

ON TRACK FOR A QUIETER FUTURE

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1 INTRODUCTION

Railways around the world are experiencing a renaissance as an environmentally-friendly means of transport that offers many advantages for sustainable development. This is a far cry from the situation of 50-60 years ago. In the 1960s Beeching's cuts reduced the size of the UK network and closed many rural stations. Then, in the early 1980s, when passenger numbers reached a low point, the Serpell report included an option to reduce the network to just three main lines and a few commuter routes, accounting for just 16% of the current network. Now, however, new lines are being opened, such as the Elizabeth Line through central London and the Borders Railway in Scotland and plans are well-developed for a high-speed line, HS2, connecting London, Birmingham, Manchester and Leeds with train speeds up to 360 km/h. Remarkably, since 2007 over 25,000 km of high-speed lines have been opened in China and there are now substantial metro networks in at least 30 Chinese cities. However, especially in the West, proposals for new lines or expansions of networks are often met with objections from local residents which are based on noise and vibration. Consequently expensive noise mitigation measures have to be included.

Railway operations will clearly never be silent; to move large quantities of passengers or freight at high speeds will inevitably result in the generation of significant noise and vibration through both mechanical vibration and aerodynamic noise. In common with many engineering noise sources, the sound power produced by a moving train is actually a very small fraction of the power required to move it: approximately one millionth. Nevertheless, it is clearly important to minimise the noise and vibration produced by railways without adding unnecessary cost or complication to their construction and operation. To be able to propose cost-effective mitigation measures it is essential to understand the sources of noise and vibration and the parameters that can influence them.

Theoretical models are an important part of this process and have been developed for many railway noise phenomena.¹ As a general principle, models should be of sufficient detail to cover the most relevant parameters in a reliable way, but not over-complicated so as not to lose insight. This approach of developing appropriate models for the purpose of designing mitigation measures is illustrated through several examples and case studies from the author's experience; these cover aspects of rolling noise, aerodynamic noise, ground vibration and curve squeal.

2 ROLLING NOISE

2.1 The TWINS model

In most situations the dominant source of noise from railway operations is induced by the rolling of the wheels along the rails. This produces noise predominantly in the frequency range 100-5000 Hz and has a typical speed dependence of $30 \log_{10} V$, where V is the train speed. The established mechanism is shown schematically in Figure 1. The roughness (or unevenness) of the wheel and rail surfaces induces relative motion of the wheel and rail and noise is then radiated by the resultant vibration.

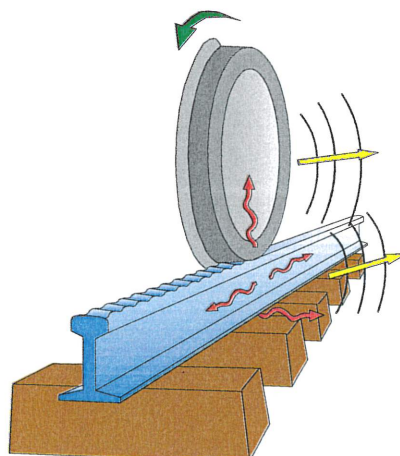


Figure 1. Schematic representation of the mechanism of railway rolling noise generation.

Early models of rolling noise were developed in the 1970s by Remington.^{2,3} He used analytical models for the wheel and rail impedances and for their radiation efficiencies and took account of many features that are still considered as important: relative displacement excitation by the roughness, the contact filter effect, track decay rates, etc. Subsequently this basic model was developed and extended to include other features that were found to be significant⁴⁻⁹ and the models were implemented in the TWINS (Track-Wheel Interaction Noise Software) package, which was validated using extensive field tests.¹⁰⁻¹² Although the TWINS rolling noise model was developed over 25 years ago, it is still widely used in the railway industry and considered as the state-of-the-art. Figure 2 shows a flowchart of the basic model.

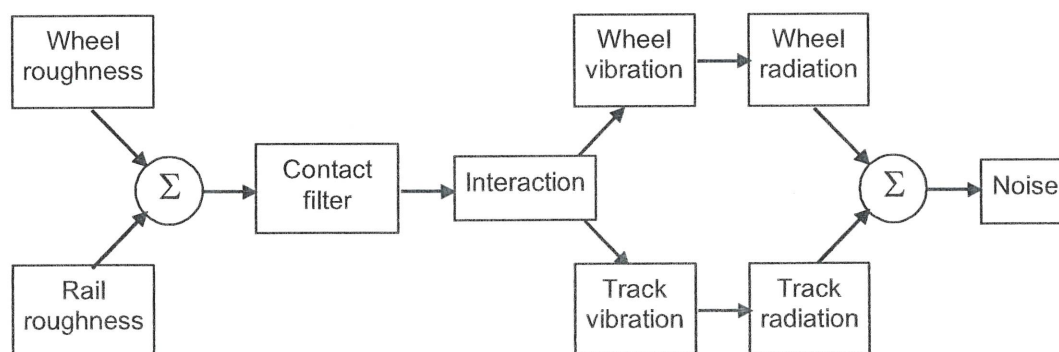


Figure 2. Flowchart of the TWINS rolling noise model.

2.1.1 Roughness excitation

The wheel and rail surfaces may appear smooth but they actually contain small variations in profile known as roughness. The frequency of excitation f is related to the roughness wavelength λ (in m) and the train speed V (in m/s) by $f = V/\lambda$. Relevant wavelengths are therefore between about 5 and 200 mm and 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, giving rise to the contact filter effect. Remington derived an analytical model of the contact filter² and later developed a numerical model based on a set of discrete point-reacting springs within the contact zone.^{13,14} In practical applications tabulated values from this model can also be used.¹ Measurement standards are now available for rail roughness and are being extended to wheel roughness.¹⁵

The surface roughness introduces a relative displacement between the wheel and rail in the vertical direction. This leads to vibration of the wheel, the rail or the contact spring between them, depending on the relative magnitudes of their mobilities. It is found that the rail has the largest mobility over most of the frequency region 100-1000 Hz; consequently, the vibration of the rail is approximately equal to the roughness amplitude in this region. The wheel response is greater at high frequencies, where a number of lightly damped resonances occur with strong radial components.

2.1.2 Wheel vibration

A railway wheel is a lightly damped resonant structure, with similar properties to a bell. The geometry of its cross-section is not amenable to analytical modelling and so finite element (FE) models are used.⁶ Axi-harmonic FE models can be used to take account of the axisymmetric geometry. Examples of modeshapes are shown in Figure 3. The most important modes in rolling noise are the one-nodal-circle axial modes and the predominantly radial modes, whereas the zero-nodal-circle axial modes are important for squeal noise.

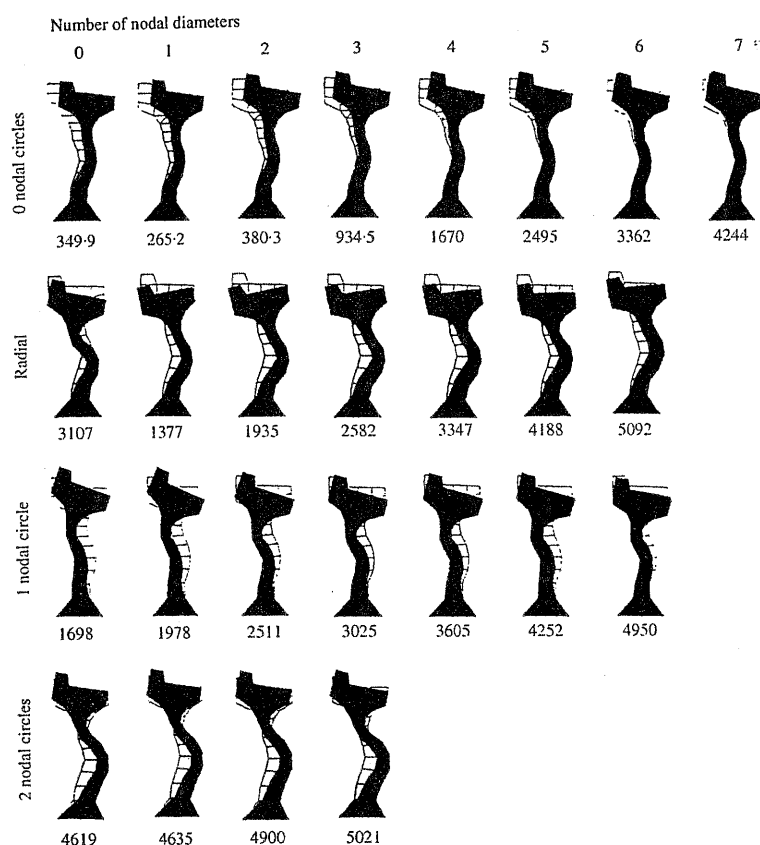


Figure 3. Examples of modeshapes of a railway wheel, showing cross-section through the wheel and listing natural frequencies in Hz.¹

2.1.3 Rail vibration

Unlike the wheel, the rail is effectively an infinite waveguide and its vibration consists of propagating waves rather than resonant modes. The rate of decay of these waves with distance along the rail determines the length of rail that vibrates with each wheel and thereby controls the radiated power. Standard measurement methods are now available to determine the track decay rate.¹⁶ A track with

soft rail pads has a lower track decay rate, and hence greater noise radiation, than one with stiff pads. This is illustrated from measured results in Figure 4.

The vertical dynamics of the rail can be represented by an analytical model of a Timoshenko beam on a two-layer elastic support; the rail pads and ballast are represented by damped elastic layers while the sleepers are represented by their mass or by a transverse beam. The lateral dynamic behaviour is less well represented by this model, as torsion should also be included.¹⁷ At higher frequency, above about 1 kHz for the lateral direction, the cross-sectional deformation of the rail section also becomes important. To account for these effects FE-based approaches making use of periodic structure theory⁷ or waveguide finite elements¹⁸ can be used; these are more efficient than full 3D FE models.

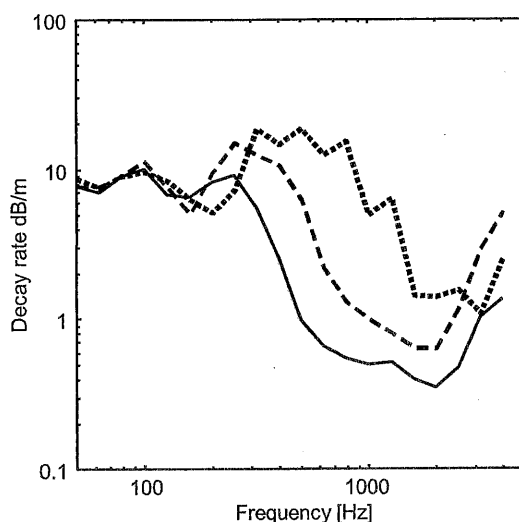


Figure 4. Measured track decay rates for the same track with different rail pad stiffness (—125 MN/m, --- 270 MN/m, ... 950 MN/m).¹⁹

2.1.4 Sound radiation

To calculate the sound radiation from a vibrating object, numerical techniques such as boundary elements (BE) and finite/infinite elements are increasingly used. Nevertheless, they are generally limited to the calculation of sound radiation at low frequencies or for simple geometries.

The acoustic radiation problem can be separated into the estimation of the radiated power and a separate estimation of directivity. The sound power in a frequency band can be found from the surface-averaged mean-square velocity, surface area and the radiation efficiency. The radiation efficiency of a wheel can be represented by simple semi-analytical formulae that have been obtained by curve-fitting to the results from axi-harmonic BE calculations.^{1,20}

To obtain the sound pressure requires additional information about the directivity. However, given the complex geometry of the passing train and the air flow around it, even if the directivity can be correctly predicted in laboratory conditions, the actual situation will be different. To obtain the average sound pressure level during a pass-by, details of the directivity are less important and simple monopole or dipole directivities can be used to good effect.²⁰

For the rail, again a full three-dimensional BE model of a length of rail would be too computationally intensive. However, as the waves in the rail generally have wavelengths much greater than the acoustic wavelength, a two-dimensional BE approach can be used for all except very low frequencies.²¹ Recently, the effect of the proximity of the ground on the radiation from the rail has also been taken into account.²²

An example of rolling noise contributions from wheel, rail and sleepers, as calculated by the TWINS model, is shown in Figure 5. These results are for a modern UK electric multiple unit (EMU) train running at 120 km/h on a track with resilient rail pads. In this case the rail is the most important source, the overall wheel contribution being about 7 dB lower. For different speeds or roughness spectra, the relative importance of different frequency bands changes, although the relative levels of each component 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 in the overall level. For other wheel designs, notably larger wheels with a curved web, the wheel contribution can be more important than the rail.⁴

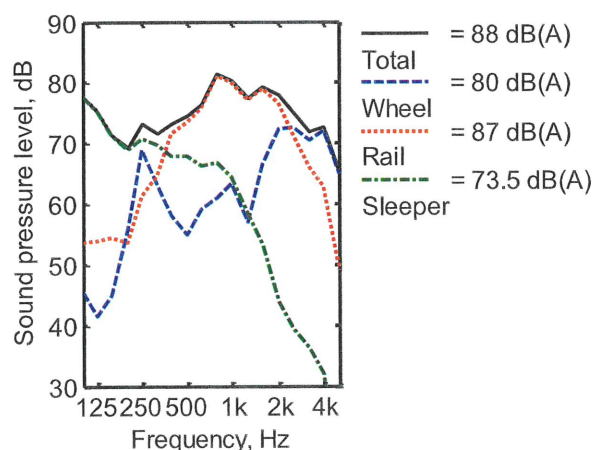


Figure 5. Example of calculated components of rolling noise at 7.5 m from the track for an electric multiple unit at 120 km/h on track with soft rail pads.²³

2.1.5 Example: effect of a track renewal on noise levels

As an example of the use of the model, results are presented of noise measured at a site in the UK.²³ Both tracks had welded rail on concrete sleepers but one track had recently been completely renewed while the other remained in its un-renewed state. Importantly, the renewed track was fitted with 10 mm resilient rail pads normally fitted to modern track while the un-renewed track had thinner, stiffer pads. As well as noise, the rail roughness and track decay rates were measured at the site. In addition, wheel roughness measurements were made on similar rolling stock. Predictions of the noise were found to be close to the measured values, as shown in Figure 6.

On the renewed track, with soft rail pads, the noise levels were 3-4 dB(A) higher than on the un-renewed track. This is due to a reduction in track decay rates (cf Figure 4) which more than offsets a small reduction in rail roughness. The track component of noise was found to be about 4 dB greater than that of the wheel for the un-renewed track and 7 dB more than the wheel component on the renewed track, as already shown in Figure 5.

2.1.6 Slab track

Slab track is usually considered to be noisier than ballasted track, with a typical penalty of 3 dB being cited. One difference between them is that the acoustic absorption of the ballast is missing in the case of a slab track. Measurements of the absorption of railway ballast show an absorption coefficient of up to about 0.6 at high frequencies, reducing at low frequencies, as shown in Figure 7. This reduces the reflected sound from the wheel and rail to some extent but the overall difference in wayside sound pressure level for a track in free field is estimated to be approximately 1 dB.

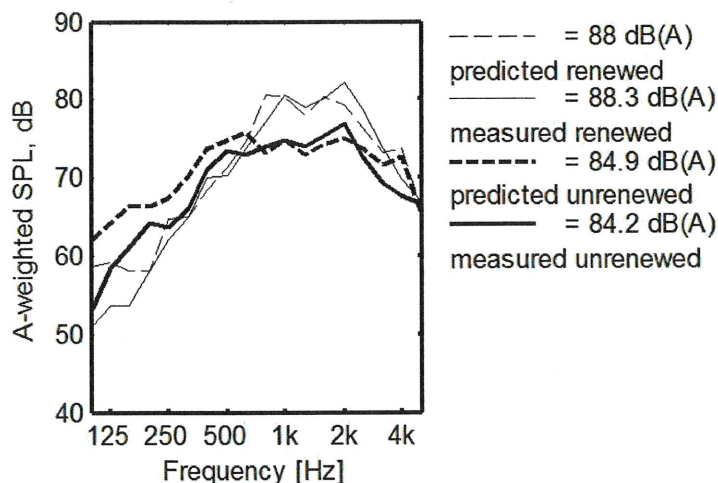


Figure 6. Noise from electric multiple units on a renewed track (thin lines) and an un-renewed track (thick lines), measured (—) and predicted (---) for 7.5 m from the track and a train speed of 110-120 km/h.²³

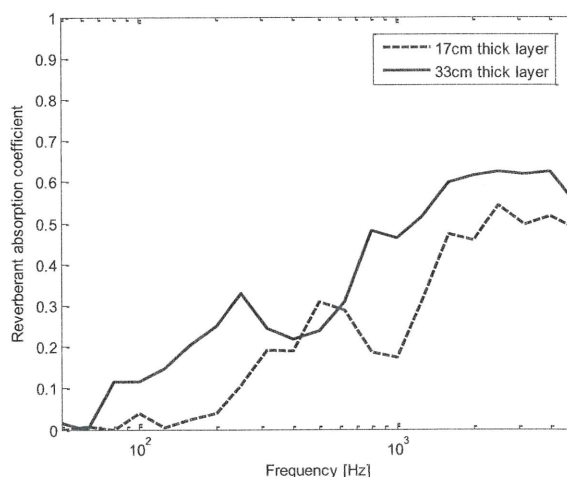


Figure 7. Measurement of diffuse incidence absorption coefficients of railway ballast in a reverberation room. Results are shown for two different thicknesses.²⁴

A more important difference between slab track and ballasted track, however, is the track decay rate. The use of softer rail fasteners on slab track means that track decay rates are typically lower, resulting in higher noise levels. However, as has been seen, ballasted tracks also use a variety of rail pad stiffnesses. The actual difference between a slab track and a ballasted track therefore depends on the types of rail fastener and the pad stiffness used in both cases and in some cases their noise levels could even be equal.

2.2 Mitigation measures

2.2.1 Summary of possible measures

The rolling noise model can be used to identify the parameters that can be modified in order to reduce noise. It is important to identify the dominant contributions and to focus attention on these. It can be seen from Figure 2 that the noise can be controlled by reducing the roughness of the wheel and/or rail, depending on which is initially dominant. This reduces the vibration levels of both wheel and

track and the corresponding sound radiation. In the UK, passenger rolling stock changed from using cast-iron brake blocks to disc brakes from the 1970s and this led to a 10 dB reduction in noise levels. The noise from freight trains in Europe remains an important issue as they have continued to use cast-iron brake blocks but here also changes to the braking system are leading to significant reductions, in this case through new brake block materials.

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.²⁵ A change in decay rate from DR_u to DR_d (in dB/m) leads to a change in noise level (in dB), in the corresponding frequency band, of

$$\Delta L = 10 \log_{10} (DR_u / DR_d) \quad (1)$$

A doubling of decay rate at a particular frequency therefore leads to a noise reduction from the rail at that frequency 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 (see below).

An alternative approach to reducing noise from the rail is to reduce the area of the radiating surface and consequently the radiation efficiency. This was attempted in the design of 'hush rail' which was tested by BR Research in the early 1990s. This had a height of only 70% of the standard rail. Unfortunately, for convenience the field test was implemented using timber sleepers, with a reduced spacing to compensate for the lower bending stiffness of the rail.^{1,26} Consequently, the sleeper noise contribution dominated, so that any benefit from the small rail was masked. A later attempt to use a small rail profile in the Netherlands in a prototype slab track was more successful.^{1,27}

For the sound radiated by the wheel, the main contribution occurs at high frequency, usually above about 1.5 kHz, as seen in Figure 5. In this frequency region, lightly damped wheel resonances are excited with a large radial component, see Figure 3. Due to asymmetry of the wheel cross-section these modes also have large axial components on the wheel web, which radiate sound efficiently.

Finally, shielding measures can be applied to reduce the radiated sound. However, conventional noise barriers are expensive and visually intrusive. An alternative is to fit shields to the bogies of the vehicles. Used in combination with very low trackside barriers these have potential to give noise reductions of up to 10 dB, as shown in field tests by BR Research.²⁸ However, care is required to ensure that there is no gap between the shield and the barrier.²⁹ A fuller discussion of mitigation measures for rolling noise is given in ^{1,30}.

2.2.2 Wheel and rail dampers

Due to its light initial damping, the wheel is a prime target for the application of damping treatments. Tuned wheel dampers have been used at least since the 1980s in Germany but have been found to be less successful elsewhere due to differences in wheel design. The wheel/rail coupling introduces a considerable effective damping which must be overcome before any noise reduction can be achieved.¹

Prototype rail dampers were developed in the 1990s, notably in the Silent Track project, as shown in Figure 8.³¹ For a track with a low decay rate, reductions in track noise of up to 6 dB can be achieved but for track with stiffer rail pads the reductions are much more modest. As wheel and rail damping treatments become more common, the prospect arises of having to test new prototypes in the field, which can be very expensive. Therefore in the German-French collaborative research project, STARDAMP, laboratory test methods were developed for the assessment of wheel and rail damping treatments. The effect of the damping treatments is assessed by combining these test methods with a simple prediction model.^{32,33}

To assess the behaviour of rail dampers, the most important parameter is the track decay rate, as discussed above. In the STARDAMP method, the decay rate of the damped track is estimated by first measuring the decay rate of a free rail with dampers attached. Results can be obtained for

dampers fitted to a 6 m long rail or a longer length. Good agreement was found with the results from in situ tests.³³

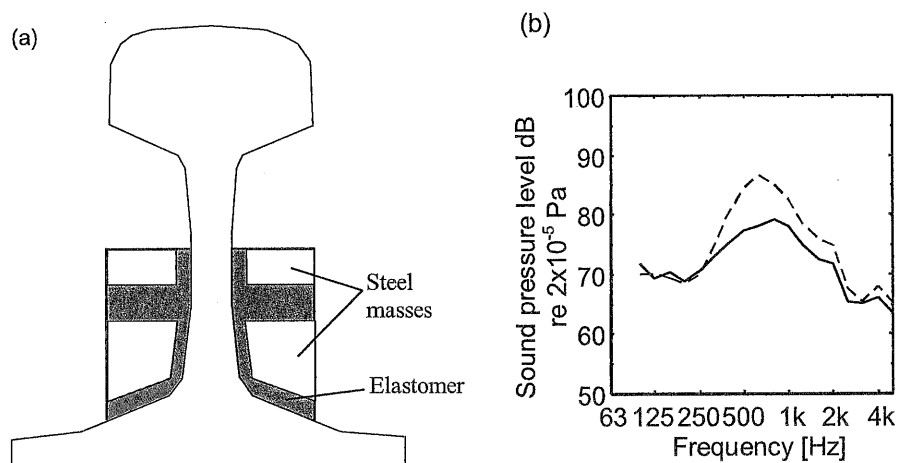


Figure 8. (a) Rail damper developed in Silent Track EU project. (b) Measured reduction in overall noise due to rail damper tested with a low-noise wheel design. - - - reference track (overall A-weighted level 90.5 dB), — track with damper (overall A-weighted level 85.1 dB).³¹

3 AERODYNAMIC NOISE

Aerodynamic noise increases more rapidly with speed than rolling noise and becomes dominant at sufficiently high speeds³⁴, as shown in Figure 9. This shows measured data from a number of high speed trains, together with a line of slope $30 \log_{10} V$ representing rolling noise and a line of slope $80 \log_{10} V$ representing aerodynamic noise; their sum appears to give a good fit to the measured data. This suggests a transition speed, at which the two components are equal, of around 370 km/h.

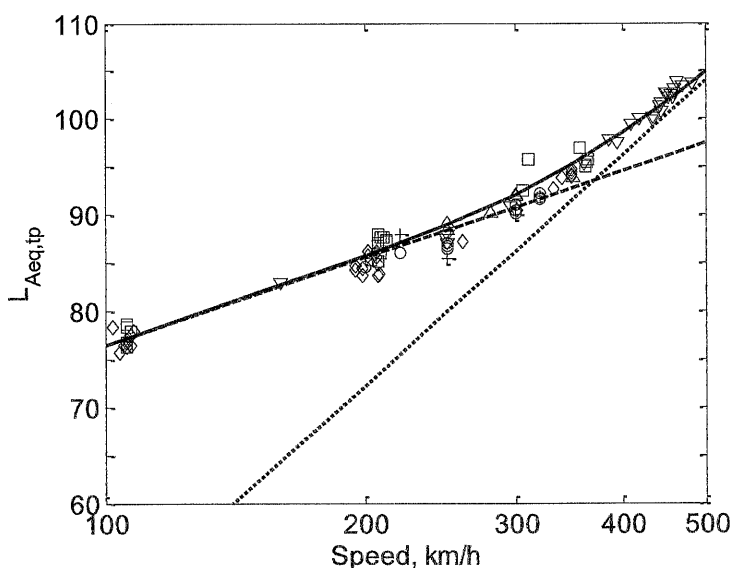


Figure 9. $L_{Aeq,tp}$ measured at 25 m from TGV-A (∇ , $\square$, $\diamond$), TGV-Duplex ($\circ$, Δ) and Thalys (+). - - -, line with slope of $30 \log_{10} V$ to represent rolling noise; line with slope of $80 \log_{10} V$ to represent aerodynamic noise; — sum of these two components.¹

The main sources of aerodynamic noise include the front of the train, the bogie regions, the inter-coach gaps and the pantographs. The pantograph only makes a small contribution to the overall pass-by noise level but becomes more important in the presence of noise barriers, as elevated sources such as the pantograph may not be shielded. Research into these sources follows two main routes: experimental, including wind tunnel testing, and computational.

3.1 Numerical approach

Numerical modelling techniques for predicting aerodynamic noise have developed rapidly in recent years but they still require considerable computational resources. There is a range of different computational fluid dynamics (CFD) techniques available, which resolve differing turbulence length scales. The number of cells, and hence also the computational effort, increase dramatically as the model complexity and turbulence resolution are increased.

Most aeroacoustic problems relating to trains involve dipole-type sources where flow over solid objects leads to fluctuating drag and lift forces. In order to resolve the flow sufficiently to enable prediction of the acoustic radiation, it is necessary to use a method that gives good resolution close to the boundaries. One such approach is Detached Eddy Simulation (DES), involving a combination of Large Eddy Simulation (LES) far from the body and Random-Averaged Navier Stokes (RANS) in the boundary layer.³⁵ Nevertheless, such an approach involves compromises in modelling the flow and results must be checked very carefully.

Although the small scale fluctuations associated with acoustic radiation could in principle be predicted directly using CFD, for example using Direct Numerical Simulation (DNS), in practice it is preferable and more practical to predict only large scale turbulence directly and use an approximate model for the smaller scales. An acoustic analogy is then used to predict the radiated sound in a subsequent post-processing step. Such a hybrid approach commonly uses the Ffowcs Williams-Hawkings acoustic analogy, which is based on a surface integral over a control surface.³⁶

This technique has been used to study the noise from cylinders that form the main components in a pantograph.³⁷ The main struts are inclined at an angle to the flow and it has been shown by computation that this leads to a significant reduction in the noise level, as shown in Figure 10(a). Moreover, for flow speeds in the critical regime, i.e. for Reynolds numbers above about 10^5 , the noise level is significantly reduced as shown in Figure 10(b).

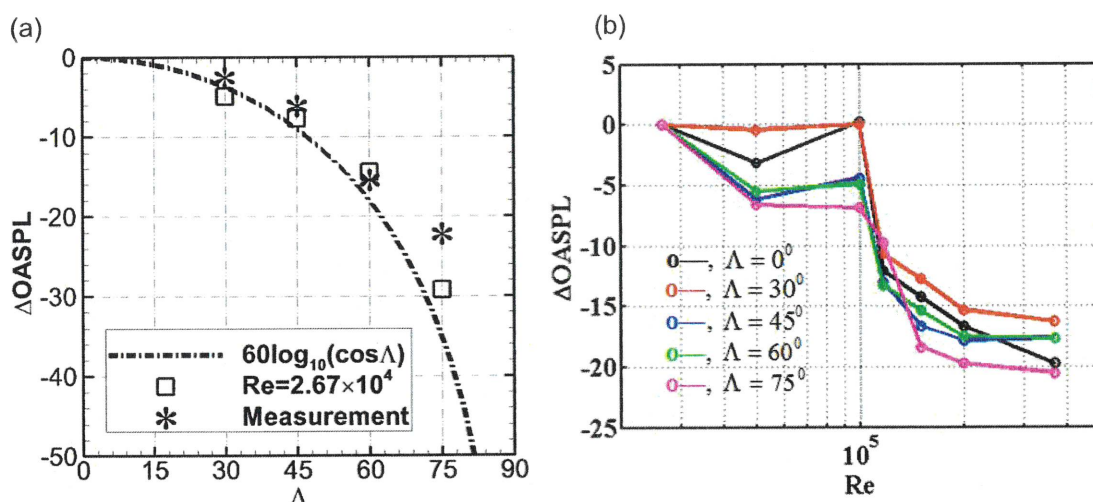


Figure 10. (a) Change in overall sound pressure level from a yawed cylinder as a function of yaw angle. (b) Change in overall sound pressure level of yawed cylinder as a function of normalized flow speed (Reynolds number) after removing $60 \log_{10} U$ trend.³⁷

3.2 Component-based approach

An alternative method of dealing with aeroacoustic problems is a semi-empirical component-based approach. This was previously developed for studying the noise from aircraft landing gear. Various cylindrical components are represented by idealised spectra, which are derived from wind tunnel tests³⁸ and/or CFD predictions. This provides a rapid calculation, allowing parametric studies to be performed easily. However, it relies on a correct model of the component noise, and it is difficult to account for interaction effects.

Applying this approach to the pantograph, a model has been produced based on a set of cylinders, including both tonal and broadband components.³⁹ Figure 11 shows an example comparison of predictions from this model with published measurements⁴⁰ on a full scale pantograph in a wind tunnel.

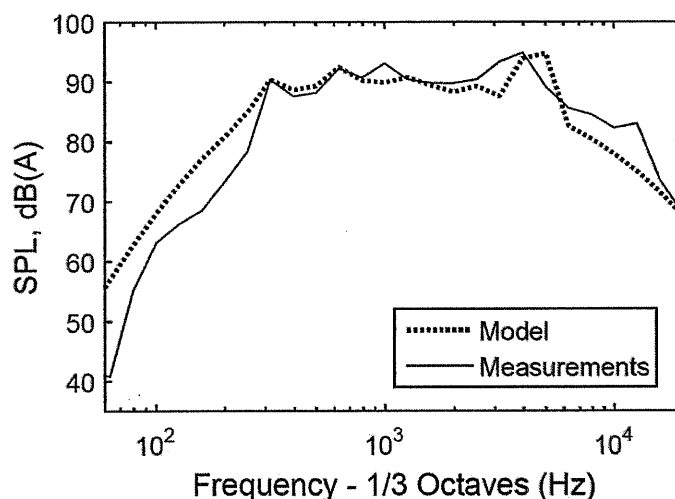


Figure 11. Sound pressure level from pantograph at 5 m for flow speed 97 m/s.³⁹

3.3 Experimental approaches

For the bogie region, the geometry of components and the flow field are more complex, so a simple component-based approach is not so readily applied. Wind tunnel tests on scale models⁴¹ have shown that the most important noise sources are associated with those components that project into the flow, such as the dampers and side-frames of the bogie. In contrast, components located beneath the train produce much less noise, as the incident flow speed is lower. Figure 12 shows example results from these tests showing the effect of extending the bogie side frame laterally into the flow.⁴¹

Field measurements can also be used to study aerodynamic noise. Microphone arrays, such as the one shown in Figure 13, can be used with beamforming algorithms to identify aerodynamic noise sources on moving trains. However, it is not straightforward to quantify their sound power using a microphone array as the beamforming algorithms are based on certain assumptions about the sound sources which are not generally upheld. In particular, the sources are assumed to be a distribution of uncorrelated monopoles, whereas the actual source directivity differs from this.

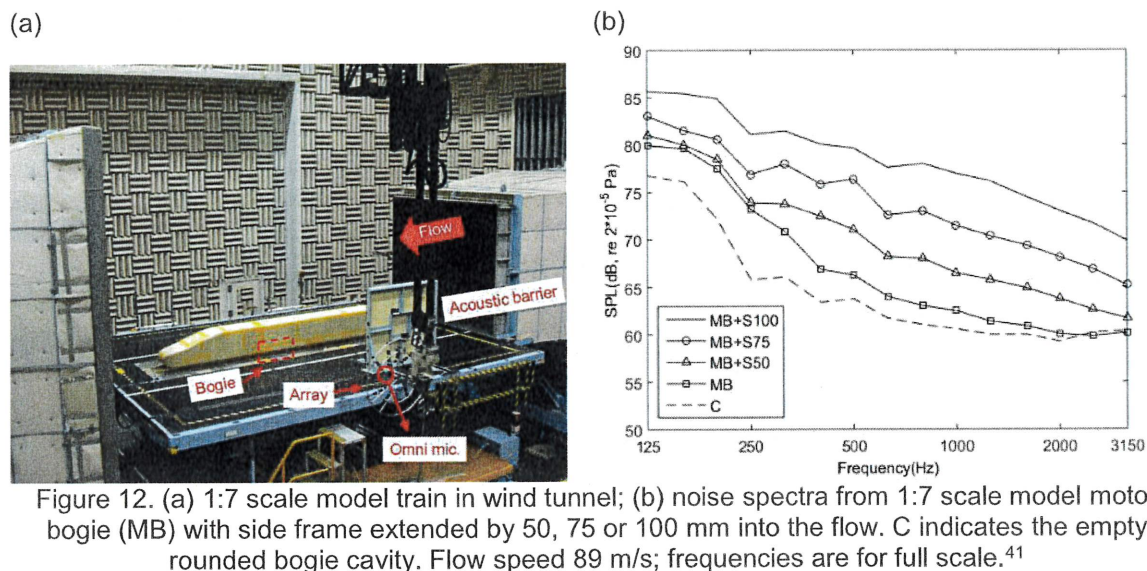


Figure 13. Microphone array with 4 m diameter in use to separate noise sources on a moving train.

4 GROUND VIBRATION AND GROUND-BORNE NOISE

Ground vibration, and the associated problem of ground-borne noise, are increasing in importance as train speeds increase and as new underground lines are built in urban environments. The opening of the 'Night Tube' in London, with 24-hour operation on Fridays and Saturdays, has highlighted this issue. A particular problem with prediction models is that the ground through which the vibration is transmitted can differ greatly from one location to another. At a given site the soil properties also vary with depth. No matter how complex the prediction models are, unless they are fed with the correct input parameters, the results will not be correct. It is often considered that models are unlikely to be able to predict vibration levels more reliably than ± 10 dB.⁴² Such uncertainty leads to the application of large safety margins in the design of mitigation measures.

Apart from rare cases of trains moving at or near the wave speeds in the ground, the main excitation mechanism is due to roughness, based on similar principles as for rolling noise, although the corresponding wavelengths are longer. A range of different modelling approaches, both semi-analytical and fully numerical, have been used to describe ground vibration from trains; for a detailed review see ⁴³.

4.1 Semi-analytical models

For a homogeneous track on a halfspace ground, semi-analytical models have been developed in which the ground response is solved in the wavenumber domain. These can also be extended to a ground consisting of parallel soil layers⁴⁴, see Figure 14. In such a model, known as MOTIV (Modelling of Train-induced Vibration), the movement of the train is allowed for such that the roughness at one wavelength will cause a response at the corresponding frequency in the moving frame of reference but for a fixed point on the ground it will cause a response at a range of frequencies due to the Doppler effect in the ground.

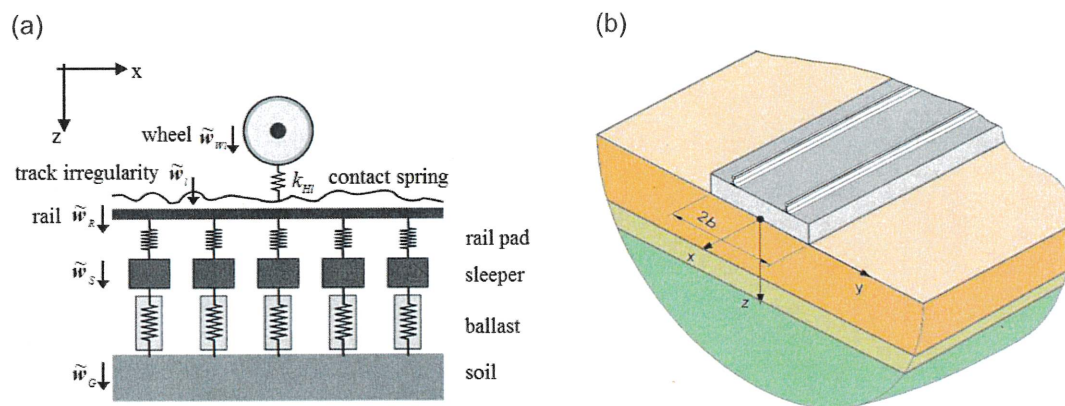


Figure 14. (a) Excitation model due to roughness and (b) the geometry of the coupled track-soil system.⁴⁵

A simpler model treats the vehicle as stationary and allows the roughness to be 'pulled through' the contact zone between wheel and rail.⁴⁵ A further simplification involves neglecting the interaction between different wheels, by treating them as uncorrelated sources of vibration. Comparing these models, Figure 15 shows the dynamic component of rail vibration (i.e. neglecting the effect of the moving load) for a track on a homogeneous ground. In narrow-band frequency resolution there are significant differences caused by the motion of the load, but in one-third octave bands the differences are more modest.

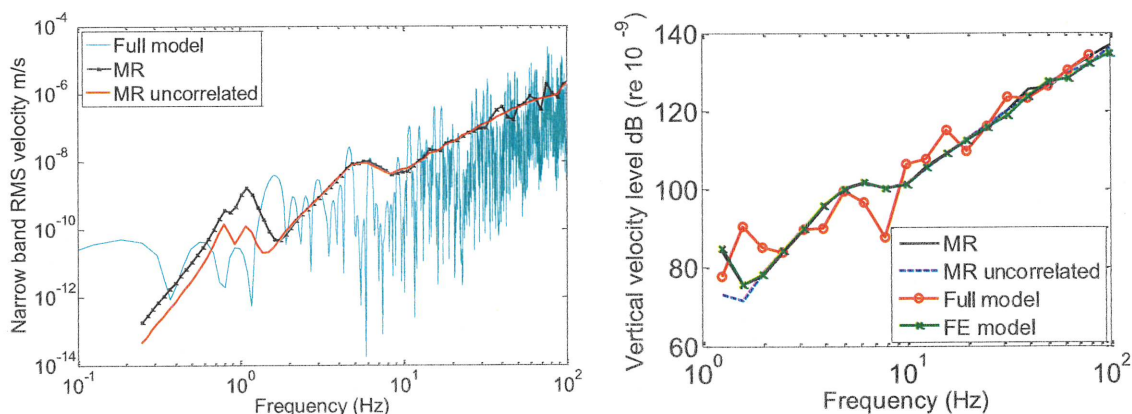


Figure 15. Dynamic component of rail velocity during passage of a train at 150 km/h. (a) Narrow band, (b) 1/3 octave band. Comparison of moving roughness (MR), moving roughness neglecting correlation between wheels, full analytical model and 2.5D FE model.⁴⁵

The moving constant load also induces vibration of the track, as the rail moves up and down under each axle. However, this does not propagate into the far field unless the train speed reaches the

wave speed in the ground. This quasi-static component can be predicted using the semi-analytical model including load motion⁴⁴, but not using the 'moving roughness' model.

4.2 Wavenumber finite element/boundary element models

An alternative approach to modelling the ground is to use numerical methods such as the finite element (FE) and boundary element (BE) methods. This has the advantage that arbitrary geometry can be considered, but the models become considerably more complex. For problems where the geometry is homogeneous in the axial direction, a wavenumber FE/BE approach can be used in which the FE/BE mesh is two dimensional and the third direction is represented by a wavenumber spectrum.⁴⁶ This so-called 2.5D approach is much more efficient than a full 3D approach, while avoiding the simplifications present in a purely 2D model. A similar approach has been used for studying rail vibration and noise.¹⁸

A model of a track on a half-space in 2.5D is shown in Figure 16. The rails are beams while the pads, sleepers and ballast are modelled using solid elements. An example result from this approach is included in Figure 15(b) where it can be seen that agreement with the analytical result is excellent for this case.

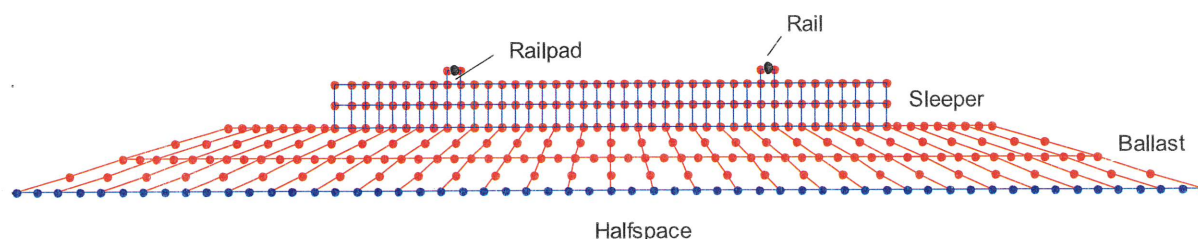


Figure 16. FE model of track included in a wavenumber FE/BE model of track/ground system.⁴⁶

The 2.5D FE/BE approach has also been used to study various mitigation measures for ground vibration, including trenches, soft barriers and ground stiffening.^{47,48} Experimental validation of the 2.5D approach has recently been carried out for a tunnel case and good agreement was found between measurements and predictions.⁴⁹

5 CURVE SQUEAL

Curve squeal is an intense tonal noise produced when a rail vehicle traverses a sharp curve. Unlike the other sources discussed here, the noise level can vary considerably in an apparently random manner from one day to another or one train to another. The excitation mechanism is associated with frictional instability and generally leads to vibration in one of the 0-nodal-circle axial wheel modes (see Figure 3).⁵⁰

A particularly serious and unusual curve squeal problem occurred in the mid 1980s when new Class 150 Sprinter diesel multiple unit (DMU) trains were introduced on some curving routes, in particular the Llandudno to Blaenau Ffestiniog line in North Wales. The squeal noise generated was apparently audible at distances up to 12 km, away while levels of 130-145 dB were measured close to the wheels. The tones generated corresponded to radial modes of the wheel excited by contact between the back of the wheel flange and the check rails. The problem is thought to have occurred due to a combination of a stiffer yaw suspension introduced for stability, a wheel shape giving modes which were easily excited by the check rail and the use of monobloc wheels with low damping. The problem was solved by fitting a constrained layer damping treatment to the wheels, as shown in Figure 17. This was eventually fitted to the whole Class 15x DMU vehicle fleet. It is effective in eliminating curve squeal, although with hindsight it could have been better optimised to reduce rolling noise as well.

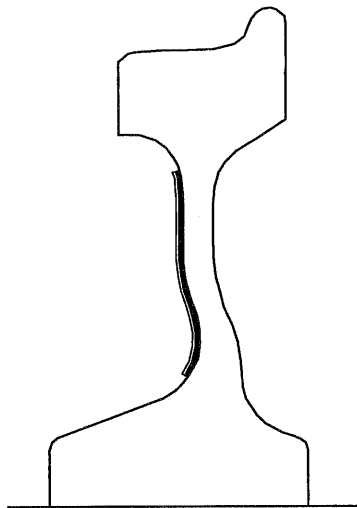


Figure 17. Constrained layer damping fitted to Class 150 DMU wheel.¹

More recent work has focused on the development of models for squeal noise.⁵⁰ A particular issue concerns the nature of the frictional instability. The conventional view was that a reduction of the friction force with increasing sliding velocity introduces negative damping into the system.⁵¹ However, other authors have focused recently on a geometrical coupling phenomenon. In reality, as can be shown by using a relatively simple model, the excitation is probably associated with both.⁵²

6 CONCLUDING REMARKS

Although railway operations will never be silent, it is clearly important to reduce their noise and vibration impact as much as possible, while not adding unnecessary cost or complication. To achieve this it is essential to understand the sources of noise and vibration and the parameters that can be used to influence them in order to propose cost-effective mitigation measures. Theoretical models are an important part of this process. It is important that such models should be of sufficient detail to cover the most relevant parameters in a reliable way, but not over-complicated in order not to lose insight. As the potential for developing ever more complex computational models increases, the use of simple models should not be overlooked. Nor should the underlying purpose of this work be forgotten, which is to give the railways and their neighbours a quieter future.

7 ACKNOWLEDGEMENTS

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