14 Noise and Vibration from Railway Vehicles

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14.1 INTRODUCTION

14.1.1 IMPORTANCE OF NOISE AND VIBRATION

Environmental noise is an issue that has seen increased awareness in recent years. Noise is often cited as a major factor contributing to people’s dissatisfaction with their environment. While this noise exposure is usually due mainly to road traffic, trains also contribute significantly in the vicinity of railway lines. Road vehicles and aircraft have long been the subject of legislation that limits their noise emissions. The European Union has therefore introduced noise limits for new rail vehicles, implemented as part of the Technical Specifications for Interoperability (TSIs) [1]. They state noise limits for new trains under both static and running conditions.

In contrast with exterior noise, the noise inside a vehicle (road or rail) is not generally the subject of legislation, apart from the noise inside the driver’s cab. For road vehicles, noise is actually used as a major factor to distinguish vehicles from their competitors and to attract people to buy a particular vehicle. As rail vehicles are for mass use, interior noise is subject instead to specifications from the purchasing organisation. These are usually limited to ensuring that problems are eliminated and that the vehicles are fit for their purpose.
Railway operations also generate vibration that is transmitted through the ground into neighbouring properties. This can lead either to feelable vibration (in the range 2–80 Hz) or to low-frequency rumbling noise (20–250 Hz). Vibration is also transmitted into the vehicle itself, affecting passenger comfort.

### 14.1.2 Basics of Acoustics

The field of acoustics is too large to cover in detail here. This chapter therefore gives only a very brief overview of some basic quantities. For further details, the interested reader is referred to textbooks on the subject [2,3].

Sound consists of audible fluctuations in pressure, usually of the air. These propagate as waves with a wave speed, denoted by $c_0$, of about 340 m/s in air at 20°C. Simultaneously, fluctuations in air density and particle motion also occur. To express the magnitude of a sound, the root mean square (rms) sound pressure is usually used:

$$p_{\text{rms}} = \left( \frac{1}{T} \int_{T} p(t)^2 \, dt \right)^{1/2}$$

(14.1)

where $p(t)$ is the instantaneous sound pressure at time $t$, and $T$ is the averaging time. Much use is made of frequency analysis, whereby sound signals are decomposed into their frequency content (e.g., using Fourier analysis). The normal ear is sensitive to sound in the frequency range 20–20,000 Hz (the upper limit reduces with age and with noise exposure) and to a large range of amplitudes (around six orders of magnitude). Because of these large ranges, and to mimic the way the ear responds to sound, logarithmic scales are generally used to present acoustic data. Thus, amplitudes are expressed in decibels. The sound pressure level is defined as

$$L_p = 10 \log_{10} \left( \frac{p_{\text{rms}}^2}{p_{\text{ref}}^2} \right) = 20 \log_{10} \left( \frac{p_{\text{rms}}}{p_{\text{ref}}} \right)$$

(14.2)

where the standard value for the reference pressure $p_{\text{ref}}$ is $2 \times 10^{-5}$ Pa. Frequencies (expressed in cycles/sec or Hz) are also generally plotted on logarithmic scales, with one-third octave bands being a common form of presentation. The frequency range is divided into bands that are of equal width on a logarithmic scale. The centre frequencies of each band can be given by $10^{N/10}$, where $N$ is an integer, the band number, although by convention, these frequencies are rounded to standard values. Bands 13–43 cover the audible range.

The total sound emitted by a source is given by its power, $W$, which in decibel form is given as the sound power level:

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

(14.3)

where the standard value for the reference power, $W_{\text{ref}}$, is $10^{-12}$ watt. The power is generally proportional to the square of the sound pressure, so that a 1 dB increase in sound power level leads to a 1 dB increase in sound pressure level at a given location. However, sound pressure also depends on the location, usually reducing as the distance between the source and the receiver increases. For a compact point source, this reduction is 6 dB per doubling of distance, while for a line source, it is 3 dB per doubling. Other quantities can also be expressed in decibels following the pattern of Equations 14.2 and 14.3.
It should be realised that sound generation is often a very inefficient process. The proportion of the mechanical power of a typical machine that is converted into sound is typically in the range $10^{-7}$ to $10^{-5}$. Sound is generated by a variety of mechanisms, but the two main types are:

- **Structural vibrations**: The vibration of a structure causes the air around it to vibrate and transmit sound, for example, a drum, a loudspeaker, wheels and rails.
- **Aerodynamic fluctuations**: Wind, particularly turbulence and flow over solid objects, also produces sound, for example, jet noise, turbulent boundary layer noise, exhaust noise and fan noise.

While the acceptability of sound levels and signal content varies greatly between individuals, it is important to include some approximation to the way the ear weights different sounds. Several weighting curves have been devised, but the A-weighting (Figure 14.1) is the most commonly used. This approximates the inverse of the equal loudness curve at about 40 dB. As the ear is most sensitive around 1–5 kHz, and much less sensitive at low and high frequencies, more prominence is given to this central part of the spectrum. The overall sound level is often quoted as an A-weighted value, meaning that this weighting curve is applied to the spectrum before calculating the total or applied as a filter to the time-domain signal.

Another overall measure of the magnitude of a sound is the **loudness**. Strictly, this is a subjective quantity, but there are ways of estimating a loudness value from a one-third octave band spectrum [4]. However, this is less commonly used than the A-weighted decibel. It should be borne in mind that an increase of 10 dB is perceived as a ‘doubling of loudness’, while a change of less than 3 dB is normally imperceptible.

### 14.1.3 Sources of Railway Noise and Vibration

In the case of railway noise, both types of mechanism mentioned previously apply. Aerodynamic noise is important for high-speed operation and is generated by unsteady air flow, particularly over the nose, inter-carriage joints, bogie regions, louvres and roof-mounted equipment such as
pantographs; this is described in Section 14.6. However, mechanical sources of noise dominate the overall noise for speeds up to about 300 km/h.

The most important mechanical noise source from a train is generated at the wheel-rail contact. Rolling noise is caused by vibration of the wheel and track structures, induced at the wheel-rail contact point by vertical irregularities in the wheel and rail surfaces; this is described in Sections 14.2 and 14.3. A similar mechanism leads to noise due to discontinuities in the wheel or rail surface (impact noise); see Section 14.4. Squeal noise occurs in sharp curves and is induced by unsteady friction forces at the wheel-rail contact (Section 14.5). Noise from traction equipment and fans is covered briefly in Section 14.7. Finally, ground-borne vibration and noise are caused by track and wheel irregularities and by the movement of the set of axle loads along the track; see Section 14.9. Transmission of noise to the vehicle interior is discussed in Section 14.8 and ride comfort is discussed in Section 14.10. A fuller coverage of these topics can be found in [5].

14.2 ROLLING NOISE

14.2.1 MECHANISM OF ROLLING NOISE GENERATION

As indicated previously, rolling noise is usually the dominant source of noise from moving trains at speeds below about 300 km/h. The sound level increases with speed $V$ at a rate of approximately $30 \log_{10} V$, that is, a 9 dB increase for a doubling of speed. It can be attributed to components radiated by the vibration of both wheels and track. This vibration is caused by the combined surface 'roughness' at their interface, as illustrated in Figure 14.2.

The relative importance of the components of sound radiation from the wheel and track depends on their respective designs as well as on the train speed and the wavelength content of the surface roughness. In most cases, both sources (wheel and track) are significant. As the noise radiation depends on the roughness of both the wheel and track, it is possible that a rough wheel causes a

high noise level that is mainly radiated by the track vibration or vice versa. It is therefore difficult to assign noise contributions solely to the vehicle or infrastructure.

### 14.2.2 Surface Roughness

Irregularities with wavelengths between about 5 mm and 500 mm cause the vibration of relevance to noise. When a wavelength $\lambda$ (in m) is traversed at a speed $V$ (in m/s) the associated frequency generated (in Hz) is given by

$$f = \frac{V}{\lambda} \quad (14.4)$$

The corresponding amplitudes range from over 100 $\mu$m at long wavelengths to much less than 1 $\mu$m at short wavelengths. Examples of wheel roughness spectra are shown in Figure 14.3 [6]. These are given in decibels relative to 1 $\mu$m (using a definition equivalent to Equation 14.2):

$$L_r = 10 \log_{10} \left( \frac{r_{rms}}{r_{ref}} \right) = 20 \log_{10} \left( \frac{r_{rms}}{r_{ref}} \right) \quad (14.5)$$

where $r$ is the roughness amplitude and expressed in one-third octave bands over wavelength. Each line represents the average of between 50 and 100 wheels.

In the TSI Noise [1] and ISO 3095 [7], a limit curve is included for the roughness of a test track. For the characterisation of new vehicles, the rail roughness should be less than this limit curve, which is also shown in Figure 14.3. This represents good-quality track. The purpose of this in the context of [1,7] is to ensure that variations in rail roughness from one site to another do not significantly affect the measurement. The measurement method for rail roughness is defined in [8].

The wheel-rail contact does not occur at a point but over a small area; see Chapter 7. The contact patch is typically 10- to 15-mm long and of a similar width. When roughness wavelengths are short compared with the contact patch length, their effect on the wheel-rail system is attenuated.

![Figure 14.3](image-url)  
**Figure 14.3** Typical wheel roughness spectra (data from [6]); also shown is the maximum rail roughness allowed for vehicle noise measurements according to TSI Noise [1] and ISO 3095 [7].
This effect is known as the contact filter. This is significant from about 1–1.5 kHz upwards for a speed of 160 km/h and at lower frequencies for lower speeds.

In early analytical models for this effect [9], the extent of correlation of the roughness across the width of the contact had to be assumed, since very detailed roughness data were not available. The following approximate formula can be used [5]:

$$|H(k)|^2 = \left(1 + \frac{\pi}{4} (ka)^3\right)^{-1}$$  \hspace{1cm} (14.6)

where $k = 2\pi/\lambda$ is the roughness wavenumber in the longitudinal direction, $\lambda$ is the wavelength and the length of the contact patch in the rolling direction is $2a$. Later, Remington developed a numerical discrete point reacting spring (DPRS) model [10]. This model is intended to be used with roughness measurements obtained on multiple parallel lines a few millimetres apart. Figure 14.4 shows the average results obtained using a series of such measurements in combination with the DPRS model [5]. This is compared with the results from Equation 14.6, which confirms the validity of the analytical model at low frequencies but indicates that the filtering effect is less severe at high frequencies than what the analytical model suggests.

### 14.2.3 Wheel Dynamics

A railway wheel is a lightly damped, resonant structure, which when struck rings like a bell, a structure that it strongly resembles. As with any structure, the frequencies at which it vibrates freely are called its natural frequencies, and the associated vibration pattern is called the mode shape.

Wheels are usually axisymmetric (although the web is sometimes not). Their normal modes of vibration can therefore be described in terms of the number of diametral node lines – lines at which the vibration pattern has a zero. A flat disc, to which a wheel can be approximated, has out-of-plane modes that can be described by the number of nodal diameters, $n$, and the number of nodal circles, $m$. A flat disc also has in-plane radial modes and circumferential modes, each with $n$ nodal diameters. In-plane modes with nodal circles occur for railway wheels above 6 kHz.

![FIGURE 14.4 Contact filter effect from numerical DPRS model (average result from six wheels) and analytical result from Equation 14.6.](image-url)
A railway wheel differs from a flat disc, having a thick tyre region at the perimeter and a thick hub at the centre connecting the wheel to the axle. A railway wheel is also not symmetric about a plane perpendicular to its axis. The tyre region is asymmetric due to the flange, and the web is usually also asymmetric, at least on wheels designed for tread braking, with the curved web being designed to allow for thermal expansion. An important consequence of this asymmetry is that radial and out-of-plane (axial) modes are coupled.

The finite element method (FEM) can be used quite effectively to calculate the natural frequencies and mode shapes of a railway wheel. Figure 14.5 shows an example of results for an Union International Union of Railways (UIC) 920-mm-diameter freight wheel [11]. The cross-section

![Modes of vibration and natural frequencies (in Hz) of UIC 920-mm freight wheel calculated using finite elements. (From Thompson, D.J., and Jones, C.J.C., J. Sound Vib., 231, 519–536, 2000. With permission.)](image-url)
through the wheel is shown, along with an exaggerated form of the deformed shape in each mode of vibration. Each column contains modes of a particular number of nodal diameters, \( n \). The first row contains axial modes with no nodal circle. These have their largest out-of-plane vibration at the running surface of the wheel. These modes are usually excited in curve squeal (see Section 14.5) but are not excited significantly in rolling noise. The second and third rows contain radial modes and one-nodal-circle axial modes. Owing to the asymmetry of the wheel cross-section, and their proximity in frequency, these two sets of modes are strongly coupled; that is, they both contain axial and radial motion. It is these modes that are most strongly excited by roughness during rolling on a straight track, due to their radial component at the wheel-rail contact point.

The modes shown in Figure 14.5 are those of the wheel alone, constrained rigidly at the inner edge of its hub. The first column of modes, \( n = 0 \), is in practice coupled to extensional motion in the axle, and the second set, \( n = 1 \), is coupled to bending motion in the axle. As a result of this coupling with the axle, which is constrained by the roller bearings within the axleboxes, these sets of modes experience greater damping than the modes with \( n \geq 2 \). The latter do not involve deformation of the axle and are therefore damped only by material losses; their modal damping ratios are typically about \( 10^{-4} \) [5].

In order to couple the wheel to the track in a theoretical model, the frequency response functions of the wheel at the interface point are required. These may be expressed in terms of receptance, the vibration displacement due to a unit force as a function of frequency. Alternatively, mobility, the velocity divided by force, or accelerance, the acceleration divided by force, can be used. Such frequency response functions of a structure can be constructed from a modal summation. For each mode, the natural frequency \( f_{mn} \) is written as a circular frequency \( \omega_{mn} = 2\pi f_{mn} \). Then, the response at circular frequency \( \omega \), in the form of a receptance \( \alpha_{jk} \), is

\[
\alpha_{jk} = \sum_{n,m} \frac{\psi_{mnj} \psi_{mkn}}{m_m (\omega^{2} - \omega_{mn}^{2} + 2i\zeta_{mn} \omega \omega_{mn})} \tag{14.7}
\]

where:
- \( \psi_{mnj} \) is the mode shape amplitude of mode \( m, n \) at the response position
- \( \psi_{mkn} \) is the mode shape amplitude of mode \( m, n \) at the force position
- \( m_m \) is the modal mass of mode \( m, n \), a normalisation factor for the mode shape amplitude
- \( \zeta_{mn} \) is the modal damping ratio of mode \( m, n \)
- \( i \) is the square root of \(-1\)

Figure 14.6 shows the radial point mobility of a wheelset calculated using the normal modes from an Finite Element Method (FEM), as shown in Figure 14.5. This is based on Equation 14.7 multiplied by \( i \omega \) to convert from receptance to mobility. At low frequencies, the magnitude of the mobility is inversely proportional to frequency, corresponding to mass-like behaviour. Around 500 Hz, an anti-resonance trough appears, and above this frequency, the curve rises in stiffness-like behaviour until a series of sharp resonance peaks are reached at around 2 kHz. These peaks correspond to the axial one-nodal-circle and radial sets of modes identified in Figure 14.5.

### 14.2.4 Track Dynamics

The dynamic behaviour of track is described in detail in Chapter 9. A typical track mobility is also shown in Figure 14.6. This is predicted using a model based on a continuously supported rail, which neglects the effects of the periodic support. A broad peak at around 100 Hz corresponds to the whole track vibrating on the ballast. At the second peak, at about 500 Hz, the rail vibrates on the rail pads. The frequency of this peak depends on the rail pad stiffness. Above this frequency, bending waves propagate in the rail and can be transmitted over quite large distances.

The degree to which these waves are attenuated, mainly due to the damping effect of the pads and fasteners, affects the noise radiation from the rail. The method of measuring these track
decay rates is standardised in [12]. Figure 14.7 shows measured decay rates of vertical vibration for three different rail pads installed in the same track. The results for the middle value of pad stiffness correspond to the mobility in Figure 14.6. The vertical bending waves are strongly attenuated in a region between 300 and 800 Hz that depends on the pad stiffness. This peak in the decay rate corresponds to the region between the two resonance peaks in Figure 14.6. Here, the sleeper mass vibrates between the pad and ballast springs and acts as a ‘dynamic absorber’ to attenuate the propagation of waves in the track. Also shown in Figure 14.7 is the limit curve from ISO 3095 [7]; it can be seen that only the result for the stiff pad exceeds this limit. The attenuation of lateral waves is generally smaller than that for the vertical direction, and a different limit applies.

FIGURE 14.6 Vertical mobilities of the wheel-rail system, showing —— radial mobility of UIC 920-mm freight wheel, —— vertical mobility of track with moderately soft pads and — — — contact spring mobility. (a) Magnitude (b) Phase. (From Thompson, D., Railway Noise and Vibration: Mechanisms, Modelling and Means of Control, Oxford, UK, 2009. With permission.)
14.2.5 Wheel-Rail Interaction

The wheel and rail are coupled dynamically at their point of contact. Between them, local elastic deflection occurs to form the contact patch, which can be represented as a contact spring. Although this spring is non-linear (see Chapter 6), for small dynamic deflections, it can be approximated by a linearised stiffness, $k_H$ [13]. This is shown as contact spring mobility ($= io/k_H$) in Figure 14.6.

The coupled wheel-rail system is excited by the roughness, which forms a relative displacement input; see Figure 14.8. Here, the forward motion of the wheel is ignored, and the system is replaced by one in which the wheel is static and the roughness is pulled between the wheel and rail (‘moving irregularity model’). Considering only coupling in the vertical direction, from equilibrium of forces and compatibility of displacements, the vibration amplitude of the wheel ($u_W$) and rail ($u_R$) at a particular frequency can be written as

$$
u_r = \frac{\alpha_W r}{\alpha_W + \alpha_R + \alpha_C}$$

$$u_R = -\frac{\alpha_R r}{\alpha_W + \alpha_R + \alpha_C}$$

(14.8)

where $r$ is the roughness amplitude and $\alpha_W$, $\alpha_R$ and $\alpha_C$ are the vertical receptances of the wheel, rail and contact spring, respectively. Clearly, where the rail receptance has a much larger magnitude than that of the wheel or contact spring, $u_R \approx -r$, that is, the rail is pushed down at the amplitude of the roughness. From Figure 14.6, this can be expected between about 100 Hz and 1000 Hz. Changing the rail receptance in this frequency region has little effect on the rail vibration at the contact point (although the changes may affect the decay rates).

In practice, coupling also exists in other directions as well as the vertical, notably the lateral, direction [14]. This modifies Equation 14.8 to yield a matrix equation, but the principle remains the same.
14.2.6 Noise Radiation

The vibrations of the wheel, rail and sleepers all produce noise. In general, the sound power $W_{rad}$ radiated by a vibrating surface of area $S$ can be expressed as [2]:

$$W_{rad} = \rho_0 c_0 S \sigma \langle \overline{v^2} \rangle$$  \hspace{1cm} (14.9)

where $\langle \overline{v^2} \rangle$ is the spatially averaged mean-square velocity normal to the vibrating surface, $\rho_0$ is the density of air, $c_0$ is the speed of sound in air and $\sigma$ is a frequency-dependent factor called the radiation efficiency. Thus, components radiate large amounts of noise if their vibration is large and/or their surface area is large and/or their radiation efficiency is high. The radiation efficiency is usually close to unity at higher frequencies and much smaller at low frequencies (where the radiating object is small compared with the wavelength of sound). Predictions of this factor can be obtained using numerical methods such as the boundary element method or, for simple cases, analytical models. Models for the wheel radiation are presented in [15] and for the rail radiation are presented in [16].

Figure 14.9 shows predictions of the noise from wheels, rails and sleepers during the passage of a pair of similar bogies. This is shown in the form of the average sound pressure level at a location close to the track (3 m from the nearest rail). The wheel is the most important source of noise at high frequencies, above about 1.6 kHz. From Figure 14.6, it can be seen that this corresponds to the region in which many resonances are excited in the radial direction. Between about 400 Hz and 1600 Hz, the rail is the dominant source of noise. Here, the rail vibrates at the amplitude of the roughness. The support structure affects the rate of decay with distance and hence the spatially averaged velocity. At low frequencies, the sleeper radiates the largest component of noise. Here, the rail and sleeper are well coupled and have similar vibration amplitudes, but the sleeper has a larger area and a radiation efficiency close to unity, whereas that of the rail reduces below 1 kHz.
Although the details of Figure 14.9 are specific to this combination of wheel and track design, train speed and roughness spectrum, it is generally the case that the most important source is formed by the sleepers at low frequencies, the rails in the mid frequencies and the wheels at high frequencies. As speed increases, the energy in the noise spectrum shifts towards higher frequencies, leading to a greater importance of the wheel in the overall level.

14.2.7 Overall Model

The complete model for rolling noise that has been described in Sections 14.2.1 to 14.2.6 has been implemented in a software package, Track-Wheel Interaction Noise Software (TWINS) [17], which is widely used in the railway industry. This is a frequency-domain model based on the moving irregularity formulation. It produces estimates of sound power and sound pressure spectra in one-third octave bands and allows the user to study the effect of different wheel and track designs on noise.

This model has also been the subject of extensive validation [18,19]. Comparisons between predictions and measurements for three track types, three wheel types and four speeds gave overall sound levels that agreed within about ±2 dB [18]. These predictions were updated in [19] along with new measurements for a range of novel constructions. Revisions to the software have improved agreement slightly. Agreement in one-third octave bands had a larger spread of around ±4 dB, but this was at least partly due to uncertainties in the measured roughness inputs.

14.3 Reducing Rolling Noise

From the theoretical understanding, it is clear that rolling noise can be reduced by:

- Controlling the surface roughness
- Minimising the vibration response of wheels and tracks by adding damping treatments, by shape optimisation of wheels or rails or by introducing vibration isolation
- Preventing sound radiation, for example, by using local shielding measures

In each case, attention must be given to the presence of multiple sources. If more than one source is important, overall reductions will be limited, unless all sources are controlled. For example, if there are initially two sources (wheel and track) that contribute equally and one of them is reduced by 10 dB without affecting the other, the overall reduction will be limited to 2.5 dB.
14.3.1 Controlling Surface Roughness

From the vehicle designer’s point of view, the main feature affecting the wheel roughness is the braking system. Traditional tread brakes, in which cast-iron brake blocks act on the wheel tread, lead to the development of high levels of roughness on the wheel running surfaces due to the formation of local hot spots. This can be seen from Figure 14.3, the greatest differences in roughness being at the peak at around 6-cm wavelength. This high roughness in turn leads to higher levels of rolling noise. With the introduction of disc-braked vehicles, for example, the Mk III coach in the UK in the mid-1970s, it became apparent that disc braking can lead to quieter rolling stock. The difference in rolling noise between the Mk III and its tread-braked predecessor, the Mk II, was about 10 dB, mainly due to the difference in roughness. Modern passenger rolling stock is mostly disc-braked for reasons of braking performance, and this brings with it lower noise levels than older stock.

However, environmental noise is usually dominated by freight traffic. Freight vehicles are generally noisier and often run at night when environmental noise limits are tighter. For freight traffic in Europe, a number of factors have meant that cast-iron brake blocks have remained the standard until recently. These include cost, the longevity of wagons (typically 50 years) and, most importantly, the UIC standards for international operation, which required the use of such brakes. However, since 1999, the UIC has been pursuing an initiative to replace cast-iron blocks with alternative materials [20]. The idea is to introduce blocks made of a composite material that do not produce hot spots and therefore leave the wheel relatively smooth. So-called K-blocks and some types of LL-blocks can give a noise reduction of typically 8−10 dB(A) compared with cast-iron blocks on a TSI-compliant track [21]. On track with higher roughness levels, the improvements are more modest.

Rail corrugation is also a source of increased noise. A corrugated track can be up to 20 dB noisier than a smooth one for disc-braked wheels. Grinding of the rail for acoustic purposes is carried out in, for example, Germany to maintain special low noise sections of track [22].

14.3.2 Wheel-Based Solutions

14.3.2.1 Wheel Damping

One means of reducing the amount of noise radiated by the wheels is to increase their damping. Impressive reductions in the reverberation of wheels can be achieved by simple damping measures. However, a wheel in rolling contact with the rail already has, in effect, considerably more damping than a free wheel, since vibration energy flows from the wheel into the track. To improve the rolling noise performance, the added damping must exceed this effective level of damping already present, which is one to two orders of magnitude higher than that of the free wheel.

Various devices have been developed to increase the damping of railway wheels by absorbing energy from their vibrations, thereby reducing the noise produced. Examples are shown schematically in Figure 14.10. These include multi-resonant absorbers (Figure 14.10a), which have been used in Germany since the early 1980s and are fitted to many trains, including the ICE-1. Noise reductions of 5–8 dB have been claimed for speeds of 200 km/h [23]. Another commercial form of damper involves multiple layers of overlapping plates known as the shark’s fin damper (Figure 14.10b). Färm [24] found reductions of 1–3 dB(A) overall, associated with wheel noise reductions of 3–5 dB(A). Constrained layer damping treatments (Figure 14.10c) consist of a thin layer of visco-elastic material applied to the wheel and backed by a thin stiff constraining layer (usually metal). Such a treatment was used on the Class 150 Diesel Multiple Unit (DMU) in the UK in the late 1980s and was applied to the whole vehicle fleet to combat a particularly severe curve squeal problem excited by contact between the wheel flange and the check rail. By careful design, sufficient damping can be achieved using constrained layer damping to make significant reductions in rolling noise as well [25,26].
14.3.2.2 Wheel Design

Reductions in the wheel component of radiated noise can also be achieved by careful attention to the wheel cross-sectional shape. In recent years, manufacturers have used theoretical models such as TWINS [17] to assist in designing wheels for low noise.

As an example of the difference that the cross-sectional shape can have, three wheels are shown in Figure 14.11. Wheel (a) is a German Intercity wheel, (b) is a UIC standard freight wheel (cf. Figures 14.5 and 14.6) and (c) was designed several years ago by the Technical University of Berlin on the basis of scale model testing [23]. Figure 14.11 also shows the predicted noise components from the wheel in each case. The track component of noise (not shown) is not affected by these changes and remains the dominant source up to 1 kHz.

These results show that a straight web (wheel (c)) is beneficial compared with a curved web (wheel (b)). This is because the radial and axial motions are decoupled for a straight web. However, it is not always possible to use straight webs if tread brakes are used, as the curve is included in the web to allow thermal expansion. Wheel (a) is particularly noisy, the main difference between this and wheel (c) being the transition between the inside of the tyre and the web and the web thickness. Increasing the web thickness and particularly the transition between the tyre and web are effective means of reducing noise but also lead to increased unsprung mass. Wheels with profiles similar to (a) have shown appreciable rolling noise reductions by the addition of absorbers, whereas wheels such as (b) have shown much smaller reductions.

Another aspect of wheel design that can be used to reduce noise is the diameter. Smaller wheels have higher natural frequencies, so it is possible by reducing the diameter to move many of the resonances out of the range of excitation (i.e., above about 5 kHz) [27]. The upper frequency is somewhat increased for a smaller wheel due to a shift in the contact patch filter, but this effect is much less significant than the shift in natural frequencies. The trend in recent years towards smaller wheels for other reasons is therefore advantageous for noise. This also reduces the unsprung mass. However, if the wheel size is reduced too much, the track noise will increase due to the reduction in the contact filter effect.

14.3.3 Track-Based Solutions

14.3.3.1 Low-Noise Track

To achieve significant reductions in overall noise, it is usually not sufficient to deal only with the wheel noise. There must be a corresponding reduction in noise from the track vibration. Two very
important parameters of the track that affect its noise emission and that are related to one another are the stiffness of the rail pad and the decay rate of vibration along the rail. A stiff rail pad causes the rail and sleeper to be coupled together over a wide frequency range. Conversely, a soft pad isolates the sleeper for frequencies above a certain threshold. The lower the stiffness of the rail pad, the lower this threshold frequency. Soft rail pads therefore effectively isolate the sleepers and the foundation from the vibration of the rail, reducing the component of noise radiated by vibration of the sleepers. Part of the designed role of the rail pad is to protect the sleeper and ballast from high-impact forces. For this reason, softer rail pads have become more commonplace in recent years. Unfortunately, softer rail pads also cause the vibration of the rail to propagate with less attenuation (see Figure 14.7). As a greater length of rail vibrates with each wheel, this means more noise is generated by the rail, as shown in Figure 14.12. There is thus a compromise to be sought between the isolating and attenuating properties of the rail pad [28].

The EU-funded research project Silent Track successfully developed and demonstrated low-noise technology for the track [5]. The most successful element was a rail damper. Multiple blocks of steel are fixed to the sides of the rail by an elastomer and tuned to give a high damping effect in the region of 1 kHz. This allows a soft rail pad to be used, to give isolation of the sleepers, whilst minimising the propagation of vibration along the rail [29]. Figure 14.13 shows the noise reduction achieved in the field tests. In this case, a low-noise wheel was used for comparison to minimise the effect of the wheel on the total noise, but, even so, some wheel noise was present at high frequencies. The overall reduction in track noise was approximately 6 dB. Various rail dampers have been developed over the last 20 years. A standardised test method was therefore developed to allow their performance to be evaluated on the basis of laboratory tests on a short length of rail [30].

![Figure 14.11](image_url)
Tests with an ‘optimum’ pad stiffness have been less successful. Although the effect of pad stiffness has been clearly demonstrated in field tests, the optimum for noise radiation is too stiff to be acceptable for other reasons, particularly track damage protection. Stiff pads are also believed to lead to a higher likelihood of corrugation growth, which, in the long term, has a negative effect on the noise. The analysis of the acoustic performance of pads with different stiffnesses is further complicated by their load-dependent characteristics and other factors such as temperature variation [31].

### 14.3.3.2 Slab Tracks

Tracks mounted on concrete slabs have become more commonplace in the last few years, notably on high-speed lines. Such tracks are generally found to be noisier than conventional ballasted track, typically by around 3 dB. This can be attributed to two features of such tracks. Firstly, they tend to be fitted with softer rail fasteners in order to introduce the resilience normally given by the ballast; this leads to lower track vibration decay rates. Secondly, they have a hard sound-reflecting surface, whereas ballast has an absorptive effect. Although the latter only affects the overall noise by around 1 dB, it has been shown recently that the difference in ground absorption below the rail also affects the radiated sound power from the rail [32].


**FIGURE 14.13** Measured noise reduction from Silent Track rail damper during the passage of a low-noise wheel at 100 km/h. (From Thompson, Jones, Waters and Farrington, A tuned damping device for reducing noise from railway track, Applied Acoustics, 68(1), 43−57, 2007. With permission.)
A number of mitigation measures have been introduced, in which absorbent material is added to the upper surface of the slab. This has the effect of reducing the reflections of sound from the slab surface. Where it is also possible to introduce some shielding of the rail noise, for example, by an integrated mini-barrier, additional attenuation is possible. Such treatments have been found to reduce noise levels from slab track back to those of ballasted track.

For street running trams, a number of embedded rail systems are used. At first sight, an embedded rail might be expected to be silent, as the rail is mostly hidden and therefore should not produce sound. In practice, the rail head is visible, and both the rail head and the embedding material around it vibrate and produce sound [33]. Embedded rail systems offer the possibility of higher rail attenuation rates, due to the damping effect of the embedding material around the rail. They can also be constructed with relatively soft supports and therefore offer the potential to produce good vibration isolation.

### 14.3.4 Local Shielding and Barriers

Conventional noise barriers at the trackside are used widely in many countries. Reductions of 10–20 dB are achievable, depending on the height of the barriers, but they are expensive and visually intrusive, especially if taller than about 2 m. Cost-benefit studies have shown that noise reduction at source can be cost-effective compared with barriers or, in combination, can allow the use of lower barriers for the same overall effect [34].

The efficiency of a barrier is improved by placing it as close as possible to the source. In the Silent Freight project, it was demonstrated that, at least for certain types of wheel, a shield mounted on the wheel covering the web can reduce the noise. A more general solution is to place an enclosure around the bogie. If used in combination with low barriers very close to the rail, reductions of up to 10 dB can be achieved [35].

Bogie shrouds and low barriers were also tested in the Silent Freight and Silent Track projects [5], but in this case, the objective was to find a combination that satisfied international gauging constraints. Unfortunately, this meant that the overall reduction was limited to less than 3 dB, owing to the inevitable gap between the top of the barrier and the bottom of the shroud [36]. There are many other practical difficulties in enclosing the bogies, such as ventilation for the brakes and access for maintenance. Nevertheless, such vehicle-mounted screens are common on trams.

### 14.4 Impact Noise and Vibration

#### 14.4.1 Introduction

In the previous sections, noise due to random irregularities on the railhead and wheel tread has been considered. In addition to this, larger discrete features occur on the running surfaces such as rail joints, gaps at points and crossings, dipped welds and wheel flats. These cause high interaction forces and, consequently, noise. In some cases, loss of contact can occur between the wheel and rail, followed by large impact forces. Noise from such discrete features is often referred to as impact noise. Whereas rolling noise can be predicted using a linearised contact spring, in order to predict impact forces and noise, the non-linear contact stiffness must be included (see Chapter 6).

Early models for impact noise were essentially empirical [37]. To predict impact forces, time-domain models incorporating the non-linearities in the contact zone have been used, for example, by Clark et al. [38] and Nielsen and Igeland [39]. These models contain large numbers of degrees of freedom to represent the track. Nevertheless, they are limited to a maximum frequency of around 1500 Hz. In order to model impact noise up to around 5 kHz, simplified models of the wheel and rail have been used in a time-stepping model in order to determine the effects of the non-linearities [40]. These are then used in a hybrid approach with the TWINS model [17] to predict the noise radiation.
14.4.2 **Wheel Flats**

A wheel flat is an area of the wheel tread that has been worn flat. This usually occurs when the brakes lock up under poor-adhesion conditions at the wheel-rail contact due, for example, to leaves on the railhead in the autumn. Wheels with flats produce high levels of noise and impact loading of the track, which can lead to the damage of track components (see Chapter 9). Typically, flats can be around 50-mm long and in extreme cases up to 100 mm. After their initial formation, flats become ‘worn’, that is, rounded at their ends due to the high load concentration on the corners. A worn flat of a given depth is longer than the corresponding ‘new’ flat.

Wheel flats introduce a relative displacement input to the wheel-rail system in the same way as roughness. Figure 14.14 shows examples of the calculated response of the wheel-rail system to a new wheel flat of depth 2 mm (length 86 mm) for a nominal wheel load of 100 kN. The model used here represents the wheel by a mass and spring and the track by a simple state-space model fitted to the track mobility [40].

When the indentation (relative displacement input due to the wheel flat) appears between the wheel and rail, the wheel falls and the rail rises. Since the wheel and rail cannot immediately follow the indentation due to their inertia, the contact force is partly unloaded. At a train speed of 30 km/h, see Figure 14.14a, full unloading occurs first. After the relative displacement input reaches its maximum, the contact force increases rapidly until it reaches its peak. The peak force in this example is about four times as large as the static load. As the speed increases, contact is lost for longer periods during the unloading phase. At 80 km/h, see Figure 14.14b, a second loss of contact can be seen to occur. However, the second impact is much smaller than the first one. Comparisons with measured impact forces [41] suggest that the simplified geometry used here leads to over-estimates of the contact force. Measured wheel flat profiles are required to give more accurate predictions.

![FIGURE 14.14 Predicted wheel-rail interaction force and displacements of wheel and rail due to 2-mm newly formed wheel flat at train speed of (a) 30 km/h and (b) 80 km/h, showing —— wheel displacement, – – – rail displacement, and ---- relative displacement excitation. (From Wu, T.X., and Thompson, D.J., *J. Sound Vib.*, 251, 115–139, 2002. With permission.)](image-url)
It is not possible to use the contact force obtained from the impact model and apply it directly within the TWINS model, because the predicted interaction force is very sensitive to details of the wheel and track dynamics used in its prediction. With a modal wheel model, the force spectrum will have strong dips at the wheel’s natural frequencies. The wheel response has only shallow peaks, just above the natural frequencies. The interaction with the track thereby introduces apparent damping to the wheel. A hybrid approach has therefore been developed [40], whereby an equivalent roughness spectrum is derived. The equivalent roughness spectrum can then be used as the input to a more detailed linear frequency-domain model, such as the TWINS model, to predict the noise due to the impact.

Example results are given in Figure 14.15a. This shows the sound power due to one wheel and the associated track vibration for a 2-mm-deep new wheel flat at different speeds for a 100-kN wheel load. Results correspond to the average over one whole wheel revolution. Figure 14.15b shows, for comparison, corresponding results for roughness excitation due to a moderate roughness (tread-braked wheel roughness). As the speed increases, the noise at frequencies above about 200–400 Hz increases in both cases. The increase in rolling noise with increasing speed is greater than that due to the flat. For the wheel flats considered here, the noise generated exceeds that due to the tread-braked wheel roughness at all speeds and in all frequency bands; however, the noise due to roughness increases more rapidly with speed, so that, at sufficiently higher speeds, it can be expected to dominate. For corrugated track, the noise due to roughness would exceed that due to these wheel flats at 120 km/h.

Figure 14.16 shows a summary of the variation of the overall A-weighted sound power level with train speed. The predicted noise level due to conventional roughness excitation increases at a rate of approximately $30 \log_{10} V$, where $V$ is the train speed, whereas the noise due to flats increases at an average of around $20 \log_{10} V$ once loss of contact occurs. For example, loss of contact was found to occur for the newly formed 2-mm-deep flat at speeds above 30 km/h and for a rounded 2-mm-deep flat above 50 km/h. This variation with speed indicates that the radiated sound due to wheel flats continues to increase with increasing speed, even though loss of contact is occurring.

![Figure 14.15](image-url)  
**FIGURE 14.15** Sound power level due to wheel and track for (a) 2-mm new wheel flat and (b) rolling noise from moderate roughness, for speeds of — — — 30 km/h, ···· 50 km/h, — — 80 km/h, and —— 120 km/h. (From Wu, T.X., and Thompson, D.J., *J. Sound Vib.*, 251(1), 115–139, 2002. With permission.)
Impact noise from wheel flats is found to depend on the wheel load. The increase in noise between a load of 50 kN and 100 kN is about 3 dB. In contrast, the rolling noise due to roughness is relatively insensitive to wheel load.

### 14.4.3 Rail Joints, Switches and Crossings

In a similar way to wheel flats, rail joints provide discrete inputs to the wheel-rail system that induce quite large contact force variations. Rail joints can be characterised by a gap width and a step height (either up or down). Moreover, the rail often dips down to a joint on both sides. Such dips are also present at welds and are usually characterised in terms of the angle at the joint.

A similar approach has been used, as previously mentioned, to study the effects of rail joints [42,43]. The sound radiation was calculated using the same hybrid method as for the wheel flats. It was found, for realistic parameter values, that the gap width is insignificant compared with the step height and dip angle.

Results are shown in Figure 14.17a for un-dipped rail joints in the form of the total A-weighted sound power emitted by the wheel and rail during 1/8 sec. The results for a step-down joint are found to be virtually independent of the step height (only results for one value are shown) and also change very little with train speed. However, for step-up joints, both the peak contact force and the sound power level increase with step height and with train speed. The sound power level from a single joint has a speed dependence of around $20 \log_{10} V$.

In Figure 14.17b, results are given for dipped joints with no height difference. Here, a dip of 5 or 10 mm is considered as a quadratic function over a length of 0.5 m either side of the joint. A dip of 5 mm corresponds to a joint angle of 0.04 radians, which is large, although within a typical range, and a dip of 10 mm corresponds to 0.08 radians, which is severe. The 10-mm dip produces a similar noise level to a 1-mm step-up un-dipped joint, although for speeds above 120 km/h, the noise level from the dip joint becomes independent of train speed.

Figure 14.18 shows the predicted noise for joints with both dipped rails and steps. The noise radiation generally increases with speed, regardless of whether loss of contact occurs. For the 5-mm dip, the noise level increases by 8 dB when the step height increases from 0 to 2 mm. For the step-down joints, the noise level is higher than that without a step, although at higher speeds, the dip has
more effect than the step. The results for the 10-mm dip are similar for both step-up and step-down joints, indicating the dominance of the dip in this case.

To compare these results with typical rolling noise results, the time base of the joint noise should be adjusted to the average time between joints. This shows [42] that rolling noise due to the tread-braked roughness considered above is similar to the average noise due to 5-mm dipped joints with no height difference (Figure 14.17b). With a height difference of 2 mm, the average noise predicted from the joints increases to almost 10 dB greater than the rolling noise. Moreover, since the time between rail joints decreases as train speed increases, it is also found that the average noise level from joints increases at about $30 \log_{10} V$, similar to rolling noise.

**FIGURE 14.17** A-weighted sound radiated by one wheel and the associated track vibration during 0.125 second due to a wheel passing over different rail joints with 7-mm gap for (a) flat rail joints, showing — — — 1-mm step-up, — — — 2-mm step-up, ······ 3-mm step-up, —— 2-mm step-down, and (b) dipped rail joints with no height difference, showing —— 5-mm dip, — — — 10-mm dip. (From Iwnicki, S. (ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)

**FIGURE 14.18** A-weighted sound power radiated by one wheel and the associated track vibration during 0.125 second due to a wheel passing over rail joints with 7-mm gap for (a) 5-mm dip, and (b) 10-mm dip, showing ······ 2-mm step-up, – – – 1-mm step-up, — no height difference, * 2-mm step-down, o 1-mm step-down. (From Iwnicki, S. (ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
14.4.4 Reducing Impact Noise

To reduce impact noise, it is clearly desirable to remove the cause if this is possible. Wheel flats can be largely prevented by installation of wheel-slide protection equipment. Monitoring equipment is now widely used to identify wheels with flats, to allow them to be removed from service as quickly as possible for reprofiling. On main lines, jointed track has been mostly replaced by continuously welded rail in the last 40 years, although inevitably, joints remain such as expansion joints, track-circuit insulating joints and switches and crossings. Even so, measures such as swing-nose crossings allow the impact forces, and thus noise, to be minimised. Attention should also be given to ensuring that welded rail joints are as levelled as possible by using rail-straightening equipment.

Countermeasures that are effective for rolling noise, such as those discussed in Section 14.3, can be expected to work equally well for impact noise. This includes, for example, wheel damping, wheel shape optimisation, rail damping and local shielding.

14.5 Curve Squeal

14.5.1 Mechanisms of Squeal Noise Generation

Railway vehicles travelling around tight curves can produce an intense squealing noise. This is a particular problem where curved track exists in urban areas, and it has been found to be annoying to both residents and railway passengers. An extensive review of curve squeal can be found in [44].

When a railway wheelset in a bogie traverses a curve, it is unable to align its rolling direction tangentially to the rail (14.19). Owing to this misalignment, lateral sliding occurs together with the natural rolling of the wheel. In sharp curves, this leads to large creep forces at the wheel-rail interface; see also Chapter 7. The leading outer wheel tends to be in flange contact, with the resultant lateral force acting inwards to ensure that the wheelset remains on the track. Longitudinal and spin creep forces also act as shown in Figure 14.19.

The presence of the creep force due to the sliding at the wheel-rail interface is the main reason for the unstable dynamic behaviour, leading to squeal noise. Two different mechanisms have been

![Diagram of forces acting on wheels of a bogie in a curve](image)

**FIGURE 14.19** (Bottom half and top half) Schematic view of forces acting on wheels of a bogie in a curve; $N$ is normal load, $F_2$ is lateral creep force, $F_1$ longitudinal creep force and $M_3$ is spin moment. (From Iwnicki, S. (ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
proposed to explain how the creep force can generate squeal. Originally, attention was focussed on the dependence of the ‘creep curve’ on the sliding velocity (also known as the Stribeck effect [45]), while, more recently, researchers have also focussed on instabilities due to coupling between normal and tangential directions.

Figure 14.20 shows a typical creep curve relating creep force to creepage. At low values of creepage, the magnitude of the creep force increases linearly. At high values of creepage, the force becomes saturated, with a maximum value of \( \mu_0 N \), where \( \mu_0 \) is the friction coefficient and \( N \) is the normal load. In practice, however, the friction coefficient \( \mu \) is not a constant. It is usually recognised that ‘dynamic’ or ‘sliding’ friction coefficients are smaller than ‘static’ ones. In fact, the friction coefficient depends on the sliding velocity, decreasing as the velocity increases. Thus, as creepage increases beyond the saturation point, the creep force once more reduces in amplitude; see Figure 14.20b.

By analogy with a damper, which gives a reaction force that is proportional to the relative velocity, the falling creep curve can be considered as a ‘negative damping’. Thus, the reaction force decreases as the relative velocity increases. Since wheel modes have very low levels of damping (see Section 14.2.3), if this negative damping exceeds a certain level, it causes instability of the wheel modes, making them prone to ‘squeal’. In this case, unstable self-excited vibration occurs, and the vibration amplitude increases exponentially until it is bounded to a limit cycle by the non-linear effects in the creep force. The critical value of the structural damping can be calculated as:

\[
\xi_{\text{lim,n}} = -\frac{N}{2m_n \omega_n V_0} \frac{d\mu}{d\gamma_L}
\]

(14.10)

with \( N \) being the normal load, \( m_n \) the modal mass of wheel mode \( n \), \( \omega_n \) its natural frequency, \( V_0 \) the rolling velocity of the train and \( \gamma_L \) the creepage in the lateral direction. For structural damping values smaller than the critical limit, self-excited vibration can occur in the wheel for the mode considered. In this approach, single-wheel modes can be excited, and the squealing frequency would be coincident with the natural frequency of the mode. Equation 14.10 highlights the importance of the slope of the friction curve for this type of excitation. However, the actual trend of the creep curve is usually not known, and assumptions need to be made for modelling purposes.

As an alternative to the negative damping effect, mode-coupling phenomena have attracted the interest of various researchers in the last decade; see [46,47] for examples. This type of instability arises from the coupling between two vibration directions and involves two adjacent modes. In the case of squeal, the non-conservative nature of the friction force can transfer energy between the

![FIGURE 14.20 Typical creep force-creepage relationships for (a) constant friction coefficient and (b) velocity-dependent friction coefficient. (From Iwnicki, S. (ed.), Handbook of Railway Vehicle Dynamics, CRC Press, Boca Raton, FL, 2006. With permission.)](image-url)
normal and tangential directions, leading to this type of instability, which is also known as ‘flutter’. As a result, two adjacent modes will tend to merge, and the vibration of the system can build up at a frequency between those of the two vibration modes. The squealing frequency would therefore not necessarily correspond to any of the natural modes of the free wheel. Again, the non-linear nature of the creep force will result in a limit cycle. This type of instability does not require a negative slope of the friction curve. In its most simplified form, it can be illustrated by means of a two-degree-of-freedom system coupled to a moving belt through a simple Coulomb model for friction [48]. This is schematically shown in Figure 14.21, where the mass \( m \) and springs of stiffness \( k_1 \) and \( k_2 \) represent the structural properties of the system and \( k_H \) represents the contact stiffness of the wheel-track system.

For mode coupling, it is not easily possible to define threshold values to determine the presence of curve squeal. In fact, this depends on combinations of various parameters such as the friction coefficient, the modal damping ratios and the relative amplitude of the mode shapes in the different directions. A necessary condition [49], however, is that both modes have components in the normal and tangential directions at the contact point.

Various efforts have been made to demonstrate which of the two mechanisms is behind curve squeal, and arguments can be made in favour of one or the other. In various laboratory measurement campaigns [50,51], it was found that the creep curve in the lateral direction, measured in an average sense, showed clear falling trends, and this could be correlated to the presence of squeal. In addition, the measured squealing frequencies can often be attributed to single wheel modes. At the same time, it has also been demonstrated that numerical models relying solely on mode coupling can show good qualitative correlation with field tests [46], and, on some occasions, a clear frequency shift was found between the measured squealing frequency and the wheel’s natural frequencies [49]. It is therefore likely that both mechanisms are important, with the possibility for one of the two to be predominant, depending on specific situations.

Several observations indicate that the highest squeal noise amplitude is often generated by the leading inner wheel of a four-wheeled bogie or two-axle vehicle. This noise can be associated with the instability mechanisms listed previously. The fundamental frequency of such squeal noise tends to correspond to a natural frequency of the wheel and is often in the range of 200–2000 Hz. The wheel modes excited in this case tend to be axial modes with no nodal circle, and their maximum amplitude is at the wheel tread (see first row of Figure 14.5).

Contact between the wheel flange and the rail, which occurs at the leading outer wheel (and possibly the trailing inner wheel) in sharp curves, has been initially found to reduce the likelihood of squeal in some cases [52]. For example, Remington concluded from laboratory experiments that flange contact reduces the level of squeal noise [53]. However, curve squeal has also been found for the outer wheels and was associated with flange contact [54–56], either with the outer running rail, with check rails (or grooved heads in tramways), or with wing rails in switches and crossings. Compared with squeal measured at the inner wheel, the cases found at the outer wheels had, in general, higher fundamental frequencies and were more intermittent in nature. In these cases,

**FIGURE 14.21** Simplified model representing mode-coupling mechanism.
additional factors have been studied and found to play a role in curve squeal. These are the presence of two contact points [57], spin creepage [58] and longitudinal creepage [59,60].

Theoretical models for curve squeal have been developed by various authors. Rudd [61] (see also Remington [53]) indicated that instability of the lateral friction force was the most likely cause of squeal and gave a simple model. Fingberg [62] and Périard [63] extended this basic model by including improved models of the wheel dynamics, the friction characteristic and the sound radiation from the wheel. Time-domain calculations allow the squeal magnitude to be predicted as well as the likelihood of squeal to be determined. Heckl [64] has also studied squeal using a simplified model and provided experimental validation using a small-scale model wheel.

De Beer et al. [50] extended these models, based on excitation by unstable lateral creepage, to include feedback through the vertical force as well as through the lateral velocity. Their model consists of two parts: a first part, in the frequency domain, can be used to determine instability and to predict which mode is most likely to be excited, and a second part, in the time domain, calculates the amplitude of the squeal noise. This model has been extended further to allow for an arbitrary contact angle and to include lateral, longitudinal and spin creepage [59]. This allows it to be applied to flange squeal as well as squeal due to lateral creepage.

Chiello et al. [65] developed a curve squeal model that accounts for both axial and radial dynamics to predict growth rates and unstable frequencies. The effect of mode coupling instability was discussed, but it was found that only a creep curve with negative slope would result in squeal.

Brunel et al. [66] introduced a transient model for curve squeal and found that even a positive friction law can lead to instability and limit cycle. This feature was associated with the coupling between the normal and lateral dynamics. More recently, Glocker et al. [46] and Pieringer [47] demonstrated that models with a constant friction coefficient can show squeal at relevant frequencies and give good qualitative comparisons with measured data.

14.5.2 Reducing Squeal Noise

In discussing solutions for curve squeal, it is of little value to quote decibel reductions. The nature of the instability is such that effective measures are those that eliminate the squeal rather than reducing it. Thus, noise barriers are generally ineffective against squeal noise. Curve squeal tests are also extremely unreproducible due to a high sensitivity to parameters such as temperature, humidity, train speed, track geometry, wheel and rail wear.

Known solutions for curve squeal include lubrication using either grease or water or the application of friction modifiers that reduce the difference between static and sliding friction coefficients. If lubricants are used, it must be ensured that they do not lead to loss of adhesion, as this could compromise safety. Grease is therefore only applied to the rail gauge corner or wheel flange. Although this location may not be the primary cause of squeal noise, this can nevertheless reduce the occurrence of squeal by modifying the curving behaviour. Water sprays have also been used effectively in a number of locations.

Friction modifiers act by reducing or eliminating the falling friction characteristic without reducing the level of friction. These can be applied either to the track at the entrance to a curve or on the vehicle. They have been shown to be very effective in eliminating squeal and can be applied to the top of the railhead without compromising traction or braking [67].

Wheel damping treatments are also known to reduce the occurrence of squeal. In this case, a small increase in the level of damping can be effective in eliminating squeal. In addition to the forms of damping discussed in Section 14.3.2, ring dampers have been used as a simple means of increasing the damping of a wheel [68].

Effective solutions can also be sought in the design of vehicles for curving in order to reduce the creepages (see also Section 17.6.5). Unfortunately, this is often in conflict with the design of bogies for stability at high speed.
14.6 AERODYNAMIC NOISE

14.6.1 SOURCES OF AERODYNAMIC NOISE

The sound power from aerodynamic sources increases more rapidly with speed than that from mechanical sources. Aerodynamic sources can be classified as monopole, dipole or quadrupole [2]. Most aeroacoustic sources on a train are of a dipole type, such as the tones generated by vortex shedding from a cylinder and turbulence acting on a rigid surface. For such sources, the sound power increases according to the sixth power of the flow speed $V$, which when expressed in decibels gives a rate of increase of $60 \log_{10} V$ [69]. The noise from free turbulence, such as jet noise, has a quadrupole source type and a speed dependence of $80 \log_{10} V$ [70]. Moreover, the frequency content also changes with speed, shifting towards higher frequencies with increasing speed; consequently, the A-weighted sound pressure level will increase at a greater rate than what these theoretical values suggest.

As a consequence of the higher speed dependence, compared with $30 \log_{10} V$ typical of rolling noise, aerodynamic noise sources will become predominant in the overall noise above a certain speed. This transition speed is often considered to be around 300 km/h [71], although more recent results suggest that it may be as high as 370 km/h in some situations [72].

Where noise barriers are placed alongside the track, the rolling noise may be attenuated by 10–15 dB, whereas the aerodynamic sources from the upper part of the train, and particularly the pantograph, remain exposed. This causes aerodynamic noise to become important at lower speeds in such situations. Aerodynamic sources are also important for interior noise in high speed trains, particularly the upper deck of double-deck trains, where rolling noise is less noticeable.

Important aerodynamic sources are found to fall into two main categories [71]. Dipole-type sources are generated by air flow over structural elements and cavities, including the bogies, the recess at the inter-coach connections, the pantograph and electrical isolators on the roof and the recess in the roof in which the pantograph is mounted. In addition, the flow over the succession of cavities presented by louvered openings in the side of locomotives is a source of aerodynamic noise, the form of which depends on the length and depth of the cavity. In the second category, which may have a dipole or quadrupole nature, noise is created due to the turbulent boundary layer.

14.6.2 MEASUREMENT METHODS

An important tool for determining the location of aerodynamic sources on a moving train is an array of microphones, mounted at some distance from the track [73]. An example is shown in Figure 14.22. Using a technique known as beamforming, the sound coming from a certain direction can be obtained by combining the signals from these microphones after applying a suitable delay to each channel. For a moving source, a technique is also required to follow the source motion and to remove the Doppler shift. The output from the beamforming is a map of sound level over the surface of the train from which the locations of the main sources can be identified. However, it is more challenging to quantify the strengths of the sources [72].

Laboratory measurements can also be made in a wind tunnel [72]. This should have a low background noise and should be treated with anechoic boundaries to prevent sound reflections. For practical reasons, testing in wind tunnels often relies on making measurements of individual components or using reduced scale models. In the latter case, scaling laws based on non-dimensional quantities such as the Reynolds number must be applied to derive the corresponding result at full scale. Nevertheless, care is required as, in certain cases, such as the flow over a circular cylinder, the noise level does not follow a linear trend with Reynolds number [74].

14.6.3 PREDICTION METHODS

Computational fluid dynamics (CFD) techniques have become popular in recent years. This is a vast and rapidly changing field that cannot be covered in detail here. Despite the rapid development,
however, it is still the case that conventional CFD methods cannot be used to calculate the aerodynamic noise from a train, so that the most promising developments are based on improved models of components or sub-assemblies [72]. In addition, Lattice Boltzmann methods have also been used recently, which are well suited to handle complex arbitrary geometries; see [75] for example.

An alternative approach is to use a semi-empirical component-based method. This has been used in particular to predict noise from the pantograph [76]. This uses a database of measured spectra from cylinders that are normalised using various non-dimensional parameters. The overall noise spectrum is assembled from the various components by neglecting flow interactions between them.

14.6.4 Control of Aerodynamic Noise

A review of countermeasures for the control of aerodynamic noise is included in [72].

These have focussed particularly on the pantograph, owing to its important location on the roof of the train. Significant noise reductions can be achieved by simplifying the design to a single-arm concept with larger main struts and by eliminating, as far as possible, any small components that generate high-frequency sound. Adding discrete holes to a cylinder can suppress the vortex shedding peak as long as the acoustic resonance of the holes does not match this peak. Roof-mounted shields alongside the pantograph are also included on some high-speed train designs.

The bogie region is also an important source of aerodynamic noise, especially as there are many more bogies on a train than pantographs. The leading bogie usually has the highest noise levels, as it is subjected to a higher in-flow velocity [77]. Attention to the shape of the train nose, particularly the underside and ‘snowplough’ region, can help to alleviate this [75]. In addition, fairings on the side of the bogies can help to smoothen the flow and to shield the noise [77].

FIGURE 14.22 An array of 90 microphones deployed at the trackside for identifying noise sources on a moving train.
14.7 OTHER SOURCES OF NOISE

14.7.1 ENGINE NOISE

Power units on trains are generally either electric or diesel. Noise from diesel locomotives is mostly dominated by the engine and its intake and exhaust. Space restrictions often limit the ability to silence the exhaust adequately, although in modern locomotives, this has been given serious attention. On electrically powered stock, and on diesels with electric transmission, the electric traction motors and their associated cooling fans are a major source of noise. Most sources of noise from the power unit are largely independent of vehicle speed, depending rather on the tractive effort required. The whine due to traction motors is an exception to this.

14.7.2 FANS AND AIR-CONDITIONING

Fans are an important source of noise on modern trains. In addition to traction motor cooling fans, the ventilation and air-conditioning systems contain fans, which produce significant noise. The most efficient solution for fan noise is to replace axial fans by radial ones, which can give noise reductions of 10 dB(A) [78], but this is not always possible.

The TSI Noise [1] places limits on standstill noise and starting noise as well as pass-by noise at constant speed. These situations are often dominated by the noise from engines and auxiliaries, principally from fans.

14.8 VEHICLE INTERIOR NOISE

14.8.1 VEHICLE INTERIOR NOISE LEVELS

All the noise sources discussed previously are also of relevance to interior noise in trains [79]. Noise is transmitted from each of these sources to the interior by both airborne and structure-borne paths, with structure-borne transmission often dominant at low frequencies and airborne transmission dominant at higher frequencies. The noise from the wheel-rail region is often the major source. In addition, on vehicles with under-floor diesel engines, noise from the engine can be significant. Noise from the air-conditioning system can also require consideration in rolling stock where this is present. There is often very limited space in which to package the air-conditioning unit and ducts.

Example spectra are given in Figure 14.23. These results show that modern high-speed trains are quieter at 300 km/h than a conventional ‘rail car’ at a lower speed [80]. In tunnels, the noise levels will increase considerably in the mid-frequency range due to a greater contribution from the walls, windows and roof [81].

14.8.2 MEASUREMENT QUANTITIES FOR INTERIOR NOISE

Conventionally the A-weighted sound pressure level has been used to specify acceptable levels inside vehicles, as indicated in the standard [82]. Internationally agreed limits are 68 dB(A) in second class and 65 dB(A) in first class [81]. However, as seen previously, the spectrum of noise inside trains contains considerable energy at low frequency. This low-frequency sound energy can be a source of human fatigue but is not effective in masking speech, for which noise in the range 200–6000 Hz is most effective [2].

Passenger requirements for noise inside a train vary from one person to another [83]. Clearly, it is desirable that the noise should not interfere with conversation held between neighbours. However, particularly for a modern open-saloon-type vehicle, silence would also not be the ideal. There should be sufficient background noise so that passengers talking do not disturb other passengers further along the vehicle (people talking loudly into mobile phones are a particular source of annoyance).
According to [84], for example, the interior noise level should be at least 60 dB(A) to avoid disturbance by other passengers. Various alternative quantities exist that can be used to define acceptable environments. These include the B-weighted level, preferred speech interference level, loudness level, alternative noise criteria (NCA), noise ratings (NR) and room criteria (RC) [85].

The interior sound level varies considerably within a vehicle. Figure 14.24 shows some example measured results where a loudspeaker has been placed at one end of an open-saloon vehicle. This was a British Rail Mk II coach dating from the 1960s, although the interior dated from the 1990s. The solid line shows the relative sound pressure level along a line down the central gangway at the height of the headrests. Results are shown in three example one-third octave bands. At low frequencies, strong modal patterns are observed due to the long acoustic wavelength. At higher frequencies, considerable decay in the sound level is observed along the coach due to the absorptive properties of the seats, carpets, etc. Additional attenuation is seen at the middle of the coach, where two glass partial screens were present at either side of the door.

Also shown are measured results at positions in front of each seat headrest. The seats were arranged in groups of four, with tables between them. At low frequencies, these measurements follow the same pattern as the gangway measurements, but at higher frequencies, considerable differences can be seen between adjacent seated positions. These spatial variations may be experienced by passengers in the vehicles; the 500-Hz frequency band, for example, is quite important for speech interference. It can also be expected that differences will occur between left and right ear positions at an individual seat, leading to binaural effects. Clearly, in a running vehicle, the source positions will differ from this, but these results serve to illustrate the general trends that can be expected.

14.8.3 AIRBORNE TRANSMISSION

Airborne sound transmission into the vehicle occurs due to acoustic excitation of the vehicle floor, walls, windows, doors and roof. The acoustic performance of a panel can be measured by placing it between two reverberant rooms and measuring the difference in sound pressure level between the two rooms [2]. The sound reduction index (or transmission loss) is the difference between the incident intensity level and the transmitted intensity level, which can be derived from such a measurement after allowing for the size of the panel and the absorption in the receiver room.
A typical sound reduction index of a homogeneous panel is shown in Figure 14.25. Generally, the sound reduction index of panels is dominated by the ‘mass law’ behaviour in a wide frequency range. At high frequencies, the coincidence region occurs where the wavelengths in the structure and air are similar. Here, a dip in the sound reduction index occurs, the extent of which depends on the damping. The mass law behaviour extends from the first resonance of the panel up to just below the critical frequency. In this region, the bending stiffness of the panel and its damping have no effect on the sound transmission (see [2] for more details).
The use of lightweight constructions such as extruded aluminium and corrugated steel leads to a low sound reduction index. This follows from the mass law, which states that the sound reduction index reduces by 6 dB for a halving of panel mass. However, such structures tend to have a performance that is even worse than what the mass law would suggest, owing to the presence of an extended frequency region over which coincidence effects occur. For example, Figure 14.26 shows measurements of the sound reduction index from a 60-mm-thick extruded aluminium floor of a railway vehicle with a 3-mm-wall thickness, taken from [86] (similar results are also found in [87]). Also shown is the ‘field incidence’ mass law estimated for a homogeneous panel of similar mass [2].

Figure 14.25 Typical sound reduction index of a homogeneous panel due to a diffuse incident field.

Figure 14.26 Octave band sound reduction index of extruded aluminium floor, showing —— measured on bare floor panel, ···· field incidence mass law for 30 kg/m², — — — measured for bare floor panel plus 12-mm suspended wooden deck, and — · — · measured for damped floor panel plus 12-mm suspended wooden deck (data from [86]). (From Iwnicki, S. (ed.), Handbook of Railway Vehicle Dynamics, CRC Press, Boca Raton, FL, 2006. With permission.)
Clearly, the extruded panel exhibits a much lower sound reduction index than this. It can be brought closer to the mass law behaviour by the use a suspended inner floor and by adding damping treatments to the extruded section.

### 14.8.4 Structure-Borne Transmission

In addition to the airborne path, considerable sound power is transmitted to the vehicle interior through structural paths. This originates from the wheel-rail region as well as from under-floor diesel engines where these are present. Structure-borne engine noise can be reduced significantly in many cases by applying good mounting practice [79]. The mount stiffness must be chosen considering the frequency characteristics of the engine. An incorrect choice of stiffness can lead to amplification rather than attenuation of transmitted vibration. Flanking paths via pipes and hoses should also be avoided.

Structure-borne noise from the wheel-rail contact is transmitted via the bogie frame through the primary and secondary suspensions as well as a host of other connections such as dampers, traction bars, etc. The dynamic stiffness of these elements are frequency dependent, often with internal resonances, and should be characterised carefully.

### 14.8.5 Prediction of Interior Noise

Deterministic methods such as FEM may be applied at low frequencies to predict the vehicle interior noise; see [88] for example. Owing to the regular geometry, an analytical model of the interior may also be used to construct the interior acoustic field on the basis of simple room modes [89].

However, at high frequencies, the number of modes becomes prohibitive for such approaches. The preferred analysis method for frequencies above about 250 Hz is therefore Statistical Energy Analysis (SEA). This can be used in both predictive mode [86,90] and experimental mode [91]. However, in predictive mode, it is not straightforward to define the coupling loss factors between the various sub-systems, especially where aluminium extrusions [87,92,93] or other inhomogeneous constructions are used [94]. Moreover, as SEA is a statistical method, it provides an average result and cannot account easily for the spatial variations in sound field, such as seen in Figure 14.24.

### 14.9 Ground-Borne Vibration and Noise

#### 14.9.1 Introduction

Vibration generated at the wheel-rail interface is also transmitted through the ground. Excitation occurs due to the passage of individual wheel loads along the track (quasi-static excitation) and due to dynamic interaction forces caused by irregularities of the wheels and track. This vibration is transmitted into nearby buildings, where it may cause annoyance to people or malfunctioning of sensitive equipment. People may either perceive the vibration directly, as low-frequency ‘feelable whole body’ vibration (between 2 Hz and 80 Hz) or, indirectly, as re-radiated noise caused by vibration of the floors and walls at higher frequencies (around 20–250 Hz).

The highest levels of low frequency ground-borne vibration are usually produced by freight trains at sites with soft soil. Heavy axle-load freight traffic, travelling at relatively low speeds, causes high-amplitude vibration at the track that excites surface-propagating waves in the ground. This type of vibration often has significant components at very low frequency (below 10 Hz) and may interact with the frequencies of buildings ‘rocking’ or ‘bouncing’ on the stiffness of their foundations in the soil. This phenomenon is especially associated with soft soil conditions, where it is found that significant levels of vibration may propagate up to as much as 100 m from the track. At these frequencies, the vibration is perceived in the building as ‘whole body’ vibration that can be felt. This is usually assessed under the principles of ISO 2631-1 [95]. High levels of vibration cause annoyance and, possibly, sleep disturbance. Complaints are often expressed in terms of concern.
over possible damage to property, although, for the levels of vibration normally encountered from
trains, such concern is unlikely to be borne out when assessed against the criteria for building dam-
age, for example, BS 7385 [96] and DIN 4150 [97].

Passenger trains also may cause significant levels of vibration, particularly electric multiple units
with high unsprung masses that dynamically interact with the rails due to the unevenness of the
wheels and the track. For the frequency range 2–250 Hz and a train speed range of 10–100 m/s
(36–360 km/h), the corresponding wavelengths of the vertical unevenness lie within the range
0.04–50 m. When considering the rail surface, at wavelengths less than about 1 m, this vertical
unevenness is most commonly caused by irregular wear or corrugation of the rail contact surface,
whereas at much longer wavelengths, it is due to undulations in the track bed. On the wheels, short
wavelength unevenness is again caused by wear, whereas discrete wavelengths up to about 3 m are
present due to out-of-roundness. In addition, dynamic forces are generated as impacts as the wheels
traverse switches and crossings or badly maintained rail joints [98,99]. However, at sites of mixed
traffic, it is usually the case that a few freight trains, perhaps running at night, are identified as the
worst cases, and it is these that dominate the assessment of potential annoyance.

High-speed passenger trains sometimes travel at speeds approaching the speed of propagation of
vibration in the ground and embankments. This has been the concern of track engineers for some
years because of the large displacements that can be caused in the track support structure, in electri-
fication masts, etc. This is particularly important on soft soil, where the wave speeds in the ground
are relatively low, and, in some cases, it may cause vibration that exceeds the limit for safety and
stability. Although the occurrences of this are comparatively rare, the topic has attracted consider-
able attention amongst researchers recently because of the expansion of the network of high-speed
railways; see [100–102].

Trains that run in tunnels also cause vibration that is transmitted to nearby buildings. This has
higher frequency content than vibration from surface tracks/trains and generally has lower ampli-
tudes. Although no direct airborne noise can be heard, vibration at the low end of the audible fre-
quency range, from about 20 Hz to 250 Hz, may excite bending in the floors and walls of a building,
which then radiates noise directly into the rooms. This rumbling noise may be found to be all the
more annoying, because the source cannot be seen and no screening remedy is possible.

14.9.2 VIBRATION FROM SURFACE RAILWAYS

All grounds are stratified on some scale, and this layered structure of the ground has important
effects on the propagation of surface vibration in the frequency range of interest. Typically, grounds
have a layer of softer ‘weathered’ soil material that is only about 1–3 m deep on top of stiffer soil lay-
ers or bedrock, depending on the geology of the site. In such a layered medium, vibration propagates
parallel to the surface via a number of wave types or ‘modes’. These are often called the Rayleigh
waves of different order (‘R-waves’) and the Love waves. The Rayleigh waves are also called P-SV
waves, since they involve coupled components of dilatational deformation and vertically polarised
shear deformation. Here, the name P-SV wave is preferred, and the term Rayleigh wave is reserved
for the single such wave that exists in a homogeneous half-space. The Love waves are decoupled
from these and only involve horizontally polarised shear deformation; they are also known as SH
waves. Since the vertical forces in the track dominate the excitation of vibration in the ground, the
SH waves are not strongly excited. They are not considered further in the present discussion.

To illustrate, measured examples of P-SV surface waves are shown in Figure 14.27 from two sites
in the UK. Such three-dimensional plots are produced by measuring the transfer function along the
ground surface over different distances and then applying a Fourier transform in the spatial domain
to express the response in terms of wavenumber ($k = 2\pi/\lambda$, where $\lambda$ is the wavelength) at each fre-
quency [103]. The phase velocity of each wave at a particular frequency is given as the ratio of the
frequency over the wavenumber ($c = \omega/k$). Each peak in the diagram represents a wave type associ-
ated with a cross-sectional mode of the soil.
For the first site in Figure 14.27a, at low frequencies (below 15 Hz), the fundamental surface wave has a quite high phase velocity, corresponding to Rayleigh waves of the underlying stiff ground layers. At high frequencies, the phase velocity converges asymptotically to a value of about 170 m/s, corresponding to the Rayleigh wave velocity in the top layer. For the second site in Figure 14.27b, a surface wave is seen between 10 Hz and 30 Hz, with a wave speed of about 100 m/s that tends asymptotically to a value of about 80 m/s at higher frequencies. However, above 20 Hz, the main activity corresponds to a higher wave speed of about 130 m/s.

The layered arrangement and properties of the ground can vary considerably from one site to another and even within a single site. Theoretical ground models are therefore essential for understanding the physics of ground-borne vibration propagation. The majority of these models assume that the ground consists of horizontal, isotropic and homogeneous infinite layers (layered half-space). Apart from a small zone in the immediate vicinity of the track, it can be assumed that the strain levels in the soil remain relatively low during the passage of a train, so that a linear elastic constitutive behaviour can be assumed.

Examples of the wave propagation of the P-SV waves in such a layered half-space are shown in Figure 14.28. These are the characteristic curves of the waves that propagate at the surface of a ground modelled as a typical soft layer of ‘weathered’ soil overlying a stiffer substratum of material. These results are calculated using a semi-analytical model [104]. The soft soil is modelled as a 2-m-deep layer with a shear wave (S-wave) speed of 118 m/s and dilatational wave (P-wave) speed of 360 m/s, and the substratum is a half-space with shear wave speed of 245 m/s and dilatational wave speed of 1760 m/s. Damping is included in both materials as a loss factor of 0.1.

Since the example ground is not homogeneous, the P-SV waves are dispersive; the phase velocity and the attenuation coefficient of the waves vary with the frequency. Figure 14.28a presents the dispersion diagram for the example ground (only the propagating P-SV modes are shown) in which the wavenumber is plotted as a function of frequency. Each line of the diagram represents a wave type associated with a cross-sectional mode of the layered soil. The corresponding phase velocities are shown in Figure 14.28b.

For this example set of soil parameters, at very low frequency, only a single mode exists, and this has a wave speed close to that of the shear waves in the substratum. Around 15 Hz, a quarter wavelength of the shear wave fits in the depth of the weathered material. Above this frequency, waves can propagate in the upper layer, with little influence of the underlying soil. At higher frequency, as the wavelengths of shear waves become small compared with the depth of the weathered material layer, the wavenumber of the slowest wave converges towards that of the Rayleigh wave in a half-space of the layer material. Higher-order propagating wave types ‘cut on’ at frequencies of 18 Hz, 35 Hz and 74 Hz.
In Figure 14.29, the maximum rail response (displacement) is shown for a unit constant load moving with different speeds. The results are given for the same example ground for two different track forms, with the properties given in Table 14.1. An important increase in vibration occurs as the load speed approaches a certain ‘critical’ value. Comparing the rail vibration level and the values of the peak-response load speed for the two different track forms, it can be seen that for the slab track, the critical speed occurs at a higher speed, and, for all speeds below the critical speed, the maximum rail response is lower than for the ballasted track. A more detailed discussion of the critical speed and the influence of the track parameters on this are given in [105,106].

The wave field for a single constant load moving along the ballasted track at different speeds is shown in Figure 14.30. For a load speed below that of any of the waves in the ground, the displacement ‘bowl’ under the single load is indicated in Figure 14.30a by the positive (upward) displacement under the track. Little effect is observed only a few metres away, although close to the track, the passage of the quasi-static displacement will be observed. Figure 14.30b shows what happens when the load travels at a speed of 115 m/s near to the ‘critical’ speed. Larger displacements occur at the loading point and displacements are observable at greater distances along the track than for the
lower load speed. Waves start to extend sideways from the track. The effects of a further increase of speed to 150 m/s are shown in Figure 14.30c. At this speed, in excess of the P-SV wave speed of the layer material (see Figure 14.28b), a number of waves are created in the track behind the load, and propagating waves may be seen travelling with significant amplitude away from the track.

If multi-body models representing the vehicles of a train are coupled to the model for the track-ground system, a theoretical model that predicts the total vibration field can be produced [107–112]. A number of models in which the vibration excited by the moving axle loads of the whole train and

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### Table 14.1

<table>
<thead>
<tr>
<th>Parameters Used for the Ballasted Track and the Slab Track in Figure 14.29</th>
</tr>
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<tbody>
<tr>
<td>Rail and railpads</td>
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<tr>
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<tr>
<td></td>
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<tr>
<td>Sleepers and ballast</td>
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<tr>
<td>Slab</td>
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by the irregular vertical profile of the track for all the axles of a train have been validated by comparison with measured vibration for a number of sites, for example, [110,113]. Examples of vibration spectra at the trackside are shown in Figure 14.31, which compares measurements and predictions from a semi-analytical model [114]. Here, the ground is a fairly soft clay and track is on a shallow embankment; the train speed was 47 m/s.

14.9.3 Ground-Borne Noise from Trains in Tunnels

To illustrate the vibration propagation from tunnels, the results from a coupled two-dimensional (2D) FEM/boundary element (FE-BE) model developed in [115] are presented. This was later extended to a so-called 2.5D model, in which the variations in displacement in the third direction are allowed for using a wavenumber transform [116].

Figure 14.32 shows the exaggerated instantaneous particle displacement at a number of points in the ground to illustrate the wave pattern radiating away from an oscillating load applied at the base of a circular tunnel at high frequency. The tunnel has a typical 3.5 m outer radius in this

FIGURE 14.31 Vertical velocity level at Steventon for train speed of 170 km/h (47 m/s): (a) at 12 m from the track and (b) at 20 m from the track, showing — predicted total level, ••• predicted level due to quasi-static loads, and —— measured level. (From Triepaischajonsak, N. et al., J. Rail Rapid Transit., 225, 140–153, 2011. With permission.)

case, without the concrete lining, and is 15 m deep at the rail [115]. Ground properties typical of a deep clay formation have been used, namely, an S-wave speed of 610 m/s and a P-wave speed of 1500 m/s (implying a Poisson’s ratio of 0.4); the density of the material has been assumed to be 1700 kg/m$^3$, and the damping loss factor is set to 0.15. Boundary elements are used to represent the ground-tunnel interface and the ground surface from +50 m to −20 m relative to the vertical centreline of the tunnel. The boundary elements allow a ground of infinite extent to be represented [115].

It can be seen that a relatively simple pattern of cylindrical wave fronts radiates towards the surface at greater distances from the tunnel. The strongest component of deformation in these waves is shear. At this frequency (100 Hz), the greatest amplitudes of response on the ground surface are at a distance of about 15–20 m from the tunnel alignment rather than directly above it.

Figure 14.33 shows the calculated response on the surface of the ground to an oscillating load at 60 Hz; in this case, unlike Figure 14.30, the load is in a tunnel. The response is calculated using the model presented in [117] for the soil and tunnel properties taken from [118]. As in Figure 14.32, the highest levels of vibration can be seen to be about 15 m to the side of the tunnel alignment, with the propagation pattern beyond this showing decaying circular wave fronts, whilst the vibration field above the tunnel is more complicated.

Again, by coupling multi-body vehicle models to the track/tunnel/ground system, the total vibration field can be produced. Figure 14.34 shows the total (quasi-static and dynamic) vibration level predicted using the coupled 2.5D FE-BE model [118]; the results are compared with the measured response from a site location in London on a conventional underground line. The response is given at the tunnel invert and on the ground surface. The predictions show a good agreement with the measurements. The fluctuations below 10 Hz are not observed in the measurements, probably due to the influence of background noise in the measurements. At the ground surface, the quasi-static component of vibration is negligible, and the ground vibration shown in the prediction is entirely due to the dynamic component. The response shows a peak at around 63–80 Hz, which is due to the resonance frequency of the vehicle unsprung mass on the track stiffness. Compared with Figure 14.31, it can be seen that the response on the ground surface is 20–30 dB smaller than that for a surface railway. Although this is unlikely to produce feelable vibration, it can nevertheless produce significant ground-borne noise in neighbouring buildings.

FIGURE 14.33  Vertical response amplitude of the surface of the ground to a load oscillating at 60 Hz; the tunnel lies at a depth of 20 m.
14.9.4 Controlling Low-Frequency Ground Vibration

Ground-borne vibration from railways can be controlled at different levels: at the source (train-track-soil interaction), in the transmission path or at the receiver (building). In the present discussion, mitigation measures at the receiver, such as base isolation of buildings and box-in-box arrangement of rooms, will not be considered further. The focus is placed on the dominant mechanism of vibration excitation and the interaction between the track and the ground, as well as on the transmission path through the ground.

In general, the treatment of low-frequency vibration, which involves surface waves with longer wavelengths and larger penetration depths, is considered more difficult, less effective and less economical than the mitigation of higher-frequency ground vibration leading to ground-borne noise.

14.9.4.1 Reducing the Track Geometric Unevenness

Long-term track usage may cause differential ballast/soil settlement and lead to an increase of the long-wavelength geometric unevenness of the track. If the dynamic component of excitation dominates the vibration, reducing the amplitude of long-wavelength components of the vertical track profile should help to reduce the vibration levels in the low-frequency range. This can be achieved by track realignment, in particular for the case of ballasted tracks by tamping the ballast. However, this has no effect on vibration very close to the track, which is dominated by the quasi-static loads. In addition, well-maintained turnouts (switches and crossings) and the removal of wheel flats will reduce large-impact forces and lead to substantial reductions of dynamic loads and further track deterioration.

14.9.4.2 Rolling Stock Modifications

It has been observed at mixed traffic sites that a particular train service may give rise to the main complaints of vibration. Thus, there is potential for vibration mitigation by reducing the dynamic vehicle loads through modification of the rolling stock characteristics.

Concentrating on the dynamic excitation, modifications to the dynamic properties of the vehicles can lead to reduction of the ground vibration. In [119], the effect of the vehicle parameters on railway-induced ground vibration was studied. It was shown that the parameters of most influence are the unsprung mass and the stiffness of the primary suspension, with higher vibration levels occurring for heavier unsprung masses and for stiffer primary suspensions. However, for the range of vehicle parameters investigated, the effect on the levels of ground-borne vibration is relatively small.
Similar findings are shown in Figure 14.35, where the reduction in ground-borne vibration at 16 m from the track due to reduction of the unsprung mass, the bogie mass and the primary suspension stiffness is given as a level difference in decibels, relative to a nominal train model. The results for this example are given in one-third octave bands and obtained using the model presented in [112] and the train parameters for the nominal model given in Table 14.2 (a four-vehicle train model was used).

It can be seen that, by reducing the unsprung mass from 1800 kg to 1200 kg per wheelset, the levels of ground-borne vibration can be reduced for the whole range of feelable vibration (between 2 Hz and 80 Hz), with a maximum reduction of 6 dB at 63 Hz. However, above 80 Hz, in the frequency range of ground-borne noise, the reduction of the unsprung mass could lead to an increase of vibration. The modifications to the bogie mass and primary suspension stiffness can also lead to small reductions between 2 Hz and 5 Hz.

**TABLE 14.2**

Parameters Used for the Nominal Vehicle Model Used in Figure 14.35

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<tr>
<th>Carbody</th>
<th>Mass</th>
<th>26,200 kg</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Pitching moment of inertia</td>
<td>$2 \times 10^6$ kgm$^2$</td>
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<td></td>
<td>Overall vehicle length</td>
<td>20 m</td>
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<tr>
<td>Bogie</td>
<td>Mass</td>
<td>5,000 kg</td>
</tr>
<tr>
<td></td>
<td>Pitching moment of inertia</td>
<td>6,000 kgm$^2$</td>
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<tr>
<td></td>
<td>Half distance between bogie centres</td>
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<tr>
<td>Wheelset</td>
<td>Mass</td>
<td>1,800 kg</td>
</tr>
<tr>
<td></td>
<td>Half distance between axles</td>
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</tr>
<tr>
<td></td>
<td>Total axle load</td>
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</tr>
<tr>
<td>Primary suspension</td>
<td>Vertical stiffness per axle</td>
<td>850 kN/m</td>
</tr>
<tr>
<td></td>
<td>Vertical viscous damping per axle</td>
<td>20 kNms/m</td>
</tr>
<tr>
<td>Secondary suspension</td>
<td>Vertical stiffness per bogie</td>
<td>600 kN/m</td>
</tr>
<tr>
<td></td>
<td>Vertical viscous damping per bogie</td>
<td>20 kNms/m</td>
</tr>
</tbody>
</table>
The effect of the unsprung mass is especially important for the cases of freight wagons with friction-damped suspensions used to carry materials such as cement, where the suspensions may seize up and result in high vibration levels. Modern passenger trains often have lower unsprung masses than those from prior to the 1990s. This is done in order to reduce track damage, and it has been achieved by avoiding electric motors to be hung directly on the axle. Instead, the motors are hung from the bogie, and vibration isolation is included in the transmissions. However, the means for reducing the unsprung mass significantly beyond this, for the sake of ground vibration reduction, are probably limited.

Regarding the suspension stiffness, some bogies exist for freight wagons that are designed to reduce the track forces by using two-stage suspensions, and these wagons have also been observed to cause much less vibration than conventional freight vehicles at vibration sensitive sites and for frequencies below 10 Hz. It should also be noted that changes in the geometrical parameters (e.g., bogie and axle distance or two-axle wagons) affect the axle and bogie passage frequencies and result in shifts in the one-third octave band spectrum of the quasi-static and the dynamic response [119].

14.9.4.3 Ground-Based Measures

To prevent vibration transmission in the ground, open trenches and ‘in-filled’ trenches (buried walls) can be considered as possible solutions, as shown in Figure 14.36. For a homogeneous ground, both measurements and simulations [120] show that, for an open trench to be effective, the depth should be more than about half the Rayleigh wavelength. However, for a layered ground, the performance is also influenced by the depth and stiffness of the soil layers. For a layered ground with a soft weathered layer above a stiffer substratum, significant reductions can be achieved if the trench cuts through the upper layer. The width of the open trench has been shown [120] to have a relatively small effect on the benefit. For practical reasons, trenches are usually filled with a soft barrier material. However, this leads to a significant reduction in their performance, as vibration can be transmitted through the barrier material as well as be diffracted beneath it [120].

A stiff wave barrier consisting of a concrete slab wall, a row of steel or concrete piles or a sheet pile wall embedded in the ground can also be effective in the mitigation of vibration [121,122]. In this case, the stability of the barrier is easy to achieve, and the installation of the barrier can be more straightforward. Such mitigation screening measures are more effective at sites with a soft soil.

As an alternative to vibration screening, the vibration may be reduced by installing stiff inclusions in the soil under or near the track [123,124]. This method can be achieved by stiffening the soil locally or replacing it with concrete, as shown in Figure 14.37.

Heavy masses or walls made of concrete or stone gabion baskets next to the track can lead to a reduction in vibration levels above a certain frequency. This frequency can be estimated as a mass-spring system resonance frequency, which is determined by the vertical dynamic stiffness of the soil.
and the mass of the block or wall \([125]\). Therefore, by increasing the mass of the block or wall, the mass-spring resonance frequency is reduced and the performance at lower frequencies is improved. This is the case also for sites with soft soil, where the mass-spring resonance is low because of the low dynamic stiffness of the soil. At frequencies higher than the mass-spring resonance, and provided that the stiffness difference between the soil and the installed wall is large enough, the propagation of the surface waves is restricted and the incident waves are scattered, resulting in a reduction of the transmitted wave field. These measures could be designed in conjunction with noise barriers, such as gabion wall noise barriers, in order to obtain feasible mitigation solutions for both ground-borne vibration and airborne noise.

Ground improvement methods are widely used in civil engineering, mainly to strengthen foundations to support the infrastructure or to avoid ground liquefaction during earthquakes. Such ground improvement methods can also be used as vibration mitigation methods at sites with soft soil. Stiffening of the subgrade under the track improves the bearing capacity of soft soils and helps in avoiding excessive track settlements. Various techniques can be applied to achieve the desired subgrade stiffening, for example, vibro-compaction, jet grouting or excavation and replacement by a new material such as concrete. Such methods can reduce the vertical track settlement and thus decrease the long-wavelength track unevenness that creates low-frequency dynamic vibration effects.

Ground and ballast improvements using geosynthetics such as geogrids that can be placed under the ballast or as a reinforcement material inside the ballast layer can also improve the vibration performance of the track. These methods, that are mainly used for stabilising the track on soft soils and for reducing the ballast degradation, can reduce the track unevenness and therefore reduce the dynamic vibration excitation at low frequencies. Over the past years, geogrids have been used mainly in combination with other ground improvement or vibration mitigation methods, and they are relatively convenient; they are easy to install on new tracks but can also be applied on existing railway tracks during ballast renewal.

### 14.9.5 Controlling Ground-Borne Noise

As ground-borne noise is dominated by higher frequencies than feelable vibration, the main way in which it is controlled is by introducing soft or ‘resilient’ elements in the track to provide some degree of vibration isolation. The principle of vibration isolation is illustrated in Figure 14.38, using a simple mass-spring system. The ratio of the force transmitted to the foundation to that applied to the mass is called the transmissibility. At very low frequency, this ratio is unity; the whole force is transmitted as it would be in the static case. At the natural frequency of the system \(f_n\), the force is increased. Above \(\sqrt{2}\) times the natural frequency, the transmissibility reduces to below unity and continues to decrease with increasing frequency. Here, a hysteretic damping model has been used (constant damping loss factor \(\eta\)) that reflects the behaviour of elastomeric materials. The effect of changes to the damping in the support, the support stiffness and the supported mass is also shown in Figure 14.38.
It can be seen that the amplitude of the resonance is dependent on the damping in the support but that the degree of vibration isolation at higher frequencies is not (for this damping model). When reducing the support stiffness, the natural frequency of the system is reduced and a greater degree of vibration isolation is seen at higher frequencies. Conversely, when reducing the mass of the system, the natural frequency is shifted to higher frequencies. In the railway context, the spring represents the stiffness of resilient elements in the track and the mass represents that of the wheelsets and the parts of the track above the resilient elements.

Vibration isolating designs for tracks are commonplace in modern underground railway systems to reduce ground-borne noise, and the subject is an important part of track design. They can also be used for surface railways; however, the insertion loss achieved will be less than that for a track in tunnel, unless a high impedance foundation, for example, a concrete raft, is introduced beneath the track. In order to isolate the track dynamically, resilient elements can be included at different levels in the track structure. The lower the stiffness of the support, the lower the natural frequency of the system and the greater the degree of vibration isolation at higher frequencies. The choice of support stiffness is, however, limited by the allowable vertical and lateral static displacements under the axle load of the train.

Figure 14.39 shows some of the basic design concepts for vibration isolating track designs, described in more detail later. The rail pad is neglected here; it has a stiffness higher than that of the resilient element in each case but is possibly still significant in the behaviour of the track design for the relevant frequency range. Moreover, the geometric track unevenness is assumed as the main source of vibration-generation mechanism. Adding resilience in the track system may also lead to smoothening of the variations in the support stiffness. In this way, the resilient track systems may lead to a reduction of vibration at relatively low frequencies [126]. An important reduction in perceived unevenness may also occur when the unevenness source is the track bed beneath the resilient element, whereas an increase is possible otherwise [127].

14.9.5.1 Soft Resilient Fastening Systems
The ballast layer forms a resilient component for a conventional ballasted track. For this reason, ballastless tracks (slab tracks) with normal pad stiffness give rise to increased vibration transmission compared with ballasted track. Designs of soft fastening systems are used to rectify this. There are
many designs of soft fastening systems, with the standard test loadings for rails being the limiting factor to which the vertical stiffness of the system can be lowered.

The most common designs are the soft baseplate systems that allow the rail support stiffness to be reduced to about 12 MN/m per fastener. They are mostly used on slab track but can also be installed on top of sleepers in ballasted track. A typical soft baseplate design is the two-stage system consisting of a relatively stiff rail pad between the rail and a metal plate, beneath which a thicker soft elastomeric pad is used. The baseplate is much wider than the rail foot to prevent excessive rail roll and resultant gauge spreading under the lateral forces of the vehicle, particularly during curving.

There are also different systems available for achieving a low vertical stiffness and, at the same time, limiting the lateral displacement of the railhead. These can achieve vertical stiffness of less than 10 MN/m per fastener; however, a potential undesirable effect is that they may lead to an increase in rolling noise if the vibration decay rates are too low.

Figure 14.40 shows the predicted level difference in ground-borne vibration at 16 m from the track due to reductions of the rail fastening stiffness. This is given in decibels, relative to a nominal surface railway model presented in [112], using a railpad stiffness of 350 MN/m and the parameters given in Tables 14.1 and 14.2. The values of the stiffness used for the comparison are selected to represent a soft railpad case (120 MN/m), a standard baseplate system case (30 MN/m), a soft two-stage baseplate system (12 MN/m) and a very soft fastening system with 9 MN/m. It can be seen that the level difference is positive in the range of feelable vibrations (below 80 Hz), with the reduction of the fastening stiffness corresponding to an increase in maximum ground vibration levels. For higher frequencies, vibration reduction is predicted, which should be the largest at the vehicle-track resonance frequency of the original track system.
14.9.5.2 Under-Sleeper Pads, Ballast Mats and Booted Sleepers

Under-sleeper pads and ballast mats can be used for ballasted tracks to lower the stiffness of the ballast layer and therefore the vehicle-track resonance frequency. Booted sleepers perform in the same way as under-sleeper pads, but they are used on slab tracks. Since for all these cases the soft material is installed below the sleeper, the sleeper mass helps to lower the coupled vehicle-track resonance frequency.

Under-sleeper pads are easy to install during a sleeper renewal operation, since they are delivered already fixed to the bottom of the sleeper. Ballast mats can be laid on tunnel invert or a prepared subgrade and have the additional advantage that the extra mass of the ballast is above the spring in the resonant system. However, if a ballast mat is too soft, there is a risk of making the ballast layer unstable under the vibration of passing trains and therefore compromising ride quality and increasing maintenance costs. For the case of the booted sleepers, these are usually bi-block sleepers, and their design and installation are integrated with the slab track.

14.9.5.3 Floating-Slab Tracks

Floating-slab tracks are used to control vibration and ground-borne noise from underground trains. The track is mounted on a concrete slab that rests on rubber bearings, glass fibre or steel springs. With this design, the highest possible mass is added above the track spring to form a system with a very low resonance frequency.

Floating-slab tracks are typically designed as part of the tunnel structure. In addition to the greater construction cost of the track form itself, great expense can come from any increase in the diameter of the tunnel that has to be made to accommodate sufficient mass for the floating slab. The slab may be cast in situ, resulting in a continuous length of concrete, or may be constructed from pre-cast sections. The continuous slab design usually has a lower deflection for a given resonance frequency and makes maximum use of the tunnel space but is harder to design in such a way that the slab mounts can be replaced.
14.10 VIBRATION COMFORT ON TRAINS

14.10.1 INTRODUCTION

The level of vibration in vehicles is a major influence on the perception of the quality of rail travel in comparison with other forms of transport. Vibration in the frequency range from about 0.5–80 Hz causes discomfort as ‘whole body’ vibration, and frequencies below this may cause nausea. The wavelengths in the vertical and lateral profiles of the track that give rise to this vibration are between about 1 m and 70 m, depending on the train speed. Of course, the comfort of passengers is an important reason for the routine monitoring and maintenance that are central to track management for all railways.

14.10.2 ASSESSMENT OF VIBRATION COMFORT IN TRAINS

It is important to understand how measured vibration levels in vehicles are used to assess the likely reaction of passengers. A comprehensive background on this subject is given in reference [128]; here, only an indicative overview is given.

The most commonly accepted principles of vibration perception assessment are laid out in the international standard ISO 2631-1 [95] and also in BS 6841 [129]. These set out terms for consideration of health, comfort, incidence of motion sickness and effects on human activities. Frequency weightings or ‘filters’ are defined that reflect human sensitivity to vibration in a similar way to the A-weighting (Figure 14.1) that is used for sound. Some of these are shown in Figure 14.41. In the assessment of ride comfort, the filter $W_b$ is used in BS 6841 to weight rms vibration in the vertical (spinal) direction for both seated and standing passengers, and filter $W_d$ is used for the two

![Figure 14.41](http://example.com/image.png)

**FIGURE 14.41** Some of the frequency weightings for whole-body vibration defined in BS 6841 [129]. (From Iwnicki, S. (ed.), *Handbook of Railway Vehicle Dynamics*, CRC Press, Boca Raton, FL, 2006. With permission.)
components of lateral vibration. Note that there is a difference between ISO 2631 and BS 6841 in that the ISO standard uses a slightly different weighting for vertical vibration, \( W_k \). However, \( W_b \) is used more in the railway industry, as is recognised in ISO 2631-4 [130]. Vibration in the frequency range 0.5–80 Hz is considered. It is measured, as appropriate, on the seat surface between the cushion and a subject or on the carriage floor. Since measurements on the seat are dependent on the seated person, measurements should be carried out for a sample of subjects. Vibration on the seat back can also be important and is evaluated using other frequency weightings.

When considering the effects of vibration on human activity, weighting \( W_g \) is used for the vertical direction rather than \( W_b \). For assessing the likelihood of vibration to cause motion sickness, weighting \( W_f \) is used for vertical vibration and the lower frequency range of 0.1–0.5 Hz is considered. No guidance is given in the standard on the influence of other components of vibration on motion sickness.

Meters and vibration analysis equipment are available that implement the frequency weighting filters, thereby evaluating the overall weighted levels of vibration. To combine the effects of vibration entering the body at the seat, seat back and the floor in different directions, the root sum of squares of these overall levels can be used.

It is for the rolling stock purchaser to set acceptable limits for the vibration measured in this way according to the type of rolling stock, taking into account factors such as the duration of journeys, number of standing passengers, line geometry standard and vehicle speed. In practice, the standards that are set vary from one railway to another.

There are a small number of single-value indicators of ride quality. One is defined by BS 6841, which allows the measurement of weighted accelerations in 12 components on the seat, seat back and floor. These overall levels are then multiplied by ‘axis multiplying factors’ to give ‘component ride values’, and these may be combined to give an overall ride index.

Another ride quality indicator for ‘average comfort’, \( N_{MV} \), is defined by EN 12299 [131]. This uses overall accelerations in the vertical and two horizontal component directions, weighted as in BS 6841 and ISO 2631, but the 95th percentile values of 60 separate 5-second measurements are taken. The measure therefore becomes sensitive to rare events of high acceleration. \( N_{MV} \) is evaluated as six times the root sum of squares of these values. Values of \( N_{MV} \) are then rated in five bands from ‘very comfortable’ (\( N_{MV} < 1.5 \)) to ‘very uncomfortable’ (\( N_{MV} > 4.5 \)). Although it is suggested that all European railways should adopt this measure of ‘average comfort’, its complexity is a barrier to its acceptance in practice.

### 14.10.3 Effects of Vehicle Design

An important aspect of passenger comfort in railway vehicles is determined by the vibration induced by irregularities of the track. This vibration is transmitted to the carbody through the suspension, and therefore, the design of the suspension plays an important role in controlling this vibration.

#### 14.10.3.1 Bogie Vehicles

At present, almost all rail vehicles are mounted on bogies. This element generally includes two levels of suspension: the primary suspension, between the wheelsets and the bogie frame, and the secondary suspension, between the bogie frame and the carbody. In addition, there are various linkages between the wheelset and the bogie and between the bogie and the carbody; these include the traction mechanisms, which are stiffer than the suspensions, and transmit the forces associated with traction and braking. Unlike in automotive vehicles, in most railway vehicles, the suspensions hardly restrict any degree of freedom between the elements that they join, so that all relative displacements and rotations between carbody, bogie frames and wheelsets are possible.

The bogie has a number of functions. They were originally adopted to facilitate curve negotiation; they also determine the dynamic stability of the vehicle, and the bogie and suspension contribute to ride quality by filtering the vibration transmitted to the carbody from the track irregularities. This filtering effect is required in both the vertical and horizontal planes. However, since
the suspension is normally built by means of passive elements, there are frequency ranges in which the vibration in the carbody is amplified by the suspension, in the same way as the transmissibility shown in Figure 14.38. In a suitable design, the unavoidable amplification associated with resonances should be outside the frequency range in which humans are most sensitive to vibration, that is, about 2–20 Hz, as seen from the $W_b$ and $W_g$ weighting curves in Figure 14.41. Oscillations at very low frequency produce motion sickness, as identified by the $W_f$ weighting. Consequently, the main resonances of the carbody mass on the secondary suspension are generally arranged to be around 1 Hz.

Frequently, the role of the primary suspension is associated with the dynamic stability and curve negotiation, whereas the secondary suspension influences the vibration comfort of the passengers. Typically, the strategy followed is that the bogie and the carbody vibrations are dynamically uncoupled; this makes it possible to separate the comfort and steering tasks. The uncoupling is achieved because the mass of the carbody is very large with respect to that of the bogie frame, while the vertical primary suspension is stiffer than that of the secondary suspension. Consequently, there are carbody vibration modes (in which the carbody moves and the bogies almost do not) and bogie modes (in which the opposite occurs).

### 14.10.3.2 Secondary Suspension

In a conventional rail vehicle, there are five vibration modes that characterise the dynamics of the carbody, which are as follows:

- Vertical oscillation of the carbody
- Pitch rotation
- Yaw rotation
- Two lateral modes that couple the roll rotation with the lateral displacement (see Figure 14.42)

Owing to the uncoupling from the dynamics of the bogie, the natural frequencies of these carbody modes depend on the masses and moments of inertia of the carbody and the stiffness and location of the secondary suspension (they may also depend on the existence of anti-roll and traction mechanisms). If the plane of the secondary suspension contains the carbody centre of mass, the two lateral vibration modes shown in Figure 14.42 become a lateral displacement of the carbody (without rotation) and a pure roll rotation. This can be technically very costly, since it may reduce the space

![FIGURE 14.42](image-url)
available in the carbody, but a similar effect can also be achieved if the secondary suspensions are inclined or by means of anti-roll mechanisms (both strategies are illustrated in Figure 14.43). If roll and lateral displacement are uncoupled, several beneficial effects are achieved, including the improvement of the passengers’ subjective comfort.

As discussed previously, the natural frequencies of the carbody modes should be located in the range between 0.8 Hz and 1.5 Hz. The lowest natural frequency usually corresponds to the vertical mode, and the highest one corresponds to the mode that contains the roll rotation. The lateral mode can potentially have frequencies as low as or lower than the vertical mode, but stiffer lateral suspensions may be required due to gauge limitations in curves and other constraints. If the lateral stiffness of the secondary suspension is very high, the filtering effect is lost, and, moreover, the critical speed for hunting may become too low.

It is convenient if half the longitudinal distance between the two pairs of secondary suspensions coincides with the carbody radius of gyration for pitch motion; this contributes to the reduction of the vibration level. This design is feasible in conventional vehicles, but it is practically impossible in articulated trains or vehicles with Jacobs-type bogies (those in which the bogie is located between two adjacent vehicles).

The fact that the height of the carbody is greater than its width makes its lateral-axis moment of inertia higher than the vertical-axis one, which may prevent stable yaw oscillations of the carbody (that can be coupled with the horizontal dynamics of the bogie). At a certain speed range, the Klingel frequency (frequency of the kinematic oscillation of the bogie) and the carbody horizontal-mode frequency are close; this can produce large oscillations of the carbody if the lateral damping of the secondary suspension is too low.

The secondary suspensions should be as linear as possible in order to facilitate the filtering of the vibration from the bogie. The dampers should be linear and behave symmetrically in the traction and compression directions. If the vehicle is equipped with coil springs for the secondary suspensions, they can have internal modes of vibration at relatively low frequencies (surge frequencies), and, as they are lightly damped, vibration above the cut-off frequency of the secondary suspension can be induced, thus exciting structural modes of the carbody. Air springs can avoid this problem; additionally, they allow the natural frequencies of the carbody to be kept constant independent of the vehicle mass. Owing to their non-linear properties, friction dampers can have a negative effect on the vibration comfort of the vehicle.

Many vehicles travelling at speeds above 120 km/h are equipped with yaw dampers (an example can be seen in Figure 14.43b). Yaw dampers have a non-linear behaviour, with a very high initial
slope in the force-velocity curve, until saturation (which is reached at very low velocity, less than 10 mm/s). If the initial slope is too high, the vibration comfort will be adversely affected, leading to a stick-slip phenomenon in the damper, as found in bogies that have friction yaw dampers.

### 14.10.3.3 Primary Suspension

The stiffness of the primary suspension is generally higher than that of the secondary suspension. The bogies have a similar set of modes of vibration to those of the carbody, but at higher frequencies. The stiffness between the wheelset and the bogie frame must be high to guarantee the running stability of the vehicle and the transmission of the braking/traction forces. In order to produce a cut-off effect, the primary suspension stiffness and damping in the vertical direction must be linear. The required longitudinal stiffness, which is much higher than the lateral one, can hardly be achieved by means of springs; usually, rubber springs and mechanisms (bars with bushings at the ends) are adopted. The traction and braking forces between the bogie frame and the carbody are transmitted through a mechanism or traction lever. The traction lever produces a high stiffness between the carbody and bogie frame in the longitudinal direction.

The natural frequencies of the bogie frame on the primary suspension are typically around 10 Hz. As this is much higher than the cut-off frequency of the secondary suspension, the dynamics of the bogie and the carbody are uncoupled, and the largest displacements of the bogie at its resonances are not transmitted to the carbody.

### 14.10.3.4 Traction Mechanisms

The traction mechanisms can affect the modes and the transmission of forces from the bogie to the carbody. Some examples are assessed below.

The sketch in Figure 14.44 shows a bogie that has radial arm primary suspensions; an example of this kind of suspension can be also seen in Figure 14.43b. The arms and the bogie frame can be considered as a four-bar mechanism, and the centre of rotation of the bogie frame is at the intersection of the arm lines (which are drawn in the figure in dashed lines). The position of the centre of rotation of the pitch motion should be at the location of the traction lever; otherwise, the pitch mode of vibration could transmit forces to the carbody through the traction lever.

The design in Figure 14.45 represents a bogie that has a Watt mechanism at each primary suspension. The Watt four-bar mechanism is formed by the two rods that have the same lengths and the axlebox. The wheelset axle must be in the middle between the joints of the axlebox. By considering the relative motion with respect to the bogie frame, the displacement of the centre of the axlebox is approximately a vertical straight line (the precise curve is a lemniscata). Consequently, the high-frequency vibration of the axle box centre of mass does not produce any horizontal displacement of the wheelset that may generate horizontal inertial forces that would be transmitted to the bogie frame through the rods. On the other hand, the pitch mode shape of this bogie consists of rotation around the centre of mass. This is only true if the traction lever is in the centre of mass of the bogie frame. If the centre of mass and the traction lever do not match, then the pitch mode produces horizontal forces that can be transmitted through the traction lever.

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**FIGURE 14.44** Bogie with arm linkage.
Carbody Structure

Finally, the vibration level inside the vehicle body is also affected by its low-order structural resonances. Typically, both vertical and lateral first-order bending modes occur at frequencies around 10 Hz, and resonances of the floor occur at frequencies just above this. The excited amplitude of these should be kept to a minimum by structural design of the carbody, avoiding the coincidence of important modes and using damping treatments. The carbody structural modes should be kept above about 10 Hz to avoid the frequency range to which humans are most sensitive. This is one reason for the trend towards light, stiff materials such as aluminium extrusions in the manufacture of rolling stock.

An additional trend is the introduction of vibration-isolated ‘walking’ floors in passenger coaches. This is primarily aimed at reducing vibration in the audible frequency range that is important for the interior noise environment (Section 14.8) but can be effective in reducing vibration above about 20 Hz [132].

The carbody vibration, in all three directions, is felt by passengers through the seat and the seat back. Seat dynamics must also therefore be considered. The coupled system of seat and human body exhibits a vertical resonance typically between 4 Hz and 6 Hz and at a similar frequency for fore-aft vibration due to the stiffness of the backrest. With the very soft seats used on some old rolling stock, these resonances can cause the vibration at the floor level to be made worse for the passenger, rather than better, by the soft seat.

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