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A. BRACCIALI - M. PIPPERT - S. CERVELLO

Railway noise: the contribution of wheels

BASICS, THE LEGAL FRAME, LUCCHINI RS PRODUCTS

LUCCHINI RS

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FOREWORD

The quest for silent products and processes derives directly from industrialization. Craftsmanship was somewhat quieter, with the exception of some special activities, like forging, normally carried out with the help of the energy available from nature, typically in the form of water falling from a given height.

The XIX Century, with the development of steam energy and the XX Century, with the generalized availability of electricity, made available, at constantly decreasing difficulty and cost, a large amount of energy that reduced the need for workforce and introduced machinery of all sizes and, unfortunately, extremely high sound power emission.

One of the forms of noise generation that all of us are exposed to, even if not working in noisy workshops comes from means of transport.

It is easy to classify means of transport into classes: those travelling by sea (ships, vessels), those flying (aeroplanes), and those travelling over land (cars, trucks, buses, motorbikes, etc.).

Noise from ships is relevant only to areas near harbours and is therefore considered to be somewhat less important than other types of noise disturbances from vehicles.

Airport noise is certainly one of the most important issues of modern times, when the passage from land-based vehicles to air-based vehicles was stimulated by the development of jet and turboprop engines that almost completely replaced the piston engine. Unfortunately jets are much noisier, and many efforts have been made in the last few decades to reduce the specific fuel consumption (i.e. the efficiency), the reliability and, last but not least, the noise emitted by jets. High dilution multi-shaft gas turbines with fans of generous dimensions often declared the fortune of aeroplanes in highly sensitive contexts. Landing and departing procedures were often developed to spread out and to homogenise aircraft noise over the widest possible area in order to reduce the exposure of particular areas close to runways and approach paths.

Car noise is difficult to be tackled. It depends on a large number of parameters, including the condition of the road and tyre surface, the degree of maintenance of the vehicles and the vehicle speed. What makes road noise almost impossible to be carefully planned --- an activity that, as we have seen, is one of the most important for technicians dealing with aircraft noise pollution --- is that noise generation due to vehicles pass-by can only be estimated on a statistical basis. Private traffic is unscheduled by definition, while public transportation follows official timetables. It would prove extremely unpleasant and unpopular to close a road or a motorway as noise levels are exceeded, and certainly politicians try to avoid this extreme measure as much as possible.

Railway noise, the object of this publication, is a somewhat special type of noise. As for cars, it largely depends on speed; as for aircrafts, in many cases different routes can be used for freight trains that, as we will see, are responsible for the majority of the noise emitted; as for ships, trains pass through rather unpopulated areas. People normally love modern trains in most regions of the world: if they are pulled by electric locomotives (or made of EMUs) noise is the only form of pollution directly affecting the receivers. There are some well-known and acknowledged experiences that confirm that railway noise is particularly tolerated by human beings, contrarily to what happens with continuous road noise even with lower equivalent level.

One of the more frequent objections raised by infrastructure owners and train operating companies when dealing with complaints from citizens is that "railways were already there when you were born". If this is true in many cases, there is, however, a greater expectancy of quality of life than probably one hundred years ago. Similarly to what happened with chimneys and railway depots with dozens of steam locomotive smoking away, nowadays pollution is not considered a sign of progress anymore: our tolerance to harmful substances has (apparently) decreased. Noise from trains will follow the same trend in the next years: fewer and fewer people will want to live in a very noisy environment due to train pass-by, shunting or whatever. As long as we believe that railway is the most environmentally friendly means of transportation, and under the hypothesis that we all should sustain collective transportation in competition with private transportation, we must be ready to recognize that resources must be spent to minimize the impact of continuously growing railway traffic by acting on the noise sources rather than shutting down lines and reducing the service.

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Matthias Pippert
Steven Cervello

ACKNOWLEDGMENTS

Wheelsets noise reduction in Lucchini has a long history. Since the early '90s, the company invested in this field by searching for skilled human resources and the most advanced equipment. Today the LucchiniRS R&D department is still deeply involved in many internal and European projects on railway noise. As an outcome of these activities, this book is published today mainly as the result of the will of Dr. Giuseppe Lucchini, President of Lucchini RS, who decided in 2007 to start this series of technical books on railway products.

We are linked to the company in different ways - academic consultants, R&D personnel, experts on legal and standardization issues. An effort like this has been possible only thank to all the people who helped us constantly with their support and criticism throughout the entire publication process.
Especially, we are grateful to Mr. Roberto Forcella, Commercial and Marketing director and Mrs. Gabriella Giampà, Marketing and Communication responsible, both from Lucchini RS, who coordinated and followed the whole work. Special thanks also to Mrs. Renata Besola who looked after the editorial project and Mrs. Jane Wilks who revised the manuscript ensuring the linguistic appropriateness.

Looking forward for a quieter world,
Andrea Bracciali, Matthias Pippert, Steven Cervello
September 2009

Railway Noise: the Contribution of Wheels
BASICS, THE LEGAL FRAME, LUCCHINI RS PRODUCTS

PART 1:
Railway noise generation & control mechanisms

Andrea Bracciali

Part 1: Railway noise generation & control mechanisms

Andrea Bracciali

1 INTRODUCTION TO ACOUSTICS

1.1 Atmosphere, pressure, sound and levels

We are all immersed in the atmosphere. Environment pressure is given by the weight of the air column above our heads. As, by definition, pressure is the ratio of a force over the area on which it acts, it was measured in the past in what we can call nowadays "strange" units of measure. *Imperial Units* were used to (and in many cases are still used) define pressure as the ratio of the force of one pound acting on a square inch (psi); also very common in the rest of the world was the use of the kilogram as a force and the use of a square centimetre as the surface on which this kilogram was acting.
As a matter of fact, atmospheric pressure was mostly measured referring back to Torricelli's barometer comprising of a glass pipe and a column of mercury; that's why many people may remember that when they were children pressure was measured in mmHg, a strange unit measuring the height of the mercury column in a closed pipe.
With the introduction of the *International System of Units* (SI), force is a derived quantity (derived from Newton's law of dynamics) and is expressed in N (Newton); the area is obviously expressed in square meters and the atmospheric pressure is therefore measured in $Pa=N/m^2$. The standard atmospheric pressure value is defined as 101325 Pa (=1013 hPa =0.1013 MPa). Incidentally, this value is quite close to 1 kg_f/cm^2, that was in fact defined as an "atmosphere". People who love old instruments know this as 760 mmHg.

Atmospheric pressure varies --- bad weather is associated to low pressure and good weather to high pressure. Nevertheless, atmospheric pressure changes are rather low and can be felt better by instruments than by people. It is only when climbing or descending very rapidly that our ears recognize another important fact --- i.e. atmospheric pressure changes with altitude.
Our ears are a complex system designed by evolution not only to respond to sudden average changes of atmospheric pressure (and after all lifts and aeroplanes are a rather recent invention) but to catch even the smallest instantaneous variation of atmospheric pressure.

These instantaneous changes have no direct effects on the environment: they don't produce wind, storms or other perceivable environmental effects, but can be felt and recognized by our ears and our brain: that's sound.

Sound can be defined, therefore, as any variation of the atmospheric pressure that can be felt and recognized by the human ear. As any transducer, also the human ear has limits in the amplitude and frequency of detectable atmospheric pressure variations: some of them will be perceived as extremely disturbing while some others will be simply "filtered out" by our auditive system.

By definition, sound pressure is the instantaneous variation of ambient pressure:

$$p_{tot}(t)=p_0+p(t)$$

where p_0 is the average (slowly varying) atmospheric pressure and $p(t)$ is the time-dependent pressure variation. If an instruments were available, we would see something similar to what's shown in Figure 1.1: a rapidly varying signal superimposed to the (quasi) static ambient pressure.

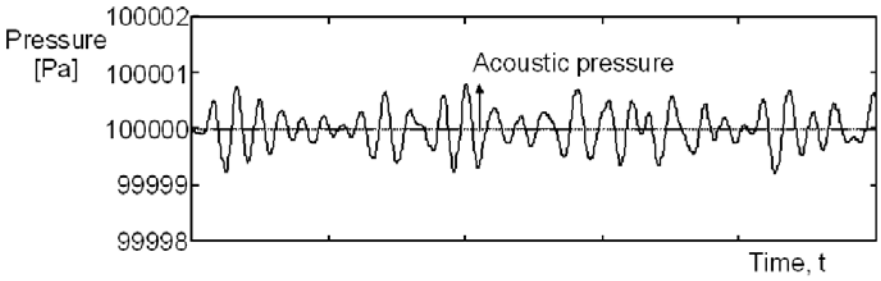


Figure 1.1 Small fluctuations of sufficient amplitude and of certain frequencies (approximately between 20 and 20000 Hz) of the ambient pressure are perceived as noise [1].

The human ear perceives the variations of sound pressure from the average line: that's why ambient pressure is removed from the signal and the square value of the pressure is considered, see Figure 1.2. As the value is rapidly changing, a useful indicator is the average of the squared pressure over the time of observation *T*.

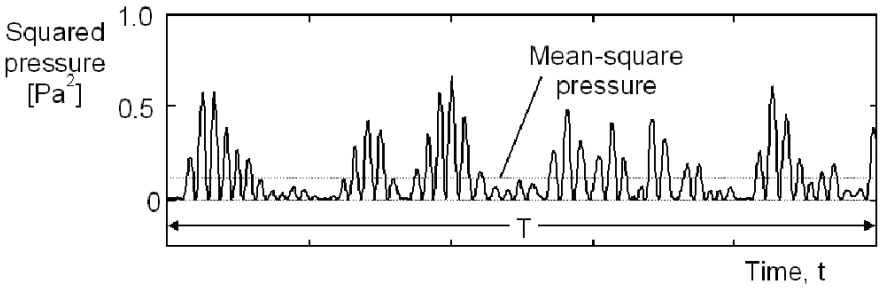


Figure 1.2 Squared pressure and average squared pressure over a given time T [1].

Mathematically, the average of a continuous function is defined as the integral of that function over the interval starting at time t_1 and ending at time t_1+T divided by the considered time interval T:

$$\overline{p^2} = \frac{1}{T}\int_{t_1}^{t_1+T} p^2(t)\,dt$$

It has been demonstrated by the branch of acoustics named *Psychoacoustics* that auditory sensation is not linear with the amplitude. It is said that the response of the human ear to noise is not linear but *logarithmic*, in the sense that a logarithmic function wheel simulates it.

Users of personal computers can verify it when adjusting the output level (volume) of the sound board (Figure 1.3): starting from 100%, volume can be adjusted in a number of discrete values. Supposing that the maximum level corresponds to the digital value of 128 and that the volume can be adjusted in 16 steps, each step is equal to 8/128=6.25%.

Figure 1.3 Typical noise adjusting panel of a personal computer sound board

The aural sensation is that the first steps give a limited sensation of volume reduction; it is only in the last steps that a real volume reduction is appreciated at each step. This can be explained with the table in Figure 1.4, where the levels and the absolute and relative attenuation are given in both linear and logarithmic scale. High-fidelity experts are used to work in logarithmic scales, and professional amplifiers have the volume jog with values normally expressed in dB. In electronic component catalogues, *linear* and *logarithmic potentiometers* can be found: the first type gives a set of linearly spaced values (=the difference of two consecutive values is constant), while the second type gives a set of logarithmic spaced values (=the ratio of two consecutive values is constant).

Volume adjustement with a linear potentiometer						Volume adjustement with a logaritmic potentiometer					
Initial value	*Final value*	*Absolute attenuation %*	*Relative attenuation %*	*Relative attenuation dB*	*Global attenuation (dB)*	*Initial value*	*Attenuation for each step dB*	*Final value*	*Absolute attenuation %*	*Relative attenuation %*	*Global attenuation (dB)*
128	120	-6.25%	-6.25%	-0.6	-0.6	128.000	-6	64.152	-49.88%	-49.88%	-6.0
120	112	-6.25%	-6.67%	-0.6	-1.2	64.152	-6	32.152	-25.00%	-49.88%	-12.0
112	104	-6.25%	-7.14%	-0.6	-1.8	32.152	-6	16.114	-12.53%	-49.88%	-18.0
104	96	-6.25%	-7.69%	-0.7	-2.5	16.114	-6	8.076	-6.28%	-49.88%	-24.0
96	88	-6.25%	-8.33%	-0.8	-3.3	8.076	-6	4.048	-3.15%	-49.88%	-30.0
88	80	-6.25%	-9.09%	-0.8	-4.1	4.048	-6	2.029	-1.58%	-49.88%	-36.0
80	72	-6.25%	-10.00%	-0.9	-5.0	2.029	-6	1.017	-0.79%	-49.88%	-42.0
72	64	-6.25%	-11.11%	-1.0	-6.0	1.017	-6	0.510	-0.40%	-49.88%	-48.0
64	56	-6.25%	-12.50%	-1.2	-7.2	0.510	-6	0.255	-0.20%	-49.88%	-54.0
56	48	-6.25%	-14.29%	-1.3	-8.5	0.255	-6	0.128	-0.10%	-49.88%	-60.0
48	40	-6.25%	-16.67%	-1.6	-10.1	0.128	-6	0.064	-0.05%	-49.88%	-66.0
40	32	-6.25%	-20.00%	-1.9	-12.0	0.064	-6	0.032	-0.02%	-49.88%	-72.0
32	24	-6.25%	-25.00%	-2.5	-14.5	0.032	-6	0.016	-0.01%	-49.88%	-78.0
24	16	-6.25%	-33.33%	-3.5	-18.1	0.016	-6	0.008	-0.01%	-49.88%	-84.0
16	8	-6.25%	-50.00%	-6.0	-24.1	0.008	-6	0.004	0.00%	-49.88%	-90.0
8	0	-6.25%	-100.00%	-∞	-∞	0.004	-∞	0	0.00%	-100.00%	-∞

Figure 1.4 Levels and attenuations for volume for linear (left) and logarithmic (right) potentiometers. The last column in each frame expresses the reduction as 20 times the logarithm of the ratio of the final level divided by the initial level.

In engineering, the use of *levels* is common as they give an easy tool to compare a quantity with a given reference value. That's why in acoustics the same concept is used, taking as a reference quantity the sound pressure corresponding at the threshold of hearing of a pure sine tone at 1000 Hz. This p_{ref} is tiny: a 20 millionth of a Pascal, i.e. more than 5 *billions times* lower than the atmospheric pressure. The basic definition of the *sound pressure level* is therefore the following:

$$Level = 10\log_{10}\left(\frac{quantity}{reference\ value}\right) \qquad L_p = 10\log_{10}\left(\frac{\overline{p^2}}{p_{ref}^2}\right)$$

This equation follows the rules of any logarithm-defined function. The unit of measure is the decibel (dB), i.e. ten times a Bel (that's due to the 10 in front of the logarithm). This means that, for example, a double pressure $2p$ pressure is perceived as 10*log10(4)=6 dB higher than a pressure p. Acousticians say that *doubling the noise increases the level by 6 dB*, and this well corresponds to the human sensation of a double annoyance. Trying to explain to politicians that reducing the level from, let's say, 88 dB to 82 dB, is not a 6.8% reduction of noise but 50% reduction of noise sometimes can be very hard!

Another element is that increasing the amplitude of source by a factor 10 increases the level by 20 dB. Note that noise amplitudes below the reference level give *negative* levels: -6 dB is not wrong, it only means that the level is approximately one-half of the reference value.

Just to have an idea of the enormous range of sound pressure that the human ear is capable of discriminating and "surviving", take a look at Figure 1.5, where a short list of typical levels is shown. The ratio between the threshold of pain and the threshold of hearing is 130 dB, i.e. the sound pressure at the threshold of pain is approximately 3 million times higher than that at the threshold of hearing!

	dB(A)
Threshold of hearing at 1000 Hz	0
Rustling of leaves	20
Quiet bedroom	35
Human voice at 1 m	60
Telephone ringing at 3 m	75
Jet aircraft taking off, at 300 m	100
Unsilenced motorcycle at 60 cm from exhaust	100
Riveting on a large steel plate at 2 m	117
Threshold of pain	130

Figure 1.5 Typical values of noise levels for different sources [1].

An interesting question of enormous importance for railway noise is: how can the total noise emitted by two or more sources be obtained by single

partial measurements? The answer to this question is not trivial. As basically sinusoidal functions, the sine waves from two sources can have exactly the same frequency, the same amplitude but opposite (180°) phase: in this case the noise sources cancel completely (that's what researchers continuously look for: the so-called *active cancellation of noise*, where additional sources are introduced in a noisy environment trying to reduce the disturbance at least to the receiver's ear).

A simple case, which well applies to railway noise, is where the sources are *incoherent*. Coherence is a rather complicated mathematical function establishing the degree of linear relationship between two signals, but in our case it will be enough to say that two sources can be considered incoherent if the noise from one source cannot be predicted from (it is not due to) the noise from the other source. As an example, environmental noise due to a car *and* an aeroplane passing simultaneously certainly falls into the category of incoherent sources.

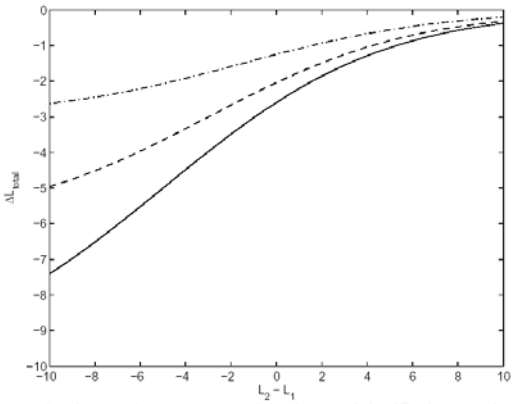
In this case, the basic equation is:

$$\overline{p^2} = \overline{p_1^2} + \overline{p_2^2} \Longrightarrow L_\Sigma = 10\log_{10}\left(10^{L_1/10} + 10^{L_2/10} + \ldots\right)$$

This equation, which is simpler than it appears, says that two sources of equal level give an overall level 3dB higher: acousticians are not mad when they say that 80+80=83 dB! Many practical consequences arise from this definition:

- If two sources have the same level, *removing one source reduces the total level by only 3 dB* (and that's why acousticians are *really happy* when a 3 dB reduction can be achieved!)
- If a source is much lower than another, it can be, in practice, omitted from the calculation of the overall level (for example, if a source is 20 dB quieter than the other one, the conclusion that 80+60=80 dB leads to an absolutely negligible error of only 0.04 dB).

A useful chart is shown in Figure 1.6, where the advantage that can be obtained on the total noise by a reduction of 3 dB, 6 dB or 10 dB from one of two sources of different levels is shown.



Change in L_{total} due to a reduction of 3 dB (– · – ·), 6 dB (– – –) or 10 dB (——) in L_1

Figure 1.6 Nomogram to estimate the reduction of total noise ΔL_{total} due to a reduction of 3 dB (-· -· -), 6 dB (- - -) or 10 dB(——) in the level of a source L_1 given for a given initial difference L_2-L_1 between the levels of the two sources [1].

1.2 Sound power and sound pressure; emission from vibrating surfaces and directivity

Sound pressure is what we hear but does not describe the source emission completely. The classical example is the comparison of a noise source with a heater. A heater has a power, that you can read on the label, but the sensation of comfort or discomfort, linked to the temperature which you are exposed to, is not related only to the power: it also depends on the environment, on the presence of insulated or conducting walls and so on. The same argument can be applied to noise: a noise source has a sound power *W* that generates sound pressure *p* that is perceived as noise.

Sound power can be evaluated from sound pressure when the source is placed in a special environment (called *free field* as there are no reflections of the outgoing sound waves) by properly adding the sound pressure multiplied by an elementary area around the source (the concept of *sound intensity I*) (Figure 1.7). Sound pressure is a *scalar* quantity and is defined only by its magnitude; sound intensity is a *vector* quantity and is defined by a magnitude, orientation and direction. Sound power can be estimated also in more complex environments (like a workshop or general industrial premises) by using *sound intensity probes*, which will not be discussed here.

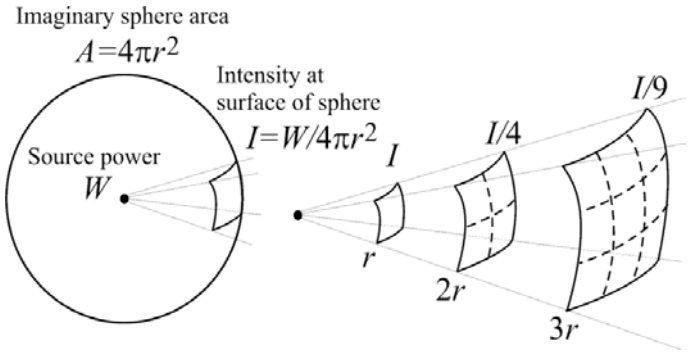


Figure 1.7 The sound power W of a noise source is spread over a spherical surface resulting in the sound intensity I which decreases with the square of the distance from the source [1].

In any case, moving away from a source, the area increases and, to keep sound power constant, sound pressure and sound intensity must decrease. It is a common experience that less annoyance is readily obtained by moving away from a particularly loud source and that the benefit is clear and immediate.

We will not further develop the concept of sound power measurement, which is already complicated enough for stationary sources and is almost inapplicable (also in principle) to moving sources.

Before leaving the subject, it is fundamental to anticipate that the most important portion of railway noise, i.e. the so-called *rolling noise*, is due to a mechanism of transformation of mechanical vibration into noise. Everybody knows how a drum works and we have all experienced a loudspeaker, where the coil excites a cone that vibrating generates sound pressure (or good music, hopefully!).
The basic relation linking the vibration field of a body to the sound power emitted by that body is

$$W_{rad} = \rho_0 c_0 S \left\langle \overline{v^2} \right\rangle \sigma$$

where $\rho_0 c_0$ is a characteristic of the medium (the air impedance), S is the area of the surface of the emitting source, σ is the radiation efficiency (depending on the shape of the body and on the boundary conditions) and the velocity v of the surface is squared and averaged over the surfaces and in time. Beyond the implications that this equation may have, that are

extremely interesting and complex at the same time, the role of the average velocity of vibration is clear and one of the methods that engineers use to reduce noise is to use damping treatment to reduce the amplitude of vibrations where they are at their maximum (i.e. at resonances). Calculations and experiments show, for example, that wheels of typical shape and size have an almost unit efficiency above approximately 500 Hz: that's why, as will be shown in the next chapters, wheel noise is normally dominant over rail noise at higher frequencies.

It is important to mention that another important characteristic of noise sources is *directivity*. Quite rarely a noise source is *omnidirectional*, and listening to a noise source resting on a flat floor and moving on a circumference (constant distance) around the source immediately gives the sensation that some positions are louder (or quieter) than others. Acousticians have identified the basic source as the so called *monopole*, also known as *pulsating sphere*, generating spherical waves in the infinite space. This source, which by definition is omnidirectional, can be combined in various ways giving origin to *dipoles*, *quadrupoles* and so on. Dipoles made of monopoles oscillating in phase have a marked directional emission: trains are often simulated as dipoles because it was observed that a directional emission is more corresponding to experimental results obtained during pass-by measurements. Directivity measurements are made with arrays of microphones placed around a circumference over the noise source in a free field or anechoic (i.e. without reflections, or completely sound absorbing) environments (Figure 1.8). It can be shown that low frequency sources are, in general, more omnidirectional than high frequency sources.

Figure 1.8 Sound directivity measurements of a track (left) and of a wheel (right). The environment has to be free from reflection, i.e. a free space or an anechoic (or semianechoic) chamber.

Modelling of a moving train can be done considering the train as a set of moving point sources or, in a simplified but well working approximation, as a moving linear source. The second approach, obtained considering the total sound power of the train uniformly distributed over the train length, gives the possibility to derive some general properties of noise observed from passing trains. Acoustic general rules, like that already shown in Figure 1.7, say that *sound pressure level decreases by 6 dB when doubling the distance from a point source*; similar considerations for linear sources lead to the conclusion that *sound pressure level decreases by 3 dB when doubling the distance from a line source*. This conclusion explains why in certain situations (for example when very long trains pass close to the receiver) the benefit that can be obtained by increasing the distance from the source is lower than expected.

From now on, we will only talk about sound pressure level, L_p, that must be measured during train pass-by at a prescribed set of distances in order to get a comparable set of measurements. No directivity or sound power measurements are normally possible, and this means that from the acoustical point of view the characterisation of the source is, in effect, rather limited and rough.

1.3 Frequency analysis of noise signals. Frequency weighting curves. Error analysis

Any measurable physical phenomenon can be either periodic or non-periodic. In the first case the phenomenon repeats itself after some time (the period *T*), while in the second case it is not possible to make a reasonable forecast of what will happen in the future on the basis of what we have observed till now.
Measurable periodic phenomena give rise to periodic signals, that can be treated by using the analysis tools developed in the XVII Century by the French mathematician Jean-Baptiste Fourier. Engineers talk about Fourier Analysis, meaning that the periodic properties can easily be defined in terms of *harmonics*.
By the definitions from the Fourier analysis, any periodic signal can be expressed as the sum of a (possibly) infinite number of special functions, i.e. sine functions, each defined at a frequency multiple of the fundamental

frequency, that is defined as the inverse of the period (f_0=1/T) expressed in Hertz (Hz). This is what is called a Fourier series. In the special case of a sine wave, only the first term of the Fourier series exists, i.e. the infinite set of numbers of the signal can be very efficiently compacted to only two numbers: the amplitude of the sine wave and its frequency.

As an example, a phenomenon repeating every second can be seen as the superposition of a sine function with (fundamental) frequency 1 Hz, another sine function with frequency 2 Hz, another sine function with frequency 3 Hz and so on. Each sine function will be multiplied by a specific factor (so-called amplitude) that can be calculated according to the Fourier theory. Plotting the amplitude of each sine component vs. the frequency the *frequency spectrum* can be plotted. This is, by far, the most common representation of signals from dynamic systems: torque, gear meshing vibrations, and many others are normally described by their spectra. It should be said that Fourier analysis is performed in the complex number domain, and that each frequency amplitude spectrum should also include a frequency phase spectrum. Railway noise has a marked random character (see later) where the frequency phase spectrum values are random. That's why phase is often neglected when dealing with railway noise frequency spectra.

Signals can be transient or continuous. Earthquake vibrations are by definition transient, while wind speed can be considered continuous for some time and then slowly varying (a special category of phenomena is that of *quasi-static phenomena*, whose average value changes slowly while the dynamic value changes rapidly). Noise can be periodic (a level crossing bell) or random (a waterfall). While the description of a periodic phenomena can be made by the observation of what happens in one period, random phenomena should be measured *forever* to analyse all their properties.
For evident practical reasons, measurements can not last forever. Investigation times are often decided according to available resources, limiting observation time to some seconds, minutes, hours and so on, depending on the phenomenon. An approximation that is therefore necessary is to expect (or, better, to suppose) that what has been measured will repeat in the following observation time. If this can not be considered exact for random phenomena, it is in any case possible *to estimate statistical properties of random signals from a limited observation* of a phenomenon. With this trick, acousticians make periodic also those phenomena that are

not periodic. The analysis of random signals is treated in specialized books to which the reader is referred for further information.

Frequency perception by the human ear is again non linear. Musicians consider the basic note (A) at 440 Hz and the concept of *octave* is well known. It means that the following A note will be the eighth key on the keyboard of a piano, to which corresponds a doubling of the frequency (=880 Hz). Psychoacoustics says that the discrimination of frequencies worsen with the increasing of frequency. For example, two tones distant 83 Hz are perceived as very different (440 Hz=note A, 523 Hz=note C); the same difference at a much higher frequency, well within the audible range, for example from 10000 to 10083 Hz, is not perceived at all. This means that not only the amplitude is better described in logarithmic scale, but also that frequencies are better described logarithmically. Octaves are therefore used in acoustics too, as shown in Figure 1.9.

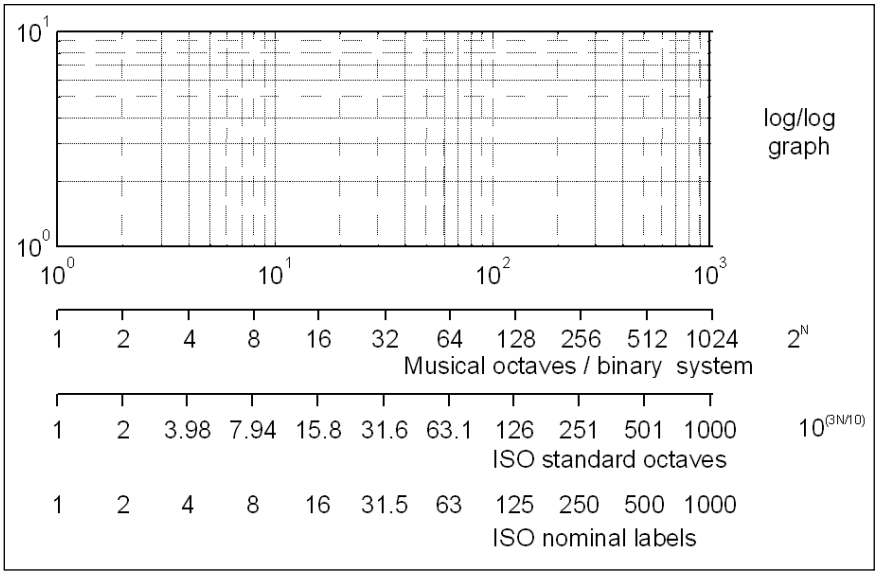


Figure 1.9 Log-log chart with amplitude expressed in decades (10 times=20 dB) and frequency expressed as octaves (musical, standard and nominal) [1].

When dealing with noise identification of tonal sources, octave description can result to be too approximate. That's why acousticians normally talk in terms of *one-third octave bands*, where the frequency doubling between

two following frequencies is divided into three parts. In this case, starting from 100 Hz, the first eleven frequencies are:

100 125 160 200 250 315 400 500 630 800 1000 Hz

Starting from the first frequency, the following can be calculated with the rule $f_{n+1}=f_n*2^{1/3}$. The interesting property is that $(2^{1/3})^{10}=10.08\approx10$, and the sequence can be extended at upper and lower frequencies by multiplying or dividing the above sequence by 10. Most of the diagrams in this book are spectra plotted in log-log scales, with dB on the y-axis and one-third octave frequencies on the x-axis.

Another fundamental characteristic of the human ear is that it is not equally sensitive to all frequencies. For example, a level of 80 dB at 100 Hz (equivalent to an average pressure of 0.2 Pa) is perceived as much less loud than the same level at 1000 Hz. The audible frequency range is conventionally believed to be between 20 Hz and 20 kHz for a young person in good health, although ageing affects auditive capabilities. To convert a noise into the auditive sensation, acousticians have developed the *frequency weighting curves*. There are several types of these curve depending on the strength and on the type of the source, but the most widely used is certainly the *A-weighting* curve shown in Figure 1.10.

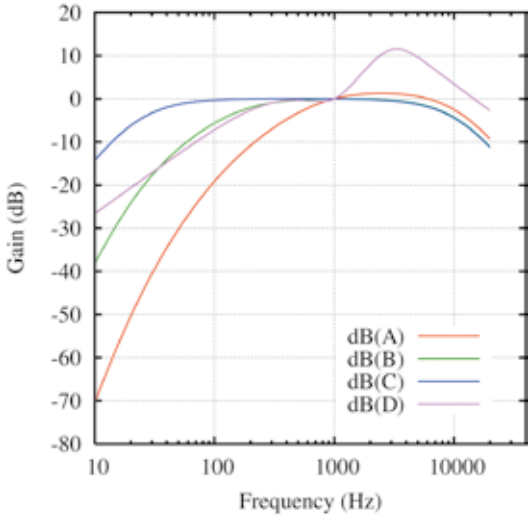


Figure 1.10 Standardised frequency weighting curves expressed as gain (or attenuation) vs. frequency

From Figure 1.10 it can be seen that the tone at 100 Hz is approximately *attenuated* by -20 dB, i.e. the 80 dB level is perceived as equivalent to a disturbance of 60 dB. The spectrum of a given noise can then be weighted with the A-curve in order to obtain a *weighted spectrum*. The overall value describing the disturbance on the receiver can be obtained by reverting the calculation performed with Fourier analysis: the various sine components at different frequencies are weighted and added to give the *A-weighted overall level* L_{pA}, expressed in dB(A).

We conclude this chapter with some considerations on railway noise pass-by measurements. The pass-by of a train is inevitably a transient phenomenon. Also in the case of nominally equal vehicles (like homogeneous long EMU passenger trains or long freight trains made of the same type of freight car), noise sources are different along the train. As any other type of measurements, noise measurements are affected by two types of errors: *bias errors* and *random errors*. Bias errors are tackled by proper calibration of the measuring chain, while random errors can be reduced by *averaging* a sufficiently large number of measurements. As random errors are such, their average value tends to zero and it is *rejected* by the averaging process. Theory of errors is based on the relationships that link the desired "precision" (statistical uncertainty) of a measurement with the duration of the measurement. It could be shown that, for a given uncertainty, the measurement duration is longer for low frequencies. It means, for example, that the precision of the measurement of a pass by lasting, let's say, 20 s, is lower if the train mainly emits at low frequencies rather than at high frequencies (that's what normally happens at low speeds).

Considering that the source is not stationary, that averaging is almost impossible and that the components at low frequencies are often dominant (especially at low train speed), it results that precision of railway pass-by noise measurements is really very low. ISO standards define three grades of accuracy (Precision, Engineering and Survey) with the corresponding uncertainties; under the hypothesis that a pass-by noise measurement is sufficiently "precise" to fall in the Engineering precision category, the uncertainty is in the order of ± 2 dB. This means that giving noise measurements with a large number of digits is wrong and illusory; furthermore, *this means that measuring several times*

the same train passing exactly in the same conditions (i.e. ensuring repeatability) results anyway in a "cloud" of values varying of several dB around the average value. Laws and directives saying, for example, that "maximum noise during pass-by must not exceed the limit of *n* dB", are ambiguous by definition. Only statistics should be used in railway noise measurements.

1.4 Instrumentation for railway noise measurements. Measuring quantities

As the sound pressure level L_{pA} is the best indicator to quantify *the effect* of a sound source on a human being, it is not surprising that on the market there are a large number of instruments called *sound level meters*. The description of all possible instruments lies outside the scope of this book, and its simple to verify with any Internet browser that prices can range from 30 US$ to some thousand Euros.

In principle, a sound level meter, or SLM for short, includes a microphone, a preamplifier, and an electronic circuitry including a frequency weighting network and a time weighting network. We will not discuss the details of microphones and preamplifiers here, leaving this subject to specialists; as frequency weighting has already been described, instead we discuss here what time weightings are.

The weighting network has been implemented since the beginning of instrumentation, obviously analogue, and some low-cost purely analogue sound level meters still exist, as the type shown in Figure 1.11.

The basic SLM has a switch to set frequency weighting to A, C, or other curves (sometimes there is also a "LIN" position meaning that there is no frequency weighting at all) and a switch to set the time weighting (or time response) to FAST or to SLOW. Time response network is simply an RC circuit that makes the value readable by the needle of the indicator. If the noise source is relatively steady, a good readout can be obtained by using the SLOW time response (corresponding to 1 s), filtering away the inevitable small fluctuations of the signal. This setting makes the movement of the needle "doughy" but very easy to read.

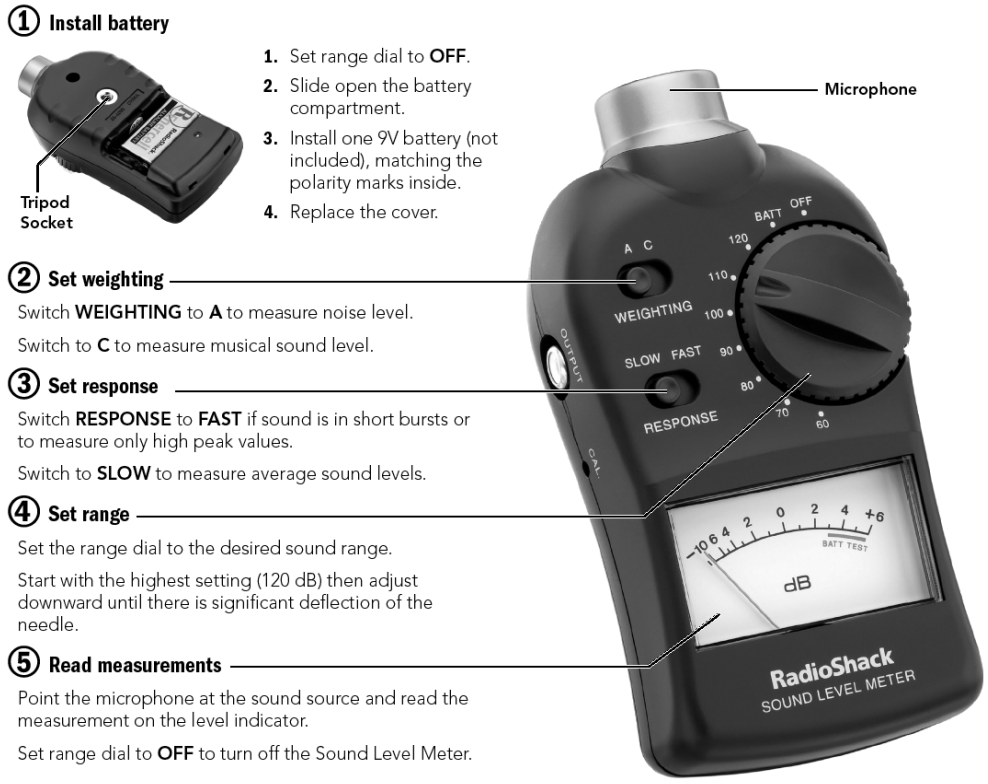


Figure 1.11 Example of an analogue Sound Level Meter [3].

If the signal is much less stable, or if the technician is interested in the details of the signal, the time constant FAST (=0.125 s) can be used, and in this case it will be harder to follow a rather "nervous" needle.

Modern high performance SLMs are almost all digital, have large memory banks and can all interface with digital computers. At the other extreme, there are fully digital SLMs: the hardware analogue part is limited to the microphone and to the preamplifier. Once the signal is digitised through a Analogue-to-Digital Converter (ADC), the software installed in the computer can perform all the operations simulating the use of a frequency network and time response filters.

The typical setting for noise measurements is described in standards for noise measurements; the EN ISO 3095:2005 standard, which is currently (2009) in force, specifies for example a set of microphones to be placed along the line to measure pass-by noise (Figure 1.12). Due to the transient nature of the pass-by, time response must always be set to "FAST".

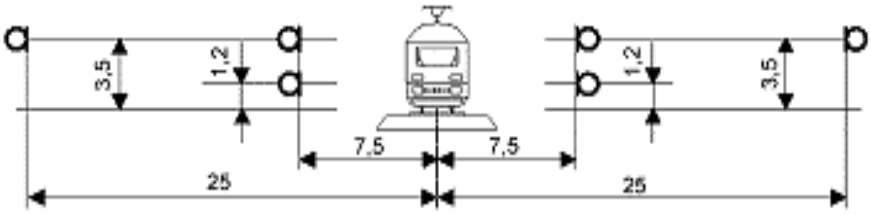


Figure 1.12 Microphone positions for pass-by measurements during train pass-by (extract from EN ISO standard) [4].

For research purposes, it is possible to use many more microphones and accelerometers mounted, for example, as shown in Figure 1.13.

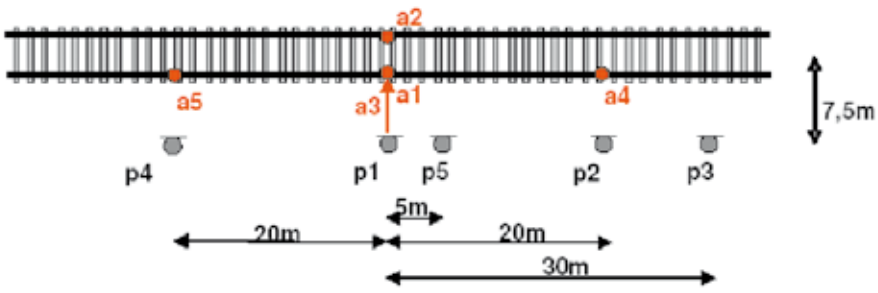


Figure 1.13 More measuring microphones and accelerometers on the rail can be used for research purposes [1].

Microphones are normally mounted on tripods, protected from environmental agents (and birds!) and connected to a "front-end" where the cables from all the microphones are collected. The front-end can either be stand-alone, including all the electronics, a screen, a keyboard and a pointing device, or can be simply a laptop interfacing with the front-end that in this case can also be just an ADC board also providing power supply to the microphones (Figure 1.14).

Figure 1.14 Typical mounting of a measuring microphone for pass-by measurements. The waterproof microphone cartridge is protected by a windscreen and by protection from birds [5].

The typical signal recorded during a train pass-by by an external microphone is shown in Figure 1.15, where the *buffer-to-buffer* time, T, and the pass-by time, T_p, can be seen. Pass-by time is defined as the time between the initial rising edge of the signal (-10 dB before the front buffer) and the final falling edge of the signal (-10 dB after the rear buffer).

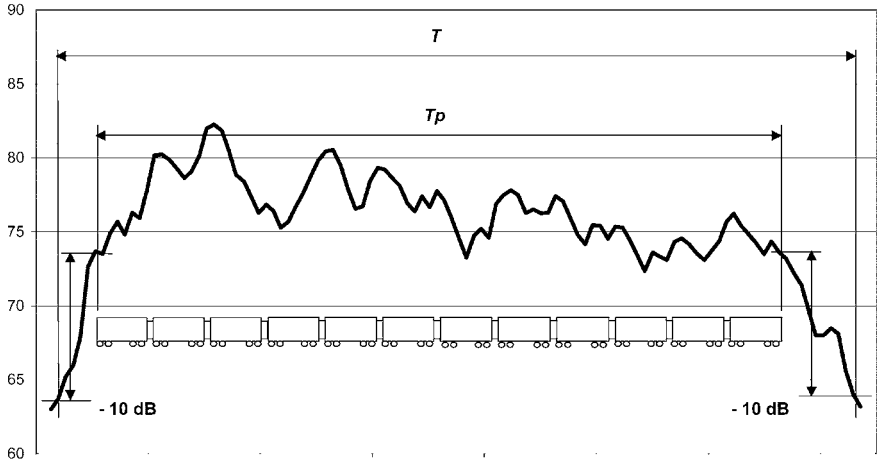


Figure 1.15 Typical shape of L_{pA} (measured with time constant "FAST"), with indication of buffer-to-buffer and pass-by times [4].

The levels can therefore be readily calculated by the definition of the equivalent sound pressure level as

$$L_{p\mathrm{Aeq},T} = 10\lg\left(\frac{1}{T}\int_0^T \frac{p_\mathrm{A}^2(t)}{p_0^2}\mathrm{d}t\right)\mathrm{dB}$$

for the buffer-to-buffer time and

$$L_{p\mathrm{Aeq},T_p} = 10\lg\left(\frac{1}{T_2 - T_1}\int_{T_1}^{T_2} \frac{p_\mathrm{A}^2(t)}{p_0^2}\mathrm{d}t\right)\mathrm{dB}$$

for the pass-by time.

Signals are stored and processed in order to calculate the quantity requested by the standards. A list of various indicators (levels) is shown in Figure 1.16. The interested reader is referred to standards and textbooks on acoustics for further details.

L_{Amax}	Maximum A-weighted sound pressure level during the measurement interval (the integration time of the sound level meter affects this result).
$L_{\mathrm{Aeq,T}}$	Equivalent continuous sound pressure level. It is the (A-weighted) level of the constant sound that has the same mean-square pressure over the integration time T as the actual sound.
L_{AE} (SEL)	Sound exposure level. This is used to characterise single events. It is the (A-weighted) sound pressure level of a constant sound of duration 1 second that has the same L_{Aeq} as the original sound.
L_{N}	The noise level exceeded for N% of the time period under consideration. For example L_{10} was used in the UK to describe traffic noise. L_{90} can be used to describe background noise.
L_{DEN}	Day/evening/night level. An equivalent continuous sound pressure level that is weighted by +5 dB for the evening period and +10 dB for the night period, used in EU legislation.

Figure 1.16 Some indicators used for the description of annoyance due to train noise [1].

1.5 Advanced measurements

Equivalent sound pressure levels are an average of the energy of the signal during a given time (the buffer-to-buffer or the pass-by times) and, as any average, do not highlight the contribution of each source, whilst the maximum A-weighted sound pressure level (with FAST time constant) L_{pAFmax} is too rough a representation of the phenomenon and doesn't clarify who was responsible for that maximum and at which frequency.

Although there are not many other possibilities for the execution of standard railway noise measurements, it is, in any case, interesting to show some other techniques available at research level.

Extremely interesting results can be obtained by the use of an array of microphones. An array of microphones consists of a medium to high number of microphones (approximately from 8 to 100 microphones) mounted in different shapes (vertical, horizontal, X, spiral, etc.). The principle of beam forming is used to follow the single sources and to quantify the emission at different frequencies. The results are extremely useful and attractive, as colour maps can be used to readily identify the main culprit for emission. Figure 1.17 to Figure 1.19 give an idea of different types of array of microphones.

Figure 1.17 A rectangular array of measurements used during pass-by measurements [6].

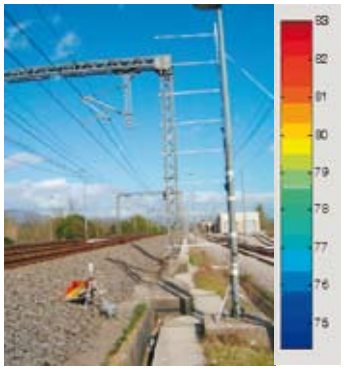
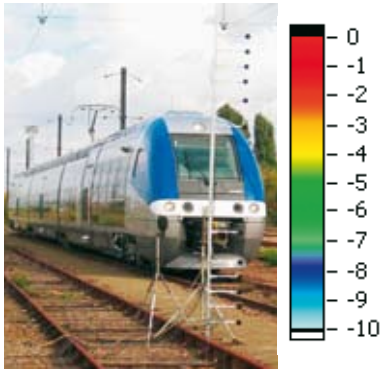


Figure 1.18 A linear (vertical) array of microphones with indication of the noise maps at different frequencies [6].



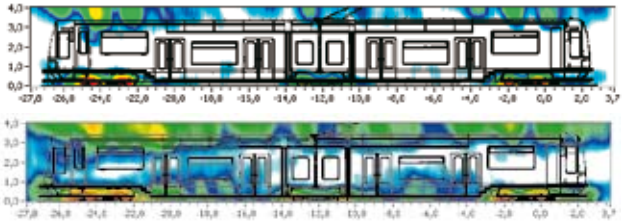


Figure 1.19 Another example of a vertical array of microphones [6].

It is interesting to mention the *Pass-By Analysis method* (*PBA*) too that, since the late 90s, has allowed for the assessment of combined roughness by use of rail vibration measurements and that proved to be an efficient and cost-effective tool for various purposes in relation to railway rolling noise. Originally developed in EU projects (METARAIL, STAIRRS) and closely related to the TWINS model, the assessment of combined wheel-rail roughness can be used for several purposes:

- assessment of wheel roughness on a smooth track
- separation of roughness and sound radiation effects when evaluating noise control measures such as wheel or rail dampers
- separation of rolling noise from other sources
- assessment of the effect of vehicle and track noise control measures
- monitoring wheel/rail roughness and track characteristics over time
- assessment of rail roughness with smooth wheels
- calibration of on-board indirect roughness measurement

As has been described above, noise measurement precision is dependent also on the acquisition time. A different approach to source quantification is therefore to try to measure the noise standing *on the train* instead of trackside. Unfortunately airflow turbulence is such that the use of measuring microphones is impossible. A device comprising of a microphone protected by a semi-cylindrical steel or aluminium case was developed and tested in a wind tunnel up to 300 km/h and found to be insensitive to airflow thanks to its shape. The device was used rather extensively in the 1994-2000 period by one of the authors (Figure 1.20).

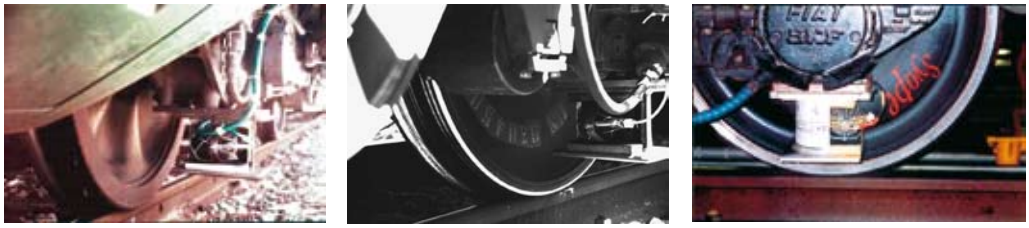


Figure 1.20 Axlebox mounted device to measure wheel-rail noise at the source continuously.

1.6 Estimation of environmental impact

The final user of external noise measurements, calculation and planning is the citizen living close to the railway line, yard or station. Environmental engineers use Geographical Information Systems (GIS) and CAD modelling to locate, describe and model a certain environment where noise sources are located with their power, emission figure, frequency composition, etc. (Figure 1.21).

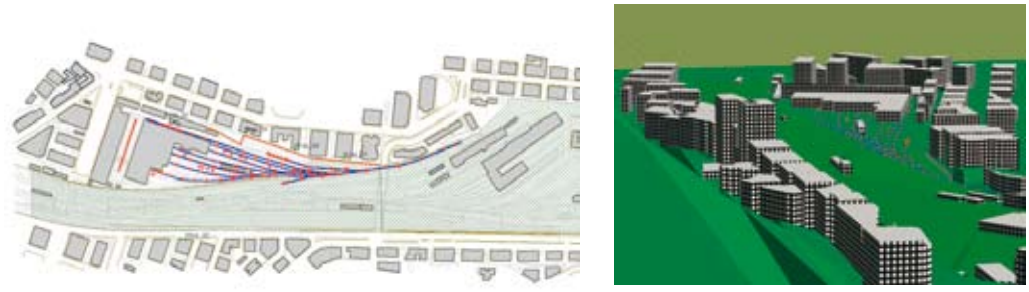


Figure 1.22 Noise map estimated with the use of commercial software packages for environmental impact simulation [6].

Using software available on the market, it is possible to derive a digital map of the area with contour lines with the same equivalent level for the chosen period (day, night, 24h, etc.) (Figure 1.22). The subject, although extremely interesting, lies outside the scope of this book.

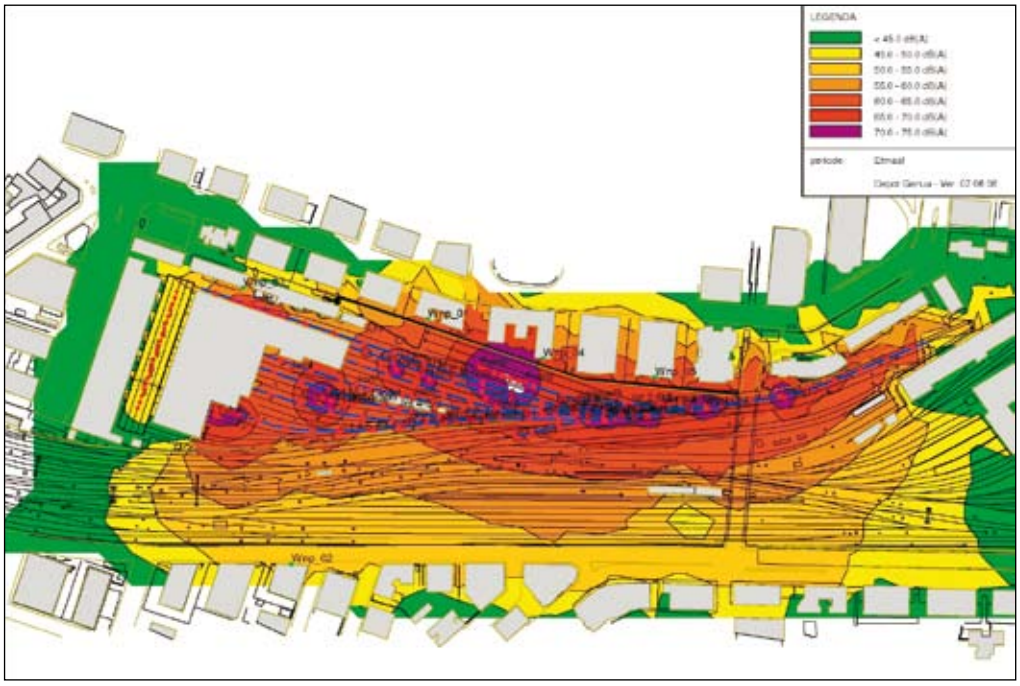


Figure 1.21 GIS map of a yard and 3-D reconstruction of the buildings around it [6].

2 RAILWAY NOISE FEATURES

2.1 Background

By observing noise from outside a railway line, distinct features can be observed also by people with little or no background in noise.

First of all, the classical pass-by noise, if observed from a certain distance in a quite "regular" space (what the acousticians call the "free field" situation, with a definition that very well resembles the physical situation, without obstacles or absorbing surfaces), is a clear function of train speed.
Noise levels depend quite heavily on train speed; on the contrary, time duration of the noise disturbance is evidently inversely proportional to train speed. It is therefore necessary to identify a parameter that allows to compare, from a numerical point of view, different situations.

As the train slows down, noise does not disappear but becomes more related to the sources installed on the vehicles. Technicians know that a train, in the broader sense, is a complex machine that includes many mechanisms, each of them producing noise.
If we consider the case of a self-propelled passenger set of vehicles (the so called EMU – Electrical Multiple Unit), it is easy to list a number of items installed on board: traction converter, traction electric motors, battery chargers, air compressors, air conditioners and fans, etc.
Although the level of these noise sources may be dependent on the speed, normally they emit a constant sound power when they are activated. It is clear that this noise is perceived by a listener also when the train is stopping at a platform, for example. This noise, called stationary noise, represents a sort of lower limit to train noise, i.e. a stationary train is not completely quiet. To make an electric train completely silent it is a good idea to lower the pantograph… which sometimes by itself constitutes a non-negligible source scratching on the overhead line.

Similar considerations, but at an even more important level, can be depicted for Diesel locomotives and for DMUs (Diesel Multiple Units), as the thermal engine, and especially the exhaust system, can be an important source at all speeds. Idling, full throttle or intermediate

positions of the engine regulator may result in completely different noise characteristics. The use of hybrid vehicles, with electric traction motors fed by batteries in urban areas and with Diesel engine generating electricity (and recharging batteries, obviously) on the outskirts is unfortunately not possible due to typical train masses and requirements in terms of acceleration and normal slopes encountered during service.

In the last 50 years the psychological barrier of 200 km/h in normal operation has been decidedly overcome. Since the '70s of the last Century, trains have been running regularly at 300 km/h and extensions are certainly technically possible as world speed records largely exceed 500 km/h. Unfortunately, drag forces are proportional to v^2, where v is the speed of the train, and engine power depends therefore on v^3. This means that running at 300 km/h requires 3.375 times the power required at 200 km/h, and that increasing, for example, to 400 km/h requires a further increase by a factor 2.37. In other words, doubling the speed requires 8 times higher power. That's why the "conventional" limit of 300 km/h still suits most of the needs of modern railways keeping an eye on energy saving.
In any case, acousticians are well used to dealing with aeroacoustics sources. Depending on the modelling of the source, it can be seen that sound power emitted can be dependent up to v^8, implying an enormously increasing noise with speed. At times, at a speed dependent on how the train is built, aerodynamic noise can become absolutely prevalent against the other sources of noise.

Summarizing, noise can be divided into (Figure 2.1):

- Traction noise, due to auxiliary sources, which may be prevalent at speeds below 30-50 km/h and that is almost constant, or gradually increases with speed;
- Rolling noise, due to the interaction of wheels and rails, which is by far the most important source of disturbance in the 50-250 km/h speed range and that depends on train speed as later described;
- Aerodynamic noise, which increases quickly with speed and becomes dominant after v=250-300 km/h.

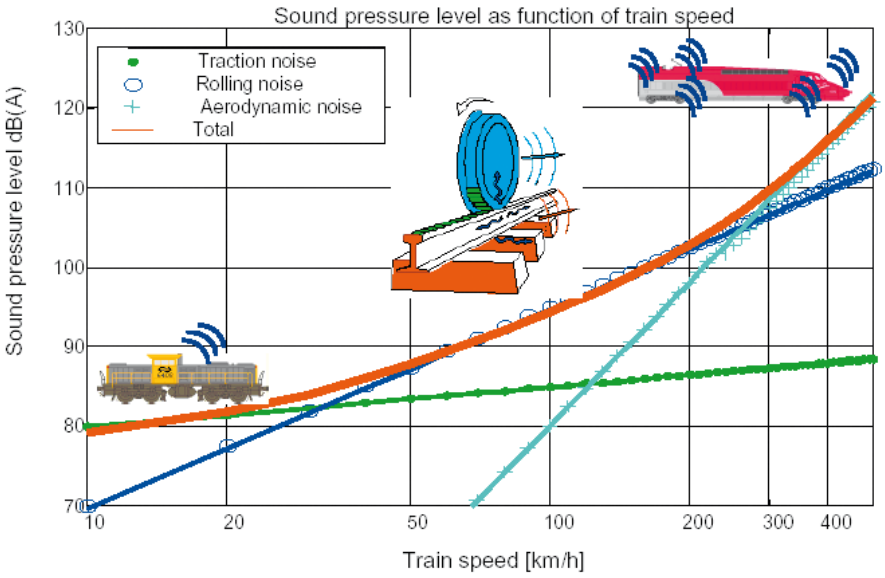


Figure 2.1 Typical dependency on speed of railway external (trackside) noise. Values of maximum Sound Pressure Level during train pass by (LpA, dB ref 20 µPa) [7].

2.2 Rolling noise basics

The central part of the diagram in Figure 2.1, which is compressed by the logarithmic scale of the x-axis, is in fact the most important one as the vast majority of trains runs at speeds between 50 and 250 km/h. In this speed range the trend of the maximum Sound Pressure Level, A-weighted and with time constant *FAST* (=0.125 s), is quite regular and can easily be found by a least squares fit of the cloud of values obtained during several pass-bys at different speeds. In literature a typical law for this segment of the overall noise curve is found to be

$$L_p = L_{p0} + 10 * \log_{10}(v^3/v_0^3) = L_{p0} + 30 * \log_{10}(v/v_0)$$

where L_{p0} is the value measured at the reference speed v_0. As can be seen, rolling noise depends on speed from a power of (approximately) 3. This means that doubling the speed leads to an increase in the noise level of $10 * \log_{10}(8) = 9.0$ dB, or that halving the speed leads to a reduction of -9.0 dB. This value should be kept in mind as, although not really feasible, one of the measures that can be adopted to reduce noise is to reduce train speed.

Rolling noise is generated by a complex interaction mechanism that involves primarily the wheel and the rail but in which non negligible components are due to other parts of the vehicle and of the track (sleepers, ballast).

The fundamental flow chart of rolling noise generation is shown in Figure 2.2, while some sketches representing the noise generation mechanism are shown in Figure 2.3. These results come from the work of Prof. David Thompson of the Institute of Sound and Vibration Research in Southampton (UK), that in the 90's developed the theory and a software package named TWINS (Track-Wheel Interaction Noise Software) that is capable of predicting noise emission from a variety of wheel and rail types and combinations.

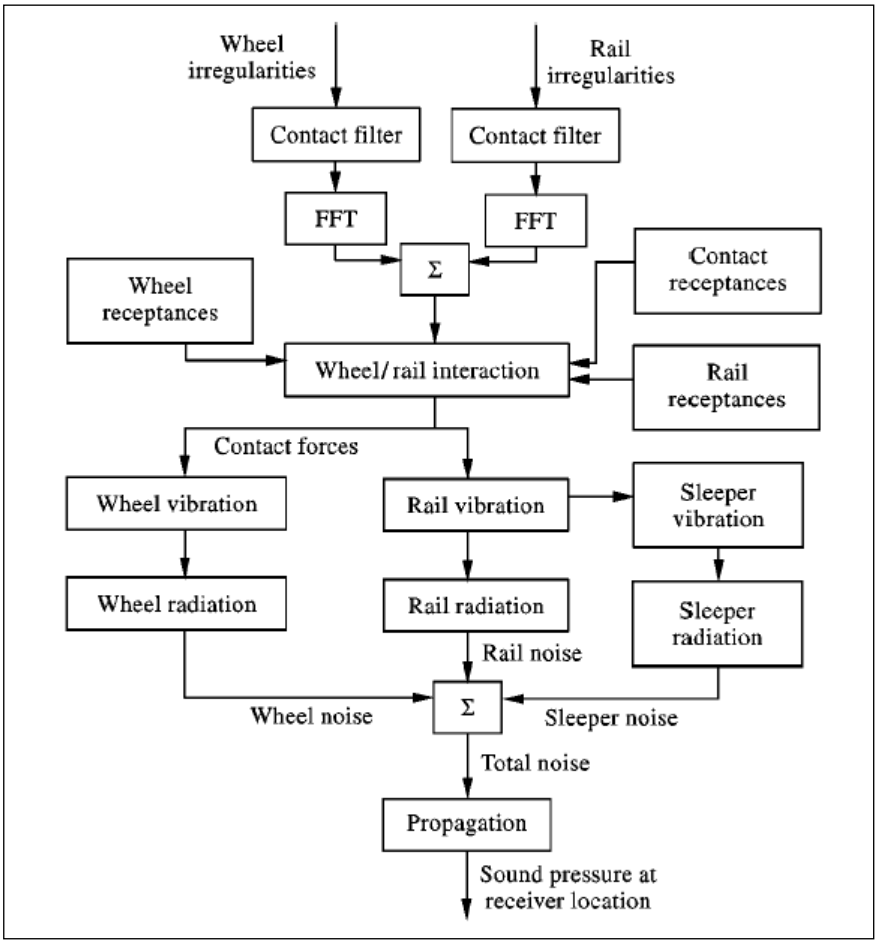


Figure 2.2 Rolling noise generation mechanism flow chart [8].

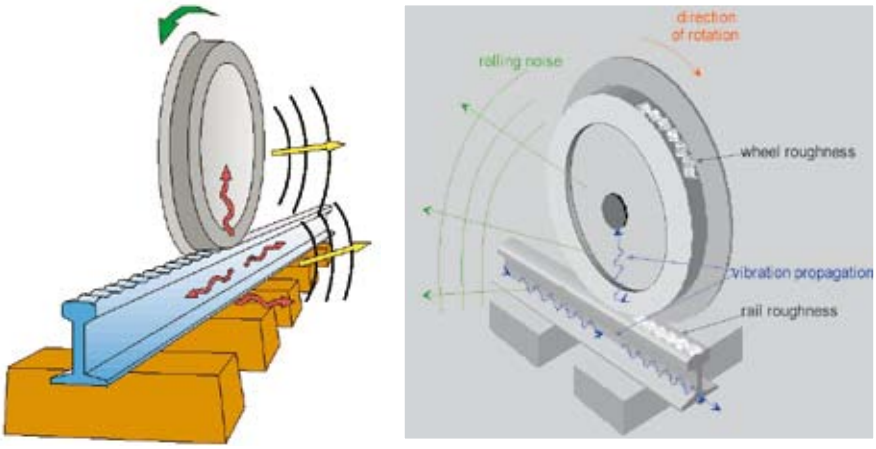


Figure 2.3 Rolling noise generation mechanism sketches [5].

The process can be schematically described as follows, in a rather simplified way:

1) Wheels and rails are not perfectly round or straight. There are inevitably some irregularities that are commonly known as "out-of-roundness" (OOR for short) for wheels and as "roughness" for rails. The majority of the rest of this book will be devoted to these subjects and to their influence on noise generation.

2) Although made of a very stiff material, steel wheels and steel rails have a finite stiffness. This means that when a wheel is pressed on the rail by the train weight (the so called axleload) a finite size area, named contact patch, is established. This area is elliptical in most cases and the sizes of the ellipse depend on the load, on the wheel and rail curvature and on the elastic properties of the material (namely the Young's modulus, E, and the Poisson's modulus, ν). Contrarily to what most people think, *hardness* is not involved in the contact patch size definition. The theory developed by Hertz considers the *elasticity* of bodies in contact, in the sense that no permanent deformations (*plastic* deformations) are produced during contact or, simply, that the yield stress of steel is not reached. Hardness is the attitude of a material to oppose to the indentation made by a penetrator: it is not therefore related to the phenomena of interest here.

As the Hertzian contact area has finite dimensions, very roughly speaking, it could be said that all irregularities "shorter" than the patch dimensions are "squashed" and that only longer irregularities are actually felt by the wheel-rail contact.

3) The concept of "short" and "long" irregularities is explained by the concept of *wavelength*. As already seen in the introductory chapter on noise, any signal, either periodic or non-periodic, can be seen as the superposition of an infinite number of sine waves of multiple frequency and different amplitude. This concept, that is normally considered in the time domain, can easily be extended to the space domain. By changing the independent variable time t with the distance x, frequencies f become wavelengths λ. The result of the calculation of the basic components on which a signal (either in the time or in the space domain) can be split is the so-called *spectrum*. The spectrum, i.e. a plot with the amplitude of all the components vs. frequency (for signals in the time domain) or wavelengths (for signals in the space domain), can be calculated by using the Fourier Transform, i.e. a mathematical tool that has been implemented on digital computers (dealing therefore with *sampled data*). As many calculations need to be performed to estimate the spectrum of a given signal, a particularly effective computer routine was developed in the '60s and called FFT (Fast Fourier Transform). The use of spectral components is particularly effective instead of using signals "as they are" because they give origin to very compact and efficient representations of a long lasting phenomenon.

4) Once filtered by contact patch filter and transformed in the frequency domain, wheel and rail irregularities can be summed up to form the input to wheel and rail.

5) Wheel, rail and the contact patch, that has its own peculiar properties, are therefore excited by the resulting displacement. The force that is effectively transmitted to the wheel and to the rail is modified by the intrinsic elastic properties of these components. The *receptance*, defined as the displacement of an elastic body under a unit force acting at a given frequency, is the useful function that can transform displacements into forces. The proper use of receptances is the last step of the upper part of the diagram in Figure 2.2 that is devoted to

the estimation of the contact forces. It is clear that in this part of the calculation scheme, wheel and rail properties are relatively known and that the variation of rail irregularity represents the most important source of contact force variation along a given railway line.

6) When contact forces are known, they can be used to simulate the emitted noise by using the lower part of the diagram. Contact forces are turned into wheel vibrations, rail vibrations (and sleeper vibrations) by using the *frequency response function* of these components. As noise is, to some extent, proportional to the averaged square surface velocity of a vibrating surface, sometimes these frequency response functions are given in terms of the velocity per unit force, i.e. the concept of *mobility*. Wheel and rail surface velocity can, therefore, be estimated by the relative mobility transfer functions.

7) Wheel, rail and sleeper vibrations are therefore "modified" by wheel, rail and sleeper radiation properties. Radiation properties are quite complicated to explain as they determine to which extent the velocity field of the free surface of any elastic body is *effectively* turned to noise. In the classical case of a vibrating piston surface delimited by an infinite plane (the so called *infinite baffle*), theoretical solutions indicate how much of the vibration is transformed into noise. For more complicated geometries, loads and, generally speaking, boundary conditions, only computer simulations by possibly using a mixture of finite and boundary element (FEM and BEM) techniques may calculate radiation properties accurately enough.

8) Noise generated by radiation of the wheel, rail and sleeper can therefore be summed up to give the total noise at the source level.

9) Total noise at the source is transferred to the receiver by the particular environment involved; also in this case it is therefore possible to identify a propagation transfer function that depends on the peculiarities of the objects found along the transmission path, their acoustic properties and so on.

While rail vibrations are measured by using conventional accelerometers glued on the various portions of the rail (head, web, foot), wheel vibration

measurements are complicated by the rotation of the wheelset. In this case it is necessary to use sliding contacts or telemetry systems. A typical experimental set up to measure wheel vibrations is shown in Figure 2.4.

Figure 2.4 Accelerometers mounted on a wheel. Signals are normally transmitted via radio telemetry [6].

Although the rest of the book will mainly be devoted to the analysis of the available countermeasures to reduce rolling noise, it is, in any case, interesting to briefly define other types of noise arising from the wheel-rail contact point.

2.3 Other sources of external noise

Tracks are not always straight and rails are not infinitely long. Moreover, track formation is not always levelled with ground as cuttings, bridges and other types of superstructure are normal.

We will limit the discussion to the following types of noise directly related to wheel-rail contact:

- Impact noise
- Squeal noise
- Flanging noise

A small chapter will be devoted to some basic considerations on ground-borne noise and vibrations and on aerodynamics noise.

2.3.1. Impact noise

Impact noise is mainly due to rail joints. Modern railways are built by welding already rather long rails (108 m to 120 m) by using either electrical (*flash butt*) or thermal (*aluminothermic*) welding procedures to form the so called *continuously welded rail*, or CWR for short. With this practice, it is clear that a CWR rail is not free to expand or to reduce its length according to temperature variations.

To get an idea of the lengths and stresses involved in the process, in many countries the track must, without speed restrictions, withstand temperature variations of around 80°C centred on a yearly average temperature T_0 of the particular site. As an example, if T_0=20°C it results in a $T_{max}=T_0+\Delta T$=20°C+40°C=+60°C and $T_{min}=T_0-\Delta T$=20°C-40°C=-20°C.

A one kilometre long rail would increase its length by $\Delta L=\alpha\ L\ \Delta T=12\bullet10^{-6}*1000*40=0.48$ m. As this expansion is constrained as the rail is continuous, the internal stress is given by $\sigma= E\ \alpha\ \Delta T = 2.1\bullet10^{11}*12\bullet10^{-6}*40=$ 101 MPa, where values for E and α are those typical for steel. It is clear that stresses are particularly high in compression (during summer, leading to possible track bucking) and in traction (during winter, leading to possible rail cracking).

Although modern track is designed to withstand these extremely demanding loads, rail joints are inevitable as long as rails are used as electrical conductors for signalling equipment and, for electrified railways, for current return to electric substations. The *insulated rail joint* (IRJ, see Figure 2.5) is needed every time a conventional *track circuit* is involved (audio frequency track circuits don't need IRJs).

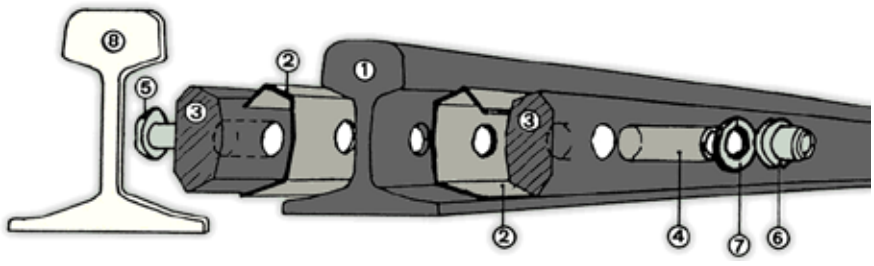


Figure 2.5 Typical insulated rail joint composition. Fishplates (3) are bolted to rail segment (1) with "spikes"(4-5-6-7) with the interposition of insulating layer (2).The two rails to be jointed are insulated with the plastic shim (8) [9].

Moreover, switches and crossings (S&C) represent a discontinuity as their *sleeper panel* is not normally able to transmit longitudinal forces due to CWR. S&C are therefore inserted in the main track by interposition of rail joints. In the crossing panel, vehicle guidance in guaranteed by check rails while the intersection of rails in the frog area generates unavoidable impacts.

Railway stations are full of joints and switches, and unless S&C with moveable parts are used (either with moveable points or with moveable swing rails) where the discontinuity is removed, the stations will always be characterized by the classical "clickety-clack" noise.

Another important source of impact noise is local irregularities of wheel tread known as *wheel flats*. Wheel flats, as the name suggests, are due to wheels sliding during braking due to poor and/or insufficient adhesion. Wheel tread material wears out destroying the circular shape of the tread locally introducing a flat spot (chord) in the wheel circumference. Almost everyone has travelled on trains with wheel flats and the continuous "ta-ta-ta-ta" noise is particularly disturbing. Another feature of relatively small wheel flats is that they are localized on a pretty narrow area (with transverse dimension in the order of 10 to 15 mm) (Figure 2.6).

Figure 2.6 "Artificial" wheelflat obtained by grinding a wheel tread for research purposes. The appearance of "natural" wheel flats is absolutely similar.

When running in curves, wheel flats noise may disappear as the contact point moves transversally thanks to lateral wheelset movement and the

consequent lateral shift of the contact point in areas that were not previously damaged by sliding. In any case, wheel flats should be avoided as much as possible as the loads, due to impacts, lead to structural failures in roller bearing cages, concrete sleepers and even in wheels, rails and axles.

Modelling of impact noise is particularly complex as wheel-rail contact may be considered linear only for small load variations. Generally speaking, and leaving more complex arguments to specific textbooks, it can be said that impact noise does not follow the speed law valid for rolling noise and that at higher speeds the wheel tends to "fly" over wheel flats and step-down joints. An interesting conclusion from research is that excitation mechanisms during impacts are similar to those involved during rolling noise generation, so, in principle, measures adopted to reduce rolling noise should be effective also to reduce impact noise.

It is clear that impact noise constitutes a particularly annoying and prevailing source; the obvious remedy would be to eliminate this discontinuity as much as possible. If this is not normally possible for S&C and IRJ, it must be said that also flash butt and aluminothermic rail welds may constitute an important source of local rail irregularity. An accurate geometry after welding of the rail joint should be restored by proper grinding of the rails after removal of welding remains.

2.3.2. Squeal noise

Curve squeal noise is a high level, strongly tonal noise that is generated in narrow curves for specific and local conditions. It is due to a combination of attitude of wheels while a vehicle is curving combined with friction characteristics at the wheel-rail interface. Many people living close to metro, light rail, tramways and railway lines close to stations are affected by this particular type of noise. It is a shouting whistle with levels that may be 20 or 30 dB higher than the loudest noise due to rolling noise. As squealing happens normally in narrow curves where rolling noise is normally limited, the disturbance given is even higher as the *differential* between the normal noisy environment and the noise during squealing can be extremely high. A typical output of a noise recording from a pass-by of a squealing vehicle is shown in Figure 2.7, where the noise due to particular eigenmodes that

are excited by local friction conditions is highly dominant over the other frequency components.

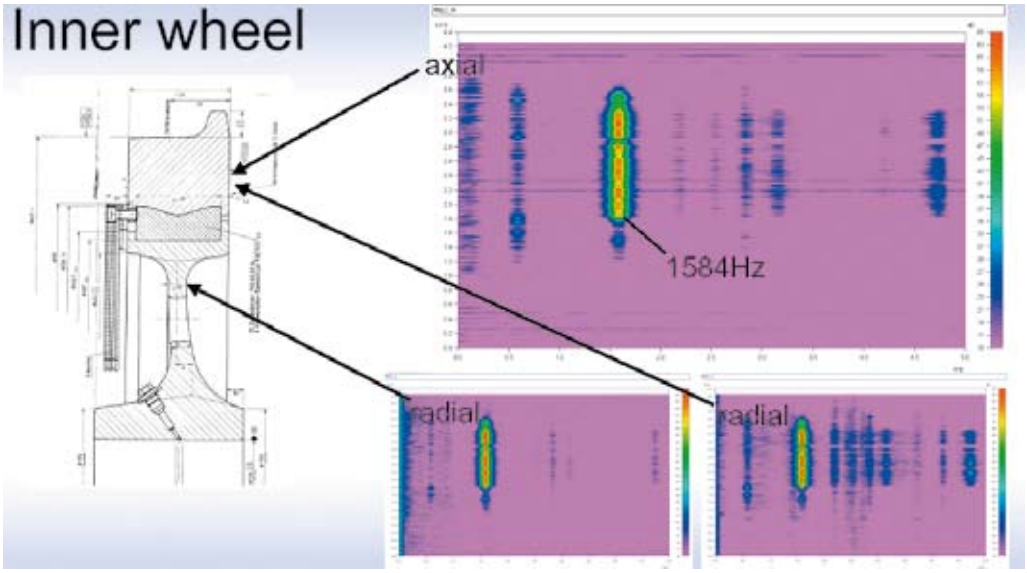


Figure 2.7 Squeal noise from a composite tram wheel. Despite the use of rubber elements between the web and the tyre and the application of an external damper, squeal is present at extremely high levels [5].

The description and the discussion of squeal noise is particularly difficult and the interested reader is referred to specific textbooks. Nevertheless, some basic concepts will be introduced here such that the possible countermeasures described later can be fully appreciated.

Generally speaking, a traction or braking force can be transmitted only if a certain *micro slip velocity* is present between the wheel and the rail. This micro sleep velocity, know as *creepage*, is always present during traction or braking. In a car travelling on a motorway at constant speed, the distance measured by an odometer mounted on motor wheels is always greater than the distance measured by an odometer mounted on trailer wheels.
The higher the torque to be transmitted, the greater the creepage is, until a limit is reached, the so called *adhesion limit*. After adhesion limit is passed, the friction coefficient decreases (Figure 2.8). The slope of the diagram, i.e. its first derivative, is equivalent to a damping characteristic.

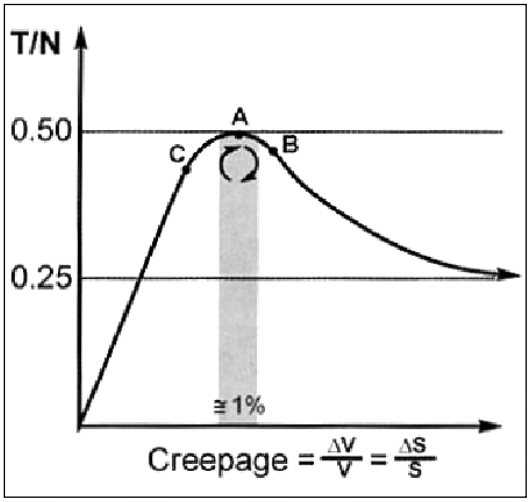


Figure 2.8 Friction characteristic of railway wheel-rail contact. Beyond the adhesion limit the coefficient of friction decreases, causing the so called "stick-slip loop" [5].

The generation of squeal noise is due to the *negative slope* part of the diagram that leads to a mechanism that is called *stick-slip*. There are some common experiences where stick-slip phenomena can be observed: a violin bow acting on a string, a windscreen wiper, chalk on a blackboard with a certain angle and pressure, a wet fingertip sliding on a glass edge. In all these cases the motion is a series of contact of the bodies (the *stick* phase) followed by sliding (the *slip* phase).
Conversely to continuous sliding, and anti wheel blocking systems (ABS) for cars are designed to avoid this, a slip phase is followed by a stick phase because the system has a certain *elasticity* in it.

The classical representation of a mechanical system fully reproducing the stick-slip phenomenon is shown in Figure 2.9. Starting from the initial resting position, the belt is started moving and the weight of the block is enough to let it travel coupled with the belt. When the force exerted by the spring passes the adhesion limit, the block starts moving back. As the *dynamic friction coefficient* is lower than the adhesion limit, the block goes back. Only when its speed returns exactly equal to that of the belt, adhesion is regained. The resulting motion is almost sinusoidal and lasts as long as the belt is kept in motion.

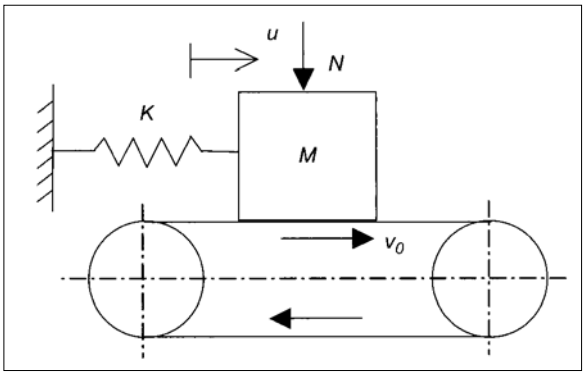


Figure 2.9 A simple mechanism where stick-slip occurs if the friction-creepage curve has a negative slope after the adhesion limit [10].

Adding damping to the mechanism can fully eliminate the stick-slip phenomenon only if it is larger than a certain value, below which it has almost no effect. This can be seen thanks to a complex mathematical model introducing the concept of *negative damping*, i.e. the region of Figure 2.8 where the slope becomes negative: in this region *the contact forces decreases as the creepage increases*. This mechanism is therefore injecting energy instead of dissipating it under thermal form. If the negative damping is larger than the natural damping of a wheel, stick-slip phenomena may occur. A *limit cycle* establishes, instead of an infinitely growing cycle, thanks to non-linearities of the adhesion coefficient curve. As any instability problem, it can be represented by transfer functions and verifying the stability by plotting the imaginary part of the open loop transfer function on the Nyquist plane, but this kind of analysis lies beyond the scope of this book.
For a railway vehicle, curve negotiation always happens with wheels having a non-radial attitude. This is inevitable for any rigid body vehicle with two wheelsets or for the wheelsets within a bogie for the classical two-bogie vehicle. The determination of the *yaw angle* (also said *angle of attack*), i.e. the angle formed by the wheelset axis with the radius of the curve, is rather complex and can be fully calculated also for a "quasi-static" curve negotiation (i.e. where transient forces are exhausted and the vehicle runs under the equilibrium of centrifugal force, weight and wheel-rail contact forces) only by means of simulation programs where creep forces are considered together with bogie elastic characteristics.

Railway curves are normally canted to reduce the effect of centrifugal forces on passengers. A particularly interesting special case is that of a vehicle running along a curve at a proper speed where the centripetal force due to the weight component resulting from track cant compensates completely the centrifugal force due to the curvature. Also in this case, where a point mass would run along the curve in steady conditions, a bogie has a number of contact forces due to the constraints between the wheelsets and the bogie frame, owing to the stiffness of the primary longitudinal suspension (also called *yaw stiffness*).

In this special case, it can generally be said that (Figure 2.10):

- the leading axle runs displaced toward the external part of the curve;
- the leading outer wheel therefore always runs with its flange in contact with the rail gauge face, with a relatively high rolling radius;
- the leading inner wheel travels on its tread (no flange contact) on a relatively low rolling radius;
- the trailing axle tends to run almost centred, or even slightly displaced towards the centre of the curve. The difference of the path run by the two wheels is much more compensated than for the front wheelset.

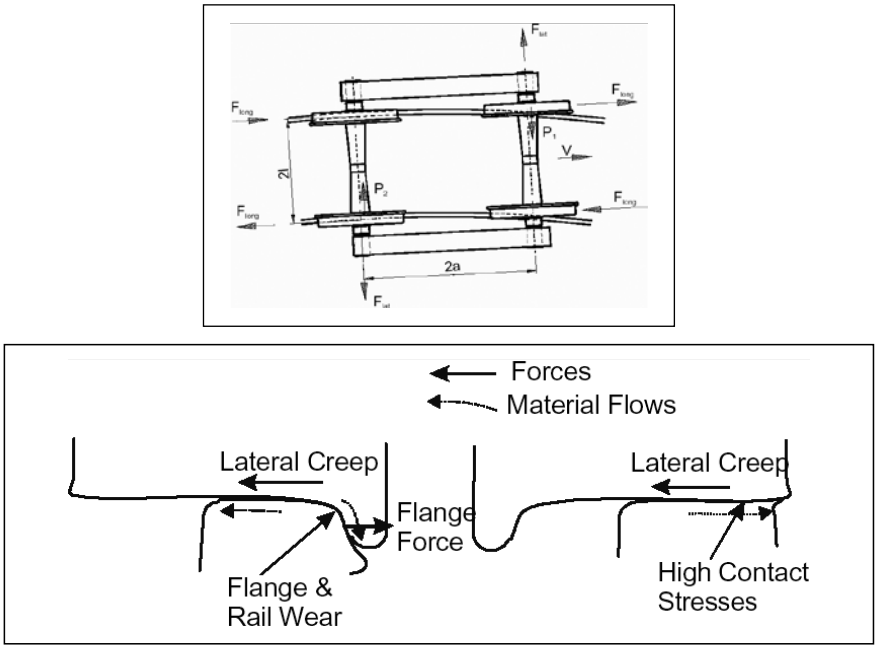


Figure 2.10 Top: bogie attitude in a curve run at the equilibrium speed (=no cant deficiency). Bottom: wheel-rail forces in the transversal plane [11].

This situation for the front wheelset can be changed by increasing the speed, resulting in a *cant deficiency* (or in a positive non-compensated centrifugal acceleration) or by adopting a particularly soft yaw stiffness. Unfortunately there are considerable limits of wheel and rail wear for the first case (kinematics forces are proportional to v^2) and stability problems in the second case. The conflict between steering and stability is well known in railway engineering and will not be further discussed here; let us only say that stability at normal to high speeds requires yaw stiffness that is so high that self-aligning of wheelsets under normal contact in a curve is almost negligible.

Concentrating our attention on the leading inner wheel where the common normal to profiles is almost vertical, the non-zero angle of attack gives rise to a lateral velocity component that is, in turn, converted into a contact force by the force-creepage relationship shown in Figure 2.8. This force cannot grow forever and will saturate the available adhesion, leading to possible stick-slip phenomena depending on local adhesion conditions, wheel damping and rail dynamics.
Generally speaking, the rail has a much lower mobility than the wheel, i.e. its surface velocity will be much lower than wheel surface velocity under the same force. This results in a wheel component much higher than that due to the rail.
It is worth underlining that the problem of squealing does not appear on the leading outer wheel. Wheel-rail contact angle on the wheel flange is much more inclined while on the inner rail it is almost horizontal; the contact point between wheel and rail is moreover displaced in front of the vertical plane passing through the wheelset, leading to a sliding speed and a resulting damping which is much higher.

As we have seen, the genesis of the squeal problem lies in the *lateral* dynamics of the wheel-rail contact; it is obvious that the phenomenon will appear when a lateral natural frequency of the wheel is excited. A solid wheel has a large number of natural frequencies (called *eigenfrequencies*) and of associated natural deformed shapes (called *eigenmodes* or simply *mode shapes*) that can be rather easily found by experimental or numerical techniques known as *experimental* or *numerical modal analysis*. The probability of coincidence of the excitation with one of these frequencies

is therefore rather high; considering furthermore that wheel damping is normally extremely low for lateral modes, it is easy to understand that once the stick-slip process has started it tends to "tune" on a very limited number of lateral eigenmodes, causing the extremely tonal squealing noise.

In the remaining part of this paragraph some measures to reduce noise will be described. Attention will be focused on the practical application rather than on theoretical possibilities. As squealing is a typical *unstable* phenomenon, a measure can be said to be effective only if it *completely eliminates* the problem, i.e. if after applying it the problem fully disappears.

Independently rotating wheels

The use of IRWs has become a common practice in many bogies of fully low-floor trams. The need to have a vehicle's floor at around 250 mm above street level prevents any possible use of a conventional wheelset with a solid axle. IRWs are therefore mounted on shafts protruding from the bogie frame or, preferably, on a so-called *dummy axle*, i.e. a steel forged frame connecting the two wheels of an "equivalent wheelset". The second solution allows for the introduction of a primary suspension giving at the same time good safety levels in the respect of the "equivalent" wheelset gauge.
The use of IRWs can be shown to be detrimental to vehicle dynamics as the "gravitational effect", i.e. the self-centring effect due to tapered wheels under the effect of the weight of the vehicle, is not sufficient to avoid the permanent contact of one of the wheel flanges of the vehicle against the rail gauge face. If this can be tolerated anyway for a tram, the effect on squeal is unfortunately equally absolutely negligible. The absence of the torsional constraints between the two wheels allows the complete cancellation of longitudinal creep forces that are often responsible for rail corrugation; unfortunately, as has been demonstrated, curve squeal noise is due to *lateral* creepage forces and to the consequent unstable limit cycle of vibration. As a result, most modern trams squeal exactly as their older predecessors.

Self or actively steering vehicles

Primarily to reduce flange forces and wear, a number of passively (self) and actively steering bogies were proposed. Cross bracing, wheelset orientation derived from carbody-bogie rotation, active yaw dampers and articulated asymmetrical bogies are only a few of the solutions developed. In principle, the advantages obtained by the use of these solutions must be carefully

evaluated in terms of the number of components, reliability, maintainability, safety and, above all, total life cycle cost of the solution.
To our knowledge, train manufacturers, never approached the design of a steering vehicle to prevent wheel squeal. It must be said that there are much more effective countermeasures.

Wheel damping

As rail contribution to squeal noise is considered to be limited, a number of potentially interesting solutions have been presented to increase wheel damping. As has been shown, squealing noise appears only if damping is below a given *threshold* or, conversely, squeal does not happens if wheel damping is artificially increased above the same threshold.

Solid (*monobloc*) and tyred wheels normally have a very low damping, as will be described in more detail in the chapters relative to rolling noise; roughly speaking, "standard" wheels sound like bells when hung up with a rope and hit by a hammer (trying to reach the *free-free* condition. i.e. a freely suspended wheel that is free to vibrate after an initial impulse). Composite wheels with rubber elements between the web and the tyre show rather different behaviour under these conditions, but nevertheless the size of the elements involved are such that squeal is not automatically prevented by the use of rubber. Durability of elastic elements impose a very light damping and, often, a high stiffness. These two rubber properties are such that vibrations are efficiently transmitted from the tyre to the web originating squeal noise anyway.

As mentioned in the case above, vehicles are rarely designed from the beginning with anti-squeal damped wheels. When it is the case, the wheel design can benefit from the insertion of the damping device from the beginning. It is in fact normal to discover that a certain vehicle displays squealing features shortly after it enters in service, i.e. when the vehicle has been homologated by safety authorities. Modifications to the wheel to fit damping devices may require a new homologation, with all the associated costs.

Some possible devices attached to wheels are:

1. a ring inserted into a groove machined under the wheel tyre (similar to elastic rings used to axially restrain the movements of bearings and other parts in conventional mechanical shafts);

2. a set of viscoelastic dampers tuned to several frequencies, mounted with screws and nuts in a "T-shape" circular groove;
3. "shark fins", i.e. a set of plates mounted onto a ring which is concentric to the wheel tyre and blocked to it with a number of bolts, normally made with two layers with or without damping material;
4. damping layer treatments, i.e. one or two plates (internal and/or external) glued onto the wheel web with the interposition of a high damping polymer.

Only treatment 4 can be retrofitted without requiring a new wheel homologation; on the other hand, only treatments 1 and 3 can be used with block braked wheels due to the limited resistance of polymer to high temperatures.

It is worth mentioning that wheel eigenmodes are highly affected by changes in the tyre height given by wheel reprofiling. A "broad-band" damper like solution 4 ensures its efficiency during the whole life of the wheel, even increasing its performances as the mass of the wheel decreases (and the damping ratio therefore increases), while a tuned damper may result very efficient on the wheel at a certain level of wear and behave poorly at some other wear stage. Data of wheel damping at different wear stages would be of great help to design tuned dampers.

Friction management

While treatment on wheels is effective in any case, it is justified only when extra costs associated with it are compatible with the resolution of the squeal problem. As already mentioned, squealing occurs in rather specific locations and, moreover, under not always repeatable conditions (it is commonly said that squeal is an "erratic" phenomenon). In most cases it would therefore be preferable, if possible, to treat a specific location, i.e. a given curve.

First of all it is necessary to mention that railway engineers have been using wheel flange lubrication since the early years of the railway age. Most of the steam locomotives had wheel flange lubricating devices and also nowadays any new locomotive has wheel flange lubrications systems. Squeal noise is due to friction *on the running table* of the rail and of the wheel, an area that is not used for guidance but for the transmission of longitudinal forces during traction and braking. While the friction coefficient in the wheel flange

area should preferably be as low as possible (zero in principle, but from a practical point of view in the order of 0.1), its value on the top of the rail should be in the order of at least 0.3 to allow full traction and braking performance of high power/mass ratio vehicles.

It is a common experience that rain has a positive effect on train dynamics in critical conditions. Derailments are due to *flange climbing* effect, which is promoted by a high friction coefficient; at the same time, wear is extremely high in dry and unlubricated environments (like many metros); squeal normally disappears when any third body is inserted at the wheel-rail contact. Braking performance is clearly affected by rain, but train distance and signalling in general has obviously been designed to allow for perfectly safe circulation under rainfall conditions.

Many administrations decided to use water where squeal noise is a critical problem and the reduction on adhesion does not lead to an inadmissible reduction in train performance. Unfortunately water is not without problems in extreme environments (ice formation) and in metro lines (where high humidity and water collection and removal may be problematic).

The use of other substances is therefore slowly entering into practice. Classical lubricants (oil or oil-based, like grease) are detrimental for adhesion, leading to both service (motor wheel spin) and safety issues (sliding during braking). Moreover, the too low adhesion coefficient may be critical for tramways, as pedestrians, cyclist and motorcyclist are expected to walk and to travel over greased rails. It is clear that also having small water pools in a city is not practical, although the proper combination with sand used in anti-spin devices may resemble the situation of a beach…

A large consensus has been obtained by the so called *friction modifiers*. A friction modifier is a water based substance that looks similar to a latex paint with the following interesting features:

1. the adhesion coefficient is reduced to levels that are considered optimal for traction and braking, therefore not affecting vehicle performance in normal service;
2. the force-creepage increases more and more, i.e. also for high creepage level the "negative damping" characteristics is never present, completely avoiding the stick-slip loop;

3. if applied on board with solid sticks, friction modifier consumption is limited as a thin film is readily established on the wheel tread;
4. the transfer to following wheels happens with the usual mechanism of lubricant transfer, guaranteeing a certain degree of protection for several vehicles;
5. when distributed with trackside lubrication equipment, the protection of the curve is immediately effective with low investment;
6. there is no danger for pedestrians, cyclists and motorcyclists as friction coefficient remains relatively high.

The drawback of friction modifiers is their purchase cost. In any case, the real efficiency of any measure must be evaluated considering the overall cost and not just the purchase cost. Two examples of dispensers of lubricant / friction modifiers are shown in Figure 2.11.

Figure 2.11 Trackside dispensers of lubricant are normally used to reduce the formation of corrugation but are also extremely effective against squeal noise [5].

2.3.3. Flanging noise

Flanging noise is quite often confused with squeal noise. It is due to the rubbing of the leading outer wheel flange on the rail gauge face. For the reasons already described, damping is sufficiently high to avoid the formation of limit cycles and squeal noise is not generated in this case.

Nevertheless, flanging noise can be a serious problem where the use of water based lubricants is reduced by frequent rain, where rust is present on the rail and where the wheel flange lubricating system is poorly tuned. This latter

case is unfortunately rather frequent as, after reprofiling with sensible wheel diameter reduction, nozzles are not always adjusted to correctly spray in the flange area.

A skilled acoustician should be able to recognise and distinguish the presence of flanging noise with or without the contemporary presence of squeal noise quite easily. First of all, with some luck, it should be recognized that squeal noise is generated by the *inner* wheel and that flanging noise is generated by *outer* wheels. With a grease pot and a brush, the high rail gauge face can be treated very easily for a test: the flanging noise should readily disappear.

Remedies to flanging noise are therefore related to the wheel flange lubrication equipment: use a grease of a more appropriate quality (for example with a high solid content), change the philosophy of grease spraying in order to enter the curves with wheel flanges properly protected by the lubricant, check the mechanical part of spraying systems (nozzles clogged or badly oriented). The use of solid stick can prevent most of the problems due to classical flowable grease-based equipment.

2.3.4. Ground borne noise and vibrations

This subject is complex and its comprehensive description is certainly not within the scope of this book. Some definitions are, in any case, useful to avoid further confusion on the role that the different factors (wheels, rails, superstructure, soil, building, etc.) have in the generation and transmission of these disturbances.

First of all, it is necessary to say that all the components, including soil, behave as elastic media. If this is clear for a wheel or for a sleeper, that have finite dimensions, the modelling of a rail or of the soil, that can be considered infinite, requires much care.

Engineers are used to dealing with "resonances", i.e. the natural frequencies (or *eigenfrequencies*) at which a finite dimension element tends to vibrate. Considering a prismatic bar, longitudinal, transverse and shear waves are transmitted from an excitation point toward the ends of the bar. At the bar ends the discontinuity gives rise to an *impedance mismatch*, and the waves

are almost fully reflected back. The phenomenon repeats until vibrations are damped out by internal damping of the material. The superposition of these waves increases the amplitude only in "resonance" conditions, i.e. at particular excitation frequencies.

A different situation happens for 2D-infinite (rails) or 3D-semi infinite (soil) spaces. In the case of the rail, energy is propagated by the input point by the rail that acts as a *waveguide*. Unless boundary conditions introduce an impedance mismatch, travelling waves never come back. The reader familiar with long and highly tensioned ropes (like those of ropeways), knows that hitting the rope with a fist leaves the input point unvaried after the impact. Only after the travelling wave has reached the end of the rope, it comes back to the input point. The whole process may take several seconds depending on many parameters that we will not discuss here.

Wavenumber-domain solutions are complicated to explain. Generally speaking it can be said that any wave has a *propagating* part that remains unaltered and a *decaying* part that reduces its amplitude with distance, converting elastic and kinetic energy into heat.

In infinite elastic media, two fundamental wave speeds can be defined, related to the compressional (and dilatational) and shear wave motions. Surface propagation of waves happens for half-space structures, like soil. Rayleigh waves are characterized by the fact the soil particles perform elliptical motions; as Rayleigh waves have the lowest speed they are therefore very efficient to transport energy also for very long distances. For soft soils, typically found in the upper layers of the ground, Rayleigh waves speed can be in the order of 100 m/s, a speed that can be reached by modern high-speed trains. In this case, a train reaches a *sonic* condition, i.e. the vehicle travels at the same speed as the waves that it produces in the elastic medium that it is travelling in. This condition determines the so-called *sonic wall*, a regime where amplitudes and stresses are exceptionally high. As this case is rather complex and applicable only to very soft soils, it will not be described further.
For the scope of this book, vibrations induced by railways can be divided in two classes:

1. vibrations at a very low frequency range, let's say up to several ten Hertz, are generated by heavy freight trains running at moderate speeds and transmitted to considerable distances depending on the soil properties.

These vibrations may induce damage to buildings and reduce comfort therein. They are perceived as very low frequency rumbling and felt by vibrating objects (tables) or parts of the house (floor, walls). Classical "amplifiers" of these vibrations are glass display cabinets, where crystals services may clink loudly. This phenomenon is what should be properly called *groundborne vibrations*.

2. vibrations at higher frequencies, let's say in the frequency range of 20-250 Hz, are felt by humans as noise. Surface velocity is in fact turned into noise following a formula where time- and space-averaged squared velocity is directly proportional to sound power. In this case, vibration is not perceived by touching a vibrating surface but rather by listening to noise that the surfaces emit. This noise, known also as *re-radiated noise*, should properly be defined as *groundborne noise*.

A sketch of the general situation is shown in Figure 2.12.

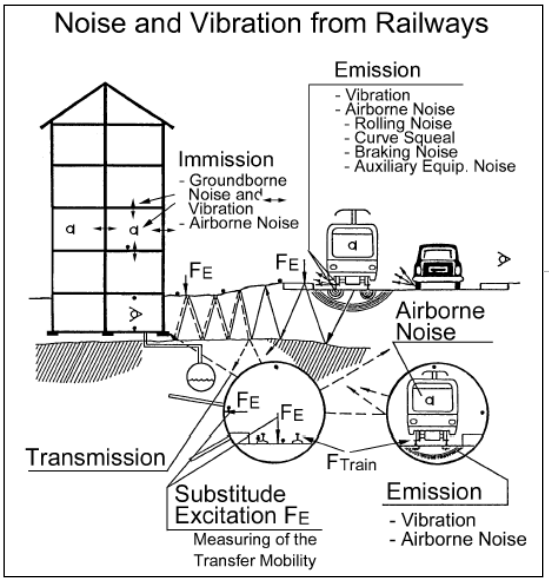


Figure 2.12 Principles of groundborne noise and vibration generation, transmission and emission in a building [12].

The indicators to quantify the disturbance and the possible damage to buildings are found in literature and in many national and European Standards and will not be discussed here. It is only interesting to say that measures against high groundborne vibration levels are somewhat limited. It is extremely rare to carry out a preliminary soil characterization to prevent the problem from the onset, for example by installing trenches, buried walls or the so-called *Wave Impedance Blocks* (or WIB).

On the contrary, measures against groundborne noise can be efficiently taken at track design level. The standard design for metro tunnel was, in the past, for example, the fastening of rails to the tunnel invert with a two-level elastic suspension by using elastic rail pads and elastic baseplates. Unfortunately, this fastening system can not efficiently cut low frequencies as it has to be sufficiently rigid to avoid premature damage to elastomers and to keep gauge constant against gauge spreading forces.

Generally speaking, to effectively cut the lowest possible frequencies, track formation should resemble a mass-spring system with a very low natural frequency. In the last few decades the track design for the metro system changed considerably. The best solution is the so-called *floating slab track*, where track support is made of relatively thick (around 0.5 m) and heavy concrete plates of different lengths (2 to 10 m depending on the design), discretely or continuously supported by elastomers. Very effective insulation properties can be obtained by such large masses resting on soft supports. Further schemes can be observed in Figure 2.13.

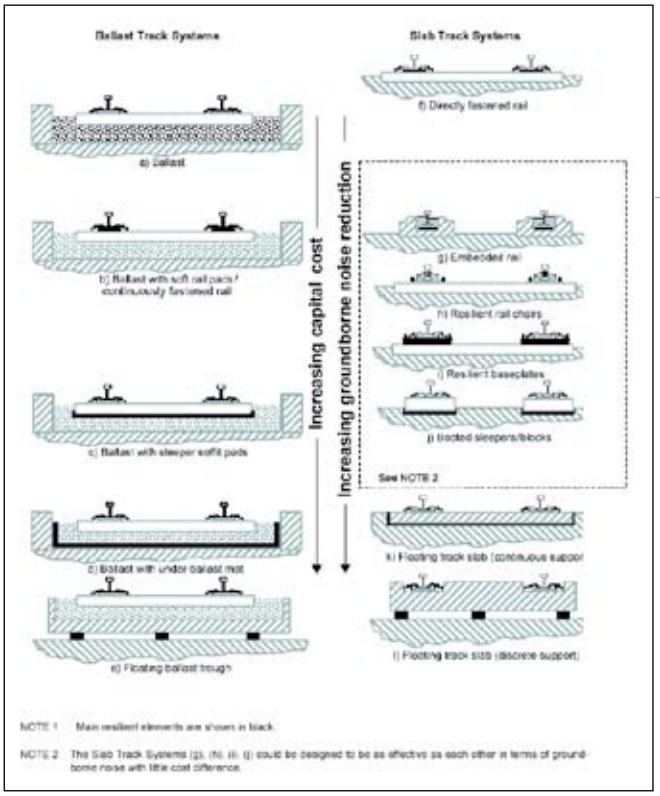


Figure 2.13 Track formation for ballast and slab systems with indications of cost and efficiency [12].

For further information the interested reader can refer to the specific literature on the subject.

2.3.5. Aerodynamic noise

Also this subject is complex and its comprehensive description is well beyond the scope of this publication. It can be said, in general, that aerodynamic noise starts to be dominant at around 250 km/h depending on local features of the specific train. It increases much faster than rolling noise, with a typical dependency on speed like $60*\log_{10}(v)$, i.e. on the 6^{th} power of the train speed.

A great importance on aerodynamic noise generation are the following features, with an order of importance that depends on the specific train:

- bogie shape and size;
- pantograph number, position and size;
- streamline nose shape;
- gaps between coaches;
- any irregularity on the side (handles, doors, windows, grilles, etc.)

Aerodynamic noise is normally dominated by components below 500 Hz. Estimates require the use of CFD (Computational Fluid Dynamics) numerical codes and are in practice impossible due to the size of the train and the need for a sufficiently fine discretization mesh. Tests in appropriate wind tunnels can help to intervene on models, while field measurements on high speed trains require the use of antennas on microphone arrays in order to "follow" and identify each source.
For further information the interested reader should see the specific literature on the subject.

2.4 European noise reduction policies

Before proceeding to analyse the possible noise reduction measures, it is interesting to describe what's been found in some European funded projects about railway noise reduction, namely the projects STAIRRS

and IMAGINE. The results are very briefly summarized in Figure 2.14 and in Figure 2.15, where it is shown that the measures at the source have the lowest cost/benefit ratio.

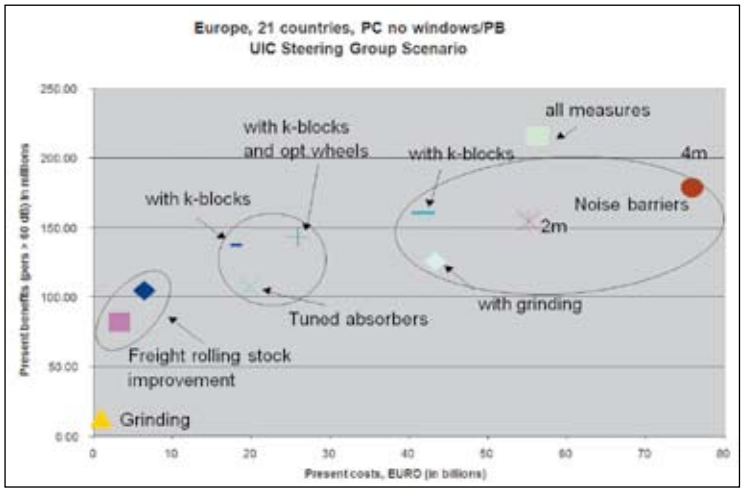


Figure 2.14 Cost-benefit chart from the EU funded project STAIRRS [6].

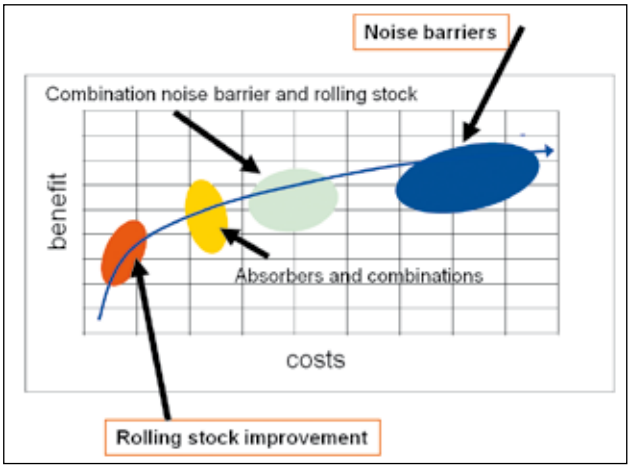


Figure 2.15 Simplified version of the same chart, showing the types of interventions directly [6].

In recent years, people's tolerance of noise barriers has decreased. They represent a physical and visual obstacle, sometimes damaging a nice landscape or even obstructing the view of people living on the ground or the first floor. The European Union requires today that railway noise must be tackled at the following levels:

1. first of all, any attempt must be made *at the source*, i.e. designers, engineers, politicians, those responsible for the environment and authorities *must* try to avoid *noise generation* as much as possible;
2. after this has been done, and only as a remedy to those situations where measures at the source proved to be insufficient, it is possible to act *on the transmission path*, i.e. installing noise barriers;
3. if even the use of noise barriers is not enough and the noise level on the façade of a particular building is too high, the last measure is to act directly *on the receiver*, for example changing windows of wall insulation.

Despite these declarations and requirements, the use of barriers is still preferred for many reasons that will not be discussed here. The chapter on legislation will give a deeper description of the political situation in Europe and of the most recent legal tools available to public local administrations to limit the number of complaints from citizens living on their territory.

As the European Union, in its legislation frame, expressly requires tackling railway noise at the source, leaving measures on the transmission path and on the receiver only as a final remedy, probably also the person in Figure 2.16 could benefit from noise reduction at the source!

Figure 2.16 A rather striking example of the visual impairment caused by a motorway noise barrier [6].

3 ROLLING NOISE: WHAT CAN BE DONE?

3.1 Introduction

In the last chapter a number of noise types were described and some general lines of intervention were depicted for some "less common" noise types, like squealing noise, impact noise, flanging noise and ground borne noise.

It is a matter of fact that the vast majority of people is affected by rolling noise in open air. In par. 2.2 a general description of how rolling noise is generated was given, along with the basic laws of variation with speed.

In this chapter the details of all the factors involved in rolling noise generation are discussed. The aim of this chapter is not to provide a theoretical basis to perform numerical simulations or to make sophisticated measurements, it is rather to show, at a sufficiently easy level, the problems related to *railway noise control*.

Acousticians never talk in terms of noise *reduction* but rather in terms of noise *control*. Controlling something means that the fundamental laws have been understood, that available measures on the market are known and compared, that cost analyses have been performed to avoid waste of time and money. An acoustic consultant should be able to provide reliable indications on the steps to follow to approach the specific noise problem and to find out the *optimum* solution in terms of *efficiency*. Sometimes the situation is complicated by the fact that the constraints are not completely clear, and that even the goal may also not be clear.

The basic question that everybody asks when approaching railway noise: "Is it the wheel or the rail?". Research in this field has led to the conclusion that *both* the wheel and the rail (or, more appropriately, the track) are responsible for noise in different ranges of speed and of frequency. This may explain why many attempts to reduce railway noise have been so frustrating: under the hypothesis that both of the sources contribute equally to overall noise, eliminating in principle one of the sources leads to a reduction in overall noise of barely 3 dB. Some examples of noise prediction taken from the literature obtained by using simulation packages are shown in Figure 3.1.

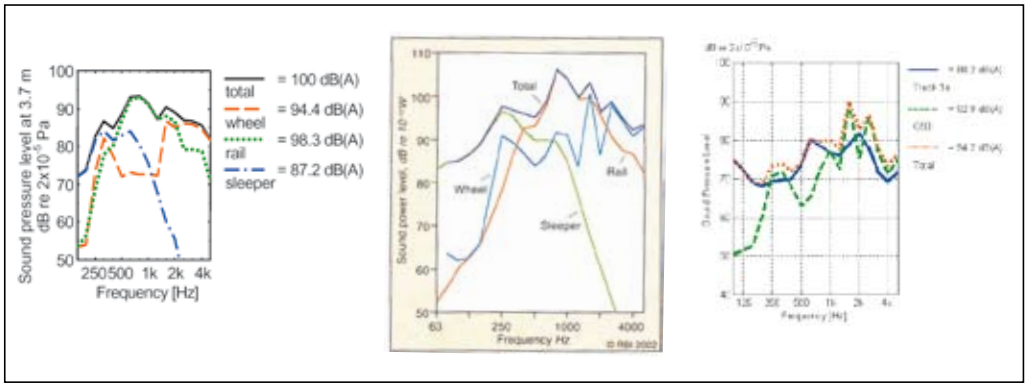


Figure 3.1 Some examples of calculation of wheel, rail and sleeper contribution to total noise. In general, rail noise is dominant at lower frequencies while wheel noise is dominant at higher frequencies. Sleeper noise is normally important below 250 Hz, i.e. at even lower frequencies ([1], [13], [1], from left to right).

Generally speaking, the relative contribution of the wheel and the rail may depend on speed, while the spectral distribution of the sources is rather different:

- the track, i.e. the combination of sleepers and rails, emits particularly at low frequencies, that are harder to be controlled (see chapter 4 on noise barriers);
- the wheel has a greater efficiency and importance at higher frequencies, as wheel resonances are normally in that region;
- at low speeds, rail noise is normally dominant. This is particularly important when crossing city suburbs at relatively low speed;
- at high speed, wheel noise is normally dominant. Small villages close to high speed line can therefore benefit more from wheel treatments.

There are, therefore, two *crossover* regions, which must be carefully evaluated case by case but for which, in general, the following can be cited:

- the frequency at which wheel emission becomes dominant over rail emission: the crossover frequency is normally between 1 kHz and 2 kHz;
- the speed at which wheel noise becomes dominant over rail noise: the crossover speed is often in the order of 120 km/h.

In any case, it must be said that in the specific field of railway noise many steps have been taken in the last few years to give more options to technicians and

politicians that want to do something different from what has been done in the past, when the only solution was the erection of noise barriers.

3.2 Wheel and rail roughness: how to act on contact interaction

The first parameter considered in rolling noise generation mechanism is "roughness". It is important to say that the concept of roughness known in the mechanical industry is far from its meaning here. Amplitudes of interest for wheels and rails may be of several hundred microns and wavelengths can range from mms to some meters, depending on the train speed and on the frequency of interest. If, for example, 31.5 Hz are of interest for a train running at 300 km/h, the "roughness" must be investigated at least up to a wavelength of 2.6 m. It is clear that roughness metres available in workshops are absolutely useless to characterize such irregularities.

Wheel roughness and rail roughness are certainly uncorrelated and, as there is no relationship, it could look logical to sum these quantities to get the total roughness. The situation is slightly complicated by the fact that wheels and rails are continuous bodies following elasticity rules. Contact mechanics is a complex science, where Hertz, Kalker and Johnson, just to mention some, have given fundamental contributions. Without entering the difficult mathematics describing these phenomena, it is often said that the contact area between the wheel and rail, with usual loads, curvature radii and Young's modulus, is an ellipse that has approximately the area of a coin (the situation is rather more complex in curve where also the wheel flange contacts the rail gauge face).

From an absolutely qualitative point of view, it looks reasonable that what's shorter than the ellipse dimensions is not "felt" by the wheel and the rail. In other words, elastic deformations are acting as a high pass filter in the wavelength domain, in the sense that roughness wavelengths shorter than the typical contact patch area dimension are sensibly attenuated. The *contact patch filter* therefore reduces the total level of roughness to $r_{tot}=r_w+r_r+r_c$, where r_w and r_r are the wheel and rail roughness spectra and r_c is the attenuation spectrum offered from the contact patch filter (Figure 3.2). It is evident that short wavelength defects can be tolerated much more than long wavelength defects.

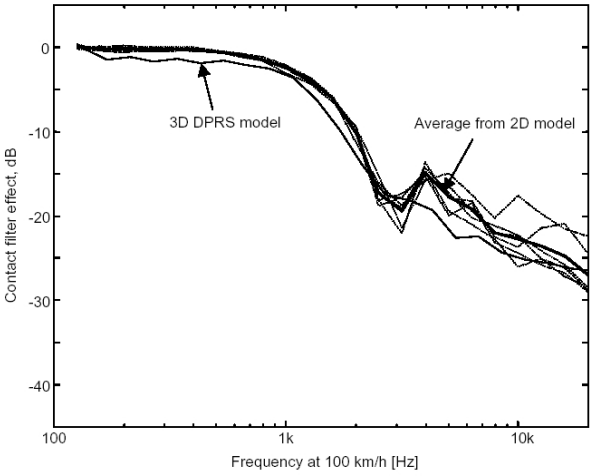


Figure 3.2 Spectrum of the attenuation offered by the contact patch filter calculated by different techniques. Wavelengths can be obtained in this case by the relationship λ=(100/3.6)/f. For example, the 1 kHz frequency corresponds to 27.7 mm (attenuation ≈ -2 dB), while the 3 kHz frequency corresponds to 9.3 mm (attenuation ≈ -20 dB) [14].

It will be shown that rail roughness values are somewhere between the levels of a block braked wheel and a disc braked wheel: in the first case it is said that *wheel roughness is dominant*, and little advantage can be gained by further improving the quality of rail surface; in the second case *rail roughness is dominant* and polishing the rails accurately may lead to a remarkable reduction in rolling noise.
As will be shown, recent developments with modern brake pad materials allow the bar to be raised, reducing further wheel roughness, while the application of modern rail grinding techniques can dramatically reduce rail roughness.

In the chapter about legislation it will be shown that the indicator chosen by the European Union is a weighted average of the noise emitted during the day, the evening and the night:

$$L_{den} = 10 \lg \frac{1}{24} \left(12 * 10^{\frac{L_{day}}{10}} + 4 * 10^{\frac{L_{evening}+5}{10}} + 8 * 10^{\frac{L_{night}+10}{10}} \right)$$

Freight trains have a noise emission of 8 to 10 dB(A) higher than passenger trains due to tread braking, and the weighting function introduces a further 10 dB penalty for nightly emission, when disturbance is higher while people are sleeping.
These two conditions are a large obstacle to the other policy of the European Union, i.e. the increase of railway traffic shifting freight from lorries to freight

trains. As long as slower freight trains can travel only during the night for line congestion, this modal shift will hardly be possible. The adoption of synthetic brake blocks (that leads to a tread roughness lower than that of disc braked wheels) is therefore the only possibility to increase freight traffic noticeably. The situation is extremely fluid, but in any case specific public funding will probably be necessary if an effect is desired in the short term. By the usual renewal rate of freight trains, whose life is in the order of several decades, and under the hypothesis of starting to manufacture new wagons only with synthetic blocks, it could take 40 years before the full advantage is obtained and, due to the logarithmic nature of noise, it could take more than 20 years before a 3 dB(A) reduction is reached.

3.2.1 Wheel roughness and wheel maintenance

Dealing first with wheel irregularities, it must be said first of all that noise engineers have little or no control over it once the braking system is defined.
Even after a perfectly circular reprofiling, wear starts to introduce irregularities in this perfect shape. As already seen, the result of this wear are known as *out of roundness* (OOR). In any case, due to the limited length of the wheel circumference, the OOR profile is periodical.

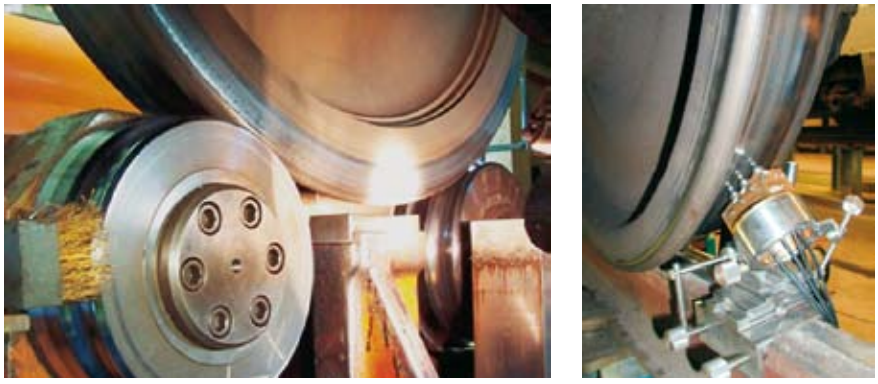


Figure 3.3 Machining to restore a worn profile is normally done by using special machine tools (under floor lathes, left). During some research activities the OOR of the wheels has been measured by using transducers similar to those shown on the right [5, 6].

Before dealing with noise, a short preface is necessary to describe how braking affects environmental pollution.

Historically, railways originally had tread (or block) braking. Brake blocks, also called pads or "shoes", are made of cast iron, respecting the principle of having a different hardness (and resulting wear) from the other elements of the friction pair, i.e. the wheel (made of steel). Cast iron blocks have specific features, like a rather high weight, limited cost, easy supply and a peculiar friction coefficient dependency on the sliding (vehicle) speed. As coefficient of friction decreases with speed, the braking system of tread-braked vehicles has been designed to exploit the capabilities of such material at the best. As is clear, kinetic energy is converted into heat at the wheel tread – brake block interface, leading to high temperatures reached during braking. Wheelsets were designed with wheels with curved web in order to accommodate large radial displacement of the wheel tyre due to thermal expansion.

Tread braking is by far the least expensive braking system, including one or more brake cylinders (two-axle vehicle normally have one brake cylinders, vehicles with bogies typically have one cylinder per bogie), some leverages with bars, trusses and beams, and brake blocks. Some self-adjusting devices to recover from block wear were developed since the early years of the railway era in order to limit compressed air consumption and to speed up the intervention of the brake blocks as much as possible. A photograph of a shoe and a shoe-holder is shown in Figure 3.4. To have an idea of the tests that any braking pair (wheel-shoe) must withstand, Figure 3.4 also shows a snapshot taken during a braking test at a test rig. Temperatures of several hundred °C are normally reached during these tests.

Figure 3.4 Classical arrangement of a brake block on a freight wagon (left). Braking tests on a rig are normally used either to assess the properties of blocks or to validate freight car wheels (right) [5].

Unfortunately, tread braking with cast iron blocks rapidly leads to high wheel roughness or, more generally, to wheel tread OOR. Complex thermo mechanical phenomena of interaction between the cast iron brake block and the steel wheel tread are such that a set of "hot points" form during long braking, leading to a progressive *polygonization* of the wheel tread (Figure 3.5).

Figure 3.5 A heavily polygonized freight car wheel [6].

Incidentally, this characteristic of cast iron braking was believed in the past to be an important feature to prevent wheel sliding and spinning. As anti-blocking systems have been developed only in recent years, it was important to avoid wheel blockage during emergency braking and to such goal the high roughness "guaranteed" by cast iron blocks was welcome.

Only in the last few decades the use of disc brakes has become common in the railway industry, and in different proportions depending on the vehicle and on the application:

- passenger cars are nowadays all equipped with disc brakes, having a larger braking power with two, three or even four disks mounted on the axle;
- locomotives have been equipped with disc brakes over the past few years, mainly for problems linked to limited space due to the presence of electric motors and the transmissions (the most common solutions are discs mounted on the transmission shafts or on the wheel web);
- EMUs and DMUs are moving towards the same direction, i.e. the use of sector discs mounted on the wheel web;
- freight cars still resist with the original solution of cast iron brake blocks.

It is worth noting that some countries adopted in the period 1950-1970 vehicles

with "mixed-braking", i.e. a combination of disc braking (around 80% of the braking power) and tread braking (the remaining portion). Railwaymen call these blocks "cleaning blocks", suggesting one the reasons for their adoption, i.e. the achievement of a sufficient roughness level to guarantee adhesion.

From a dimensional point of view, the peak-to-peak amplitude of these (typically periodical) OOR can reach 60-80 μm for the cast iron tread braked vehicles, while a disc braked wheel has an OOR stochastically distributed normally contained in a band of 10 μm.

All this results in a much lower noise emitted by a disc-braked vehicle compared to a cast iron block braked vehicle: this difference is normally in the range of 8 to 10 dB(A). It is a common experience nowadays to recognize by the noise which type of train is passing by: freight trains are much noisier than modern passenger rolling stock with disc brakes.

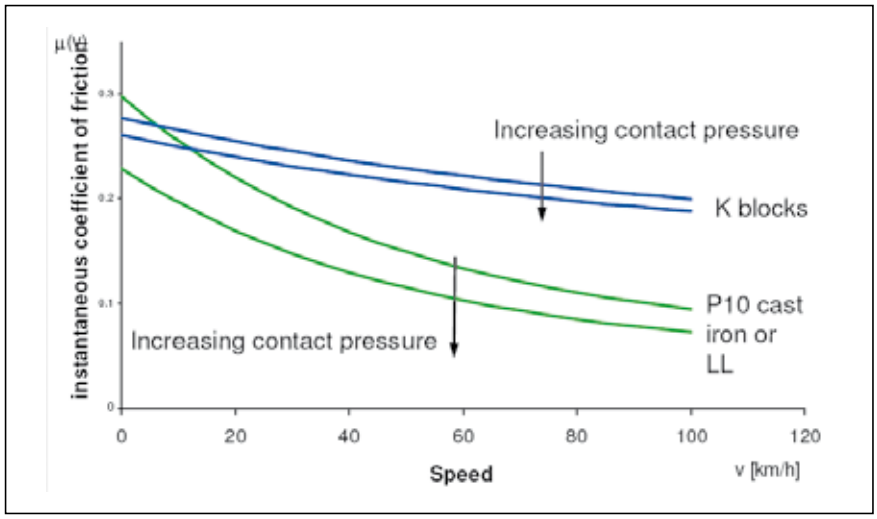
As an evident and straightforward way to reduce wheel roughness, the adoption of disc braking on new rolling stock has been proposed. This turned out to be unfeasible as the freight car is as much as inexpensive as possible (consider, for example, that it normally has no electrical equipment at all), and the adoption of disc braked wheel sets would have led to unacceptable high costs and to maintenance procedures that are incompatible with the normal railway practice.

An alternative that was tested was the use of drum brakes, i.e. a combination of disc brakes (in the sense that the braking surface is not the wheel tread but is a specific one) with cast iron blocks. In any case modification to the wheel set and to the braking system is extremely onerous. Although results were (rather obviously) good from the acoustical point of view, the experiment didn't lead to a mass application of this solution.

The only possibility to reduce noise while keeping the existing wheelsets and (more or less) the same braking system is offered by brake blocks made of different materials, namely sintered and synthetic material. These materials were classified by UIC in three categories (Figure 3.6):

- "K" blocks, with a high coefficient of friction with speed dependency that does not copy the behaviour of cast iron. This type of block requires a modification to the vehicle braking system;

- "L" blocks, with a low coefficient of friction, for possible direct use where the block braking system is not the primary one (i.e. as a replacement for cast iron on locomotives and dual-braked systems used for passenger vehicles);
- "LL" blocks, with a coefficient of friction with behaviour very similar to cast iron. These blocks can be used as a direct replacement of cast iron brake blocks.



3.6 Coefficient of friction for different brake shoes material according to UIC classification

The replacement of a standard and well known material with a completely different one has not been free from "teething" problems (see Figure 3.7) [5].

Figure 3.7 Some examples of the now solved initial critical problems of composite blocks [6].

The visual appearance of tread braked wheels where composite shoes are used is particularly appealing (Figure 3.8); roughness measurements indicate that the surface quality is even better than that of disc braked wheelsets (Figure 3.9).

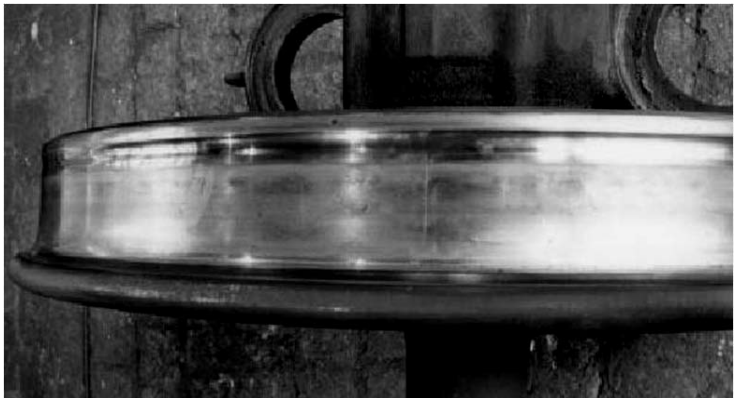


Figure 3.8 Visual appearance of a wheel tread braked with composite shoes [6].

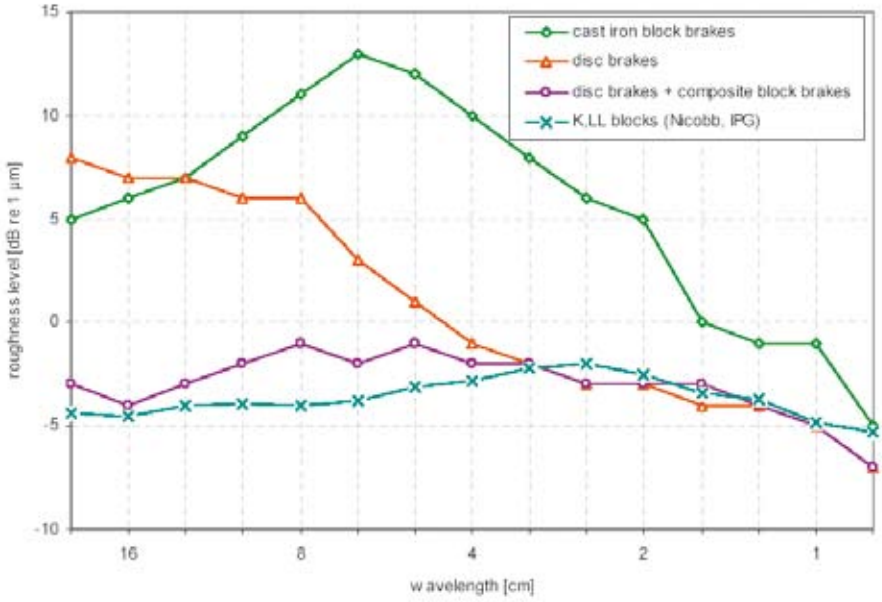


Figure 3.9 Wheel roughness obtained by using different tread braking solutions. The advantage that is obtained by using composite brake shoes on disc braked wheel sets is evident [5].

In the next paragraph the surface quality of rails is analysed in detail; we can anticipate here that the roughness of composite blocks braked wheels is of the same level as rail roughness that can be obtained with state-of-the-art acoustic grinding techniques (Figure 3.10). As a result, the combination of wheel roughness and rail roughness obtained with respectively composite shoes braking and special grinding techniques is much lower than all the other existing combinations and can lead to noticeable reduction of rolling noise (Figure 3.11).

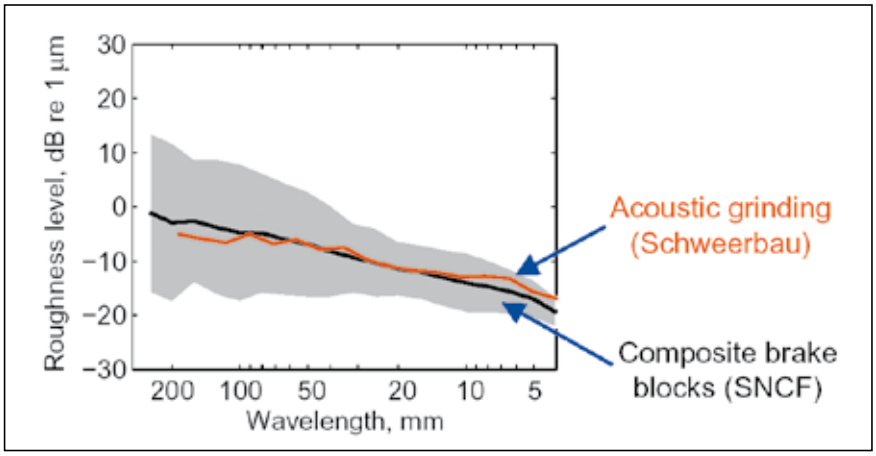


Figure 3.10 Comparison of rail roughness obtained with acoustic grinding (red line), average (black line) and distribution of wheel roughness obtained with composite brake blocks (grey shaded area) [5].

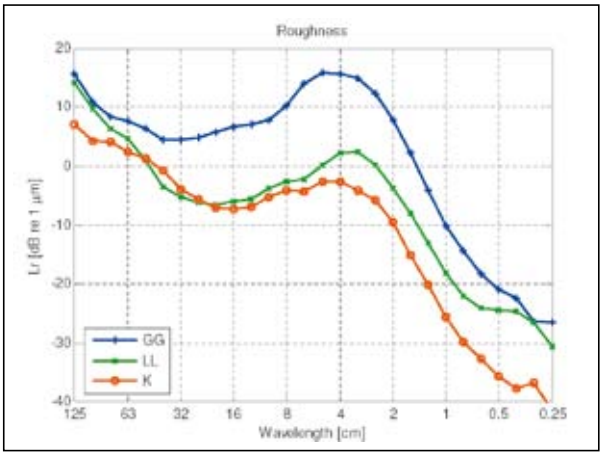


Figure 3.11 Combined wheel-rail roughness (including the effect of contact patch filter) for different braking technologies. The use of K-blocks results in a combined roughness consistently lower (more than 10 to 15 dB) in most of the wavelength bands [5].

3.2.2 Rail roughness and rail maintenance

Rail level irregularity, as it should be probably more appropriately called, may have distinct features. Generally speaking, a track in "good conditions" has a rail with a few visible features on it. Rail surface appears bright and uniform.

On the contrary, a track in a tight curve very often has a "low rail" (tracks are normally canted) that shows some periodic visible irregularities, producing a classical "roaring" noise when the trains pass above it. This special case of rail level irregularity is called *corrugation*, and is an endemic problem that happens in most of the railways in the world depending on a number of parameters. Corrugation often has a rather precise period, i.e. a constant and largely prevailing wavelength, generated by a so-called *wavelength-fixing mechanism*.

In the case of an uncorrugated smooth rail, no specific wavelength components are normally found in the irregularity spectrum. In this case, scientists talk in terms of *asperities* or, as it has nowadays become common, *rail roughness*. The absence of tonal terms implies that rail roughness is a *wide band physical phenomenon*. As a system responds at the frequency at which it is excited, normal rail roughness is in practice capable of exciting all the eigenmodes of a railway wheel. The subject will be dealt with in more detail when discussing wheel response and measures to reduce the response.

As it is recognized that rail roughness level plays a fundamental role in noise generation, especially in presence of disk braked vehicles with low roughness, how rail roughness can be controlled in practice is interesting. Before dealing with this subject, it should be discussed how roughness is measured in practice. It should not be forgotten that rail roughness that has importance for noise generation *is not visible* and that two apparently equal tracks can result in external noise during pass-by that differs by even more than 5 dB(A). Experience is therefore not a good tool to establish in advance if a rail is rough or not: amplitudes of a few microns can not be estimated by visual inspection.

Rail roughness is the result of the interaction of rail surface with the

wheels of trains passing over the rail. Wear can be uneven leading to rail roughness higher in one section than in other section; normal maintenance procedure, including *rerailing*, may reset the story of a part of a line. It is therefore impossible to give a reasonable estimation about the rail roughness status of an entire line by sampling only some locations.

This leads to the problem of rail roughness measurement, as normal laboratory equipment can work on a few cm measuring length and measuring trains do not have the proper sensitivity to measure very low roughness levels (*submicron*, i.e. with amplitude often $r < 1$ μm).

It can not be forgotten, anyway, that noise problems are normally located in quite confined areas: a hospital, a school, a group of houses. In all these cases the track length is not tens of kilometres but a few hundred meters. Short measurements can be done manually with portable lightweight devices, of which a certain variety exists on the market. Quite recently, a "road test" was done within the frame of a European funded project to assess the practical applicability of a protocol for rail roughness measurements to be used for new rolling stock type testing. Although not the primary scope of the road test, it was found that all the equipment used for the test gave the same (from a statistical point of view) result on a prescribed rail section.

Two different families of equipment are available: fixed distance (around 1.2 m) instruments and trolleys. The vertical profile can be measured with contact or contactless displacement sensors; LVDT, inductive proximeters and accelerometers are used in the various solutions. Where accelerometers are used, the profile is obtained with a double integration process.

Beyond all other differences, including portability, weight and software capabilities, it appears evident that fixed length equipment have strong limits on maximum measurable wavelength and that the measurement of even some ten meters may become a complicated task. On the other hand, trolleys are sensitive to walking speed. Some examples of instruments available on the market are shown in Figure 3.12.

Figure 3.12 Some examples of available equipment to measure rail roughness (top: fixed-length equipment, bottom: carts) ([5] top, [15] bottom left]).

The only way to keep a line or, in principle, an entire network under control without major disruption to normal service is to use the principle of an *indirect rail roughness measurement*.
As all indirect measurements, rail roughness is not directly measured but it is estimated from the measurement of another physical quantity. The outcomes of roughness are high vibration level on the wheel set and, as we have seen, high noise level. It is clear that both of these properties are also affected by wheel roughness, and that's way vehicles where transducers are placed to estimate indirectly rail roughness are disc braked or, better, not braked at all.

Axle box vibration measurement is a rather common and popular method to try to estimate rail surface status. Nevertheless, many transfer functions are involved between the rail surface and the axle box vertical vibration: the response of the wheel set, the response of the axle box case, the behaviour of the roller bearings. Experiences in the world have shown that correlation between axle box acceleration and rail roughness is not so straightforward.

German Railways DB adopted a different philosophy more than ten years ago. They took a passenger coach, removed the braking system from one of the bogies, removed a toilet, conditioned from the acoustical point of view the enclosure obtained, and suspended a microphone over a hole drilled in the coach floor (Figure 3.13). A correction with speed was used in order to compare results from track sections run at different speeds.

Figure 3.13 Interior of the DB measuring coach. The microphone in suspended to isolate it from car body vibrations and located in the former toilet area over the bogie to collect rolling noise [5].

Special grinding is planned where noise level exceeds a certain value. This technique, named Specially Monitored Track (Besonders Überwachtes Gleis, BÜG in German), is based on the definition of the average level of noise, an intervention level (3 dB higher than the average) and an expected level after grinding (3 dB lower than the average). Figure 3.14 shows the levels and the philosophy of the method, where grinding is followed by regular monitoring with the BÜG car that allows to plan in advance the next grinding operation.

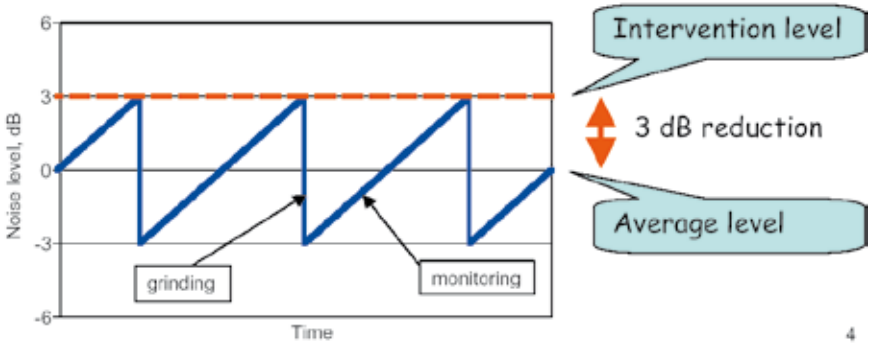


Figure 3.14 Principle of application of the Specially Monitored Track technique used in Germany [5].

The Specially Monitored Track is supported by a special grinding technique named *shuffle grinding* or *oscillating block grinding* (or simply *block grinding*). The visual appearance of a corrugated rail and of a block ground rail are compared in Figure 3.15, where the difference is evident. As shuffle grinding is rather different from conventional grinding; some details are given in the following.

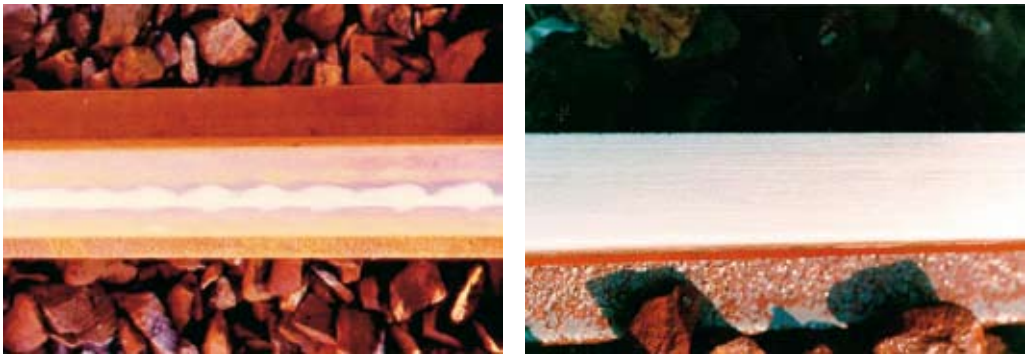


Figure 3.15 A corrugated rail (left) and a rail ground with oscillating stones in the frame of the Specially Monitored Track activity (right) [16].

As is widely known, grinding is the only available technology to remove defects on the running table of rails *and* to reprofile the rails where the transverse profile doesn't meet quality requirements for several reasons (increased equivalent conicity, flange angle too low, presence of heavy rolling contact fatigue phenomena, etc.).

"Classical" grinding operations are based on the use of special vehicles (*grinding trains*, Figure 3.16) equipped with a certain number of rotating stones. The number of stones has a direct impact on productivity, normally expressed in meters of finished track per hour, and on the number of *passes* that are needed to obtain the desired results. The transverse profile is therefore restored thanks to accurately planned metal removal, resulting in a set of *facets* that define, with a certain degree of approximation, the desired profile. The axis of the stone lies in the vertical plane normal to the track axis (Figure 3.17).

Figure 3.16 A high productivity grinding train with 48 rotating stones. Sparks and dust are a common feature of this machinery and both dust suction and water spraying are used to reduce pollution and risk of fire.

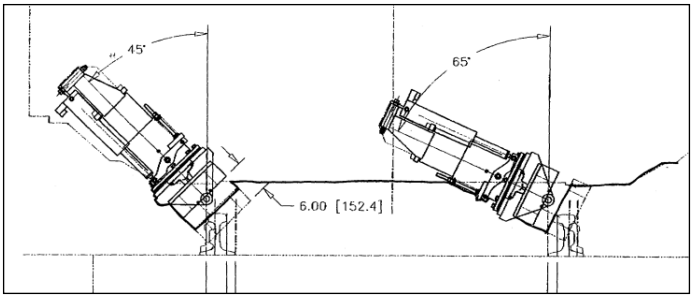


Figure 3.17 A grinding stone with its electric motor (left) and a sketch of how different stones are oriented to obtain the desired reprofiled rail facets.

Operations with conventional grinders leave a residual roughness after grinding which is linked to kinematic parameters of the machining (grinding train speed, rotational speed of grinding stones). A ground rail shows, therefore, some scratches with a spacing of 20 to 30 mm that are perfectly compatible with safe operation of the rail but that, as will be shown later in this chapter, may affect noise performance.

Block grinding is based on a reciprocating mechanism where a set of grinding stones are mounted in the position of the "piston". The crankshaft is moved by an electric motor and the rod makes the stones oscillate around a central position.

Figure 3.18 A block grinder (left) and the oscillating equipment (right).

Figure 3.19 Close-up of the reciprocating mechanism of a block grinder (left), a grinding block (centre) and the arrangement of multiple blocks, resulting in an equivalent longer rigid block length (right).

The main advantage of block grinding is that both feeding and working speeds act in the same direction, while conventional grinding stones have the two motions orthogonal. A shuffle block leaves almost no residual roughness (Figure 3.20). The main disadvantage of shuffle grinding is that reprofiling is not possible --- blocks wear assuming the shape of the rail.

This is not a drawback at all for new rails, electing block grinding as the best technique for preventative grinding on newly laid down rails. Defects of longer wavelengths may be corrected by block grinding, largely reducing the problems of impact noise due to bad geometry welds.

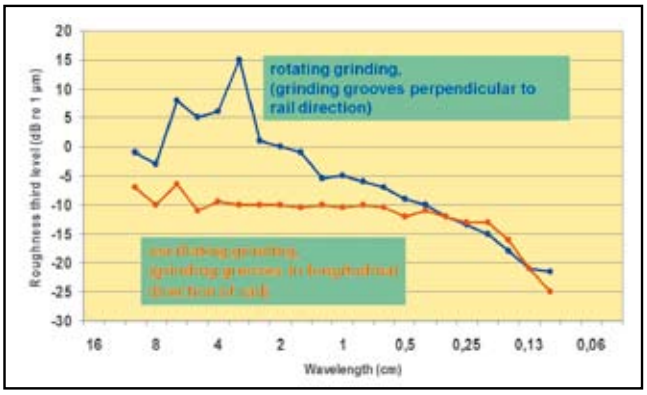


Figure 3.20 Residual roughness after rotating grinding (blue line) and after block grinding (red) [6].

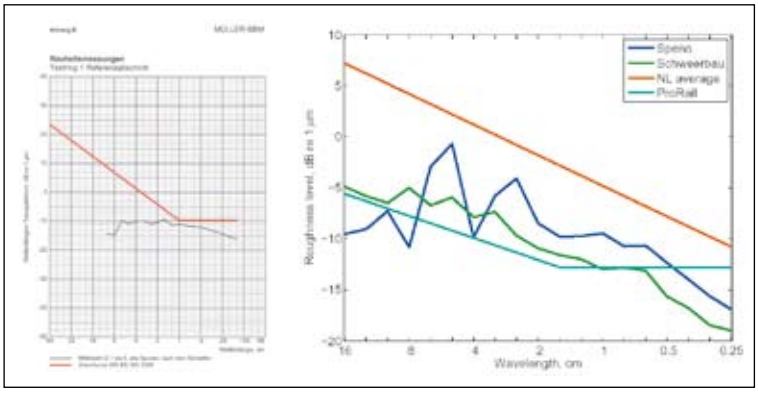


Figure 3.21 Residual roughness after block grinding performed by different companies compared to current rail roughness limit included in standards for external noise train type testing measurements (right) [6, 5].

One of the advantages of obtaining very low roughness after grinding is that the roughness growth process is not linear. If the initial roughness is very low, the growth will be particularly slow preserving the results for a long time and requiring fewer grinding operations and expenses. Two examples of such results are shown in Figure 3.22.

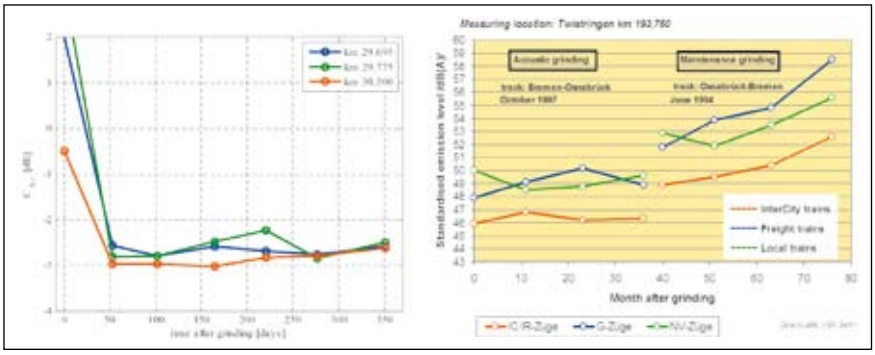


Figure 3.22 Roughness levels in time of block ground rails (left). Noise emitted by different categories of trains running on block ground and conventionally ground tracks (right) [5, 6].

The greatest advantage of block grinding can be obtained only with the simultaneous use of composite block braking. Calculations to get a quantitative evaluation must be conducted by properly combining the wheel and rail roughness and the effect of contact patch filter. Figure 3.23 shows that a real advantage can be obtained only by using *both* acoustic grinding and composite block braking. This means that wheel roughness from disc braked wheelset is *dominant* over acoustically ground rail roughness: the adoption of composite block braking could lead to further dramatic improvements in noise reduction.

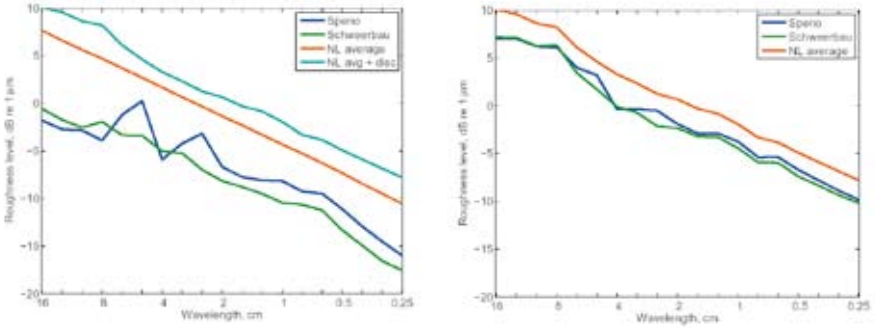


Figure 3.23 Calculated combined roughness for the combination of block grinding and disc brakes wheel sets (left) and for the combination of block grinding and tread braked wheels with composite blocks (right) [6].

A third grinding option is offered by *offset grinding*. In this technique the grinding stone axis is not perpendicular to rail axis but *skewed* with respect to it. Stones work on the lateral face rather the on the front face, wearing out conforming to current rail profile (Figure 3.24).

Figure 3.24 Offset grinding in operation (left) and a comparison of the visual appearance of a conventionally (darker surface) and offset (brighter surface) ground rail (right).

Compared to block grinding, offset grinding leaves a set of marks along the rail, potentially resulting in a higher noise level (Figure 3.25). A comparison with conventional grinding shows, nevertheless, that this residual roughness has a peak corresponding to a wavelength in the order of 6 to 8 mm, where the contact patch filter has a large effect. This contribution is then much better tolerated than the classical 20÷30 mm peak of the residual roughness left by conventional grinding (Figure 3.26).

Figure 3.25 Block ground (left) and offset ground (right) rails [6].

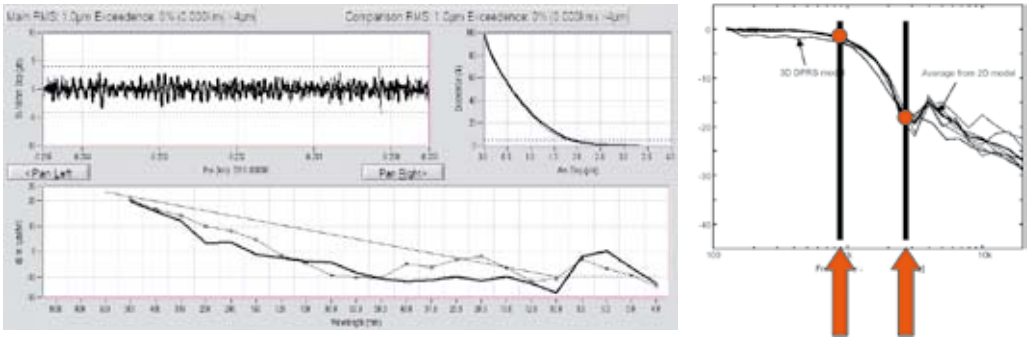


Figure 3.26 Left: comparison of residual roughness after conventional (thin line) and offset (thick line) grinding. Amplitude and statistical distribution are almost identical but frequency content is rather different, with a peak at 20 mm for conventional grinding and at 6.3 mm for offset grinding. Right: estimated attenuation for a 30 mm peak and a 10 mm peak of residual roughness.

A last grinding technique deserves the attention of railway technicians: the so-called *High Speed Grinding* (HSG) where the basic principle is the use of small grinding stones skewed by a large angle (Figure 3.27).

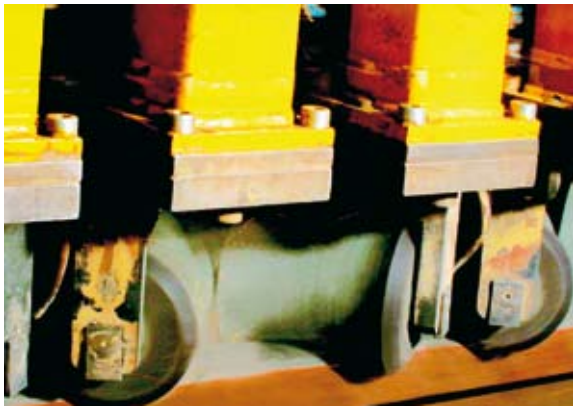


Figure 3.27 Grinding wheels of the HSG train.

The kinematics principle is shown in Figure 3.28, where it is clear how a component of peripheral speed of the grinding stone (V_S) results in sliding. Very low material removal can be done as grinding train speeds up to 80 km/h. It will be interesting to see in the future if this technology will be able to give the same interesting acoustic properties as, for example, shuffle grinding.

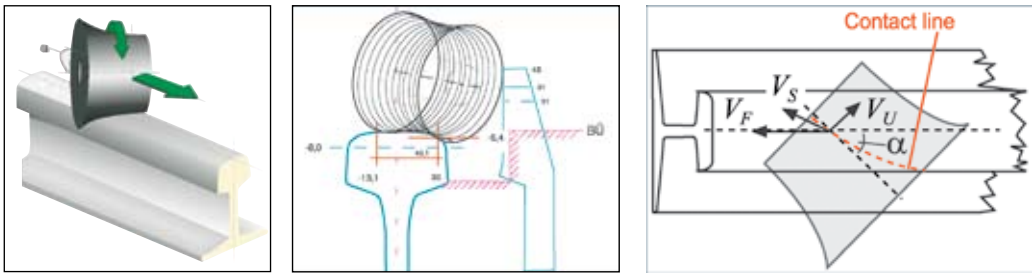


Figure 3.28 Kinematics principle of operation of HSG.

3.3 Wheel-rail contact point geometry: a parameter out of control

It is a common experience to observe different rubber tyres for our cars that have much different noise generation. The research to a quieter sculpture has led to particularly quiet tyres while preserving the obviously necessary safety against aquaplaning and ensuring durability and drivability.

The same thing could be thought of in principle for railway wheel tread. It is evident in this case that neither the material nor the details of the surface can be changed – a wheel is always a wheel, after all.

Nevertheless, in order to "soften" somewhat the contact point to reduce excitation to wheel and rail, different transverse geometry have been simulated. The basic principle is to smooth as much as possible the contact by acting on the curvatures of the bodies in rolling contact. Clearly, wheel radius can not be changed for noise purposes, and also railhead crown radius cannot be modified. The only parameter that can be adjusted somewhat is the transverse profile of the wheel.

Studies have shown that also in the most favourable conditions the improvement that can be obtained is limited. These studied focused only on acoustic-related issues, while wheel-rail coupling has a fundamental influence on running dynamics and safety. In any case, it was shown that this way does not lead to any interesting potential improvements of noise emission. Not surprisingly, higher noise levels observed during pass-by of trains with "hollow" wheels (i.e. wheels with concave tread to wear instead of the initial convex tread) were confirmed by theoretical studies.

The concentration of forces at the side ends of the contact patch leads to higher excitation.

In any case, therefore, contact point geometry is not a control parameter for noise emission for clear operational reasons.

3.4 Wheel design for low noise

The other way to reduce noise emitted by the wheel is to design it differently. Some parts of the wheel – the interfaces to the rest of the world, like the tread profile and the hub – are fixed, while the shape and size of the hub appears to be a promising area of investigation.

Early attempts started from the concept of decoupling the part subjected to wear (the tread) and the web, similarly to what happens in normal tyred wheels, but with insertion of rubber elements with elastic and damping properties (Figure 3.29). Unfortunately these elastic wheels, that have been adopted largely on trams and metro vehicles, have strong limitations for use on railways. One of the worst accidents in the modern railway era happened in 1998 near Eschede, in Germany, when a ICE train derailed after failure of an elastic wheel. After that episode, it can be said that there are no trains running at high speed with elastic wheels. Similarly, no freight trains can be equipped with elastic wheels of this kind for thermal reasons due to tread braking and subsequent damage of the rubber elements.

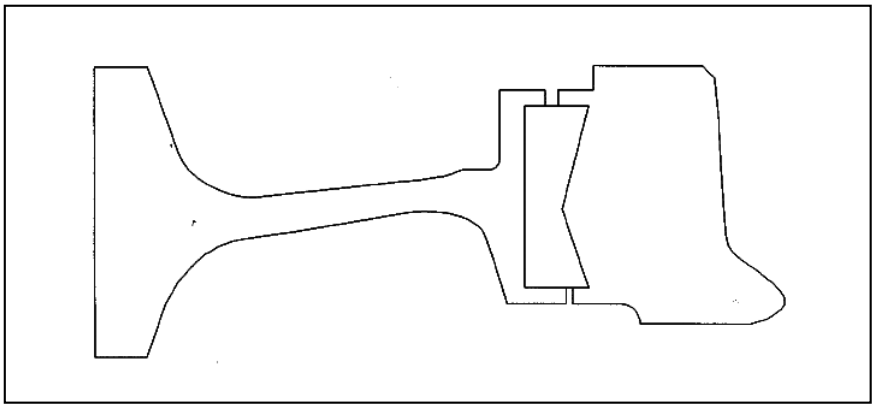


Figure 3.29 Sketch of an elastic wheel with rubber elements between tyre and web [17].

The concept of damping with a constrained layer is at the basis of various design of damped wheels. Some of them were successful for squeal noise reduction while others are now currently available for speeds largely in excess of 300 km/h. The basic principle is well known in acoustic engineering and is based on the damping of natural frequencies by energy dissipated by a polymer inserted between the vibrating body and a constraining added plate (Figure 3.30). Bending of the wheel web produces shear deformations in the viscous polymer that are dissipated into heat with consequent reduction of amplitude at the resonances. Textbooks on noise control largely discuss properties and basic theory of constrained layer damping.

There are several advantages in using this kind of technique that will be described in the last part of this book. The main drawback is that also this treatment can be used only on disc braked wheel set for the same problem of viscoelastic material thermal damage.

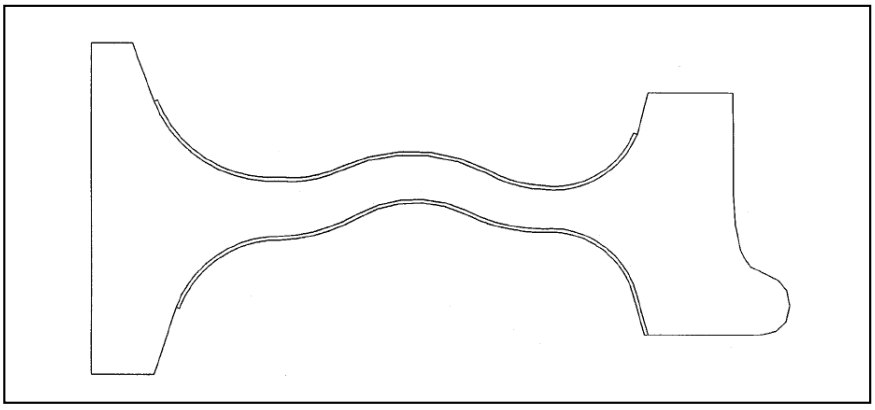


Figure 3.30 Sketch of a constrained layer damped wheel [17].

Some attempts were made in Italy when the very stiff double corrugated web of the prototype ETR500 was substituted with a new generation of particularly light wheels. Some sketches and photographs are shown in Figure 3.31 and in Figure 3.32.

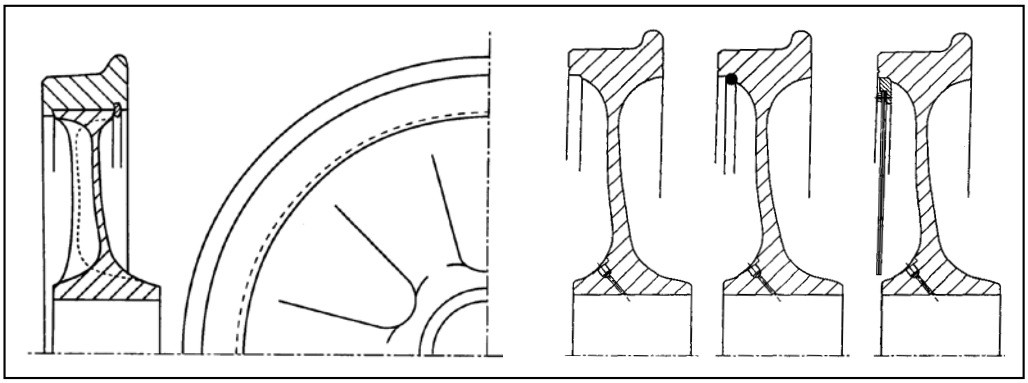


Figure 3.31 Original design of wheels with double corrugated web for ETR500 Italian high speed train (left). Several solutions tested in the early '90s to reduce noise of new light ETR500 wheel (right) [18].

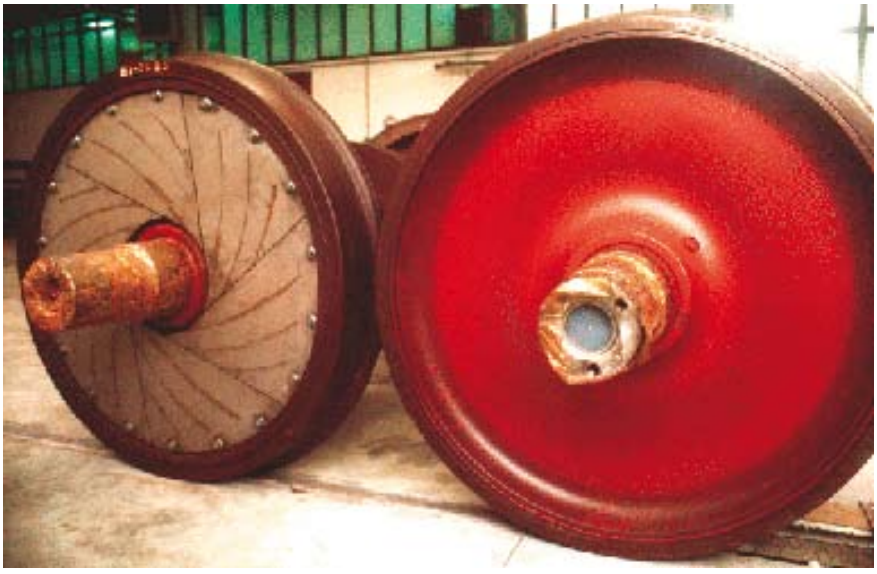


Figure 3.32 Wheel damped with layered and damped "shark fins" (left). Wheel with ring damper (right) [18].

Other sources of data are those reported regarding the Lok2000 locomotive built by former SLM (Figure 3.33). It can be seen how most of the measures have limited effect on rolling noise.

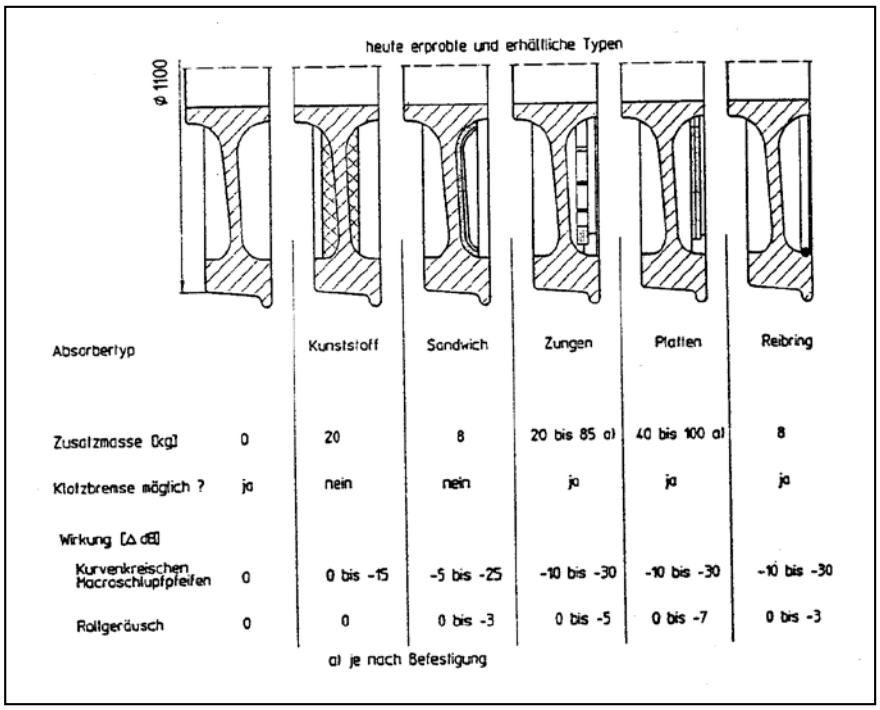


Figure 3.33 Noise reduction obtained with several reductions on Lok2000 locomotive [19].

A particularly interesting review of the research in this particular sector in the last two decades can be found in a recent paper by Thompson and Gautier, to which the reader is referred for further details [20]. Almost all possibilities were explored at a research and testing level, leading to particularly interesting results, among which we can briefly cite:

- wheels with acoustically optimised web shape and size (Figure 3.34). They consist of a straight web with a proper increase in section close to the tyre, changing eigenmodes and radiation properties. The increase in mass was around 40 kg and the overall noise reduction was in the order of 2 dB (around 5 dB in the 1600-5000 Hz frequency band where wheel noise is dominant);
- wheels with tuned absorbers (Figure 3.35). The wheel noise component was reduced by around 5 dB. A different design reduced noise by 1-1.5 dB. In any case the design was not compatible with thermo mechanical requirements for use with tread braking;

- composite wheels, made of an aluminium web and a steel tyre, with or without tuned absorbers (Figure 3.36). They resulted in the same mass as the original "full steel" wheel with a reduction of 4.5 to 6 dB of wheel component noise depending on the speed;
- wheels optimised from a mechanical point of view (very light) with screens with a special mounting arrangement that increased wheel damping (Figure 3.37). They resulted in a reduction of the wheel component of noise of approximately 5÷6 dB from 150 to 300 km/h;
- perforated wheels (Figure 3.38), obtaining by machining a wheel centre in order to create a sort of an "acoustic short circuit". No noise reduction was observed.

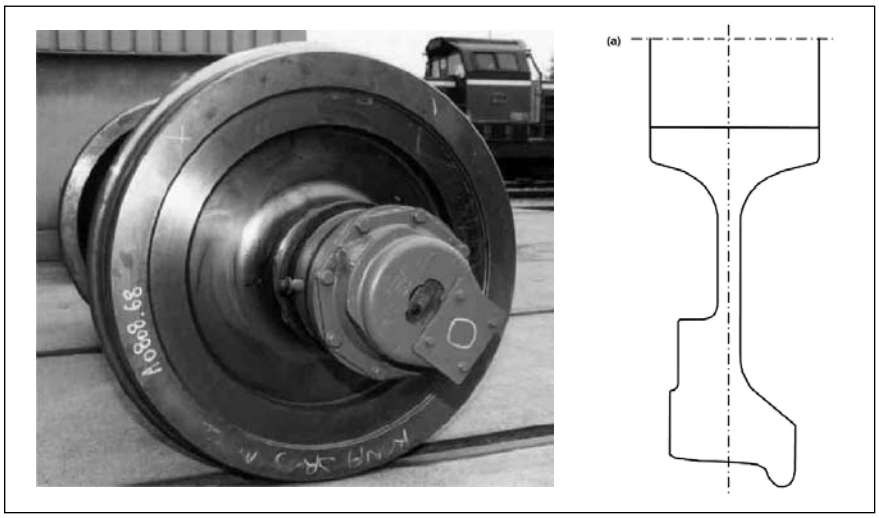


Figure 3.34 Wheel with optimised web shape [20].

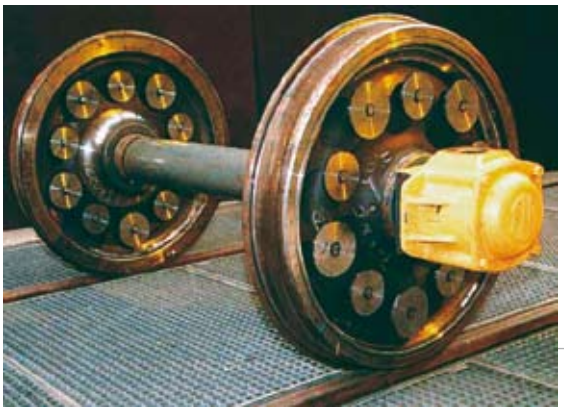


Figure 3.35 Wheels with tuned absorbers [6].

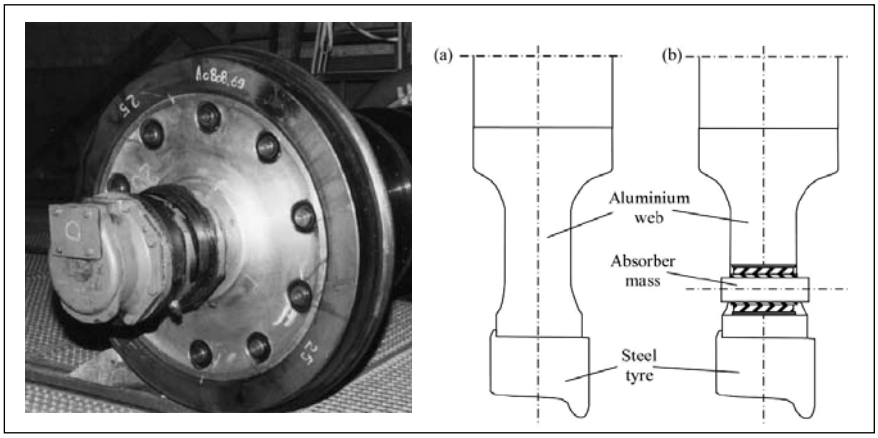


Figure 3.36 Composite wheel with aluminium web and absorbers [20].

Figure 3.37 Optimised wheel with noise screens [20].

Figure 3.38 Perforated wheel [6, 20].

Another project that is interesting to cite is the one developed by DB in the early years of this millennium [21, 22]. The aim of the research was the

design of a low residual stress monobloc wheel to be used on freight cars which was at the same time "optimised" from an acoustical point of view. The acoustical study (Figure 3.39 and Figure 3.40) led to a rather peculiar web shape.

Although the noise abatement results from line tests were promising (even if lower than expected), the most difficult condition to fulfil was the constraint on residual stresses after braking at the brake bench.

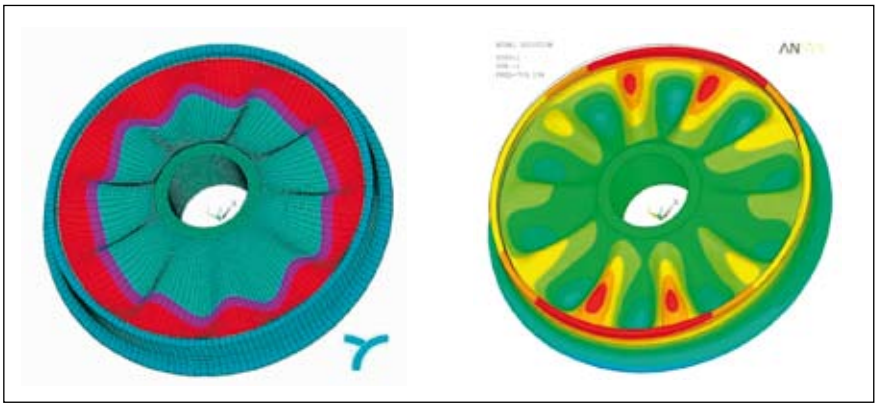


Figure 3.39 FE 3D model of the double corrugated DB wheel (left) and eigenfrequency at 709 Hz (right) [21].

Figure 3.40 Low noise prototype monobloc wheels for block braked freight wagons [22].

Although none of these research projects led to commercial products, the value of these researches is of the utmost level, showing practical limits of noise reduction measures applied to wheels. Currently available commercial products are:

- ring damped wheels. This is a classical solution to avoid squeal noise (see the chapter on this subject) but have no or very little effect on rolling noise;

- wheels with damping devices (constrained layer, tuned absorbers) (Figure 3.41 and Figure 3.42). In this case it is necessary that the damping provided by the dampers be greater than the damping freely supplied by the wheel/rail contact. Thermo mechanical problems affect this type of wheel too. Overall noise reductions can be up to 5 dB. Wheel damping with friction (full metal) devices, insensitive to high temperatures, developed at a research stage, will be shown in the last part of this book.

Figure 3.41 Wheel damped with multilayered dampers

Figure 3.42 Wheels with constrained layer damping

3.5 Track design for low noise

In a bottom-up description approach, a track formation is identified by the following elements:

- either a ballast bed or a concrete slab;
- either concrete sleepers (laying on ballast bed) or (possibly) some "sleeper-like" elements (*booted sleeper*, *Cologne's egg*);
- a rail fastening system, with one or more elastic levels;
- the rail

To the author's knowledge, the only rail specifically developed to reduce noise was the SA42 manufactured for the STV project in The Netherlands in the '90s (Figure 3.43).The applications of this design appear to be rather limited.

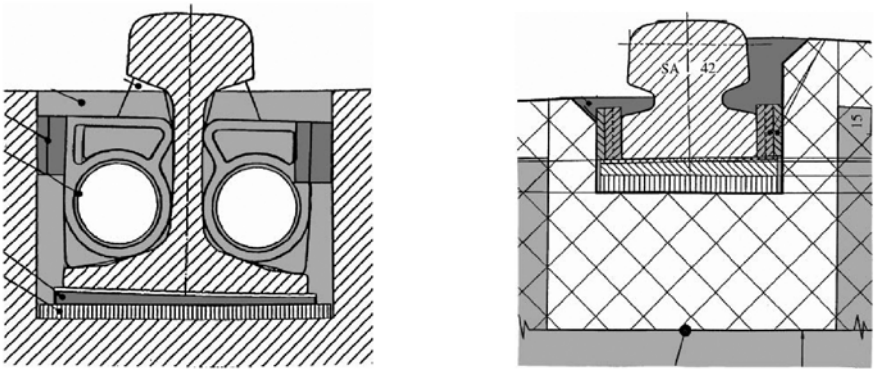


Figure 3.43 Comparison of a classical Vignole rail (UIC 60) with the SA42 rail [23].

Other tests by using "saddle rails", made of an aluminium cover on a reduced section standard rail (Figure 3.44), and rails with coatings (laser cladding) were mainly designed for reducing noise squeal.

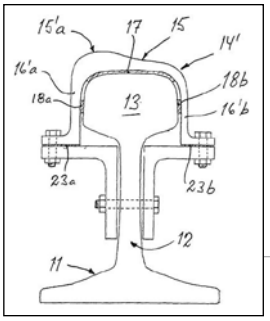


Figure 3.44 Sketch of the composite "saddle" rail (excerpt from the patent), designed to reduce squeal noise [24].

Another type of rail, named *BBEST*, was developed in the UK by Balfour Beatty Rail Technologies as an element for a complete replacement of the conventional track (Figure 3.45).

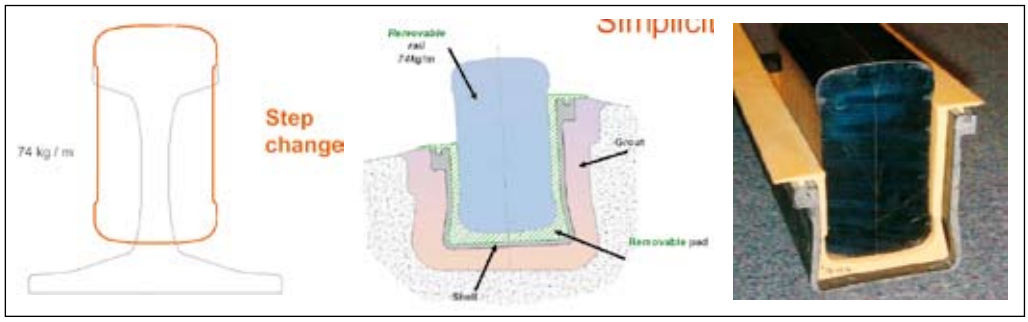


Figure 3.45 Rail section (left), sketch of application on concrete slab (centre), components (right) [25].

Noise behaviour of BBEST track is reported in Figure 3.46.

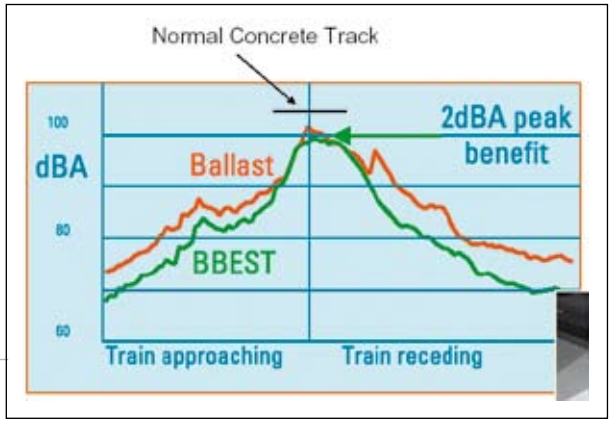


Figure 3.46 Noise behaviour of BBEST track [25].

The choice between sleepers on ballast and concrete slabs depends on many factors and, overall, on the philosophy that each country wants to espouse. Germany and Japan, for example, have decided to use slab track, while France and Italy prefer to go on with conventional tracks. This is certainly not the place to analyse and discuss the two systems, it will suffice to say that, typically, slab track is noisier due to concrete noise reflection and to higher rail noise emission. While the former can be dealt with using special coverage of the surface (Figure 3.47), the latter is due to the need for softer rail pads to compensate for the absence of ballast flexibility.

Figure 3.47 Different types of slab an ballasted track coverage with absorptive panels may be used to reduce both noise emission and noise reflection (especially from the concrete slab) [25].

This leads to the analysis of the influence of track radiation properties. We have already seen that the rail can be regarded as a waveguide that carries the energy far away from the contact point with the wheel. The softer the pad, the longer the vibrating portion of the rail is that can convert mechanical energy into noise (Figure 3.48).

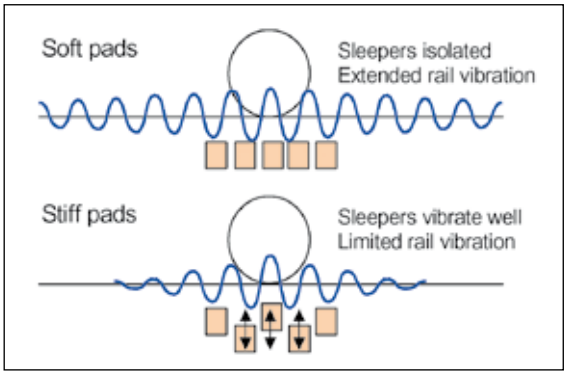


Figure 3.48 The length of rail interested in vibrations strongly depends on rail pad dynamic stiffness [5].

The assessment of the "effective length" of vibration of a rail can be made with the concept of *track decay rate*: the rail is excited by a known input (an impact hammer with a load cell) at different locations and the response is collected at a number of points along the rail (Figure 3.49). Signal processing leads to the evaluation of the decay rate measured in dB/m. The higher this decay is, the lower the emitting portion of the rail, with positive consequences on emitted noise.

Figure 3.49 Measuring the dynamic response of the track with roving instrumented impact hammer and accelerometers glued onto the rail to derive track decay rates [5].

To reduce rail noise, acousticians therefore would like to have rather stiff pads (with a stiffness in the order of 500 MN/m) in order to get sufficiently high vibration reduction along the rails (Figure 3.50).

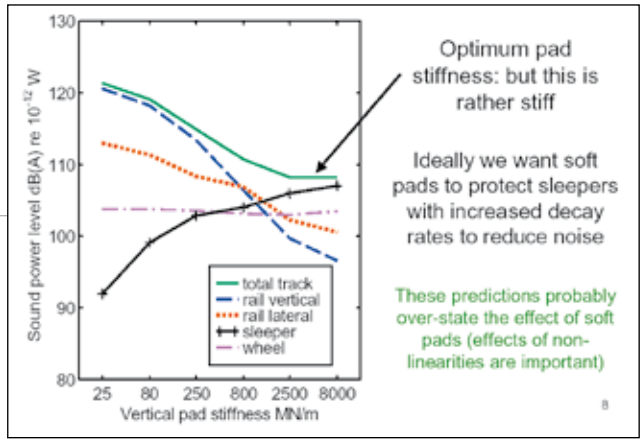


Figure 3.50 Noise emitted by track and wheel for different values of rail pads stiffness. It can be seen that to reduce noise a rather stiff rail pad is needed [5].

There are anyway other factors equally or more important to decide the rail pad stiffness. There is strong evidence that track geometry is kept for longer if rail pads are much softer. Another important factor that leads to the choice of soft rail pads is that exceptional loads due to wheel flats are less inclined to damage pre-stressed concrete sleepers currently used. The use of heavier and bigger sleepers gives more stability and less flexibility to the track, again confirming the need for soft rail pads. With the development of chemistry of rubber, nowadays "standard" rail pads have a dynamic stiffness in the order of 100 MN/m or lower, with marked non-linear characteristics.

Simulations show that the negative effect on noise emission of the use of "soft" rail pads could be compensated by the use of rail dampers (Figure 3.51).

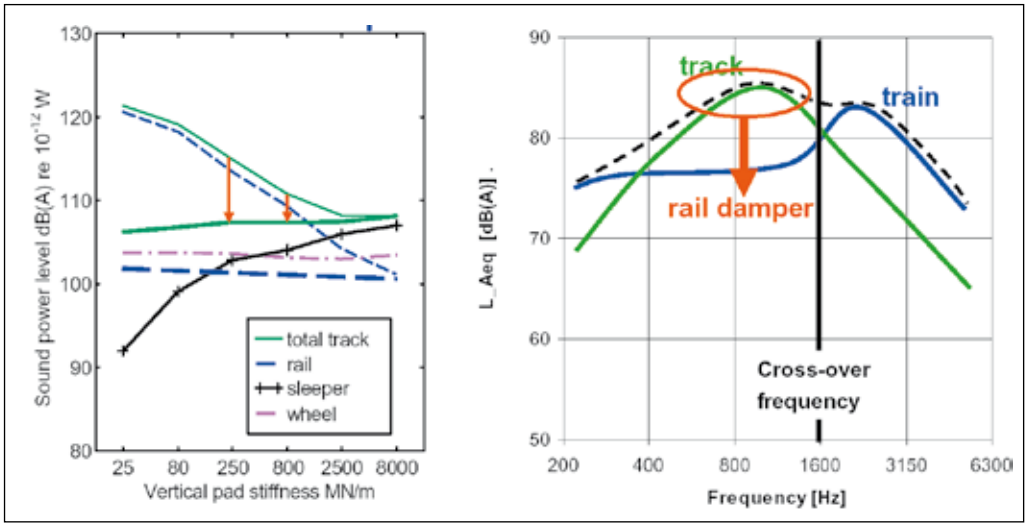


Figure 3.51 Reduction of noise emitted by the rail with the use of rail dampers [5].

Rail dampers are available in different shapes and sizes but are all based on the common principle that a vibrating rail can be "blocked" only by a damper (or a set of dampers) tuned on the main frequencies of vibration of the rail. It is again the case to underline that the rail has no natural frequencies (it is infinite) leading to a particularly broadband behaviour of the rail. It is therefore fundamental that the dampers have a very good efficiency in a broad frequency range, leading to "multilayered" solutions of steel sheets alternated to rubber sheets (Figure 3.52) or to solutions where steel bars of different size are "embedded" in a rubber block (Figure 3.53).

Rail dampers can then be fixed to the rail via mechanical coupling (bolts and nuts) or by bonding. The first case is simpler but may require maintenance, the second case is certainly more expensive but in principle doesn't require maintenance.

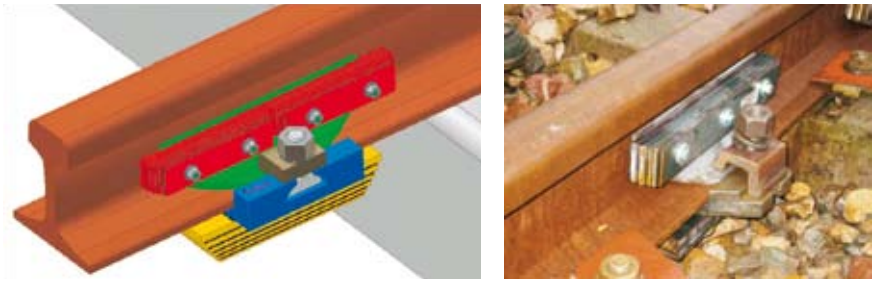


Figure 3.52 Rail dampers made of multilayered still and rubber elements fastened to the rail with screws [6].

Figure 3.53 Rail dampers made of steel bars embedded in rubber and glued onto the rails [6].

With the use of rail dampers it is possible to reduce the overall noise by 2 to 4 dB, while the rail noise component can be reduced by more than 5 dB (Figure 3.54, Figure 3.55 and Figure 3.56).

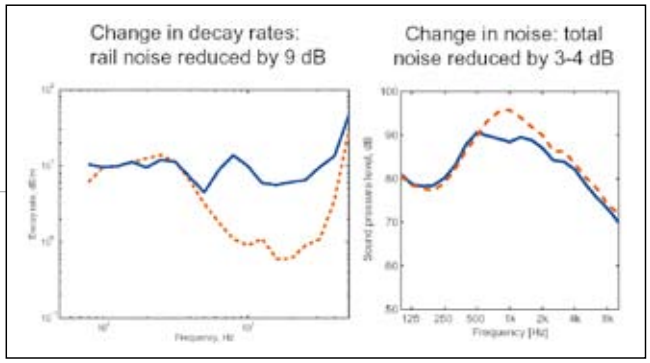


Figure 3.54 Increase in decay rates (left) and reduction of overall noise (right) when rail dampers are used [5].

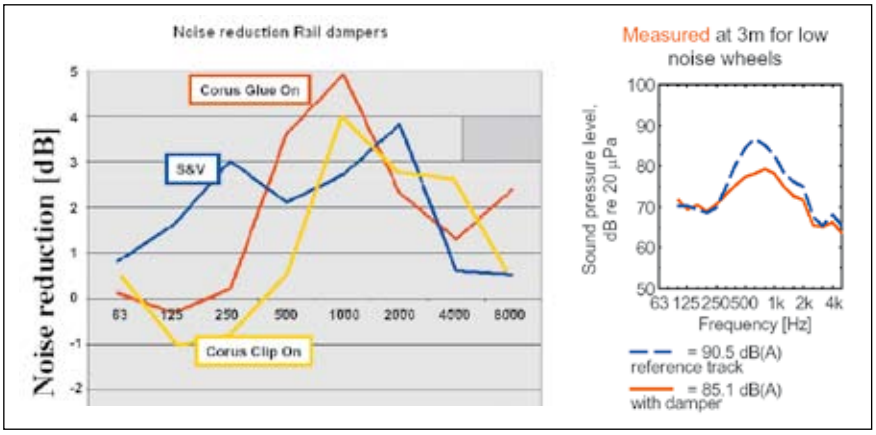


Figure 3.55 Noise reduction that can be obtained by the use of rail dampers of different kinds [5].

distance from centre of the track	Train category			
	ICE	IC	regional	freight
3 m	2,5 dB	3,2 dB	3,4 dB	4,1 dB
8.5 m	2,1 dB	2,7 dB	3,0 dB	3,4 dB
avg speed (km/h)	162	165	138	90

Figure 3.56 Results of the use of rail dampers in the Silent Track project [5]

4 NOISE BARRIERS

If measures at the source are insufficient or impracticable, the legislator says that interventions on the transmission path can be applied.

Two possibilities exist to "intercept" sound pressure waves travelling from the source to the receiver with the use of surfaces with acoustic impedance: noise barriers close to source or close to the receiver. The first case can be seen in Figure 4.1: a freight car is equipped with side skirts and very low barriers are positioned along the track.

Figure 4.1 Bogie shrouds and low trackside barriers combination tested on a freight wagon [6].

Solutions of this kind have not encountered the favour of train operating companies and infrastructure owners for a number of reasons: vehicles must be retrofitted, accessibility to bogie components is limited (visual inspections are not possible anymore), heat removal from braking components may be reduced (airflow is modified), trackside barriers may be an obstacle to maintenance operations (ballast tamping, reprofiling). As a matter of fact, no solutions of this kind have been implemented in service.

If the buildings along the line are not too high, the other solution is offered by noise barriers. Sometimes these rules are not exactly followed, as can be seen in Figure 4.2.

Figure 4.2 Sometimes noise barriers are installed where their efficiency is evidently extremely limited. They only offer clean surfaces to writers for unattractive graffiti…

The other limitation to the use of noise barriers is the orography of the terrain. In the presence of deep valleys (like in Switzerland and in most of the Alpine countries), noise can reach the receiver passing over noise barriers installed at the trackside.

The design and evaluation of the performances of noise barriers is described in numerous textbooks on noise. The main reason for the development of noise barriers is due to the extremely high annoyance given by road traffic; on the contrary, railway noise is much less disturbing for the same values of noise indicators. We give here only some fundamentals on the estimation of noise barrier performance, referring the reader to general acoustic knowledge sources.

Predictions are often based on the theory of diffraction. The Fresnel number is defined as:

$$N = 2\Delta x/\lambda = 2\,(i+r-d)/\lambda$$

where i is the distance from the source to the top of the barrier (incident ray), r is the distance from the top of the barrier to the receiver (diffracted ray), d is the straight (direct) distance from the source to the observer and $\lambda=c/f$ is the wavelength of the noise component at frequency f being c the speed of sound (≈340 m/s).

The Fresnel number is then inserted in the classical Maekawa chart or entered in the Kurze-Anderson equation for line source (or moving point source)

$$IL = 5 + 15 \log_{10}\left(\frac{\sqrt{2\pi N}}{\tanh\sqrt{2\pi N}}\right)$$

where *IL* is the Insertion Loss at a given wavelength for a given "path length difference" Δx.
A simple example for f=340 Hz and a Δx=1 m leads to N=2 and to *IL*=13.25 dB. It can be seen how also for limited path length difference the abatement is absolutely relevant. Figure 4.3 plots the insertion loss as a function of frequency for different path length differences. It is clear that the advantage at low frequency is less significant than at high frequencies and that the increase of IL is linear with logarithmic increase of frequency. Noise at higher frequencies is therefore better shielded by high barriers or, alternatively, lower barriers are needed to get the same IL for high frequency sources.

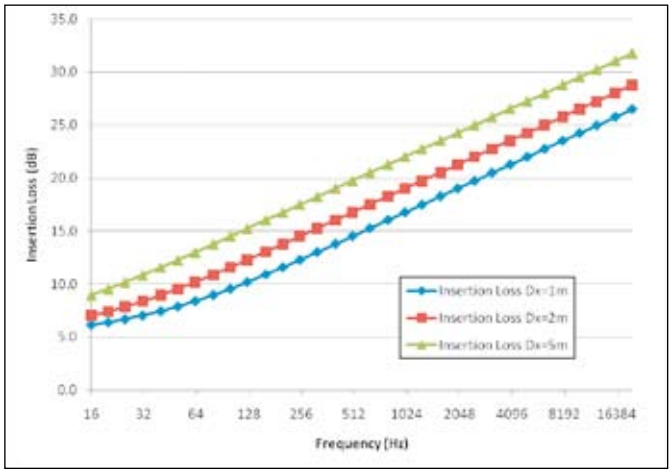


Figure 4.3 Chart for the estimation of Insertion Loss for noise barriers with a path length difference of Δx=1 m, Δx=2 m and Δx=5 m vs. frequency.

The main limitation of noise barriers is, as already mentioned, the diffraction that generates an "additional" source at the top of the barrier (*Figure 4.4*).

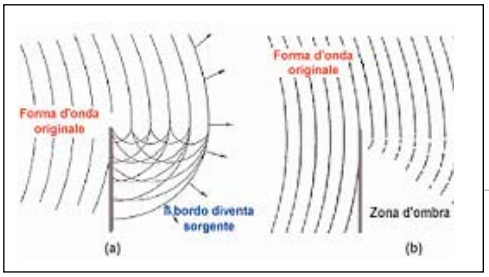


Figure 4.4 Diffraction effect and possible advantage obtained by "shading" [26].

Barrier "crowning" with different devices has been developed to reduce the diffraction effect. If the abatement of the top edge of the barrier were infinite there would be no noise in the area "shaded" by the barrier. Different types of devices, purely reflective (*Figure 4.5*) or absorptive (*Figure 4.6*), were developed and commercially available. Absorptive solutions are based on the use of porous material that tends to deteriorate with atmospheric agents.

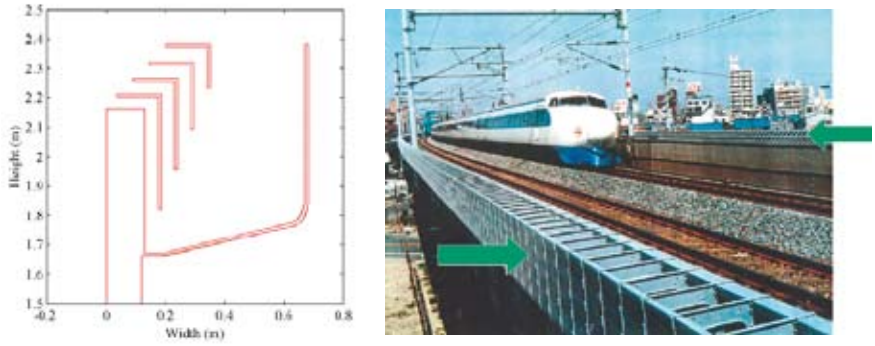


Figure 4.5 The Calmzone® device is purely reflective (left). Application on Japanese high speed railways (right) [27, 28].

Figure 4.6 Barrier crown with absorption devices of different kinds [26, 27].

A more complex analysis, including the multiple reflection between the barrier and the vehicle and the effect of the shape of the top of barriers lies outside the scope of this book.

Noise barriers have, on their side, the fact that noise reduction offered is rather well predictable and that their efficiency is constant along time and distance. Drawbacks are the high initial cost, a sometimes unacceptable visual impact (see chapter 2) and a limitation of access to the track for safety and rescue reasons.

5 CONCLUSIONS ON ROLLING NOISE

Summarizing all the results that have been described in this part of the book, it can be said that an alternative to noise barriers exists.

It certainly requires an interdisciplinary approach as:

- wheels must be designed possibly with straight web and fitted with efficiently high damping devices;
- braking with composite brake blocks is a must for freight wagons and could be of great help also for passenger coaches and locomotives, with the aim of further reducing the roughness of the already smooth tread of disc braked wheelsets;
- efficient, long lasting and relatively low cost rail grinding should extend from Germany to all other countries where railway noise is an issue;
- rail damping should be used locally, where rail noise component can be dominant.

If properly designed and maintained, the full circle of planning, investing, checking and correcting can give the expected results. Political efforts should therefore be directed towards financing and implementing an interdisciplinary approach to railway noise which can result both in a large consensus from the public and consistent money saving.

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R. van Aken: *Raildampers Product release and application*
M. Hecht: *Curve squeal*
M. Hiensch: *Preventing railway Squeal Noise through railhead optimisation*
A. Kuijpers: *Noise control by rail roughness control*
E. Jansen: *Acoustic measurements on rolling stock retrofitted with composite brake blocks*
R. van Mil: *Silent Passenger trains*
J. Oertli: *Composite brake blocks international view*
J. Peen: *Whispering Train Programme Technical knowledge The Life Cycle Costs*
C. Roovers: *ProRail Railway Noise Knowledge Centre*
C. Roovers: *Innovative solutions for noise of railway yards*
H. Stark: *Introduction Whispering Trains & Homologation composite brake blocks*
D. Thompson: *Rail Dampers*
D. Thompson: *Noise control through rail grinding*
E. Verheijen: *Rail dampers in the Netherlands - Acoustical knowledge*

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D. Benton: *Engineering Aspects of Rail Damper Design and Installation*
M. Beuving: *Optimal lay-out of railway depots*
P-E. Gautier , F. Poisson, E. Bongini , S. Molla: *SILENCE: combined reduction of noise from rolling stock and infrastructure*
J. Gräber, St. Dörsch: *Composite Brake Blocks (CBB) - an Overview*
D. Hartleben: *Grinding of Rails - Acoustic Benefits*
B. Hemsworth: *Noise Reduction at Source: EU Funded Projects*
F. Margiocchi: *Dampers on the rails in France: complete assessment of rail absorber performances on an operated track*
H. Paukert: *UIC activities on noise reduction*
M.Viscardi, N.Rusciano, S. Marucci: *Acoustic Characterization of DMU MINUETTO*

PART 2:
The Political and Economical Relevance of Rail Noise Abatement

Matthias Pippert

Part 2: The Political and Economical Relevance of Rail Noise Abatement

Matthias Pippert

1 INTRODUCTION / HISTORICAL BACKGROUND

Railway noise is a relatively new issue of environmental politics. For a very long time noise abatement was not an important concern for the railways. That is not a surprise because in the past noise from transport was not considered as such an important problem as it is today. Furthermore, the speed of trains was lower than today and the traffic was not as concentrated on a few trunk lines as it often is today.

Noise awareness, as far as railways are concerned, seems to have started in the big cities where a lot of people lived close to railway lines with intense traffic, being disturbed not only by the railway noise but by other noise sources of the metropolis as well. Consequently an early example for a railway which tried to abate noise with respect to the population was a railway which mainly operated in such a metropolis. The Berlin city railway (S-Bahn), part of Deutsche Reichsbahn, already in the 1920s started to use rails with a length of 30m instead of 15m in order to avoid rail joints. Rail joints were a major source for railway noise in that time, before the era of endless welded rails. The same method was used on trunk lines for fast trains which ran through towns and cities. At that time the German railways were also developing new solutions for the connection between rails and sleepers, switch tongues and frogs aiming at reducing noise emissions.[1]

Better working conditions for railwaymen were the reason for the invention of exhaust silencers when diesel engines went into operation. The German Federal Railways started to analyse the problem of engine noise systematically in the 1950s when major series of post-war shunting engines entered operation. Exhaust silencers were one instrument, but also attempts to insulate the driver's cab from the engine noise or to use damping methods for the engine bonnet were made. Engine noise was not only a problem for

[1] Baumann, Deutsches Verkehrsbuch, 1931, p. 268-9.

the train drivers but also for the shunters often standing or working next to the modern and loud diesel engines.[2]

Until very recently and with only few exceptions, there were nearly no regulations for noise emissions of railway vehicles. Typical legal requirements affected only the construction of new railway lines or major improvements of speed and capacity of existing lines. Typically legal immission limits were defined and calculated basing on the noise levels at the façade of houses. If it was expected that noise emissions of the newly built railway line would exceed the immission limits noise barriers or other "passive" means of noise barriers had to be constructed.

But within the last 15-20 years railway noise has become a major problem which even may impede the growth of rail transport if fast and substantial progress in noise abatement fails to appear. This is especially true in the case of freight transport, but also regional passenger transport is sometimes hurt by protests of the local population.

Figure 1.1 Noise is often annoying and can even bad for your health. (Source: Allianz pro Schiene / Taubert)

The conclusion is that the railways face a major challenge. Abating noise is becoming a precondition for continued growth and modal shift. Railways should not abate noise awareness but accept the challenge and be a reliable partner for the public aiming at reduced noise levels. First effective attempts have been made and some more are to follow. Nevertheless, increased political support for the railways could be very helpful even when it comes to noise abatement.

[2] Zboralski, Zulässige Schallstärken beim Bau und Betrieb von Fahrzeugen, 1957, p. 500-1

2 THE ENVIRONMENTAL PERFORMANCE OF RAILWAYS

Motorised transport implicates environmental problems. The major impact categories to be considered when comparing the different modes are

- energy consumption,
- CO_2 emissions,
- toxic emissions (especially particles and NO_x) and noise.

In terms of nearly all impact categories railway is rather the ecological solution than the ecological problem. The most urgent problem seems to be CO_2 because this substance is most relevant for abating the greenhouse effect.

2.1 Abating railway noise means climate protection

Unfortunately, reliable consolidated data for the European rail and road transport do not yet exist, but for instance in Germany the ratio of the specific CO_2 emissions per tonne kilometre is about 1 : 4,2 between rail and road freight transport. The respective ratio for passenger transport is 1 : 2,1 per passenger kilometre. The whole chain from the primary energy source to the wheel of the train or the car is included in this calculation. Of course, the relation is even better for the railways when you look at countries (e.g. Norway or Switzerland) with an even higher share of electric operation or with very high shares of renewable energy e.g. water or wind in the railways' overhead wire.

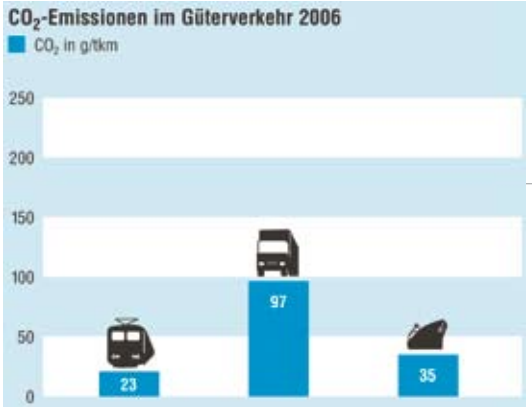
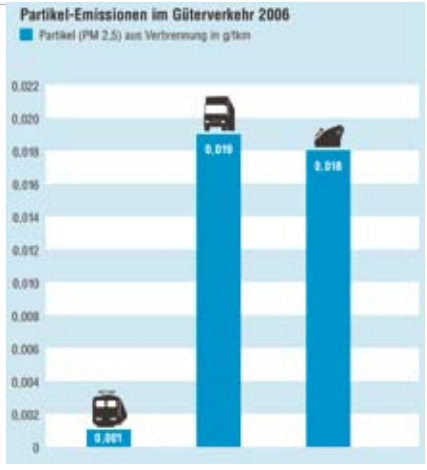


Figure 2.1 CO_2 emissions in German freight transport: Relation between rail and other modes (Source: Allianz pro Schiene, database Umwelt & Verkehr, based on calculations by Institut für Energie- und Umweltforschung, 2008)

Figure 2.2 CO_2 emissions in German passenger transport: Relation between rail and other modes; air transport: only domestic flights. (Source: Allianz pro Schiene, database Umwelt & Verkehr, based on calculations by Institut für Energie- und Umweltforschung, 2008)

Similar or even more impressive ratios are true for energy consumption and the kinds of toxic emissions named above. The railway is therefore ***the*** solution for climate protection and for the shortages in global energy supplies as far as transport is concerned.

Figure 2.3 Particle emissions in German freight transport: Relation between rail and other modes (Source: Allianz pro Schiene, database Umwelt & Verkehr, based on calculations by Institut für Energie- und Umweltforschung, 2008)



Polls show that in Germany with its relatively dense and intensely used railway network 20 % of the population feel themselves extremely or moderately disturbed by rail traffic while the respective figures are 32 % for air and 60 % for road transport. But the railways have no reason to be calm about the noise problem. The local relevance of railway noise may be so high that neighbours consider it to be the most important environmental problem. Everywhere where the number of freight trains is rising considerably protests occur and they can become very powerful. This is especially true where new railway lines for freight transport are being discussed or where former idle lines are about to be re-activated for long-distance freight transport.

Time is running: Measures for climate protection are urgent and so are measures for increasing the capacity of rail freight transport. That means that noise abatement is also urgent, because noise has become a relevant obstacle for modal shift.

2.2 Noise emissions - the "Achilles' heel" of rail transport

Noise emissions have become the "Achilles' heel" of rail transport in terms of environmental problems. Although less people are affected by railway noise than by noise from road transport, it has become a severe problem for the railways. A considerable part of the population is seriously affected by railway noise.

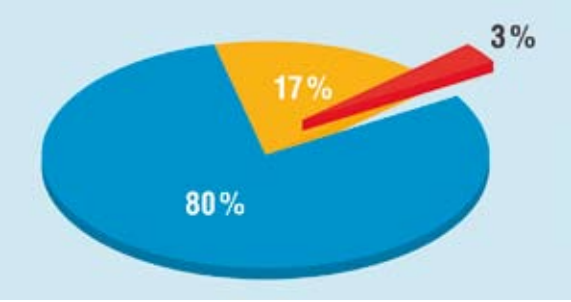


Figure 2.4 Annoyance because of noise: 3 % of the Germans are extremely or seriously annoyed by railway noise; 17 % are moderately or somewhat annoyed while 80 % feel not annoyed by railway noise. (Source: Federal Environmental Agency, poll "Umweltbewusstsein in Deutschland 2004")

When you compare the noise level in a distance of e.g. 25 m from the vehicle and relate it to the same amount of traffic (e.g. 1.000 persons per hour or 1.000 tonnes per hour) the noise levels of rail and road transport at the same speed are more or less the same, although it depends on the types of vehicles used. But these figures can be questioned: On one side trains usually run faster than road vehicles and on the other side the use of noise barriers is not considered. A clear figure like specific noise emissions per tonne kilometre or per passenger kilometre cannot be given.

Only a few decades ago noise emissions from the railways were not considered as a big problem. Several reasons can be identified which now lead to more awareness in terms of railway noise:

(1) Nowadays noise is everywhere. Life without noise, in contrary to the

past, has become nearly impossible in the industrialised European countries. Noise comes from industry, from all means of transport, and other sources, resulting in increased noise awareness of the population. Therefore single sources of noise which could have been neglected in the past today contribute to an overall noise burden which leads to serious health problems and is not longer tolerated by many people.

(2) The general trend of growing transport volumes leads, in principle, also to the growth of transport-related noise. For a long time the railways had been considered as a "loosing" transport mode, but this trend has turned to the contrary. Rail transport is growing and in several countries even the modal share is rising. That means that people who are affected by rail noise immission expect that the problem might become more serious in the future instead of losing its relevance.

(3) The rolling noise emission level of railways is related to the speed of trains, and the speed of trains has risen in the last decades. This is also true for freight trains. Before the continuous air brake was invented in the 1920s the speed of trains did usually not exceed 45 km/h. Afterwards the usual speed of European freight trains was 65 km/h until 80 km/h became the usual speed limit in the 1960s. Only some fast freight trains already reached 100 km/h as limit. Today, 100 km/h is the rule for "ordinary" freight trains while "fast" freight trains often run at 120 km/h. Although this raise of speed seems to have been necessary due to competition and capacity reasons it has resulted in the major noise problem railways have today.

(4) Rail traffic has been concentrated on a few trunk lines. A lot of parallel and spare lines which were still in use even for long distance freight transport a few years or decades ago have lost a lot of their traffic or even have been closed at all. While rail freight transport in Europe has been started to grow again, the noise burden in several countries has been concentrated on fewer lines than before.

It is a big progress in transport and climate policy that in recent years it was possible to reopen regional lines for passenger transport which had been abandoned in the private car-minded decades of the 1960s, 1970s and 1980s. But although regional passenger transport is much more silent than rail freight transport sometimes protests occur because the railway

already has got the reputation to be very noisy. The Vinschgaubahn (Val Venosta Railway) between Merano and Malles in Northern Italy (Alto Adige) is one example where parts of the local population were in opposition to the reopening because they feared the noise. But after operation had resumed people could be assured of the advantages of the railway because modern passenger trains are much more silent than passenger trains used to be a few decades before.

Figure 2.5 Silent Diesel Multiple Unit (Stadler GTW 2/6) serving the Val Venosta, a tourist and noise-aware region in Northern Italy (Source: Allianz pro Schiene / Pippert)

The railways are proud to be the most environment-friendly mode of transport. The environmental advantages are very good arguments for the political backing for rail transport. But when it comes to noise the railway's advantages are not very clear. Therefore it is a question of credibility for the railway industry and its suppliers to abate noise emissions. It is obvious that the railways in Europe are not acting today on a level playing field because e.g. tax regulations, the share of investment funds and other financial decisions of the states often encourage rather road and air transport instead of the railways. But on the other side the railways would be well advised to do their own homework in terms of noise abatement in order to have better arguments when struggling for a more levelled playing field.

2.3 First and most important step: retrofitting the European freight wagon fleet with composite brake shoes

Traditionally, freight wagons were fitted with brake systems using cast iron blocks. When braking these blocks are pressed onto the wheel tread and thus the train slows down. It can be loud when a freight train is braking but the more serious noise problem which is caused by the cast iron blocks is that the wheel treads are roughened and the rolling noise is increased.

Figure 2.6 Traditional cast iron brake shoe for freight wagons: wheel tread is roughened while braking. Thus the rolling noise is increased. (Source: Allianz pro Schiene / Pippert)

In the 1960s several railway companies started to use composite brake blocks but it was not until the 1990s that such brake blocks were considered as reliable enough for the general use in freight transport on the European network. With composite blocks (or "K-blocks") there is nearly no roughening of the wheels so that one reason for noise emission is omitted. The difference between the rolling noise emissions of wagons with cast iron brake blocks and with composite brake blocks is about 9-10 dB. That means a reduction of about 50 % in the perception of affected people and a reduction of 87-90 % in terms of noise energy.

In 2003 the members of the Community of European Railways (CER) committed themselves to procure only freight wagons with composite brake blocks in the future. It took some time until all railways and even the private wagon owners really pursued this self-commitment, but since 2007 the Noise TSI for freight wagons is in force. This document of European law sets emission limits for new wagons which cannot be achieved with cast iron blocks. That means that only wagons with composite brake blocks or even better solutions, e.g. disk brakes, have the chance to be homologated for operation on the Trans-European rail network.

Figure 2.7 Composite brake shoe ("K-block") for freight wagons: wheel tread is not roughened while braking. Thus the rolling noise is reduced. (Source: Allianz pro Schiene / Pippert)

Although new cars are now much more silent than the existing fleet, it is clear that freight transport will remain loud unless the bulk part of the existing fleet is retrofitted with the new type of blocks or similar solutions. The "natural" renewing process would take about 25-35 years according to the usual lifetime of freight wagons. It is broadly accepted within the European "rail community" that all freight wagons with a certain minimum kilometric performance (e.g. 10,000 km per year) and a minimum remaining lifetime (e.g. five or ten years) should be retrofitted in order to make fast progress in terms of noise reduction.

Unfortunately, if you want to retrofit old wagons with K-blocks instead of cast iron blocks it is not enough just to change the blocks. The problem is that the braking power is transmitted in a different way. The brake gear needs to be changed as well and even different types of brake cylinders have to be fitted. The costs depend on the type of wagon and the price level in the respective country. The Deutsche Bahn estimates the costs for retrofitting the German wagon fleet at c. 4,500 €per wagon. The costs are slightly lower for two-axle and four-axle wagons while for a six-axle articulated wagon with three bogies (e.g. Sggmrs for containers) costs are higher because a second brake cylinder is needed. The Swiss railways which have already benefitted from a state-funded retrofit programme calculated ex-post 13,800 CHF for a two-axle wagon and 21,800 CHF for a four-axle wagon. The higher amount for Switzerland seems to be due to higher wages and the inclusion of costs for engineering, project management and homologation.[3]

Most railway companies have agreed that the procurement and operation of freight wagons with composite blocks is practically not

[3] Bundesamt für Verkehr, Lärmsanierung der Eisenbahnen. Standbericht 2008. 1. Januar - 31. Dezember 2008, 2009, p. 13

more expensive than the procurement and operation of "traditional" wagons with cast iron blocks. The lifetime of composite blocks is much longer than the lifetime of cast iron brake shoes. However, the results of the first few years of operation show that not all problems have been solved yet. Further improvements and new patterns of maintenance seem to be necessary. First of all, the composite blocks do not lead the heat off which emerges during braking. Secondly, the K-blocks perform a higher force on the wheel treads. Both phenomena lead to increased wear of the wheels. This is especially true when both wagons with cast iron blocks and wagons with composite blocks are used in the same train because in this case the K-blocks perform a bigger share of the braking power. One solution might be to introduce new types of wheels. The other solution would be to accelerate the process of retrofitting so that K-blocks become the standard.[4] Experience of HUPAC, a Swiss company which operates trains for combined transport across the Alps, showed that during the first years the sum of LCC of wheels and brake shoes were higher with K-blocks than with cast iron blocks, but a balance seemed to be possible with further improvement of K-blocks and maintenance strategies as well as concentration on only one or two types of composite brake shoes instead of the broader range of composite block models used during the first time.[5]

From 2004 until 2007 Railion Nederland tested the "quiet Dolomit shuttle" which ran between Hermalle (Belgium) and Veendam (Netherlands) with a fixed train consist of Tapss-wagons (two two-axle bogies). All wagons of this shuttle operation were retrofitted with K-blocks of the Cosid 810-type. The result was a 9 dB noise reduction and interesting new knowledge about wear and LCC. On one side the lifetime of brake shoes and wheel treads was less than expected. Asymmetric wear of the wheel tread shaped up as a major problem. On the other side no flats occurred when using K-blocks. Reprofiling of the wheels was regularly necessary after slightly more than 200,000 km. With cast iron blocks flats often occurred and reprofiling was necessary in intervals – though in average of more than 200,000 km - between nearly 0 km and more than 800,000 km as a random problem. This led to the conclusion that "it seems that the maintenance of wheels has

[4] Miehlke, Lärmarme Güterwagen, 2007/8

[5] John, Betriebserfahrungen der HUPAC AG mit K-Sohlen, 2006.

become more predictable which can even lead to cost reductions."[6] If the problem of asymmetric wear of composite blocks was solved and better methods of reprofiling the wheel treads were found, the maintenance cycles of wheels and brake shoes could be put together at 230,000 km and thus maintenance could even become cheaper than with cast iron blocks.

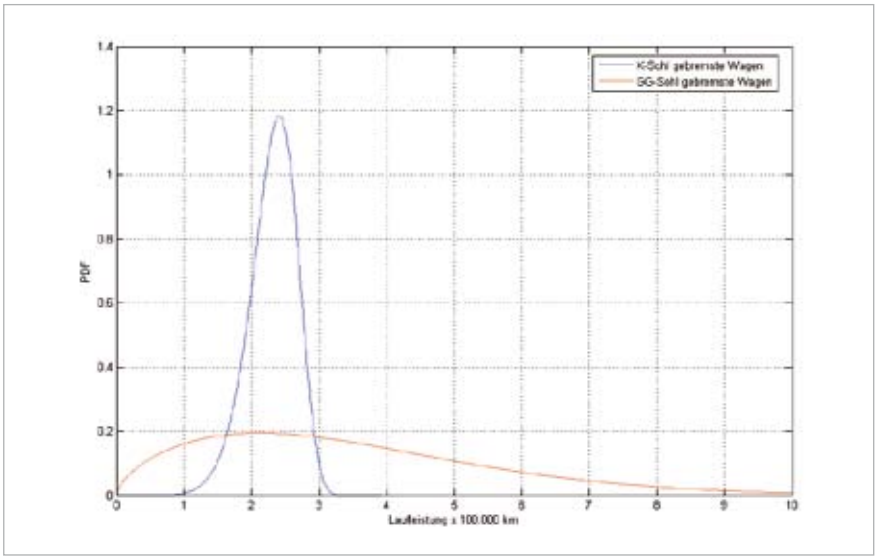


Figure 2.8 Probability distribution of kilometrage of wheels between re-profiling: Wheels with cast iron brake shoes have to be re-profiled in random intervals while with K-blocks the intervals are usually slightly above 200.000 km. (Source: Lloyd's Register Rail Europe; see also: Peen/Pos, Der leise Dolomit-Shuttle, 2007, p. 34)

Another attempt to make freight trains more silent are the "LL-blocks" ("LL"="Low-Low" →low cost, low noise). It is intended to introduce composite blocks which do not roughen the wheel tread but show the same pattern of braking power transmission so that the brake equipment of the wagon only needs minor changes or can even remain unchanged except for the new blocks. Such blocks seem to be a luring perspective for railway companies, wagon owners and governments which are requested to contribute to the retrofitting with public funds. But it is still unclear whether this solution is really viable. Some types of "LL-blocks" have been developed and tests have been taking place for some time. It seems that the retrofitting with LL-blocks is cheaper than

[6] Peen / Pos, Der leise Dolomit-Shuttle. Lärmreduzierende Maßnahmen durch K-Sohlen und Radabsorber; 2007, p. 33, own translation (MP)

K-blocks but not gratis and it is not yet clear whether LL-blocks lead at least to the same results in terms of noise reduction and cost-effective maintenance as the K-blocks. Therefore railway companies plead for the K-blocks until the advantages of the LL-blocks have been proved in practice.

It is an open question how the retrofitting of freight wagons will be financed. The railways have, until now, nearly no obvious commercial advantages of using more silent trains. Therefore they request state funds for this modernisation. Swiss voters authorized a comprehensive noise abatement programme for the railways in 1998. This included, as the first step, the retrofit of all passenger coaches and freight wagons with K-blocks, financed by the Swiss government. Only after this had been started the second step commenced which was the construction of noise barriers and the installation of noise-protecting windows in houses in the neighbourhood of frequently used railway lines. The idea behind this order was that measures for the rolling stock were cheaper and effective everywhere while noise barriers have much higher LCC and help only locally. It was clear that measures for the rolling stock would not be sufficient and in certain areas noise barriers would still be necessary, but all in all the Swiss strategy was cheaper by 40 % than if only noise barriers had been used. The Swiss noise abatement programme was a clear commitment to the railways. It was part of the Swiss transport strategy aiming at modal shift towards the railways and especially avoiding road freight transport across the Alps. Since the noise programme started the Swiss government spent 14,72 € per capita and year for railway noise abatement while the respective figure for e.g. Germany is 0,66 €.[7]

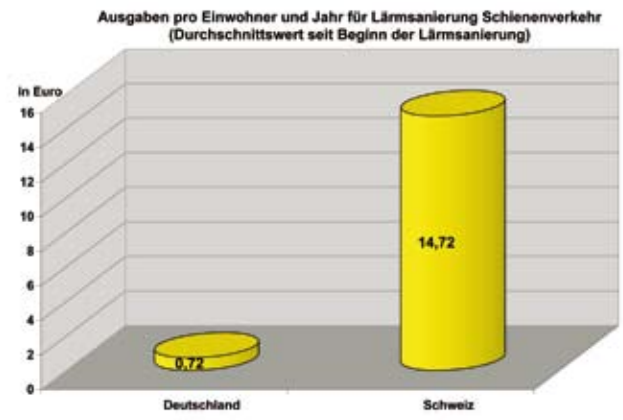


Figure 2.9 Expenses per capita and year for rail noise abatement in Switzerland and Germany. (Source: Allianz pro Schiene)

[7] Data as of 2007; Flege, Schienengüterverkehr und Umwelt: Trends, Probleme, Lösungen, 2007, sl. 20.

If all national governments in Europe funded the refurbishment of the freight wagon fleet they could save a lot of money which otherwise would be necessary to erase noise protection walls. In spite of that, governments are still reluctant to fund K-blocks and the reasons seem to be a mélange of regulative credos, the habit of rather financing infrastructure than vehicles, the hope for successful and cheap LL-blocks, the fear of foreign freight wagons, general reluctance to finance railways – and the discussion of noise-related track access fees.

In the author's view direct funding is the fastest and most effective way of financing the retrofit. The measure is clear. The wagon fleet to be addressed is also clear and direct contact between government and wagon owners could be arranged. Effective rules could be established to make clear that the funded wagons do a minimum kilometrage on the national network. In the case of Germany the freight railways would need about six month to start the process and after 5 – 7 years the retrofitting could be done, using the regular maintenance intervals. Also the German private wagon owners' association has clearly voted for a fast retrofitting.[8] One counter-argument is that while 130,000 German wagons would be retrofitted some 470,000 foreign wagons would remain loud and still allowed to enter the German network. But this argument is not convincing because about 85 % of the operational performance on the German network is done by German wagons. The Swiss wagons which also have a considerable share are already being retrofitted. New wagons do have K-blocks. Although being allowed, most "loud" wagons from other countries will not do a lot of kilometres on the national German network. The situation would be more or less the same in other countries with a large network like France, Italy or Poland.

Due to the present economic crisis this year and the next would be the best time to start with the systematic retrofitting project. Such a programme would contribute to economic recovery programmes financed by the national states. Temporarily a lot of wagons are idle and the railways' maintenance workshops have spare capacity and face the challenge to keep their staff busy. After the crisis the modal share of rail freight transport will probably resume to increase and it would be a good idea to make it with silent wagons. However, the European Commission favours a harmonised bonus

[8] Heinrici, Waggonhalter wollen Lärm mindern, 2007

system for the 1435 mm-network and hopes thus to achieve the retrofitting of all relevant wagons (up to 370,000) until 2015.[9]

Noise-differentiated track access fees might lose effectiveness due to a lot of interfaces between the government, the infrastructure manager, train operating company, the wagon's owner (often a leasing company), the wagon's user and even the shipper if long-term contracts are concerned. Furthermore, it would presumably take, at least, one or two years to establish an effective system of noise-related track access fees and this would not even take place in all EU member states. The author's rough estimation is that with track access fees the process would take twice the time and only include half of the wagons.

On the other hand, noise-differentiated track access fees are in general a good idea. It would not be necessary to change the national systems of track access fees, but a bonus system according to the kilometrage per retrofitted wagon axle could be established. If the countries with bigger networks still hesitate to finance the refurbishment, such a bonus system is obviously the best choice for small countries. Additionally to their investments in the national network and the national wagon fleet, Switzerland also established a bonus system for wagons with K-blocks in order to give incentives for foreign wagons. Unfortunately these bonuses have not been used to great extent by foreign wagon owners. Also the Netherlands established such a system in 2008, paying a bonus of 0.04 € per wagon kilometre for wagons with K-blocks or LL-blocks provided that the wagon was set in service before 2008 and retrofitted not earlier than 1st Jan. 2008. The bonus is calculated according to the kilometrage on the Dutch network and was at first restricted to the maximum of 60,000 km per wagon within three subsequent years. Meanwhile this was doubled to 120,000 km within three subsequent years in order to get better response from wagon owners. A similar scheme is also in force for retrofitted passenger coaches.[10]

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[9] Commission of the European Communities, Communication from the Commission to the European Parliament and the Council. Rail noise abatement measures addressing the existing fleet, 2008, esp. pp 4-7.

[10] Snepvangers, Stimulation of retrofit, 2008; ProRail, Aanvulling 4 op de Netverklaring 2009 Gemengde net, 15th July 2008

In spite of all counter-arguments it is very clear that track access fees are at least a good second best approach to achieve more silent wagons with K-blocks. The worst case would be if the conflict between both "parties" continued and the result was to do nothing because both sides failed to triumph. Furthermore, the noise reduction of 9-10 dB which can be achieved by K-blocks will not be enough for the future. Another 10 dB is the minimum to be achieved if the railways want to lose substantial parts of their noise problems in freight transport. New solutions exist as prototypes with potentials between 3 dB and 9 dB additionally to the K-blocks. These projects do not seem to be self-financing and the process of identifying the best solution could be improved by a well defined noise-related access fee scheme. Such a system should privilege silent wagons and especially give incentives for wagons which are intensely used.

One important argument has not been mentioned so far: If a train consists of e.g. 20 wagons, it makes nearly no difference when only a few wagons have silent brakes or silent bogies. The best improvement would be if intensely used block trains consisted of 100 % silent wagons and even this could be favoured by track access fee schemes.

3 THE LEGAL AND POLITICAL FRAMEWORK FOR THE ABATEMENT OF RAILWAY NOISE

Immission limits for new or improved railway lines were the "traditional" type of rail noise regulation. Such limits are still in use and urge the infrastructure managers to include noise barriers and other noise protection measures when new construction projects are planned. Noise emission limits are, however, a quite new requirement in the homologation of railway vehicles. Except for a few forerunners in national law they were invented by EU regulation. The first step was the TSI for High Speed Trains of 30th May 2002.[11] Since June 2006 new railway vehicles have to fulfil the requirements of the Noise TSI for conventional rail transport. Freight wagons were exempt at first but since February 2007 have to respect the TSO noise limits as well. TSI Noise brought a reduction of 5-10 dB, according to the general type

[11] Decision of EU Commission 2002/735/EG of 30th May 2002; Official Journal EC L 245/402, esp. p 419-420. In the following the paper will focus on conventional rail transport.

of vehicle, compared to the vehicle generations invented before. When it comes to freight wagons, the Noise TSI favours the retrofit with K-blocks. It has already been announced that stricter noise limits might be in force from c. 2016.
A new issue is the action planning of the European Noise Directive (END). It comprises requirements for strategic noise maps and noise action plans based on these maps. The END focuses on agglomerations and main transport infrastructure, but does not express a clear target how far the noise action plans should go. Nevertheless, it may happen that the railways will not meet the requirements of noise action plans set in force by the regional authorities although TSI Noise is respected.

3.1 Homologation of vehicles (TSI noise, national law)

3.1.1 National law

Before the Noise TSI came into force, some countries, e.g. Austria, Italy and Switzerland, already had national noise emission regulations for rolling stock. They differ from the Noise TSI in terms of limits, definitions or measurement methods. However, it can in general be assumed that a vehicle that fulfils the Noise TSI limits also complies with the current national limits of these countries. In the future, national noise limits will hardly have any relevance for homologation regulations for the European railways because, in the interest of interoperability, the Noise TSI will almost certainly also become the benchmark for vehicle homologation outside of the interoperable network.

In 1993 the **Austrian** government decreed a regulation about the noise emissions of new railway vehicles.[12] It defined limits for interior and exterior noise at both running and standstill. The regulation had to be applied for all new vehicles in Austria (§ 5 (1)-(3) SchLV) and for foreign traction units homologated for the Austrian network for the first time (§ 10 (2) SchLV).

[12] Verordnung des Bundesministers für öffentliche Wirtschaft und Verkehr über die Lärmzulässigkeit von Schienenfahrzeugen (Schienenfahrzeug-Lärmzulässigkeitsverordnung – SchLV); in: Bundesgesetzblatt für die Republik Österreich, Jahrgang 1993, 154. Stück, 25. Juni 1993, doc. no. 414.

Special regulations applied for vehicles imported second-hand to Austria (§ 5 (4) SchLV). Foreign carriages and wagons being operated under the conditions of RIC or RIV were exempt from the Austrian noise limits. For Austrian freight wagons it included a step-by-step approach as the noise limits were reduced by 5 dB (A) after 31st Dec. 1996 and another 5 dB (A) after 31st Dec. 2001 (§ 7 SchLV, see footnote 6). This Austrian regulation somewhat also affected the German procurement policy. Traditionally the Austrian and German railways cooperate quite well in the sense that electric locomotives operate regularly over long stretches of the neighbour's railway network. This is quite easy because of a common electricity system, loading gauge and train control systems. Since 1993 new German locomotives intended to operate in Austria have to fulfil the requirements of the Austrian noise regulation. This is one reason why Deutsche Bahn procured a series of class 182 "Taurus" locomotives which had been developed for Austria. The Landesnahverkehrsgesellschaft (LNVG), the Public Transport Authority for the federal state of Lower Saxony in 2005 called for tenders for the procurement of 11 diesel locomotives for Regional Express services. As the Noise TSI was not yet available and no German noise regulation was applicable, LNVG stipulated that the new engines fulfilled the Austrian noise emission regulation.

Figure 3.1 Class 246 (Bombardier TRAXX DE) diesel locomotive for regional passenger services in Lower Saxony: more silent than required by Noise TSI (Source: Dr. Ulrich Bitterberg)

The PTA chose the Traxx P 160 DE (class 246). The methodology of Noise TSI and the Austrian "SchLV" is somewhat different which can be shown by a confrontation of the respective values for external noise:

Noise emission values of Traxx P 160 DE (class 246 LNVG)				
Requirements	Indicator	Limit TSI [db (A)]	Limit SchLV [dB (a)]	measured value [dB (A)]
External noise standstill TSI	$L_{pAeq,60s}$	75	-	67
External noise standstill SchLV	$L_{pAeq,60s}$	-	80	74
External noise passing at 160 km/h	$L_{pAeq,Fmax}$	-	95	91
External noise passing at 80 km/h	$L_{pAeq,Fmax}$	-	86	80
External noise start-up TSI	$L_{pAeq,Fmax}$	89	-	81
External noise running TSI	$L_{pAeq,Tp}$	85	-	82

Source: Bitterberg/Schätzer/ Zapf, Die Baureihe 246 – neue Traxx-Diesellokomotiven für den schnellen Regionalverkehr in Niedersachsen, 2008, p. 408; similar values also known for Siemens EuroRunner (ÖBB class 2016), see Anhorn, Ingo: How to design a low-noise locomotive? Concept and noise emissions of the locomotive Eurorunner 20, 2006, sl. 21

In March 2000 **Switzerland** introduced a Federal law on noise remediation for the Swiss railways.[13] Based on this law a regulation about the noise remediation of the railways was decreed in November 2001 and last changed in February 2005. The Swiss noise remediation programme comprises measures on the rolling stock, the existing railway lines and on houses burdened with railway noise immissions. The regulation defined target values for refurbished passenger carriages and freight wagons. The target values are for

a) passenger carriages: $L_{PAeq,Tp} = 84$ dB(A)
b) freight wagons: $L_{PAeq,Tp} = 86$ dB(A)

and have to be complied with at 80 km/h. The conditions for the acoustic measurements are similar to those defined in the Noise TSI, but refer explicitly

[13] Bundesgesetz über die Lärmsanierung der Eisenbahnen vom 24. März 2000, doc. no. 742.144.

to the prEN ISO 3095/January 2001 and to the definition of rail roughness of the High Speed TSI. As in Noise TSI for conventional railway vehicles, the values given have to be respected in a distance of 7.5 m from the middle of the track and at 1.2 m over the railhead.[14] Actual target values for locomotives and Multiple Units were not given as the Swiss noise remediation programme focuses on the brake shoes of carriages and wagons.
The programme started with the remediation of passenger carriages and measuring campaigns have shown that the target values have been reached. The values for the most common types of standard gauge carriages were 82.8 dB(A) for EW I and EW II and 80.2 dB(A) for Bpm. Values for the EW III carriages have not yet been published. New vehicles have not been included in the programme as those carriages have low-noise equipment (e.g. disc brakes) from the delivery and comply with the emission limits. Also metre gauge carriages of Rhätische Bahn (RhB) and Zentralbahn (zb) have been included in the remediation programme and values in the range of 81-82 dB(A) have been achieved. Presumably due to the steep gradients on these lines some modifications of the brake systems had to be provided and tested before the whole fleet could be refurbished.
Provisional monitoring shows that also the target values for freight wagons are reached.[15]

In July 2006 new regulations were decreed giving noise emission limits for new railway vehicles. Measurements have to show the noise levels at 100 %, 75 %, and 50 % of the regular maximum speed. These noise levels then have to been standardised to 80 km/h using the same formula as European Noise TSI (applied when speeds are > 80 km/h). In general the same methodology is applied as with the remediation target values. The actual noise limits are:

Vehicle type	**Noise limit (Transit Exposure Level)**	**Respective limit TSI**
Locomotives	83 dB (A)	85 dB (A)
Multiple units	82 dB (A)	81 / 82 dB (A)
Passenger carriages	80 dB (A)	80 dB (A)
Freight wagons	84 dB (A)	82 / 83 / 85 dB (A)

[14] See Annex I of the Verordnung über die Lärmsanierung der Eisenbahnen (VLE) vom 14. November 2001 (Stand am 22. Februar 2005), doc. no. 712.144.1, p. 12.
[15] See Bundesamt für Verkehr BAV, Lärmsanierung der Eisenbahnen. Standbericht 2008. 1. Januar – 31. Dezember, 2009, p. 9-11. and 13.

The values are stricter than those of the noise remediation programme and in most cases even somewhat stricter than those of the European Noise TSI. The Swiss regulation does not define a special value for start-up noise and stationary noise, but the limits have also been applied for traction units at all operational situations between 0 km/h and 40 km/h.

Figure 3.2 Class 428 Electric Multiple Unit (Stadler FLIRT) of the German TOC "cantus": one of the most silent EMUs at present in operation. (Source: Allianz pro Schiene / Pippert)

Additionally, a limit is defined for Multiple Units with a length of equal or more than 50 m length (75 dB (A) in a distance of 25 m and height of 3.5 m above railhead)
All regulations apply also for narrow gauge railways although exceptions are possible in special cases.

All limits are also valid for new foreign vehicles which are to be homologated in Switzerland. But for vehicles which are older only the limits of the remediation programme are applied. The Noise TSI has not been transferred yet into Swiss national law. It is intended to adjust the national noise limits to TSI in 2010.

In 1998 also **Italy** defined noise emission limits for railway vehicles in the national regulation (Decreto 459/98). While the Swiss and the EU regulation

refer to values in a distance of 7.5 m (height 1.2 m) from centre of the track, the Italian regulation decrees levels for a distance of 25 m (height 3.5 m). This may lead to a difference of 5-8 dB (A), but in this case calculation cannot replace measurement. Furthermore, like Austria but unlike Switzerland and EU the maximum level is chosen instead of a Transit Exposure Level.

The values are defined for vehicles entering service from 1st January 2002 onwards, whereas for vehicles entering service from 1st January 2012 onwards, stricter limits will apply. Only limits for pass-by noise are defined which have to be measured at speeds which are different according to service and vehicle type. A rough estimation shows that already the values for 2002 are quiet strict compared to the Noise TSI.

Noise limits				
Pass-by noise (L_{pAmax}), distance 25m, height 3.5 m				
Speed	Service type	Vehicle type	Limit 2002	Limit 2012
250 km/h	Passenger	Traction unit	90	88
		Hauled vehicle	88	86
160 km/h	Passenger	Traction unit	85	83
		Hauled vehicle	83	81
	Freight	Traction unit	85	83
		Hauled vehicle	90	88
90 km/h	Freight	Traction unit	84	82
		Hauled vehicle	89	87
80 km/h	Diesel locomotive		88	86
	Diesel Multiple Unit		83	81

Source: DPR 459/98 of 18th Nov. 1998; Bracciali/Cervello/Moroder, Qualificazione acustica dei treni della Val Venosta, 2005, slide 3.

No special regulations are given in the Italian decree of 1998 how to deal with foreign vehicles and with a future (seen from 1998) European Noise TSI.

For vehicles that are not licensed according to TSI, as a rule older vehicles, the national homologation regulations still apply when vehicles are imported second-hand or for the first time homologated for international service into the respective country. Noise limits should be seen as infrastructure relevant

criteria that must be met by the vehicle being licensed. While foreign vehicles which fit to the Noise TSI limits have to be accepted on the Trans European Network by the national authorities, foreign vehicles might not be allowed for operation on other national railway lines in countries where the national noise limits are stricter than the European law dictates.

It is unlikely that additional countries will introduce noise limits into their national homologation regulations on railway rolling stock that are stricter than the provisions in the Noise TSI. If anything, it is to be expected that the application of the Noise TSI provisions will be extended to cover railway lines that do not belong to the interoperable network.

3.1.2 Noise TSI (EU Commission Decision 2006/66/EG)

The Noise TSI for conventional railway vehicles (excluding high-speed transport) was set out in the Commission's Decision 2006/66/EG. It has been in force since July 23, 2006 for locomotives, multiple units and passenger carriages, and since February 1, 2007 also for freight wagons. The directive sets the upper limits for the external noise emitted by rolling stock when stationary, starting up and pass-by noise when travelling at 80 km/h. The Noise TSI applies directly only to the 'Trans-European Network' as defined by the EU, but also when only small individual sections of the network are used. In addition, there are plans to expand it to cover the entire European conventional railway network. In the long-term, only the sub-sections of the network with clearly isolated operations will be exempt from the European licensing regulations.[16] For most types of rolling stock therefore vehicles will have to conform with the Noise TSI to obtain a license. The separate development of rolling stock that does not comply with the Noise TSI would be neither advisable nor make economic sense. Compliance with the mandatory noise limits is defined in the EU Directive 2001/16/EC as being a basic essential requirement, which means that EU member states are not allowed to derogate from the regulations when licensing rolling stock for use. Only in exceptional and unique cases will it be legally possible, when licensing rolling stock, to disregard the noise limits defined in the Noise TSI – namely if rolling stock has been developed specifically for operations

[16] Directive 2001/16/EC article 2, letter k

outside of the interoperable network and no intersection with other parts of the network is intended. Examples of such cases are narrow-gauge trains, DC multiple units for local transport systems (not equipped for multiple systems), rolling stock with a particularly high axle load for operations on non-public networks of the coal and steel industry, and in certain cases for rolling stock operating with third-countries if no borders between member states within EU are crossed.

TSI noise limits differ according to the type of rolling stock. The Noise TSI does not stipulate noise remediation measures for older rolling stock, but specific rules apply when older vehicles are substantially refurbished. There is a considerable systematic difference in how the TSI for noise limits are applied to freight wagons and to other types of rolling stock, particularly regarding the rules on upgrades. The rules for freight wagons are therefore presented separately below.

Limits and special rules for traction units and passenger carriages

The limits for start-up noise, pass-by noise and stationary noise can be seen in the following table. For purchasing projects that were current at the time there was a two year transition period. This has now expired for all types of vehicle. Rolling stock that was covered by this transition period is allowed to be 2 dB (A) louder than stipulated for new vehicles (number 7.5.1 Noise TSI). One exception is provided for diesel multiple units with a rating of more than 500 kW per engine, which have a transition period of five years for start-up noise (number 7.5.2) The individual testing methods are regulated by and listed in Noise TSI. With few exceptions (see EU Official Journal 2006, L37/23ff), the Noise TSI is based on the ISO standard 3095 (in the draft version from 2001 prEN ISO 3095:2001). Most measurement readings are defined as being 7.5 metres from the track's centreline and 1.2 metres above the top of the rail. The quality of the track is defined for the measurement process (reference track), which is important in understanding how testing is carried out. This is crucial because a track of insufficient quality can magnify rolling noise, making the test results incomparable. The testing process therefore involves a great deal of effort, and cannot be carried out at any arbitrary place on the network.

Noise limits for locomotives, multiple units and passenger carriages according to Noise TSI[17]

Vehicle category	Stand-by noise	Start-up noise	Pass-by noise
Electric locomotives	≤ 75 dB (A)	≤ 82 dB (A) [P < 4.500 kW at the rim] ≤ 85 dB (A) [P ≥ 4.500 kW at the rim]	≤ 85 dB (A)
Diesel locomotives	≤ 75 dB (A)	≤ 86 dB (A) [P < 2.000 kW at the shaft] ≤ 89 dB (A) [P ≥ 2.000 kW at the shaft]	≤ 85 dB (A)
Electric multiple units (EMU)	≤ 68 dB (A)	≤ 82 dB (A)	≤ 81 dB (A)
Diesel multiple units (DMU)	≤ 73 dB (A)	≤ 83 dB (A) [P < 500 kW per motor] ≤ 85 dB (A) [P ≥ 500 kW per motor]	≤ 82 dB (A)
Passenger carriages	≤ 65 dB (A)	---	≤ 80 dB (A)

Additional explanations:
Special rules apply to the measurement of start-up noise and stand-by noise which partially even differ according to the vehicle category (e.g. diesel vs. electric). The special rules for the stand-by noise refer to the operation of the ancillary units. The interval for measuring the stand-by noise is generally 60 seconds.

The pass-by noise limit is defined for a speed of 80 km/h in a distance of 7.5 m from track centreline and height of 1.2 m above railhead. Additional, the pass-by noise at maximum speed (but not exceeding 190 km/h) has to be measured. The latter will than be recalculated to 80 km/h using the formula $L_{pAeq,Tp}$ (80 km/h) = $L_{pAeq,Tp}$(v)-30*log (v/80 km/h). The higher value must not exceed the limit given in the chart above.

[17] According to Section 4.2.2.1 of Noise TSI (Official Journal E.C. L 37/12) the terms „diesel" or „diesel engine" refer to all types of combustion engines which means that the limits also apply to vehicles which are operated with other types of fuel e.g. agro fuel or natural gas. For Great Britain and Ireland some of the noise limits are higher (Sections 7.7.2.1 and 7.7.2.3), presumably due to the smaller loading gauge in those countries. In Estonia, Latvia and Lithuania for the time being the Noise TSI limits for traction units and passenger carriages are not yet in force (Section 7.7.2.6).

If existing vehicles are substantially refurbished or modernised ("substantially" meaning that a new homologation is necessary), it has to be proved that the relevant noise levels have not been increased (section 7.6.2 of Noise TSI).

Additionally, the Noise TSI also defines noise limits for the internal noise inside railway vehicles.

Maximum limits and special rules for freight wagons

Freight wagons are distinguished by the number of axles in relation to wagon length over buffers. The coefficient of the number of axles per metre of wagon length is called APL. A limit on the start-up noise of freight trains is not considered meaningful. With regards to refrigerator wagons in particular, Noise TSI (Section 4.2.1.2) defines additionally a limit for the stationary noise of freight wagons.

Maximum noise limits (pass-by at 80 km/h) for freight wagons

Freight wagon	APL [1/m]	LpAeq, Tp [dB(A)]
New	≤ 0.15	82
Renewed	≤ 0.15	84
New	> 0.15 ≤ 0.275	83
Renewed	> 0.15 ≤ 0.275	85
New	> 0.275	85
Renewed	> 0.275	87

Additional explanations:
Values measured at V=80 km/h, distance 7.5 metres from the track's centreline and 1.2 metres above the top of the rail, on a reference track. Additional measurements are also taken for freight wagons at their maximum speed. As with other types of rolling stock, these results are recalculated to 80 km/h. Both values must conform to the limits.

The following special rules apply to freight wagons:

- Noise remediation upgrades are not required. If modifications are made for other reasons, it must be demonstrated that noise emissions have not increased.
- If the performance of a wagon's brakes has changed as a result of modifications or upgrades, requiring the renewal of its operating license, the wagon must conform to the appropriate maximum limits given in the table shown above. However, if the wagon has been equipped with composite brakes and not with additional sources of noise, then no additional verification is necessary.
- If the wagon has been equipped with additional sources of noise, it must adhere to the current maximum limits for stationary noise emissions for new rolling stock.
- Until further notice, exemptions apply for Finland, Estonia, Latvia and Lithuania (see Noise TSI, nos. 7.7.2.2 and 7.7.2.4).

Limits to be tightened in the future

In section 7.3 of the Noise TSI it is explicitly recommended that lower limits be used for vehicles that are ordered 10 years after the TSI comes into force, or enter service after 12 years. The recommended reduction for multiple units is 2 dB (A) and for all other types of vehicle 5 dB (A). By 2013 at the latest, the process of revising the Noise TSI will begin. The outcome will be a tightening of the limits to a greater or lesser degree. Because of the fact that there are types of locomotives available today that already comply with the 5 dB (A) reduction in the TSI limits for a subset of noise types, it is probable that a differentiated revision of the limits will take place, with some limits being reduced by more than 5 dB (A).

As shown below (part 3.4) infrastructure operators and authorities which commission regional passenger rail services (PTA) may prescribe noise limits that are outside of the homologation regulations, or favour vehicles with certain noise standards. In those cases, it can be assumed, they will use the Noise TSI limits as their basis, or at least employ its methodology (definition of noise and measurement methods), if they intend to require differing values.

3.2 Immission limits

Immission limits for new or improved railway lines are the "traditional" type of rail noise regulation. Before the Environmental Noise Directive (END; see Section 3.3 of this paper) was introduced, only national regulations existed which probably implies that a lot of national solutions and approaches exist. In spite of that it seems that a lot of similarities exist and in this section mainly the German approach is described probably showing the typical problems and consequences of immission regulation for railway noise.

The Federal Immission Protection Law (Bundesimmissionsschutzgesetz) was passed in 1974 and comprises regulations for noise as well as for pollutants.[18] Details concerning traffic noise are given in a special decree from 1990 (Verkehrslärmschutzverordnung) which was last amended in 2006.[19] The decree gives the following immission limits which are differentiated for different areas and for day (6:00h a.m. until 10:00h p.m.) and night (10:00h p.m. until 6:00h a.m.):

Immission limits ("Beurteilungspegel" = "noise rating level")		
Type of area/building	Day [dB (A)]	Night [dB (A)]
Hospitals, schools, sanatoriums, retirement homes	57	47
Housing areas	59	49
Centre zones, villages, mixed zones	64	54
Industrial/commercial areas	64	54

The relevant locations for meeting the noise immission limits are the façades of buildings or external living areas. These values are relevant for road and railway (including tramway and metro) noise. The details given

[18] Bundesimmissionsschutzgesetz vom 15. März 1974, Bundesgesetzblatt I, S. 721 und 1193.

[19] Sechzehnte Verordnung zur Durchführung des Bundesimmissionsschutzgesetzes (Verkehrslärmverordnung – 16. BImSchV), Bundesgesetzblatt I, S. 1036; last changed by article 3 of the law of 19th September 2006 (Bundesgesetzblatt I, S. 2146).

in the following refer to the rail transport only. The noise limits are only applicable when a new railway line is to be constructed or when one or more main tracks shall be added. If other types of infrastructure investments lead to a noise level increase of 3 dB (A) or more or if the noise level after the investment equals or exceeds 70 dB (A) at day or 60 dB (A) at night, the immission values given in the chart above have to be respected after the construction has been completed.

The so called "Beurteilungspegel" (noise rating level) is rather calculated than measured because the noise immission has to be estimated in advance. This is part of the planning approval process. The calculation is based on the projected numbers of trains with detailed assumptions about the share of different train categories. Correction factors are applicable for

- Train categories
- Vehicle types (share of vehicles with disk brakes)
- Train length
- Train speed
- Different types of superstructure
- Damping because of ground quality or meteorological conditions
- Topographical situations, noise barriers, buildings, civil engineering works etc.

The technical rules for calculation and application of these factors are described in the so called "Schall 03", published by the former German Federal Railway in 1990.[20] A new edition is due to be published very soon by the German Federal Transport Ministry. With the new edition the methodology is amended and new developments in terms of train categories and vehicle emissions are regarded.[21]

[20] Richtlinie zur Berechnung der Schallimmissionen von Schienenwegen – Ausgabe 1990 – Schall 03

[21] Möhler/Liepert, Schall 03 2006 Richtlinie zur Berechnung der Schallimmissionen von Eisenbahnen und Straßenbahnen, 2008; Kurze/Moehler/Liepert/Onnich, The new German prediction model for railway noise „Schall 03 2006" – Potentials of the new calculation method for noise mitigation of planned rail traffic, 2008

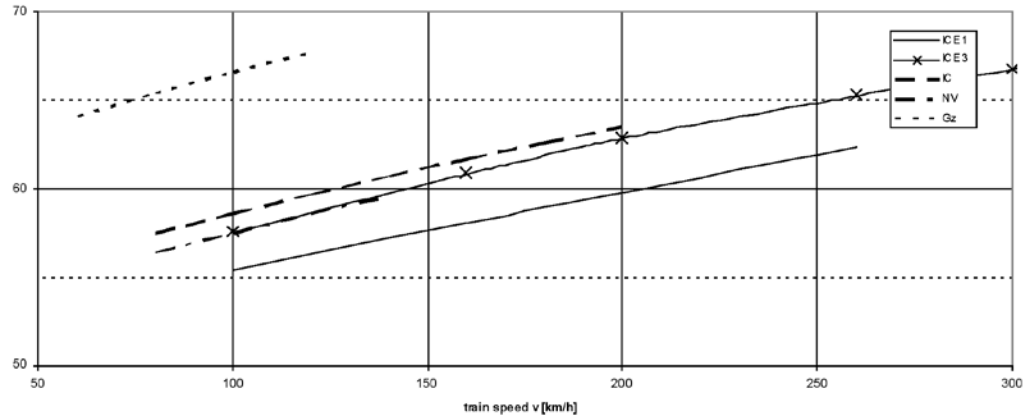

Figure 3.3 Dependency of velocity on calculated noise levels LpAeq at 25 m distance for different train types with 1 train per hour according to "Schall 03 2006". (Source: Kurze/Moehler/Liepert/Onnich, The new German prediction model for railway noise „Schall 03 2006" – Potentials of the new calculation method for noise mitigation of planned rail traffic, 2008, p. 188)

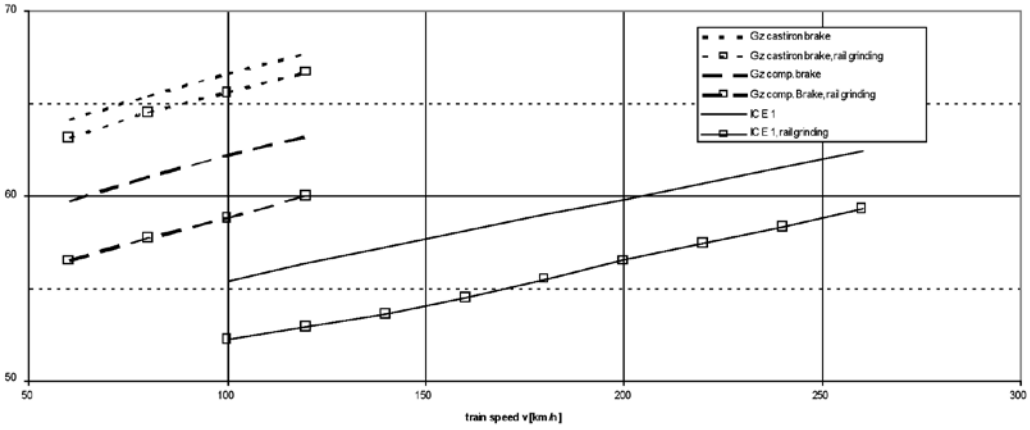

Figure 3.4 Calculated noise levels LpAeq for trains with cast iron brake and composite brake with and without rail grinding according to "Schall03 2006". (Source: Kurze/Moehler/Liepert/Onnich, The new German prediction model for railway noise „Schall 03 2006" - Potentials of the new calculation method for noise mitigation of planned rail traffic, 2008, p. 189)

After this calculation has been done, the so called "Schienenbonus" ("rail bonus") is applied, which means that the result is reduced by 5 dB (A). The rationale of the rail bonus is that several studies have shown that the annoyance for people living next to transport routes is somewhat less relevant for them, if they are confronted with the same measured physical

noise levels, when the noise comes from trains and not from roads. This is due to the quality of train noise which is different from that of roads. When a train passes by the noise increases slowly, is then very loud for some time and afterwards decreases again, while noise from roads is emitted continuously with sometimes very loud elements. The studies showed that a difference of about 5 dB (A) is reasonable which means that 70 dB (A) from roads are as annoying as 75 dB (A) from railways.

If the calculation of noise emissions shows that the immission limits will be exceeded because of the project in question, amendments have to be made or special noise protection measures like barriers, walls or sometimes protecting windows have to be projected.

In the context of planning approval processes for railway lines with open access it is difficult to refer to measures at the rolling stock, because the use of the line will not be able to be restricted to certain silent classes.

The planning reflects of course only forecasts about the noise emission of the train traffic. If the train frequency is increased afterwards without major investments, the neighbourhood is affected but has no clear instrument to urge the infrastructure manager to extend noise protection. On the other side, if train traffic decreases or noise emissions decrease because of new technical solutions, noise protection measures may become to a certain extent superfluous or even annoying in itself.

One consequence are noise remediation programmes like in Germany and in Switzerland. These programmes address existing lines where, in the case of Germany, the immission levels given in the *Verkehrslärmschutzverordnung* (see chart above) are exceeded. Recently, the budget for the German programme has been increased from 50 Mio. € per year to 100 Mio. € per year. The railways hope that at least a part of this amount will be used for the refurbishment of freight wagons because this would give substantial more comprehensive results in the short run than only erecting protection walls.

The Italian decree 459/98 gives in its articles 4 and 5 limits for new railway lines, distinguishing between lines for a maximum speed not exceeding 200 km/h (art. 5) and lines with a maximum speed of more than 200 km/h (art. 4). As in Germany the immission limits are differentiated according to the type of area and

building to be protected which supports the statement that such approaches are typical for the so far national noise immission regulations. However, indicators, definitions and actual values for the limits differ from country to country.

3.3 European Environmental Noise Directive (END) and noise action plans

The European Environmental Noise Directive provides a new approach of noise abatement. The consequences of this approach for the railways cannot exactly be foreseen today. The directive 2002/49/EC from July 25, 2002[22] requires authorities and member states in certain regions to measure and map noise pollution as a public service. Noise action plans are to be drawn up based on these assessments. The directive currently specifies which information shall be provided about noise pollution but the level above which noise pollution action plans have to be drawn up has to be decided upon by the member states. Furthermore, there is no European regulation to specify which minimum targets have to be attained. The directive only refers to national immission limits. Nevertheless, railway companies should assume that where railway traffic significantly contributes to noise pollution, they will in the future be required to limit the use of noisy railway vehicles or to take measures alongside the infrastructure. These requirements could be imposed in different ways, either legal, political or regulatory.

Above all, the following requirements are relevant for railway traffic:

Up to June 30, 2007, noise maps were required for
- Metropolitan areas with more than 250,000 inhabitants and
- Main railway lines with a frequency of more than 60,000 trains per year (equivalent to an average of around 3 to 4 trains per hour and direction)

Up to June 30, 2012, additional noise maps are required for
- Metropolitan areas with over 100,000 inhabitants and
- Main railway lines with a frequency of more than 30,000 trains per year (equivalent to an average of around 1.7 trains per hour and direction)

[22] 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

Compliance with the time limit for the first phase was not consistent. In the meantime however, many noise maps have been produced. Noise maps for the main railway lines in the German national rail network (DB Netz AG) are compiled and made available by the Federal Railway Authority. These cover 4,000 kilometres of track for phase one:
http://laermkartierung.eisenbahn-bundesamt.de/
http://www.eba.bund.de/cln_007/nn_204680/DE/Fachthemen/Umgebungslaermkartierung/laermkartierung__node.html?__nnn=true

Another example are the Dutch noise maps provided by the railway infrastructure manager ProRail which can be found here:
http://www.prorail.nl/Geluid/De%20Geluidkaart/Pages/Hoe%20werkt%20de%20Geluidkaart.aspx

Figure 3.5 Dutch Example for a strategic noise map with isophones (Source: www.prorail.nl/internetresources/geluidskaart/geluidkaart.htm)

The Environmental Noise Directive requires the unification of the national methodologies of noise-related information. It dictates to give information about two different noise levels – L_{den} and L_{night}. "den" in L_{den}" stands for "day, evening, night" which means that it is a noise level for 24 hours, combining three different measures, namely L_{day} (12 hours), $L_{evening}$ (4 hours) and L_{night} (8 hours). These measures shall be measured or calculated for all

days of the year and at the end average L_{den} and L_{night} should be given for a typical year in terms of climate conditions and noise patterns. The standard times are 7:00h a.m. until 7:00h p.m. (L_{day}), 7:00h p.m. until 11:00h p.m. ($L_{evening}$) and 11:00h p.m. until 7:00h a.m. (L_{night}), although the states can define different times provided that the pattern of 12 / 4 / 8 hours is respected and the same rules apply to all sources of noise. The respective noise levels have to be measured at or calculated for the most noise burdened façade of houses in a height of 4 m. Member states are still allowed to use their own measures and measurement methods but they are required to adjust their methodology to the requirements of giving reliable information about L_{den} and its components as well as to inform the Commission clearly about such variations.

The strategic noise maps are required to show, using intervals of 5 dB (A), which areas are affected by noise of the respective noise levels and estimations about the number of inhabitants living in those areas. The noise maps shall give detailed information on both L_{den} and L_{night} and show the contribution of each type of noise, e.g. railway noise, industrial noise, road noise etc. Where national noise immission limits exist, the strategic noise maps should also show where these limits are exceeded at present.

In metropolitan areas, the effects of noise must be determined for all railway lines when noise maps are compiled. The maps have to be checked and updated if necessary every five years, and additionally when there are significant developments in the noise situation. The competent authorities are obliged to draw up noise action plans. The concrete measures and the noise reduction targets are at the discretion of the authorities, but they should take into account the relevant limits of national law and most urgent areas in accordance with the results of the strategic noise mapping process. Quiet areas should be protected against increased noise.

The time limit for drawing up the plans is July 18, 2008 and July 18, 2013 respectively, according to the noise mapping phase. To date, there are no legal requirements as to which noise pollution reduction targets should be met. However, depending on local conditions (noise pollution, sensitivity of the population, density of the population, track noise), considerable reduction measures can be planned. These measures could affect the railways even though, taken by themselves, their noise emissions do not

cause the immission threshold values to be exceeded. In annex V of the directive 2002/49/EC, explicit measures for the following fields are given:

- 'traffic planning,
- land-use planning,
- technical measures at noise sources,
- selection of quieter sources,
- reduction of sound transmission,
- regulatory or economic measures or incentives'.

Based on these suggestions, or even exceeding them, it can be assumed that the authority responsible for the noise immission protection will choose one or more of the following possibilities for controlling the situation:

- Influencing the planning approval procedures when rail lines are upgraded or newly constructed;
- Agreeing track-side noise reduction measures with the infrastructure operator, and participate in the costs if necessary;
- Agreeing operative measures with the infrastructure operator, if necessary including different types of restrictions on the use of noisy railway vehicles;
- Imposing regulatory conditions on the infrastructure operator, for example noise contingents;
- Reaching agreements with the train operating companies (TOCs) that make most use of the line in question, or that are particularly responsible for the noise pollution;
- Imposing regulatory conditions on those TOCs;
- Reaching agreements with the Public Transit Authorities (PTAs) on measures for avoiding noise emissions (vehicle quality, changes to timetables, investments in infrastructure, other operational measures).

The most likely outcome from the point of view of a TOC is that it will be confronted with measures carried out by the infrastructure operator or by the PTA, i.e. it will in most cases be only indirectly confronted with environmental authority measures.

As far as heavily frequented rail freight lines are concerned, it will not be typical for the environmental authorities to require a TOC to carry out measures since particularly those lines are generally used by several TOCs.

The environmental authorities, however, can also directly contact the PTA over emissions originating from their regional passenger railways. Particularly for railways, which together with their infrastructure are owned by local or regional authorities, it is conceivable that they will be contacted directly.

Noise action planning is still a new instrument, and to date, no substantial experience has been gathered on its application. It is to be expected however that this instrument will see further development within the service life of railway rolling stock that is currently being procured. European legislation might also be extended to set clear targets for noise reduction, as will the regulations determining how much control the authorities have over infrastructure operators and transport operating companies.

The extent to which railways will be affected by noise action plans is not yet clear, but it is likely that the consequences will be severe, especially for lines which are heavily used by freight trains. Also commuter lines in urban areas with dense traffic will probably be affected, as dense traffic means a lot of trains and noise emissions and those lines serve agglomerations and cross a lot of sensible areas, like residence areas, schools, hospitals and so on.

Infrastructure Managers of busy railway lines will face attempts of the authorities to reduce the noise emissions of their railways. The solutions could be e.g. noise barriers, silent vehicles, measures at the superstructure, or noise differentiated track access fees. "Noise ceilings" have also been proposed. It means that only a certain noise level per day is allowed for a particular stretch of a railway line. When the actual emissions exceed this level, which could be measured at certain points along the tracks, all traffic (or at least freight trains) has to be stopped. This might cause serious problems for the infrastructure manager who could not provide for the train paths the railway companies have booked. Therefore, infrastructure managers will certainly try to avoid such interruptions by setting incentives for the train operating companies to use more silent vehicles.

Although 60,000 or 30,000 trains per year do not seem to be a very high usage of a train line, such lines contribute to the overall noise burden and may therefore be affected especially in agglomerations or in the neighbourhood of industry or roads. Such lines can even be relevant by themselves if the majority of the trains are freight trains and run at night. Additionally,

In general environmental authorities, public transit authorities, infrastructure managers and train operating companies should negotiate for a common approach and find the most effective and most economical way to reduce railway noise in the respective region.

3.4 What about the financial burdens of noise abatement?

In order to achieve a level playing field between transport modes, measures for noise abatement should be funded - at least partially - by the states. - That is a correct statement, but one should reflect that the European railways in general are dependent from the state budget. The infrastructure is usually to a great extent financed by the member states, sometimes with contributions from the EU-budget. Passenger operations which are considered as public services are usually financed or co-financed by local, regional or national authorities. In a lot of cases the governments also provide for the vehicles especially for passenger operations. If there are public funds for noise abatement in general, the railway industry should, of course, negotiate to get a high share for its own noise abatement purposes.

It is a matter of transport policy and of the bargaining position of the railways to which extent public funds are provided from the national, European or local budget, be it for infrastructure investments, vehicle procurement, operations or noise abatement. The better the image of the railways is the better is their bargaining position. That means that quality of service, environmental performance and the successes of noise abatement provide good opportunities. Railways are well advised to choose a pro-active approach for noise abatement and should be able to propose measures which are economical and effective.

A few general comments shall suffice in this chapter, some more detailed comments will be given in chapter 4.

In general, measures at the rolling stock are cheaper than noise barriers. Noise barriers have to be renewed after a few decades and although also measures at the rolling stock do sometimes not last for the whole lifetime of the vehicle, the life cycle costs (LCC) of noise barriers are usually higher than the LCC of rolling stock measures.

Measures at the rolling stock are often, although not always, cheaper than often thought. It is not justified to refuse the development of silent railway vehicles with the general argument that noise abatement would be too expensive. It is always necessary to look at the actual kind of vehicle and the actual noise problem. For instance if a vehicle is to be refurbished, noise reduction of 5 dB (A) might be easy and also economically viable, although not gratis, to achieve, while the 6th Decibel may be too complicated and too expensive.

A general problem is the financial relation between infrastructure managers and train operating companies. If train operating companies procure silent vehicles at additional costs they contribute to savings on the part of the infrastructure managers who can reduce the investments for noise barriers. Therefore, the infrastructure managers should pass the money they save to the train operating companies or the vehicle owners, e.g. by track access fee reductions.

4 "STATE OF THE ART" AND NOISE-AWARE STRATEGIES FOR VEHICLE PROCUREMENT, MODERNISATION, AND OPERATION

As mentioned above, measures at the rolling stock are usually cheaper than noise barriers. Furthermore, silent vehicles are silent everywhere on the network. Of course, the latter is only true if the tracks are in a sufficient state as otherwise the positive effects of silent vehicles may be balanced by negative effects of e.g. rail roughness or badly maintained rail joints.

Financing of silent vehicles could partially be done by incentives. These incentives could be noise-differentiated track access fees or incentives from the PTA which commission passenger services. Regulations on noise emission values should be focused on new vehicles. For economical reasons the railways should be able to keep old vehicles as spare capacity. This would not counteract noise abatement, if the railways restricted the operation of very loud vehicles to certain types of services, be it voluntarily or driven by incentives.

4.1 Railway vehicles: state of the art, recent developments and a mid-term perspective for the future

In recent years the railways and rail supply industry have made a big step

forward in terms of noise reduction. In the past noise reduction was only a lower-ranking criterion for well-designed railway vehicles. When the construction of high speed lines began the actual noise immission limits had to be respected for the neighbouring areas. This led to cost-intensive measures like noise barriers, artificial tunnels, cuttings, etc. In order to reduce the costs for these civil engineering works systematic attempts to reduce the noise emission of fast trains were made. The result was that high speed trains, although quite loud at high speed, are much quieter than conventional trains when running at "conventional speeds". In the German rules of "Schall 03" (as of 1990) the pass-by noise level of an ICE high speed train at 120 km/h is considered to be lower by 6 dB (A) than the pass-by noise level of a conventional loco-hauled Intercity train at the same speed.

When the attempts to introduce European or national noise regulations appeared, new methods of acoustic-aware design were due to be applied also for conventional railway vehicles. The result is that new vehicles which meet the requirements of the Noise TSI are more silent by 5-10 dB (A) compared to their predecessors.

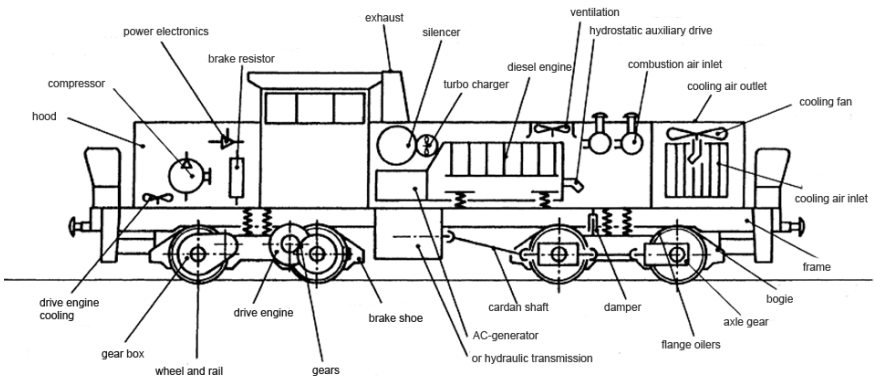

Figure 4.1 Main noise sources of a diesel locomotive; showcase with noise sources of hydraulic and electric power transmission. (Source: TU Berlin, Institut für Land- und Seeverkehr, FG Schienenfahrzeuge, Prof. M. Hecht)

Noise reduction is not gratis and the railways and the rail supply industry have been very reluctant to accept the challenge of systematically reducing the noise emissions of conventional rolling stock. In fact the costs of noise reduction on the vehicle side depend on strategy, targets, and circumstances.

If you have a conventional vehicle design and want to make it more silent afterwards, this might be a very expensive undertaking. If you need additional parts, this will probably mean additional costs to a smaller or greater extent. The cheapest way to design silent railway vehicles is to integrate acoustic management into the development from the beginning on. The questions to be dealt with are: Where are the sources of noise? Which are the ways the noise is transmitted to other parts of the vehicle? What are potential amplifiers for the acoustic vibrations?

Acoustic Management Process

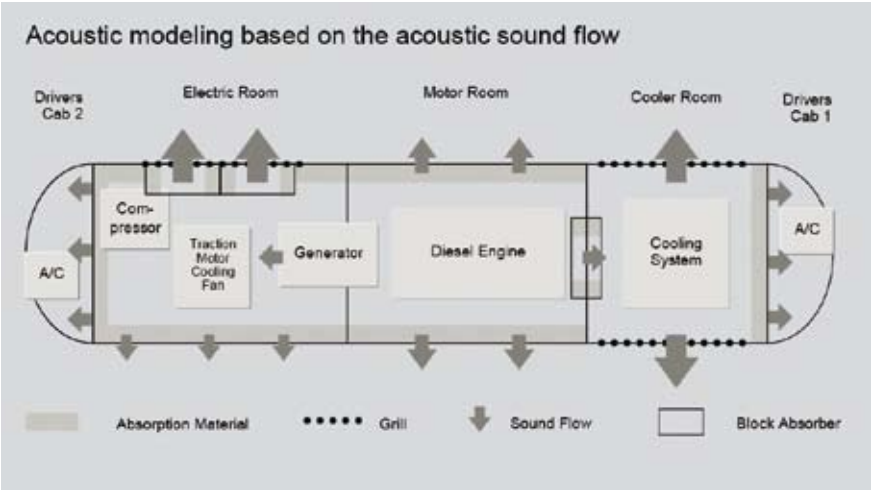

Figure 4.2 Acoustic modelling based on the acoustic sound flow. (Source: Siemens AG, 2009)

Channels of transmittance may be avoided or damped. A good design of air and exhaust flows can avoid noise. Sometimes already small amendments of the fan or the gear, the gear wheels, or the power transmission have a big positive effect. Spare space in the general vehicle concept makes it easier to systematically control the acoustic flows. It is still not an easy task and problems may occur when the weight allowance provides restrictions or when it is complicated to find reliable ways for leading off the heat.

It has been shown that the production costs of silent locomotives usually are not really higher than those of loud ones, provided that acoustic management has been done from the first phase of designing onwards. There may be a

Figure 4.3 Swiss Federal Railways (SBB) class Re 460 (SLM "Lok 2000"): the first "silent" electric locomotive (Source: Allianz pro Schiene / Pippert)

few additional parts for damping but the bulk of additional costs derive from the engineering process. Large-scale production thus reduces substantially the additional costs for noise abatement per single vehicle.

The early bird in offering a very silent locomotive was the Swiss SLM (Winterthur) which developed the very silent "Lok 2000" for the Swiss Federal Railways. Siemens did quite well with the electric "Taurus" locomotive and the diesel electric "EuroRunner", both classes designed for the Austrian Federal Railways (ÖBB), meaning that the Austrian noise regulations had to be fulfilled. (Actually the "EuroRunner" performed even better than the national regulations required.[23])

Figure 4.4 Austrian Federal Railways' class 2016 (Siemens Eurorunner ER 20): "silent" diesel locomotive, designed to meet Austrian and TSI noise limits (Source: Siemens AG, 2009)

Bombardier offers the "TRAXX DE" which has been proved to be even better than Noise TSI requires (see above, section 3.1.1 of this chapter). Also silent multiple units are successful on the market. Stadler has sold

[23] Friess/Marl/Schieber: The EuroRunner – an Environmentally-friendly Locomotive from Siemens, 2003, p. 22; Anhorn, How to design a low-noise locomotive? Concept and noise emissions of the locomotive Eurorunner 20, 2006

many FLIRT EMUs which are reported to be more silent than TSI requires. Alstom presented and realised a concept for low-floor multiple units. Most of the traction and ancillary equipment was placed on the roof thus giving space for both controlling the rolling noise and noise transmittance between wheels, gear and car body and for controlling the acoustic flows resulting from the aggregates on the top.

Figure 4.5 Prototype of Alstom hybrid locomotive, redesigned diesel locomotive of East German origin, now being tested in port areas and shunting operation (Source: ALSTOM Lokomotivenservice GmbH)

Hybrid concepts and energy storage for the operation on non-electrified lines and for shunting services are being examined at present in order to reduce energy consumption. These concepts have also positive effects on acoustics as "silent" energy can be used for start-up and acceleration at low speeds. Thus also the diesel engines may be scaled down which means that they are more silent. Several suppliers experiment with DMU concepts equipped with on-board storage of electric energy. Voith Thurbo is developing a power unit for DMUs with hydraulic storage of energy. General Electric and Swedish Train Technology are pioneers with energy storage for traction purposes on diesel shunting and freight locomotives. Alstom Stendal is testing a prototype of a shunting engine with a hybrid concept thus reducing the noise emission by 15 dB according to their own statements.

Although, in general, acoustic upgrading of existing locomotives is more complicated and expensive than procuring silent new vehicles, there are some positive examples. The Technical University of Berlin and the private railway company Havelländische Eisenbahn (HVLE) were successful in refurbishing a "Blue Tiger" diesel engine from Bombardier. With a few amendments especially concerning the fans it was achieved that the engine

noise of the locomotive now complies with the limits of Noise TSI. Only a limited amount of money was necessary and provided by the German Federal Ministry for the Environment.

Figure 4.6 Havelländische Eisenbahn (HVLE) class 250 (Bombardier "Blue Tiger"): example for redesigning an existing locomotive in order to meet TSI noise limits (Source: TU Berlin, Institut für Land- und Seeverkehr, FG Schienenfahrzeuge, Prof. M. Hecht or Havelländische Eisenbahn)

All in all it seems that the limits of the next stage of the Noise TSI which is announced to be set in force around the years 2016-2018 will be achieved, even if they come out to be stricter than assumed in 2006. However, a substantiated analysis should be done before decreeing new noise emission limits in order to make it neither too challenging nor too less ambitious. Instead of lowering all indicators by 5 dB (A) a differentiated approach seems to be adequate.

It could be questioned how far noise reduction should go beyond the 2nd stage of TSI Noise. That stage could be a future state of the art which will make it unlikely to go further with reasonable efforts. At least for passenger operation at conventional speeds other environmental challenges could become much more prominent.

The situation is much different, of course, for the prevailing problem of freight train noise. K-blocks brought a reduction by 8-10 dB. Some new bogie designs have the potentials for further 3 dB. The LEILA bogie has proved to be more silent by 9 dB compared to standard Y25 bogies with K-blocks, although international homologation has not been achieved yet (April 2009) and the reduction of production costs as well as generating sufficient demand for starting mass production are still a challenge. Even when this has be done the noise problem of rail freight transport may still exist and request a further reduction of 5 dB (A) and more. This is especially

true when green transport policy becomes true in the European Union with an accelerated modal-shift towards the railways.

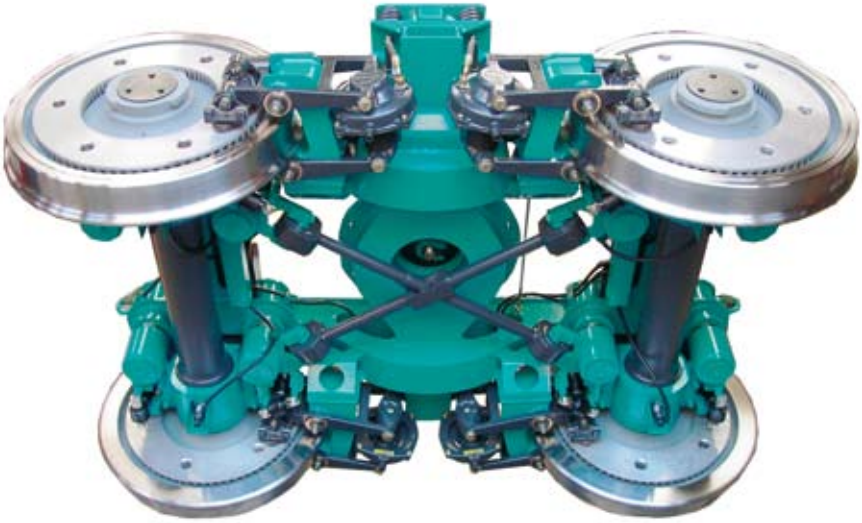

Figure 4.7 LEILA bogie designed by Technical University of Berlin and Josef Meyer AG Rheinfelden (Switzerland) for low-noise and energy efficient freight trains. Cross-anchors and inner bearings can be seen on the photo.
(Source: TU Berlin, Institut für Land- und Seeverkehr, FG Schienenfahrzeuge, Prof. M. Hecht)

But substantial progress in noise reduction will not at all be balanced by increase of freight volume on the rails. Doubling the number of trains means ***ceteris paribus*** an increase of noise emission by 3 dB while a noise reduction by changing brakes and bogies may reduce the emitted noise energy by 98-99 %. That means that a freight train with cast iron brake blocks is as loud as 64 freight trains with LEILA bogies.[24]

4.2 General comments on noise-aware procurement, modernisation and operation

First of all, noise emissions must not be neglected in the procurement process of new vehicles. It is always much more expensive to deal with

[24] Hecht/Keudel, In Messfahrten nachgewiesene Vorteile des LEILA-Güterwagendrehgestells, 2007

severe noise problems of a vehicle design afterwards. To procure silent vehicles is not only a good argument for marketing and public relations, but it is necessary to keep the good-will of the society for the railways and for modal-shift to this energy-efficient transport mode. When defining the specification sheet for a new series of railway vehicles one should consider that loud vehicles will lose value faster than more silent vehicles because during the vehicles' lifetime new regulations and requirements may lead to higher operation costs for loud vehicles and less opportunities for their use. A recent project of the German NGO Allianz pro Schiene, which was funded by the Federal Ministry for the Environment, showed that future differences in the development of the fair market value of second-hand railway vehicles allow for differentiated financing conditions provided by banks or leasing companies. Based on careful assumptions about the German market conditions, additional procurement costs of 1 – 4 % might be balanced by reduced financing costs if ambitious environmental criteria in terms of noise emissions, exhaust emissions and energy efficiency are respected.[25]

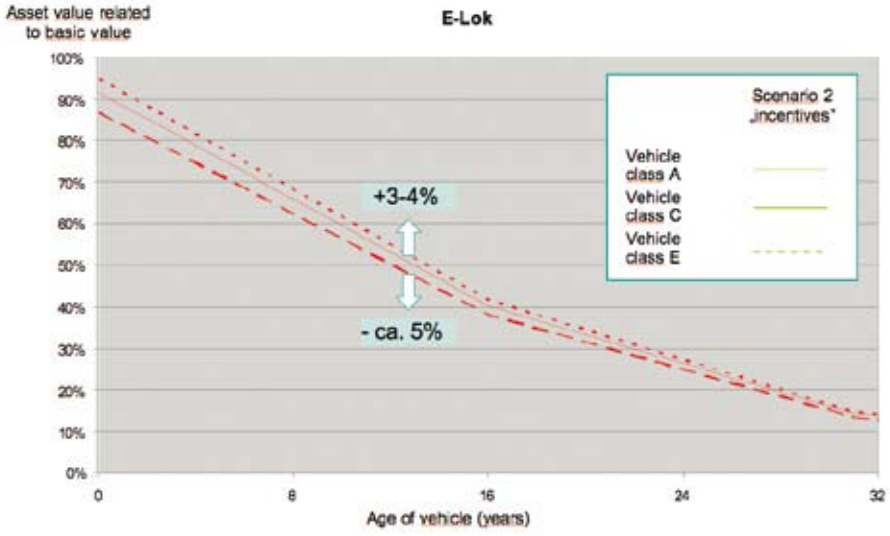

Figure 4.8 (Source: Allianz pro Schiene / SCI Verkehr GmbH)

[25] The project was finished in spring 2009. The results will be provided on http://www.allianz-pro-schiene.de/deutsch/Themen/Umwelt/Fahrzeugfinanzierung/ but have not yet been published at the time of writing this paper.

Noise remediation of existing vehicles is usually difficult. The special situation with freight wagons has been discussed in section 2.3 of this paper. As far as locomotives are concerned, there are examples for reductions of 2 – 4 dB with minor investments but this always means additional engineering and often the need of new homologation procedures.

When existing vehicles are substantially refurbished, systematic noise-aware design should be done, although the remaining structure of the old vehicle will set limitations. TSI Noise at present requires documentation that in such cases the noise emissions have not been increased, but certain enhancements in terms of noise reduction seem always possible and desirable in case of such refurbishments.

Even measures in operation have a potential for noise reduction, although not as big as with noise-aware procurement and design. Most of these measures are similar to the requirements of energy-efficient driving, meaning that the actual maximum speed is reduced and unnecessary acceleration and braking is avoided.

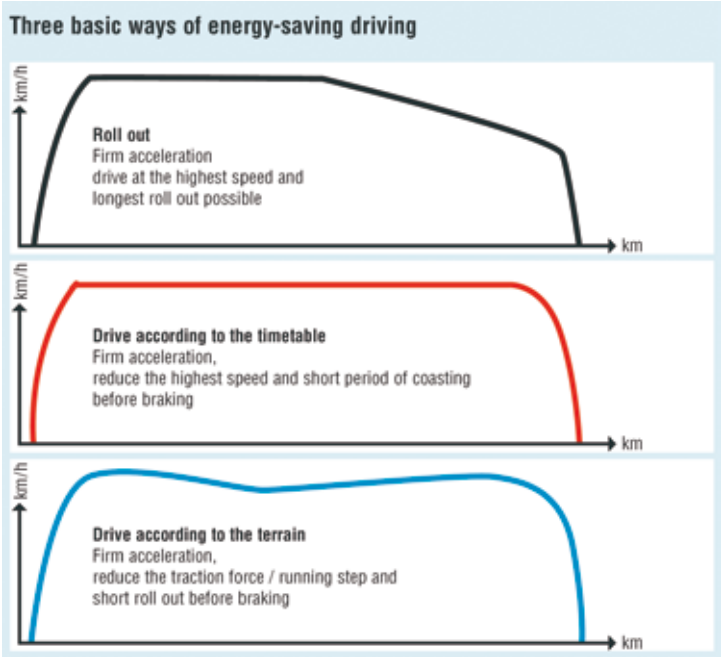

Figure 4.9 Different styles of energy-efficient driving. "Eco-driving" has also acoustic advantages for the railways. (Source: Deutsche Bahn AG)

Further options are to analyse and control noise emissions at maintenance facilities or from vehicles temporarily not in use (stand-by functions).

And if you think that your railway company does not have noise problems at all because of very thin traffic flows, then think of one German private railway with only one pair of not very long freight trains per day. The railway ceased to have noise problems with the population of a near-by residence area with typical middle-class houses, after they changed the timetable and thus not any longer woke up everybody with acoustic level crossing signals - each morning at 4 o'clock.

5 PERSPECTIVES AND CONCLUSIONS

Railways should be aware of noise emissions. That is essential for traffic growth on the rail, and achievable at reasonable costs. Major progress has been made in recent years, but a lot of new solutions have not yet come to every-day-use. Due to the long lifetime of railway vehicles, it may take decades until enhancements in terms of noise reduction are effective on the whole network. Nevertheless, a an overall reduction of railway noise in Europe by 10 dB (A) or even more seems to be achievable within the next ten years, although the reduction depends to a large extent on the effective public funding for the refurbishment of freight wagons with K-blocks or even better solutions.

Although the costs of procuring silent vehicles are not as big as often considered in the past, public funding is necessary, especially for R&D, retrofitting older vehicles, and incentive schemes. Public funding for the railways and their competitiveness is indispensable for climate protection policy in transport. Public funding for noise reduction helps the railways to keep and even extend their lead in terms of environmental performance. Authorities and members of parliaments should consider that measures at the vehicles although often considered to lie in the responsibility of the railway companies can substantially reduce the amounts for public investments in the railway infrastructure because noise protection measures like barriers, walls and artificial tunnels can be avoided. Public expenditures for silent railway vehicles, especially freight wagons, will certainly lead to reduced public expenditures for railways and noise protection in the long run.
In spite of this request for public funding, the railways should, as part of credible and responsible management strategies, integrate noise awareness in all steps of procurement, designing and operation.

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PART 3:
Lucchini RS products for railway noise reduction

Steven Cervello

Part 3: Lucchini RS products for railway noise reduction

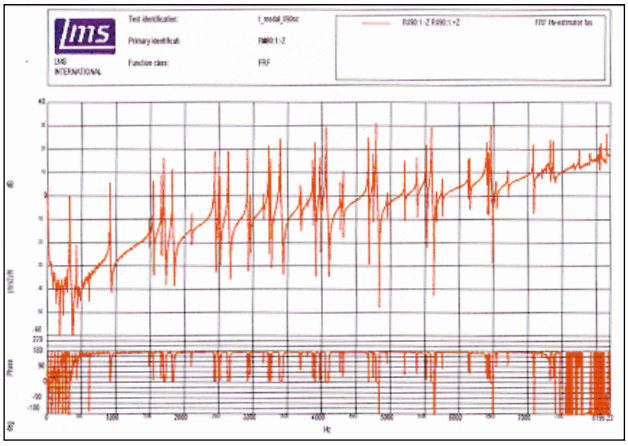
Steven Cervello

1 THE LUCCHINI RS R&D DEPARTMENT

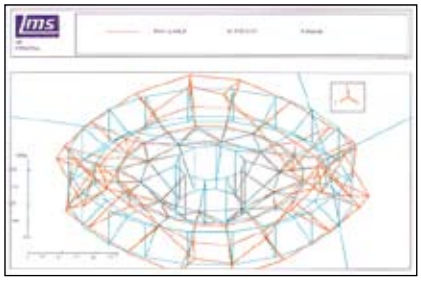
When noise became an issue which the people started to complain about and the city administrators started to fix limits on, Lucchini RS understood that in the future low noise wheels could become a 'must' and that an investment in new R&D projects was necessary.
The first step was to set up a fully equipped laboratory of vibro-acoustic, comprising all the instrumentation available at that time for the complete characterization of the noise emitted by wheels. A semi-anechoic chamber was designed, built and tested.
The laboratory is today equipped with an 8 channel LMS Scada system capable of performing structure modal analysis and a customized, portable dynamic channel acquisition system that is also used for field measurements like pass-by acoustic tests. Further equipment are various kinds of accelerometers, microphones, impact hammer and an electrodynamic shaker.
The scope of the laboratory was obviously to help understanding the vibro-acoustic phenomena of wheels and develop innovative methods for reducing wheel noise emission but also perform a sort of qualification in controlled conditions of wheel performance and so to compare the standard wheel with the damped one providing an acoustic parameter characteristic of the wheel that would enable the customer to choose the best solution at a stage where it would not be possible to perform in-service tests as the vehicle would still be under development.
The first experimental activity was a full modal analysis of a ETR500 trailer wheel suspended and free to vibrate (Fig. 1.1). Around 70 acceleration measuring points were

Fig. 1.1 The first modal analysis performed on the ETR500 trailer wheel

used and the excitation was given with an instrumented hammer in the radial direction (on the rolling plane) and in the lateral direction (on the flange).

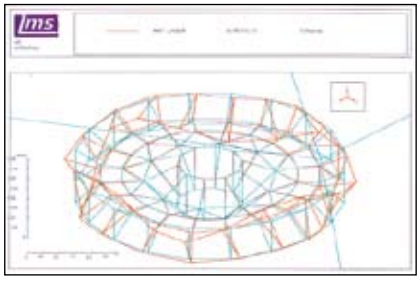


Point Frequency Response Function from 0 to 8 kHz

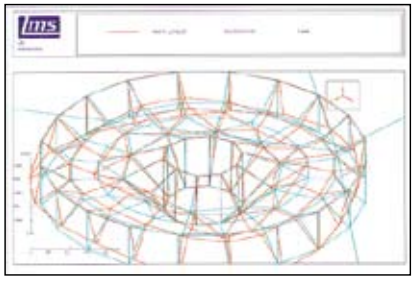


Rim bending mode at 910 Hz

Radial mode at 1485 Hz



Rim bending mode at 1660 Hz



Radial mode at 2102 Hz

Fig. 1.2 Examples of experimental modal analysis results on a ETR500 trailer wheel

The result was a very fine modal analysis that identified frequencies, damping factors and especially mode shapes.
The main understandings were that two main mode shape families could be identified:

- radial modes – looking at the wheel section, the rim tends to rotate and accordingly the web tends to move axially; these modes are excited by both radial and axial forces; in some particular cases the rim moves simply in the radial direction with a similar effect on the web. All these kind of modes together characterize the so-called rolling noise.
- rim bending modes – looking to the rolling surface of the wheel, the rim tends to bend generating a number of waves on the circumference that increases with the resonance frequency; differently from the radial modes, in this case the web is not moving; for this reason these modes are excited more by lateral forces that take place while running through curves.

With these basic concepts in mind it was possible to design damping solutions and decide how they should interface with the wheel structure. Defining the interface was an important point as a damper to function should absorb the energy of the vibrating structure and in the case of a damping material it means that the movements of the wheel are transferred to the material so, as a consequence, it will start to dynamically deform and dissipate this mechanical energy in the form of heat.
As an example, a damper applied to the web would not dissipate the rim bending modes, whereas the damping of the rim would be effective on the majority of the modes.
To have a better idea of the performance of a damping solution, acceleration measurements seemed sometimes not to be complete enough as the result of the measurement depends on the specific point where the accelerometer is applied; also wanting to reduce the number of measurements, a microphone measurement at a couple of meters from the wheel seemed to give more complete information as the pressure at that point would depend more on the complete way of vibrating of the wheel surface. This is not actually true as the wheel emits noise with a specific pattern or directivity diagram; for this reason it was decided to define a new measurement setup procedure made of an arc of 8 microphones (Fig. 1.3 - 1.4) and from them to calculate the Sound Power emission of the wheel.

Fig. 1.3 - 1.4 The Lucchini RS semianechoic chamber with the arc of microphones installed

The vibro-acoustic characterisation procedure developed by Lucchini RS enables the comparison of different low noise wheels.
The parameter chosen to qualify and quantify noise emission is the sound power. The sound power emitted by a source can be measured through sound pressure measurements on a grid as described in the ISO 3744, sound intensity measurements at discrete points (ISO 9614) and sound intensity measurements by scanning (ISO 9614-2).
The ISO 3744 approach was chosen, as the new laboratory is a semi-anechoic room certified to simulate free field conditions over a reflecting plane for frequencies higher than 250 Hz. This is a basic condition for the ISO 3744.
Here the sound power which depends on the exiting force is normalised by this force and it is measured in dB10 [re pW/N^2].
The wheel hub leans through a rubber block on a support (40cm high from the ground); the wheel plane is horizontal.
The exiting force is an impact made with an instrumented hammer in a fixed position on the rim rolling surface, in the radial direction.
The measurements are done in the frequency range of 250 - 6000 Hz by 8 microphones mounted on a vertical arc (90° and r = 2m). The arc is turned in 5 positions (every 45°) so to cover a half hemisphere with 36 measurement positions.
We were aware that the sound emission of a wheel suspended and free to vibrate was not completely representative of the condition observed in service. We could expect that as the wheel is rotating on the rail, it would receive an extra damping and that for this reason the damping performance of an applied damper should be far higher than the damping provided by the wheel rotation.

Field measurements were then necessary to prove the final performance of the proposed solution; in order to complete the characterization the low noise wheels the laboratory measurement equipment was completed with a portable multi channel acquisition system for microphones and accelerometers.
Nevertheless field measurements are always complicated and expensive to organize and normally would be done on the final chosen solution.
In Lovere in 1994 a very special test rig was developed: the BU300, a roller rig for full-scale wheelsets, is able to run up to 300km/h and to simulate very realistic running conditions.

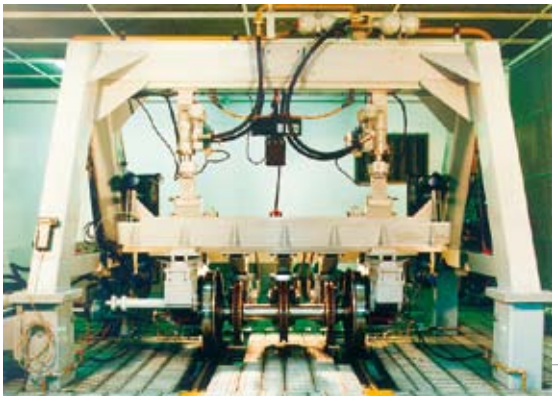


Fig. 1.5 BU300 roller rig

As shown in Fig. 1.5, a complete wheelset including the primary suspension, can be mounted under a beam on which vertical, lateral and longitudinal actuators apply dynamic forces able to reproduce curving and straight running conditions wile the wheels are rolling on two discs of two meters of diameter and with externally the profile of the UIC60 rail.
This test rig has many applications and has been also successfully used for acoustic tests.
Due to the fact that the test rig itself is very noisy during operation, Sound Pressure Level would not be possible carried out, so it was decided that more reliable measurements could be done by accelerometers applied on the wheel surface; the cables were then introduced by a radial hole in the axle bore and at one end connected to a special multi channel slip ring.
The acceleration measured on the rim by the accelerometers gives only local information of the vibration behaviour of the wheel. To have an idea of the Sound Pressure Level produced at a certain distance it was decided to estimate

the Sound Power Level emitted by the sound source in the hypothesis that a major part of the wheel surface would be vibrating like the point in which the accelerometer is installed; obviously it is just an approximation.

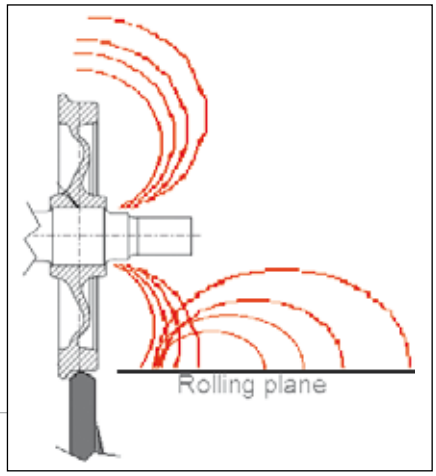


Fig. 1.6 Scheme propagation of noise emitted by the vibration of the wheel

The Sound Power can be calculated from the speed movement of the vibrating surface in the direction normal to the surface itself.

From the Fast Fourier Transform of the measured acceleration it is possible to derive the speed spectra of the vibrating surface as:

$$v(f) = \frac{1}{f} \cdot a(f)$$

In the simple hypothesis that two thirds of the wheel web/rim surface is vibrating in the same way as where the measurement has been performed, it is possible to calculate the corresponding power emission as:

$$W(f) = \rho_0 \cdot c_0 \cdot \sigma \cdot v_i(f)^2 \cdot S_i$$

Where σ is the *radiation efficiency* which we considered equal to 1 (usually $0 \leq \sigma \leq 1$)

The Power Level has is calculated by:

$$L_W = 10 \cdot \log_{10}\left(\frac{W(f)}{W_{ref}}\right)$$

with $W_{ref} = 10^{-12}$ W

Considering for example a microphone in free field, at a distance D from the wheel, it would read a Pressure Level that in the hypothesis of a source near to a solid wall (see Fig. 3. 6) is:

$Lp = (L_W - 20 \cdot \log_{10}(D) - 11) + DI - 3$

DI = Directivity Index in this case it's equal to 6

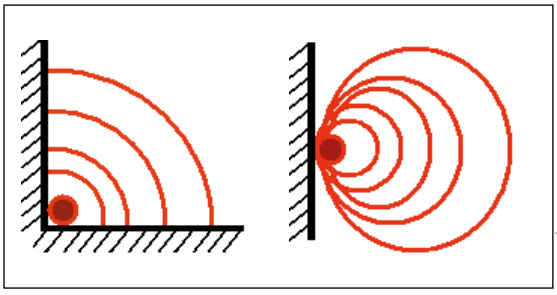


Fig. 1.7 Sound Source near a solid wall, side and above view

This last described method is the nearer attempt to experimentally simulate the real running conditions of the wheel under the vehicle.

Throughout the last 15 years at the Lucchini RS laboratories, various solution were developed and tested some of them were actually successful and are today on the market or going through the final qualification and acceptance from the railway authorities.

Starting from the beginning, the fist tested ideas were the ring-in-a-groove damping treatment, at that time, recognized as useful only to reduce squeal noise and its efficiency was expected to be low; nevertheless the analysis of different ring damped wheels proved to be good training for both the researchers and the entire equipment, especially in terms of data processing to find out damping and sound power.
Times were mature in 1997 to start the development of a different type of damping treatment that was named after the Greek goddess of silence: *Syope*®.
The following chapters describe the experiences made during the development of the main solutions, the *Syope*® and *Syope*® Braw, the *Galene*® and the *Hypno*®.

Syope®

Syope®
braw

2. THE *Syope*® TREATMENT

2.1 Introduction

In 1998 Lucchini RS completed the development of the so-called Syope wheel, a noise-damping treatment that could be applied to any kind of solid wheelset.
Named after the ancient Greek word for 'silence', Syope offers the optimum combination of noise reduction and product reliability. It consists of a constrained damping plate of aluminium or steel that is fixed in place using a visco-elastic bi-adhesive polymer.

The well-known constrained layer damping solution was identified as the most convenient as:

- the treatment was expected to be automatically approximately equally effective on all wheel eigenmodes, while other treatments need to be 'tuned' on the specific wheel eigenmodes. Similarly, the treatment was expected to be relatively independent from the wheel tyre thickness modification after re-profiling, that significantly changes wheel modes (see a later paper, [1], as proof of this statement);
- the treatment is self-centring and radial centrifugal forces are supported by the metallic panel and not by polymer, virtually eliminating any stress during the service;
- it was expected to possibly apply the treatment to any axial symmetric wheel without modifications to the current geometry, virtually without changing the design and applying the treatment to the replacement wheel;
- the peculiarities of the treatment let suppose that no structural modifications were introduced in the wheel and that no safety analysis would have been necessary to be performed on the treated wheel.

At the same time, Lucchini RS was aware that:

- it was necessary to find a polymer with extremely good properties, considering the expectations in terms of lifetime in service of a wheel (up to five years) and the extremely demanding conditions of the railway environment;
- the only possible mounting of the wheels on the axle is the press fit one, while shrink (hot) fit normally in use in Italy was not possible as the polymer could not survive to the 200-250° C heating;

- the treatment is only possible for disc-braked wheelsets as block braking introduces too much heat in the wheel tread that is transferred to the wheel web damaging the polymer.

All of these problems were carefully considered and Lucchini RS started the development of the *Syope*® treatment initially as a premium solution for high speed trains. Only after many years would the market appreciate the product as a valid solution for ordinary service and for light railways.

2.2 Description of the *Syope*® treatment

Basically, the treatment consists of a steel or aluminium layer constraining a special adhesive polymer sheet (Fig. 2.1 Scheme of the Syope wheel). Polymer was developed by 3M for aerospace applications and was selected after a careful evaluation of mechanical, chemical and physical properties. Such a polymer based on acrylate technologies has several important features that gave many industrial applications the possibility to solve not only sound reduction but also bonding aspects and sealing needs.

Polymer has a special structure which is completely homogeneous in every part, this structure has, during the application process, micro movements that fill all the micro profile of the materials involved with a strong improvement in performances also just after a few minutes from the application. This aspect can also solve many of the problems due to thermal differentials, especially when the materials bonded together are not the same and then are affected by different contractions or expansions; this is one of the reasons for its choice in aerospace applications where the thermal variation is wide and fast.
The special polymer also protects the surface of the wheel web from corrosion, and has a high and constant adhesive resistance over time. Its ability to withstand harsh conditions, extremes of temperature and humidity and substances such as fuel, alcohol and salt has been demonstrated during an extensive programme of tests. Trials included adhesion resistance tests, accelerated weathering, outdoor weathering, thermal cycling and fatigue resistance.

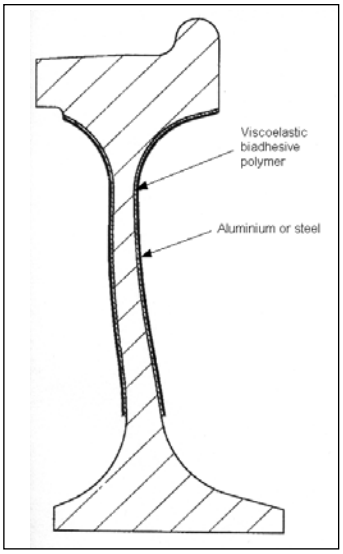


Fig. 2.1 Scheme of the Syope wheel

2.3 Laboratory Vibro-acoustics Tests

The normalized sound power emitted at each natural frequency was determined by using the already mentioned procedure. It consists of collecting the output of an array of microphones mounted on an arc centred at the wheel centre, with the wheel resting on a "soft" support and giving an impact with an instrumented hammer. Ensuring that the highest natural rigid frequency of the wheel on its support is lower than 1/3 of the lowest elastic mode of the wheel results in the so-called "free-free" response of the wheel, e.g. the impact gives an initial speed to the excitation point but the wheel is then free to vibrate without either other excitation or constraints. As the floor is reflecting, the entire sound energy is measured by the microphones.

The output from all the microphones is properly added to estimate the power output from the hemispherical surface defined by the rotation of the array around the wheel axis. Using the reciprocity theorem, the arc of microphones was kept standing while the impact point was moved around the circumference. The total power was normalized with the excitation, resulting in the estimation of reduction of noise from the wheel in service (Fig. 2.2).

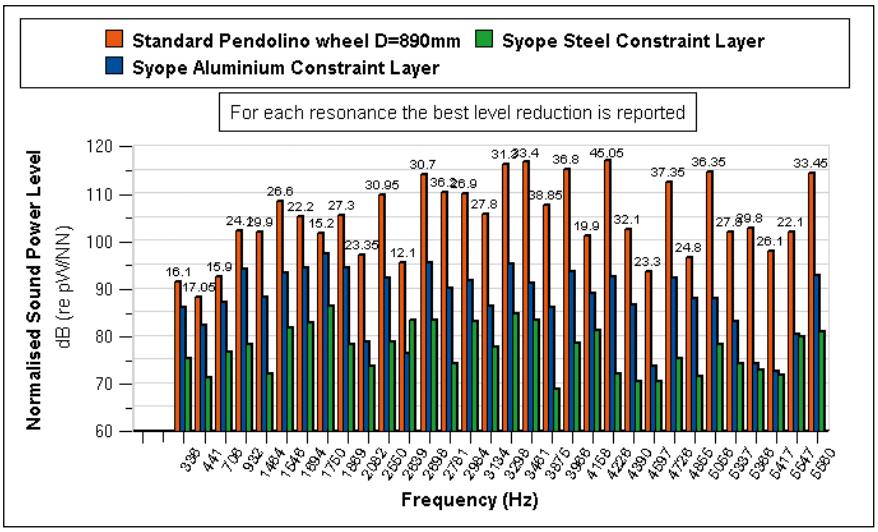


Fig. 2.2 Normalized sound power levels obtained in 1997 for the Italian Railways high speed wheel (monobloc, curved web, d=890 mm) with steel and aluminium constraining plate. Reductions obtained at each frequency with steel plate are shown

These data were considered very promising but were believed to be only partially reachable in practice for the already mentioned existence of "rolling damping". Working alongside with Fiat Ferroviaria, a set of line tests was therefore planned with the test trainset ETR470-0.

In the mean time a full scale test was organized on the roller rig BU300, two accelerometers were fixed on the wheel rim and measured through a multi channel slip ring (Fig. 2.3).
From there an estimation of the sound power pressure at 7,5m was performed and then compared with the in-service pass-by tests made on the ETR470.

Fig. 2.4 shows the comparison of standard and Syope wheels first through microphone sound pressure level measured at 7,5 m from an ETR500 passing by with 4 bogies equipped with Syope wheels and the rest with standard wheels and then through an estimation of sound pressure at 7,5 m by accelerometer measurement on the wheel rotating on the test rig.
At a first analysis the absolute values for each measurement method are quite different, but this is quite obvious considering that in the case of pass by tests the measurements are influenced by the various sources of the vehicle whereas the accelerometer measurement are influenced only

by the wheel itself, secondarily the rail stiffness of the BU300 roller rig is constant and in any case different from normal railway tracks made of ballast and slippers.

Fig. 2.3 Syope wheel installed on the BU300 roller rig, on the lower part of the rim there are the accelerometers mounted.

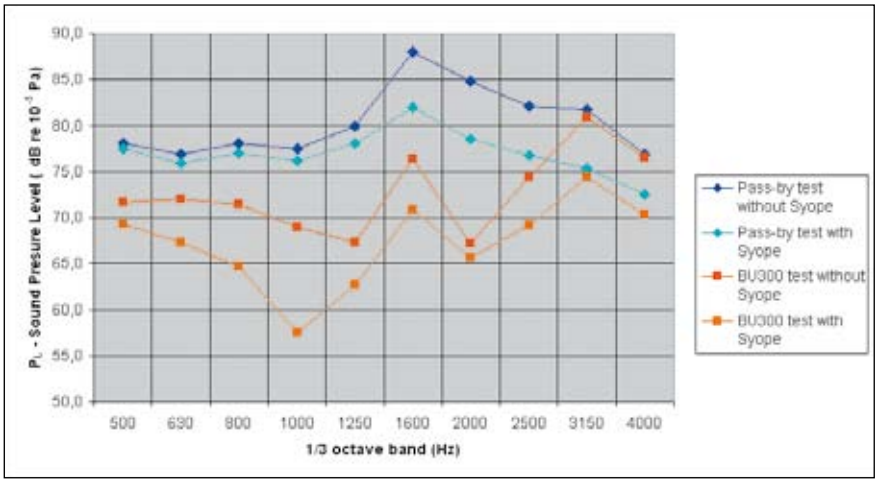


Fig. 2.4 1/3 octave band spectra of Sound Pressure Level at 7,5 m of the Syope and Standard wheels running at 160km/h; comparison between field and test rig tests

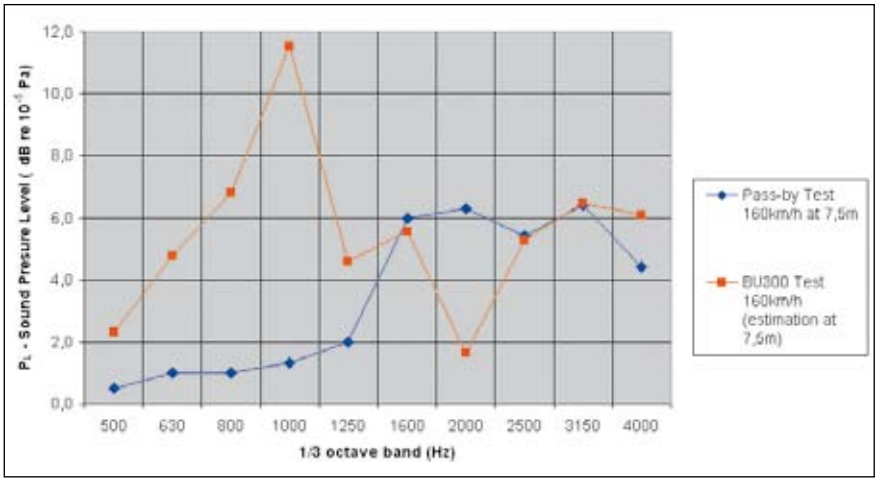


Fig. 2.5 Sound Pressure Level Reduction at 7,5 m of the Syope compared to Standard wheels running at 160km/h; comparison between field and test rig tests

Nevertheless if we compare the difference between Standard and Syope wheels (Fig. 2.5), it is interesting to recognize that for frequencies higher than 1,2 kHz the differences in the levels are quite similar except for the 2 kHz band that is actually a band with no wheel resonances as it can be seen from the narrow band spectra (Fig. 2.6).
For lower than 1,2 kHz frequencies, the difference at the roller rig is much higher than in service, but it should pointed out that in this frequency range, the influence from the track is more important than the influence of the wheel, obviously this effect is not measured on the test rig.
In any case the test performed on the BU300 test rig was useful to estimate the reduction level that can be obtained with Syope wheels taking into account the effect of rolling damping and for higher range frequencies the level reduction estimation can considered very reliable.

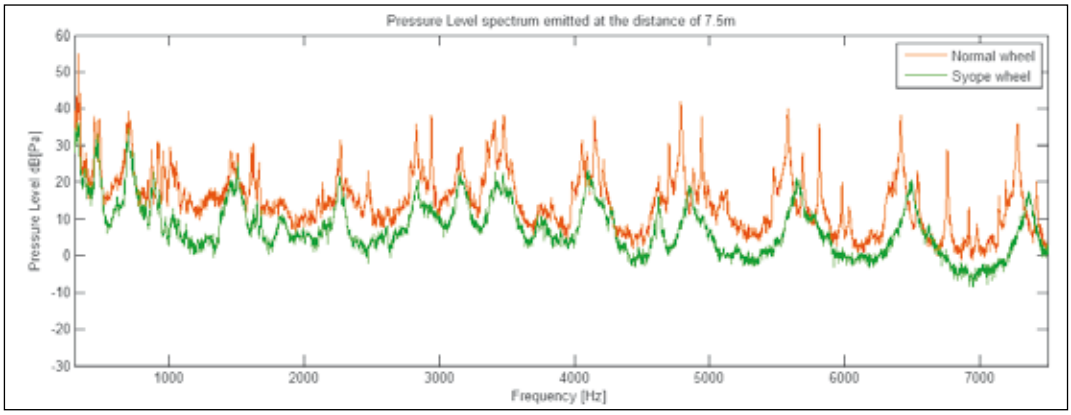


Fig. 2.6 Narrow band Sound Pressure Level estimated at 7,5m from accelerometer measurements on the wheel while rotating on the roller rig at 160 km/h

2.4 1997: high speed tests with ETR470-0 trainset

ETR470-0 is a tilting train of the Pendolino family that Fiat Ferroviaria was using at that time as a test train to develop different technologies. Lucchini RS supplied 4 wheelsets with *Syope*® treatment with steel constraining plates. Test runs up to 250 km/h were performed and noise emission was measured with conventional standing microphones and with a linear array of microphones in order to horizontally separate the sources (Fig. 2.7).

Fig. 2.7 Left to right: experimental Syope® wheels on the ETR470-0 trainset, trackside microphones during a 250 km/h pass-by, preparation of linear array of microphones

A pass-by at 220 km/h is shown in Fig. 2.8; the presence of an additional source, installed to develop a computer program able to remove the Doppler effect and to identify the wheels contribution, emitting noise at exactly 2 kHz is evident near the front driver's cab. Noise reduction during a pass-by at 220 km/h is shown in Fig. 2.9. Although it is evident that the exceptional results obtained in the laboratory tests were not reached, the around 5 dB(A) reduction was considered very promising and the results were presented to FS in May 1999.

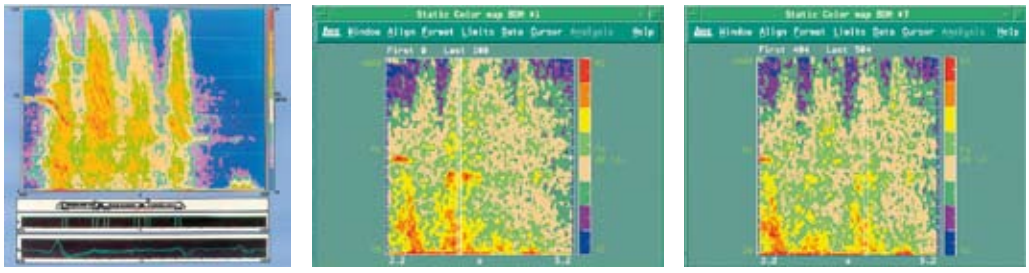


Fig. 2.8 Left to right: time-frequency analysis (spectrogram) average on 21 microphones, analysis of a standard wheelset and analysis of a Syope® wheelset

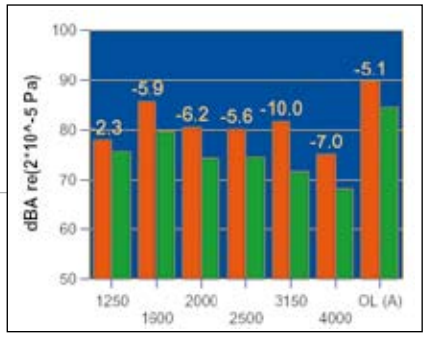


Fig. 2.9 One-third octave band and overall noise levels measured during a pass-by at 220 km/h of the ETR470-0 trainset. Data for untreated (red) and Syope® treated (green) wheels

2.5 2000: HIGH SPEED TESTS WITH A ETR500 PLT TRAINSET

2.5.1 Introduction

After the results obtained with the ETR470-0 trainset, the Rolling Stock Technology Unit of the Italian railways (Trenitalia Unità Tecnologie Materiale Rotabile - UTMR) decided for a more extensive and precise noise measuring campaign. At that time a trilateral cooperation of the main European Railways (DB, FS, SNCF) was setting up a high speed campaign in the three countries to investigate aerodynamic drag at high speed and the influence of bogie fairings to reduce energy consumption of high speed trains.
Times were right to test the *Syope*® wheel on the new multivoltage ETR500 PLT trainset number 51, and Trenitalia contacted the University of Florence to perform on-board and trackside noise measurements on their behalf. A great advantage came from the fact that also technicians from DB were in charge of measuring trackside noise with a spiral array of microphones that was able to efficiently separate the sources in both vertical and lateral directions.
The train was prepared in order to solve one of the main limitations of the ETR470-0 campaign, e.g. the different roughness of the standard and *Syope*® wheels. Most of the wheels (approx. 75%) were reprofiled in the Milano Fiorenza workshop, and tests on the Roma-Firenze Direttissima line started two weeks after, when the train had run approximately 3000 km. Incidentally, this is the same distance that will be recommended [2] around five years later.

The coincidence of the aerodynamic drag test campaign allowed the testing of all the combinations of the following parameters:

- speed, in the range 195-300 km/h;
- fairings (with special acoustic treatment);
- *Syope*® wheels;
- reprofiling.

2.5.2 Results

The results were processed and published independently by Trenitalia - University of Florence [3] and by DB measuring group [4] and can be

summarized as follows: in the 190÷295 km/h range, the use of *Syope*® treatment reduces noise by 4 to 5 dB(A), offering similar or better performances compared to fairings. Speed confirmed its importance while turning the wheels reduced by approximately 2 dB(A) the noise emitted by standard wheels. The advantage offered by *Syope*® wheels is better at the lower limit of the speed range tested, as fairings are particularly efficient at the higher speeds where aeroacoustic noise becomes prevailing.

The train composition and the typical noise "signature" from the pass-by are shown in Fig. 2.10, while photographs and results are shown in Fig. 2.11 and in Fig. 2.12.

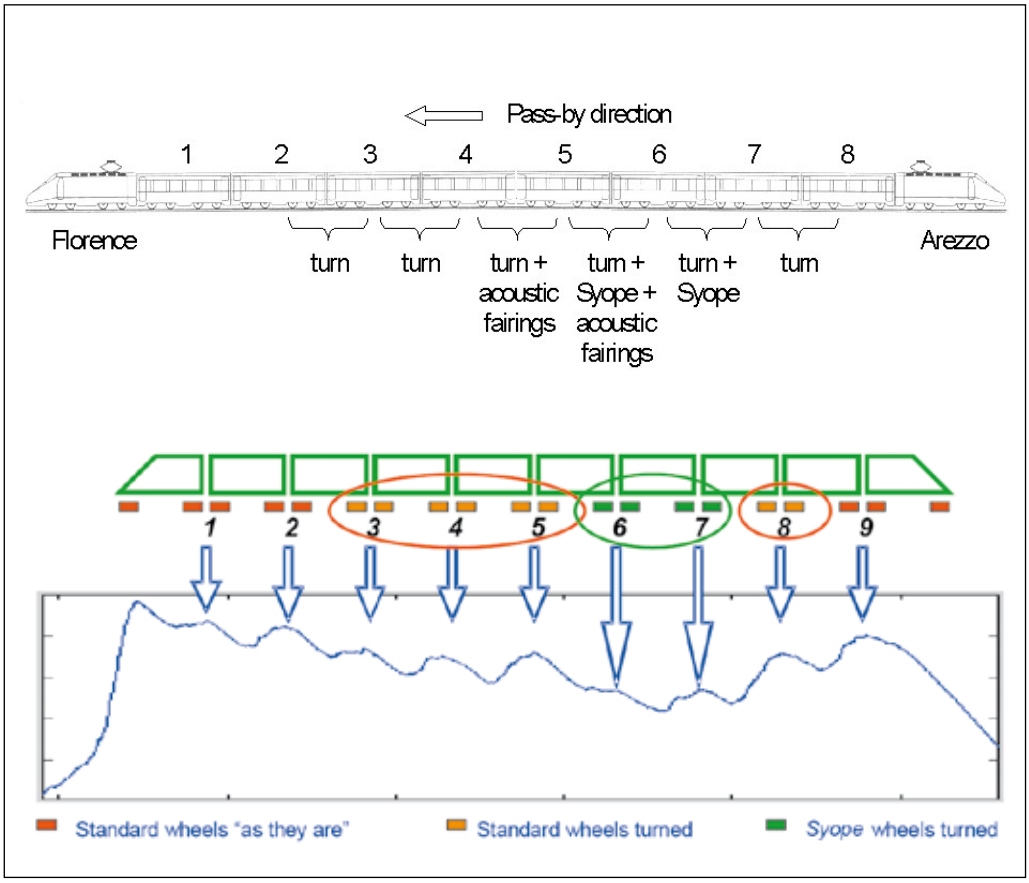


Fig. 2.10 Composition of the test ETR500PLT-51 trainset (left) during the test campaign in Renacci, 2000 (left). The typical noise signature (LpAmax,F) at 7.5 m of a pass-by is shown on the right, where the effect of wheel reprofiling and Syope® wheels is evident

Fig. 2.11 Some images from the test campaign in Renacci, 2000, on the ETR500 PLT trainset 51. Left: train passing by in front of the spiral array of microphones by DB; mid: axle box noise measuring device developed by University of Florence in front of a Syope® wheel

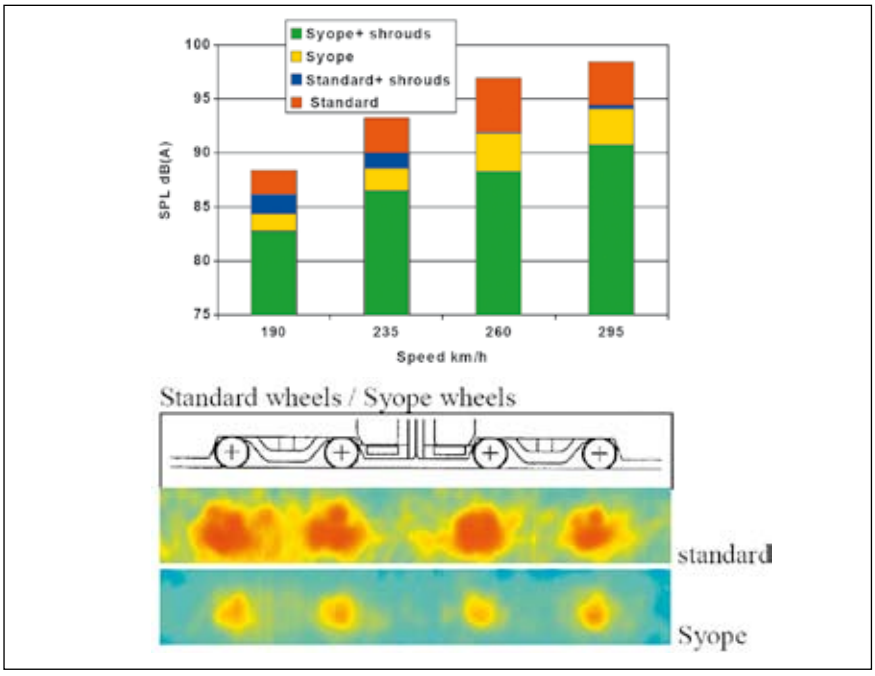


Fig. 2.12 Noise emission comparison between standard and Syope® wheels, found by (left) University of Florence - Trenitalia and by (right) DB

After the completion of the test campaign, it was decided by Trenitalia UTMR to leave the wheelsets in service. A first check was carried out on 18 February 2002 at Milano Fiorenza depot, after approximately 200,000 km, and it was found that no damage was visible on the *Syope*® panels. It is worth noting that the coach number 7 in the experiments (90 83 5 889 200-2, Bar-Dining Car, one bogie with *Syope*® wheels) had in the meantime become part of ETR500PLT-31 and was therefore not inspected.

2.6 2000: low speed tests on a narrow gauge light railway

While all the tests shown up to now were conducted on high speed trains, Lucchini RS was also facing the problem of noise reduction at lower speeds. This task is even more challenging, as it was known and accepted that at low speed the track becomes the dominant source and that remedies taken on the wheel can be of low or no effect.

Lucchini RS supplied *Syope*® wheels for two different type of trains of Circumvesuviana, a local narrow gauge (950 mm) light railway with very intense traffic situated in the Naples area. The network, 140 km long, crosses densely populated areas where the disturbance induced by noise can be significant.

Extensive test campaigns were carried out in the 50-90 km/h range, showing a reduction of the overall rolling noise of 4 to 5 dB(A) also in this lower speed range, without any modifications to the track [5].

A special problem that affects most of the Circumvesuviana trains is, moreover, squeal noise in some narrow curves close to a couple of luxury hotels whose customers were strongly complaining about high level tonal noise especially at early morning. Although it is recognized that squeal noise is an erratic phenomenon, that often appears and disappears without a specific and clearly identifiable reason, it was not observed anymore during pass-bys of the trains equipped with *Syope*® wheels (see Fig. 2.13).

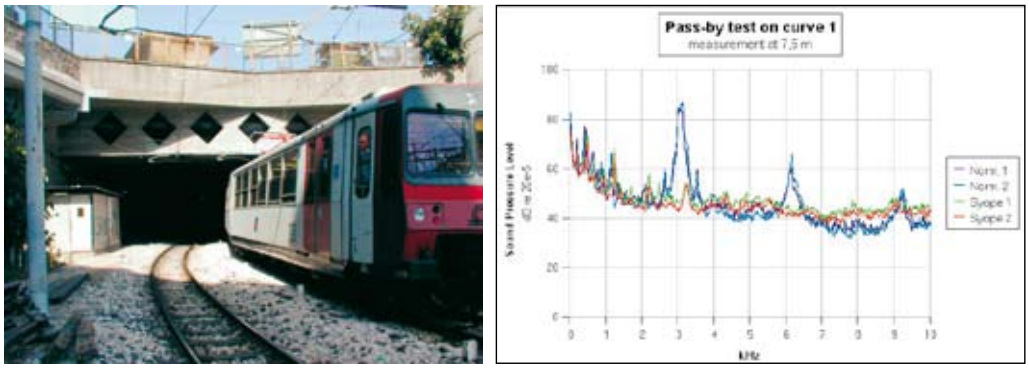


Fig. 2.13 A Circumvesuviana EMU train ETR200 negotiating a narrow curve close to Porta Nolana terminus station, just below a luxury hotel, where squealing is common (left). Noise spectra of ordinary ("Norm.") ETR200 compared to another trainset equipped with Syope

2.7 Safety certification by italcertifer

It is evident that any modification applied to a wheel may compromise safety. Immediately after the decision to perform tests, Lucchini RS submitted to Italcertifer, the Italian body for railway certification, all the technical documentation about the project, including the aforementioned reports and technical instructions.
Italcertifer, with the support of Trenitalia UTMR, evaluated the test results and the documents and, also on the basis of service checks, released on 28 August 2002 the evaluation of the mechanical and acoustical properties of the *Syope*® treatment [6].
The most important conclusions that were reached can be listed as follows:

- "The *Syope* treatment does not require any geometrical or structural modifications to the wheels on which it is applied. The mounting of panels… does not require further mechanical mounting systems… as a consequence, it does not alter the resistance properties of the wheel and has no structural functions (therefore it is not subjected to external loads…)"
- "Mechanical and adhesive properties of *Syope* treatment… have been verified with lab and in-field tests following the reference standards."
- "The manufacturing and mounting process is defined by supply procedures and material checks, mounting procedures and final check procedures. These procedures allow to trace all the manufacturing and check phases of the *Syope* treatment."
- "Moreover, the wheel production process with *Syope* treatment differs from the standard wheel solely for the panel mounting phase"
- "We certify that trackside noise measurements reveal a noise emission reduction of *Syope* treated wheels compared to standard wheels. For the microphone placed at 7.5 m from the line centreline, such reduction is not lower than 4 dBA in the speed range 200 to 300 km/h… the emission reduction is particularly concentrated in the frequency range above 1 kHz."
- "Since December 2000 FS Trenitalia has used 16 wheels with *Syope* treatment on two ETR500 trainsets used for both tests and commercial service. We certify that these trainsets have been used up to 320 km/h and that up to now they have run approximately 300,000 km without any problem due to the *Syope* treatment".

2.8 Application cases

Lucchini RS presented the results of the laboratory and line tests already mentioned at Innotrans 2002 in Berlin. At that time, the reconstruction process of the Merano-Malles line, along the Venosta valley on the Alps near the Austrian border, was at an advanced stage. The line, formerly managed by Italian State Railways FS, was closed in the early '90s as it was considered not productive. The Autonomous Province of Bolzano decided to re-activate the line and was searching for a low environmental impact rolling stock. They finally chose a DMU articulated train manufactured by Stadler, Switzerland, that incorporates many state-of-the-art solutions. People in charge asked Lucchini RS to supply wheels with *Syope*® treatment, and the line was finally opened on 5 May 2005 and immediately had great success, gradually almost totally replacing the coach services along the valley.

The train has the structure shown in Fig. 2.14 and is capable of running at speeds of up to 100 km/h. Trailing bogies are quite conventional and are equipped with *Syope*® wheels (disc brakes on the axle) while the central motor car has brakes mounted on the wheel web, a solution that intrinsically offers a low noise. For new rolling stock of this kind (DMU), Italian laws [7] require a maximum $L_{pA,F}$ of 83 dB(A) after 1/1/2002 and 81 dB(A) after 1/1/2012 measured at 80 km/h with the microphone at 25 m from the track centreline (height=3.5 m).

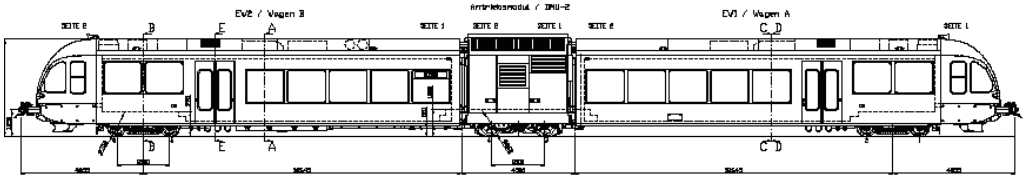


Fig. 2.14 Stadler GTW 2/6 of Val Venosta regional railway

Tests were conducted on 12/13 July 2005 on a line section with a very limited slope of approximately 2‰, a condition that allowed measurement with full throttle climbing up to Malles and with engines off descending down to Merano. It was therefore possible to evaluate the contribution of rolling noise to overall noise at full power.

The highest maximum level of noise of 79.0 dB(A) was recorded during a 100 km/h full power run, while the noise at 80 km/h with both the engines off was stable around 74.0 dB(A) [8]. These values are much lower than what is currently requested by Italian law and will also satisfy the requirements of 2012; moreover, the rolling noise contribution to the overall noise is particularly limited (9 dB(A) less than the 2002 limit, 5 dB(A) less than overall noise) and contributes only slightly to noise pollution.

Once again, what is satisfactory is the result obtained by a low speed regional train. It is worth noting that these results were obtained on a track with a roughness spectrum largely above limits stated by EN ISO 3095:2005.

Another noticeable application was the supply of high speed trainsets of the "Pendolino" type to Czech Railways. In 2000 Fiat Ferroviaria undertook an order of construction for ten tilting trains, but their number was then reduced to seven; the first set was delivered in 2004 as Pendolino CD 680 (Fig. 2.15). While testing from Breclav to Brno on November 18, 2004, the Pendolino reached a speed of 237 km/h and created a new Czech railway speed record. Since 2006 the service has been extended to Slovakia and Austria.

Fig. 2.15 Czech 680 Pendolino

These trains are very similar to the prototype Pendolino ETR470 and the wheels were exactly the same; although no noise measurements were performed by Lucchini RS or by the University of Florence, there is no reason to forecast different results from those obtained during the tests in Italy. The Czech application is, anyway, noticeable as adverse environmental conditions brought to the light a feature that was not possible to observe in Italy, i.e. a high number of freezing and thawing out cycles that repeatedly

allowed water to turn to ice and back leading to partial detachment of panels. This inconvenience generated a sealing procedure that has been applied since then in all the subsequent wheels supplied.

In 2005 Circumvesuviana decided to adopt the *Syope*® wheel as the retrofit standard for the entire existing fleet and for the new trains that are object of a tender. This is by far the largest single order of Syope wheels until now (totalling approximately more than 3000 wheels) and is particularly interesting as the Railway Administration will change the entire maintenance operation procedure (passing from shrink fit to press fit) with the inevitably associated costs that are always lower, in any case, than those linked to the use of noise barriers.

More recently, the Train Operating Companies Trenitalia SpA (part of the Italian State Railways, Holding FS SpA) and Cisalpino AG ordered a new tilting train named ETR600 (12 trainsets) and ETR610 (14 trainsets) respectively and emitted so-stringent a tender that the manufacturer (Alstom Ferroviaria) had to include the *Syope*® damping treatment (Fig. 2.16).

This step is crucial as the European Directives assign to the Infrastructure Owner (RFI in Italy) the responsibility for noise pollution disturbance. The use of noise reduction measures at the source is therefore a noticeable example that fulfils the same legislation that states noise reduction measures should be adopted preferably at the source and then on the acoustic path (leaving the measures on the receiver as the last option).

The first trains are currently undergoing homologation tests and no data are available from the noise emission point of view. It is worth noting that the wheel has changed radically from that used in the previous versions of Pendolino, and this will very likely lead to further improvements in noise emission reduction.

Fig. 2.16 The Cisalpino ETR610-001 during a transfer run (13.06.2007)

2.9 Panel strength at the end of wheel life

2.9.1 *Syope*® wheel recovery at the end of their life

Although external loads on the panel are mainly centrifugal and are supported by the mechanical structure of the panel (whose elasticity modulus is much greater than that of the polymer) and by the constraints given by the shape of the wheel web, some concerns remained about the effective durability of the treatment in real service.

The whole set of Syope wheels, four wheelsets, finished their life under ETR500 PLT-56, were returned to Lucchini RS plant in Lovere and were treated as normal steel scrap.

Thanks to the cooperation with the Maintenance structure of Trenitalia, it was instead possible to monitor the status of the Syope wheels installed in 2000 under ETR500 PLT-51, four wheelsets. These wheels were collected at the Trenitalia Workshop in Vicenza at the end of July 2005 at the end of their useful life, e.g. 1,100,000 km and five years.

Once again, Trenitalia recognized [10] that the use of *Syope*® wheels had been absolutely "transparent" to the final user, also on a high value train that was regularly inspected: "About maintenance aspects, the life cycle of mentioned wheelsets was absolutely identical to that of all the other wheelsets used on ETR500 trains…". Nevertheless, an overview of the panel conditions in the workshop showed that external panels had some

parts that were apparently detached. It was decided to take the wheels back to Lucchini RS laboratories in Lovere to verify, after a full life, the final conditions of the constraining panel.

2.9.2 Panel tear tests at the end of wheel life

As the external action on panels is due to the combination of the centrifugal force and vertical vibrations (peak and random) induced by actual wheel rolling on actual rail, it is readily understood that these actions can never produce a detachment of the panel. It was therefore necessary to "invent" a test procedure that had no resemblance with reality trying to tear off the panel from the wheel web. Several decisions were taken:

- the external action should be able to detach the panel from the wheel, i.e. it had to be applied along the wheel axis;
- no mechanical action was permitted on the panel before the external action was applied, i.e. it was not possible to make bores, threads or other fixing surfaces;
- no thermal loads had to be applied on the panel, i.e. any welding to the panel was forbidden.

The only possibility left was to bond a disk onto the inner border of the panel, where the surface is minimum and hence the possibility of detaching the panel is maximum. A structural glue was chosen and two rings, one for the internal panel and one for the external panel, were prepared. The rings were rigid enough to avoid deformation during tension tests and were pulled by a statically determinate system of chains and rings. The system is shown in Fig. 2.17 and in Fig. 2.18.

Fig. 2.17 Tear test preparation. Left: the glue is applied to the internal ring. Mid: curing time was min. 24 hours. Right: Universal traction machine with Syope wheel mounted ready for tear test

Fig. 2.18 Results of tear test on the internal panel (the ring detached at 51 kN without detaching the panel, above) and on the external panel (the ring detached at 21 kN without detaching the panel, below)

From these tests it was concluded that the reserve of safety was still largely sufficient and that five years of service of the prototypes were not sufficient to reduce the adhesion properties of the polymer. As a result, safety is absolutely guaranteed in any railway application.

2.10 CONCLUSIONS

Syope® treatment underwent a decade of development, including all the aspects of research, development and industrial processes. As a result, it proved to be a safe and reliable noise reduction treatment that preserves its functionality for the full life of the wheel.
Although technical aspects are certainly a prerequisite for any wheel related product, recent findings show that also Life Cycle Cost analyses [11] are favourable, highlighting how the impact of this measure is advantageous compared to noise barriers.

The blending of advantageous technical and economical issues are the reason for the continuously increasing success of the *Syope*® product.

Italian State Railways FS policy not only forecasts the use of noise barriers installed by RFI (infrastructure owner) but starts introducing the concept that also mass produced trains for the main train operator Trenitalia can be effectively equipped with noise reduction devices that can avoid the use of noise barriers in "border" noise pollution situations. This is a major leap towards the total and combined approach to the reduction of railway noise, that can be reached only through the deep cooperation of all those involved.

Coherently to its commitment to always provide the market with technologically advanced and safe products, Lucchini RS is currently developing, with the continuous support of the University of Florence and of its main customers, other solutions for tread braked wheels [12] and for wheels with discs mounted on the web.

2.11 FURTHER DEVELOPMENTS: SYOPE BRAW

Web braked wheels optimize the space occupied by the wheelset, but the residual space for mounting acoustic dampers is very limited.
The innovation consists in applying on the surface between the rim and the web, not taken up by the disc brake, a metallic profile with a special visco-elastic material.
This solution, as for the normal Syope wheel, does not require any mechanical fixing to the web that could reduce the wheel structural resistance.
The mounting of a disc brake will contribute to damp the wheel radial modes that deform the web.
The Syope Braw solution applied to this kind of wheel contributes to damp the bending rim modes.
The combination of Syope Braw with the disc brake enable us to obtain a particularly low noise wheel.
The results obtained from the tests in the Lucchini RS semi-anechoic laboratory, show that it is possible to foresee a reduction of the rolling noise of at least 4 dB in service.
The proposed idea results in good performance also against the squealing

noise generated by the wheel when running through tight curves where rim modes are normally excited.

Laboratory acoustic tests were carried out in the semianechoic room to compare acoustic emission
from different wheel configurations (Fig. 2.19): 1) standard, 2) with a web mounted disc brake, 3) with the Syope Braw application, 4) with both the disc brake and Syope Braw.

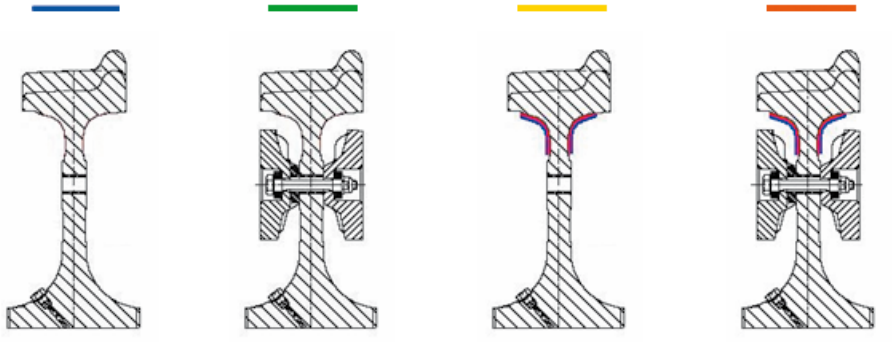


Fig. 2.19 Tested configurations

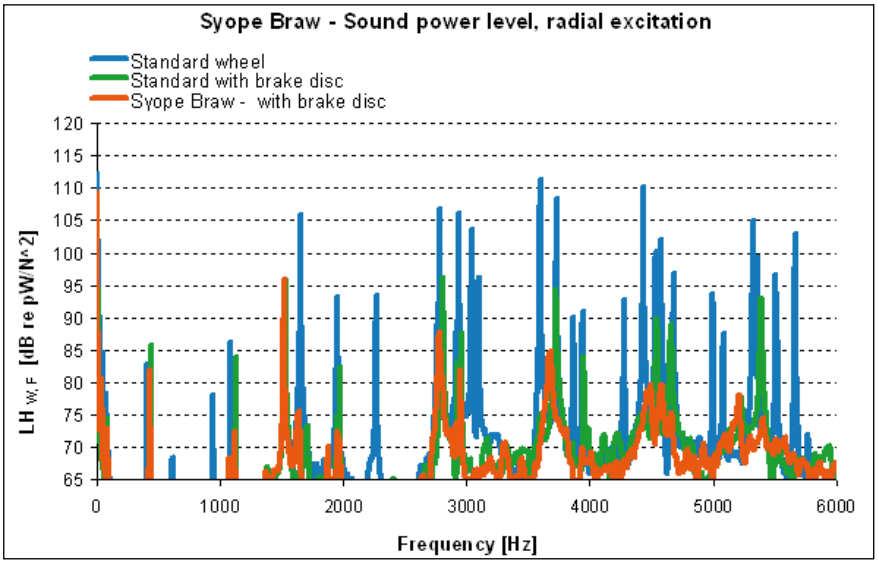


Fig. 2.20 Comparative Normalized Sound Power Level for three wheel configurations with axial excitation

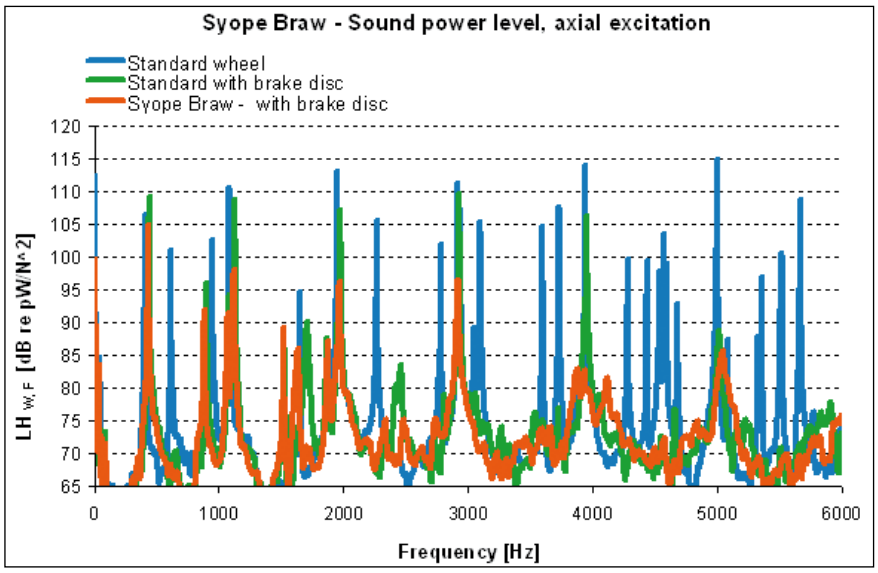


Fig. 2.21 Comparative Normalized Sound Power Level for three wheel configurations with radial excitation

Figures 2.20 - 2.21 show the results of the laboratory comparative tests performed on the three wheel configurations.
It is interesting to see how for the axial excitation, in which especially the rim modes are excited, in the case of the disc brake mounted wheel, the rim resonances are not damped but they are when they mount the Syope Braw solution. For this reason it is expected that a good effect of the Syope Braw solution can be seen when approaching curves: that is the case in which the tread is more excited.

As for the normal Syope, the Syope Braw system will last for the entire life of the wheel, the special polymer used to attach the steel plate also protects the surface of the wheel web from corrosion and has a high and constant adhesive resistance over time.
Its ability to withstand harsh conditions, extremes of temperature and humidity and substances such as fuel, alcohol and salt has been demonstrated during an extensive programme of tests by 3M and Lucchini RS.
Temperatures verifications on in service wheels show that the damper, during braking, will not exceed a temperature of 90°C .

The main benefits that can be obtained with Syope Braw wheels are:

- High reduction of both rolling noise emission and squealing,
- Syope Braw system can be applied to any existing wheel without requiring new design verifications.
- Low weight increase (8 kg per wheel) and low space occupied.
- Syope Braw system will not modify the wheel fatigue resistance.
- Excellent against wheel corrosion.
- No implication on the life cycle cost or maintenance procedures of the wheel.
- The mounting of the damper does not require any mechanical fixing to the wheel and guarantees high safety levels.
- The assembling of the Syope Braw wheels does not differ from standard ones.

Galene®

3 GALENE

3.1 Introduction

Even greater attention to noise annoyance is given to urban transportation, typically trams and metros. For these applications, where the speed is normally quite low (40 – 90 km/h), Lucchini RS produces resilient wheels. This technology requires a more complex design as the wheel is made of a central web and a tyre all made of steel and compressed in between a number of rubber blocks. This solution introduces a first suspension stage for the vehicle and has become today a normal product for all modern trams with low floor that reduces the space for traditional suspensions.
A first positive result is that the damping, introduced by the rubber blocks, make resilient wheels more silent than solid wheels, at least when considering rolling noise.
Another implication of low floor trams is that wheels are not rigidly connected with an axle anymore, but are mounted through a bearing on a special structure called "portal axle".

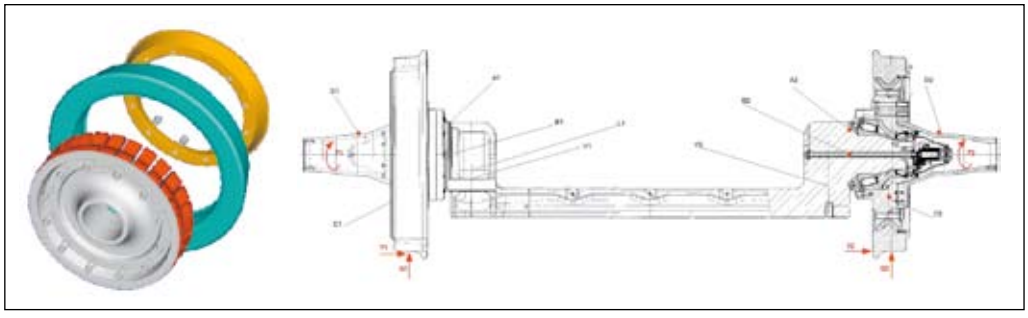


Fig. 3.1 Left: Resilient wheel components, center wheel, rubber bloks (red), tyre and (blu) and closing ring (yellow) . Right: complete resilient wheelset for low floor tram vehicles

The consequence of having independent wheels is that the wheelset loses its physical property of auto-centring its self respect to the rails. The tyre flange is then subjected to slip more against the rail during and after curving through narrow curves.
This condition can generate, especially with dry weather, a sort of dynamic phenomenon in which the tyre flange stick slips against the rail at a frequency that corresponds to the one of its resonances. This results in a high frequency

squealing tone (between 1000 and 3000 Hz) and not even the presence of the rubber is able to damp this vibration. This kind of noise is considered very annoying by the inhabitants; the reduction of this vibration is not easy as the damper should be actually tuned to the resonance frequency.
A simple solution like Syope, that would be normally effective against squealing noise, is not feasible as there is not enough surface on the tyre where the constrained visco-elastic damper can be applied.

3.2 The dampers

The damping system developed for this application is tuned absorbers. They are made of a double sheet of stainless steel with a thin layer of visco-elastic polymer in the middle.
The damping plates are then mounted to the vibrating component in a clamped-free configuration.
The geometrical dimensions of these plates are of fundamental importance to enable correct damping of the vibrations: if the resulting modal frequencies of the dampers are near to the vibrating component frequencies these will be damped as the vibration energy will pass to the plates and then be dissipated in the form of heat energy.
Correct dimensioning of the absorbers requires extensive modal testing on prototype shapes mounted on the component, necessary to define and validate numerical models used for further geometry optimisation and to finally verify the actual performance of the optimal solution.

Fig. 3.2 Various shapes of dynamic absorbers

The case here presented is about a city tram found to have a very high squealing resonance at 2000 Hz which would greatly annoy the citizens.

3.3 Absorber development

First laboratory modal tests showed that 2000 Hz was clearly the 3rd resonance of the tyre.
The problems found in the development of the absorbers was the very small space left for the application due to the fact that for the powered wheels, the gear was directly linked to the majority of the centre-wheel; whereas for the trailer wheels, much space was already occupied by the braking system.
The development of the absorber was carried out through both numerical and experimental approaches. Fig. 3.3 shows by the FEM model the 2000 Hz mode shape of the tyre and of the absorber.
The laboratory tests were performed on a completely powered wheelset (two powered wheels and one portal axle).
The portal axle leans through two rubber blocks on the laboratory floor; the exiting force is an impulse in two fixed positions, first on the rim rolling surface, radial direction, then laterally on the tyre flange.
The acceleration measurements are done by one triaxial accelerometer applied on the rim, external side.
After various optimisation iterations of the absorber's geometry, it was possible to define a solution which proved to be able to considerably reduce squealing noise.

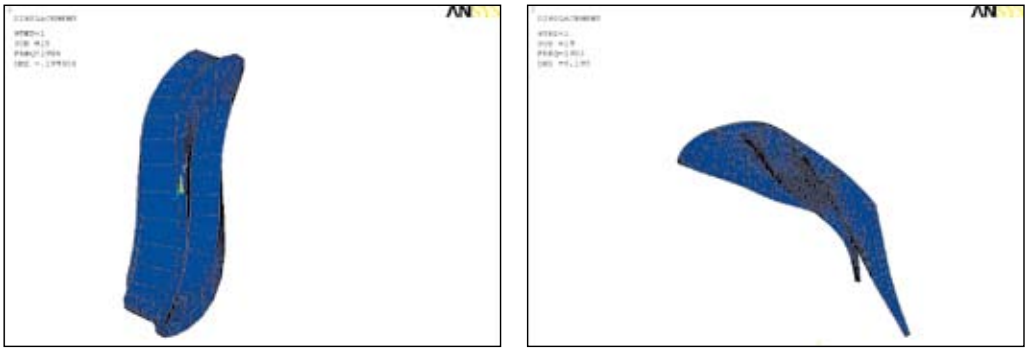


Fig. 3.3 Tyre and absorber mode shape

In figure Fig. 3.4, the experimental FRFs of the radial (first graph) and lateral (second graph) acceleration response of the tyre without absorbers (green

line) and with the final version of the absorbers mounted (red line) are shown. Most of the resonances are reduced but it can be seen that especially the 2000 Hz frequency is considerably damped demonstrating that this type of technology can be particularly effective when dealing with one particular frequency.

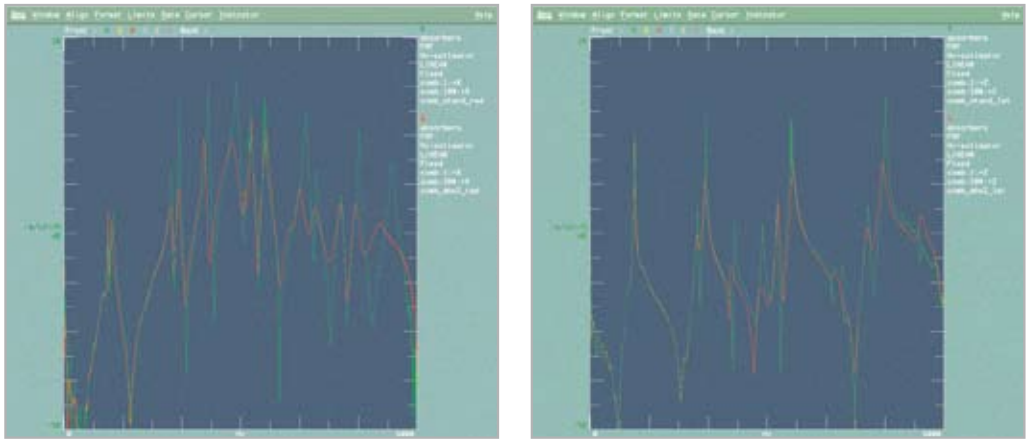


Fig. 3.4 Frequency Response functions for the standard (green) and damped (red) tyre

3.4 In service acoustic tests

After the development of the absorbers it was decided to produce anumber of prototypes to be tested in service conditions

Tram configuration

The tyre profiles were newly machined and no lubrication was active on the wheel during the tests.

In figure Fig. 3.5 there is the layout of the train set with the position of the powered (PA and PB) and trailer (T) bogies, the wheel position where the 4 microphones were placed (M1, M2, M3 and M4) and the front driving direction. A photograph follows of the wheel with the absorbers and the position of the microphone on the gear in front of the wheel.

During the tests, the wheels were equipped with absorbers in different configurations which are listed in Fig. 3.6.

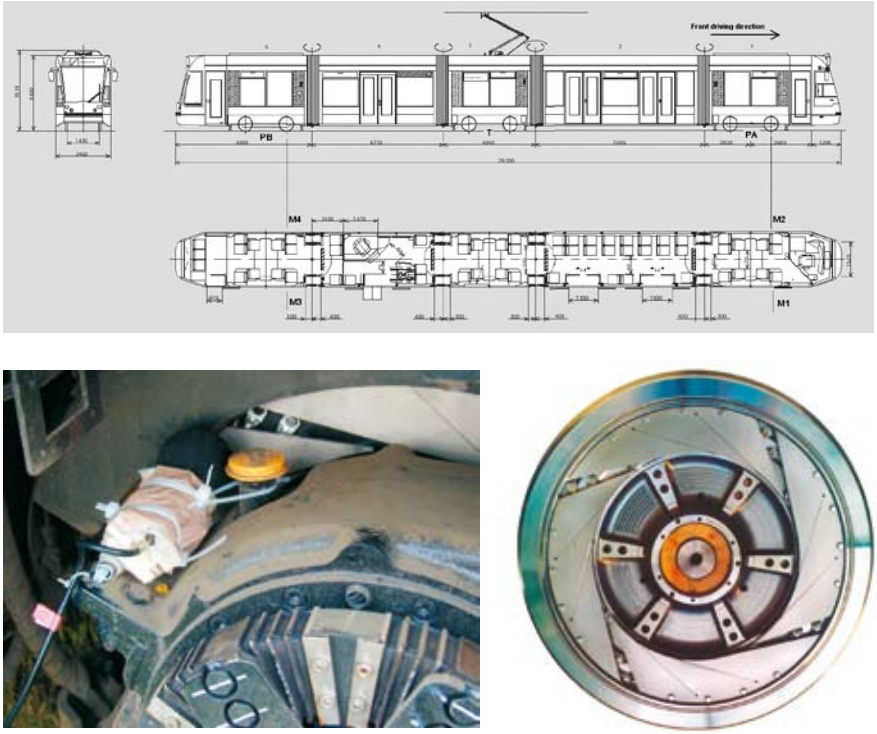


Fig. 3.5 Configuration of the tram for the acoustic tests (above); the microphone mounting (below left); the Combino resilient wheel with the absorbers

Configuration A	Bogies PA, T and PB equipped with wheel absorbers
Configuration B	Bogies T and PB equipped with absorbers and PA without
Configuration C	Bogies PA, T and PB equipped without absorbers

Fig. 3.6 Tram configuration during the tests

Test sites

The test sites name and description are listed in Fig. 3. 34.

Curve A	Narrow curve on the ring track inside the maintenance depot
Curve B	Medium curve on the ring track inside the maintenance depot
Curve C	Large curve on the ring track inside the maintenance depot
In the city	Different curve track conditions around the city

Fig. 3.7 Chracteristics of testing sites

Two types of tests were performed:

1) On board acoustic tests

The tram was in configuration B and 4 microphones were mounted near to 4 different wheels (see figure 2 and 3):

- M1 front right powered wheel of bogie PA
- M2 front left powered wheel of bogie PA
- M3 front right powered wheel of bogie PB
- M4 front left powered wheel of bogie PB

Microphone signals were recorded at a sampling frequency of 10 kHz after an antialiasing filer set at 5 kHz during different running conditions:

- At curves A and B at normal speeds
- At curves C at different speeds
- At different curves around the city

2) Pass-by acoustic tests

The tram was in two different configurations: A (all wheels with absorbers) and C (all wheels without absorbers).
Two fixed microphones were placed in the inner side of curve C with the tram passing at different speeds and direction.

The microphones were:

- M1 placed at 7,5m from the central track, at 1,5m height from the ground
- M2 placed at 1m on the ground from the nearer rail

Microphone signals were recorded at a sampling frequency of 16.384 kHz after an antialiasing filer set at 8 kHz.
The curve track site was part of the ring track inside the maintenance depot and was the one called C .

Test results

In order to analyse the data, all the microphone signals have been first "A" weighted in the time domain and then processed in the time and frequency

domain using a Hanning window of 1024 samples and an overlap of 30%, obtaining a coloured map of the sound pressure level with a time resolution of 0,031 s and a frequency resolution of 9,77 Hz from 0 to 5000 Hz Figure 3.35 shows, as an example, a time-frequency diagram of microphone 2 and 4 with reference to curve B, run 2. Here it's clear that the 2000 Hz is the dominant source for M2 and that is then considerably reduced due to the absorbers' performance.
Fig. 3.9 summarises and compares the maximum sound pressure levels of the analysed microphones and average and standard deviation values are calculated.
The same type of analysis applied for the on board tests is used for the pass-by tests (Fig. 3.10).

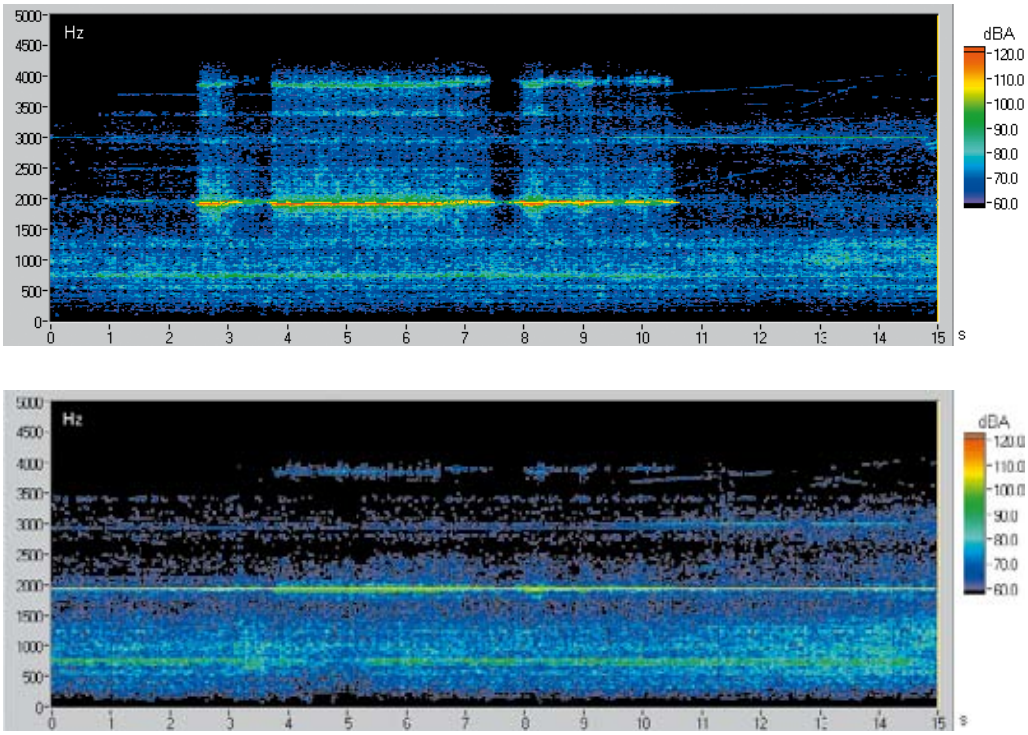


Fig. 3.8 Time - frequency of the sound pressure levels for two microphones, one installed near the standard wheel, the other near a wheel with absorbers

		On board test - Curve A at the depot					
		Lp_{max} at 1934Hz			OA Lp_{max}		
		Standard	Absorbers	Delta	Standard	Absorbers	Delta
	Run N°	dBA	dBA	dBA	dBA	dBA	dBA
	1	128	83	45	130	107	23
	11	125	87	38	130	108	22

		On board test - Curve B at the depot					
		Lp_{max} at 1934Hz			OA Lp_{max}		
		Standard	Absorbers	Delta	Standard	Absorbers	Delta
	Run N°	dBA	dBA	dBA	dBA	dBA	dBA
	2	125	108	17	125	110	15
	12	126	109	17	126	112	14

		On board test - Curve C at the depot					
		Lp_{max} at 1934Hz			OA Lp_{max}		
	Speed	Standard	Absorbers	Delta	Standard	Absorbers	Delta
Run N°	km/h	dBA	dBA	dBA	dBA	dBA	dBA
9	30	124	98	26	130	112	18
8	25	110	110	0	115	110	5
7	25	120	91	29	125	110	15
6	20	110	102	8	112	109	3
5	20	126	100	26	126	100	26

		On board test - around the city					
		Lp_{max} at 1934Hz			OA Lp_{max}		
		Standard	Absorbers	Delta	Standard	Absorbers	Delta
	Run N°	dBA	dBA	dBA	dBA	dBA	dBA
	1	126	102	24	128	123	5
	3	122	80	42	130	105	25
	7	120	96	24	124	116	8
	8	134	85	49	133	122	11
Average		122	98	24	125	112	13
Standard deviation		7	10	14	6	7	8

Fig. 3.9 Results of the on board test

		Pass-by test - Curve C at the depot											
		Lp_{max} at 1934Hz						OA Lp_{max}					
		Microphone M1 at 7,5 m			Microphone M2 at 1 m			Microphone M1 at 7,5 m			Microphone M2 at 1 m		
	Speed	Standard	Absorbers	Delta	Standard	Absorbers	Delta	Standard	Absorbers	Delta	Standard	Absorbers	Delta
Run N°	km/h	dBA	dBA	dBA	dBA	dBA	dBA	dBA	dBA	dBA	dBA	dBA	dBA
<-- 10	30	84			101			88			104		
--> 9	30	74			92			81			96		
<-- 8	25	88			104			94			108		
--> 7	25	88			100			89			103		
<-- 6/2	20	92	53	39	105	74	31	94	74	20	107	89	18
--> 5/1	20	82	54	28	97	72	25	87	74	13	100	90	10
Average		85	54	34	100	73	28	89	74	17	103	90	14
Standard deviation		6	1	8	5	1	4	5	0	5	4	1	6

Fig. 3.10 Results of the pass-by tests

3.5 Conclusions

The processing of all the signals recorded during the tests show that the absorbers developed for the resilient wheel are able to reduce squealing noise considerably.
At the frequency of 1936 Hz where the highest squealing resonance takes place there is:

- an average reduction of 24 dB (Lp_{max}) when measuring with microphones mounted on the bogie near to the wheel along different type of curves
- an average reduction of 34 dB (Lp_{max}) when measuring the pass-by noise of the tram first completely equipped with absorbers then without, at a speed of 20 km/h, along curve C.

For the overall sound pressure level (Lp_{eq}) there is:

- an average reduction of 13 dB (Lp_{eq}) when measuring with microphones mounted on the bogie near to the wheel along different type of curves
- an average reduction of 17 dB (Lp_{eq}) when measuring the pass-by noise of the tram first completely equipped with absorbers then without, at a speed of 20 km/h, along curve C.

Hypno®

HYPNO

4.1 Introduction

Developments of absorbers for freight wheels were not followed initially mainly for market reasons. Freight operators would be very unlikely to pay for products with this kind of added value.
But on the other hand, freight vehicles are recognized as the noisiest railway vehicles and people living near railway lines have a much higher concern considering the fact that this kind of vehicle normally travels at night.
But the main problem that we face when trying to deal with this specific problem is that because the wheel is tread braked, the rim can reach very high temperatures (around 500°C; this is the case for example when traveling down the Alps railway line).
In these conditions viscoelastic polymers cannot be used as they would melt and the wheel rim would also tend to thermally expand itself creating possible geometrical or interface problems to mechanical parts mounted on it.
The tread brake introduces relevant defects of the rolling surface and generally a higher roughness that excites wheel vibration more. An Important step forward seems to be obtained by using K composite brake blocks instead of the traditional cast iron blocks.
The result is that the surfaces are smoother and the noise emission becomes more similar to a standard solid wheel with disc brakes.
But even with this solution that is nowadays being generally adopted, a relevant reduction of noise emission needs to be obtained in order to comply with the new European noise regulations.
Lucchini RS research in this field has resulted in the development of the so called Hypno® solution.
No polymers are used and damping is obtained through friction between metal plates.
This Project was carried out by Lucchini RS under the European Funded Project called Silence and coordinated by DB. The solution was developed for two wheels (BA004 and Sura25) considering the specification of freight operators and was tested on the Trenitalia brake test rig and under the DB Cargo vehicles.

4.2 Design concept

The Hypno® damping system dissipates vibration energy by generating friction forces through relative micro-movements between two metal plates. Such relative movements are generated by the vibrating rim on which one of the metal plates is mounted; the second metal plate is fixed to the hub and connected to the complete surface of the other plate with special rivets that must enable small relative movements between the plates (Fig. 4.1).

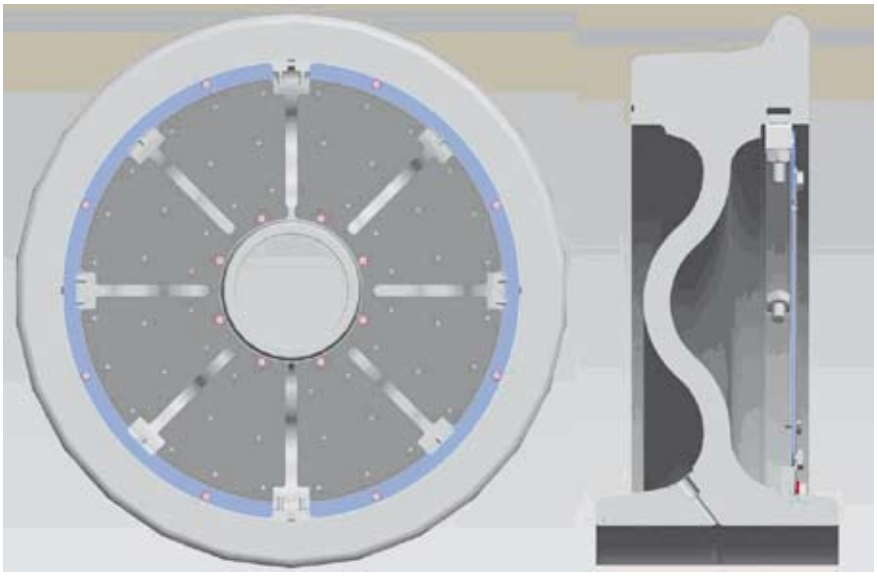


Fig. 4.1 Hypno® design concept

The first metal plate is constrained to the hub using an elastic metal ring that is inserted into a rectangular groove machined over the hub, while two different connection systems have been developed to fix the second plate.
The second metal plate is constrained to the rim by means of four circular sectors which are screwed to the wheel rim triangular groove.
The metal plates are connected both to the ring and to the four circular sectors by screws. Both the plates have 8 radial cuts to enable the air ventilation of the wheel and cooling during braking. To enable easier radial expansion the metal plates are divided into two or four symmetrical parts.
The Hypno damper was developed for both the BA004 standard German freight wheel and the Sura25 the newly UIC certified freight wheel designed by Lucchini RS for a 25 ton axle load (Fig. 3. 40).

Due to the different web shape, on the BA004 it was only possible to mount the damper on the inner side of the wheel, whereas for the Sura25 there was enough space on the outer side.
The two prototypes were produced and tested in the Lucchini RS acoustic laboratory and on the Trenitalia brake test rig.
The Sura25 wheel performed better. This was justified by the fact that having the damper mounted on the outer side and so connected to the outer side of the rim that has higher mobility than the internal side, the presence of the flange that makes this side more rigid, more dynamic movement would be introduced into the damper and so more damping would be achieved.
Subsequently it was decided to test a complete wheelset on the Lucchini RS roller rig BU300 and compare two Sura25 wheels one with the absorber and one without.
This test was very interesting as the rim vibration was measured by accelerometers on each wheel during wheelset rotation; signals were acquired through rotating electric contacts.

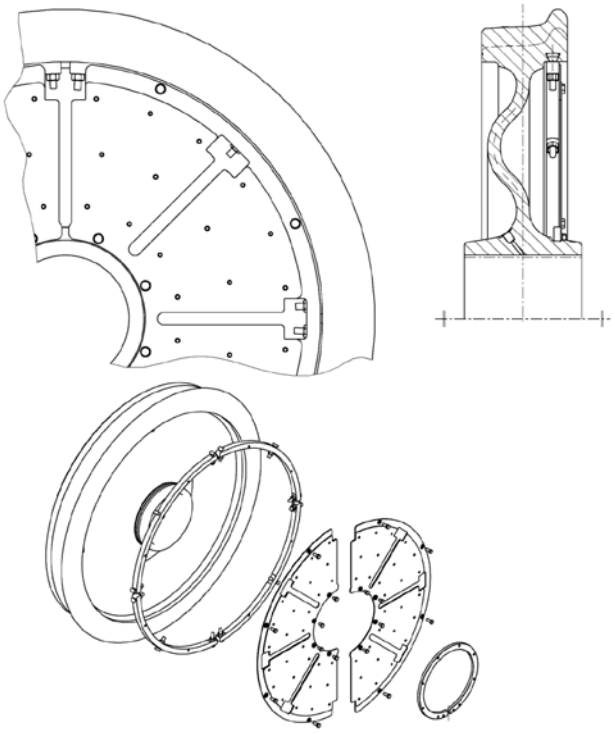


Fig. 4.2 Hypno® damper overview

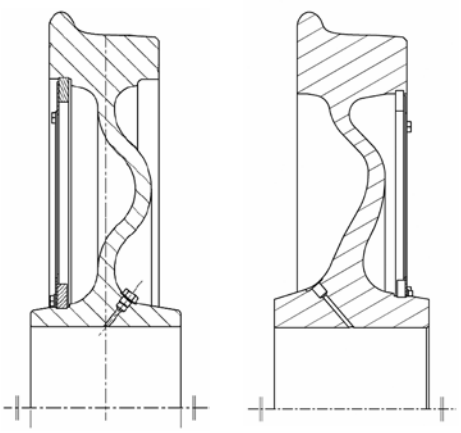


Fig. 4.3 The two Hypno prototypes; BA004 wheel (left), Sura25 wheel right

4.3 Laboratory acoustic tests

The test setup was the typical one developed by Lucchini RS, consisting in the determination of the Normalized Sound Power from microphone measurements on an arc (Fig. 4.4).
In the case of the Sura25 wheel, the reduction of the resonance picks was very good from 1 kHz up : from 10 to 20 dB in the case of axial excitation (Fig. 4.5) and from 15 to 30 dB in the case of radial excitation (Fig. 4.6).
The damper seemed to have a better efficiency against rolling noise where radial dynamic forces were involved.

Fig. 4.4 Vibro – acoustic measurement setup

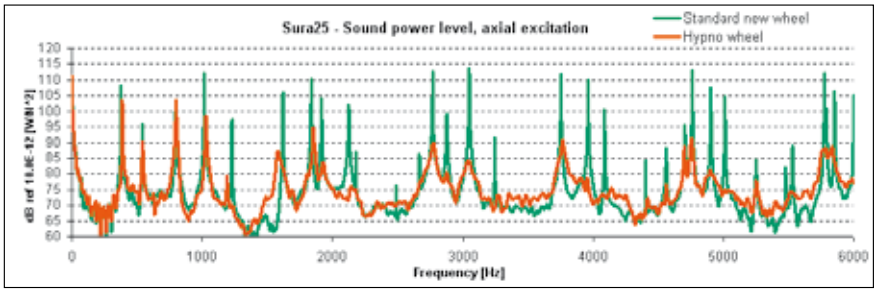


Fig. 4.5

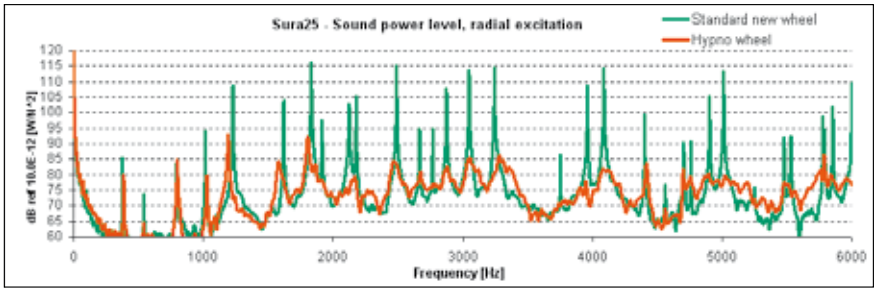


Fig. 4.6

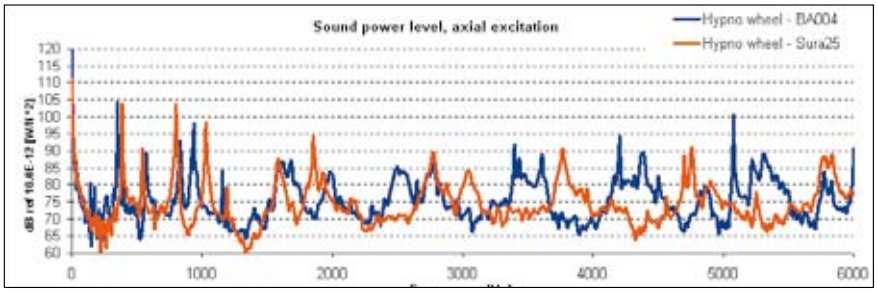


Fig. 4.7

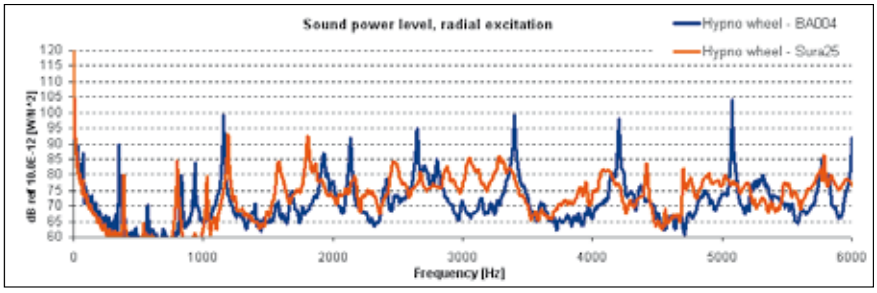


Fig. 4.8

The BA004 with the Hypno damper compared to the Sura25 wheel, showed a similar performance in the case of axial excitation (Fig. 4.7), whereas for radial excitation higher resonance peaks could be found for the BA004 wheel (Fig. 4.8). This confirms the fact that the best position for dampers to be applied is on the outer side where the rim has greater mobility.

4.4 Brake tests

It was important to verify the ability of the proposed damper to stand the very critical conditions that the wheel undergoes during high power braking.
Basically, according to the UIC 515-5 the worst conditions are reached when descending the S.Gottardo railway. In this case, the wheel brakes with a constant power of 50 kW for about 45 minutes at a speed of 60 km/h.
During these tests the temperature of the rim exceeds 500°C and the rim has a radial and axial movement of about 1 to 2 mm.
For this reason it is important that the damper is able to adapt to the relative movements of the rim where it is attached and that no residual deformation remains after the cooling of the wheel.
The tests were performed according to the UIC510-5 which is the Standard used for new wheel design qualification on the UIC certified Trenitalia test rig (Fig. 4.9). After the complete sequence of tests the damper was still in good condition and, in particular, no buckling or any kind of residual deformation could be observed.

Fig. 4.9 The Hypno wheel mounted on the Trenitalia Brake test rig

4.5 Vibro-acoustic tests performed on the BU300 test rig

Field testing is normally a very expensive activity. During this project it was decided to characterize the wheel vibration in rolling conditions. This was done on the BU300 test rig, a unique facility that enables testing wheelsets rolling in very realistic running conditions.
In this occasion we developed a new version of the Hypno wheel (right photo of figure Fig. 3. 47 SURA25 wheelset with the noise absorber "Hypno";); the intent was to enable the visual inspection of the web as this is one of the requirements of some operators; visual inspections are made on the wheel web during periodic maintenance of the vehicle. The principle of the solution is the same as for the first prototype.
The following tests are an interesting example of how it was possible to compare easily and in very realistic dynamic conditions to different versions of the prototypes.
In this case, two SURA25 solid wheels (one of them equipped with the Hypno dampers) were press-fitted on a test axle and mounted on the BU300 roller rig (Fig. 4.10 and Fig. 4.11).
The wheelset was subjected to various running conditions in order to simulate the behaviour of the Hypno dampers in every in-service conditions. Fig. 4.12 shows the combination of the vertical and lateral loads (FQ1, FQ2 and FY) applied at the wheelset and the speeds at which the tests were performed.

Fig. 4.10 SURA25 wheelset with the noise absorber "Hypno"; the second version on the right side enables the visual inspection of the web

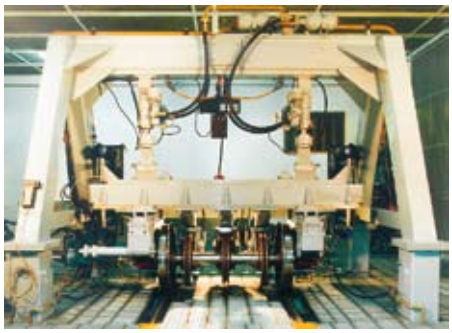


Fig. 4.11 - Test wheelset mounted on BU300

Test number	FQ1	FQ2	FY	Velocity
	kN	kN	kN	km/h
01	74	69	0	70
02	74	69	0	75
03	74	69	0	80
04	74	69	0	85
05	74	69	0	90
06	69	74	-10	75
07	79	64	10	75
08	69	74	-15	75
09	79	64	15	75
10	69	74	-10	90
11	79	64	10	90
12	69	74	-15	90
13	79	64	15	90

Fig. 4.12 Combination of vertical under lateral loads applied on BU300 test rig

4.5.1 Measurement setup

It is not possible to use acoustic microphones inside this test rig due to the high noise produced by the engines and various electric systems that make the test rig function.

As previously made in other projects, accelerometers were placed directly on the wheel rim and from the acceleration measurements an estimation of the Sound Power emission was made and from there a calculation of the Sound Pressure Level that a micropon would have measured at a distance of 7,5 m.
Two mono-axial accelerometers were attached on the outer side of each wheel: one in the radial direction and one in the axial direction of the wheel (see Fig. 4.13) in order to measure the radial and the axial components of the acceleration of the wheel.

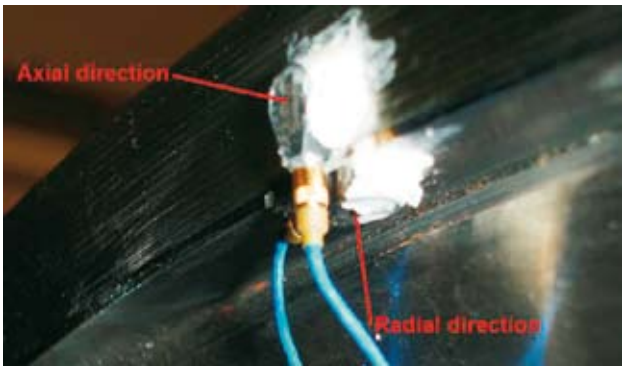


Fig. 4.13 Accelerometer application point and measure direction

These four accelerometers are fixed with a special glue and the signal cables pass through a radial hole on the axle collar and are then connected to a special sliding ring contact system, in order to transmit the acceleration signal while the wheelset is rotating (Fig. 4.14 and Fig. 4.15).

Fig. 4.14 Sliding contacts terminal

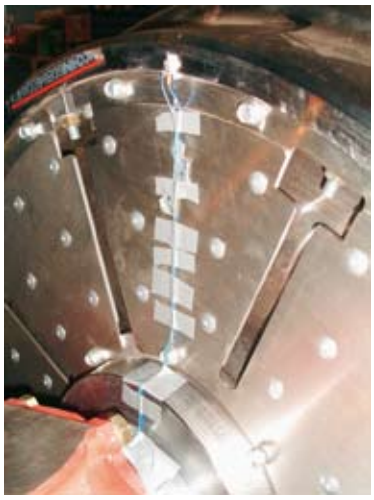


Fig. 4.15 Accelerometers attached to the "Hypno wheel"

Fig. 4.16 The new version of the Hypno damper that enables the visual inspection of the web

4.5.2 Measurements

Measurements of the vibration were performed at various speeds up to 90 km/h in straight and curve running conditions.

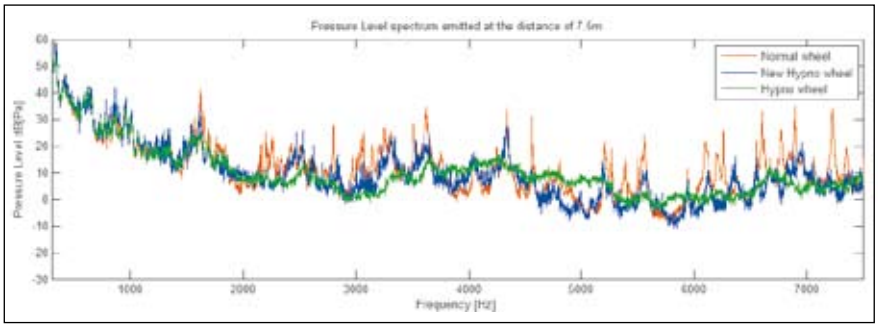


Fig. 4.17 Example of Sound Pressure Level estimated from acceleration measurements on the wheel while rotating at 80km/h on the roller rig BU300

In Fig. 4.17 Example of Sound Pressure Level estimated from acceleration measurements on the wheel while rotating at 80km/h on the roller rig BU300 there is an example of Sound Pressure spectra estimation which radiates

from the Normal wheel and from the two kind of Hypno wheel prototypes: the peaks.
It can be seen that the standard wheel maintains it frequency resonance as if it was free to vibrate; in the case of the first Hypno prototype, resonances are almost completely cancelled from 2 kHz and the reduction of the peaks is in the order of 15 ÷ 20 dB(W) and an overall reduction of about 6 dB. For the second prototype some resonance remain, even if at a lower level but resulting in an overall level reduction of about 4 dB; the result is nevertheless very interesting if we consider the fact that damper surface and its mass is much lower.

4.6 Conclusions

The tests performed on the SURA25 wheelset shows clearly the great contribution given by the "Hypno" absorbers to reduce the noise emission The innovative aspect of this solution is that for the fist time an effective damper is able to resist to the very high temperatures (500°C) and the derived thermal elongations that are reached when the wheel is braked. The testing method shows that a design stage it is possible to effectively evaluate the noise reduction of different solution in very realistic running conditions reducing the need for expensive in service tests.

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Lucchini RS committed to try to help its customers to reduce several types of railway noise that are described in this book, developing since mid '90s a number of solutions that are now available and that can be tailored on the specific application. Extensive laboratory activities and measuring campaigns helped to identify the most promising techniques, together with a long lasting presence in European funded projects and a continuous cooperation with Universities, Research Centres, railway administrations and rolling stock manufactures.

The book is structured in three chapters in order to give the reader a general information as wide as possible without entering mathematical details that are extremely complicated in acoustics.

The first part of the book, written by Prof. A. Bracciali, is an introduction to railway acoustics dealing with railway noise generation and control mechanism. Although the goal of the book is to describe mainly the wheels, this part shows how reducing the overall external noise is a *teamwork* task, involving also the infrastructure and a number of specifically devoted maintenance techniques.

The second part of the book, written by Dr. M. Pippert, is a comprehensive excursus of the legislation in force in Europe and in the world on the subject of railway noise abatement. Legal and political frameworks are described and discussed, together with a number of arguments that are in favour of the train as the most environment friendly transportation mean.

The third part of the book, written by Dr. S. Cervello, describes Lucchini products to reduce railway noise that are the direct consequence of a long development of laboratories equipment and of technical skills of Lucchini R&D employees. Low noise wheels for high speed, regional and mass transit (metro, tram) trains are described, together with research in progress for heavy haul trains.This book represents the first Lucchini RS publication on the topic of railway noise control. It is the second book of a series of technical books whose publication started in 2008.

In particular, this book deals with railway noise, a by-product of railway transportation which is annoying a large part of the people living in densely populated areas.