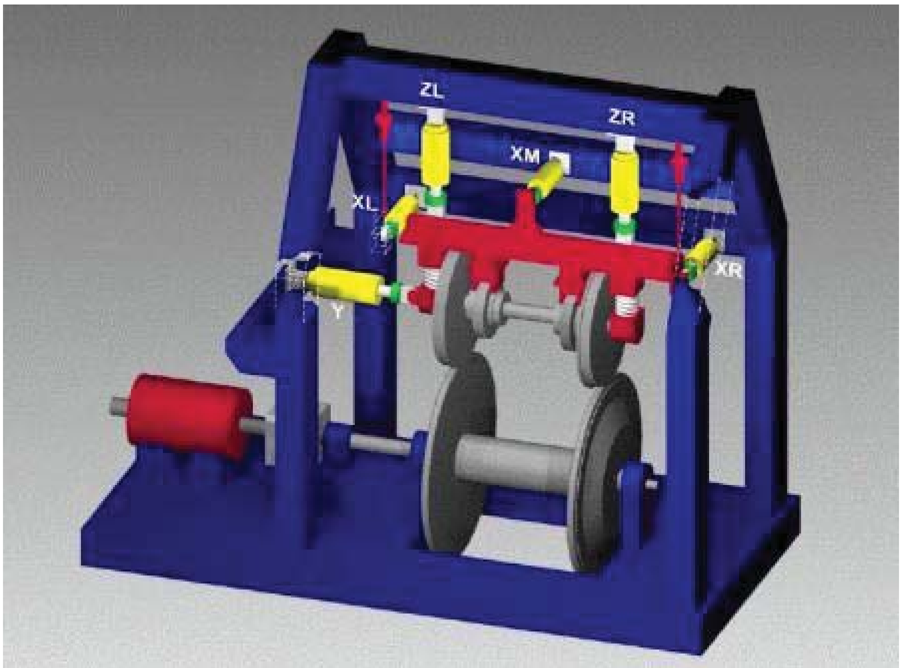


Fig. 51: *General setup of the RASP roller rig*

Within the frame of a measuring campaign in October 2012, the two wheelsets provided by GHH Radsatz equipped with different kinds of dampers were tested on the RASP ('Rad-Schiene Prüfstand') of DB Systemtechnik (see Fig. 51).

The following wheel/damper combinations were used:

- BA308 without damper
- BA308 with damper Vicon-Rasa HT (Schrey & Veit)
- LK900 without damper
- LK900 with ring damper (Valdunes)
- LK900 with DAAVAC damper (Pinta-Enac)
- LK900 with plate damper (GHH Radsatz)

Wheelset BA 308:

Wheelset BA308 has been tested in combination with wheel dampers of type Vicon-Rasa HT (Fa. Schrey & Veit).

Fig. 52*: Left: Wheel BA 308 equipped with wheel damper of type Vicon-Rasa (Schrey und Veit); Right: Individual damper assemblies. Each wheel needs three assemblies to cover the full circumference.*

Wheelset LK900

Wheel LK900 was tested equipped with the ring damper (Fig. 53 center) and with the plate damper (see Fig. 53 right). The plate damper consists of metal plate segments separated by an elastomeric layer.

Fig. 53: *Left: Wheel LK900 with the DAAVAC wheel damper (Pinta-Enac); center: ring damper (Valdunes); right: plate damper (GHH Radsatz).*

7.4.2 Measurements on RASP

Roughness measurements

Like in the case of a wheel rolling on a real track also on the test rig the roughness of rail and wheel is the excitation mechanism responsible for rolling noise. The roughness of test wheels and rail-wheel has been measured (see Fig. 54 for the measurement setup).. For both the BA308 wheels and the LK900 wheels the roughness has been measured after installation on the test rig, right before conducting the acoustic measurements. The roughness measurement was repeated after finishing the tests for the respective wheel sets in order to monitor changes, which might have occurred during the test runs. The BA308 wheelset was produced for acoustic measurements in the laboratory during the LZarG project. It had never run on a real track but had already been used on the RASP test rig within LZarG. Hence, the initial roughness was not the one of a completely new wheel.

Fig. 54: *Sensor of the wheel roughness measuring device on the running surface of a test wheel.*

The surfaces of the wheels are very smooth. Particularly at longer wavelengths the measured spectra are far below the curve from the ISO standard. The roughness is almost independent of wavelength up to λ ~15 cm. This indicates that the noise radiated from the wheels should have only a weak dependence on velocity. The roughness changed slightly between the initial and final measurement.

Measurements of sound pressure levels

Measurements of rolling noise were performed by placing two microphones at positions denoted by MP1 and MP2 in Fig. 55. MP1 was the microphone position close to the test wheel. It was located in a distance of 30 cm from the wheel and 30cm above the ground (the vertical position of the rail-wheel contact is almost at ground level). MP2 was used to monitor the noise from the rail-wheel. It was located at a distance of 60 cm from the rail-wheel and 93 cm below the vertical position of the rail-wheel contact.

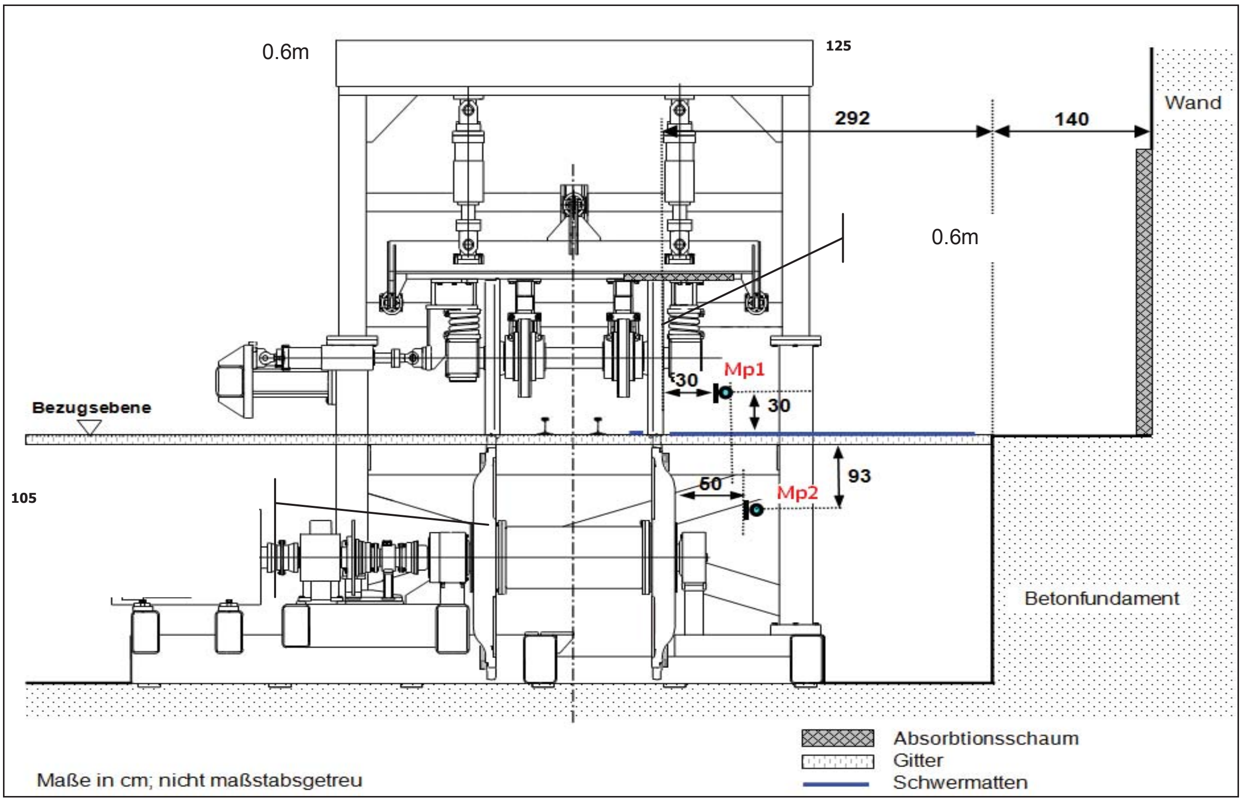


Fig. 55: *Position of the two microphones. Mp1 is close to the test wheel while Mp2 is located close to the rail wheel.*

In a first series of acoustic measurements, a thorough quantification of the background noise was performed. The background noise mainly consisted of contributions from fans, the hydraulic system and engine noise from the motors driving the 'rail-wheel'. It could be shown that the background noise had its main contribution at frequencies below 400 Hz and, hence, in a frequency range, which is not relevant for the sound radiation of wheels.

For each of the six wheel/damper combinations (see Section 7.4.1), acoustic measurements with four different vertical wheel loads and for three different speed values were carried out. The vertical load ranged from 50 kN per wheel up to 100 kN per wheel, speeds of 80, 100 and 120 km/h were used. These values for vertical load and speed cover the typical ranges for freight traffic. Table 11 lists the parameter combinations used when testing each wheel/damper combination. Additionally, the sound pressure levels were monitored while slowing down from 120 to 80 km/h. This is denominated by "Ramp" in Table 11. These speed scans can be used to check if resonances occur at certain speeds. Hence, a total of 16 acoustic measurements were carried out for each wheel/damper combination.

Table 11: Parameter combinations for vertical load and speed applied for each wheel/damper combination

Vertical load	Speed in km/h
2 x 50 kN	80
2 x 50 kN	100
2 x 50 kN	120
2 x 50 kN	Ramp
2 x 65 kN	80
2 x 65 kN	100
2 x 65 kN	120
2 x 65 kN	Ramp
2 x 80 kN	80
2 x 80 kN	100
2 x 80 kN	120
2 x 80 kN	Ramp
2 x 100 kN	80
2 x 100 kN	100
2 x 100 kN	120
2 x 100 kN	Ramp

The following figures show examples of the acoustic measurement results. For full documentation see [SR 19]. In Fig. 56 the measured effect of the different wheel dampers attached to the LK900 wheel is displayed. The curves in the upper half show the measured sound pressure levels for the bare LK900 wheel and for the wheel equipped with DAAVAC damper, ring damper, and plate damper, respectively. The effect of the dampers becomes better visible when calculating difference spectra. These are shown in the lower part of Fig. 56.

All measurements in Fig. 56 have been performed with speed v=80 km/h and vertical load Fz=100kN.. At higher frequencies ring damper and plate damper reduce the noise by up to 12 dB in certain 1/3 octave bands. The effect of the DAAVC damper is considerably lower. The noise reduction effect of the DAAVAC damper is smaller reaching about 5 dB in the relevant frequency range. No noise reduction was found below 1500Hz for all dampers.

The second example in Fig. 57 contains sound pressure measurements with the pure BA308 wheel and with the BA308 wheel + block damper for different speeds (v=80 km/h, 100 km/h, 120 km/h). Sound pressure levels increase slightly with increasing speed. Noise reductions up to 15 dB in single 1/3 octave bands were measured independent on speed. No noise reduction was observed below 1000 Hz within the accuracy of the measurements. However, the BA308 wheel has a dominant vibration mode (‘R2 mode’) at a frequency of about 1050 Hz, which is clearly visible in the spectra in Fig. 57. This could indicate that the damper is tuned to a frequency range that is too high for this particular wheel.

Similar results were found when varying the vertical load keeping the speed constant. The total noise emission decreases slightly with increasing vertical load due to the increased damping of the wheel. On the other hand is the performance of the damper is independent of the vertical load.

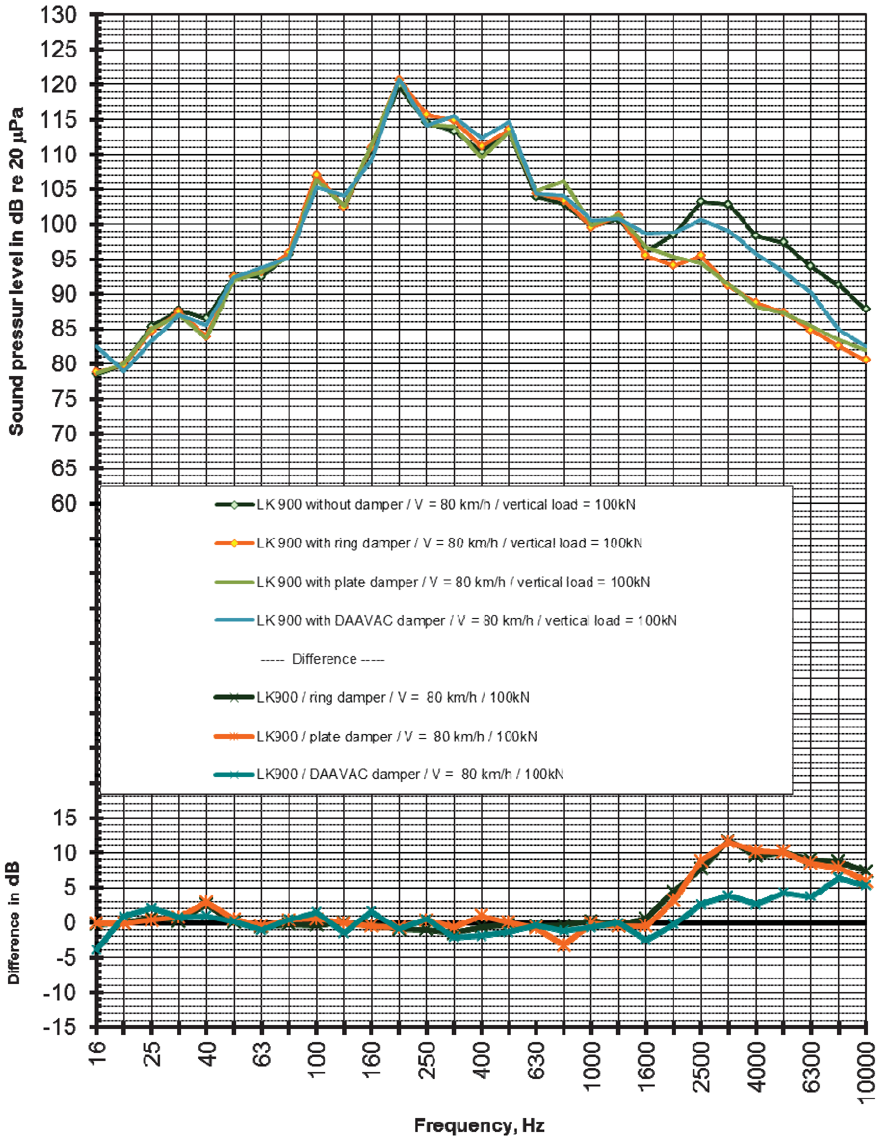


Fig. 56: *Comparison of the noise reduction obtained by the three damper types 'ring', plate', 'DAAVAC'. The upper half shows the spectra for the bare LK900 wheel and for the LK900 wheel with the three damper types. The curves in the lower half show the differences. Vertical load = 100 kN, speed = 80 km/h.*

The direct comparison of the LK900 and the BA308 wheel is shown in Fig. 58. The sound pressure levels of LK900 are considerably below those of the BA308 wheel in the frequency range from 500 to 2000 Hz. This is particularly caused by the high frequency of the dominant R2 mode (LK900: fR2=2266 Hz; BA308: fR2 = 1041 Hz). Differences up to 6 dB were measured. Outside of this frequency range the acoustic behaviour of both wheel types is almost identical.

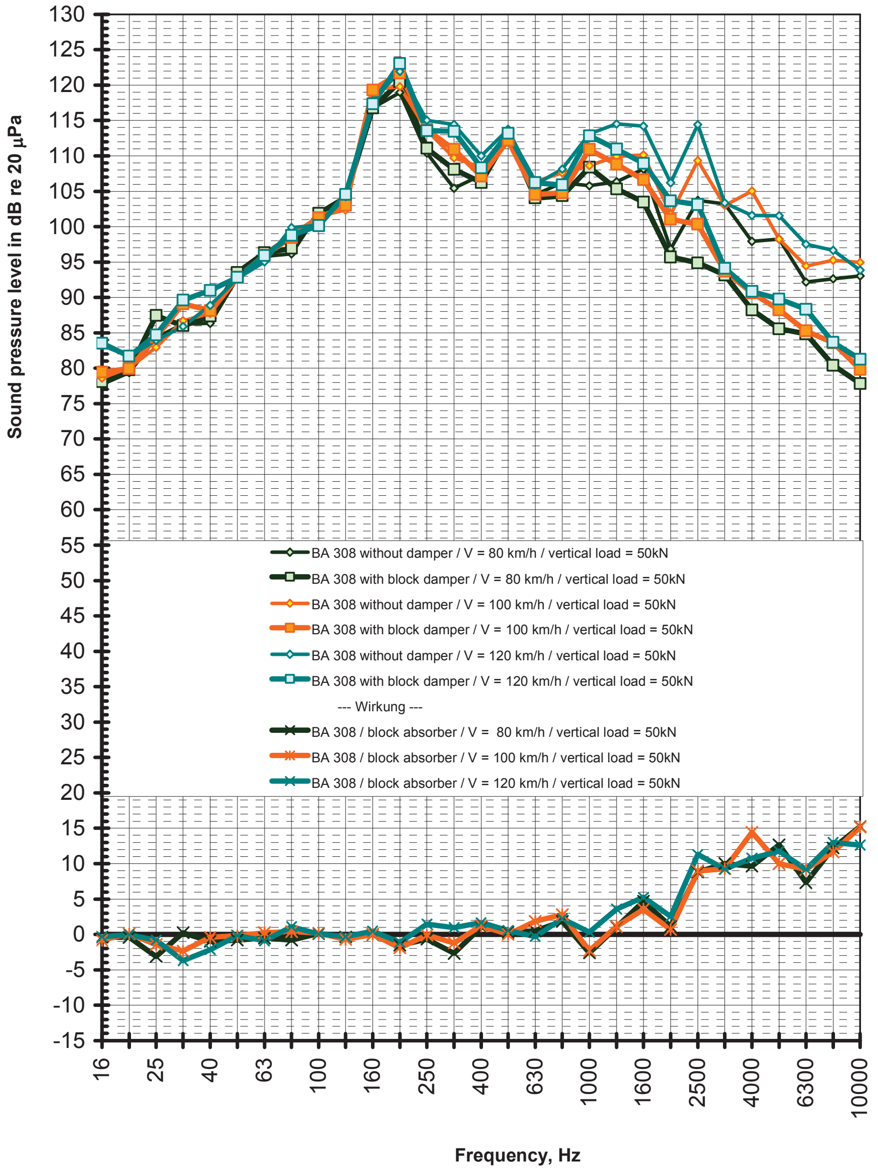


Fig. 57: *Measured sound pressure levels for the BA308 wheel with and without block damper. The curves in the upper half are absolute levels while the curves in the lower half show the differences. Vertical load = 50 kN, speed = 80 km/h, 100 km/h, 120 km/h.*

An example for the velocity scans is given in Fig. 59 for the bare LK900 wheel. The measurement started at t=0 with v=80 km/h. Then the velocity was continuously increased up to 120 km/h (t =40 s) and thereafter continuously reduced to 80 km/h. The results are shown in A-weighted equivalent sound pressure level LAeq (t).

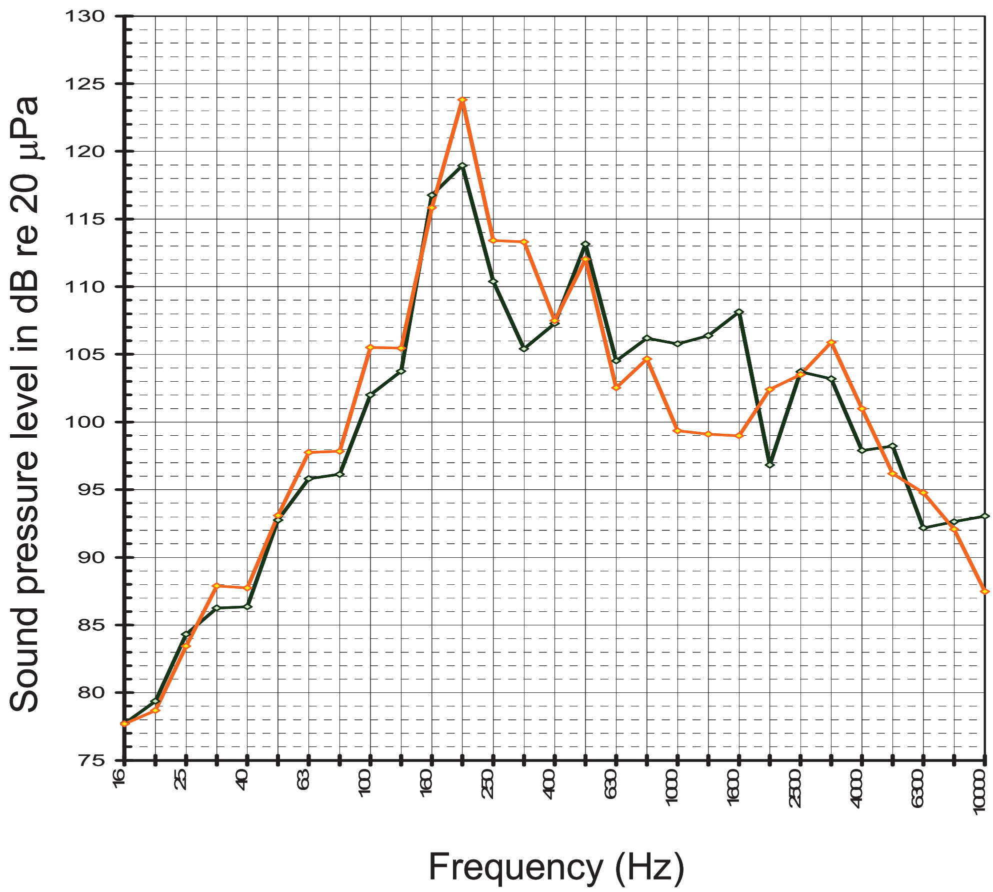


Fig. 58: *Direct comparison of LK900 (brown curve) and BA308 (green curve).*

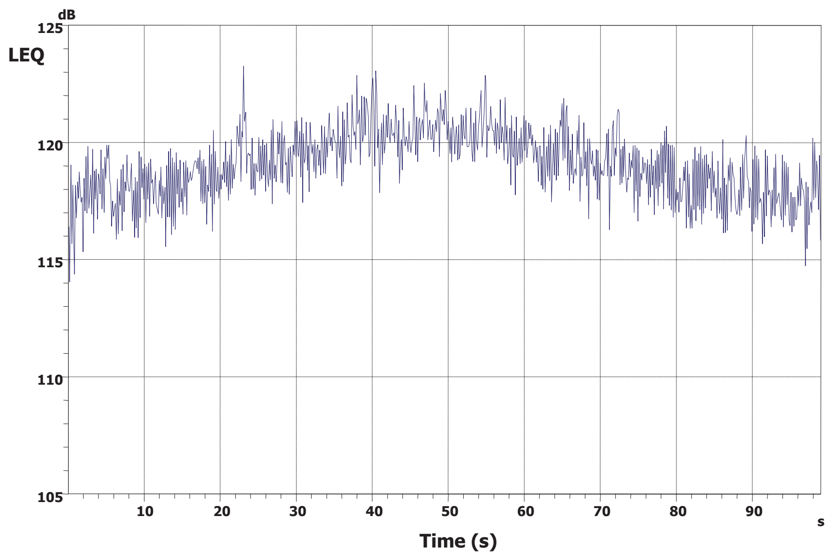


Fig. 59*: Dependence of the measured A-weighted equivalent sound pressure level LAeq (t) on velocity. v=80 km/at t=0, then continuous increase of velocity up to 120 km/h, which was eached at t=40s. Continuous decrease of velocity from v=120km/h to v=80km/h starting at t=55s.*

Summary: Single number representation of the measured noise reduction

The following tables give a summary of the measured noise reductions for all wheel/damper combinations and for all investigated parameter combinations. In order to provide a single number representation of the noise reduction effect of the dampers, overall levels LAeq have been calculated from the spectra. In order to avoid the levels being dominated by the low frequency background noise, the summation has been restricted to frequencies f > 800 Hz. However, as the dampers were not tuned to the wheels the obtained results do not allow a comparison of the different dampers, but results were only obtained to compare the roller rig measurements with the Stardamp method.

Table 12: A-weighted sum levels for frequencies $f \geq 800$ Hz and their difference for the BA308 wheel and block damper.

Measurement		Parameter		L_{Aeq} (no damper)	L_{Aeq} (with damper)	Difference ΔL
BA 308	block damper S & V	speed km/h	Load kN	dB(A)	dB(A)	dB
308-00-000	308-A1-000	80	50	114,6	112,4	2,3
308-00-010	308-A1-010	100	50	117,6	115,2	2,4
308-00-020	308-A1-020	120	50	121,5	117,3	4,3
308-00-100	308-A1-100	80	65	114,2	111,5	2,7
308-00-110	308-A1-110	100	65	117,7	114,7	3,0
308-00-121	308-A1-120	120	65	120,7	117,0	3,7
308-00-200	308-A1-200	80	80	113,0	110,1	3,0
308-00-210	308-A1-210	100	80	116,8	113,9	2,9
308-00-220	308-A1-220	120	80	120,3	116,6	3,7
308-00-300	308-A1-300	80	100	112,2	109,0	3,2
308-00-310	308-A1-310	100	100	116,5	113,4	3,2
308-00-320	308-A1-320	120	100	120,7	116,3	4,4

Table 13: A-weighted sum levels for frequencies $f \geq 800$ Hz and their difference for the LK900 wheel and ring damper.

Measurement		Parameter		L_{Aeq} (no damper)	L_{Aeq} (with damper)	Difference ΔL
LK900	ring damper	speed km/h	Load kN	Mp1 (wheel) dB(A)	dB(A)	dB
LK900-oA-000	LK900-R-000	80	50	112,6	108,3	4,3
LK900-oA-010	LK900-R-010	100	50	116,0	110,2	5,8
LK900-oA-020	LK900-R-020	120	50	119,6	112,0	7,7
LK900-oA-100	LK900-R-101	80	65	112,6	108,4	4,3
LK900-oA-110	LK900-R-110	100	65	115,9	110,4	5,4
LK900-oA-120	LK900-R-120	120	65	118,6	112,6	6,0
LK900-oA-200	LK900-R-200	80	80	111,1	106,8	4,3
LK900-oA-210	LK900-R-210	100	80	115,0	109,8	5,2
LK900-oA-220	LK900-R-220	120	80	117,2	111,7	5,6
LK900-oA-300	LK900-R-300	80	100	111,0	107,7	3,3
LK900-oA-310	LK900-R-310	100	100	114,0	110,3	3,7
LK900-oA-320	LK900-R-320	120	100	116,1	112,2	4,0

Table 14: A-weighted sum levels for frequencies $f \geq 800$ Hz and their difference for the LK900 wheel and plate damper.

Measurement		Parameter		L_{Aeq} (no damper)	L_{Aeq} (with damper)	Difference ΔL
		speed	Load	Mp1 (wheel)		
LK900	plate damper	km/h	kN	dB(A)	dB(A)	dB
LK900-oA-000	LK900-S-000	80	50	112,6	108,5	4,1
LK900-oA-010	LK900-S-010	100	50	116,0	110,6	5,4
LK900-oA-020	LK900-S-020	120	50	119,6	112,4	7,2
LK900-oA-100	LK900-S-100	80	65	112,6	109,8	2,9
LK900-oA-110	LK900-S-110	100	65	115,9	111,5	4,4
LK900-oA-120	LK900-S-120	120	65	118,6	113,6	5,0
LK900-oA-200	LK900-S-200	80	80	111,1	107,5	3,7
LK900-oA-210	LK900-S-210	100	80	115,0	109,8	5,2
LK900-oA-220	LK900-S-220	120	80	117,2	112,6	4,6
LK900-oA-300	LK900-S-300	80	100	111,0	108,9	2,1
LK900-oA-310	LK900-S-310	100	100	114,0	110,6	3,4
LK900-oA-320	LK900-S-320	120	100	116,1	112,9	3,2

Table 15: A-weighted sum levels for frequencies $f \geq 800$ Hz and their difference for the LK900 wheel and DAAVAC damper.

Measurement		Parameter		L_{Aeq} (no damper)	L_{Aeq} (with damper)	Difference ΔL
		speed	Load	Mp1 (wheel)		
LK900	DAAVAC damper	km/h	kN	dB(A)	dB(A)	dB
LK900-oA-000	LK900-D-000	80	50	112,6	110,1	2,5
LK900-oA-010	LK900-D-010	100	50	116,0	113,4	2,6
LK900-oA-020	LK900-D-020	120	50	119,6	115,1	4,5
LK900-oA-100	LK900-D-100	80	65	112,6	111,4	1,2
LK900-oA-110	LK900-D-110	100	65	115,9	114,2	1,6
LK900-oA-120	LK900-D-120	120	65	118,6	116,6	1,9
LK900-oA-200	LK900-D-200	80	80	111,1	110,5	0,6
LK900-oA-210	LK900-D-210	100	80	115,0	113,2	1,8
LK900-oA-220	LK900-D-220	120	80	117,2	115,4	1,8
LK900-oA-300	LK900-D-300	80	100	111,0	110,0	1,0
LK900-oA-310	LK900-D-310	100	100	114,0	112,7	1,3
LK900-oA-320	LK900-D-320	120	100	116,1	114,7	1,4

7.4.3 Prediction of field test results

The sound pressure level measured in a field test during pass-by of a test train is caused by the three noise sources (assuming that aggregate noise and aerodynamic noise can be neglected):

- wheel
- rail
- sleeper

The sound pressure levels caused by each of these three sources add up energetically at a receiver location giving the total sound pressure level (see Fig. 60). When using a single microphone it is impossible to separate the contributions of the different sources. This requires a numerical model as it is e.g. implemented in the TWINS software. When a damper is added to the wheel, only the source "wheel" is reduced. The sources "rail" and "sleeper" remain unaffected. The effect of a wheel damper is estimated from the difference of sound pressure levels measured with and without damper respectively.

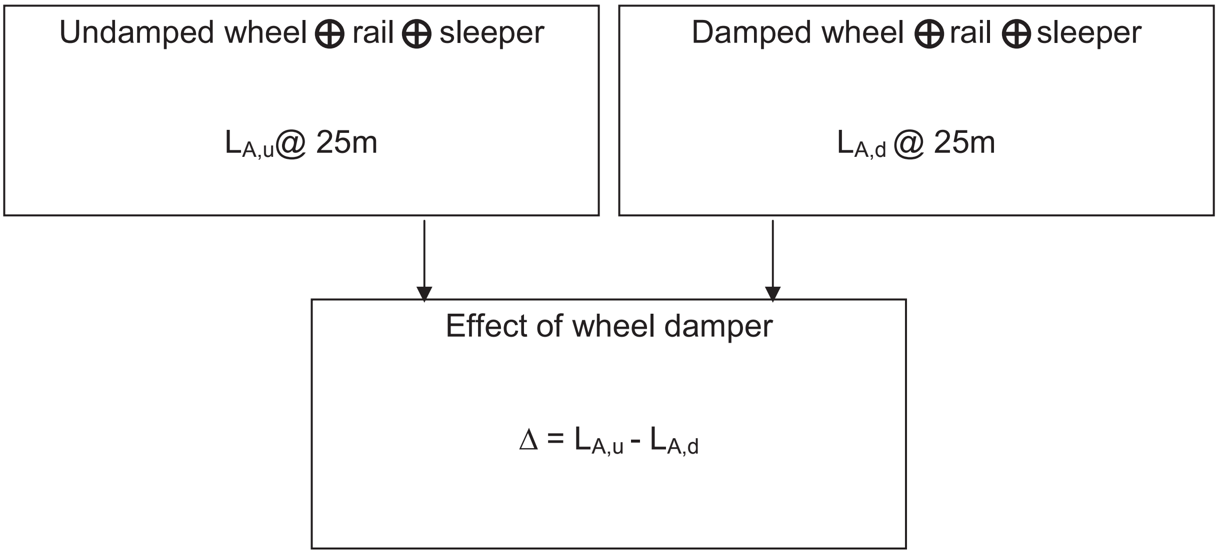


Fig. 60: *General scheme of calculating the effect of a wheel damper in a field test. (+ means energetic summation of the respective sound pressure levels). Only the source "wheel" is modified by adding a wheel damper.*

On the other hand, the relevant noise sources measured on the roller test rig are:

- wheel
- rail-wheel
- background noise

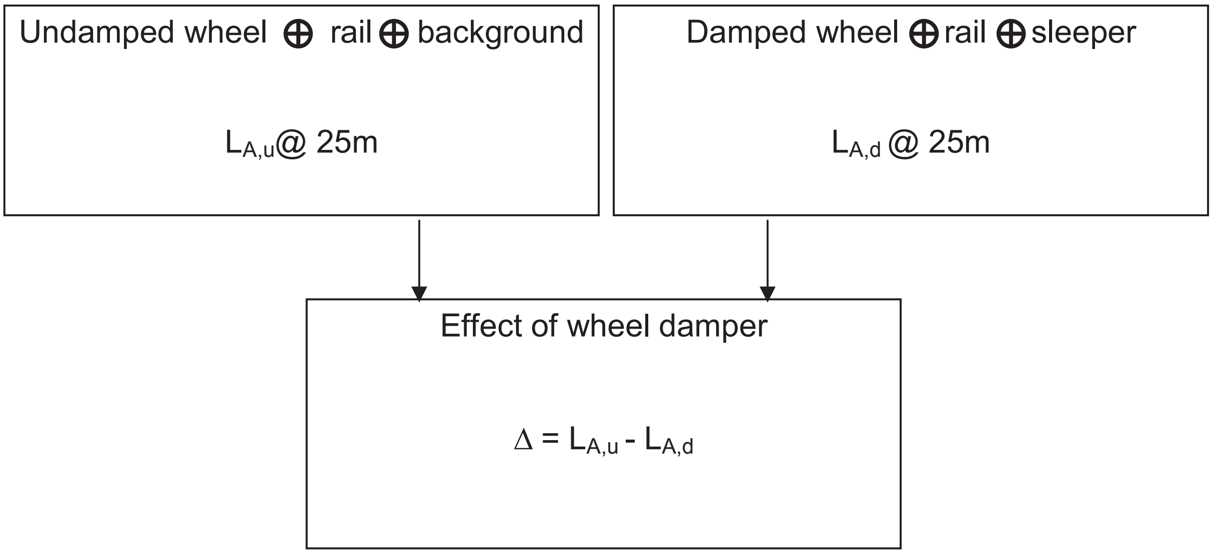


Fig. 61: *General scheme of calculating the effect of a wheel damper from the results of a rig test. Only the source "wheel" is modified by adding a wheel damper.*

It can be assumed that the total noise from the rail-wheel and the background noise is lower than the total of the noise from rail and sleeper in a field test. Therefore the procedure according to Fig. 61 is expected to overestimate the effect expected from a wheel damper in reality.

A simple procedure for predicting the noise reduction measured in a field test on basis of the noise reduction measured in a rig test would be the following:

1. Test rig measurement of the noise reduction obtained by adding a wheel damper (Δ_{rig}(f)).
2. TWINS calculation with the parameters of the test wheel and of the track foreseen for a field test.
3. Reducing the wheel contribution obtained in step 2. by Δ_{rig}(f).
4. Recalculation of the total of the contributions from rail/sleeper/wheel taking the sound pressure spectra for rail and sleeper from step 2. and the spectrum for the wheel from step 3.
5. Subtraction of the "total spectrum" obtained in step 4. from the "total spectrum" obtained in step 2.

7.4.4 Conclusion

Tests were performed with wheels of type BA308 and LK900 equipped with different types of wheel dampers. During test runs on the roller test rig RASP of DB Systemtechnik in Brandenburg/Kirchmöser the sound pressure has been measured and the effect of wheel dampers was estimated by subtracting sound spectra measured for damped wheels from the corresponding spectra of the un-damped wheel.

Measurements on roller test rigs can supplement the other test methods for wheel dampers devised within STARDAMP for the following reasons:

- Measurements can be performed on full wheelsets and the effect of wheel dampers determined from measurements of the sound emission from the wheels.
- The excitation of the wheels is caused by the combined roughness of rail wheel and test wheel leading to contact forces very similar to those acting in field tests.
- Parameters like speed, vertical forces, position of the rail/wheel contact, rolling angle etc. can be chosen within a wide range and parameter scans can be readily performed. For instance is it possible to continuously change the speed over a wide range up to 300 km/h and to continuously monitor the sound emission from the wheels. Such scans enable the detection of unwanted resonances of the wheel damper, which might remain undetected in field tests, which are usually limited to a few pass-by speeds of a test train.
- Rail and wheel roughness can easily be monitored. This is extremely costly in field tests and is therefore frequently omitted.
- Different wheel/damper combinations can be compared under realistic conditions. Tests should mainly focus on such relative measurements. The precise prediction of absolute sound pressure levels as they are to be expected in field tests is difficult and needs further investigation. A first estimation can be made on the basis of the procedure described above.

Particular attention should be paid to the roughness of rail and wheel. The test wheels should have typical roughness, i.e. it is recommended to use run-in wheels which have been used in regular trains. The roughness should at least exceed the roughness of the rail-wheel, thus making measurements less sensitive to changes in the rail roughness during the course of the measurements. Further acoustic optimization of the RASP by e.g. installing additional absorbing materials and by reducing background noise is recommended.

8. The STARDAMP Software

This section describes the STARDAMP software, which was developed in the project and defines its requirements. The software is divided into two parts:

- STARDAMP tool – used to make rolling noise predictions to assess wheel or rail dampers.
- Modal parameter file editor (mp-editor) – used to create input files for the STARDAMP tool for damped and undamped wheels from a finite element model of the wheel section (described in a separate user manual [SR 23] – see also Sec. 5.5).

The STARDAMP tool implements TWINS-like predictions of rolling noise in a user-friendly way. Compared with TWINS the number of options is limited in order to allow access by non-expert users.

The use of the software to evaluate rail dampers and wheel dampers is described in detail in [SR 22]. In addition the software can be used to assess the effect of a combination of wheel and rail dampers simultaneously. It can therefore be used in the following modes:

- baseline calculation (no dampers);
- rail dampers;
- wheel dampers;
- both wheel and rail dampers.

The software is written in Matlab and can be supplied as a compiled stand-alone application to run in Windows. It has a graphical user interface (GUI) which can be operated in English, French or German.

Each model is defined in terms of “traffic” and “track”. All the information related with the train such as wheel geometry, speed and axle load are contained in the traffic, while the physical and geometrical properties of the track are contained in the track model.

8.1 STARDAMP software for rail dampers

The flow chart in Fig. 62 summarises the process adopted for evaluating the effectiveness of a rail damper. A baseline version of the model (left branch) is compared to the same model but with different track characteristics that correspond to the damped track (right branch). The damper effect is considered by using measured Track Decay Rates values.

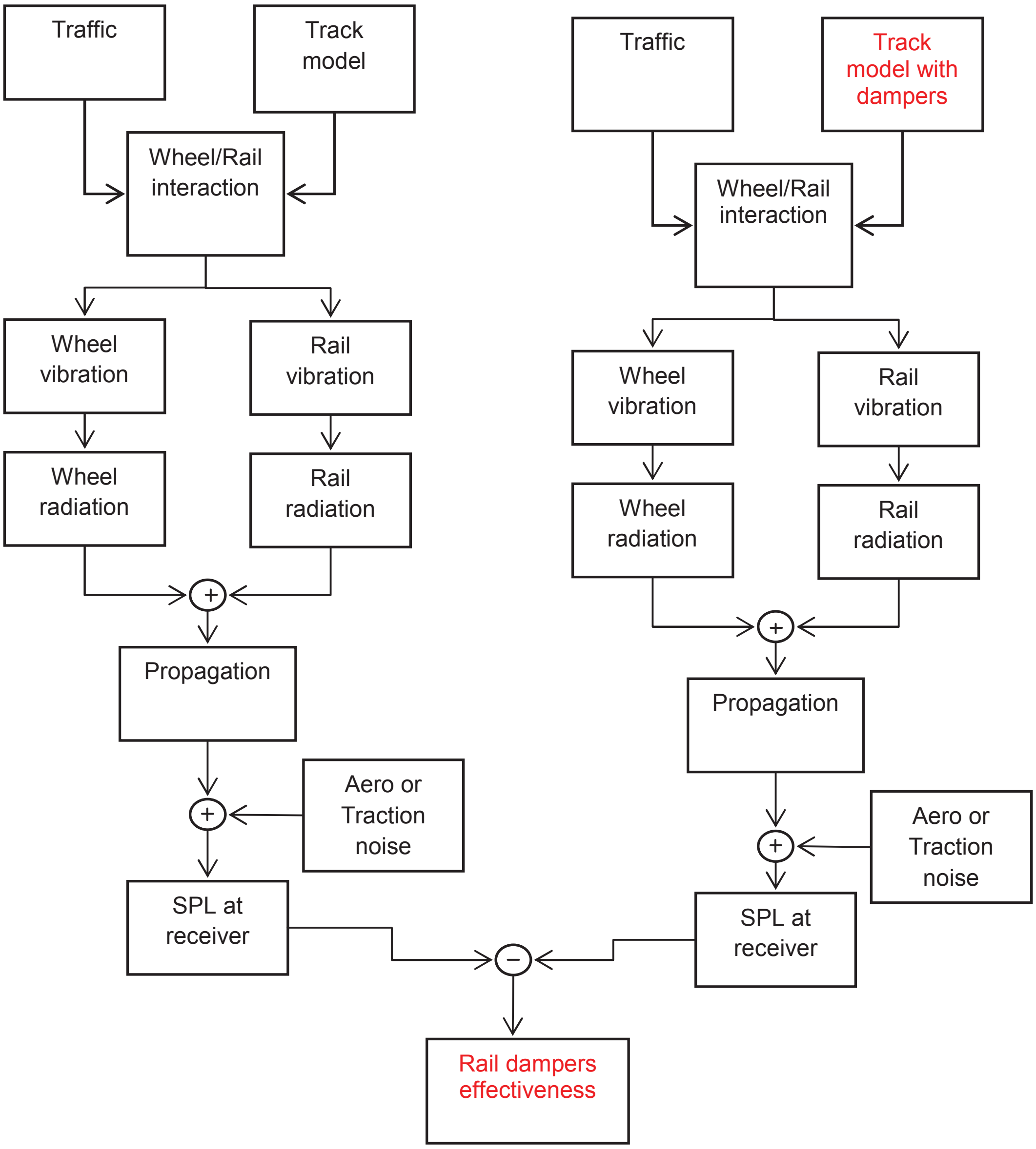


Fig. 62: *Flow chart of the process for rail dampers assessment.*

Fig. 63 shows the graphical interface of the tool as it appears when set in the mode for assessing rail dampers.

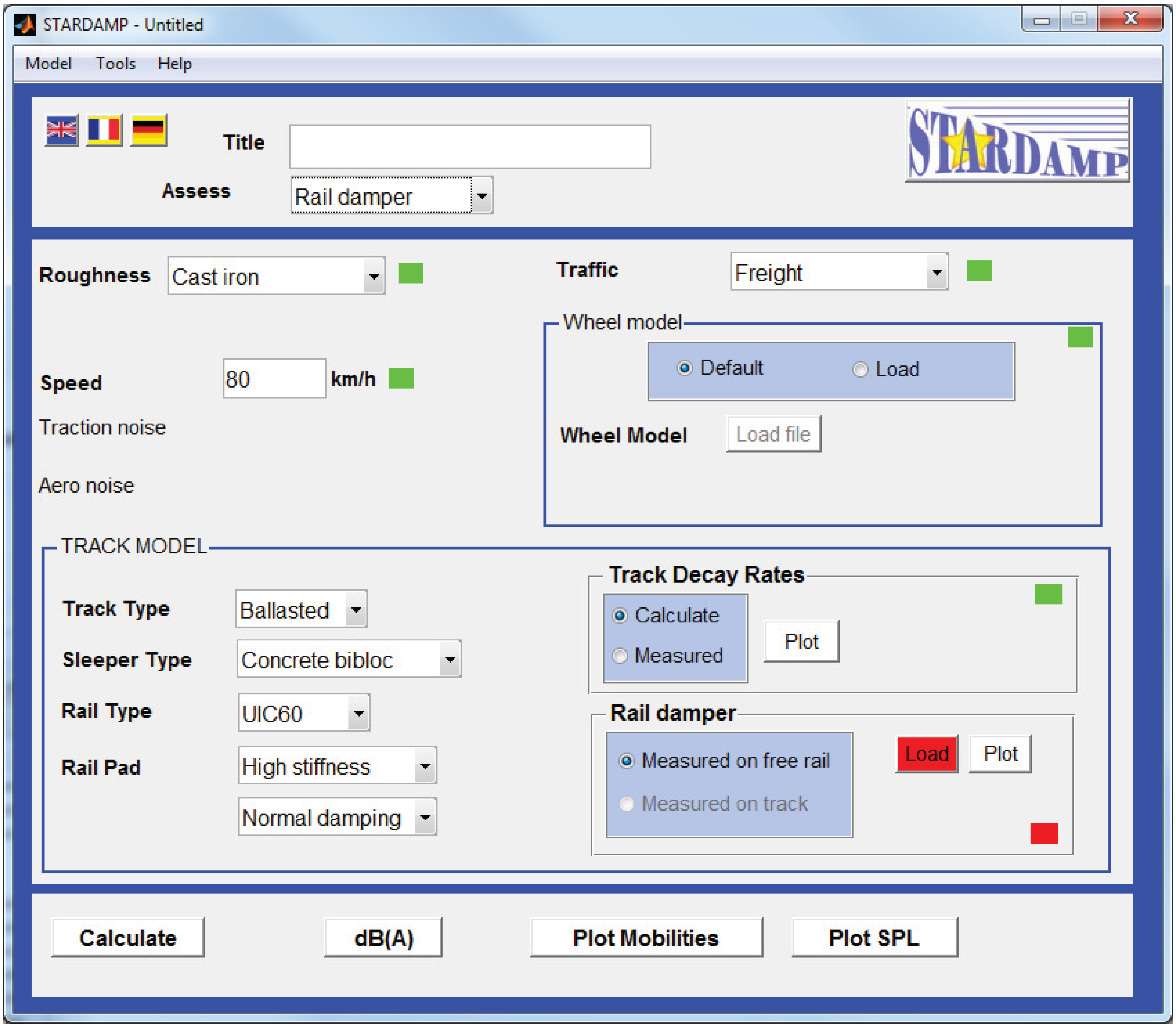


Fig. 63: *The software as it appears when set for rail damper assessment*

The software allows three default types of trains (denoted “traffic” in the tool interface) to be modelled, which are represented by means of the wheel modal parameters, vehicle length and the load per wheel. Optionally a user defined wheel modal parameter file can be loaded; it has to be of the type generated with the modal parameters editor.

Linked with each of the three default traffic types, several parameters are fixed. 9 summarises the default train speed values and the wheel types.

Table 16– Summary of traffic type

Parameters	High-speed	Regional	Freight
Default speed (km/h)	300	160	80
Wheel type	TGV	LK900	BA308

Concerning roughness three different spectra are offered as default and they can be used with any of the train models described in Table 16. These roughness levels are representative for wheel with disc brakes, cast iron brakes and K-block brakes. and stem from earlier measurement campaigns performed by SNCF. The wheel roughness level is added to the TSI reference roughness for the rail in order to achieve a combined roughness for the excitation. Alternatively a user defined roughness file can be uploaded into the software. This should correspond to the total roughness of wheel and rail and the contact filter must be excluded since it is already accounted for in the tool.

Two track types can be chosen: ballasted and slab track. Regarding the slab track option there are no sleepers and the rail pads connect the rail to a rigid ground. The ballasted track type, on the other hand, is representative for the ballast, the sleepers, the rail pads and the rail.

Sleeper type: when sleepers are included they can be modelled as concrete monobloc, concrete bibloc or wooden. Monobloc sleepers are considered like systems having their own modal behaviour, whereas bibloc sleepers are seen as two rigid masses during the calculation.

Four rail types are modelled in the software: UIC60, UIC54, S54 and S49.

Rail pad stiffnesses and loss factors can be chosen from a set of predefined values, otherwise they can be manually set according to specific needs.

Effect of rail dampers

The track decay rate TDR is the parameter commonly adopted to describe the exponential decay of the vibration amplitude from the excitation point (see Sec. 3.1).

In order to assess the effect of rail dampers installed on a track, the software is designed to make two rolling noise predictions. These are for the same train/track combination with and without the rail dampers. Therefore, before running the software with the rail damper option activated, it is necessary to evaluate the decay rates for the chosen rail damper. This can be done with damper measurements in the laboratory (as described in Sec. 5.3) or with measurements in the field.

The software allows three different possibilities for comparison between a base configuration (no dampers) and a rail damper configuration:

- with damper measurements in lab and calculated TDR for tracks;
- with damper measurements in lab and measured TDR for tracks;
- with damper measurements in field and measured TDR for tracks.

The fourth option (with damper measurements in field and calculated TDR for tracks) is not allowed since there is not enough confidence that the calculated TDR will be representative of the track where the dampers have been measured when considered without dampers. Therefore this option could give a misleading interpretation of the rail damper performance.

Input data and their format are described in detail in [SR 22].

Operation of the software

Once the parameters are set, as necessary, the button “Calculate” can be used to run the simulations. In order for the rail damper effect to be evaluated two runs are performed in the software: the first without considering any added dampers and then the second with the dampers as set by the user. Green flags (see Fig. 63 and Fig. 64) indicate that the input data are complete whereas red flags are used to indicate that more inputs are required. During the computations the values in the GUI cannot be edited.

In order to increase the robustness of the results, for each calculated case three contact points are considered (nominal and nominal ± 10 mm) and the resulting sound pressure level is the energy average of these three.

When the calculations are completed the user is asked to decide about the saving options, as shown Fig. 64. This should help to manage the files and to prevent later undesired loss of results. In order to save the file correctly a directory has to be set first. This will be the directory where the model is going to be saved. The user should then define the model name. By doing this another folder having the model name will be created in the folder selected before. Inside the model folder three files will be created. The first one contains the setting information of the model, which means the train speed, the wheel roughness etc. The other two files have either the input information for the numerical calculations and the outputs.

Finally if a model is run over an already saved one the user is asked whether to update the existing file or to save as a new one before being able to analyse the results.

Fig. 64: *Saving menu appearing straight after calculations. Note calculation options are greyed out.*

After a calculation, or if a file having results associated with is loaded, the three results buttons in the bottom part the interface can be used to show the results.

The button named dB(A) opens a new window which summarises the global sound pressure levels associated with the model. As the user will be interested in assessing the rail dampers, the second row of the table is the one to be observed (Fig. 65). It gives the overall level contributions of the track and for the wheel with and without dampers.

The buttons “Plot Mobilities” and “Plot SPL” return graphs similar to those shown in Fig. 66. The SPL graph contain two sets of curves: one for the undamped track and the other for the damped track.

Finally by resetting the “Calculate” (which has changed name to “Reset”) the GUI will be restored for evaluating new configurations.

As an option the SPL values can be visualised also in octave bands, rather than 1/3 octave, and at 25 m distance from the track, instead of 7.5 m. Moreover both A-wighted and non A-weighted modes are available for plotting the Sound Pressure Levels.

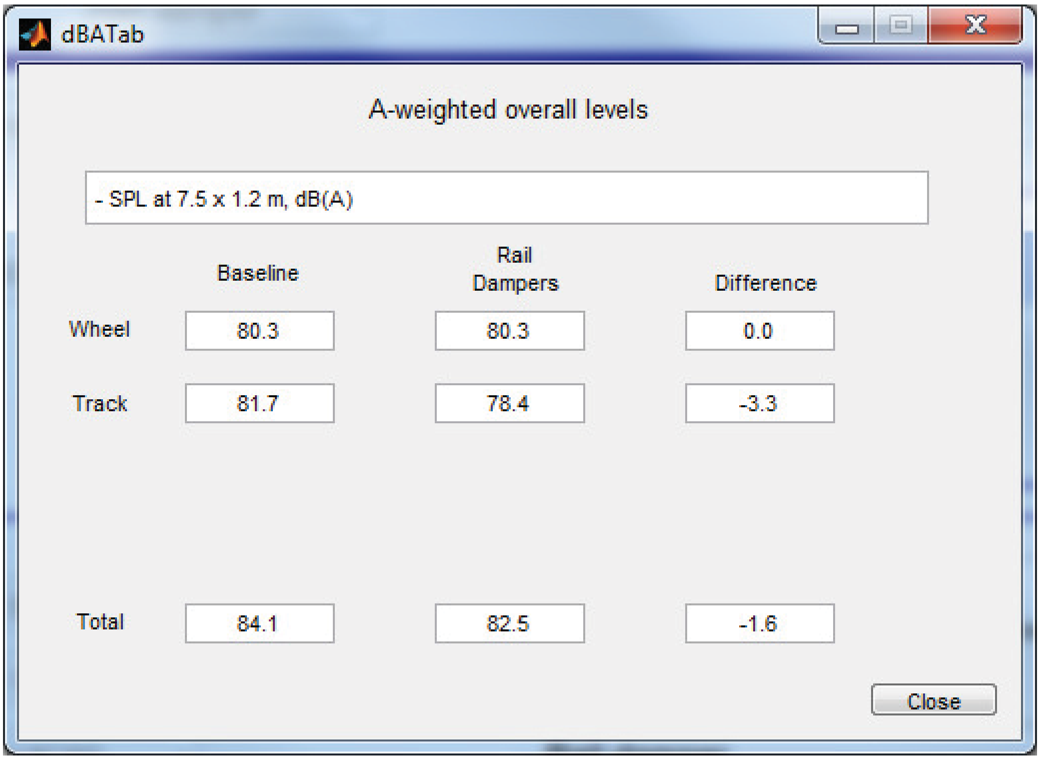


Fig. 65 : *Summary of overall sound pressure level.*

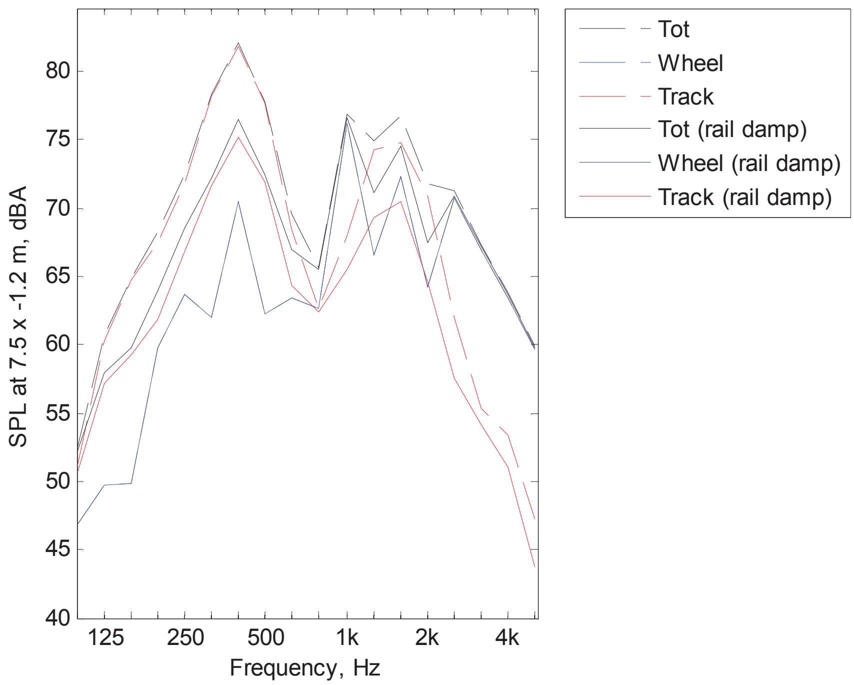


Fig. 66 : *Sound pressure levels in one-third octave bands. Dashed lines represent the model without dampers, continuous lines the model with dampers.*

8.2 *STARDAMP software for wheel dampers*

The same software can be used to assess wheel dampers in the same way as for rail dampers. The flow chart representing the operation of the tool for wheel dampers assessment is given in Fig. 67. The dampers are included in the model by adjusting the damping ratios and the natural frequencies in the wheel_modes file via the mp_editor.

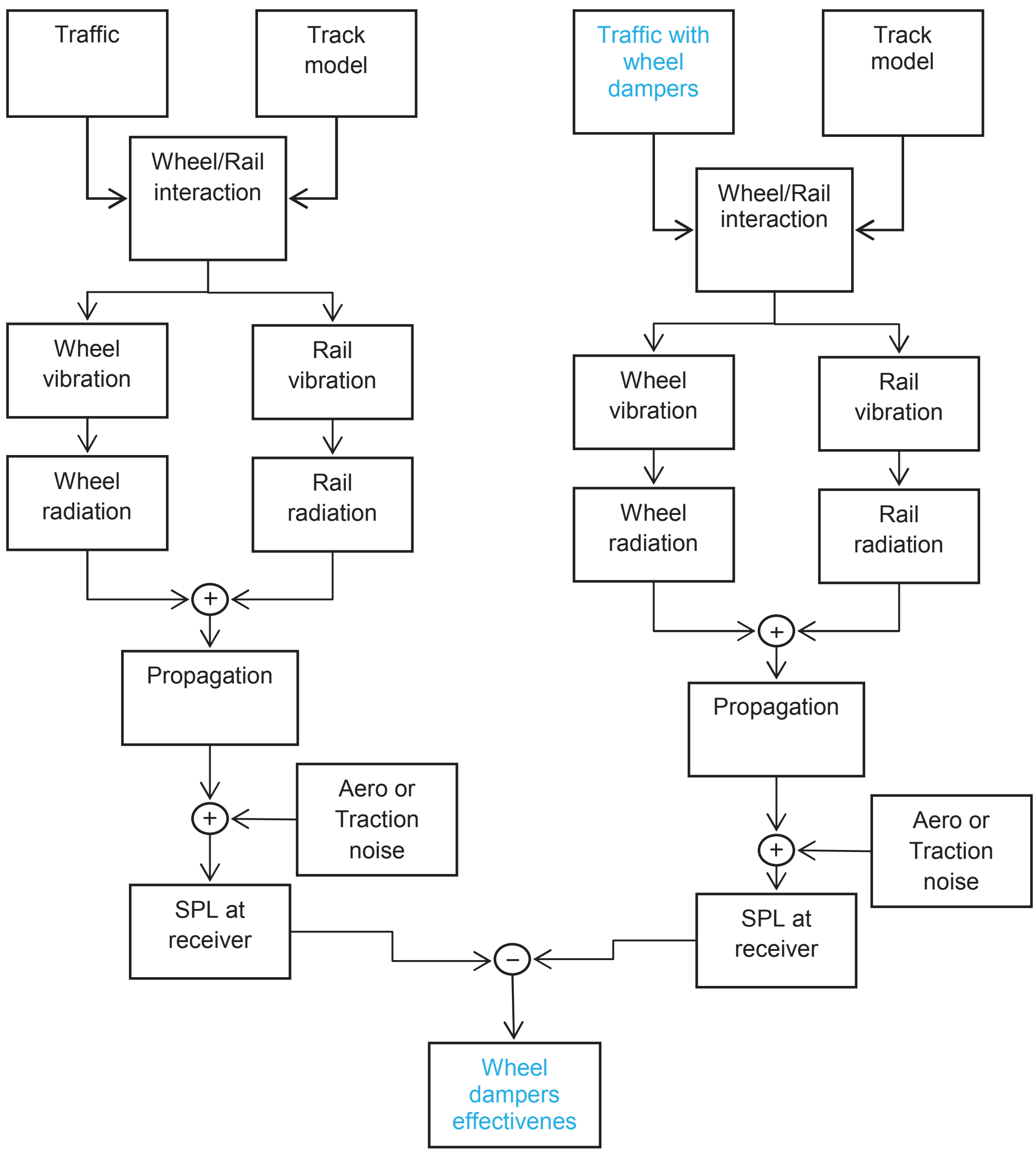


Fig. 67 : *Flow chart of the process for wheel dampers assessment.*

By choosing the “Wheel Damper” option in the “Assess” popup-menu the graphical interface will appear as in Fig. 68.

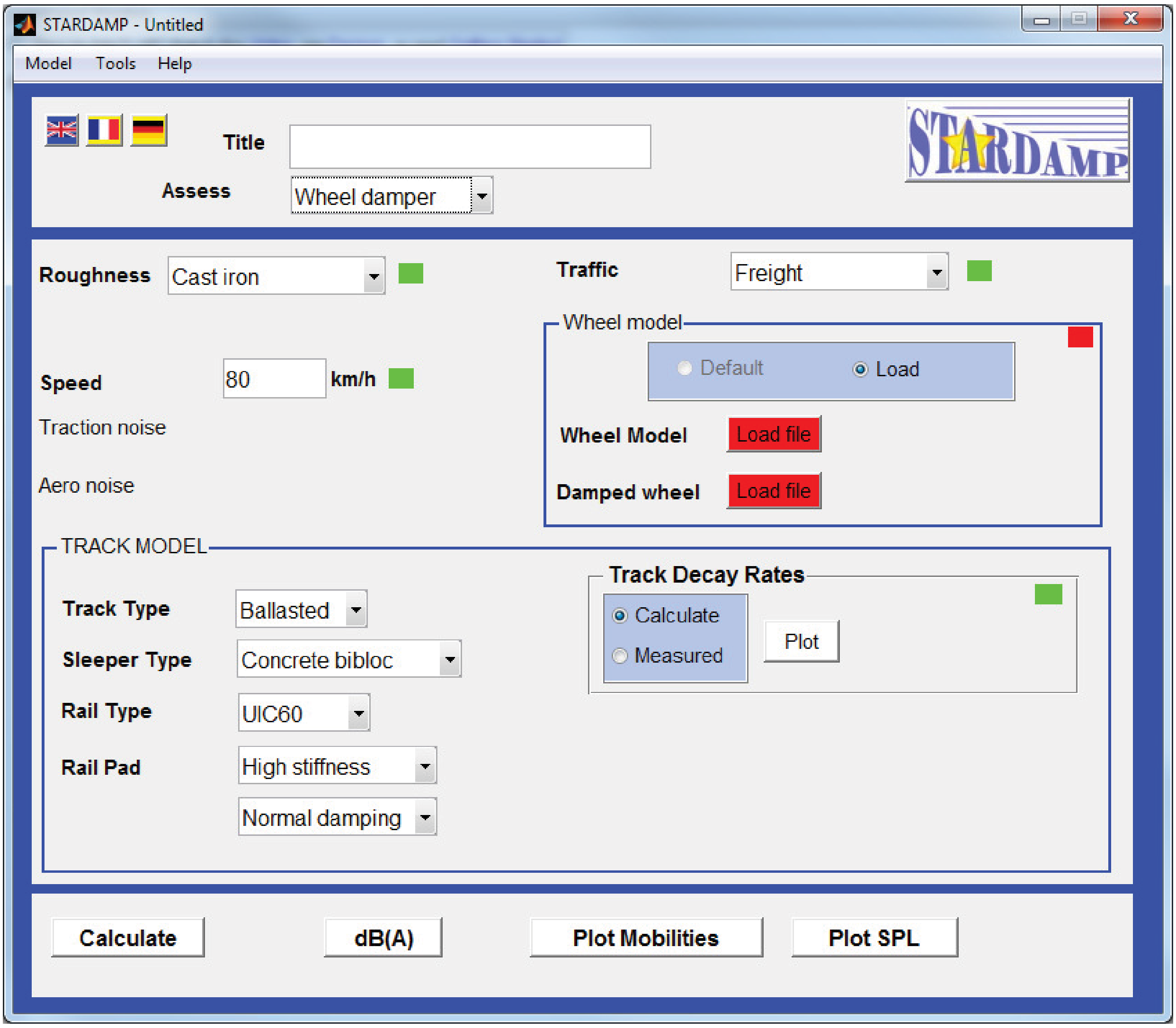


Fig. 68: *The software as appears when set for wheel dampers assessment.*

The main difference between this case and the rail damper one is that the user has to load two wheel modal parameters files, whereas for rail dampers these could be the predefined ones. These files should both be generated with the modal parameter editor as described in the mp-editor user manual [SR 23].

All the other functionalities described for the rail dampers in the previous section are available in the same way.

Effect of wheel dampers

Several techniques exist to reduce the wheel vibration levels with the purpose of lowering the wheel noise. It is known, however, that these measures mostly affect the wheel component of noise, so that the effect on the total noise has to be evaluated case by case.

The aim of this tool's functionality is that to observe the effect of modifying the wheel structure by comparing the noise levels calculated adopting two different wheel models. These two models should represent the same wheel with and without dampers.

The methodology implemented in the software does not account for any modification which occurs in the wheel mode shapes due to the presence of the dampers, nor shielding. The software performs a check on the wheel mode shapes. If the two wheel modes files loaded have different values for them the user is warned and asked whether to continue or reload the two files.

Fig. 69 shows as an example the sound pressure levels in one-third octave bands and Fig. 70 the table with overall levels obtained comparing the same wheel model with increased damping.

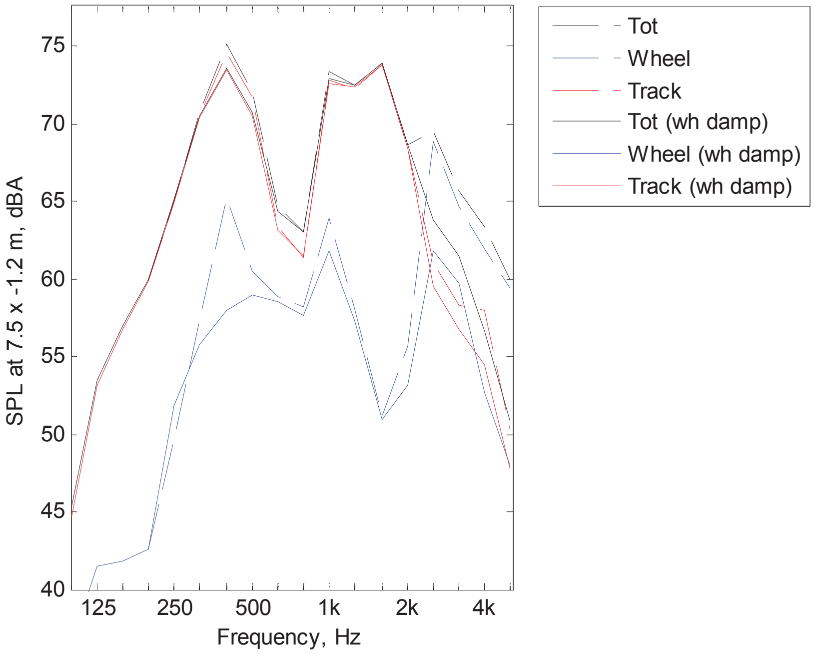


Fig. 69: *Example of the results for wheel damper assessment. Continuous lines represent the model without dampers, dashed lines the model with dampers.*

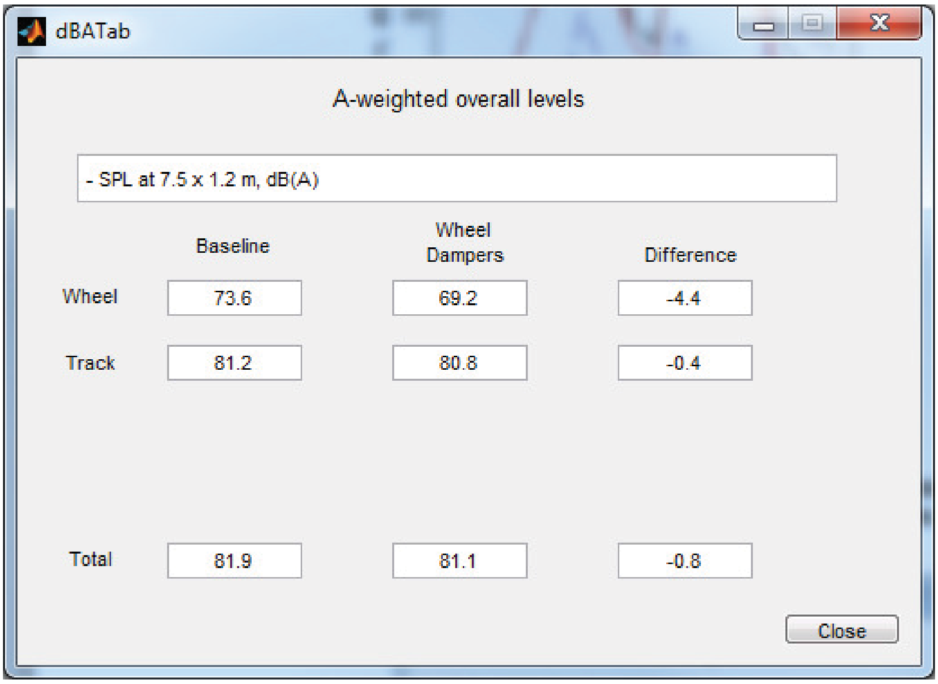


Fig. 70: *Overall values for a wheel damper assessment case.*

8.3 The mp-editor

In the TWINS model, the structural dynamics of the wheel is calculated from an FE model implemented in any finite element package software. In order to enter the model into the TWINS software a reduced set of data from the modal analysis of the wheel is written to a *modalparameters* file(*filename.mp*, 'an “M”, “P” file') (see Fig. 9 in Sec. 4). This contains the modal frequencies, modal masses (as defined by the FE package's eigensolution normalisation), modal damping and mode shapes. These should include all the modes that contribute significantly to the wheel vibration response in the range of frequency for rolling noise. Since the wheel modes, even with damping treatments, are relatively lightly damped, this 'modal basis' needs only contain modes to a frequency only a little higher than the frequency range of interest in order to avoid the modal truncation.

The mode shape response is recorded in the modal parameters file at only a small subset of the nodes of the original finite element model.

Very often, investigative studies with the rolling noise model require the details of the modal basis to be modified, for instance to introduce the effects of higher damping in particular modes, or to 'tune' the modal frequencies to those of specific measurements made on a particular wheel. The *mp_editor* utility therefore provides the ability to view the modes of the wheel in tabular form identified in terms of mode type and frequency in a form that can directly be edited by the user (see Fig. 71).

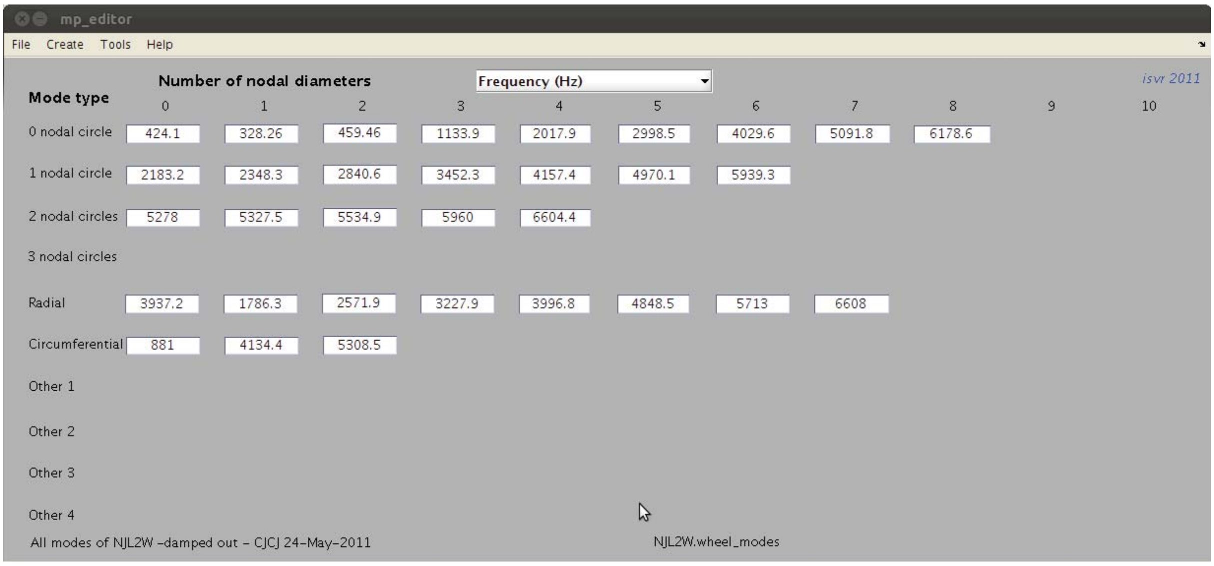


Fig. 71: *Main dialogue box of mp_editor*

The construction of a wheel model requires two main steps: An FE analysis and the importing of the subsequent analysis results into *mp_editor.* Installation and application of the *mp_editor* is explained in a comprehensive handbook [SR 23].

The mp_editor allows to test wheel models by calculating the radial and axial contact point receptances of the wheel. This is useful for comparing with measured receptances when tuning a wheel model and it can help identify if an error has been made in building wheel_modes file or in editing the data. An example graph is shown in Fig. 72.

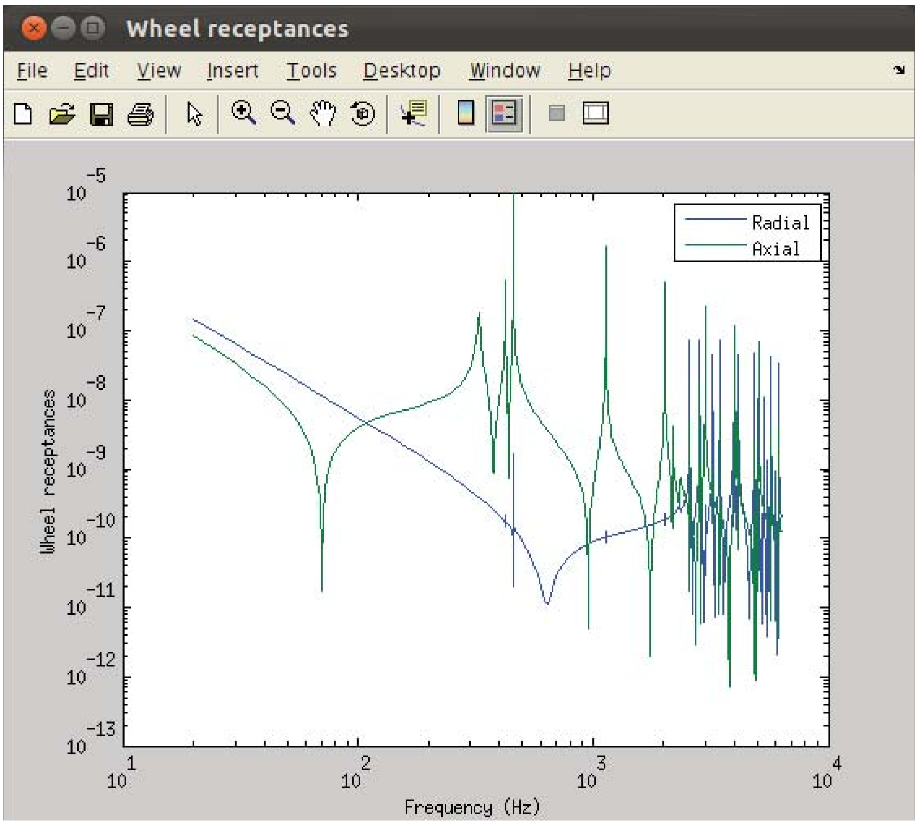


Fig. 72: *Example plot of radial and axial point receptances.*

9. Durability and mechanical integrity

Dampers in service are subjected to a number of environmental conditions that cause ageing of their components and might thus decrease the effectiveness of the noise reduction. Therefore it is important to assess their behaviour in the long term.

Besides the sustainability of the damping effect, the mechanical integrity should be also assessed to ensure that the damper itself and particularly their fastening systems guarantee a proper installation and functionality.

There are currently no standard procedures developed specifically for the assessment of rail and wheel dampers with regard to these criteria. However, there are European standards that describe methods for testing the effect of the environmental conditions on damping materials that could be adapted for dampers in regular application.

It was an aim in STARDAMP to define test procedures for rail and wheel dampers to simulate the long term impact of environmental conditions (ageing) and of varying load cycles caused by the passing of trains (mechanical integrity). Use of existing standards should be made as far as possible with suggestions for adapting established procedures for the testing of other systems to suit rail and wheel dampers in service.

End-users of dampers shall be provided with information about possible tests to estimate the acoustic performance and mechanical integrity of dampers in the long term.

Because of the different designs and materials of each damper it is not possible to develop a general procedure that can be applied to all products. This is especially true for the wheel dampers, which are usually specifically developed for certain wheel designs and therefore vary much more than the designs of rail dampers. Since it is always the combination of wheel and damper that has to be assessed, each design might require modifying the test and assessment procedures.

It was the task of the TU Berlin to perform tests of the long-term acoustic properties of rail- and wheel dampers to show the applicability of the proposed methods and to define general recommendations. The test results are documented in

[SR 5[SR 5] for rail dampers and in [SR 15] for wheel dampers. A separate handbook gives general recommendations for tests related to the long-term behaviour of dampers [SR 2].

9.1 Heat ageing of wheel dampers

Wheel dampers are often made of elastomeric materials combined with metallic parts. The properties of the elastomers are highly dependent on environmental conditions such as temperature and also vary over time due to the ageing of the material. Environmental conditions that apply to Central Europe are summarized in Table 17.

Ageing is a key factor in the use of wheel dampers because it is necessary to assure that despite the degradation of the material the dampers can achieve a certain noise reduction over their whole life cycle.

Therefore it is important to test the effect of this ageing, for which there are two possibilities:

1. Direct periodical measurement on a real test wheel equipped with dampers. The main advantage of this method is that the test is carried out under real environmental conditions. However, its obvious disadvantages are the long test period and high costs.
2. In order to obtain results in a reasonable time, dampers can be subjected to accelerated ageing processes with their performance being measured before and after the ageing.

Table 17: Operating conditions in Central Europe

	Minimum	Maximum
Air temperature (°C)	-25	45
Relative humidity (%)	35	100
Sunlight exposure (h/day)	0	12
Train speeds (km/h)		250 (passengers) 120 (freight)

The standard ISO 188 [5] describes a general procedure for the accelerated ageing of vulcanised rubbers, which are often used in wheel dampers. This method is based on the heat ageing of samples in an air oven. According to ISO 188, the required temperature and duration of the test must be chosen according to the standard ISO 23529 [6]. This temperature must be higher than the maximum temperature of the considered operating conditions but also respect the limit given by the manufacturer for the correct performance of the dampers in order to avoid permanent damage.

The maximum operating temperature according to table 15 is 45°C. As the ageing temperature must be higher than this value, it was chosen to be 60°C for the tests within STARDAMP. The duration of the ageing was also chosen in agreement with ISO 23529 and was set to 168 hours.

The dampers were then exposed to the chosen temperature during the chosen period in an air oven according to the ISO 188 standard (method A) [5]. In order to guarantee a regular heating, the minimum distance of the test samples to the walls of the oven must be at least 5cm and there must exist a clearance of at least 1cm between samples. A sketch of the heat ageing in the air oven is shown in Fig. 73. After the ageing, the dampers were cooled down to room temperature.

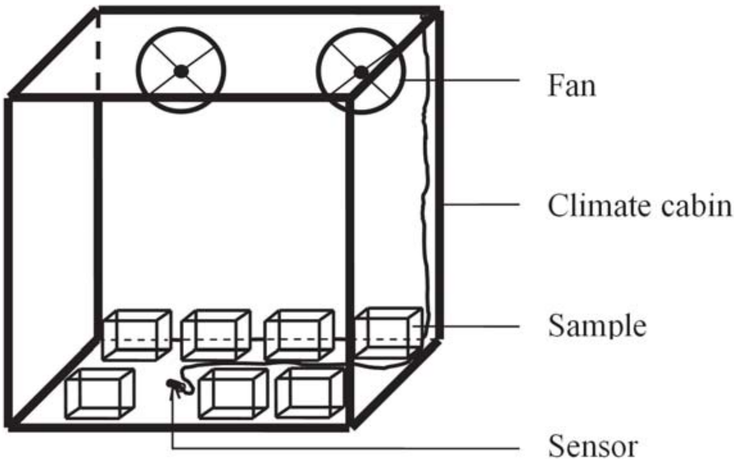


Fig. 73: *Sketch of heat ageing in air oven*

Before and after ageing the modal damping is measured. This will document any change in the damping ratios which might have ocured during heat treatment. Results of the aging tests for the combinations BA308 wheel/SuV damper and LK900 wheel/plate damper are summarized in Table 18 and Table 19 as examples.

Table 18: Results of the ageing tests for the BA308 wheel and SuV damper

wheeltyp:	BA308	BA308
typ of damper:	SuV	SuV (aged)
room temperature:	21°C	20,4
wheel temperature:	19,8°C	22,2

before aging					**after aging**				
0ln Mode					0ln Mode				
mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	339	2	0,005	0,26%	0	327	6	0,018	0,89%
	508					508			
1	73	-	-	-	1	73	-	-	-
	153	-	-	-		153	-	-	-
	292	-	-	-		292	-	-	-
	584	-	-	-		584	-	-	-
2	399	-	-	-	2	399	-	-	-
3	941	4	0,004	0,19%	3	903	4	0,005	0,24%
4	1864	40	0,022	1,08%	4	1864	-	-	-
5	2762	106	0,038	1,91%	5	2713	125	0,046	2,31%
6	3717	14	0,004	0,19%	6	3720	109	0,029	1,47%
7	4745	97	0,020	1,02%	7	4719	52	0,011	0,55%
8	5852	14	0,002	0,12%	8	5886	47	0,008	0,40%
1ln Mode					1ln Mode				
mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	997	-	-	-	0	997	-	-	-
	1515	9	0,006	0,28%		1464	15	0,010	0,52%
1	1111	-			1				
	1303	5	0,004	0,18%		1287	21	0,016	0,81%
	-	-							
	1713	13	0,007	0,36%		1713	dB n.a	-	-
	2026	16	0,008	0,40%		2026	dB n.a	-	-
2	2302	16	0,007	0,35%	2	2258	59	0,026	1,30%
3	3172	25	0,008	0,39%	3	3201	28	0,009	0,44%
4	4556	85	0,019	0,94%	4	4556	dB n.a	-	-
5	5479	48	0,009	0,44%	5	5479	dB n.a	-	-
6					6				
7					7				
8					8				
Rn Mode					Rn Mode				
mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	2193	10	0,004	0,22%	0	2095	dB n.a	-	-
1					1	2969	-	-	-
	3288	86	0,026	1,30%		3110	62	0,020	1,00%
						3596	-	-	-
						4282	-	-	-
2	1184	15	0,013	0,64%	2	1174	13	0,011	0,53%
3	1604	13	0,008	0,39%	3	1566	39	0,025	1,23%
4	2538	68	0,027	1,33%	4	2441	32	0,013	0,66%
5	3288	85	0,026	1,29%	5	3288	dB n.a	-	-
6	4440	28	0,006	0,32%	6	4464	27	0,006	0,30%
7	5131	23	0,004	0,22%	7	5148	36	0,007	0,35%
8					8				
2Ln Mode					2Ln Mode				
mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	-	-	-	-	0				
1	2541	67	0,026	1,32%	1	2443	25	0,010	0,51%
	2868	69	0,024	1,20%		2765	50	0,018	0,90%
	3029	3dB n.a	-	-		3029	dB n.a	-	-
2	3456	64	0,019	0,93%	2	3455	81	0,023	1,17%
3	3717	13	0,003	0,17%	3	3750	49	0,013	0,65%
4					4				
5					5				
6					6				
7					7				
8					8				

Table 19: Results of the ageing tests for the LK 900 wheel with plate damper

	before aging	after aging
wheeltyp:	LK900	LK900
typ of damper:	GHH plate damper	GHH plate damper
room temperature [°C]:	22	23,6
wheel temperature [°C]:	20,6	24,1

0Ln modes (DASA)

mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	281	3 dB n.a	-	-	0	281	3 dB n.a	-	-
	383	-	-	-		383	-	-	-
1	81	3 dB n.a	-	-	1	81	3 dB n.a	-	-
	158	3 dB n.a	-	-		158	3 dB n.a	-	-
	-	-	-	-		-	-	-	-
	493	5	0,009	0,47%		491	14	0,028	1,39%
	-	-	-	-		-	-	-	-
	-	-	-	-		-	-	-	-
2	334	3	0,008	0,42%	2	334	2	0,007	0,36%
3	920	4	0,004	0,20%	3	920	4	0,004	0,19%
4	1685	13	0,008	0,38%	4	1685	11	0,006	0,31%
5	2547	12	0,005	0,24%	5	2551	12	0,005	0,24%
6	3457	16	0,005	0,23%	6	3462	10	0,003	0,15%
7	4401	10	0,002	0,11%	7	4396	7	0,002	0,08%
8	5341	5	0,001	0,05%	8	5342	6	0,001	0,06%

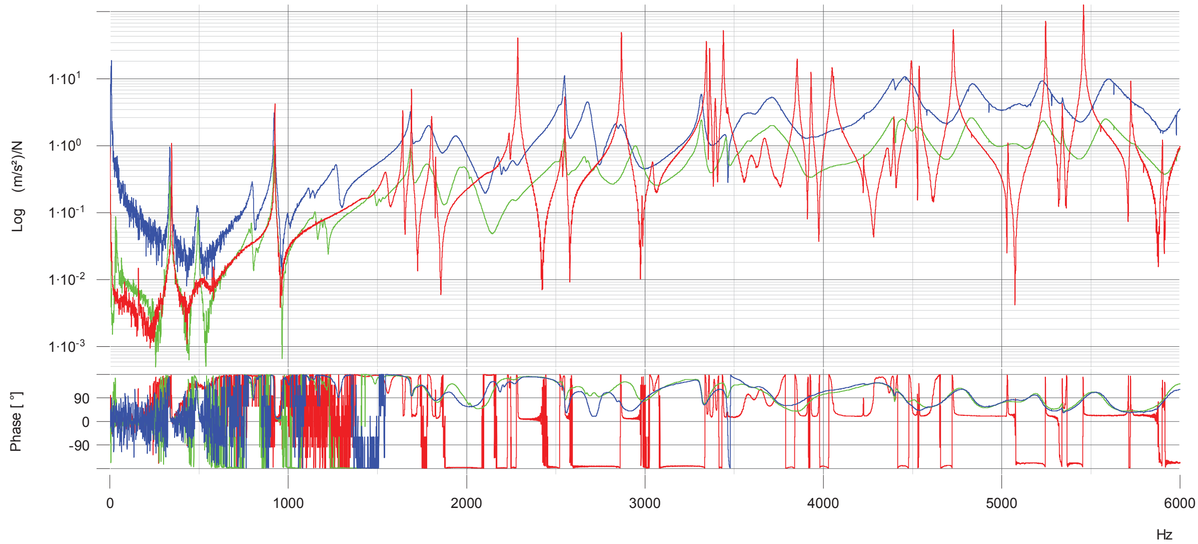
1Ln modes (DASA)

mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	-	-	-	-	0	-	-	-	-
	-	-	-	-		-	-	-	-
	1640	3 dB n.a	-	-		1640	3 dB n.a	-	-
1	1475	53	0,036	1,79%	1	1475	3 dB n.a	-	-
	1776	71	0,040	1,99%		1786	63	0,036	1,78%
	2256	3 dB n.a	-	-		2258	3 dB n.a	-	-
2	1803	3 dB n.a	-	-	2	1803	3 dB n.a	-	-
3	2545	14	0,005	0,27%	3	2545	9	0,004	0,18%
4	2983	3 dB n.a	-	-	4	2983	3 dB n.a	-	-
5	3344	3 dB n.a	-	-	5	3344	3 dB n.a	-	-
6	3854	3 dB n.a	-	-	6	3854	3 dB n.a	-	-
7	5061	3 dB n.a	-	-	7	5061	3 dB n.a	-	-
8	6091	76	0,012	0,62%	8	6109	90	0,015	0,74%

Rn modes (DASA)

mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}	mode	Eigenfrequency TU B	Δf_{-3dB} = f2-f1	damping coefficient	c / c_{crit}
	f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]		f_{res} [Hz]	Δf_{-3dB}	η	η /2 [%]
0	3394	-	-	-	0	3394	-	-	-
1	-	-	-	-	1	-	-	-	-
2	2285	-	-	-	2	2285	-	-	-
3	2868	-	-	-	3	2868	-	-	-
4	3447	-	-	-	4	3447	-	-	-
5	4049	3 dB n.a	-	-	5	4049	3 dB n.a	-	-
6	4830	87	0,018	0,91%	6	4837	87	0,018	0,90%
7	5585	103	0,018	0,92%	7	5601	106	0,019	0,95%
8					8				

Measured wheel accelerations for both radial and vertical direction before and after heat treatment are displayed in Fig. 74 and in Fig. 75. For estimating the change in performance of the damper, the SEMA method with subsequent calculation with the STARDAMP software can be applied. The full documentation of the tests is contained in [SR 15].



- Constant offset between measurements before and after ageing could be due to an error in transducer sensitivity

Fig. 74: *Radial accelerations on the single wheel LK 900, bare and with the GHH plate damper (aged and unaged) green: before heat treatment, blue: after heat treatment, red: bare wheel without damper*

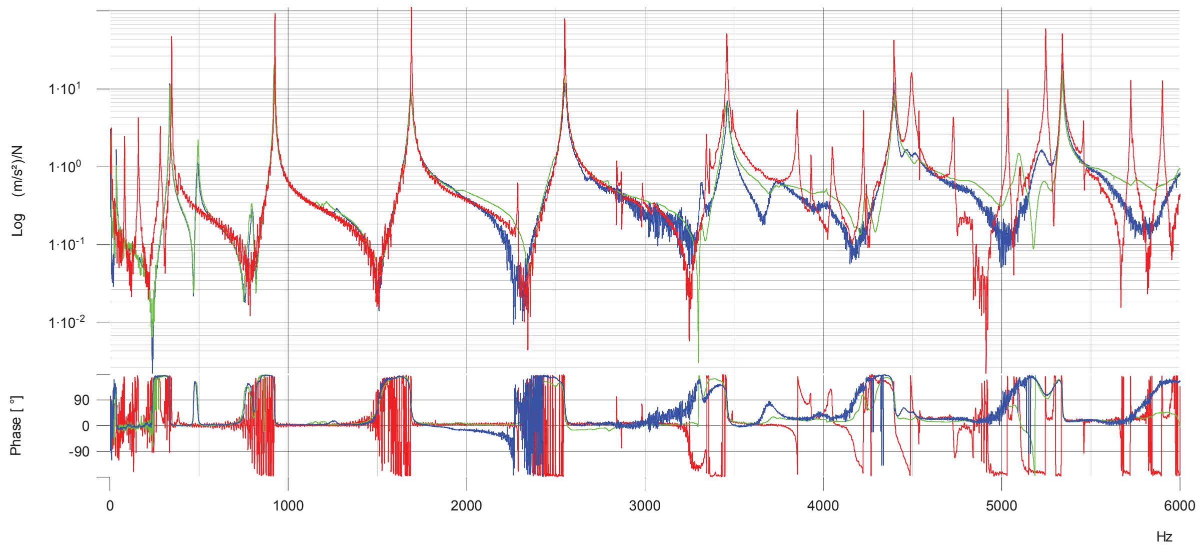


Fig. 75: *Axial accelerations on the single wheel LK 900, bare and with the GHH plate damper (aged and unaged) green: before heat treatment, blue: after heat treatment, red: bare wheel without damper*

9.2 Heat ageing of rail dampers

Rail dampers were subjected to heat treatment by TU Berlin similarly to the wheel dampers as described above. A temperature of 55°C and an ageing time of 168 hours were chosen.

Decay rates were measured before and after the heat treatment according to the ‘6m rail procedure’ (see Sec. 5.3). As an example the results obtained for the Tata Steel damper are displayed in Fig. 76. There are only small changes in DR, which fall within the typical error bars of the measurement. The full documentation of all aging tests with rail dampers is contained in [SR 5].

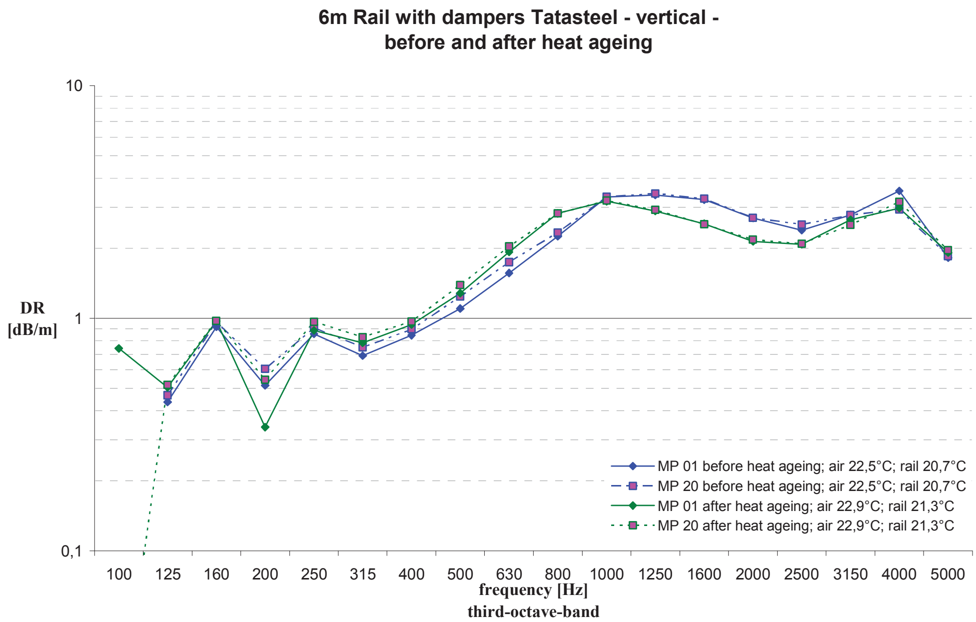


Fig. 76: *Measured decay rates for the Tata Steel damper in vertical direction before and after heat ageing*

9.3 Handbook

In addition to the tests described in the previous two sections, which were dedicated to testing the dampers provided by the STARDAMP project partners, a handbook [SR 2] has been composed with the aim to introduce test procedures for rail and wheel dampers to simulate the long term impact of environmental conditions (ageing) and of varying load cycles caused by the passing of trains (mechanical integrity).

The handbook makes use of existing standards as far as possible and suggests ways of adapting established procedures for the testing of other systems to suit rail and wheel dampers in service.

Future end-users of dampers shall be provided with information about the tests to estimate the acoustic performance and mechanical integrity of dampers in the long term.

Because of the different designs and materials of each damper it is not possible to develop a general procedure that can be applied to all products. This is especially true for the wheel dampers, which are usually specifically developed for certain wheel designs and therefore vary much more than the designs of rail dampers. Since it is always the combination of wheel and damper that has to be assessed, each design might require modifying the test and assessment procedures. Based on analysing the operating conditions, the handbook provides guidelines for appropriate testing.

During operation, the dampers are exposed to a number of loads and environmental conditions that can alter their acoustic performance. These conditions are of very different nature: mechanical loads, vibration, thermal loads, degradation due to various physical phenomena, etc. Since each case is different, it is not possible to make generalizations, neither to describe all the particular cases. For each case it must be analysed which are the possible hazards that might influence the acoustic performance of the dampers and how relevant they are. The variables that determine the operating conditions can be classified in two main groups that correspond to the two groups of assessments considered. A first group is formed by the environmental conditions, whose influence is assessed through tests of acoustic performance before and after ageing processes. Environmental conditions to be taken into account can be sorted in two categories. First of all those which have to be taken into account under all conditions and the second group consists of those which are suitable for a specific environment.

Environmental conditions, which are mandatory to be considered, are temperature, sunlight exposure and ozone exposure. Environmental impacts, which should be considered under specific environmental conditions are corrosive atmosphere and contact with fluids.

The second group of parameters includes those factors related to the railway operation that have an influence on the long term behaviour of the dampers. The operation conditions affect both the mechanical integrity and the acoustical performance:

- Axle load
- Speed
- Thermal loads induced by braking
- Vibration and shocks occurring in typical operating conditions
- Type of traffic (freight, passenger etc.)
- Track parameter (e.g. curve radii, slopes or cants can influence the distribution of forces in the track, which is relevant for the mechanical integrity tests)

The handbook [SR 2] describes procedures and defines guidelines for the following tests:

- Ageing tests:
    - Heat ageing
    - Resistance to low temperatures
- Stability against fluids
- Corrosion test in artificial atmosphere (Salt spray test)
- Stability against UV radiation
- Stability against ozone

The handbook is also included in the guidelines [SR26].

10. Field tests

10.1 Field test protocol

A central target of the STARDAMP project was to develop a methodology, which allows assessing the performance of dampers for rail and wheel without the necessity to perform field tests with real trains on real tracks. Nevertheless in special situations there will remain a certain demand for field testing e.g. for demonstration and homologation purposes. In order to perform these tests in the most efficient way it is necessary to define a measurement protocol, which allows the execution of field tests in a way to ensure reliable results, which are not restricted to the special situation of the test. Such a measurement protocol has been developed in STARDAMP. The field test performed by SNCF (see Sec. 10.2) was carried out according to this protocol and can be regarded as a first test for its applicability.

The following two subsections give an overview of the test protocol (separately for the cases of rail damper and wheel damper). For the full documentation of the measurement protocol, see [SR 21] and guidelines [SR26].

10.1.1 Field test protocol for rail dampers

The test protocol for rail dampers must correspond to the two following objectives (or scenarios):

Objective 1: Assessment of the noise reduction due to rail dampers on a specific site (e.g. hot spot)

Objective 2: Reproducible assessment of rail dampers in comparison with other noise mitigation products.

There are two different basic approaches for determining the difference between noise levels emitted with and without the damping devices:

Method A) *Comparison of pass-by noise before and after the installation of the dampers*

Two separate measurement series of pass-by noise levels of commercial trains are conducted and compared. The first measurement is performed shortly before the installation of the rail dampers and subsequently it is repeated with the dampers at exactly the same position as before. The comparison of the pass-by noise emitted for the two configurations is performed for different classes of train categories and train speed.

Method B) *Comparison of pass-by noise at two track sections; with and without dampers.*

Two track sections are chosen for the test. The rail dampers to be tested are installed in one of the sections and the other one is kept as a reference. The two sections should be situated near-by, or preferably adjacent to each other, and be identical in regard to e.g. construction and rail roughness. The difference in noise emission between the two test sections during the pass-by at constant speed is equivalent to the performance of the damper for the particular train category and speed.

Both of these methods have specific benefits but also drawbacks and which one to use should be determined according to the objective of the measurement. The advantage of method A is that it is not required to account for differences between track sections that could influence the damper effect. On the other hand, the trains passing by at the two separate measurement occasions may have significantly different acoustic behaviour. This is especially true for freight trains for which the wheel roughness vary largely between wagons with different types of brakes. For Method B it is the other way around, it accounts for the difference in rolling stock but not for different track sections.

If the objective is to assess the noise reduction at a certain site, either one of these methods can be chosen depending on what is the most convenient solution for the specific case. Most times method B is preferred since the effort is generally lower.

For a reproducible assessment of the efficiency of a rail damper product it is recommended to combine both methods and to perform pass-by noise measurements at two track sections both before and after the installation of rail dampers. However, it should be noted that such measurements could be quite costly and time consuming.

The basic measured quantities are the Transient Exposure Level, TEL or A-weighted equivalent continuous sound pressure level $Lp_{Aeq,Tp}$, train speed and pass-by time Tp. It is recommended to determine the frequency spectrum.

The measured data is to be divided into classes according to train category and speed. The train categories are

a) Passenger trains exclusively with disc-braked coaches

b) Passenger trains with at least one block braked coach

c) Freight trains primarily with cast-iron brake blocks,

d) Freight trains primarily with composite brake blocks

If necessary, additional train categories can be chosen (e.g. high-speed trains, regional trains). Each measured train category is further divided into speed classes, which are determined according to the following predefinition:

- For speeds below 100 km/h, within the class the deviation in speed from the mean value may amount to 5 %
- For speeds above 100 km/h, within the class the deviation in speed from the mean value may amount to 5 km/h

The number of train pass-by measurements that are required for a reliable result depends both on the measured train category and the measurement approach (Method A or Method B). The fact that rail dampers may show a different effect for different train categories can be explained by the fact that rail dampers only reduce the noise from the track and the effect on the total noise is higher for train types with low noise emission. It can also depend on the combined roughness spectrum and if a frequency range is excited for which the damper typed has been tuned.

Generally, freight trains show a larger variance than other rolling stock categories and therefore a larger number of passing freight trains should be measured. If only Method A is chosen (measurement before and after installation of rail dampers) a higher number of measurements are required to ensure statistical reliability.

For further details including e.g. measurement positions, measurement equipment, required number of train pass-bys per train category, data analysis and documentation, see [SR21].

10.1.2 Field test protocol for wheel dampers

When assessing the efficiency of wheel dampers it is, in most cases, not practical to measure commercial trains in operation as in the case for rail dampers described above. A specific test train should be prepared for the measurement campaign. The test train should both have bogies with dampers mounted to the wheels as well as bogies with reference wheels in their initial state without dampers. The efficiency of the wheel dampers is assessed by comparing the noise emission of damped wheels and reference wheels at pass-by with constant speed. All wheels of the test train should be of the same type and the wheel roughness comparable.

There is the possibility to compare the noise emission before and after the mounting of dampers but it is not as practical as the approach proposed above. The train has to be brought to a workshop for the mounting of wheel dampers which could mean that the measurement conditions change between the two occasions. The test train would also have to be at disposal for a longer period of time which would imply higher costs. On the other hand it is the only way to ensure that there are no differences between damped and reference wheels. A combination of both approaches is optional.

The roughness of the damped and the reference wheels has to be measured before the measurements. It is recommended to monitor the total roughness with accelerometers on the rail during the measurements and to repeat the wheel roughness measurements of selected wheels after the tests have been finished.

Some general requirements are:

- All coaches/cars of the test train should be of the same type and unloaded.
- The test train should run under normal operating conditions.
- The wheels should have run at least 1000 km before the tests (2000 km recommended).
- There should be no visual defects on the wheel treads or irregularities such as flats.
- It must be ensured that the measurement results are not influenced by the noise from the power unit. A minimum traction power to maintain a constant speed shall be applied during the pass-by noise measurement

A minimum of three bogies of the test train should be equipped with damped wheels. For locomotive-hauled trains it is recommended that the test train consists of minimum 6 wagons or coaches. The half of the wagons should be equipped with dampers and the other half should serve as reference. If the treated wheels are likely to be very quiet in comparison to the reference wheels, a buffer car with silent wheels (e.g. wheels with dampers) shall be used. Buffer bogies should also be prepared between locomotive and test bogies. These wheels are not to be included in the data evaluation. For trains without a locomotive (multiple units), the absolute minimum number of cars is three (train a: see Fig. 77) and the minimal recommended is four (train b). In any case the absolute minimal number of bogies equipped with damped wheels is three.

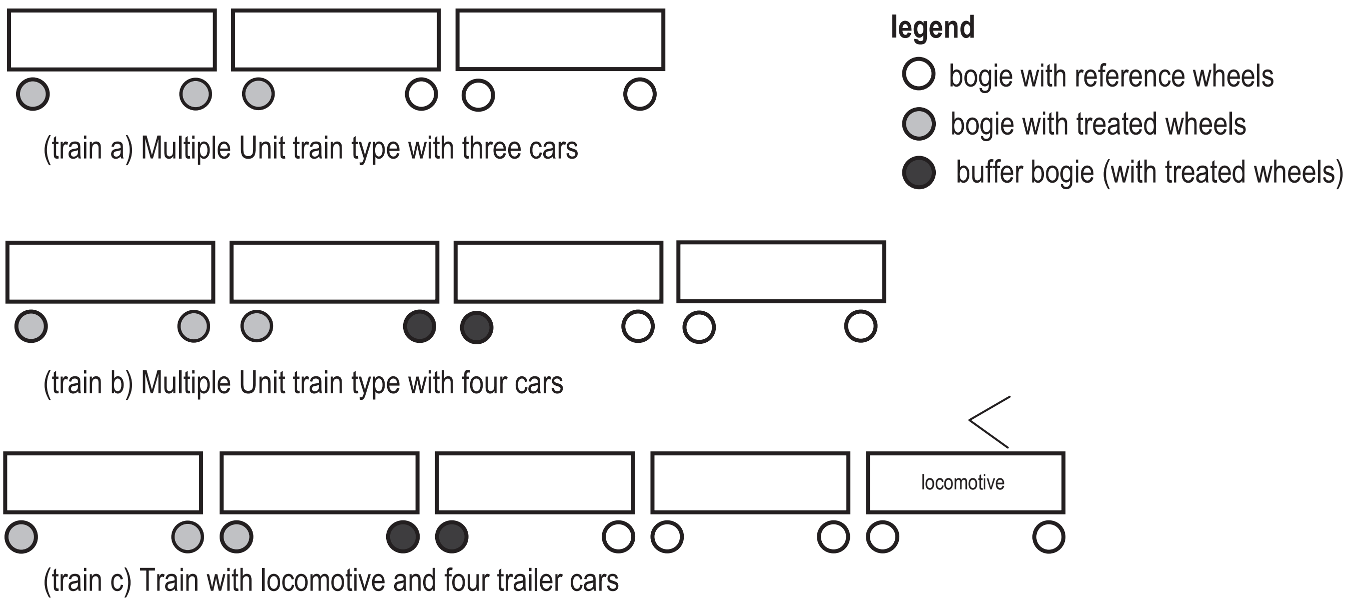


Fig. 77: *View of the composition of the test train for several train configurations*

For further details including e.g. measurement positions, measurement equipment and documentation, see [SR21].

10.1.3 Test site requirements

When performing field tests for reproducible assessment of dampers, the test site should be chosen such that ideally the noise measurements are independent of its specific conditions. The test site environment has to be generally suited for acoustic measurements. The track sections should be straight, without crossings or bridges and the surrounding space must be unobstructed for free sound propagation.

The test track section should be characterized by measuring the rail roughness according to the European standard EN 15610 [9] and the vertical and lateral track decay rates according to the European standard EN 15461 [1].

Optimum test track conditions differ depending on whether it is the efficiency of rail- or of wheel dampers that is to be assessed:

For wheel dampers it is important that the noise emission of the track is relatively low and that the contribution of the wheel can be distinguished from other noise sources in the total noise. If the track noise is dominating it will not be possible to see any effect of wheel dampers in the total noise reduction no matter how good the wheel damper actually is. It is recommended to perform the measurement on a track that complies to the TSI-requirements for minimum TDR. It is preferred that the rail roughness is relatively high since the influence of the wheel roughness is not as decisive and the comparison between different wheels with varying roughness is more reliable.

For rail dampers on the other hand the efficiency is depending on the initial damping of the test track. If the damping is already high, installing dampers will show little effect on the noise emission. It is highly recommended when choosing a test site for the assessment of rail dampers to first make sure that the dynamic behaviour of the test track is such that maximum efficiency of the dampers is possible. For practical purposes it is recommended to perform TDR- measurements prior to the planning of field tests. For more details regarding the recommended procedure, see [SR 21].

It is recommended to grind the track (test section and reference section) before the measurement campaign in order to have uniform rail roughness.

10.2 Field tests

A first application of the test protocols were the field tests planned in STARDAMP [SR18] and presented here. The field tests were performed by SNCF both to test the practicability of the measurement protocols and to produce experimental data for the validation of the STARDAMP method. A test train was partially equipped with wheel dampers and the sound pressure level was recorded during pass-by on a test site equipped with rail dampers and an adjacent undamped reference track.

The acoustic performance of rail dampers was determined by calculating the difference in pass-by noise levels emitted with and without the dampers. The acoustic performance of wheel dampers is determined by calculating the difference in noise level emitted during the pass-by of both bogies with damped wheels and bogies with reference wheels (without dampers). The operating conditions were carefully selected to come as close to the requirements stated in the measurement protocols as possible.

This section presents the main results obtained in the field tests. For full documentation see [SR 18].

10.2.1 Test Train

The test train was an SNCF regional train named VB2N on which wheel dampers of type Vicon RASA RSI manufactured by Schrey & Veit [see 5.3.1, Fig. 78] were installed during a maintenance operation.

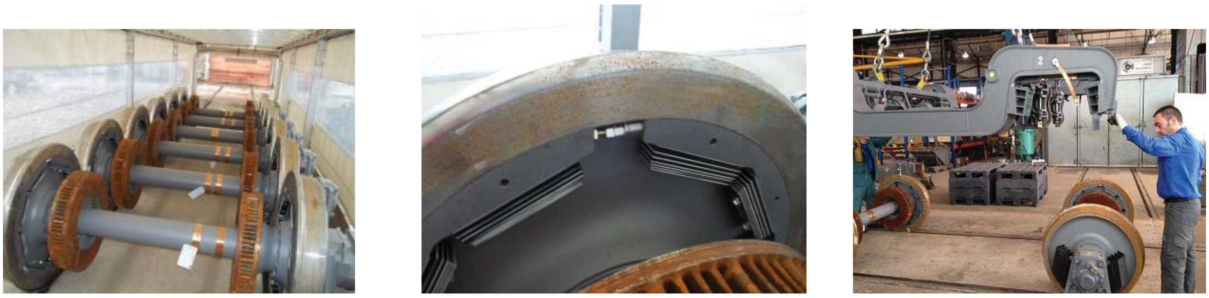
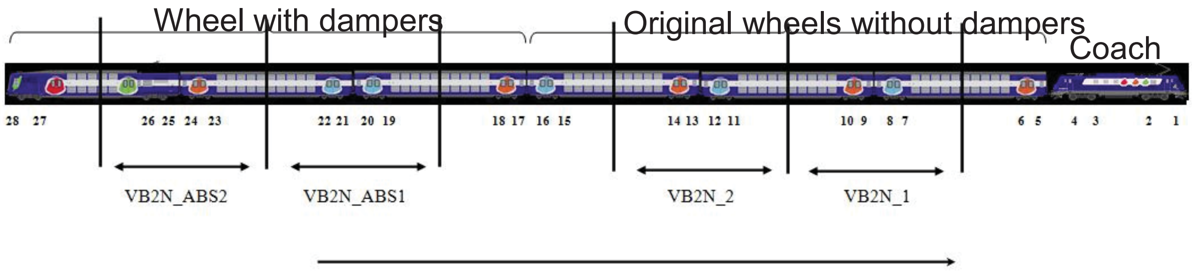


Fig. 78: *VB2N wheels equipped with wheel dampers*

The composition of the test train was the following :



The wheels equipped with dampers are numbered from 17 to 28. The roughness of the wheels has been measured and the spectra are shown Fig. 79. The curves present average roughness spectra of all the test train wheels included in each portion of the train (with and without wheel dampers) and shows that the roughness of the wheels is quite homogeneous. All the wheels equipped with wheel dampers were new and all the wheels without wheel dampers were turned on the axle prior to installation.

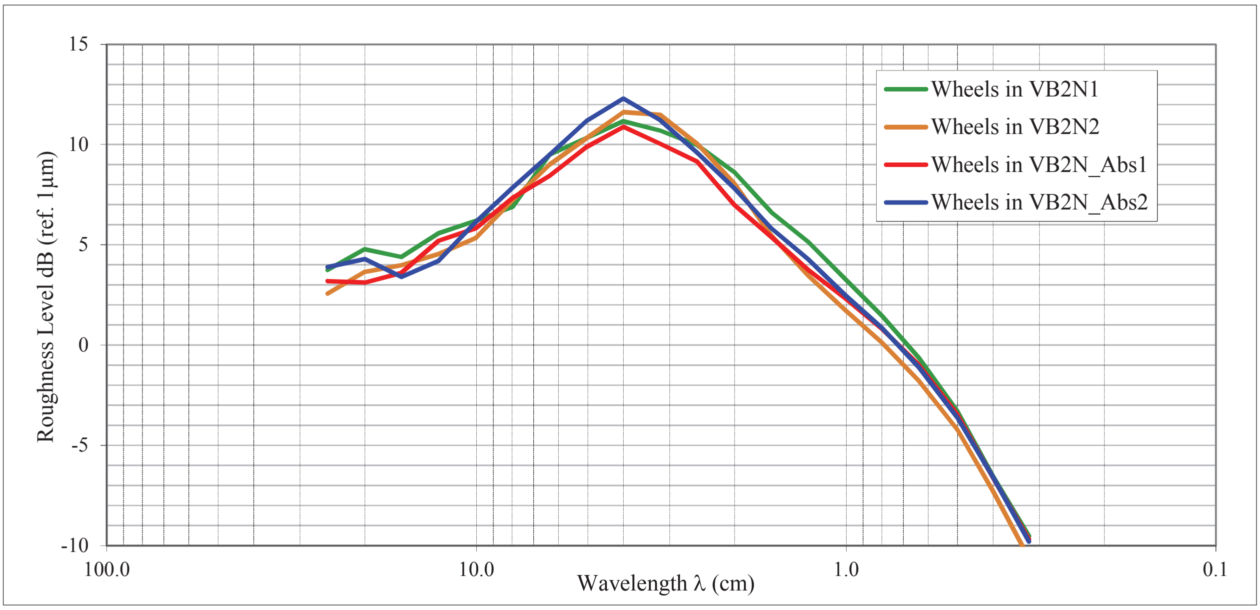


Fig. 79: *Measured roughness spectra of the VB2N wheels*

10.2.2 Site description

The measurements have been performed on the railway line between Paris and Le Havre, on the track situated between the kilometre point 74,800 and 74,400. This site does not fully comply with the requirements concerning the environmental characteristics defined by the EN ISO 3095.

In order to manage the budget, to avoid extra costs and to limit time consumption it was decided to realise the measurements with a commercial train rather than using a dedicated test train. This also limited the choice of the test track where the rail dampers could be installed.

This site has been divided into two adjacent areas. The first is called **reference section (ZR)** which remained in its original state and an **evaluation section (ZA)** in which the rail dampers have been installed. Each area is 200 meters long.

Fig. 80: *Test track for field tests*

The track has the following characteristics :

Radius of curvature	Infinity
Level gradient	0
Rail	UIC 60 (Long welded rail)
Fastening system	Fastclip + Elastic pad 9mm
Sleeper	Concrete bibloc

10.2.3 Application of the measurement protocol [SR21]

Sensors and microphones as listed in table 20 were installed at ZA and ZR. Their position is shown in Fig. 81.

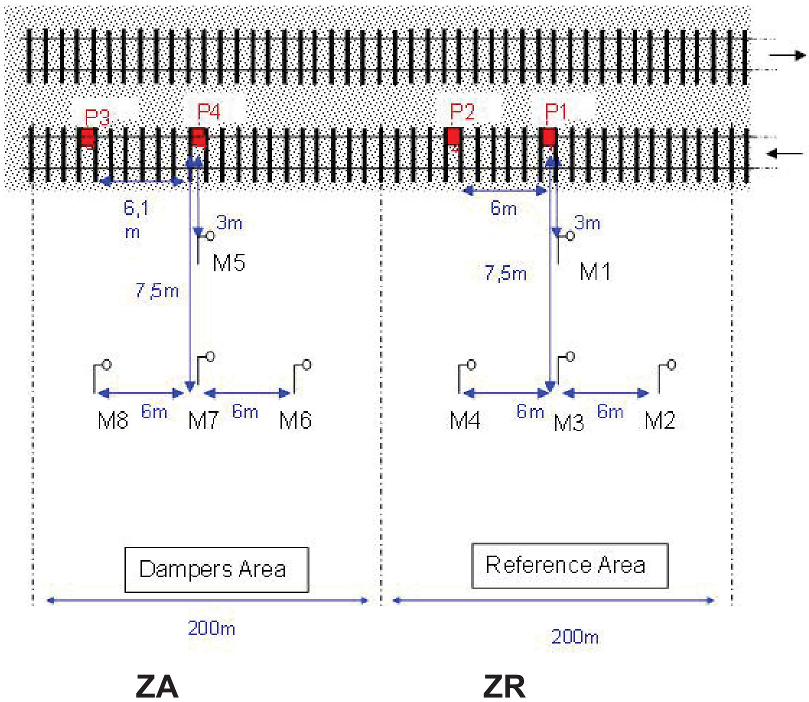


Fig. 81: *Schematic presentation of test section and reference section including position of sensors and microphones*

Sound pressure levels measured by each microphone on both the test site and the reference site during the passby of the same train before the installation of rail dampers differed by at maximum 0,1 dB and, hence, validated the homogeneity of the site (Fig. 82), i.e. homogeneity of track decay rates and of rail roughness.

Table 20: Position of sensors and microphones

Area	Point	Sensor	Distance from the rail	Sensor height/top of the rail	Position along the track
ZR	P1	pedal D50	On the rail (external)		0
	P2	pedal D50	On the rail (external)		6
	M1	Microphone	3 m	1,2 m	0
	M2	Microphone	7,5 m	1,2 m	-6
	M3	Microphone	7,5 m	1,2 m	0
	M4	Microphone	7,5 m	1,2 m	6
ZA	P3	Pedal D50	On the rail (external)	0	6
	P4	Pedal D50	On the rail (external)	0	0
	M5	Microphone	3 m	1,2 m	0
	M6	Microphone	7,5 m	1,2 m	-6
	M7	Microphone	7,5 m	1,2 m	0
	M8	Microphone	7,5 m	1,2 m	6

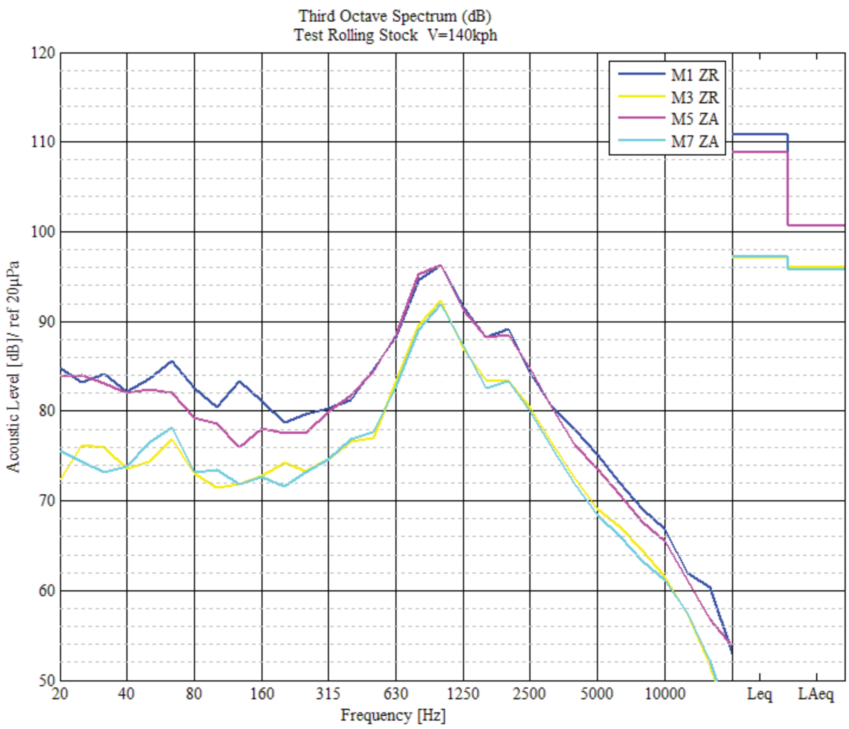


Fig. 82: *Third octave spectrum (dB): one rolling stock at 140 kph on both areas before the installation of rail dampers*

10.2.4 Wheel dampers performance

The wheel dampers performances were evaluated from the comparison of the A weighted equivalent continuous sound pressure level integrated over time during pass-by of the respective section of the test train defined by VB2N_1 and VB2N_2 for the reference wheels and by VB2N_ABS1 and VB2N_ABS_2 for the damped wheels (according the description of the train – see above) from the following equation:

$$\Delta 1 = L_{Aeq\_VB2N\_ABS1} - L_{Aeq\_VB2N\_1}$$
$$\Delta 2 = L_{Aeq\_VB2N\_ABS2} - L_{Aeq\_VB2N\_2}$$

This evaluation was made on the both area ZR (reference area) and ZA (dampers area). Table 21 summarizes the results for the different microphone positions and three speeds. Wheel dampers induce an attenuation between 0.8 and 1.9 dBA on the global noise level (at 7,5m from the track) emitted depending on the type of train, the speed, the variability induced by the measurements, the variability (even if the homogeneity of the track has been checked) induced by the area of measurements and the impact of rail dampers on the contribution of the track (modification of the response of the wheel/rail contact).

Table 21: Measured noise reduction in dB

Test train				
	Speed	80 kph	110 kph	140 kph
ZR	Δ1 dB(A) for M1	-1,3	-1	-1
	Δ1 dB(A) for M3	**-1,9**	**-0,8**	**-1,5**
	Δ2 dB(A) for M1	-1,7	-2,1	-1,7
	Δ2 dB(A) for M3	**-1,3**	**-1**	**-1,3**
ZA	Δ1 dB(A) for M5	-2,3	-0,5	-1,3
	Δ1 dB(A) for M7	**-2,3**	**-1,5**	**-0,1**
	Δ2 dB(A) for M5	-2	-2,3	-1,7
	Δ2 dB(A) for M7	**-1,7**	**-1,9**	**-1,9**

The spectra in Fig. 83 illustrate the effect of the wheel dampers on the reference section (left) and on the test section with rail dampers (right). These figures show that the wheel dampers induce an attenuation in the frequency band between [315-500] Hz and from 1500 Hz on.

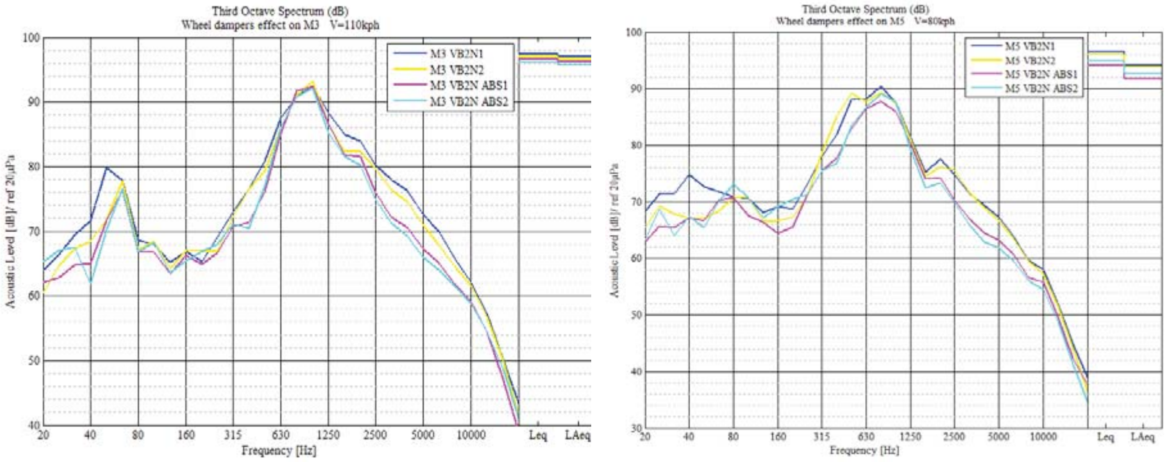


Fig. 83: *Left: Measured sound pressure levels from wheels with/without wheel damper on the reference section (ZR); right: Measured sound pressure levels on the section equipped with rail dampers*

10.2.5 Rail dampers performance

The rail dampers installed on the test track were of type Silent Track ® manufactured by Tata Steel [see 5.3.2 and Fig. 84].

Fig. 84: *Silent Track ® damper manufactured by Tata installed on the test track*

Before measuring the performances of the rail dampers, the two track sections have been characterized by measuring:

- Track decay rates before and after installation of rail dampers
- Rail roughness
- Passby noise level

The measured TDRs are displayed in Fig. 85.

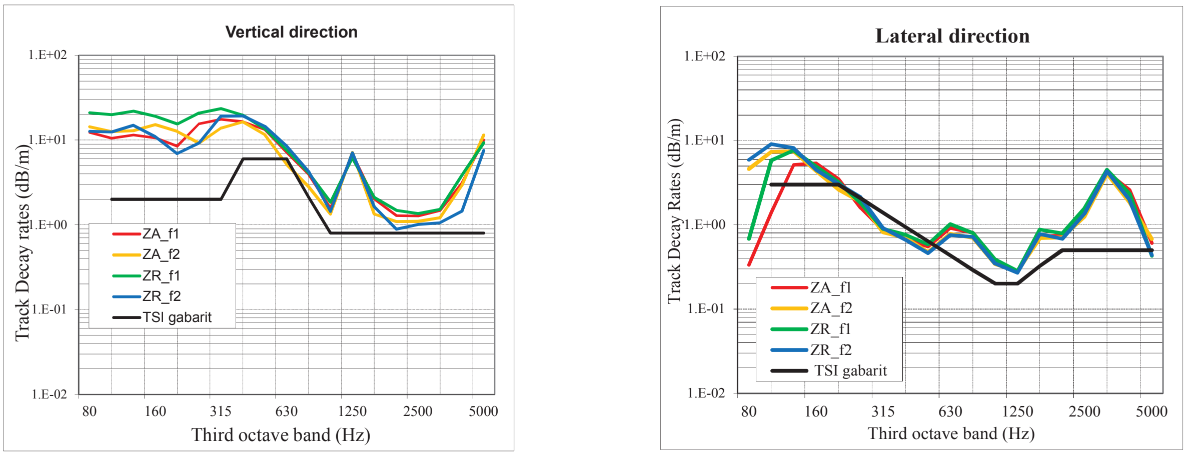


Fig. 85: *Track decay rates in vertical and lateral direction for each file of rail (f1 & f2) and for each section of track tested (ZR = reference area, ZA = futur area for rail dampers)*

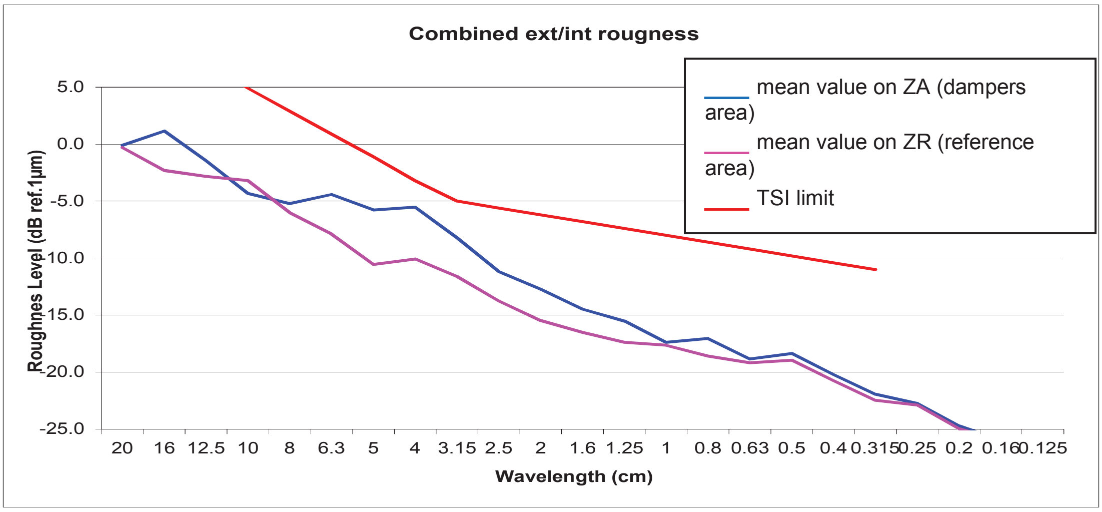
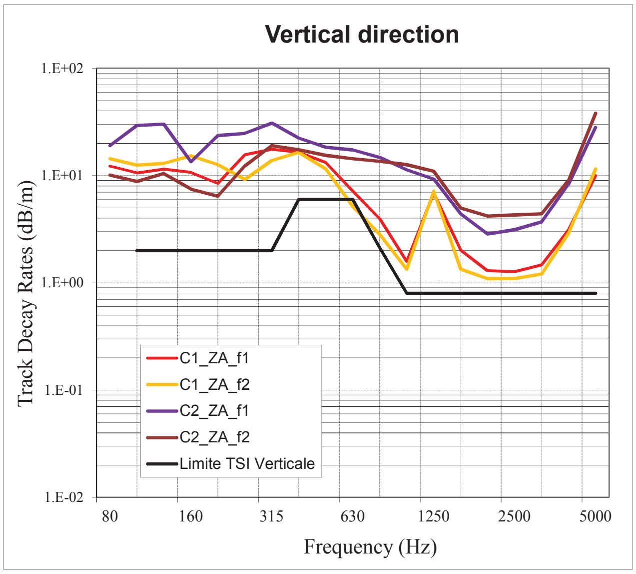


Fig. 86: *Rail Roughness (mean value) on each test area*

Both areas do not fully respect at all frequencies the TSI limits in terms of TDR but have excellent levels of rail roughness (and thus the excitation due to the wheel roughness will be dominant) even if one can observe a difference between the two areas. The roughness correction was applied according to the principal and procedure of the small deviations detailed in annex of the current version of EN ISO3095 (“Method to accept small deviations from acoustic rail rough requirements”).

Rail dampers impact on track decay rates

The following figure demonstrates the increase of TDR (vertical and lateral direction) caused by the installation of rail dampers on ZA. The curves denoted by C1 correspond to the initial situation and those denoted by C2 to the situation after installation of the rail dampers. The rail dampers have a strong impact in the frequency band between 630Hz and 5kHz, as expected, in the vertical direction (except in 1250 Hz band, which is strongly influenced by the pinned-pinned mode) and on the total spectrum in the lateral direction.



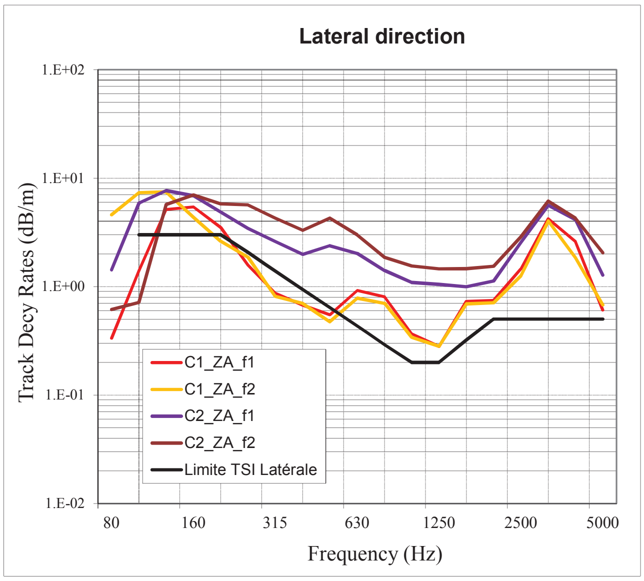


Fig. 87: *Track decay rates measured on the test track before and after the rail dampers installation*

C1 *= measurement* ***before*** *the rail damper installation*
C2 *= measurement* ***after*** *the rail damper installation*
f1,f2 = each file of rail for checking the homogeneity

Rail dampers performance and impact on noise level

The rail dampers performance was evaluated by subtracting the A-weighted equivalent sound pressure levels measured at the reference section (ZR) from those measured at the test section (ZA):

$$\Delta ABS = L_{Aeq,T_p}(ZA) - L_{Aeq,T_p}(ZR)$$

	Test train	Test Train	Test Train	Regional train
Speed (kph)	80	110	140	160
ΔABS dB(A) on M1 (3m from the track)	-3.6	-3.1	-2.6	-3,8
ΔABS dB(A) on M3 (7.5m from the track)	0.0	-0.1	0.0	-0,4

From the table above, the measured effect of the rail damper is in the range of 2.6 – 3.8 dB(A) at a distance of 3m from the track. At 7.5m, the effect is much less than expected and reaches only 0.4 dB(A), which means that based on the measured sum level no effect at all of the rail dampers could be measured within the error bars of the measurement. The reason for this anomalous behaviour becomes visible in the spectra in Fig. 88 (left: speed 80 km/h, right: 140 km/h). At 7.5m one can observe a resonance in the spectra measured at ZA (red curve) around the frequency of 1250 Hz for both train speeds, while this phenomena does not appear in any of the other curves. This peak dominates the sum level measured at ZA (7m) and explains the anomalous behaviour. However, the reason for the occurrence of this peak is unclear. Apart from the 1250 Hz band also at the 7m distance a clear effect of the rail damper is observed. Further investigations have to be performed to identify the origin of this phenomenon e.g. including the possibility of additional radiation of track components.

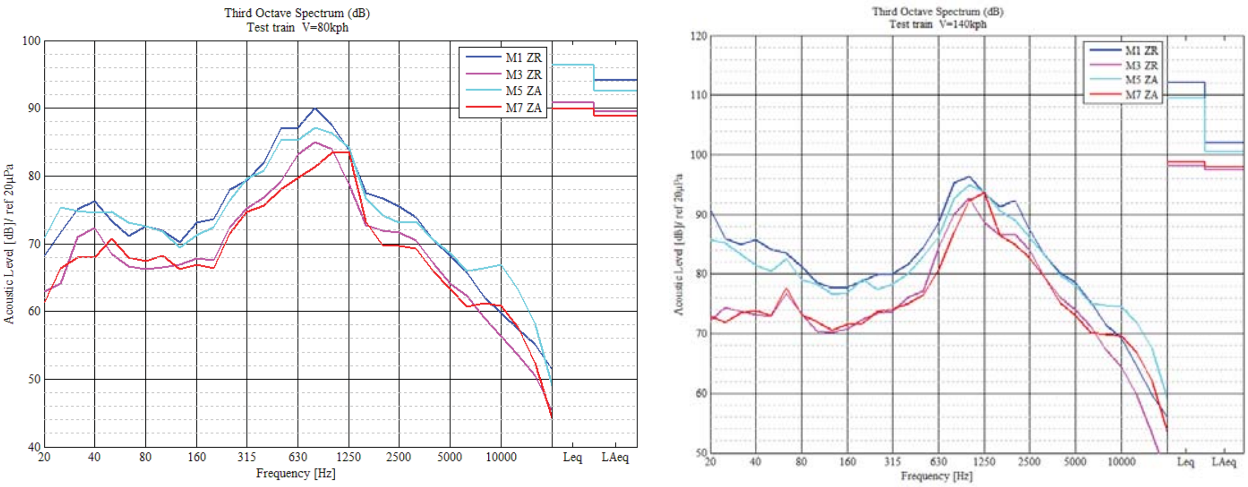


Fig. 88: *Measured sound pressure levels on the reference section (ZR) and on the test section (ZA) at distances of 3m and 7.5m; left: train speed 80 km/h, right: train speed 140 km/h.*

10.2.6 Conclusions

These field tests also represent a first application of the test protocols. It has turned out after the measurement campaign that some adjustments and improvements should be included in the field test measurement protocol. It is required to attach accelerometers to the track and to use an additional « near field » microphone to better separate the contributions of wheel and track. This will support the data analysis when unexpected phenomena occur as it was the case in the STARDMP field test.

Apart from these requirements, the results and specific field test conditions encountered in the STARDAMP field test are representative of the complexity and risks when performing dampers assessment and potentially homologation exclusively on the basis of field tests. Even though being more representative, tests carried out with real trains further increase the level of risks. Real trains on real tracks make the tests extremely costly and time consuming, particularly due to the need to apply for temporarily approval for the use of prototypes in real trains/on real tracks. This experience was also made for the STARDAMP field tests clearly pointing out the need for alternative test methods based on pure laboratory testing.

The following performances of wheel and rail dampers for each of the prototypes tested here have been measured in the field tests:

- The tested **wheel dampers** induce an attenuation in the range **[0.8 – 1.5] dBA** on the global pass-by noise level in agreement with the results obtained from simulation with the STARDAMP tool.
- The tested **rail dampers** have reduced noise in the frequency band between 630Hz and 5kHz. The reason for the increase in the 1.25 kHz band measured at a distance of 7.5 m from the track is not yet clear. The test track in STARDAMP is representative

of previous studies of rail dampers assessments and the effect of rail dampers on TDR could be reproduced [10]. On this basis, we can expect an overall noise reduction between 2.5 to 3 dB(A) as predicted by the STARDAMP tool (for the rail dampers only).

11. Validation of the STARDAMP method

In order to gain common recognition by the stakeholders in the field of damping technologies, the tool application had to be demonstrated, and its reliability validated. It was indeed necessary to check that virtual prediction of a damping device efficiency leads to similar results to the classical field-testing method. This was especially true for a tool that offers the possibility to use default values as important input parameters (like roughness spectra, wheel models and track decay rates).

The validation procedure included two main steps: verification and validation. Both of them are processes by which evidence is generated, and credibility thus established, that the model has adequate accuracy and level of detail for its intended use. Details and results from the validation of the STARDAMP tool can be found in the report [SR 25].

11.1 Verification step: Sensitivity analysis on academic cases

The first step of verification was a sensitivity analysis, made by varying the most influent parameters, like track decay rates, combined roughness or speed of the train, to check if the results showed the variation expected from a simplified analytical approach. The input files were simplified (unity values or same multiplication factor over the whole frequency range), so that the results could easily be predicted.

The goal of this step was to check that the tool gives significant and relevant results for simple cases, that the mathematical model behind the tool is correct, and that there are no programming errors that could affect the results.

The reference case was defined in the STARDAMP tool according to the table 22 below. This was the case with whom all other results were compared.

Table 22 - Configuration in the STARDAMP tool for the Reference unitary case

Test Nr	Rough. (dB)	Traffic	Speed	Wheel	Track	Sleeper	Pad stif.	Pad damp.	TDR (dB/m)
1	From file (Unity)	Regional	100 km/h	Default (LK900)	ballast	Concrete monobloc	medium	normal	From file (Unity)

Then, several other tests were performed:

- Variation of the speed: 144 and 200 km/h,
- Variation of the roughness: doubled and quadrupled roughness (in metric units),
- Variation of the track decay rate: doubled and quadrupled TDR + checking of the definition of a damped rail.

All these calculations gave the results expected by analytical calculations, making it possible to conclude on the good functioning of the tool toward these parameters.

An example of a verification case is shown in Fig. 89 below. It corresponds to the case were the initial unitary combined roughness is doubled or quadrupled. As expected for this simple case, the total Sound Pressure Level is directly proportional to the level of combined roughness.

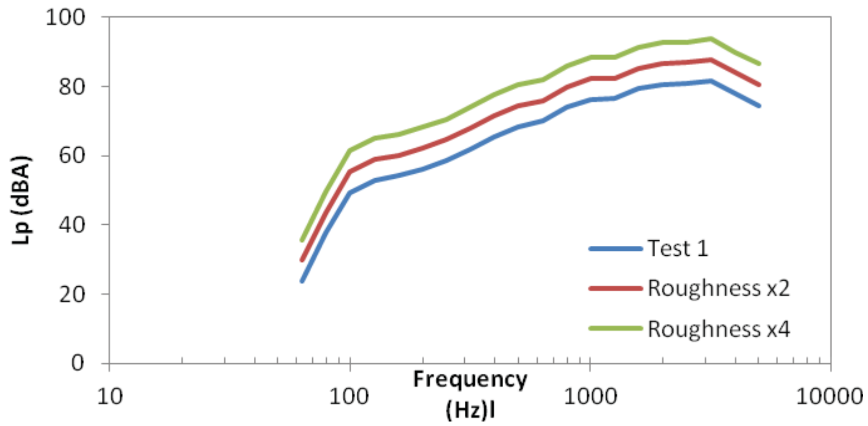


Fig. 89: *Verification case on the influence of the combined roughness on the total SPL*

11.2 Validation step: Comparison with TWINS

The validation step checks if the computational model accurately represents the underlying mathematical model and its solution, and that the result of the calculation are relevant regarding real cases. In order to do so, calculation results can be compared with measurement data, or they can be validated against calculation results from software that has already been fully validated.

The STARDAMP tool has been implemented with the same mathematical model as TWINS, which is a software tool that has been fully and successfully validated in previous research work (see ref. [11] and [12]). A way to check that there were no programming errors in the STARDAMP tool was to calculate the same scenarios in both tools and compare the results. As far as possible, the same inputs were used in the two tools, and the comparison was made on the SPL spectra for the wheel, the rail, the sleepers and the sum of them, in the interval of interest which is set to [100, 5000 Hz].

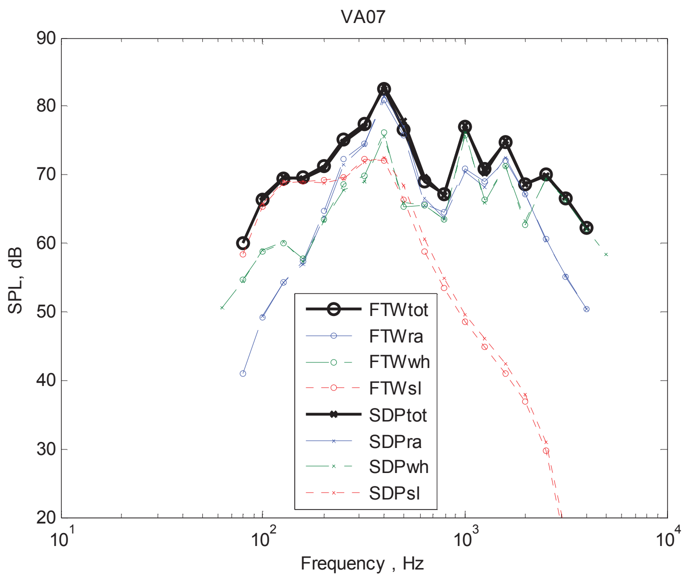


Fig. 90: *Example of a verification case between STARDAMP and TWINS calculations: Bibloc sleeper model, freight traffic and measured TDR*

The comparison showed a good agreement between the results, confirming that the codes implemented in TWINS and STARDAMP were corresponding. Results show deviation between TWINS and STARDAMP results no greater than 1 dB for all components (track and wheel). Nevertheless, a difference was pointed out in the behaviour of monobloc sleepers, but a further investigation showed that STARDAMP offered the more reliable model, by taking all geometrical characteristics of the sleeper into account.

It was a necessary step to assess that the results obtained with the STARDAMP tool have a good coincidence with real cases, by giving good accordance with results from TWINS, whose reliability has been fully validated.

Another validation step, which is still on-going in the framework of a STARDAMP tool user group, is a comparison of calculation results with measurement data from the STARDAMP measurement campaign and other previous well documented research projects. This is the occasion to test the tool in many real situations, using different types of track and rolling stock, for different speeds, and test the accuracy of default values against real cases. The first comparison results were quite encouraging, and even if some discrepancies sometimes occurred, they were most of the time explainable by the quality of the measurement campaign or inputs. But confidence in STARDAMP calculation results is mostly based on the validation towards TWINS, which was giving really good results.

All these steps contributed to conclude that the STARDAMP tool is a valuable software tool to help the users predict levels of noise reduction that can be expected from the installation of wheel dampers, rail dampers or a combination of both of them. The simplified interface makes it easy to use, even for non-acoustician experts, although a basic knowledge is still required to fully analyse the results of the calculation and to draw the right conclusions. But despite an apparent simplicity, the software offers a good level of reliability which makes it a good alternative to expensive field tests that would have been required to verify the effectiveness of a damping configuration.

12. Documentation of the STARDAMP results

The results of STARDAMP are documented in the following reports and documents:

[SR 1] B. Asmussen: Definition of general requirements of end users (DB)

[SR 2] TU Berlin: Measurement protocol durability assessment handbook for laboratory tests (TUB)

[SR 3] M. Toward, D. Thomson: Laboratory tests for rail dampers (ISVR)

[SR 4] P. Kitson: STARDAMP testing – long rail tests at Tata Steel (Tata Steel)

[SR 5] G. de Ana Rodríguez, C. Sánchez Martín, D. Bumke, K. Buggisch: Measurement report rail dampers – acoustic and fatigue tests (TU Berlin)

[SR 6] H. Venghaus: Round Robin Test – Rail Dampers (Schrey&Veit)

[SR 7] M. Starnberg, B. Lawrence: Measurement report - Track decay rate measurements on real track (DB)

[SR 8] M. Toward, D. Thompson: Results of the Round Robin Test for rail dampers: Summary and conclusions (ISVR)

[SR 9] D.J. Thompson: Laboratory test method for rail dampers – specifications for ‘free rail’ tests (ISVR)

[SR 10] M.G.R. Toward, D.J. Thompson: Laboratory test method for acoustic rail dampers – WP 1/1.3 rail. ISVR Consulting report 8810-R02, July 2012 (ISVR)

[SR 11] H. Venghaus: STARDAMP project WP1.3 Round Robin Test – wheel dampers (Schrey&Veit)

[SR 12] B. Betgen, P. Bouvet: VIBRATEC report for STRADAMP reference 562.012.RA.03.A Measurement protocol proposal for wheel dampers and measurement results obtained at VIBRATEC (Vibratec)

[SR 13] B. Betgen, P. Bouvet: VIBRATEC report for STRADAMP reference 562.012.RA.02.A: Illustration of the “EMA + TWINS” calculation procedure and calculation results for Stardamp wheels (Vibratec)

[SR 14] Alstom transport: Measurement report wheel dampers (ATSA)

[SR 15] G. de Ana Rodríguez, C. Sánchez Martín, M. Metzger, D. Bumke, B. Wette: Measurement report wheel dampers – acoustic and fatigue tests (TU Berlin)

[SR 16] F. Demilly, T.Ingouf, Y. Flament: Valdunes Round Robin Tests – Wheel dampers - WP 1 / 1.3 Measurement report - LK900 Wheel set performed at INSA / Valdunes laboratory (Valdunes)

[SR 17] M. Starnberg: Measuring TDR on undamped real tracks – Best practice rules (DB)

[SR 18] C.Mellet, F.Margiocchi: Measurement report field test (SNCF)

[SR 19] B. Asmussen: Tests on the RASP test rig of DB Systemtechnik in Brandenburg/Kirchmöser (DB)

[SR 20] B. Betgen, H. Venghaus: Results of the round robin test wheel dampers - summary and conclusion (Vibratec, Schrey & Veit)

[SR 21] F.Margiocchi, C.Mellet: Measurement protocol field tests (SNCF)

[SR 22] G. Squicciarini, D.J Thompson: Description of the STARDAMP software (ISVR)

[SR 23] C.J. Jones : Help file and documentation for mp_editor (ISVR)

[SR 24] L. Krüger, Measurement Report Stardamp WP 1.3 Round Robin Test Wheel Damper June 2012

[SR 25] M. Starnberg: Report on the validation of the STARDAMP software (DB)

[SR 26] L. Pesqueux, STARDAMP project - guidelines (Alstom)

[SR 27] Laufradsatz LK900, drawing A-1-107947 A, GHH Radsatz, 01/06/2011

[SR 28] Vollrad LK 900, drawing A-1-107974 A, GHH Radsatz, 01/06/2011

[SR 29] P. Kitson : Damper Durability Test Procedure (Tata Steel)

13. STARDAMP guidelines

The guidelines of the STARDAMP project are presented and given in [SR26]. In a first part, the existing protocols to evaluate the damper performance are summarized. In a second part, methods developed within STARDAMP are given.

The STARDAMP assessment method comprises two steps:

1. Laboratory testing of dampers in order to measure specific physical properties described in [SR26] (§3.4.3, §3.4.4, §3.4.5)
2. Application of a software tool (TWINS based) described in [SR26] (§3.4.2) for calculating the performance of a damper for a specific application with the results of the laboratory tests as input parameters

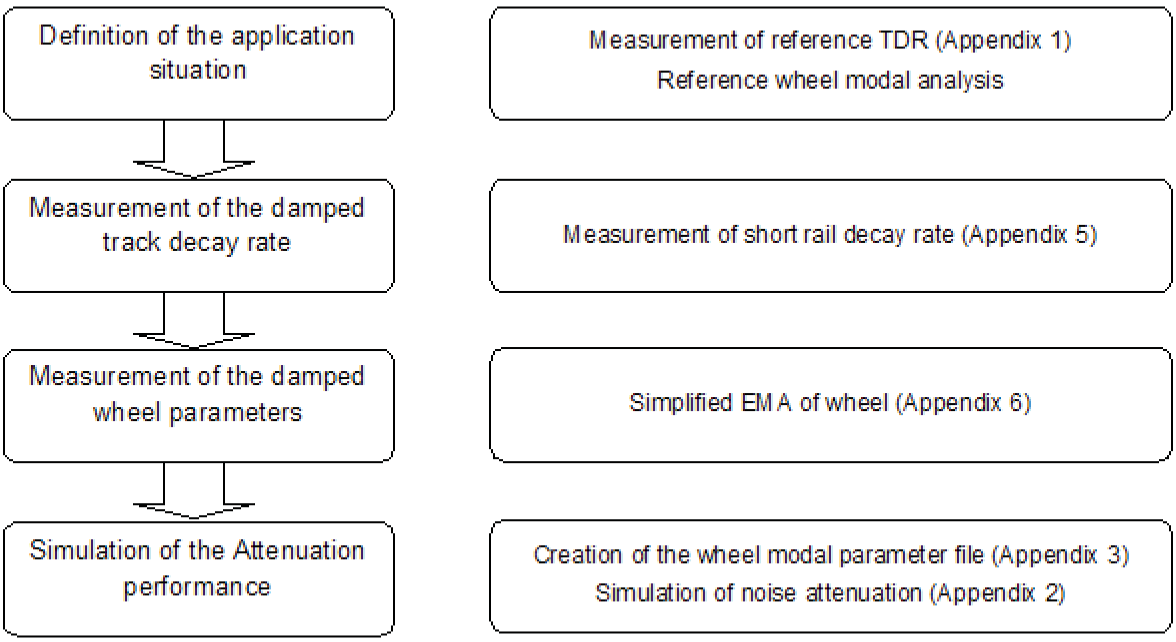
The STARDAMP method enables to assess the performance ΔLp of dampers in a specific application.

Measurements done with standardized protocols obtain the input data for noise prediction.

Wheel parameters are formatted for simulations via the MP_editor program.

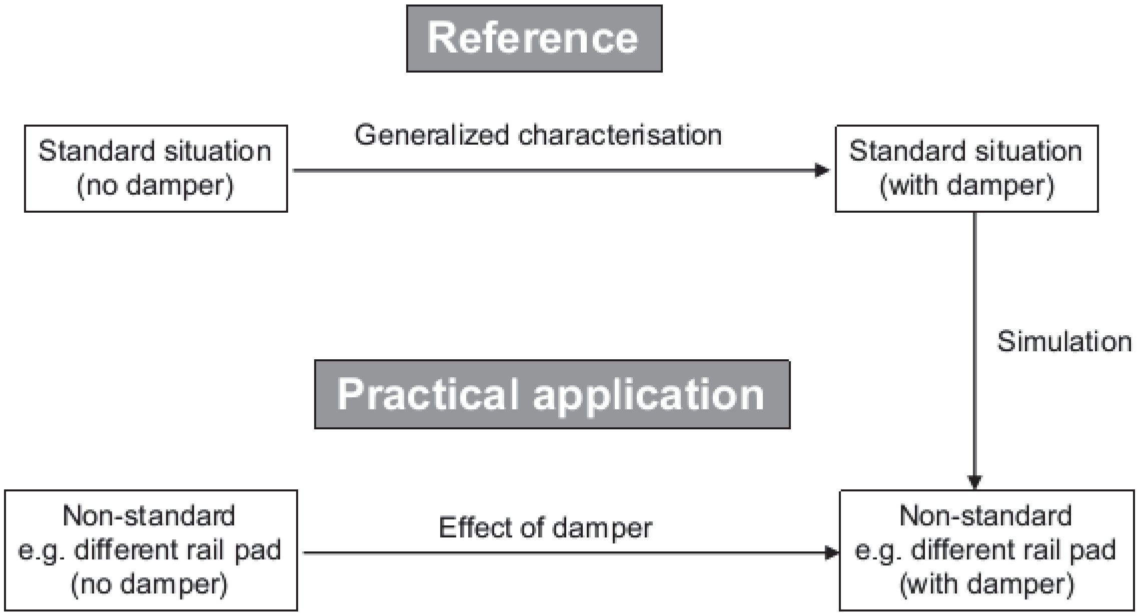
The STARDAMP simulation tool assesses the wheel and/or rail dampers performance.

The Figure below summarises the steps and recommendations of the STARDAMP methodology and refers to [SR26] (§3.5):



A reference system has been defined to better estimate the performance of a damped solution. With this method we can easily compare different products with each other.

Transfer from the reference situation to a non-reference situation can be achieved by application of the STARDAMP software.



The STARDAMP method relies on the combination of several measurements and computer simulation. Laboratory measurement protocols have been prepared, tested and optimized.

For rail dampers, demonstration was made that decay rate measurements on a 6m suspended rail are sufficient, combined with the undamped track decay rate, to characterize the damped track decay rate.

- For wheel dampers, demonstration was made that a simplified experimental procedure (the simplified experimental modal analysis SEMA), which only needs measurements of a reduced set of FRF, was often sufficient to characterize the damped wheel parameters.
- Fatigue testing guidelines were produced to evaluate the long-term performance of both wheel and rail dampers.

Field testing of damper properties have been carried out, and field test methodologies have been optimized:

- A complete field test protocol has been tested and implemented at SNCF.
- An analysis of the variability of track decay rate measurements have been carried out at DB, and recommendation while performing Standard EN 15461 measurements.

Complete guidelines including all test methodologies and recommendations as standalone annexes have been compiled and made available to the general public for the use of the STARDAMP method in future projects [SR 26].

14. Dissemination of the STARDAMP results

The STARDAMP project and its results have been presented on several conferences and workshops and in journals. A final workshop exclusively dedicated to STARDAMP has been organized by the consortium.

14.1 STARDAMP final workshop

For presenting the STARDAMP results to all interested stakeholders in the field of damping technologies for reducing rolling noise, a public workshop was organized on 18th of October 2012 at DB Systemtechnik in Munich with the following agenda:

9:00 Welcome and STARDAMP project overview

Alfred Hechenberger (DB Systemtechnik)

Bernd Asmussen (DB Systemtechnik)

Alexander Spieshöfer (Projektträger MVt)

9:30 Challenges of innovative noise reduction

Maria Starnberg (DB Systemtechnik)

9:45 Motivation for STARDAMP

Florence Margiocchi (SNCF)

10:00 Damping of rail and wheel (fundamentals)

David Thompson (ISVR, University of Southampton)

10:30 Coffee break

11:00 Measurement Protocols, Laboratory measurements, Round robin test

General remarks -Helmut Venghaus (Schrey&Veit)

Lab testing of rail dampers -Paul Kitson (Tata Steel)

Lab testing of wheel dampers -Thomas Gerlach, Francois Demilly (GHH-Valdunes)

12:00 Demonstration of the STARDAMP-software tool

Introduction -David Thompson (ISVR, University of Southampton)

Application to wheel dampers -Benjamin Betgen (Vibratec)

Application to rail dampers -David Thompson (ISVR, University of Southampton)

13:00 Lunch

14:00 Coffee and chance to see exhibition

14:30 Field tests and validation

Florence Margiocchi (SNCF)

14:50 STARDAMP method – recommendations and best practice rules

Lise Pesqueux (Alstom Transport)

15:15 Discussion

16:00 DB Rig tests for wheel dampers

Bernd Asmussen (DB Systemtechnik)

16:20 Testing rail dampers in Switzerland

Enzo Scossa Romano (SBB)

16:40 Conclusions and final remarks

Florence Margiocchi (SNCF)

17:00 End

The workshop attracted 95 participants from 12 European Countries, which was a clear indication for the large interest in the subject of the project and in particular for the huge demand for a generally agreed reliable assessment method for dampers. Participants represented railway operators, infrastructure managers, suppliers, consultants, research universities and universities.

Fig. 91: *STARDAMP final workshop, Munich*

14.2 STARDAMP publications

Maria Starnberg: *Challenges of innovative noise reduction*, presentation Stardamp Workshop, 18/10/2012

Florence Margiocchi: *Motivation for STARDAMP*, presentation Stardamp Workshop, 18/10/2012

David Thompson: *Damping of rail and wheel – fundamentals*, presentation Stardamp Workshop, 18/10/2012

Helmut Venghaus: *Measurement Protocols, Laboratory measurements, Round robin test-general remarks*, presentation Stardamp Workshop, 18/10/2012

Paul Kitson: *Laboratory Testing of Rail Dampers*, presentation Stardamp Workshop, 18/10/2012

Thomas Gerlach, François Demilly: *Measurement Protocols, Laboratory measurements, Round Robin tests- Lab testing of wheel dampers*, presentation Stardamp Workshop, 18/10/2012

David Thompson, Benjamin Betgen: *The STARDAMP software tool*, presentation Stardamp Workshop, 18/10/2012

Florence Margiocchi: *Field tests and validation*, presentation Stardamp Workshop, 18/10/2012

Lise Pesqueux: *The STARDAMP method Recommendations and best practice rules*, presentation Stardamp Workshop, 18/10/2012

Bernd Asmussen: *Roller Rig Tests for Wheel Dampers*, presentation Stardamp Workshop, 18/10/2012

B. Betgen, P. Bouvet, D. Thompson, F. Demilly and T. Gerlach: *Assessment of the efficiency of railway wheel dampers using laboratory methods within the STARDAMP project*, proceedings of Acoustics 2012 Nantes conference

B. Betgen, P. Bouvet, G. Squicciarini and D. J. Thompson: *The STARDAMP Software: An Assessment Tool for Wheel and Rail Damper Efficiency,* DAGA 2013

Florence Margiocchi: *Stardamp Deufrako Project – Standardization of damping technologies for the reduction of railway noise,* UIC Sustainability Conference, Venice, 2012

Upcoming events in 2013

Thomas Gerlach, Christian Kemp-Lettkamp: *Standardization of damping technologies for the reduction of railway noise*, to be presented at 17th International Wheelset Congress, Kiev/ Ukraine, 22nd – 27th September 2013

B. Betgen, P. Bouvet, G. Squicciarini and D. J. Thompson: *Estimating the performance of wheel dampers using laboratory methods and a prediction tool,* to be presented at IWRN 11

Florence Margiocchi: *Stardamp Deufrako Project – Standardization of damping technologies for the reduction of railway noise,* Internoise 2013, 15-18/09/13, Innsbruck

Florence Margiocchi: *Stardamp Deufrako Project – Standardization of damping technologies for the reduction of railway noise,* WCRR 2013, 25-28/11/13, Sydney

14.3 Distribution of the STARDAMP software

The STARDAMP software has been distributed to all the partners of the project. After the end of the project, it will be made available for non-partners of the project as a commercial product.

VIBRATEC will be in charge of the distribution of the software. VIBRATEC will manage all end users and sales contacts for all services:

- Sale of the software
- Maintenance,
- Technical Support,
- Rolling Noise Technical Trainings,
- user's meeting.

Some of this services will be co-organised with ISVR (for example: maintenance and training).

The first training session and user's meeting will be planned in the 2nd semester of 2013.

14.4 Revision of the standard EN 13979-1

The EN standard EN 13979-1, named "Railway applications – Wheel-sets and bogies – Mono-bloc wheels - Technical approval procedure - Part 1: Forged and rolled wheels" describes the technical approval of railway wheels, including the acoustical assessment in chapter 8 and annex E of the standard. As also the TSI approval of railway wheels refers to this standard, it is the main design guideline for wheels in Europe. The main requirement concerning the acoustic behaviour in this standard is that a new wheel design should be less noisy than a reference wheel. For the assessment of the acoustical behaviour two methods with their limitations are described. The first method is based on calculations. A validated calculation method is required, therefore the TWINS software is explicitly recommended. The second method is to perform field tests.

This standard is under revision in the European working group CEN/TC 256/SC 2/WG 11 at the moment. The working group 11 works on wheel set standards and its members are from railway operators, car builders and wheel set producers. As there is a big overlap of the acoustical section of the standard with the content of the STARDAMP project concerning wheel noise, the progress made in the STARDAMP project shall be introduced into the standard.

In general there are two possible fields of improvement from the project: The field test method of the standard could be updated with the field test measurement protocol of STARDAMP. The second possible improvement concerns the calculation method described in the EN Standard. So far due to the restrictions of the TWINS software the standard recommends to perform field tests for wheels with dampers. Here the STARDAMP software tool is proposed to be introduced as an alternative approach based on calculation. Nevertheless it is necessary that the STARDAMP software tool is validated first.

The benefit would be high for all users: Being in compliance with the European standard EN 13979-1 the STARDAMP method for wheels could be used for the development and at the same time for the approval of the acoustical behaviour of wheels with dampers. At the same time the applicability and efficiency of the standard EN 13979-1 would be improved with regard to acoustics.

15. Progress in the field of the project outside STARDAMP

Installation and testing of rail dampers have continued during the last two years in several European countries. The following list gives some examples:

a. Infrabel has installed rail dampers on several test sites in Belgium. Four different types of product are currently being tested. Installation has been performed on tracks with different rail pad stiffness. Also installations on steel bridges are included.

b. In the United Kingdom one installation of rail dampers has been carried out on the Thameslink project. Full assessment of their effectiveness has not yet been completed.

c. In Germany rail dampers have been installed on several locations within the 'Konjunkturprogramm II'. Measurements were performed to quantify the achieved noise reduction. It is foreseen to include rail dampers in the German noise legislation in the near future.

d. In Austria several rail dampers have been tested during the last years. To become an overall view of this new railway noise reduction technology, the tests contain different products. Straight track and curves, ballast track and situations on steel bridges were chosen as test sections.

e. SBB has carried out an extensive network-wide measurement campaign in order to estimate the potential effect of rail dampers. Four types of dampers have been tested. Decay rates have been measured on softly supported rails and track decay rates on several real tracks in Switzerland. The overall target was to estimate the potential of rail dampers on the Swiss railway network. Close cooperation has been established with the STARDAMP activities (see Sec. 3.5).

16. Lessons learned and recommendations from the project

The measurements performed in the project show the necessity of accurately performed measurements as well as precisely described test procedures. The influence of different parameters on the acoustic result was investigated in the Stardamp project. This should be taken into account for further measurements. Crucial points were especially the measurement of TDRs and the measurement of damping ratios for highly damped wheel modes. Better control of variability and source of uncertainty is needed. This includes in particular

- temperature
- variability of measurements: clear measurement protocols shall help to minimize the variability
- variability between dampers (related to the manufacturing process and to the variability of the fixation system of the dampers on the wheel)

For the assessment of the mechanical integrity of wheel dampers it was difficult to find a practical test method due to the high accelerations of the running wheel and the mass of the wheel. The high accelerations cannot be simulated with complete wheels on commonly available testing machines. As no general rules for assessment can be given, it is recommended to choose tests, which are related to the concept of each damper.

Concerning durability and ageing of dampers further results from long term tests and experiences from the service are needed.

Application of the STARDAMP software in practice should be further monitored. During the validation phase of the STARDAMP software it turned out to be important to share experience among the users. Therefore it is recommended to organize regular user's meeting to ensure the performance of the tool, to simulate a wide variety of applications and to establish the scope of application of the tool.

The acoustic field test performed by SNCF involved a modification of rolling stock, mainly in the wheel area. Getting access to the French railway network with the modified rolling stock implied checking the conformity with the law 01 of July 2004. The requirement is to perform a safety check of the wheel/damper system as the damper is fixed into a groove near the wheel rim.

Thirteen topics from the law 01 of July 2004 had to be assessed and evaluated by experts of rolling stock. The main subject concerned the dynamic behaviour of the new wheel modified. Finally, it took about 6 months to get the approval for the modification of the rolling stock and the access to the French railway network. The approval was accepted only for this test campaign. So for a series modification, the requirement for reliability of the system will be more demanding. This clearly demonstrates the need for testing procedures that do not require field tests.

17. Additional work

A number of additional activities have been carried out during the execution of the STARDAMP project, which were originally not included in the work programme.

The IRSID method was additionally included in the round robin test for wheel dampers (see sec. 7.3) as it can provide information complementary to the EMA method. Though originally not included in the STARDAMP work programme, it was decided in the starting phase of STARDAMP to include it in the round robin test for wheel dampers and to explore the potential of the method as it can provide important information about the impact of the damping system on the screening effect.

In the course of developing the assessment method for rail dampers, it has turned out that accurate track decay rate measurements are crucial for the reliability of the method.. Since TDRs measured on a real track are an important input parameter for the STARDAMP software, it was decided that the factors causing the discrepancies were to be further examined in a measurement campaign. Based on the conclusions from the measurement campaign best practice rules have been worked out ([SR 17]).

During the validation phase of the STARDAMP software it has turned out to be important to share experience among the users. Therefore it is recommended to offer participation in a user group.

18. Conclusions

Within the frame of the STARDAMP cooperation a method has been developed, which allows to predict the efficiency of dampers for rail and wheel. A software tool has been developed for calculating the efficiency of rail dampers and of wheel dampers and of the combination of both. Basic input parameters required by the software are decay rates and track decay rates for rail dampers and modal parameters for wheel dampers. Test procedures have been defined to generate these input parameters in order to ensure maximum reliability of the results and in particular to guarantee that different users of the method obtain equal results. Based on the ‘6m rail method’ elaborated for rail dampers in STARDAMP, easy criteria or limit curves can be defined to ensure that rail dampers meet certain requirements (e.g. for homologation purposes).

The STARDAMP software tool is divided into two parts. The first part is used to make rolling noise predictions to assess wheel or rail dampers. The second part contains the modal parameter file editor (mp-editor), which is used to create input files for the STARDAMP tool for damped and undamped wheels from a finite element model of the wheel section. The STARDAMP tool implements TWINS-like predictions of rolling noise in a user-friendly way. Compared with TWINS the number of options is limited in order to allow access by non-expert users. A comprehensive documentation of both software and the full STARDAMP procedure have been elaborated.

Reference conditions have been defined in terms of wheel and track. The performance of a damper may be calculated with the STARDAMP tool either with respect to reference conditions or with respect to conditions defined by the user. Results are given both as single number and in the form of spectra.

Sources for uncertainties and variability of the parameters influencing the performance of a damper have been identified and suggestions for further investigation have been given.

Procedures for rail and wheel dampers to simulate the long term impact of environmental conditions (ageing) and of varying load cycles caused by the passing of trains (mechanical integrity) have been defined and documented in a handbook.

The STARDAMP method has been validated on different levels. This included round-robin tests for generating the input parameters for the laboratory tests and a field test with a train equipped with wheel dampers on a damped track. A measurement protocol for efficient conduction of field tests has been elaborated.

19. References

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[7] EN ISO 3741, Acoustics – Determination of sound power levels of noise sources using sound pressure. Precision methods for reverberation rooms

[8] W. Behr, Innovationsprojekt “Leiser Zug auf realem Gleis” (LZarG) final report, T.TVI32(1), DB Systemtechnik

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