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Contract Number: IST-2000-28419

Harmonised **A**ccurate and **R**eliable **M**ethods for the EU Directive on the Assessment and Management **O**f Environmental **Noise**

DEFINITION OF OTHER SOURCES AND INFLUENCE OF SOURCE MEASURES

***Deliverable 12 part 3* of the HARMONOISE project**

Project Co-ordinator: AEA TECHNOLOGY RAIL BV

Partners:	AEA			
	TNO	DLR	LCPC	VTI
	SP	M+P	Autostrade	TUG
	DGMR	TRL	Arpat	JRC-EI
	DeBakom	SNCF	Delta	
	DB	CSTB	Kilde	

Document identity: (reference file name) HAR12TR-030625-AEA11.doc

Date: (date of issue) 29 January 2004

Level of confidentiality: Consortium

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Project funded by the EC under the Information Society and Technology (IST) Programme

Amendments

Version number	Amendment details	Date (YY-MM-DD)
1		2003-06-25
2		2003-12-02
3		2003-12-10
4		2004-2-2

Approval by the Project Steering Committee

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SUMMARY

The aim of Harmonoise WP 1.2 is to develop a source model for the description of railway noise. Up to now, sources such as rolling noise, traction noise and aerodynamic noise have been considered in detail. The results can be found in different technical reports and deliverables [Harmonoise D11 to D13]. But the description of effects as railway bridges, braking noise, curve squeal, rail and wheel dampers, wheel shields, small barriers and impact noise, which have also a considerable influence on the railway noise, have been excluded in our considerations up to now. For most of these effects, a detailed model is not appropriate for the harmonoise model or available. However, simple models for most of the effects have been summarized in the state-of-the-art report D10 or can be found in some of the national calculation schemes. In this report, models for each of the effects mentioned above are reviewed. In most cases, correction terms to the rolling noise spectra will be introduced to model the effects. Corrections for radiation characteristics have to be included for railway bridges, wheel shields and small barriers. Different projects perform investigations to develop better models and the simple solutions mentioned here should be replaced as soon as better solutions are available. Deliverable D13 part 2 *Practical datacollection for the Harmonoise source model: measurement guidelines & analysis*, contains measurement protocols for the “other sources” as far as available.

The various sources and noise control measures described previously are described jointly in a practical calculation model which can be regarded as an extension to the harmonoise rolling noise model. Although theoretical models are available for some of the sources, they are generally not appropriate for the harmonoise source description and are therefore in this report given as correction spectra to the rolling noise.

Finally this report gives a table with an overview of the frequency range and dynamic range, measurement protocol and data for each source. Also the relevant source heights for the various source terms are included.

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1 Introduction

The aim of Harmonoise WP 1.2 is to develop a source model for the description of railway noise. Up to now, sources such as rolling noise, traction noise and aerodynamic noise have been considered in detail. The results can be found in different technical reports and deliverables [1,2,3,4]. But the description of effects as railway bridges, braking noise, curve squeal, rail and wheel dampers, wheel shields, small barriers and impact noise which have also a considerable influence on the railway noise have been excluded in our considerations up to now. For most of these effects, a detailed model is not appropriate for the harmonoise model or available. However, simple models for most of the effects have been summarized in the Harmonoise state-of-the-art report [5] or can be found in some of the national calculation schemes. In this report, models for each of the effects mentioned above are reviewed. In most cases, correction terms to the rolling noise spectra will be introduced to model the effects. Corrections for radiation characteristics have to be included for railway bridges, wheel shields and small barriers. But we should keep in mind that different projects perform investigations to develop better models and that the simple solutions mentioned here should be replaced as soon as better solutions are available. The report [6] contains measurement protocols for the “other sources” as far as available.

[1] *D12 part 1, Definition of track influence: roughness in rolling noise*, HAR12TR-020813-AEA10, E. Verheijen and A. van Beek 2003.
[2] *D12 part 2, Definition of track influence, track composition and rolling noise*, HAR12TR-030403-AEA10, M. Beuving 2003.
[3] *WP 1.2 Rail sources: Modelling of railway traction noise for input to rail traffic noise models,* HAR12-030710-Tno01, M. Dittrich 2003
[4] *WP 1.2 Aerodynamic noise*, HAR12TR-030828-SNCF01, C. Talotte, SNCF
[5] *WP 1.2. State of the art, HAR12TR-020118-SNCF10*, A. van Beek e.a.2002
[6] WP 1.2, D13 part 2, *Practical datacollection for the Harmonoise source model: measurement guidelines & analysis,* Paul van der Stap e.a. 2004.

2 Other sources than rolling noise

2.1 Railway bridges (DB/AEAT)

Depending on the bridge type, the sound level emitted from a train passing the bridge can be increased and the maximum of the sound-emission spectrum can be shifted to lower frequencies compared to the sound level measured at an open track. Generated by excitations of bridge vibrations, these effects are mainly determined by two factors, the transmission of vibrations from the train to the bridge structure and the frequencies of the natural bridge oscillations. Both factors strongly depend on details of the bridge construction and the track structure used. Because bridges are build individually for each case, source models for bridge noise have to be considered as rough estimations. But in the Harmonoise state-of-the-art report, a categorization of bridges has been proposed depending on the bridge type and the track type:

Bridge type	Track type
Steel bridge	Without ballast, without vibration isolation
	With ballast, without vibration isolation
	With ballast, with vibration isolation
Composite construction bridge	With ballast bed, without vibration isolation
	With ballast, with vibration isolation
Solid reinforced concrete bridge	With ballast bed, without vibration isolation
	With ballast, with vibration isolation
	With rigid-slab track

In the state-of-the-art report, also values for the sound-level increase for different types of bridges and tracks have been proposed. But we have to keep in mind that the bridge noise may vary significantly. Figure 1 shows the sound-pressure level measured in 25 m distance of different steel bridges without ballast during the pass-by of trains. Variations of up to ± 5 dB(A) are possible depending predominantly on construction details.

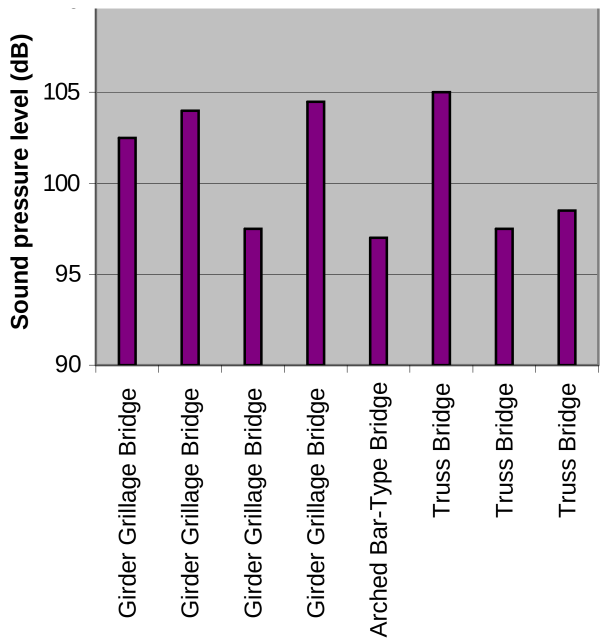


Figure 1: Sound pressure levels of different steel bridges without ballast measured in a distance of 25 m during the pass-by of trains with a velocity of 120 km/h. Bridges of the same type differ in its geometrical parameters.

Although it is well known that the main part of the sound is radiated by the bridge structure, we propose to model bridge noise by a correction factor to the rolling noise. The additional source has to be assumed over the whole length of the bridge at the equivalent source height of 0 m. For the description of the bridge noise, the sound power level has to be given as third-octave spectrum. To obtain data for different bridge categories, measurements have to be performed beside the bridge and beside the open track. If the track conditions are comparable (e.g. the same track and the same roughness category), the correction term is given by the difference of the measured sound power levels.

Concerning the vertical radiation characteristics of bridges, the construction details have a large influence. If we consider a bridge consisting of a deck and side parts, the lateral and the downward noise radiation should be equal. In this case we should prefer monopole characteristics. If we consider a concrete bridge without side parts, the downward radiation from the deck should be even higher than the lateral radiation of the bridge. Concerning the lateral radiation characteristics, no accurate information is available.

2.2 Braking noise (AEAT)

Definition: The *excess* noise produced by a certain train at a certain speed, due to the activated braking system.

2.2.1 Physical phenomena

Two cases of physical phenomena responsible of braking noise can be identified depending on the braking system (disk brake or block brakes).

Disk brake:

Brake screech occurs from a coupling of the eigenmodes of brake disk and brake mechanism during the braking process with increasing force applied on the disk.

This phenomenon appears in high frequencies (at around 10 kHz). The energy is concentrated on several picks which could emerge from 20 to 25 dB. The braking noise does not seem directly linked with the braking intensity. We should notice here that these results have been obtained in a station and then include maybe a part of amplification due to the building.

Theoretical and experimental investigations and development of modelling tools are still in progress [1]. Further work has to concentrate on a better theoretical understanding of the generating mechanism and on the development of constructive measures (in particular acoustically optimised brake pads) in order to reduce disk brake screech.

Block brakes

For block brakes, the main difficulty of the problem comes from the fact that the block rubs on the running surface of the wheel influences the roughness generation and then the rolling noise. Two sorts of contact have to be taken in consideration: the contact patch between rail and wheel and the contact between block brake and wheel. Break screech for tread brakes with composite brake blocks does not occur with the same high intensity as for cast iron blocks. This is another positive effect of the so-called “K-brakes” besides the well-known fact that they considerably reduce rolling noise. Prediction methods and standardised measurement methods for screech determination and design solutions still need to be developed.

[1] Q.S. NGUYEN, A. OUESLATI, X. LORANG, F. MOIROT: "Le crissement des freins: un problème d'instabilité dynamique", Colloque SFA Bruit et Vibrations dans le domaine ferroviaire, dec 2003 (in french)

2.2.2 Measurement protocol proposed in the Dutch calculation scheme:

The Dutch calculation scheme gives a short description of the method to assess braking noise.
The total noise (rolling+traction+braking) is measured at 7.5 m from the track (1.2 m height), during pass-by of train at a speed *V* of 25 km/h, 50 km/h, 100 km/h and maximum speed.
In case the noise level at such a speed does not exceed the noise level of this train at constant speed (with deactivated braking system), the braking speed is neglected.
Using the least squares method, the braking noise can be determined as a function of log(*V*) for each one-third octave band. Braking noise is assigned to a source line at axle height (0.5 m).

2.3 Curve squeal (AEAT/DB)

Definition: The *excess* high-frequency noise produced by train and track when a train runs through a curve (or switch).

The occurrence of curve squeal noise is dependent on many parameters: weather, curvature, track transverse profile, speed, bogie design.

Curve squeal means the additional noise which can occur when a vehicle passes tight curves. A description of the physical effects can be found in the Harmonoise state-of-the-art report [1]. The typical nature of the sound of curve squeal, a screeching sound, causes problems when using an average A-weighted correction factor. The usual relationship between the sound level, to which citizens are exposed, and annoyance does not hold for highly tonal sounds.
Because curve squeal includes also pure tones and its harmonics, the noise is rather disturbing and should be considered by the final calculation scheme. But an accurate prediction of the curve squeal noise as a function of vehicle and track parameters is still not possible today due to the statistical nature of the effect. Nevertheless, a very simplified approach can be found in the German “Schall 03” . This calculation scheme assumes a correction term of 3 dB(A) for curves with radii less than 500 m and a correction term of 8 dB(A) for curves with radii less than 300 m. But it has to be considered that this correction term is an averaged value and therefore the error can be immense when comparing predicted values with single measurements. A frequency-dependency of the correction term has not been identified yet but it is known that the tonal contributions of the curve squeal are predominately in the frequency range from 1 to 5 kHz. The terms include contributions of the wheel and the rail and should be used as a correction of the rolling noise. The Report [3] and the paper [4] contain a measurement protocol for curve-squeal.

[1] *WP 1.2. State of the art, HAR12TR-020118-SNCF10*, A. van Beek e.o.2002
[2] Friedrich Krüger, *Statistische Erfassung von Kurvenquietschen bei Nahverkehrsbahnen,* Zeitschrift für Lärmbekämpfung 45(6), Nov. 1998.
[3] *WP 1.2*, D13 part 2, *Practical datacollection for the Harmonoise source model: measurement guidelines & analysis,* Paul van der Stap e.o. 2004.
[4] E. Verheijen e.a., *A measurement protocol for curve squeal noise*, paper presented at Internoise 2000, Nice.

2.4 Impact noise (AEAT)

Impact noise is a special case of railway noise that occurs in those conditions where a noise caused by an impulsive increase of wheel rail contact force is produced. In the past, not so many studies where done on this kind of noise, and it is still difficult to predict where can occur. Two different reasons can cause impact noise: a discontinuity in the rail or a wheel flat.

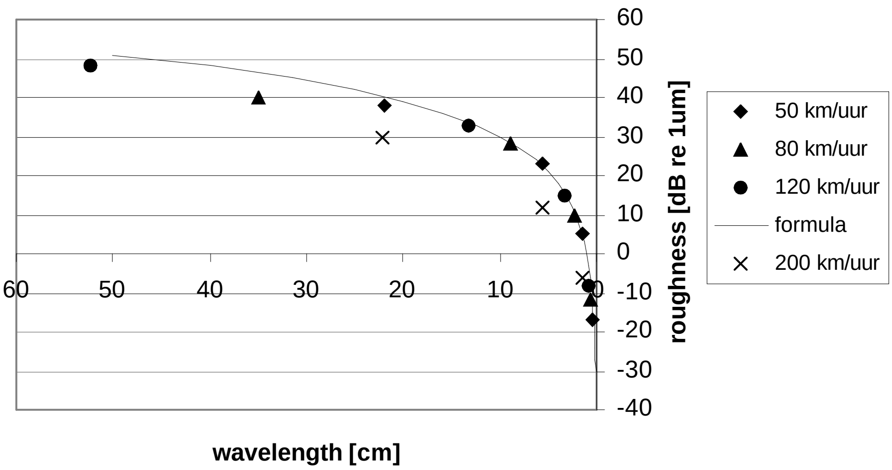
2.4.1 Rail joints

Discontinuities in the rail are present on rail joints, on switches and sometimes even on welded rails, if a sharp difference in the vertical height of rail occurs. This obliges the wheel to accelerate in the positive or negative vertical direction, inducing inertial loads. Unfortunately, it is not possible to know a priori which are distance and difference in vertical level between two adjacent rails unless they are not measured. These two quantities depend on the type of rail joint, on temperature, on rail wear, etc. and the difference in Lp(A) (averaged over 0.125 s) could be great. As an example, the difference between a 0 mm and a 2 mm step-up joint for a lightly dipped rail could reach 8 dB [1]. In general it could be seen that an increase in sound level is expected both in case of a step-up or a step-down joint: speed relationship (V) is approximately $20*\log_{10}(V)$ for a step-up, but much less for step-down joint. When the dip become larger than 10 mm, this dominates the impact noise mechanism, and the same effect is seen both on step-up and step–down joints [1].

A model based on an equivalent railroughness for rail joints is presented in the next paragraph.

The effect of rail joints on rolling noise is extensively studied by modeling in [1]. Wheel /rail impacts due to rail joints are simulated in the time domain. The impact forces are transformed into the frequency domain and converted into the form of an equivalent roughness spectrum. These

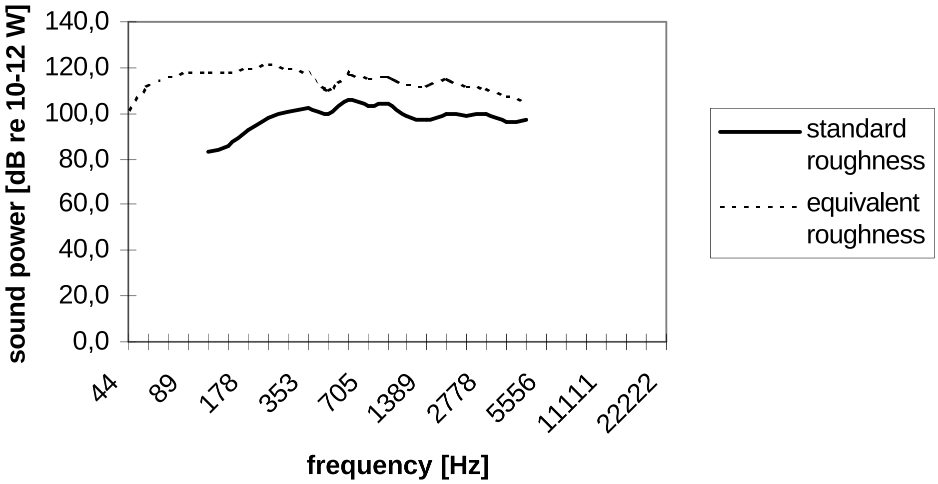
equivalent roughness spectra are presented for various speeds. An example is given as illustration of the theory.
In this paragraph, the results for one joint geometry are evaluated. It is tested if the equivalent roughness spectra in the frequency domain for various speeds can be transformed into a general consistent roughness spectrum in wavelengths. This is done for a passing over a rail joint with 7 mm gap and 1 mm step-up with a 5 mm dip (the upper spectrum in fig 12. The spectra for 50, 80, 120 and 200 km/h are translated to the wavelength domain.
From this it can be seen that the 200 km/h spectrum is not in line with the other results, probably because other mechanisms like greater loss of contact occur. But for 120 km/h and lower speeds it is shown that this equivalent roughness spectrum can be described as a function of the wavelength, R[dB re 1 um] = 30 log λ[cm]. The result is shown in the next figure.



The presented equivalent roughness spectra can be regarded as the total roughness in the Harmonoise rolling noise model. This is the energetic sum of wheel and rail roughness:
$R_{total} = (R_{wheel} \oplus R_{rail}) + C_f$

If the equivalent roughness level is much higher than wheel roughness, this total roughness level can be translated to an equivalent rail roughness level, so that the excitation by rail joints can be described as a rail condition.
The correction for the contact filter in a normal situation in rolling noise has probably other significance in this situation. However, ignoring this issue, it is possible to find a value for Rrail, using a standard value for the contact filter for rolling noise. In this example it was found that the wheel roughness of a cast iron braked wheel can be ignored to translate this equivalent total roughness spectrum to an equivalent rail roughness spectrum, using a standard contact filter.

In the next figure a comparison of this equivalent rail roughness spectrum is made with a calculation for standard condition without rail joints.
This is shown in the next figure.



The equivalent roughness levels given in [1] are representative for the impact forces that occur in 125 ms when a wheel is passing a rail joint.
For the given different speed this is representative for a distant from 2 up to 4 meters (120 km/h).

Conclusions from this example:
The speed dependency of the excitation of rail joints can be described as an “equivalent rail roughness” in as a function of wavelength for speeds up to 120 km/h and be fitted in the harmonoise rolling noise model by this way.
However, as the excitation is strongly dependent on joint geometry, and static wheel load, the “equivalent rail roughness” should be described for these different situations.
The equivalent roughness level is valid for several meters of track around the joint. This should be taken into account when constructing a representative roughness level for a jointed track with regularly distributed joints.

[1] T.X. Wu and D.J.Thompson, *On the impact noise generation due to wheel passing over rail joints, ISVR, Southampton*, June 2001.
[2] T.X. Wu and D.J.Thompson, *A hybrid model for wheel/track dynamic interaction and noise generation due to wheel flats*, ISVR, Southampton, January 2001.

2.4.2 Wheel flats

Wheel flats cause a relative leak of contact between rail and wheel, so that the wheel, under its own weight and wagonload, accelerates in the vertical direction. The effect is a great difference in the contact force between the wheel and the rail, which excites an excessive vibration in the wheel and in the rail, causing a peak in the noise level. At the present, a mechanism similar to that for rail joints could be assumed [2]. Wheel noise peak is periodic (period is of course wheel rotation) and very short in time, and it can be measured only as a difference in Lp between a normal contact condition and the impact situation, while using a fast averaging time (0.125 s). A single wheel may sometimes contain several flats, so that special attention should be given while trying to attempt which is the difference between typical rolling noise and impact noise. Wheel flats show an increase in noise (for that wheel) with a speed dependence of $20*\log_{10}(V)$ [2]. Wheel flats deeper than 2 mm are the main cause of wheel noise up to 200 km/h.

In Norway are indications that a flat wheel may occur on a few percent of Norwegian suburban trains, and that the SEL-levels for the passage of such trains can increase up to approx. 6dB(A). The similar effect of a single flat on a long freight train is less noticeable.

3 Special track design

3.1 Slab track (AEAT)

Slab track is the general name for a track superstructure without ballast. Although slab track is regarded as a standard application for metro and tram systems, the number of *railway* slab tracks is still small, but it is growing. Especially on new (high-speed) lines ballastless tracks are applied.

Various slab track designs have been explored in the past two decades, and are still being explored. Some designs are based on concrete biblock sleepers, laid on a reinforcement grid, that are afterwards concreted (e.g. Rheda 2000 in Germany [1]). Other designs use subsequent concrete slabs of a few meters length (e.g. Gemona in Italy). In both these types of design, the rail is supported at discrete positions by fasteners, like with ballasted track. Also continuously supported track has been applied, e.g. Best in the Netherlands. Here, the rails are laid in a gutter in the continuous slab, and are embedded in cork rubber.

From the viewpoint of noise emission, the various slab track designs differ from ballasted track mainly with respect to the following:

- The concrete slab (with or without embedded sleepers) is much stiffer than the ballast bed with sleepers. Usually the higher stiffness of slab tracks is (partly) compensated by the application of smoother rail pads. This will generally lead to increasing rail vibrations compared to ballasted track.
- The coefficient of absorption of the concrete surface of the slab is much lower than that of ballast. For the purpose of compensating the increase of noise due to more reflective energy, sometimes-absorptive material is applied on the surface of the slab.

The net effect on noise emission is dependent on the design characteristics of the slab track. Generally, a few dB higher levels are found than for ballasted tracks with concrete or wooden sleepers. E.g. the German calculation scheme *Schall 03* assumes an increase of noise emission of 3 dB(A) for trains on a slab track (without absorptive material) relative to trains on track with concrete sleepers.

Generally speaking, the track transfer function of slab track will differ from ballasted track. Because of the variation between different designs of slab tracks, it will be necessary either to measure or to model the specific slab construction. The same measurement protocol as for ballasted track can be used for this purpose.

[1] D. Briginshaw, *Latest Design Of Rheda Slab Track Installed*, International Rail Journal, November 2001.

4 Source measures

4.1 Rail dampers (AEAT)

Rail dampers are vibration absorbers clamped to rails. The principle of operation is that vibrations in the rails are attenuated by the damper. As the effective vibration length (per wheel/rail contact) is reduced, the track decay rate will raise. This parameter is the main parameter affected by the damper in modeling tools like TWINS. The track decay rate can be measured using an excitation hammer and accelerometers at different distances from the excitation position. Vertical as well as lateral decay rates are of interest.

The effect of a noise damper on the noise level can be measured directly using two cross sections on the same track, the first with dampers, the second without dampers. The SPL difference between both sections should be determined. An average can be calculated per type of train in a certain speed range.

The damper will generally not influence the noise emitted by the vehicle. The reduction of rail vibrations is not “felt” by the wheel through the contact patch. Only the rail and sleeper vibration levels are affected.

The track transfer function, defined from roughness excitation to noise, will change by application of a rail damper.

The effect of a rail damper **cannot** be simply assessed in a laboratory arrangement with a short piece of rail. The standard damping by the fasteners is essential, reflections should be avoided, and the vehicle noise is absent.

4.2 Wheel dampers (AEAT)

The wheel damper acts on the wheel vibrations and wheel radiation.

The effect can be measured on a test train of which 2 or 3 vehicles are equipped with wheel dampers.

The effect of a wheel damper **cannot** be simply assessed in a laboratory arrangement with a single free wheel. The standard damping by the axle load is essential, and the track noise is absent.

5 Calculation model for other sources and measures

The various sources and noise control measures described previously can be described jointly in a practical calculation model which can be regarded as an extension to the harmonoise rolling noise model. Although theoretical models are available for some of the sources, they are generally not appropriate for the harmonoise source description and are therefore given as correction spectra to the rolling noise. The pass-by equivalent sound pressure level by vehicle length can be described with:

$$L_{p,eq}(f, h=0m) = L_{r,eff}(f) \oplus L_{r,eff,impact}(f) + C(f) + L_{Htr}(f) + 10\log(N/L)$$

$$L_{p,eq}(f, h=0.5m) = L_{r,eff}(f) \oplus L_{r,eff,impact}(f) + C(f) + L_{Hveh}(f) + 10\log(N/L) + \Delta L_{p,squeal}(f) + \Delta L_{p,\,brake}(f,v)$$

Where:
$L_{p,eq}(f,h)$ = equivalent sound pressure spectrum at source height h during pass by at 7,5m
$L_{r,eff}(f)$ = combined effective roughness frequency spectrum (wheel+rail) at given speed
$L_{r,eff,impact}(f)$ = effective equivalent roughness due to impact
$\oplus$ = operator symbol for energy summation
C(f) = contact filter spectrum
$L_{Htr}(f)$ = transfer function from effective roughness to sound pressure, per axle, per meter, for the track
$L_{Hveh}(f)$ = transfer function from effective roughness to sound pressure, per axle, per meter, for the vehicle
N/L = number of axles per meter
$\Delta L_{p,squeal}(f)$ = correction spectrum for squeal noise, can be different for gradual or sharp curve (i.e. as in points)
$\Delta L_{p,brake}(f,v)$ = correction spectrum for braking noise at given speed v, will depend on brake type (block, disc or other) and on speed range: squeal at lower speeds, friction noise at higher speeds for block brakes.
(all as 1/3-octave spectra).

The following table gives an overview of the frequency range and dynamic range, measurement protocol and data for each source. Also the relevant source heights for the various source terms are included.

Correction spectra for braking and squeal noise typically are allocated to the source height of the wheel, whereas the other sources are allocated to both the wheel and track source heights.

The correction terms for curve squeal and braking noise will depend on rolling stock type. The equivalent roughness term for impact excitation is somewhat speed dependent and also strongly dependent on the width and type of rail joint or wheel flat. For each source table 5.1 gives the term on which the correction will work.

Other source	**Indicative Frequency range**	**Indicative Dynamic range**	**Measurement protocol dB(A)**	**Measurement protocol spectral**	**Correction in:**	**Source height:**
bridges	< 2 kHz	0 – 15 dB	Protocol bridges	Protocol bridges	$L_{Htr}(f)$	0m
Curve squeal	700 Hz –4 kHz	0 - 30 dB	Curve squeal	-	$\Delta L_{p,\ squeal}(f)$	0.5m
Braking noise	> 10 kHz	Not sufficient knowledge	Braking noise protocol	Braking noise protocol	$\Delta\ L_{p,\ brake}\ (f,v)$	0.5m
Impact noise: flat wheels	200 Hz – 6 kHz	0 - 15 dB	-	-	$L_{r,eff,impact}(f)$	0m, 0.5m
Impact noise: rail joints	200 Hz – 6 kHz	0 - 15 dB	Protocol impact noise	-	$L_{r,eff,impact}(f)$	0m, 0.5m
Slab track	-	+2-3dB(A)	Standard track protocol HN*	Standard track protocol HN*	$L_{Htr}(f)$	0m
Rail dampers	< 2 kHz	0 – 6 dB(A)	Standard track protocol HN*	Standard track protocol HN*	$L_{Htr}(f)$	0m
Wheel dampers	2 – 8 kHz	0 - 3 dB(A)	Standard Rolling noise protocol with test train and reference train	Standard Rolling noise protocol with test train and reference train	$L_{Hveh}(f)$	0.5m

Table 5.1. Overview available default values and measurement protocols for other sources. Indicative frequency and dynamic ranges are dependent on individual situations.

* Results on the measures track in comparison with reference track.

If available, the default values will be given in the report D13 part 1 (Harmonoise database) and a measurement protocol will be given in D13 part 2 (Harmonoise measurement protocol).

6 Conclusions

A practical model has been provided for the prediction of various sources and noise control measures which can occur in combination with rolling noise. Correction terms are used for curve squeal, braking noise, impact noise and for various types of track variants such as bridges, slab tracks, rail dampers and wheel variants such as wheel dampers. These terms are either integrated in the transfer function or are additions to the roughness level or the sound pressure level.

Default values can be provided for these sources and measurement methods are to be provided which allow the collection of relevant data for each type of source or noise control measure. Typical frequency ranges and dynamic ranges of such correction terms have been indicated.