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BUT ARE THE TRAINS GETTING ANY QUIETER?

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Abstract

To reduce railway noise effectively a good knowledge of the source mechanisms is first required. Wheel/rail interaction, in particular, results in a multiple source environment where wheels, rails and sleepers all play a role. This is a classic noise control problem where treatments will not be successful unless they take account of the relative importance of each source and appropriate measures are applied for all the significant sources. Starting from theoretical research into source modelling, a number of practical techniques will be discussed that have been developed for reducing railway noise in the last 10-20 years. The difficulties of practical implementation will also be discussed, recognising that the railway industry faces many pressures that make it reluctant to change.

1. INTRODUCTION

The huge increases in mobility over the last two centuries have been accompanied by large increases in environmental noise. Although railways are seen as a relatively environmentally-friendly means of transport, in common with most means of transport they nevertheless represent a significant source of noise.

The early railways were often subject to considerable opposition. The following was written in 1825, in a letter to the Leeds Intelligencer (quoted in [1]): "*Now judge, my friend, of my mortification, whilst I am sitting comfortably at breakfast with my family, enjoying the purity of the summer air, in a moment my dwelling … is filled with dense smoke, …. Nothing is heard but the clanking iron, the blasphemous song, or the appalling curses of the directors of these infernal machines*". Nevertheless, although some objections such as this were attributed to environmental reasons such as noise, most were brought for economic or aesthetic reasons [1]. As early as 1863, for example, the Manchester, Buxton, Matlock and Midlands Junction Railway in England was forced to build its line in a cut-and-cover tunnel almost 1 km long so that it should not be visible from the Duke of Rutland's home at Haddon Hall [2]. Today such schemes are not uncommon to mitigate noise, but the idea is not new.

In common with many other forms of environmental noise, railway noise has become an increasingly important issue in the last half century. Opposition to new railway lines is now often focussed on their potential noise impact. This may be because noise is quantifiable in a way that aesthetics are not, but as noted by the Wilson Report of 1963 [3]: "There is a considerable amount of evidence that, as living standards rise, people are less likely to tolerate noise".

Public awareness of noise has led to the introduction of legislation in many countries. Limits have been applied to the noise emitted by individual road vehicles and aircraft since the 1970s, and these have become stricter over the intervening period. Nevertheless, it is widely accepted that, even though the European noise limits for cars were reduced by 8 dB between 1973 and 1996, they have had little effect on the noise produced in normal use which is dominated by tyre noise [4]. For rail vehicles, the issue is complicated by the interaction of the vehicles and the track, both of which contribute to the noise. Limits on the noise emitted by individual rail vehicles have only been introduced in Europe in the last five years. These have been achieved through the means of 'Technical Specifications for Interoperability' (TSI) [5, 6], which are intended primarily to *allow* interoperability of vehicles between different countries in Europe. Such limits have the potential to reduce railway noise in the long term. European legislation also requires the development of noise maps for large population centres and the subsequent development of action plans [7]. These, too, will mean that railway operators and infrastructure companies will have to consider how to minimise noise.

Noise barriers are often used to mitigate noise levels from new railway lines. For example, on the 160 km Betuwelijn freight route in the Netherlands, to be opened in 2007, there are 160 km of noise barriers, even though much of the route runs parallel to motorways. Although these barriers only represent about 2-3% of the total cost of the project (€4.7bn), if the three tunnels on the route are also included, the proportion spent on 'noise mitigation' rises to around 20%. Control at source is now increasingly seen as potentially more cost-effective than secondary measures such as noise barriers [8].

Since the 1970s research has been under way into the sources and mechanisms of railway noise. From this, theoretical models have emerged that explain the origin of the noise [9]. This has been followed by the development of many different techniques intended to reduce the noise at source. Yet, the practicalities of the railway industry are such that it faces many pressures that make it resistant to change. Moreover, its assets, both vehicles and infrastructure, have typical lifespans of 30-50 years. So, although railway noise may now be quite well understood, we are entitled to ask: 'But are the trains getting any quieter?'

Historically, great changes occurred in the railways with the replacement of steam traction by diesel and electric in the 1950s and 1960s and with the replacement of much traditional jointed track by continuously welded rail from a similar period. Since the 1970s many vehicles have been fitted with disc brakes which has the side effect of leading to quieter running, reducing noise levels by around 10 dB compared with cast-iron block braked stock [10]. But these changes have been offset by large increases in speed, particularly with the introduction of high speed trains. A doubling of the speed increases the level by about 9 dB, thus eliminating the acoustic benefit of disc brakes.

In this paper, an overview of railway noise sources is given with particular emphasis on rolling noise. Possible means of noise reduction are then discussed before expanding on reasons why implementation is often difficult. Results are drawn from a number of international research projects, many of which have been described in more detail in [11].

2. "HAVE YOU EVER THOUGHT OF USING RUBBER WHEELS?"

In discussing railway noise control, people often ask: 'Have you ever thought of using rubber wheels'. The simple answer is 'Yes, of course'. The introduction of flexibility at the wheel/rail contact is known to be beneficial in reducing the excitation of wheel and rail [12]. Yet, behind the question is usually the naïve assumption that 'rubber wheels' must be quieter because rubber is somehow a 'quiet' material. This is not necessarily the case.

To pose a slightly different question: Are trains actually louder than lorries? In both cases the dependence on speed is similar so noise levels are compared here at a common speed of 80 km/h. In [4] it was shown that the A-weighted SEL for a single heavy road vehicle travelling at 80 km/h and measured at a distance of 7.5 m remained around 87 dB between 1972 and 1998. These levels are dominated by tyre noise. A-weighted sound pressure levels at 25 m from a freight train consisting of 4-axle 100 tonne tank wagons travelling at 80 km/h are given by Hemsworth as about 84 dB [13]. To compare these two figures, corrections for distance and to SEL are required. As shown in Table 1 this gives an SEL of 88 dB for the rail vehicle which is quite close to the result for the lorry.

Table 1. Comparison of noise levels from lorries and rail vehicles (all A-weighted sound pressure levels in dB at 80 km/h).

	Correction	Rail vehicle	Lorry
Average level at 25 m		84	
Average level at 7.5 m	$10 \log_{10}(25/7.5) = 5$ dB	89	
SEL at 7.5 m for single vehicle	$10 \log_{10}(T) = -1$ dB ($T = 0.8$ s for vehicle length 20 m)	88	87
Normalise by load factor	$10 \log_{10}(2.5) = 4$ dB	88	91
Update to current practice	Rail vehicle 10 dB quieter due to braking system; lorry on porous road surface	78	86

However, each 100 tonne tank wagon can carry about 2 to 3 as much as a 40 tonne lorry. Or put another way, a rubber-tyred rail vehicle would require 2 to 3 times as many wheels as the lorry. Taking this loading factor as 2.5, this gives the rail vehicle an additional advantage of 4 dB.

Finally it may be noted that the levels given for tank wagons in [13] are for vehicles fitted with cast-iron brake blocks; more modern vehicles with either disc brakes or composite brake blocks can be expected to be about 10 dB quieter (see section 7.1 below). Similarly quiet road surfaces, such as porous asphalt, can reduce the tyre noise by around 5 dB, more on a new surface. This gives levels of 78 and 86 dB respectively. Therefore this rough comparison shows that the carriage of freight by rail has a potential noise advantage of around 8 dB over carriage of the same load at the same speed by road. Clearly the supposition that rubber wheels would be advantageous is misplaced in this context.

Table 2. Features of rubber tyres and steel wheels.

	Rubber tyre	Steel wheel
Damping loss factor	~ 0.1	~ 3×10^{-4}
Radiation efficiency	very low (also influenced by ‘horn effect’)	~ unity
Excitation mechanisms	Road/tread roughness Block impact/snap-out Air pumping	Wheel/rail roughness
Amplitude of roughness excitation	~ mm	~ μm

The various features of the two systems are compared in Table 2 from which it is clear that the high damping and low radiation efficiency of a rubber tyre are offset by the high amplitudes of excitation at the tyre-road interface. Other comparisons could be made: if the tread pattern could be eliminated and the road surface replaced by a smoother track, tyre noise levels may be reduced considerably, perhaps by as much as 20 dB. Such systems exist in mass transit systems where rubber-tyred vehicles are used, but clearly they cannot operate on conventional tracks. However, the final argument in favour of the steel wheel on steel rail system is that the rolling resistance is much less than for a rubber tyre.

3. RAILWAY NOISE: A CLASSIC NOISE CONTROL PROBLEM

The problem of railway noise can be used to illustrate the classical methodology of noise control.

The first step in noise control is to *identify the dominant source*. There are many different sources of noise from a railway, and in different situations the dominant source may vary. Notably on North American freight railways, a major issue for environmental noise is related to locomotive warning signals. It is obligatory to sound the horn in an extended sequence on the approach to road crossings [14]. There are many such crossings, especially in populated areas and so it is a major source of annoyance, particularly from operations at night. In other situations, such as stations in urban areas, the public address system may be the major source of noise in the immediate neighbourhood.

However, the most important source of noise from railway operations is usually rolling noise. This is caused by the interaction of wheel and rail during running on straight track. Other sources include curve squeal, bridge noise and aerodynamic noise. Noise inside the vehicle also includes all of these sources as well as others such as air-conditioning.

Having identified the dominant source, the following step is to *quantify the various paths or contributions*. Focussing on exterior rolling noise, the vibration of the wheel and the rail can be identified as potential sources. Early attempts to understand the problem tended to be polarised into attributing the noise solely to one or the other [15]. More recently, however, it has become widely recognised that both the wheels and rails usually form important sources which make similar contributions to the overall sound level [16]. Prediction models allow their relative contributions to be quantified (measurement methods can also be used, although they should be used with caution, as discussed in section 5 below). Clearly, effective noise control requires both sources to be tackled. For example, in a situation where wheel and rail contribute equally to the overall level, a reduction of 10 dB in one of them, while the other is unchanged, will produce a reduction of only 2.5 dB in the total.

The next step is to *understand how each source can be influenced*. Here, the theoretical models allow the sensitivity of noise to various design parameters to be investigated (measurements alone do not). Noise control principles can be considered in terms of reduced excitation, increased damping, vibration isolation or acoustic shielding and absorption.

From these principles, *actual designs can be developed and tested*, first using the prediction model, then ultimately in practical tests. It would, of course, be risky to proceed straight to this last step. The source or path that is treated may not be the dominant one, or the modification introduced may not influence the source as intended. Yet there are many examples in railway noise control where this has been done, often leading to the conclusion ‘we’ve tried that and it doesn’t work’!

Before noise control measures can be applied in normal operation, *practical constraints have to be taken into account*. The measures that have been developed in principle have to satisfy many other requirements of the operating environment. In the case of the railway these are particularly related to safety. At this point compromise is often required in the acoustic design. Ideally such constraints should be considered as early as possible in the design process, provided that they don’t stifle innovation altogether!

4. MODEL FOR ROLLING NOISE AND IMPLICATIONS FOR NOISE CONTROL

Figure 1 shows a simplified schematic diagram of the mechanism by which rolling noise is generated. This basic model was first put forward by Remington [17] and now forms the basis of the TWINS model [18, 19]. The noise is radiated by the vibration of the track, principally the rails, and of the wheels [16]. In turn, this vibration is induced by the unevenness (roughness) of the wheel and rail running surfaces. This roughness causes a relative motion between the wheel and rail, the vibration amplitude of each component depending on their dynamic properties. For roughness of wavelength λ (in m), the frequency of excitation, in Hz, is given by $f = V/\lambda$, where V is the train speed (in m/s). Relevant wavelengths are between about 5 and 200 mm. Typical amplitudes are of the order of microns. Wavelengths that are short compared with the contact patch length (typically 10-15 mm) are attenuated in their excitation of the system, the so-called contact filter effect [17, 20].

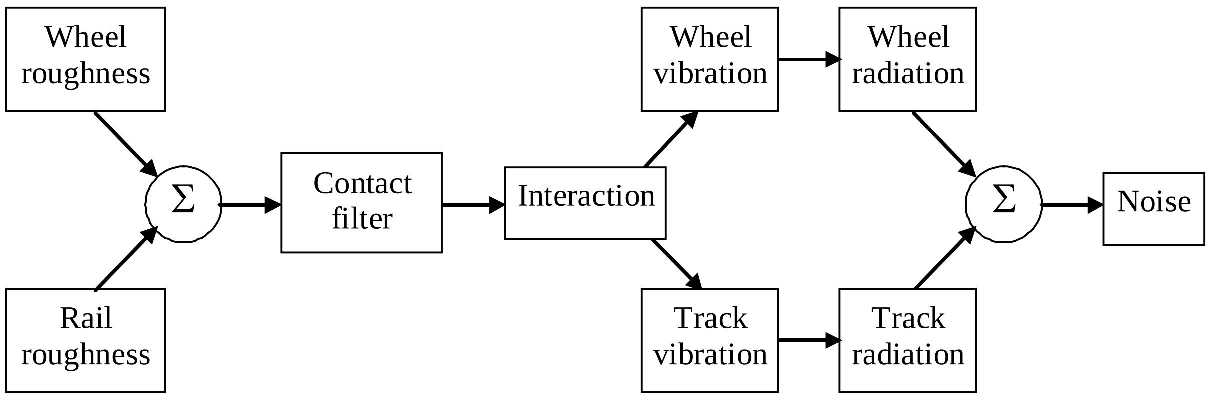

Figure 1. Model for rolling noise generation.

The surface roughness introduces a relative displacement between the wheel and rail in the vertical direction. The actual motion of the wheel and rail at the contact depends on whether the rail or wheel has the highest mobility (Y_R, Y_W). The Hertzian contact spring (mobility Y_C) also plays a role. The equations of motion for interaction via a single degree of freedom at circular frequency ω are:

$$v_R = \frac{i\omega r Y_R}{Y_R + Y_W + Y_C}\ ;\ v_W = \frac{-i\omega r Y_W}{Y_R + Y_W + Y_C} \qquad (1)$$

where v_R and v_W are the vertical velocities of rail and wheel at the contact point and r is the roughness amplitude. Comparing typical mobilities in Figure 2, it can be seen that the rail has the highest mobility in most of the region 100-1000 Hz. Consequently, $v_R \approx i\omega r$ in this region. The wheel response is higher at high frequencies, where a number of lightly damped resonances are found with strong radial components.

An example of the contributions of wheel, rail and sleepers to the overall noise as calculated by the model is shown in Figure 3. This is for a freight wheel design running at 100 km/h and in this case the rail is the most important source, the wheel contribution being about 4 dB lower overall. As the speed or roughness spectrum change, the relative importance of different frequency bands changes, although the relative levels of each source within a given frequency band remain the same. Thus for higher speeds the importance of higher frequencies increases and the wheel component becomes more important.

It can be seen from this model that the noise can be controlled by reducing the roughness of the wheel or rail, or both, depending on which is initially dominant. This reduces the vibration response of both wheel and track and the corresponding sound radiation in direct proportion to the reduction in combined roughness.

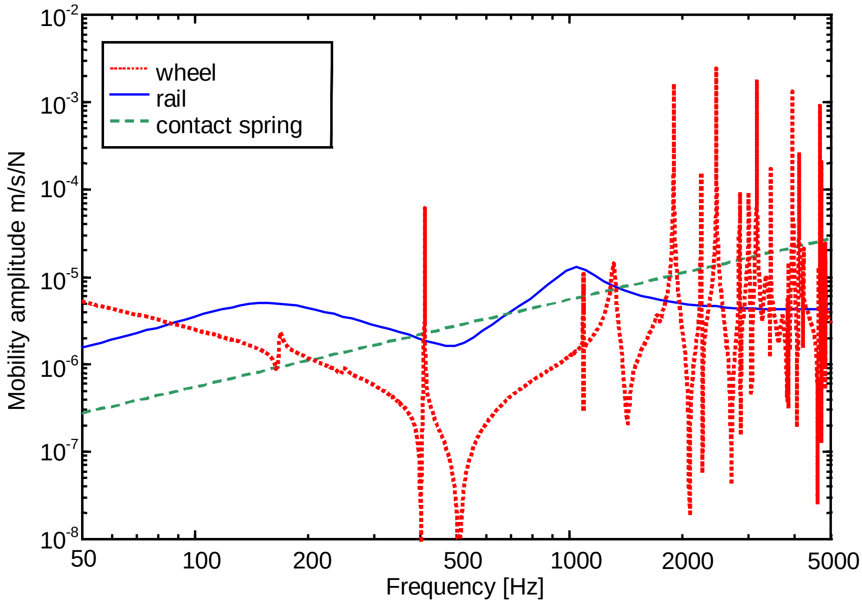

Figure 2. Typical wheel, rail and contact spring mobilities.

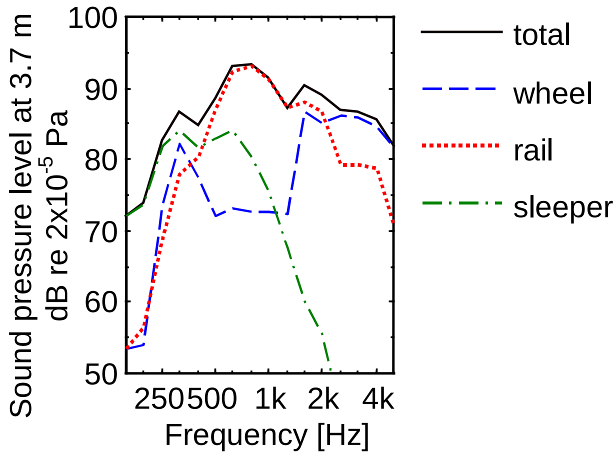

Figure 3. Components of rolling noise for a freight wheel at 100 km/h on track with medium stiffness pads [19].

Changes to the frequency response of the track, Y_R, have minimal effect on the noise radiation in the region where it dominates the spectrum, as seen from equation (1). The parameter with the strongest influence on the amount of noise radiated by the rail is the rate of decay of vibration along the rail, usually expressed in dB/m [21]. A doubling of decay rate in a particular frequency band leads to a reduction of noise from the rail in that band by 3 dB. Measures that aim to introduce damping to the rail are therefore likely to be among the most effective in reducing track noise.

For the sound radiated by the wheel, the main contribution occurs at high frequency, usually between about 1.5-2 kHz and 5 kHz, as seen in Figure 3. The upper limit is determined by the contact filter effect. In this frequency region lightly damped wheel resonances are excited with a large radial component. Due to asymmetry of the wheel cross-section these modes also have large axial components on the wheel web, which radiate sound efficiently. Both radial and 1-nodal-circle axial modes are strongly excited. Their motions are coupled, that is both sets of modes contain both radial and web axial motion [22].

There is also potential to reduce the sound radiation by applying local shielding.

The theoretical model thus provides an indication of which parameters can be influenced to reduce noise. In sections 7 and 8 below various practical implementations that have been developed are described. First however, two practical issues are discussed.

5. WHY DO MICROPHONE ARRAY MEASUREMENTS ALWAYS INDICATE THAT THE WHEEL IS DOMINANT?

Experimental methods are also available for separating the contributions from different sources. Microphone arrays have been used by a number of authors to study railway noise, e.g. [23-28]. These are particularly useful for measuring aerodynamic noise sources. However, it is noteworthy that in studies of rolling noise the microphone array tends to give greater prominence to the wheels and less to the track than the theoretical models described above. There are a number of possible reasons for this:

(a) The wheels are relatively compact sources whereas the rail is an extended source; compact sources show up more clearly in source maps.

(b) The rail source is strongest directly below the wheels and can be mis-interpreted as being wheel noise.

(c) The rail consists of a coherent line source, whereas the analysis of microphone array results is based on the assumption that the sources are a distribution of incoherent point sources.

In [29] it has been shown that this last effect can be significant. The radiation from a propagating wave in the rail occurs predominantly at an angle to the rail that is determined by the ratio of the structural and acoustic wavenumbers (typically 10-30° from the normal [30]). As seen in Figure 4(a), a peak in sound pressure from the rail is measured at a point displaced from the forcing point. However, when a microphone array is directed perpendicular to the track it actually suppresses this component of sound, 'seeing' only the much smaller contribution from the vibration near-field close to the wheel/rail contact points. The rail contribution can thereby be under-estimated by more than 10 dB, particularly in the frequency region where the decay rate is low and the rail contribution is significant, see Figure 4(b). For softer rail pads this effect can be even greater [29].

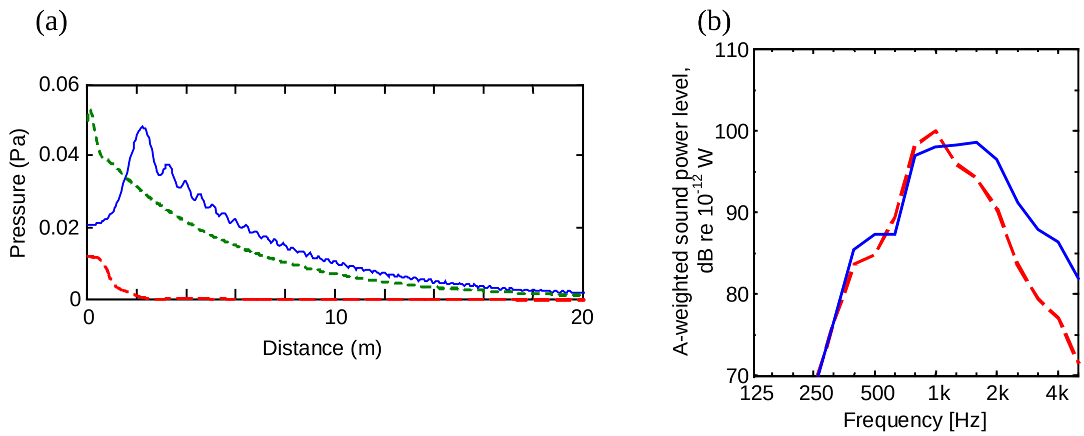

Figure 4. Results of simulations of microphone array measurements of noise from the rail. (a) Results at 1.6 kHz, —— magnitude of sound pressure at single microphone at 5.7 m from the rail versus distance along the track from the forcing position, – – – output from microphone array at 5.7 m, - - - relative source strength obtained from rail vibration (not to same scale). (b) Effect on rail component of noise, —— from TWINS prediction, – – – as inferred from microphone array [29].

6. THE INTERACTION BETWEEN WHEEL AND RAIL

Before turning to specific noise reduction measures, some implications of the wheel/rail interaction are also considered. As seen in Figure 3, both the wheel and the track produce noise, in both cases due to their vibration. Reductions to both components are needed. In most situations, although not all, measures aimed at reducing one component have little or no effect on the other. As the vehicles and infrastructure are often owned by different parties, at least in Europe, neither party is able to influence the overall noise significantly alone.

Another complication is that the vibrations of both wheel and rail are induced by the combination of their roughnesses. A typical situation is that wheels fitted with cast-iron brake blocks have large roughness with wavelengths around 40-80 mm. For a train speed of 100 km/h this excites frequencies around 400-800 Hz where the track radiates most sound, see Figure 3. In such a situation is the vehicle or the track responsible for the noise? Clearly both are.

This interplay between the properties of the vehicle and track is one of the reasons that ‘drive-by limits’ for rail vehicles have taken so long to implement compared with road vehicles. The measurement method in new version of ISO3095 [31] and the noise limits introduced within the TSIs [5, 6] are intended to quantify the ‘vehicle noise’ in a way that minimises the influence of the track. Thus the tests are to be carried out on a track with a roughness spectrum below a specified limit and with a low radiation contribution, defined in terms of the rail pad stiffness or the track decay rate. However, the track contribution to both roughness and sound radiation is not completely eliminated. Moreover, the results obtained are not directly relevant to ‘normal’ operation where the rail roughness may be higher and the track radiation greater. As for road vehicles, a situation has been created where reductions obtained in the test do not necessarily translate into reductions in normal operation.

7. REDUCING EXCITATION

7.1 Braking Systems

As noted already, cast-iron block brakes are known to lead to a corrugated wheel surface which gives a rolling noise level of up to 10 dB more than a disc-braked vehicle [10]. Whilst passenger rolling stock has increasingly used disc brakes, until recently freight vehicles in Europe have retained cast-iron brake blocks. This is due to the higher cost of disc brakes, the difficulty of running trains of mixed braking systems and most importantly the fact that UIC (International Union of Railways) rules for interoperability of freight vehicles actually required cast-iron brake blocks.

Once it was recognised that noise from freight traffic was the main railway noise source, and that there was a high reduction potential from having smoother wheels, the development of braking systems which ensured smooth wheels was commenced within the UIC programme [32, 33]. The main issues identified were then not so much the development of prototype solutions, but rather the adaptation of known solutions from passenger to freight traffic.

Composite brake blocks were already in use for passenger rolling stock and were proven solutions both in terms of general application to the railway system and acoustically, giving a noise reduction of up to 10 dB. Distinction is made between K-blocks (composite blocks) of which several are available and LL-blocks which are composite or sintered metal blocks with a similar friction characteristic to cast iron and could therefore be used as a simple retrofit in existing wagons.

Application of composite blocks to freight vehicles involved selection of suitable composite brake blocks, assessing their general braking performance, and developing wheels capable of withstanding higher temperatures due to the use of K-blocks. In parallel, investigations were carried out into other system problems such as ensuring that electrical conductivity was sufficient to operate track circuits, and winter time behaviour in Northern European countries. Two types of K-blocks were homologated by UIC in 2003 and investigations into remaining technical questions for LL-blocks have continued with the intention of allowing a preliminary homologation to be turned into a definitive one [33].

The main instrument intended to stimulate the introduction of wagons with K or LL blocks is the TSI for noise from conventional rail vehicles [6]. This requires new or refurbished wagons to satisfy noise limits which effectively rule out the use of cast-iron blocks. Examples of measured results from freight wagons are plotted in Figure 5 along with the limits, which depend on the number of axles per unit length (APL). It is clear that vehicles with cast-iron brake blocks exceed these limits. However, recent reports from Germany suggest that, since most wagons are ordered in small batches, around two thirds of new wagons are actually allowed exemptions from the TSI procedure and are still being built with cast-iron block brakes.

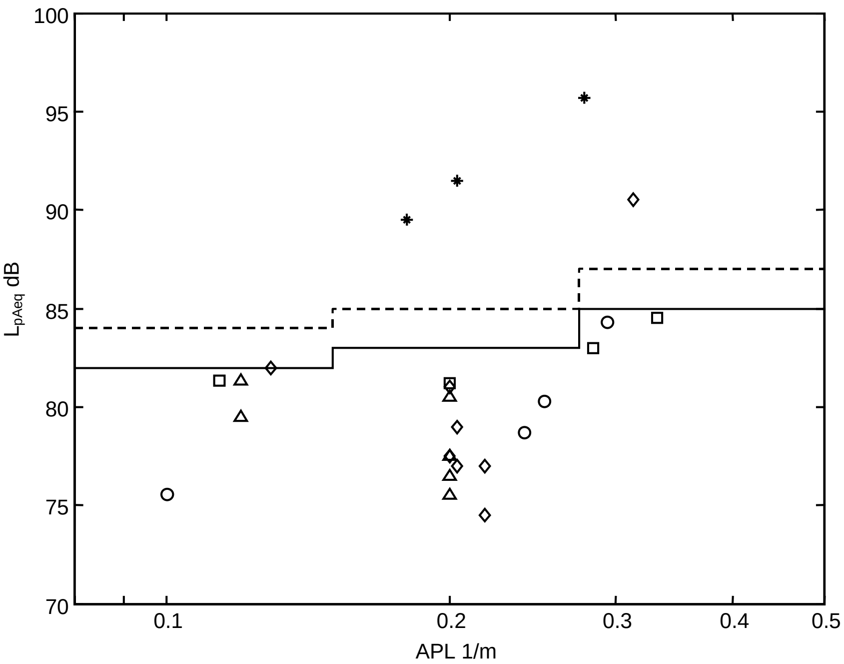

Figure 5 Equivalent pass-by noise level measured at 80 km/h at 7.5 m for various types of freight wagon in different countries, plotted against number of axles per unit length (APL). ◊, Switzerland (K-blocks); Δ, Austria (K-blocks); ○, France (K-blocks); □, Germany (K-blocks); *, cast-iron blocks; —, TSI limit for new rolling stock; – – TSI limit for refurbished rolling stock [11].

7.2 Rail roughness

Unlike the replacement of brake blocks, there is no simple measure that can be used to minimise rail roughness apart from rail grinding. Although the conventional rail grinding process using rotating stones can make the rail considerably smoother, a noticeable tonal peak is left. This has a wavelength corresponding to a resonance of the grinding stones and their drive system. This wavelength depends on the speed of the grinding train and typically occurs at about 20 to 30 mm. This corresponds to a frequency of about 2.5 to 4 kHz at a train speed of 300 km/h. This peak is gradually worn away by passing trains but it would be preferable to avoid it in the first place.

‘Acoustic grinding’ refers to rail grinding carried out specifically to reduce noise levels. Such a procedure is applied in Germany where sections of ‘specially monitored track’ may be designated [34]. These can be associated with a source term in environmental calculations that is 3 dB lower than normal. This has been officially recognised since 1998 and now covers almost 1000 km of the DB network. The track is monitored twice annually by measuring the sound using a dedicated monitoring coach. The track is ground when the level exceeds a particular limit (6 dB above the expected level after grinding). The roughness is also monitored. A typical grinding interval is 4 years. The ‘acoustic’ grinding procedure involves planing or milling followed by grinding using oscillating stones [34]. A similar acoustic grinding procedure has been selected for the new high speed line in the Netherlands.

8. REDUCING WHEEL AND RAIL VIBRATION AND RADIATION

8.1 Reducing Wheel Noise

Various measures applied to the wheel were tested in the EU project Silent Freight [35, 36], the earlier UIC project OFWHAT [37, 38] and the French project VONA [39, 40]. Examples of wheel designs from these projects are shown in Figures 6 and 7. These involved added damping treatments, wheel shape optimisation and attempts to reduce the radiation. These various aspects are discussed separately below (see also [11]).

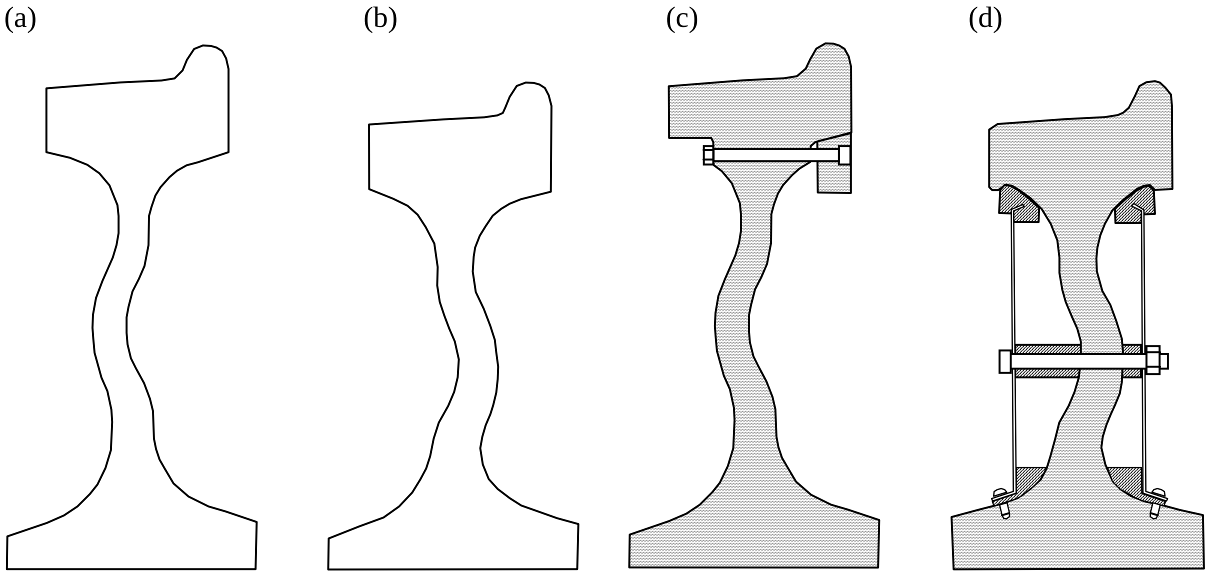

Figure 6. Prototype wheels developed in Silent Freight EU project [11, 35]. (a) Reference wheel 920 mm diameter, (b) shape-optimised 860 mm diameter wheel, (c) reference wheel fitted with ring damper, (d) 860 mm shape-optimised wheel fitted with web shields.

8.1.1 Wheel damping

As noted already, a railway wheel is lightly damped (see Table 2) and various treatments can be considered to increase its damping. To reduce curve squeal small increases in damping are often sufficient but, due to the wheel/rail interaction, considerable damping has to be introduced for a significant effect on rolling noise. The equivalent damping loss factor of a wheel when rolling on a track has been found from experiments to be around 0.01-0.02 at 2 kHz, falling with increasing frequency [41]. The added damping has to exceed this level to achieve a significant effect on rolling noise.

Many practical designs of wheel damper have been produced and introduced. These include systems based on constrained layer damping applied on the wheel web, tuned absorbers mounted on the inside of the tyre or on the web, laminated cover plates and various ‘friction’ dampers. Their effectiveness depends on the initial balance between wheel and track radiation in the overall noise. Thus dampers achieving reductions of 5-8 dB in Germany in the 1980s were found to give only 1-3 dB reduction in tests in the Netherlands and France [11].

The ‘effect’ obtained also depends on the comparison made. In some cases the wheel has to be machined to accept the damper (see Figure 6(c,d)). This would have the effect of making it noisier initially. If comparisons are made between the same wheel with and without dampers a misleading result is therefore obtained.

In the Silent Freight project tuned absorbers, tuned to a single frequency of 2.2 kHz, were attached to the inside face of the tyre (at a similar position to the ring damper in Figure 6(c)) [11]. The wheel component was thereby reduced by 4-5 dB [42]. Earlier results in the OFWHAT project with a different type of tuned absorber fitted to the wheel web also gave reductions of 4-5 dB [37, 38]. Constrained layer damping treatments were also considered in Silent Freight, although not actually implemented. They were predicted to give similar reductions to the tuned absorbers provided that a sufficiently stiff constraining layer is used and both sides of the wheel web are covered, including the inside of the tyre region [43]. The ring damper in Figure 6(c) gave more modest reductions [11].

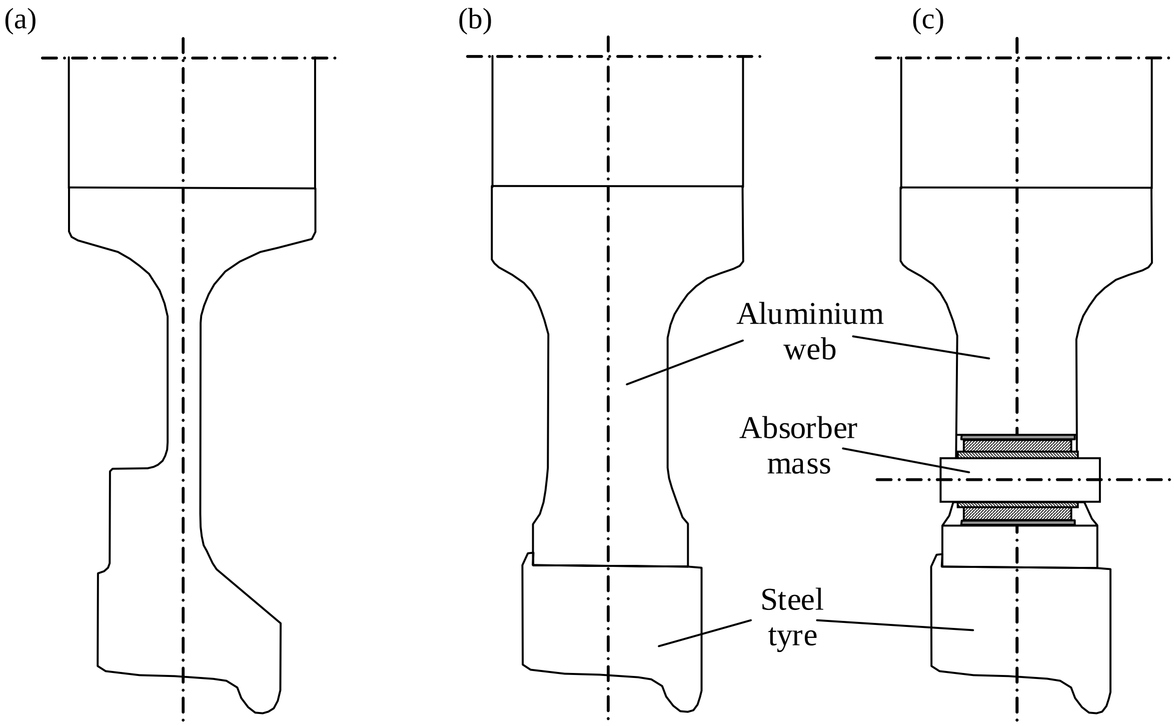

Figure 7. Prototype wheel designs developed in French project RONA [11, 39, 40]. (a) ‘Symmetrical’ cross-section, (b) aluminium web, (c) aluminium web with tuned absorbers.

8.1.2 Wheel shape optimisation

Changes in the wheel shape can also be used to reduce noise, as has been successfully demonstrated in several projects. The main principles are

(a) to decouple the radial motion of the wheel from its axial motion (by making the cross-section as symmetric as possible),
(b) to increase the natural frequencies as much as possible (reduced diameter, thicker

web, larger fillets),

(c) to reduce the radiating area (reduced diameter).

The natural frequencies of axial 1-nodal-circle modes can be increased considerably by a using a smaller diameter, although the radial modes are affected much less [22]. In one study, by reducing the diameter from 900 mm to 740 mm and using a straight web of thickness 37 mm a reduction of over 10 dB in wheel component was calculated [44].

Small wheels are already fitted to some freight vehicles. Example results are shown in Figure 8 for two wheel designs [45]. The 730 mm diameter wheel, Figure 8(a), which is fitted to car-carrying wagons, has its first radial resonance in the 2.5 kHz band. The wheel component is only dominant above this frequency. The 360 mm solid webbed wheels, Figure 8(b), are fitted to some 'rolling motorway' vehicles used for carrying lorries. These have a very low wheel contribution, as much as 20 dB less than a standard wheel. However, the track contribution is *increased* by about 2 dB as a result of the contact patch becoming shorter, leading to a loss of some of the contact filter effect [45].

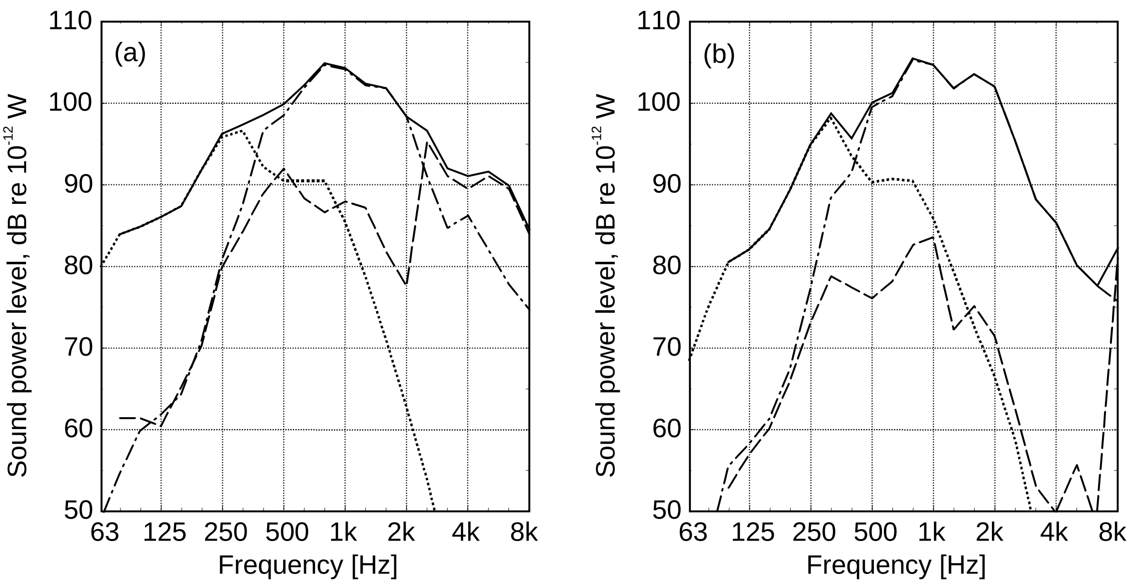

Figure 8. TWINS sound power predictions for small wheels at 100 km/h (a) 730 mm diameter wheel, (b) 360 mm diameter solid web wheel. ——— total, – – – wheel, ······· rail, – · – · sleeper [45].

On tread-braked wheels the web is curved to allow for thermal expansion during braking (see Figure 6(a)). This limits the scope for optimisation in terms of 'symmetry'. Nevertheless, the wheel shown in Figure 6(b) was found to be about 3 dB quieter than the reference wheel for only a modest reduction in diameter (920 to 860 mm) and increase in web thickness [43].

For disc-braked wheels there is more scope for using a shape-optimised wheel which can have a straight web. The wheel running surface, however, must remain asymmetrical. The wheel in Figure 7(a) was intended to compensate for this by adding mass on the inside of the tyre region. The diameter was unchanged at 920 mm due to the need to retrofit these wheels to existing vehicles. Tests on the French TGV showed reductions in overall noise of 2 dB and up to 5 dB in the wheel component of noise [11].

Also tested was a wheel with a thick aluminium web, shown in Figure 7(b). This reduced the wheel component by about 6 dB; in combination with tuned absorbers (Figure 7(c)) the benefit increased to about 12 dB. However, after the fatal accident at Eschede in Germany in 1998 caused by the failure of a resilient wheel, development of such wheels involving multiple materials was stopped.

8.1.3 Other ways of reducing wheel noise

Cover plates can be fitted to the wheel to reduce sound radiation from the web, as shown in Figure 6(d). This wheel is also shown in Figure 9(a). These web shields are resiliently mounted and are constructed of 1 mm steel to ensure a low radiation efficiency whilst providing sufficient shielding. It was estimated that the wheel component was reduced by 8 dB compared with the reference wheel due to a combination of shape optimisation and the web shields [11, 35].

An alternative concept is shown in Figure 9(b). Here the wheel web was perforated to reduce its radiation efficiency. Whilst this reduced the wheel noise at low frequencies, the effect was negligible above 1 kHz where the wheel component is most significant.

(a) (b)

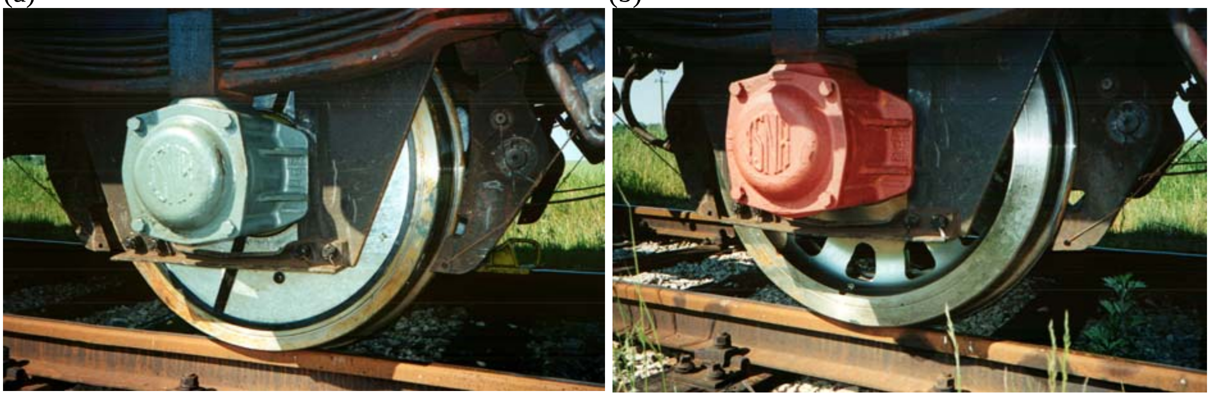

Figure 9. Prototype wheels developed in Silent Freight EU project. (a) Shape-optimised 860 mm wheel fitted with web shields, (b) wheel with perforated web (photos: ERRI).

8.2 Reducing Track Noise

8.2.1 Rail pad stiffness

In recent years, it has become common to use soft pads to protect sleepers and ballast from impact loading. Unfortunately, soft pads may lead to relatively low decay rates, as shown in Figure 10(a), and therefore higher noise levels [46]. Conversely, stiff pads increase the rail decay rates but they increase the dynamic coupling to the sleeper and therefore its noise contribution. The peak seen in the track accelerance in Figure 10(b) corresponds to the frequency at which the sleeper is decoupled from the rail; above this frequency the track decay rate drops sharply.

Figure 11 shows the dependence of noise on rail pad stiffness. A compromise can be obtained when the noise contributions from rail and sleeper are equal [46] but this occurs with quite stiff pads. Such an approach was demonstrated in the OFWHAT project [37]. However, stiff pads are undesirable as they provide insufficient protection from impact loading. Moreover they are believed to be more likely to lead to higher rail roughness levels [47]. Thus alternative methods of increasing track decay rates are sought.

8.2.2 Rail damping

Rail dampers can be used to increase the track decay rates whilst maintaining a relatively soft rail pad. Studies of constrained layer damping indicated that this provided insufficient damping, except at frequencies above 2 kHz. Therefore several designs of tuned absorber have been developed. In the OFWHAT project an absorber was designed that was attached to the rail foot [37]. Although it was only tested in combination with stiff pads, it was shown to be effective in reducing the track noise.

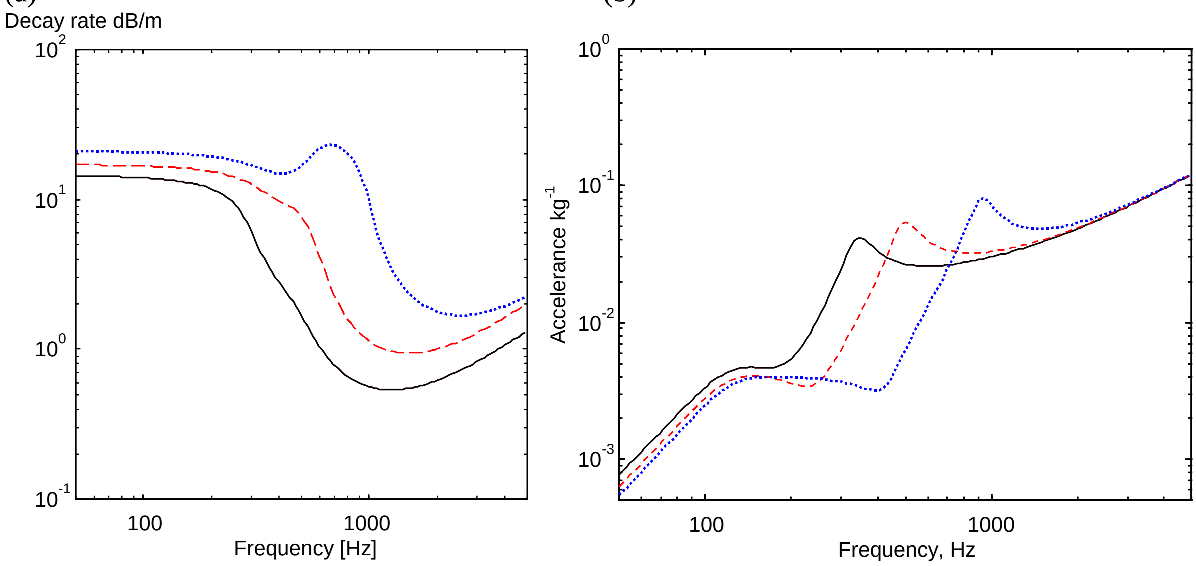

Figure 10. Predicted effect of rail pad stiffness on track dynamic behaviour. (a) Vertical decay rates; (b) vertical accelerances of track fitted with corresponding rail pads, —— 140, – – 300 and ······ 1000 MN/m.

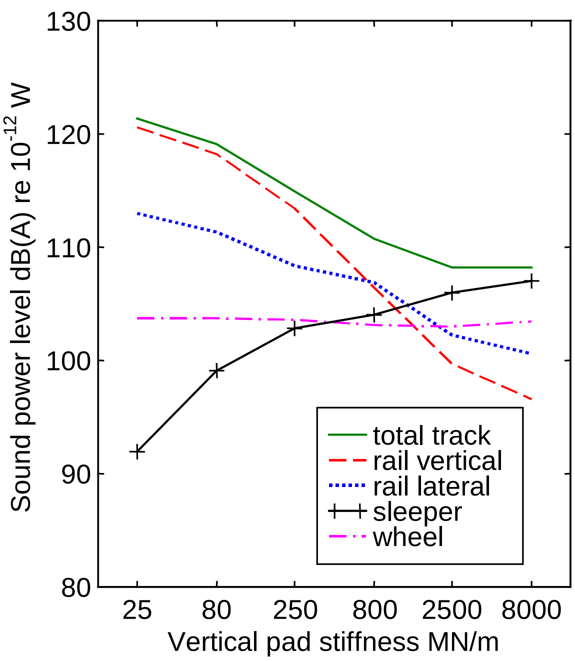

Figure 11. Example of predicted sound power due to one wheel and the associated track vibration versus high frequency rail pad stiffness. Calculations using TWINS for a standard 920 mm freight wheel at 100 km/h, with a typical tread-braked roughness.

In the Silent Track project this principle was developed further, leading to the rail damper shown in Figure 12(a) [48]. This was designed to be attached continuously along the rail, avoiding the end of the foot, where rail fasteners are applied, as well as the region under the rail head as required by rail lifting equipment. In field tests, reductions of 6 dB in the track component of noise were demonstrated, see Figure 12(b), although the effect will be smaller on track with stiff pads. A number of practical installations have followed, e.g. in the Netherlands, showing them to be a robust design.

A variant of this design which can be clipped onto the rail between sleepers is shown in Figure 13. This has been successfully tested in France and shown to give reductions of around 3 dB on sites with medium stiffness pads [49]. Several other types of rail damper are also now commercially available.

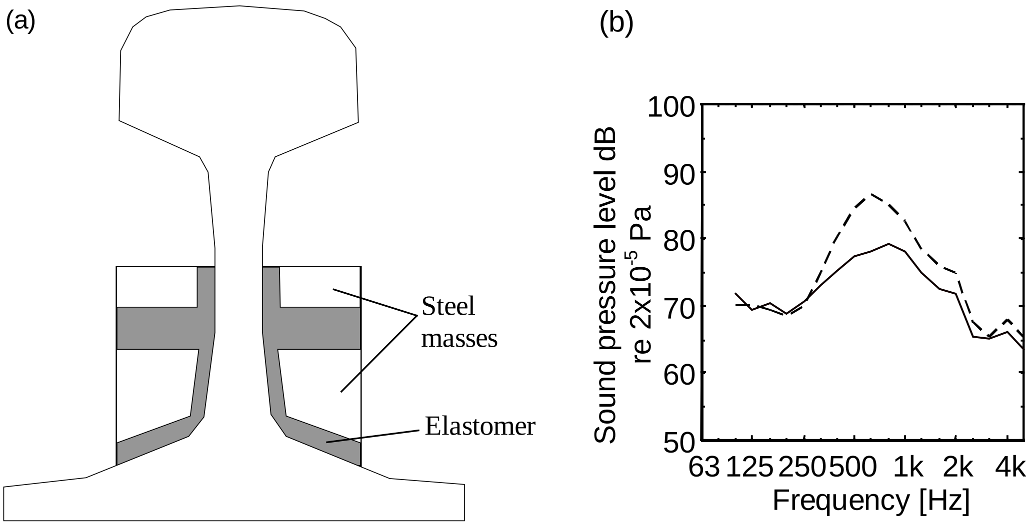

Figure 12. (a) Rail damper developed in Silent Track EU project. (b) Measured reduction in overall noise due to rail damper tested with a low-noise wheel design. - - - reference track (overall A-weighted level 90.5 dB), — track with damper (overall A-weighted level 85.1 dB) [48].

Figure 13. Discrete form of rail dampers applied at SNCF test site [49].

8.3 Local Shielding Measures

Shielding measures applied locally to the source can also reduce the wayside noise. One such application was demonstrated in [50] where reductions of 8-10 dB were shown to be possible by a well designed enclosure around the bogie in combination with low trackside barriers.

However, there are considerable practical difficulties with such a design, including maintenance and ventilation, and also the different structure and vehicle gauging requirements on different railway systems. In the Silent Freight and Silent Track projects a system of bogie shrouds and low barriers was designed that was to be compatible with a number of European railway systems. Consequently a gap of 118 mm remained between the top of the barrier and the bottom of the shroud so that the overall noise reduction was limited to at most 3 dB [51].

8.4 Slab track designs

Slab track has been used for a number of new lines in recent years, in particular on high speed lines in Germany. Slab tracks offer advantages such as reduced maintenance but they are usually 2-5 dB noisier than ballasted track. The main reason for this is the use of rail fasteners with a reduced stiffness [52]. This leads to increased noise from the rail due to lower decay rates (see Figure 11). The acoustic absorption is also lower due to the elimination of the ballast, but this only accounts for about 1-2 dB in the wayside noise.

The introduction of absorptive treatments and partial shielding, such as that shown in Figure 14, have been demonstrated on slab tracks in Germany [52]. The full treatment shown in Figure 14 gave a reduction of about 6 dB. Omitting the raised barriers lowered the benefit to about 4.5 dB. A more limited absorptive treatment which left the rail exposed gave a noise reduction of about 3 dB, which was nevertheless sufficient to make the track equivalent to a conventional ballasted track with concrete sleepers [52].

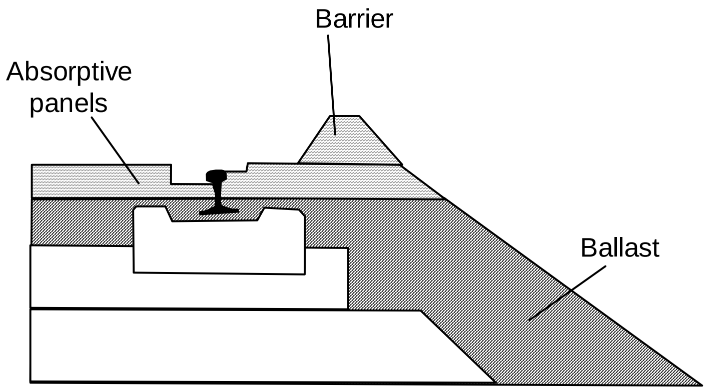

Figure 14. Absorptive treatment applied to conventional slab track [52].

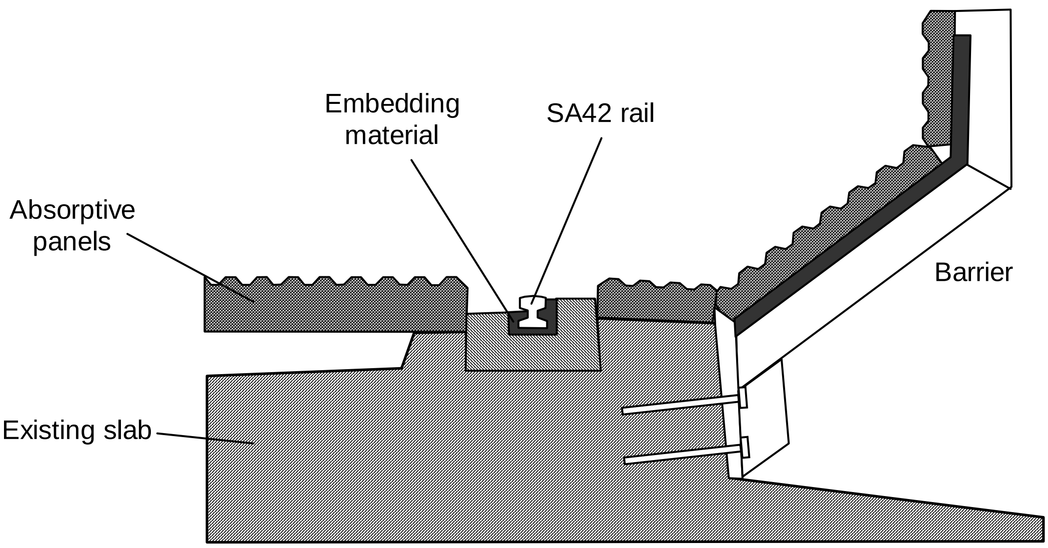

Figure 15. Slab track with SA42 rail section, absorptive treatment and integral mini-barriers developed in Dutch project STV [11, 53].

In the Dutch 'Quiet Rail Traffic' (STV) project a novel slab track design was developed, shown in Figure 15 [53]. A new small rail profile, SA42, was developed which was embedded in a channel in the slab in a relatively stiff embedding material. This combination meant that the track decay rates were relatively high, yet the high impedance of the slab caused it to have a low vibration and hence sound radiation. Reductions of about 3 dB in track noise were found relative to standard ballasted track; reciprocal vibro-acoustic measurements indicated a greater reduction of around 7 dB [11, 53]. A side effect of this design, however, is that the track is relatively stiff whereas slab tracks usually require resilient fasteners to compensate for the flexibility of the ballast.

In addition, integral barriers were tested on this novel slab track as shown in Figure 15. These were larger than the barriers on the conventional slab in Figure 14. They gave additional noise reductions of 6 dB, while absorptive treatments on top of the slab gave an extra 2 dB reduction.

9. PRACTICAL LIMITATIONS

The railway is a safety-critical environment, and important aspects that must be taken into account in any new designs include structural integrity, safe operation and electrical conductivity for operation of signalling systems. Historically, many of these aspects have become defined by conventions that are hard to challenge, being based on established practice. For example, how much should the rail be allowed to deflect under a train? In many cases this is the quantity that is limited in order to prevent the rail from suffering excessive stress and to avoid gauge spreading. However, in practice the vertical deflection itself might be increased by careful design without any adverse effects on the stress, while the gauge can be maintained by a stiffer support laterally. Such changes are often hard to introduce into the railway culture.

Although both the noise reduction and the potential for implementation of prototype designs have been demonstrated in research projects, the railway industry is cautious about introducing such new technology. For example, phase 1 of the Channel Tunnel Rail Link from London to the Channel Tunnel in the UK faced considerable pressure to manage environmental noise, but it was decided to use only ‘proven technology’, mostly conventional noise barriers and changes in alignment [54]. This policy was identified as contributing to the commercial success of the project. For the Dutch high speed line, HSL-Zuid, an initial proposal to use a novel slab track with embedded rail was changed to a conventional slab track before construction started. Clearly, from these experiences, it is unrealistic to expect large new construction projects to adopt new technology, but a way has to be found to develop such technology from the prototype stage to the point where it is accepted as proven technology and no longer perceived as a source of unacceptable risk.

The Dutch Innovation Programme Noise (IPG) [55] has helped in this objective by financing pilot installations of a number of technologies, including K and LL brake blocks and various types of rail damper. This has allowed experience to be gained in their use, including extensive life-cycle cost analysis. Meanwhile, the development of legislation, particularly the TSIs, will provide an increasing stimulus to consider noise-reducing measures as part of the procurement of new vehicles and track. Many of the techniques described in this paper are therefore likely to become more commonplace in the future.

10. CONCLUSIONS

In conclusion we return to the question: are the trains getting any quieter? Technological solutions are being developed but implementation is lagging behind. The biggest change in recent years has been the gradual elimination of cast-iron brake blocks from passenger rolling stock. Yet this is a step that was understood 25 years ago and which is still far from complete, particularly for freight vehicles. The gain of 10 dB has at least in part been offset by increases in speed. The question remains that was posed to the author when he first joined the acoustics team of British Rail in 1982: how can we obtain the *next* 10 dB reduction?

The solution is more complex, but techniques that can be used to achieve this, such as rail dampers, wheel dampers, wheel screens, low barriers and bogie shrouds, are now generally known and have been demonstrated. None of these can achieve large reductions on its own but in combination they can be effective. Nevertheless, there is understandable reluctance to introduce such novel technology into new railway projects. It is a large step from research project to full implementation and pilot implementations, such as seen in the Dutch IPG programme, are needed to bridge this gap and to generate practical experience of their use. Moreover, political and economic hurdles need to be overcome if budgets allocated to noise barriers are to be redirected to stimulate control at source. Even so, hopefully it will not be another 25 years before the trains really do get quieter.

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