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Assessment of the efficiency of railway wheel dampers using laboratory methods within the STARDAMP project

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Noise is one of the major issues for the expansion of railway traffic. Within a relatively wide speed range, rolling noise is the predominant railway noise source. In recent years, rail and wheel absorbers have been developed by different manufacturers that show to be effective devices for the reduction of rolling noise. STARDAMP (Standardization of damping technologies for the reduction of railway noise) is a Franco-German research project within the DEUFRAKO framework that unites end users, manufacturers and research institutes. The target of STARDAMP is to support the transfer from R&D of wheel and rail dampers to their regular application. A key factor in this context is the development of new testing methods for the assessment of damper performances. Today, such tests are usually performed as field tests that are costly and time consuming. The intention is to replace these by standardized laboratory measurement and calculation techniques. The present contribution deals with a proposition for a wheel absorber testing protocol, combining finite element calculations, experimental modal analysis and analytical calculations using TWINS. The influence of the main parameters is discussed and results for different absorbers are presented.

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1 Background

During the past decade, several national and European projects have been conducted in the aim of understanding railway noise sources and developing mitigation measures. SILENT FREIGHT [1], SILENT TRACK [2], STAIRRS [3] and SILENCE [4] shall be cited here. These projects have permitted to provide practical criteria to decision makers and to develop consensus between legislators, railway operators, railway infrastructure managers and the railway supply industry. Also, rail and wheel dampers have clearly been shown to be promising candidates for action plans within phase II of the Environmental Noise Directive (“END”) [5].

Yet, the step from R&D of damping devices to their regular application remains challenging. One reason is the lack of standardized measurement methodologies to assess the effectiveness of rail and wheel dampers. The definition of such methodologies is the main purpose of STARDAMP. A general description of the project has been presented at DAGA 2012 [5].

This paper deals with the proposal for a wheel absorber measurement protocol combining finite element calculations, experimental modal analysis and analytical calculations using TWINS (Track-Wheel Interaction Noise Software) [7]. Within STARDAMP, a software tool based on TWINS has been developed with the aim of providing a tool to a larger public, including non experts. The procedure described here also applies when using this software instead of TWINS.

2 Rolling noise in a nutshell

This section gives a very short introduction to rolling noise and its mitigation measures. For a more detailed description we refer to reference [8].

The origin of rolling noise can be found in the asperities that are present on any wheel tread and rail head. This surface roughness introduces a relative displacement of wheel and rail and causes both components to radiate noise. Wavelengths of roughness that are relevant to rolling noise are approximately between 5 mm and 0.5 m. When “read” at a speed of 120 km/h for example, these wavelengths produce an excitation between 6.7 kHz and 67 Hz. Note that wheel or rail defects such as flats or impacts from cracked ballast stones are not considered here.

The relative displacement imposed by the roughness is absorbed by rail and / or wheel, depending on the receptance of each component in the considered frequency range. Typical receptances of a standard ballasted track and a monobloc wheel are displayed in Figure 1. The double peaks that are visible in the wheel receptance curve are due to rotational effects.

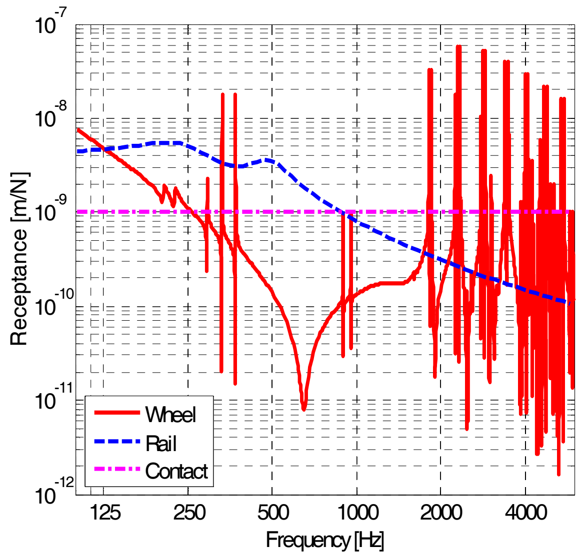

Figure 1 : Wheel (LK 900 bare wheel), rail and contact receptances (vertical direction)

Clearly, the behaviour of the rail (close to an infinite beam) is very different from that of a wheel (which has a modal response). Also, one distinguishes different frequency domains that are “dominated” by the rail or the wheel. The local deformation of wheel and rail is not taken into account by these receptances. Most often this effect is described by an additional contact receptance. It is inversely proportional to the contact stiffness, which depends on material properties and the local geometry involved (notably the contact patch dimension).

All this is suggesting that rail noise dominates at lower frequencies while wheel noise dominates at higher frequencies. This is globally true; however, the track contains other components than the rail that also radiate noise. Especially the sleeper contribution is generally important at low frequencies. Figure 2 gives an example of rail, wheel and sleeper powers calculated with TWINS for the same wheel-track combination as in Figure 1.

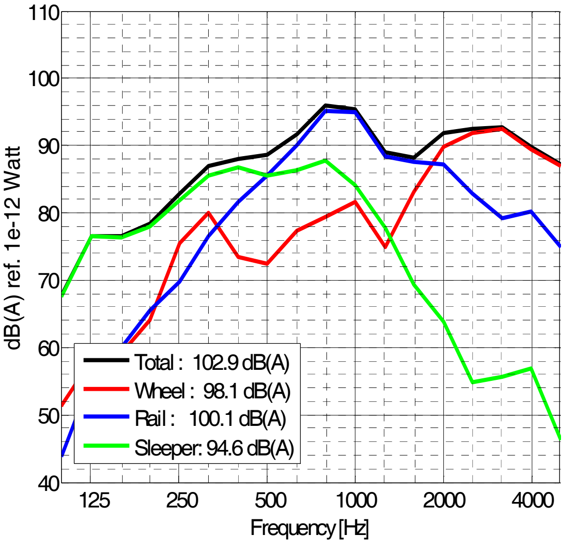

Figure 2: TWINS calculated sound powers for wheel, rail and sleeper @ 100 km/h and 8.5 to wheel load; bare LK 900 wheel

2.1 Mitigation measures

Rolling noise can be reduced by decreasing

- the excitation due to roughness,
- the rail response, and
- the wheel response.

Additionally, the propagation of sound can be restrained using screens, but this is not in the scope of this paper.

Indeed, a reduction of wheel roughness is one of the most effective mitigation measures. It can notably be achieved by replacing cast iron block brakes with disc brakes or composite block brakes. Beyond this measure, however, roughness is difficult to control. For further improvements, one has to act on rail and wheel responses.

Rail sound power notably depends on the effective length of rail that is vibrating, which in turn depends on the damping introduced by the support system. This damping is quantified by the so-called track decay rate in dB/m (a damping ratio is not appropriate because the energy loss caused by outgoing waves would also be taken into account then). The stiffer the rail fastening system is (notably the rail pads) the higher the track decay rate will be. On the other hand the sleeper contribution increases with stiffer rail pads. Also, softer rail pads are preferred by infrastructure owners because of their positive effect on the track system lifetime. For slab track, rail pads have to be quite soft in order to not destroy the concrete slab. In all situations where relatively soft rail pads are installed, rail dampers represent an effective way to increase the track decay rate. More information about the assessment of rail damper efficiencies can be found in a parallel paper [9].

In contrast to the rail, the wheel response is essentially modal in nature. The wheel modes that occur depend on the wheel geometry. A first step for the reduction of wheel noise is therefore the geometrical optimization of the wheel shape.

Similarly to the rail, wheel noise can also be reduced by the use of dynamic absorbers or dampers. Indeed, the initial damping of a free wheel seems as low as that of a church bell. When rolling on a track, however, the damping that is experienced by the wheel can be 10 times higher than the modal damping measured in free conditions. Therefore it is important to compare the additional damping introduced by any damping device with the so called "damping when rolling" and not with the damping of the free wheel.

Different types of wheel absorbers exist. Constrained-layer absorbers are mounted directly on the wheel web. Other designs include mass-spring systems that are tuned to particular frequencies. Using resilient materials with high damping coefficients, these absorbers also introduce some damping beyond their tuning frequency. A further evolution of this absorber type is the multi-layer damper involving sheets of rubber and steel of different stiffness and mass, leading to a broadband effect. Besides dispersion of energy in a resilient material, dry friction can add damping. This principle is exploited using friction rings that are clamped to the inner side of the wheel rim.

3 Wheels and absorbers tested within the STARDAMP project

Altogether, four different wheels have been used in the STARDAMP project. One of these wheels, termed LK 900, has especially been designed by GHH-VALDUNES for the project to be compatible with different kinds of absorbers. It is a 900 mm monobloc wheel with a straight but slightly inclined web. The results presented here have been obtained with this wheel.

Figure 3 : LK 900 wheel with GHH-VALDUNES VLN Ring absorber

Figure 3 shows the wheel with a GHH-VALDUNES VLN ring absorber and Figure 4 with a GHH-VALDUNES plate absorber. As mentioned before, the ring absorber adds damping to the wheel through dry friction. The plate absorber is made of an elastic layer between two metal sheets. Due to varying lengths of the different blades and the high damping introduced by shearing of the elastic layer, this "tuned absorber" is also effective over a wide frequency range.

Note, however, that neither of these absorbers has been tuned to the wheel. Indeed, the aim of the project is not to optimize wheel absorbers but to define characterisation methods.

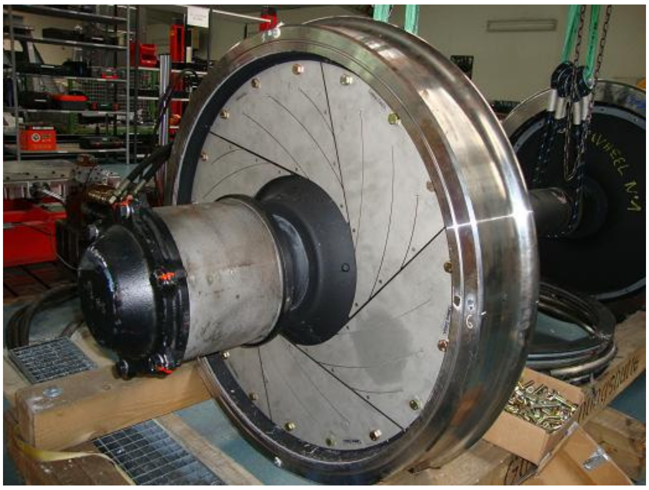

Figure 4 : LK 900 wheel with GHH-VALDUNES plate absorber

4 Wheel noise calculation procedure

4.1 Methodology for bare monobloc wheels

The presented methodology is based on TWINS calculations [7]. This software implements an analytical description of wheel-rail interaction, rail response, rail radiation and wheel radiation. The wheel response is not directly calculated in TWINS because the complex geometry of a railway wheels is unsuitable for an analytical description. The wheel modal basis is therefore calculated externally using a Finite Element (FE) model. Input to TWINS consists in a reduced number of modal displacements on wheel tread and web, largely sufficient for the calculation of wheel-rail interaction and wheel radiation. Modal damping varies very little from one bare wheel to another; default values (depending on the number of nodal diameters) are used. After definition of all track parameters (such as rail and sleeper type, rail pad stiffness and others), TWINS calculates the sound power radiated from each component (wheel rail and sleeper) for a unit roughness excitation. These results are finally weighted by a contact filter that takes into account given wheel and rail roughness spectra as well as the static wheel load and the speed of the train.

4.2 Methodology for wheels with damping devices

For wheels with damping devices, the procedure is very similar. Indeed, the FE results of the bare wheel can still be used because most absorbers have a small effect on mode shapes. Modal frequencies and damping coefficients are measured by means of an Experimental Modal Analysis (EMA). The TWINS input file is then updated with these data and calculations are performed as described above. This method generally gives far better results than a direct (FE) modelling of the wheel with mounted absorbers (because of the complexity of absorber modelling). However, the underlying hypothesis of unchanged mode shapes involves limitations; these are further discussed in section 6.

5 Results

As mentioned before, wheel dampers act through the additional modal damping applied to the wheel. The effect can be clearly seen on the wheel receptances. Figure 5 compares receptances of the bare wheel and receptances obtained with mounted ring or plate absorbers. The gain at the main receptance peaks is roughly between a factor 10 and 100. As the bare wheel receptance does not reflect the damping when rolling, however, this is not the gain that will be obtained for the wheel response. Indeed, Figure 6 shows that the gain on the vertical displacement of the wheel at the wheel-rail contact point reduces to a factor 3 to 10. Note that this displacement has been calculated with TWINS, taking into account the wheel-rail interaction. It cannot directly be measured in the laboratory.

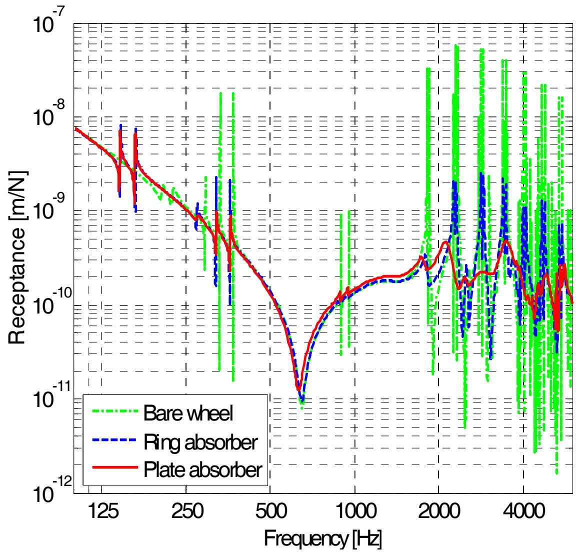

Figure 5 : Receptances of bare wheel, wheel with ring absorber and wheel with plate absorber

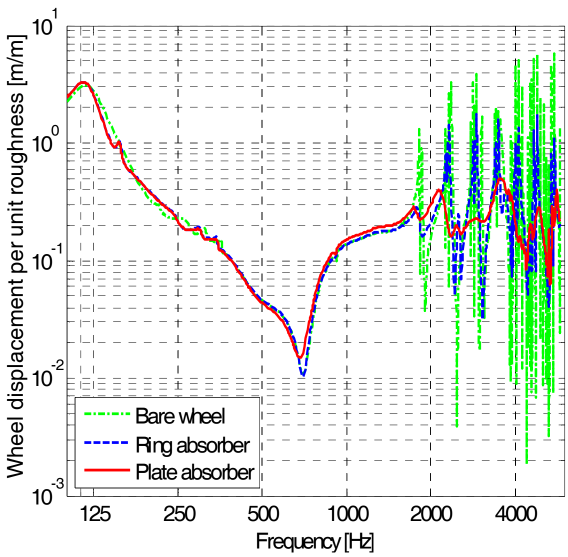

Figure 6 : Wheel response per unit roughness for bare wheel, wheel with ring absorber and wheel with plate absorber

Figure 7 and Figure 8 indicate the acoustic powers of wheel, rail and sleeper, obtained with the damped wheels (ring and plate absorber). These can be compared with Figure 2 that contains the results for the bare wheel. With 6.3 dB(A) for the ring absorber and 9.8 dB(A) for the plate absorber, the reduction in wheel power is close to what could be expected with regard to the calculated wheel displacements. Given the predominance of the rail over a wide frequency range, the overall noise reduction is much lower: 1.5 dB(A) and 1.7 dB(A) respectively. Note that the spectrum of the wheel power obtained with the plate absorber remains below the track power over the entire frequency range. The additional benefit of this absorber would therefore only be audible on a quieter track (for example with stiffer rail pads or rail absorbers).

These results clearly show that a total noise reduction produced by a given wheel absorber can only be given for one specific track.

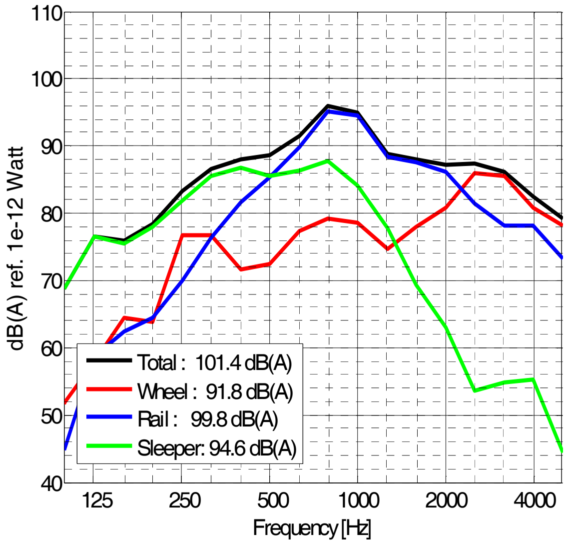

Figure 7 : TWINS calculated sound powers for wheel, rail and sleeper @ 100 km/h and 8.5 to wheel load; LK 900 wheel with ring absorber

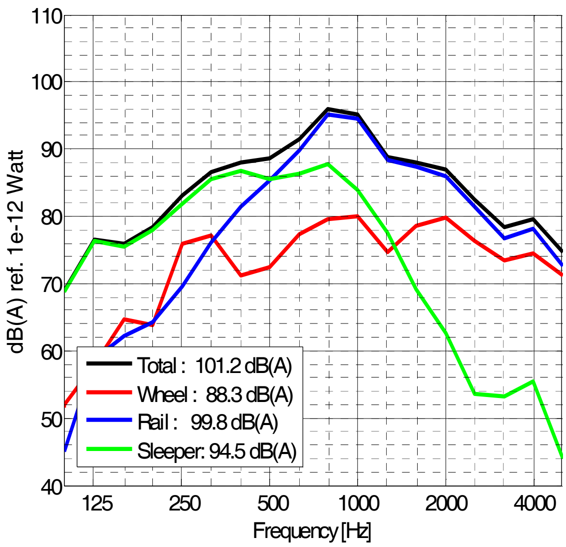

Figure 8 : TWINS calculated sound powers for wheel, rail and sleeper @ 100 km/h and 8.5 to wheel load; LK 900 wheel with plate absorber

Another parameter that highly influences the performance of any wheel absorber is the roughness of wheel and rail. The results presented above have been obtained by the use of TSI+ rail roughness [10] and a typical roughness spectrum measured on a disc braked wheel at a speed of 100 km/h. A typical tread braked wheel roughness spectrum for example is characterised by significantly higher levels in the mid-frequency range. The predominance of the rail will then be even clearer and the relative benefit of a wheel damper lower.

6 Comparison with other methods

The presented method for the assessment of wheel damper efficiencies is one of three techniques that are tested within STARDAMP.

The first alternative method is a direct measurement of rolling noise on a roller test bench. On such a test bench, the (circular) rail is fixed on a bigger driving wheel. The obvious advantage of this method is that wheel radiation is not calculated but directly measured. Also, the excitation of the wheel is the real rolling excitation, in contrast to any other laboratory method. On the other hand, one has to ensure that the radiation of the driving wheel does not pollute the measurements. Also, the dynamic behaviour of the rail is expected to be different from that of a real rail. Potential effects on the wheel response have to be controlled.

Another method, developed by GHH-VALDUNES, consists in measuring vibro-acoustic transfer functions between a force excitation of the wheel and its sound radiation in a reverberant room. As this excitation is not representative for rolling conditions, a post-processing with a model similar to TWINS is necessary. Again, wheel acoustic power is measured directly here, which represents a considerable advantage when assessing wheel absorbers that are liable to radiate sound themselves. This is in fact the case for the GHH-VALDUNES plate absorber[1].

The "Experimental Modal Analysis + TWINS" method described in this paper only accounts for the effect of added damping due to the absorber. Any modification of the wheel radiation (shielding and re-radiation) is not considered. This point remains to be controlled. A deeper comparison of results obtained with the three different methods is currently under study.

TWINS calculations with an experimentally updated wheel modal basis also reach their limits when a complex device is mounted on the web. Typical examples are web-mounted brake discs. Due to the bolted or screwed connection at discrete points, these wheels cannot be considered as axisymmetric anymore. The damping that is introduced by the friction between both parts can be measured experimentally, but brake discs also modify the mode shapes. For such cases TWINS can still be used but the method becomes less straightforward.

[1] Due to its large surface, the GHH-VALDUNES plate absorber radiates some noise itself. However, shielding and re-radiation are expected to be of the same order of magnitude. The cancelling of both effects has not been verified yet, but results from the SILENCE project [4] support this hypothesis.

7 Conclusion

The STARDAMP project aims at qualifying methods that permit the assessment of wheel and rail damper efficiencies. For wheel dampers, the "Experimental Modal Analysis + TWINS" method is a promising candidate that is applicable for all devices that mainly modify the modal damping of the wheel without affecting too much mode shapes or radiation. In these cases, the method has the big advantage of calculating wheel and rail power separately, which is important for a correct prediction of total noise reduction. Wheel and rail roughness are part of the input data and therefore numerically controlled. This is very important because any sound power reduction that is obtained with an unknown roughness spectrum is of little significance.

The comparison with alternative laboratory methods is currently in progress. These are especially interesting for the assessment of the plate absorber.

Acknowledgments

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References

[1] B. Hemsworth, "Silent freight project: final report". *ERRI* (2000)

[2] B. Hemsworth, "Silent track project: final report", *ERRI* (2000)

[3] STAIRRS (Strategies and Tools to Assess and Implement noise Reducing measures for Railway Systems) final report, available at http://www.uic.org/IMG/pdf/rapport_final_stairrs.pdf, *ERRI* (2003)

[4] SILENCE project (2008), see documentation on http://www.silence-ip.org/site/

[5] Directive 2002/49/EC of the European parliament and of the Council of 25 June 2002 relating to the assessment and management of environmental noise. *Official Journal of the European Communities*, 18.7.2002

[6] M. Starnberg, "STARDAMP: Ein deutsch-französisches Kooperationsprojekt zur Verminderung des Schienenlärms", DAGA 2012

[7] D.J. Thompson, B. Hemsworth, N. Vincent, "Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method". *Journal of Sound and Vibration* 193(1), 123-135, 1996.

[8] D. Thompson, "Railway Noise and Vibration", *Elsevier* (2009)

[9] M. Toward and D.J. Thompson, "Laboratory methods for testing the performance of acoustic rail dampers", ACOUSTICS 2012

[10] Commission Decision of 23 December 2005 concerning the technical specification for interoperability (« TSI ») relating to the subsystem 'rolling stock', *Official Journal of the European Union*, 18.2.2006