# CHAPTER 4 WHEEL/RAIL NOISE GENERATION

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# CHAPTER 4

# WHEEL/RAIL NOISE GENERATION

## 4.1 INTRODUCTION

This chapter discusses the basic theory of wheel/rail noise generation for tangent track, curved track, and special trackwork to provide a foundation for making informed decisions concerning treatment. The mechanisms involved are exceedingly complex, and the reader is referred to the literature for more detailed discussions than those provided here.

There have been substantial advances in the theories concerning wheel/rail noise generation during the 1970s and 1980s, and research in the area is continuing, especially with respect to high speed rail. The theory of wheel/rail noise generation, though highly developed, is incomplete. The effect of wheel and rail profiles on wayside noise from tangent track is being studied, but contradictions exist. Wheel squeal theory is also subject to some inconsistency with respect to field experience. In general, however, this fascinating and challenging field of noise control is rich with topics for theoretical and experimental work.

There are several descriptive terms for various types of wheel/rail noise. The terms *rolling noise* and *tangent track noise* are both used here to refer to noise produced by rail and wheel roughness and material heterogeneity. Rolling noise also occurs at curved as well as tangent track. The term *impact noise* refers to noise generated by rail imperfections, joints, and more significantly, special trackwork. Much of tangent track rolling noise at severely worn rail or with flatted wheels may include substantial impact noise due to wheel/rail contact separation. The terms *wheel squeal* and *curving noise* both refer to the noise generated at curves. Curving noise includes both wheel squeal and *wheel/rail howl*, the latter being the less common. While wheel squeal is due to a stick slip phenomenon involving nonlinear interaction of the wheel with the rail, wheel/rail howl, also common at curves, is less clearly defined.

Tangent track rolling noise is considered first, followed by a discussion on curving noise. Finally, the researchers conclude with a short discussion of special trackwork noise.

## 4.2 TANGENT TRACK NOISE

Tangent track noise is due to wheel and rail roughness, which may include rail corrugation and random surface defects. Tangent track noise is subdivided into (*1*) normal rolling noise for smooth ground rail and trued wheels; (*2*) rolling noise due to excessively roughened rails and wheels; (*3*) impact noise due to pits and spalls of the rail and wheel running surfaces, wheel flats, and rail joints; and (*4*) rail corrugation noise. These categories form a basis for discussion of tangent track noise and selection of noise control procedures. The term *wheel/rail roar* is often used to refer to rolling noise of all types, while *roaring rail* has been used to refer to corrugated rail noise.

Tangent track normal and excessive rolling noise are discussed within the context of a linear theory of wheel/rail interaction, relying on rail and wheel roughness as the primary cause. Another cause of wheel/rail noise may be heterogeneity of the rail steel moduli, for which the linear theory presented here may not be entirely adequate. Further, with sufficiently large amplitude roughness, as with severely corrugated rail, or with severely flatted wheels, wheel/rail contact separation may occur, producing impact noise. A separate theory of impact noise is presented after the discussions of normal and excessive noise, and before the discussion of rail corrugation noise.

### 4.2.1 Normal Rolling Noise

Noise in the absence of wheel and rail imperfections or discontinuities, such as wheel flats, spalls, rail corrugation, joints, and so on is termed *normal rolling noise*. Normal rolling noise is the baseline condition for which most new rail transit systems are designed, and departures therefrom constitute a degradation of condition and performance with respect to noise control. This baseline condition is the state that operating rail transit systems should be striving for or preserving. There is also an important side benefit associated with a quiet system: smooth rolling surfaces produce less vibration of various trackwork and truck components, thus extending life and, possibly, reducing operational costs.

#### *4.2.1.1 Characterization of Normal Rolling Noise*

Tangent track rolling noise has a broadband frequency spectrum, often with a broad peak between 250 and 2,000 Hz. At systems with adequate ground rail and trued wheels, traction motor cooling fan and gearbox noise and undercar aerodynamic noise may contribute significantly to operational noise, and some care must be exercised in identifying the wheel/rail noise component. Representative samples of 1/3-octave band wayside noise produced by a Bay Area Rapid Transit (BART) vehicle at various speeds are presented in Figure 4–1. In this example, traction power equipment produces the peak in the spectrum at 250 Hz. At higher frequencies, wheel/rail noise dominates the spectrum, peaking at about 1,600 Hz. With trued wheels and smooth ground rail on ballast and ties, BART is one of the quietest vehicles in operation at U.S. transit systems.

A whine may also occur because of a periodic grinding pattern in the rail running surface, as evidenced by the peak at about 1,600 Hz in the 80 mph data shown in Figure 4–1. At lower speeds, this peak is reduced in frequency, consistent with a wavelength in the rail of about ¾ in. long. Without this component, the passby noise would have been less than measured by a few decibels.

The high levels of low-frequency noise below 125 Hz are likely due to aerodynamic sources in view of the relatively large difference between the 60 and 80 mph data in this frequency range. If there is difficulty in distinguishing the wheel/rail noise component from traction power equipment noise and other under car equipment noise, then wheel/rail noise is probably not excessive, and efforts at noise control might be directed toward traction power equipment (*1*)

With normal rolling noise, the rail running surface will be free of spalls, checks, pitting, burns, corrugation, or other surface defects, which may not be entirely visible. The wheel and rail provide running surfaces which, under ideal conditions, should have similar characteristics for smoothness and low noise as any anti-friction bearing. From a practical perspective, ideal bearing surfaces are difficult to realize and maintain in track due to lack of lubricant, corrosion and contamination, and dynamic wheel/rail interaction forces.

### *4.2.1.2 Rolling Noise Generating Mechanisms*

Four generating mechanisms have been suggested in the literature as sources of normal rolling noise. These include

- Rail and wheel roughness,
- Parameter variation, or moduli heterogeneity,
- Creep, and
- Aerodynamic noise.

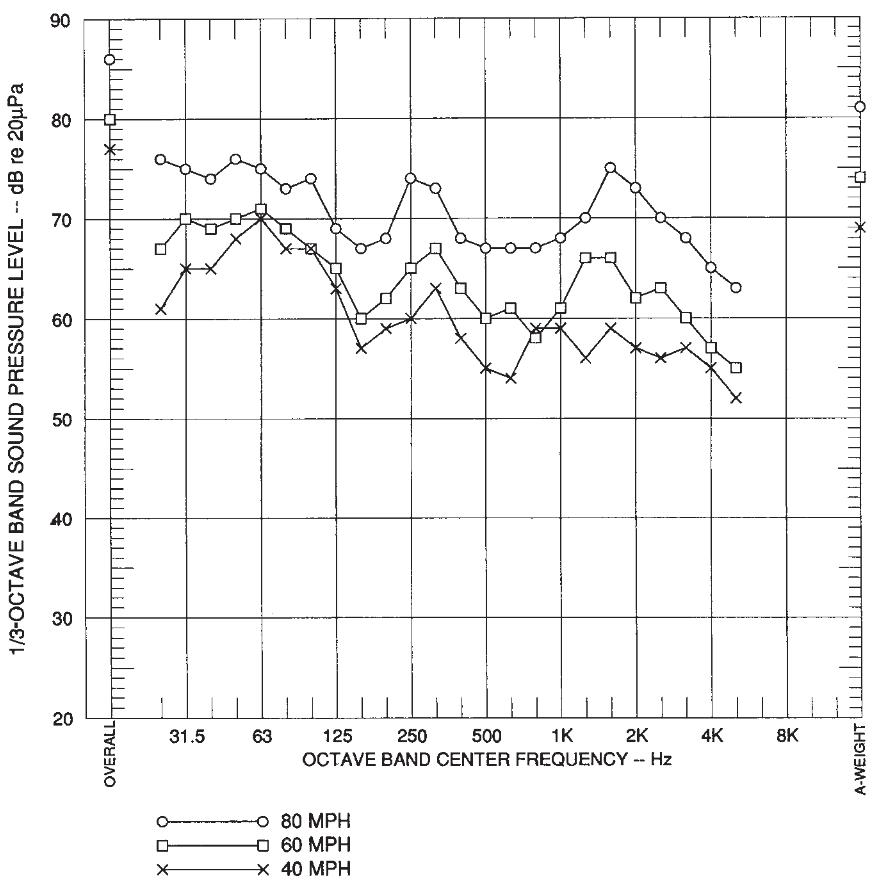



**FIGURE 4-1 PASSBY NOISE FOR SINGLE BART VEHICLE WITH ALUMINUM CENTERED WHEELS ON BALLAST AND TIE TRACK**

*Wheel/Rail Roughness.* Wheel and rail surface roughness is believed to be the most significant cause of wheel/rail noise. The surface roughness profile may be decomposed into a continuous spectrum of wavelengths. At wavelengths short relative to the contact patch dimension, the surface roughness is attenuated by averaging of the roughness across the contact patch, an effect which is described as *contact patch filtering*. Thus, fine regular grinding marks of dimensions less than, perhaps, 1⁄16 in., should not produce significant noise compared to lower frequency components.

*Parameter Variation.* Parameter variation refers to the variation of rail and wheel steel moduli, rail support stiffness, and contact stiffness due to variation in rail head transverse radius-of-curvature. The influence of fractional changes in elastic moduli and of radius-of-curvature of the rail head as a function of wavelength necessary to generate wheel/rail noise equivalent to that generated by surface roughness is illustrated in Figure 4–2. The wavelength of greatest interest is 1 to 2 in., corresponding to a frequency of about 500 to 1,000 Hz for a vehicle speed of about 60 mph. Over this range, a variation in modulus of 3% to 10% is required to produce the same noise due to rail roughness. Experimental data for the effect of modulus variation at this frequency have not yet been found. Rail head ball radius heterogeneity also induces a dynamic response in the wheel and rail. The variation of rail head curvature would have to be on the order of 10% to 50% to produce a noise level similar to that produced by rail roughness alone. Data on rail head radii of curvature as a function of wavelength have not been obtained nor correlated with wayside noise. Also, rail head ball radius variation will normally accompany surface roughness, so that distinguishing between ball radius variation and roughness may be difficult in practice.

*Dynamic Creep.* Dynamic creep may include both *longitudinal* and *lateral dynamic creep*, *roll-slip* in a direction parallel with the rail, and *spin-creep* of the wheel about a vertical axis normal to the wheel/rail contact area.

***Longitudinal creep*** is not considered significant by some researchers, as rolling noise levels are claimed to not increase significantly during braking or acceleration on

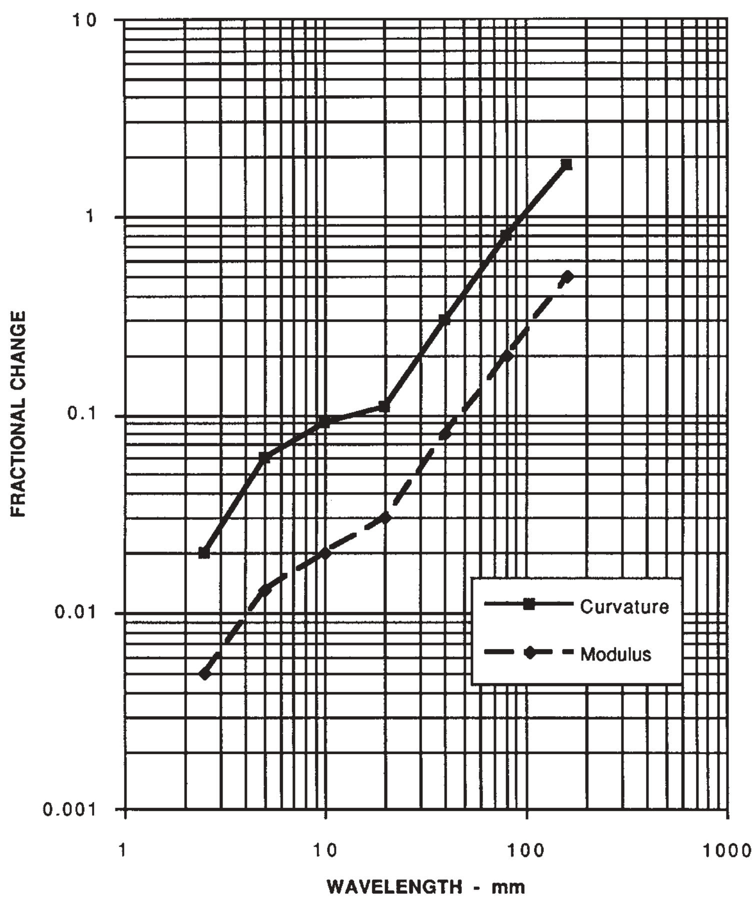



**FIGURE 4-2 CHANGE IN ELASTIC MODULUS AND RAIL HEAD CURVATURE REQUIRED TO GENERATE WHEEL/RAIL EXCITATION EQUIVALENT TO ROUGHNESS EXCITATION** *(5)*

smooth ground rail. However, qualitative changes of the sound of wheel/rail noise on newly ground rail with a grinding pattern in the rail running surface is observable to the ear as a train accelerates or decelerates, in contradiction to the notion that longitudinal creep is of no significance.

***Lateral creep*** occurs during curve negotiation, and is responsible for the well-known wheel squeal phenomena resulting from stick-slip. Lateral creep may not be significant at tangent track, but lateral dynamic creep may occur during unloading cycles at high frequencies on abnormally rough or corrugated rail. Lateral dynamic creep is postulated by some to be responsible for short-pitch corrugation at tangent track. Therefore, lateral creep, at least in the broad sense, may be a significant source of noise.

***Spin-creep*** is caused by wheel taper which produces a rolling radius differential between the field and gauge sides of the contact patch. Spin-creep has been suggested as a source of wayside noise at the Vancouver Skytrain (*2*), though, no data supporting a strong dependence of noise level on spin-creep have been obtained.

*Aerodynamic Noise.* Aerodynamic noise is caused by turbulent boundary layer noise about the wheel circumference as it moves forward and by undercar components which exhibit substantial *aerodynamic roughness*. Noise due to air turbulence about the wheel is usually not significant at train speeds representative of transit systems, while noise due to air turbulence in the truck area may be significant. Another interesting possible source is high-velocity jet noise emanating from between the tire and rail running surface, which has been anecdotally suggested as responsible for abnormally high noise levels after rail grinding at Tri-Met, where a grinding pattern was evidently ground into the surface of the rail. However, no measurements have been performed and no theoretical models have been proposed in support of this conjecture.

*4.2.1.3 Parallel Mechanical Impedance Model*

The standard linear model (*3*) of wheel/rail rolling noise generation is described in Figure 4–3. The model is a *parallel impedance model* because the vertical mechanical impedance of the rail and wheel appear in parallel with each other, and, together, with the *contact stiffness*, $K_c$ , determine the dynamic interaction forces between the rail and wheel in response to rail and wheel tread roughness. The mechanical impedance is the ratio of contact force to response velocity expressed as a function of frequency.

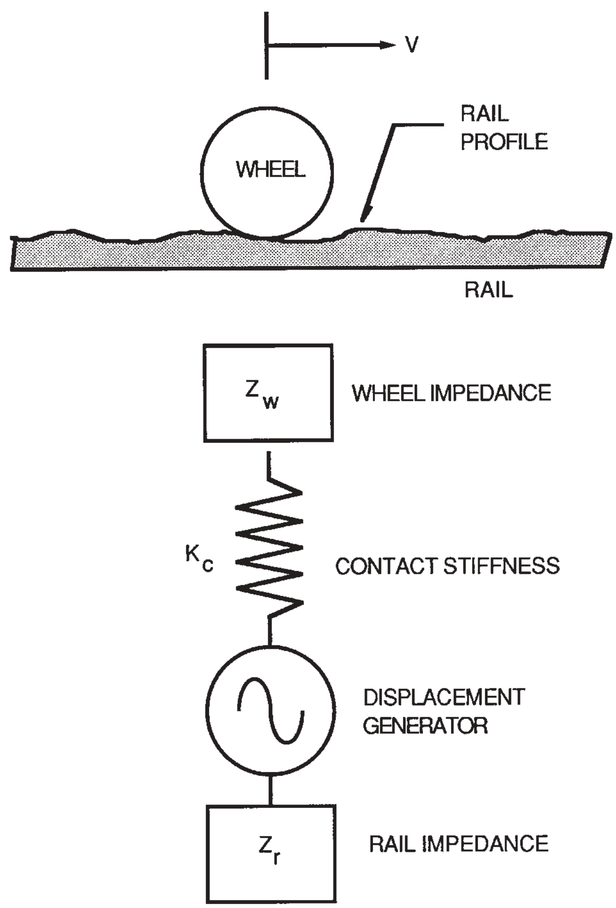



**FIGURE 4-3 PARALLEL IMPEDANCE MODEL OF WHEEL RAIL NOISE GENERATION**

The roughness is indicated as a displacement generator in series with the contact stiffness. The customary approach is to decompose the rail and wheel roughness amplitude into a spectrum of amplitude versus roughness wavenumber, $k$, equal to $2\pi$ divided by the wavelength, $\lambda$. The corresponding frequency, $f$, of the sound due to roughness is related to vehicle speed, $V$, and roughness wavenumber as

$$f = kV = 2\pi/\lambda$$

Again, the roughness may be decomposed into an infinity of waves with associated wavelengths, $\lambda$. The spectra may consist of discrete wave components, such as with corrugated rail, or may have a continuous spectrum of wave components, as with random roughness.

The model includes the various resonances of the wheel set and track support in the expressions for wheel and rail mechanical impedances. Given the dynamic contact force, the vibration of the wheel may be predicted from the transfer mobilities, or modal velocity responses, of the wheel to contact forces. A modal response is the vibration response of an object at a natural vibration mode, and is associated with a modal, or natural, frequency.

The contact force can be expressed as a function of the parallel combination of the mechanical impedances for the wheel and rail, and the roughness. Thus

$$F(\omega) = \frac{i\omega\delta(\omega)}{\dfrac{1}{Z_c(\omega)} + \dfrac{1}{Z_r(\omega)} + \dfrac{1}{Z_w(\omega)}} = i\omega\delta(\omega)Z(\omega)$$

where $i$ is the square root of $-1$, $\omega$ is the radian frequency, $2\pi f$, and $Z(\omega)$ is the frequency dependent parallel combination of the impedances.

The contact mechanical impedance, $Z_c(\omega)$, is given by

$$Z_c(\omega) = \frac{K_c}{i\omega}$$

The velocity of the wheel tread at the contact patch, $v_t(\omega)$, and rail head, $v_r(\omega)$, neglecting deformation of the contact patch, are given by

$$v_w(\omega) = \frac{F(\omega)}{Z_w(\omega)} = \frac{Z(\omega)i\omega\delta(\omega)}{Z_w(\omega)}$$

$$v_r(\omega) = \frac{F(\omega)}{Z_r(\omega)} = \frac{i\omega Z(\omega)\delta(\omega)}{Z_r(\omega)}$$

The entire discussion can be reformulated in terms of compliances, $C(\omega)$, through the relation

$$C(\omega) = \frac{1}{i\omega Z(\omega)}$$

In this case, the expressions for the contact force simplify to

$$F(\omega) = \frac{\delta(\omega)}{C_r(\omega) + C_w(\omega) + C_c}$$

The contact compliance, $C_c$, is assumed to be independent of frequency. The rail and wheel displacements in response to the contact force become

$$\delta_r(\omega) = C_r(\omega)F(\omega)$$

$$\delta_w(\omega) = C_w(\omega)F(\omega)$$

The mechanical impedance functions for the wheel and rail can be rather complicated in the audio range. (See below.) However, the above forms indicate that the ratios of the wheel and rail compliances to the parallel compliance determine the responses of the wheel and rail, respectively. In particular, if the contact stiffness is very large, the parallel impedance becomes the parallel impedance of the wheel and rail, with the contact patch stiffness contributing insignificantly to the interaction. This is the usual condition at frequencies below, perhaps, 100 Hz. If the contact stiffness is sufficiently small, the parallel combination will be controlled by the contact compliance, in which case both the wheel and rail velocities, or displacements, will decrease. If the wheel compliance is high with respect to both the contact and rail compliances, the wheel displacement will approach the combined roughness, $\delta(\omega)$, of the rail and wheel, and the rail displacement will approach zero. Conversely, if the rail compliance becomes very large relative to the wheel compliance, the rail displacement will approach $\delta(\omega)$, and the wheel displacement will decrease.

For normal running noise, where the wheels and rails are in good condition, the above model is expected to be a reasonable representation of the physics of wheel/rail interaction. The model ignores possible contact separation due to excessive roughness or corrugation, which would result in a nonlinear set of equations. Also not included in the model are lateral forces and responses, though these may be included (*4*). There are no expressions or information concerning lateral rail roughness, and lateral roughness is therefore ignored, though this may not be clearly justified with conical wheels and canted rails. The model also ignores heterogeneity of the rail steel moduli, and other parametric variations such as discrete rail supports with respect to the translating vehicle, and variation of the ball radius of the rail head, which will modulate the contact stiffness. Roll-slip- or spin-creep-generated noise is similarly not included.

Current models of wheel/rail interaction are at a considerably higher state of development than that given above. These more complex models, including the British Rail model of wheel/rail noise generation (TWINS), are discussed by Remington (*5*) and Thompson (*6*). A very thorough discussion of high-frequency wheel/rail interaction models is provided by Knothe and Grassie (*7*).

*4.2.1.4 Wheel/Rail Roughness*

Examples of measured rail roughness spectra are provided in Figure 4–4. Wheel roughness is equally important, and the roughness, $\delta(\omega)$, appearing in the parallel impedance model should be taken as the square root of the sum of the squares of the two roughnesses. The roughness spectra provided in Figure 4–4 are in terms of 1/3-octave bands of roughness wavenumber, $k = 2\pi/\lambda$, where $\lambda$ is the wavelength in in. The 1/3 octave band displacement, or roughness, levels are in decibels relative to 1 micro-in. Thus, if one were to "filter" the rail roughness by a 1/3-octave band filter centered at 16 radians per inch, the observed root-mean-square output of the filter would be 24 to 28 dB re 1 micro-in., or about 15 to 25 micro-in. The actual frequency of excitation is obtained by multiplying the wavenumber by the train speed in inches per second and dividing by $2\pi$. Thus, the wavenumber of 16 radians per inch (wavelength equal to 0.393 in.) at a train speed of 60 mph (1,056 in./sec) is 2,690 Hz. These roughness spectra are only representative, and should not be applied to all systems. Actual roughness spectra are expected to vary considerably depending on rail condition, wear, corrugation, and so on.

Employing quantitative estimates of wheel and rail roughness directly in noise prediction and identification of noise control treatments is usually not practical, because of the complexities of the modal responses of the wheel and to a lesser extent the rail, both of which may allow only an approximate estimate of wheel/rail noise. A more useful quantitative approach is to measure wayside or interior noise, and then rely on the model to estimate the qualitative effect of changes in the parameters of the system. Matching roughness spectra to observed wayside noise using an assumed model for noise generation is a practical method of parameter evaluation.

*4.2.1.5 Wheel Mechanical Impedance*

The calculated radial mechanical impedance of the tread of a resilient wheel is illustrated in Figure 4–5, based on a theoretical formula developed to study the State of the Art Car (SOAC) (*8*). The anti-resonance of the wheel, indicated by a maximum in the mechanical impedance at about 800 Hz, can be correlated with a peak in the wayside noise spectrum illustrated in Figure 4–6. The wayside noise is of a light

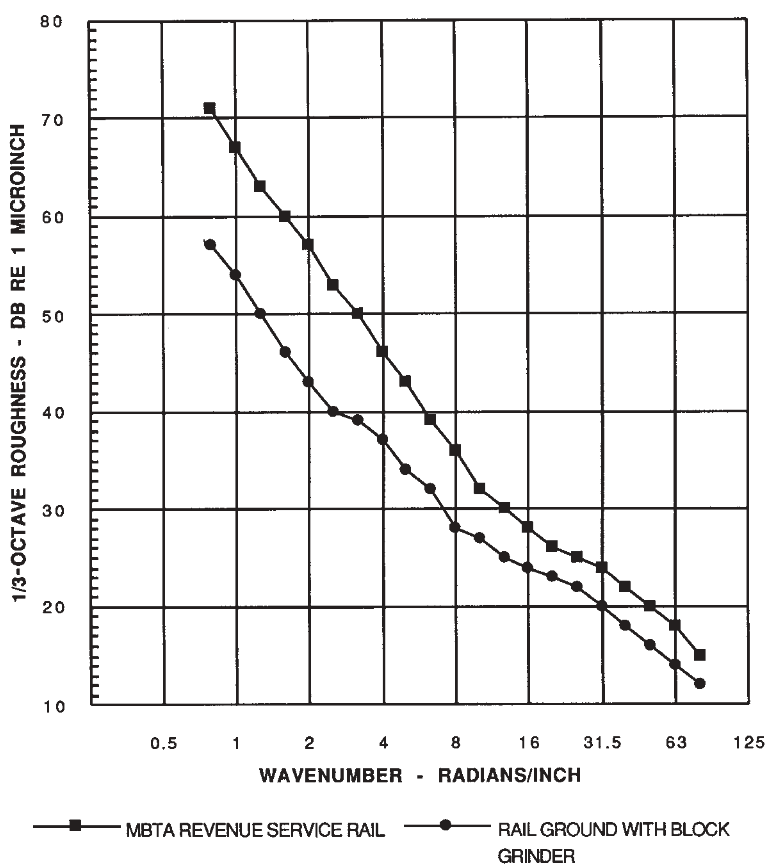



**FIGURE 4-4 RAIL ROUGHNESS SPECTRA** (*5*)

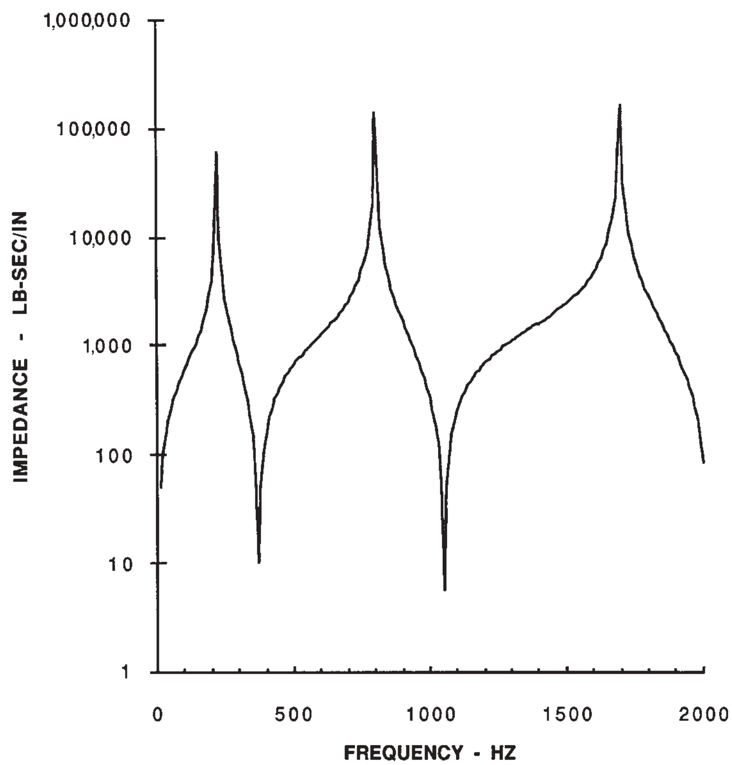



**FIGURE 4-5 RADIAL POINT MECHANICAL IMPEDANCE OF WHEEL TREAD** (*5*)

rail articulated vehicle on direct fixation track traveling at about 45 mph at a range of about 150 ft. The sample indicates a dominant peak at about 800 Hz, which contributes to the detectability of train noise in the community. Other mechanical anti-resonances exist at about 200 Hz and at about 1,700 Hz. In general, the mechanical impedance of the wheel is quite complex, and appears to have a strong effect on wayside noise.

*4.2.1.6 Rail Mechanical Impedance*

The theoretical input mechanical impedance of a 119-lb/yd rail with 150,000 lb/in. stiffness fasteners at 36-in. pitch is illustrated in Figure 4–7 depicting a point on rail head between two adjacent fasteners and another point over a fastener. The fasteners are assumed to have a top plate weight of 14.6 lb, including clips, and the rail is represented by a Bernoulli-Euler beam. The calculation was developed specifically for this manual, and includes the effect of discrete rail supports. The impedance level in dB is 20 Log10 (force/velocity in lb-sec/in.). Below 50 Hz, the two impedance functions correspond with that of a spring, while the effect of the rail mass resonating on the fastener stiffness appears between about 100 and 200 Hz, where the impedance approaches a shallow minimum. The rail-on-fastener resonance frequency is calculated to be 110 Hz. However, at higher frequencies, the impedance functions become quite complex. The input mechanical impedance at the rail head between the fasteners exhibits a sharp dip at about 600 Hz due to the pinned-pinned mode resonance of the rail on its discrete supports. The input mechanical impedance directly over a fastener, however, exhibits a poorly developed antiresonant behavior, with rather complicated shape, at the pinned-pinned mode. Both impedance functions exhibit more complicated behavior at higher frequencies. The actual pinned-pinned mode frequency is calculated more accurately by including transverse shear and rotary inertia in the beam equations. (See Chapter 10 for further discussion concerning rail corrugation.)

The 800 Hz component shown in Figure 4–6 can also be correlated with a pinned-pinned mode resonance of the rail at about 700 to 800 Hz, calculated with Timoshenko beam theory for the rail which includes rotary inertia and transverse shear. The calculation is for 115-lb/yd rail with 30-in. fastener spacing. Whether or not the pinned-pinned mode is responsible for the peak in the wayside noise spectrum has not been determined. Another track resonance that may occur in the neighborhood of these frequencies is bending of the fastener top plate. Again, the significance of fastener top plate bending on rail radiated wayside noise has not been

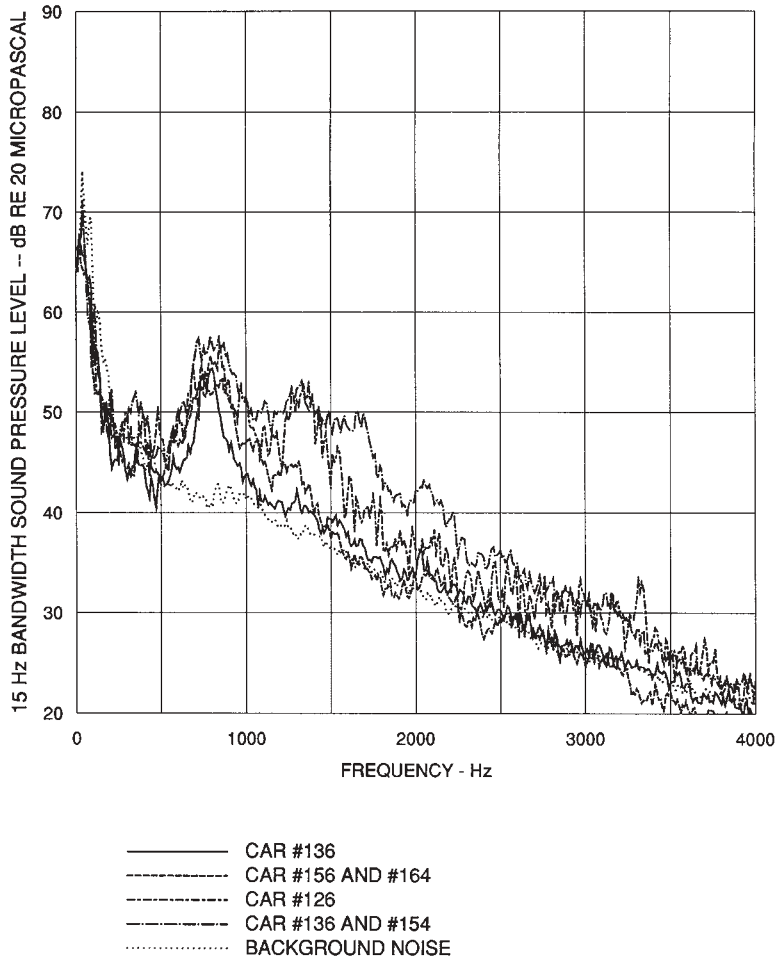



**FIGURE 4-6 WAYSIDE NOISE SPECTRA FOR EASTBOUND TRAINS AT 45 MPH ON NEAR TRACK BY 142ND AVENUE**

determined. (See the discussion concerning top plate bending in Chapter 10.)

Bending waves will propagate in the rail up to a frequency corresponding to twice the wavelength of the pinned-pinned mode wavelength. Between this wavelength and the pinned-pinned mode wavelength, vibration transmission along the rail may be inhibited, depending on the rail support dynamic characteristics, producing what is termed as the "stop band." Above the pinned-pinned mode frequency, up to another cut-off frequency, bending waves may propagate freely, resulting in a "pass band." The response of the rail and its ability to radiate noise will be affected by the widths of the stop band. A slight randomness in the support separation may significantly alter the pass and stop band characteristics (*9*).

One may conclude that at the dominant frequencies between 250 and 1,000 Hz associated with wheel/rail rolling noise there are extensive possibilities for interaction between the wheel and rail, and that these interactions must be considered when dealing with noise control and rail corrugation.

*4.2.1.7 Contact Stiffness*

Contact stiffness is calculated on the basis of Hertzian contact theory, the results of which are presented in Figure 4–8 for a 15-in. radius wheel. The contact stiffness does not vary greatly over the range of rail head ball radii, though reducing the ball radius to 6 in. from about 15 in. would appear to reduce the contact stiffness by about 16%, which, under the most optimistic scenario, would reduce contact forces by at most 1.5 dB. Note, however, that contact stresses may be increased as a result of lessened contact area. Optimizing the contact stiffness would not appear to produce a significant noise reduction, while setting the ball radius to be comparable with that of the wheel (15 in.) would "round out" the contact patch, possibly leading to lower wear rates, and, indirectly, lower noise. Typical practice, however, is to employ a ball radius of about 7 to 10 in. As discussed below, wheel tread concavity due to wear increases the lateral contact patch dimension, with the result that the contact width may be comparable with that of a less worn wheel tread on, for

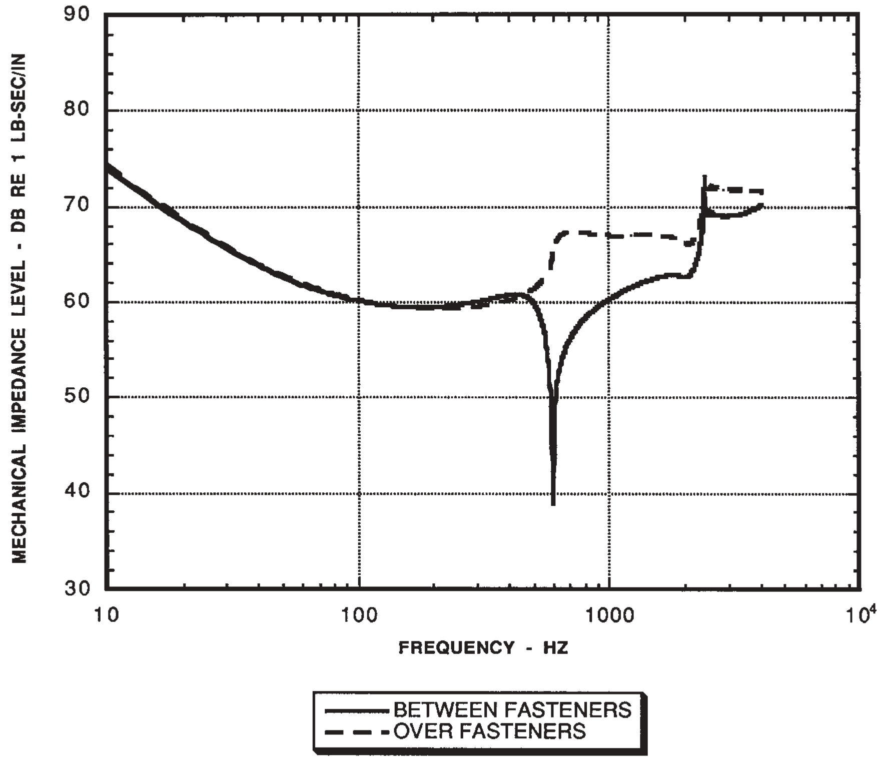



**FIGURE 4-7 RAIL INPUT MECHANICAL IMPEDANCE FOR 119 LB/YARD RAIL FASTENER SPACING OF 36 INCHES AND FASTENER STIFFNESS OF 150,000 LB/IN**

example, rail with ball radius 14 in. Although the rail head may be optimized for a particular ball radius, wheel tread wear may frustrate maintaining a specific contact width unless a vigorous wheel truing program is in place.

### *4.2.1.8 Wheel/Rail Conformity*

Increasing conformity of the wheel and rail has been proposed by Remington (*10*) as a noise reduction technique to take advantage of uncorrelated roughnesses between various parallel paths along the rail in the longitudinal direction. Significant noise reductions on the order of 3 to 5 dB are predicted for frequencies on the order of 500 Hz. However, increased wheel/rail conformity has been identified as a cause of spin-creep corrugation at the Vancouver Skytrain (*2*), leading to noise. Care should be exercised before adopting high conformity wheel and rail profiles. High wheel/rail conformity results from normal wheel tread wear if wheel truing is not frequently conducted.

### *4.2.1.9 Noise Radiation*

Measurement data suggest that both the wheel and rail are significant sources of noise, and both must be considered in developing noise control solutions. The acoustic power radiated by a wheel can be approximated by the area of the wheel multiplied by the rms velocity of the wheel surface in a direction normal to the surface, and the specific impedance of air, $\rho c$, where $\rho$ (1.2 kg/m$^3$) is the air density and $c$ is the velocity of propagation of sound (340 m/sec). Other dimensions are given in SI units as well. Sound radiation by the rail is usually modeled by assuming that the rail is a cylinder of diameter equal to the height of the rail. Noise radiation partitioning between the rail and wheel is difficult to quantify accurately because of the closely coupled nature of the wheel and rail and their proximity to one another. Even the ties can be considered significant noise radiators. For continuous smooth ground rail, much of the theoretical literature suggests that the rail is the most significant radiator of noise (*5*). Numerous experimental data suggest, however, that wheel-radiated noise is of similar significance as that of the rail (*11,12*).

Noise radiation from the wheel and rail is described in terms of radiation efficiencies. For a large radiating surface, with dimensions large relative to the wavelength of sound, the radiation efficiency is unity. For a wheel, the radiation efficiency decreases with increasing wavelength above the diameter of the wheel, and, at long wavelengths relative to the diameter of the wheel, the radiation efficiency decreases very rapidly with increasing wavelength. The radiation efficiency for the rail is affected by both the velocity of bending waves in the rail and by the diameter of the rail. The radia-

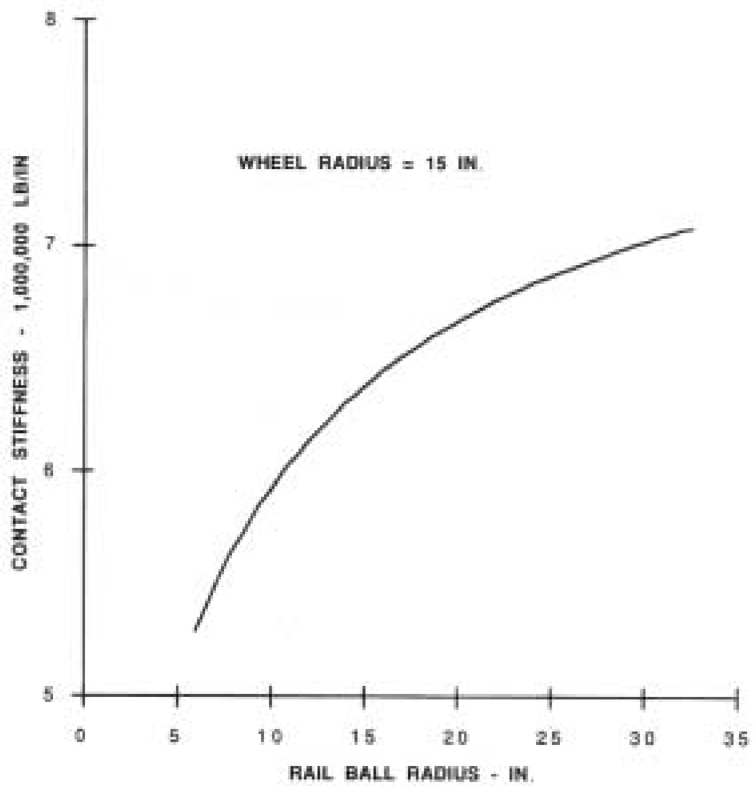



FIGURE 4-8 VARIATION OF CONTACT STIFFNESS WITH RAIL HEAD BALL RADIUS

tion efficiency becomes negligible if the bending wave velocity in the rail is less than the acoustic velocity. For the present purposes, this effect is ignored. The radiation efficiency of a rail may be modeled by a cylinder of diameter equal to the width of the rail. An example of the dependence of the radiation efficiency on frequency for a 6-in.-diameter cylinder in rigid body motion transverse to the rail center is presented in Figure 4–9. This model indicates that the radiation efficiency for a typical rail may be assumed to be unity at frequencies greater than about 1,000 Hz. At lower frequencies, the wavelength in air is long with respect to the diameter of the rail, leading to a declining radiation efficiency with decreasing frequency. However, noise radiation by the rail is still significant at 500 Hz. For very low frequencies, the radiation efficiency is declining as the cube of the frequency and as the fourth power of the rail diameter.

#### *4.2.1.10 Directivity*

Peters (*12*) has determined that the radiation pattern for sound radiated by rail vehicles is primarily that of a distribution of dipole radiators. For a dipole radiator, the sound intensity across a unit of area normal to the direction of propagation varies as the square of the cosine of the angle between the direction of maximum radiation and the direction of radiation (*13*). An example of the angular dependence of sound radiation from a dipole source is provided in Figure 4–10. Dipole radiation is consistent with noise radiation from a wheel, and, for this reason, Peters conjectures that the wheel is the dominant radiator of noise. However, much of the noise radiated by the rail may also be radiated in a dipolar fashion.

An example of the passby signature of a 5-car BART transit train is compared in Figure 4–11 with the predicted passby signature based on a dipole radiation pattern (*14*). The agreement between predicted and measured signature shape is remarkable, indicating that the dipole model is probably most appropriate, though slightly underconservative. The normalized passby signature of a single Portland Tri-Met LRV traveling on ballast-and-tie track at about 35 mph is compared in Figure 4–12 with predictions based on monopole and dipole radiation patterns. In this case, the passby signature falls midway on a decibel scale between that predicted for monopole and dipole sources. The Tri-Met vehicle has Bochum resilient wheels which may affect the radiation of noise. In both cases the dipole model underpredicts the shoulders of the passby signature. From a practical standpoint, the discrepancy is not particularly important for long trains. The data shown for the Tri-Met vehicle are for a single vehicle with three trucks, and may not be representative of the fleet or other light rail vehicles.

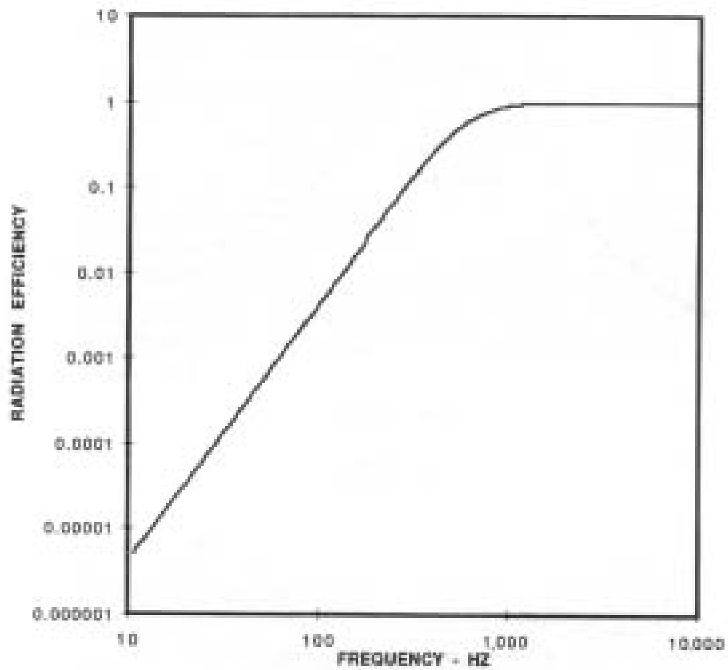



**FIGURE 4-9 RADIATION EFFICIENCY OF 6-INCH-DIAMETER CYLINDER IN RIGID BODY TRANSVERSE MOTION**

The dipole radiation pattern, or directivity factor, has an effect on computation of energy-averaged sound levels, or noise exposure levels. Maximum sound levels occurring during train passage are not strongly affected, though differences on the order of 1 dB at distances beyond one train length from the track may exist between the dipole and monopole model. The effect on attenuation with distance is not large, but there may be a significant effect when basing calculations on sound power radiated by the wheels and rails. Finally, if traction power equipment or air conditioning noise is included, the monopole radiation model may become more significant. Assuming a monopole source characteristic will result in a conservative estimate of energy-averaged sound levels such as the Equivalent Sound Level ($L_{eq}$) or Day-Night Sound Level ($L_{dn}$), though the margin is limited to 1 dB.

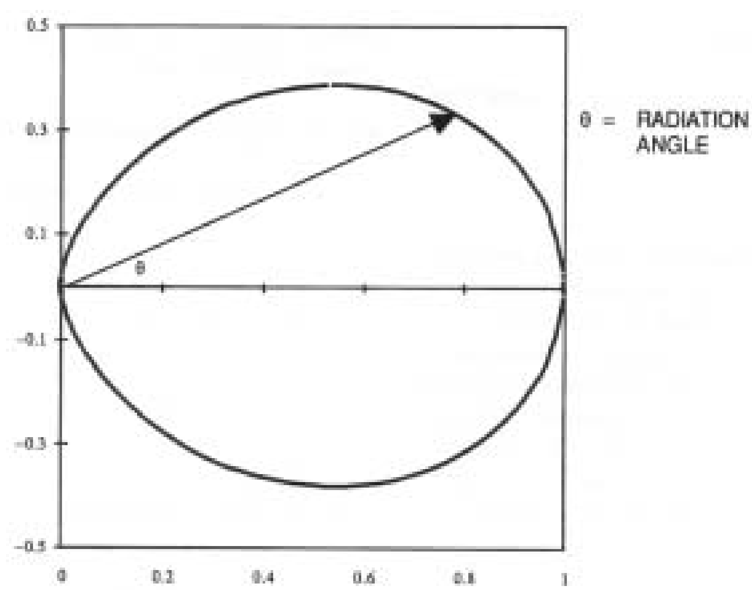



**FIGURE 4-10 DIPOLE RADIATION PATTERN**

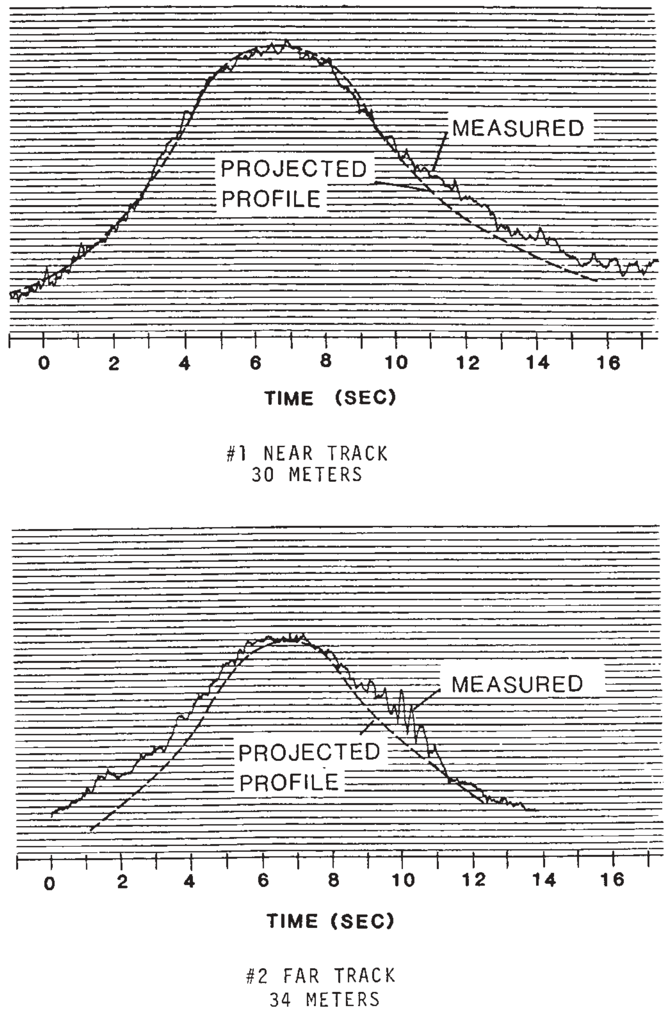



**FIGURE 4-11 MEASURED AND PREDICTED PASSBY TIME HISTORY OF A-WEIGHTED SOUND LEVEL FOR 5-CAR BART TRAIN AT 110 KM/HR (70 MPH)**

*4.2.1.11 Attenuation with Distance*

The sound radiated over an open field from a transit train attenuates with distance from the track for two main reasons, the effects of ground and atmospheric absorption notwithstanding: *Geometric Attenuation* or *Spreading Loss*, and *Excess Attenuation*. Geometric spreading loss is due to energy dispersion into three dimensions and is the most significant. From a point source, the maximum sound level attenuates at a rate of 6 dB per doubling of distance. For a line source, such as a train, and a receiver within about ½ train length from the track, the maximum sound level will attenuate at a rate of 3 dB per doubling of distance. At larger distances, the finite length of the train causes the maximum sound level to attenuate at a rate greater than 3 dB per doubling of distance. The equivalent level, $L_{eq}$, or the Day-Night Level, $L_{dn}$, will attenuate at 3 dB per doubling of distance due to geometric spreading, regardless of train length. Excess attenuation is due to sound energy absorption by the ground and the complex interference effects related to the direct and ground reflected waves.

**4.2.1.11.1 Attenuation Due to Geometric Spreading**

The maximum passby sound pressure level for dipole source characteristics in the absence of excess attenuation can be represented as

$$L_{\max}(\text{dB } re \text{ } 20\ \mu\text{Pascal})$$

$$= 10\text{Log}\left[\frac{3}{4}\,\frac{\rho c Q_D W_D'}{4\pi R}\,\{2\alpha + \sin(2\alpha)\}\right] + 94$$

For a monopole source, the maximum sound level as a function of distance, $R$, from the train is

$$L_{\max}(\text{dB } re \text{ } 20\ \mu\text{Pascal}) = 10\text{Log}\left[\frac{\rho c Q_M W_M'}{4\pi R}\,2\alpha\right] + 94$$

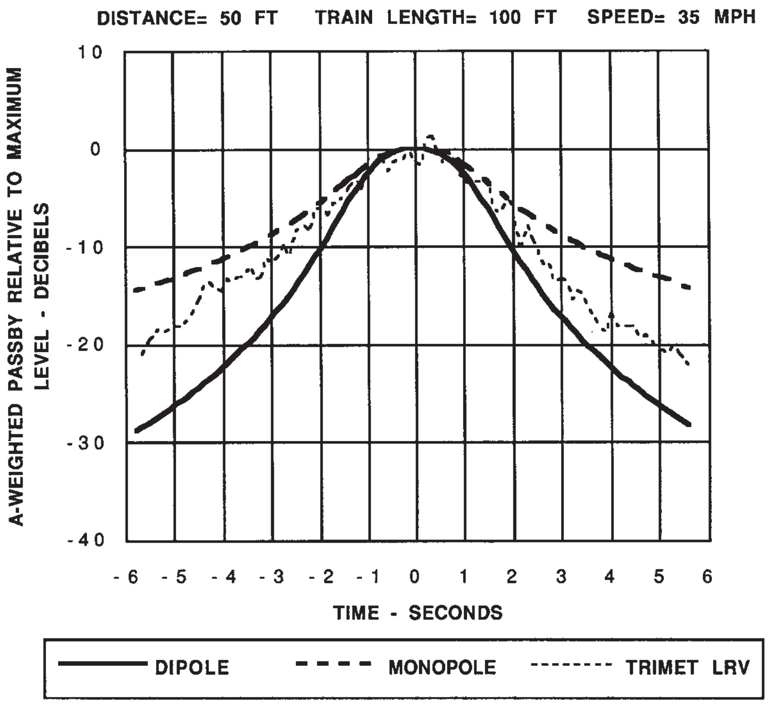



**FIGURE 4-12 TRIMET LRV PASSBY SIGNATURE WITH DIPOLE AND MONOPOLE PREDICTIONS**

The angle, $\alpha$, is given by

$$\alpha = \arctan\left(\frac{L}{2R}\right)$$

$W'_D$ is the dipole sound power per unit train length, $W'_M$ is the monopole sound power per unit train length, both in Watts per meter, $L$ is the train length in meters, and $R$ is the distance from the track center in meters. The product of the density and acoustic velocity, $\rho c$, is the specific impedance of air, equal to 407 rayls (kg/m$^2$/sec). (The SI system of units is preferred for these types of computations.) $Q_M$ and $Q_D$ are directivity factors associated with radiation patterns in the vertical plane transverse to the rail. For practical problems, the directivity factors $Q_M$ and $Q_D$ can be assumed to be uniform with respect to the angle above the horizontal plane, though this assumption is open to further investigation. The directivity factors are 1 for a fully absorptive ground surface (ballasted track with grassy embankment), and 2 to 4 for a fully reflective ground plane (embedded track), ignoring interference effects. If most of the noise is radiated by the rail, then $Q$ would be 4 for a concrete invert with direct fixation fasteners, yet 1 for ballast-and-tie track, a not insignificant difference of 6 dB, and only slightly greater than the 5 dB difference often observed between noise levels from aerial structure and ballasted track.

In practice, the sound level is measured at a reference distance of, perhaps, 15 m, or 50 ft, and sound levels at other distances are expressed as ratios of distances, speeds, and so on. In this case, the acoustic impedance, $\rho c$, and the sound power per unit track length cancel each other out. A variant of this approach is to measure the single-event noise exposure level, SENEL, or simply the SEL, for a train, and, from that, determine the sound power per unit length of train, assuming either a dipole or monopole distribution of sources. The SEL for a train passby is 10 times the logarithm of the time integral of the sound pressure squared, relative to the square of the reference pressure. For dipolar noise characteristics, the SEL is

$$\text{SEL (dB)} = 10\text{Log}_{10}\left[\int_{-\infty}^{+\infty} \frac{p^2(t)}{p_0^2}\,dt\right] = 10\text{Log}_{10}\left[\frac{3}{4}\,\frac{\rho c Q W'_D L}{p_0^2\, 4Rv}\right]$$

The integral is with respect to time in seconds and $v$ is the train speed in m/sec. For a monopole source, the SEL is

$$\text{SEL (dB)} = 10\text{Log}_{10}\left[\frac{\rho c Q W'_M L}{p_0^2\, 4Rv}\right]$$

The total sound power for the train is $WL$. A factor of $3/4$ appears in the expression for the dipole source. Thus, the single event level, SEL, for a dipole source is 1.25 dB less than the SEL for a monopole source of equivalent sound power per unit length.

The maximum sound level can be expressed in terms of the SEL directly, given the train length and distance (*15*). For the dipole distribution, the expression is

$$L_{max}(\text{dB } re \text{ } 20 \text{ } \mu Pa) = \text{SEL} + 10\text{Log}_{10}\left[\frac{V}{\pi L}\right] + 10\text{Log}_{10}[2\alpha + \sin(2\alpha)]$$

For the monopole source distribution

$$L_{max}(\text{dB } re \text{ } 20 \text{ } \mu Pa) = \text{SEL} + 10\text{Log}_{10}\left[\frac{V}{\pi L}\right] + 10\text{Log}_{10}[2\alpha]$$

The maximum sound level may be expressed in terms of the reference SELs presented in the FTA guidance manual for environmental assessment (*16*). The reference SEL levels are referenced to 50 ft from track center and 50 mph train speed. Dipole source distributions are assumed for rail cars, assuming that the noise is primarily wheel/rail, for which the maximum levels are then given by

$$L_{max}(\text{dB } re \text{ } 20 \text{ } \mu Pa) = \text{SEL}_{ref} + 10\text{Log}_{10}\left[\frac{\text{S(mph)}}{\text{50(mph)}}\right] - 10\text{Log}\left[\frac{\text{L(ft)}}{\text{50 ft}}\right] + 10\text{Log}_{10}[2\alpha + \sin(2\alpha)] - 3.3$$

For the monopole distribution, the maximum level is

$$L_{max}(\text{dB } re \text{ } 20 \text{ } \mu Pa) = \text{SEL}_{ref} + 10\text{Log}_{10}\left[\frac{\text{S(mph)}}{\text{50(mph)}}\right] - 10\text{Log}\left[\frac{L\text{(ft)}}{\text{50 ft}}\right] + 10\text{Log}_{10}[2\alpha] - 3.3$$

For train length long relative to receiver distance, $\alpha$ approaches $\pi/2$, $\sin(2\alpha)$ approaches 0, and the expressions for the maximum passby levels in terms of the SELs become the same for the dipole and monopole sources.

The attenuation of wayside maximum noise levels as a function of distance from a train with dipole noise sources is illustrated in Figure 4–13 (*17*). At distances close to the train,

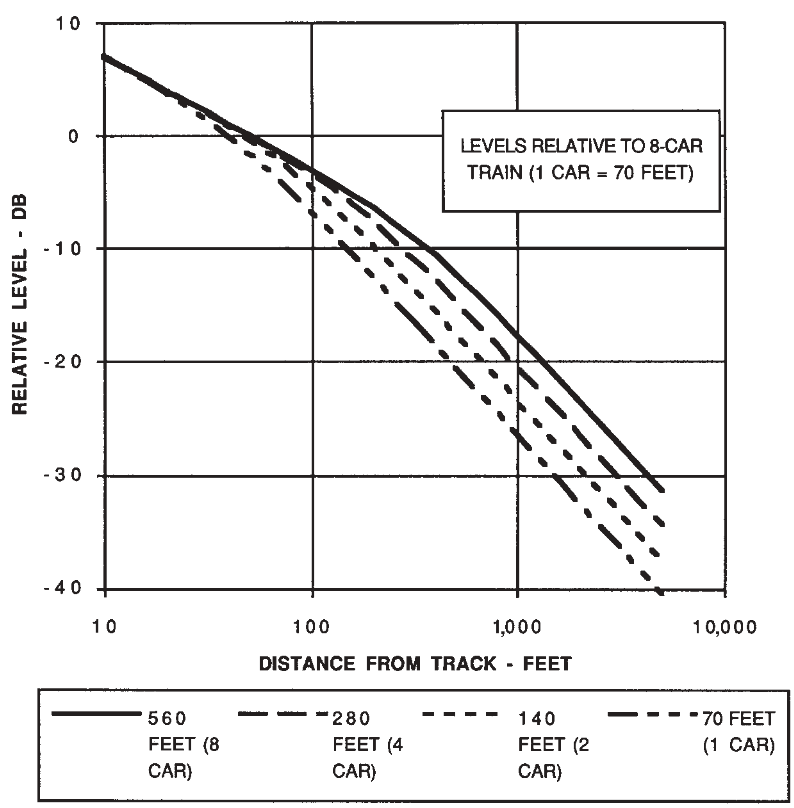



**FIGURE 4-13 GEOMETRIC SPREADING FACTOR FOR TRANSIT TRAINS (LEVELS ARE IN DECIBELS RELATIVE TO AN 8-CAR TRAIN AT 50 FEET - A CAR IS ASSUMED TO BE 70 FEET LONG)**

that is, within one train length, the attenuation versus distance is about 3 dB per doubling of distance, corresponding to a line source. At larger distances, the train begins to look like a point source, or at least a source of finite length, and the attenuation due to geometric spreading increases to about 6 dB per doubling of distance. The distance at which the attenuation versus distance goes from a line source character to a point source character is determined by whether or not the train is modeled as a dipole or monopole source. Within a distance of $2/\pi$ times the average truck-to-truck spacing, the short-period maximum noise level will be controlled by individual trucks, and the attenuation versus distance curves given in Figure 4–13 will not be representative at distances less than about 15 to 25 ft from the track. Normal train/receiver distances are greater than 25 ft, though there exist urban situations with transit structures immediately adjacent to residential structures.

The calculation of energy equivalent sound levels ($L_{eq}$) as a function of distance will depend on source characteristics, train speed, length, and so on The SEL, $L_{eq}$, and $L_{dn}$ attenuate at a rate of 3 dB per doubling of distance in the absence of ground effects, shielding by structures, or atmospheric absorption, because of the linear nature of the source when averaged over time. This particularly simple result leads to great simplification in predicting transit system noise, as opposed to predicting the attenuation with distance of maximum sound levels.

#### 4.2.1.11.2 Excess Attenuation

Where ground effects are to be included, a common assumption is that the single-event sound exposure level, SEL, or equivalent level, $L_{dn}$, attenuate at a rate of 4.5 dB per doubling of distance from the track. In this case, the maximum sound level may be calculated from the SEL using the formulas given above for 3 dB per doubling of distance to within an accuracy of about ½ dB. This surprising result is due to the fact that the maximum level will also decline by about 1.5 dB per doubling of distance in addition to the geometric spreading loss. More refined estimates of ground effect are provided in the FTA Guidance Manual (*18*).

The sound power per unit length cannot be reliably estimated from maximum sound levels and assuming a 4.5 dB attenuation per doubling of distance. A more physically realistic model would be required. On the other hand, within 50 ft from the track, sound absorption provided by the ground may not be significant, and a 3 dB attenuation per doubling of distance would probably be appropriate for most close receivers. In fact, with respect to design of noise control treatments, relying on excess attenuation to control noise impacts at receivers within about 100 ft from the track is probably not appropriate, since most receiver windows are well above the ground, and second story windows would not benefit at all. This is doubly true for aerial structure track. Judgment must be exercised in determining whether to include excess attenuation.

Atmospheric absorption is usually not significant within about 500 to 1,000 ft of the source. The excess attenuation at a frequency of 500 Hz and at a distance of 1,000 ft from the source is less than a decibel. Noise from rail transit systems is usually insignificant beyond about 500 ft from the track. For this reason, atmospheric absorption is not included in the modeling of rail transit noise versus distance. However, there may exist certain pathological situations where noise complaints are received from residents located at considerable distance from the track. Examples include low-frequency noise radiation from bridge structures, for which atmospheric absorption is not significant again. Rail corrugation may cause particularly raucous noise which may be objectionable to receivers at distances in excess of 500 or 1,000 Hz, depending on the type of community. In these cases, atmospheric absorption should be included in the consideration of sound propagation. However, one must add the effects of temperature inversions and wind gradients, both of which are discussed in the preceding chapter.

#### 4.2.1.11.3 Combined Geometric Spreading and Excess Attenuation

The following formula can be used to compute the SEL at a receiver for a reference $SEL_{ref}$ given for a reference distance and speed.

$$\begin{aligned} \mathrm{SEL} &= \mathrm{SEL}_{\mathrm{ref}} + (K_{\mathrm{speed}} - 10)\mathrm{Log}_{10}\left(\frac{V}{V_{\mathrm{ref}}}\right) \\ &\quad - (1 + G)10\mathrm{Log}_{\mathrm{ref}}\left(\frac{D}{D_{\mathrm{ref}}}\right) \end{aligned}$$

$$K_{\mathrm{speed}} = 30 \text{ dB/Decade speed}$$

$$G = 0.5 \text{ (Soft Ground)}$$

$$= 0 \text{ (Hard Ground)}$$

The speed dependence, $K_{speed}$, is given as 30 dB per decade speed for the maximum sound level, which is reasonably accurate for wheel/rail noise calculations. In the above, a decade change in speed is an increase in speed by a factor of ten. Thus, A-weighted noise levels will tend to increase approximately as 30 $\mathrm{Log}_{10}$(speed). Ten decibels are subtracted to account for the vehicle passby time duration, so that the actual variation of SEL as a function of train speed would be about 20 dB per decade speed. A lower speed dependence of 27 dB per decade speed has been measured for passby maximum sound levels by the author for vehicles with Bochum wheels. Still lower speed dependence might be obtained for vehicles with minor wheel flats, due to the speed dependence of wheel flat noise on train speed, as discussed below. The excess attenuation factor, $G$, is 0.5 if 4.5 dB attenuation per doubling of distance is assumed for "soft" ground, and is identically zero if a 3 dB attenuation per doubling of distance over "hard" ground or from aerial structures is assumed. This usage is consistent with the FTA Guidance Manual.

### 4.2.2 Excessive Rolling Noise

Excessive rolling noise without corrugation is produced by abnormally rough rails and wheels. The roughness may or may not be apparent to the eye, and may include visible pits, spalls, and other imperfections in the running surface. Poor definition of the contact wear strip or irregular width contributes to excessive rolling noise. The rail roughness spectrum illustrated in Figure 4–4 for the MBTA is an example of rough rail. Where excessive rolling noise is suspected, an inspection of the track and wheels may indicate pits and spalls and generally worn running surfaces. A quantitative measurement of rail roughness using a profilometer can be used to measure the roughness amplitude as a function of wavelength or wavenumber. Noise reductions achieved by rail grinding would confirm the existence of excessively rough rail.

Excessive rolling noise due to rail and wheel roughness with insufficient amplitude to produce contact separation is describable by the standard linear model of parallel mechanical impedances described above. In this case, the excessive rolling noise level varies linearly with roughness level. If smooth undulations of the rail surface are of sufficient amplitude, contact separation may occur, contributing to visible wear. In these cases, impact noise may occur, which is discussed below in the next section.

The radiation and attenuation with distance of noise as described above for normal rolling noise also applies to excessive rolling noise. The noise radiation should conform to that of a distribution of dipole sources.

### 4.2.3 Impact Noise from Rail Imperfections, Joints, and Wheel Flats

Impact noise is a special type of wheel/rail noise which occurs at tangent track with high-amplitude roughness, rail joints, rail defects, or other discontinuities in the rail running surface, and wheel flats. Remington provides a summary of impact noise generation (*5*), which involves nonlinear wheel/rail interaction due to contact separation, and is closely related to impact noise generation theory at special trackwork. I. L. Ver (*19*) categorizes impact noise by type of rail irregularity, train direction, and speed. The theory presented below is based on Ver's discussion.

#### *4.2.3.1 Smooth Irregularity*

A smooth irregularity may produce an impact if the train speed exceeds the critical velocity defined as

$$V_{CE} = \left[ \frac{g\left(1 + \frac{M}{m}\right)}{d^2y/dx^2} \left(1 + \frac{m}{\rho}\frac{\beta}{2}\right) \right]^{1/2}$$

Here, $g$ is the acceleration of gravity, $M$ is the total vehicle weight supported by the wheel, $m$ is the wheel mass, $\rho$ is the rail mass per unit length, and $y$ is the rail longitudinal profile. The parameter, $\beta$, is related to the rail support modulus, and rail section by

$$\beta = \left[ \frac{\mu}{4EI} \right]^{1/4}$$

*The rail need not contain a visual discontinuity to produce an impact*, because even a smooth undulation in the rail profile may cause contact separation. For an 880-lb wheel, 100 lb/yd rail, rail support modulus of 8.5×105 lb/ft$^2$, and roughness wavelength of 1.3 ft and amplitude of 0.08 in., the critical velocity at which rail separation may be expected is about 60 mph. The critical velocity is about 30 mph for a short wavelength of perhaps 2 in. and amplitude 0.05 in. (corresponding to very severe short pitch rail corrugation). (Impact theory is directly relevant to contact separation at corrugated rail.) The theory suggests that contact separation will occur at lower train speeds for low wheel loads than for high wheel loads. Thus, *light weight transit vehicles are more prone to dynamic contact unloading, lateral slip, and, perhaps, short pitch corrugation, than heavy freight vehicles.*

#### *4.2.3.2 Level and Decreasing Elevation Rail Joints and Wheel Flats*

"Level rail" joints are characterized by rail ends at equal running surface elevation, but separated by a distance, $w$. A decreasing elevation, or "step-down" joint, is characterized by an elevation drop, h. Wheel flats are characterized by the same distance, $h$, which is the defect depth of the flat relative to the normal rolling surface of the wheel. A rolling wheel must traverse the gap or wheel flat, and there exists a critical velocity above which the wheel loses contact with the rail. In this case, the critical velocity is given by

$$V_{CE} = \left[ ag\left(1 + \frac{M}{m}\right)\left(1 + \frac{m}{\rho}\frac{\beta}{2}\right) \right]^{1/2}$$

where $a$ is the wheel radius. The gap width, step-down height, or flat depth do not significantly affect the critical velocity, though these parameters do strongly determine the impact velocity of the rail and wheel, which is directly related to the peak sound pressure produced by the impact.

At high speed, the noise produced by a wheel flat will decrease with increasing train speed, and, at sufficiently high speed, the noise due to a relatively small single flat may become imperceptible above the normal rolling noise. Even with large flats, the wheel will appear to drop on the flat at low speed, but not drop on the flat at high speed (*20*). Similarly, a wheel traversing a gap will produce less noise at sufficiently high speed than at lower speeds.

#### *4.2.3.3 Step-Up Rail Joints*

Step-up rail joints are the most serious of impact noise producing rail joints, because there is no critical speed above which the peak sound pressure ceases to rise with increasing speed. An impact is always generated, and the impact velocity is directly related to speed. As a result, the impact noise from step-up joints would never be masked by normal rolling noise as might occur for step-down joints or wheel flats. The rail impulse, $m_{eq}\Delta V$, is then given by

$$m_{eq}\Delta V = Vm_{eq}\sqrt{\frac{2h}{a}}$$

where $m_{eq}$ is an equivalent rail mass determined by its bending stiffness and mass per unit length. For typical rail supports and bending moduli, the equivalent mass is approximately 40% of the mass contained within a 1-m-long section of rail.

#### *4.2.3.4 Summary of Impact Noise*

The main points of the above discussion are listed by I. L. Ver, et al. as

1) The step-up joint is the most serious rail joint with respect to producing impact noise, because there is no critical speed above which no increase in noise level may be expected with increasing speed.
2) Step-down joints and wheel flats cause separation of the wheel from the rail above a critical velocity. Below the critical velocity, sound pressure levels may be expected to increase with increasing train speed. Above the critical velocity, no increase in noise level may be expected with increasing train speed, and the impact noise may be masked by rolling noise at sufficiently high speed.
3) Below critical speed, step-up joints, step-down joints, and wheel flats of the same height generate similar peak sound levels for a given rail head elevation difference or flat depth, h. The peak sound pressure increases with increasing wheel load and height.
4) Smooth irregularities may produce impact noise due to contact separation above a critical speed. The critical speed is less at lower axle loads than at higher axle loads, and increases with increasing radius of curvature of the irregularity.
5) Rail support resilience increases the critical speed relative to rigid rail condition. For typical wheels and rail support conditions, the increase is by a factor of two relative to rigid track.
6) The impact noise *decreases* with increasing wheel radius and *increases* with increasing wheel mass. These offsetting parameters may cancel any noise reduction which might be expected by increasing wheel radius.

Modern transit systems employing continuous welded rail and trued wheels will likely not be concerned with impact noise generated by rail joints, though there will remain impact noise generated by rough rail, wheel flats, switch frogs, and crossover diamonds. Further, older systems which employ jointed rail at steel elevated structures must be concerned with rail joint condition and maintenance, and all systems must be concerned with rail grinding and wheel truing to eliminate associated impact noise.

### **4.2.4 Corrugated Rail Noise**

Rail corrugation causes excessive rolling noise of a particularly harsh character and of very high level. The terms “roaring rail” or “wheel howl” or “wheel/rail howl” are typical descriptors of noise produced by corrugated rail.

#### *4.2.4.1 Characterization*

If rail corrugation exists, the passby noise level will be high compared to that of normal rolling noise, and the spectrum will contain discrete frequency components and associated harmonics clearly revealed with 1/3-octave or narrow band analyses of the passby or interior noise. An example of a 1/3-octave band spectrum of noise due to trains traveling on corrugated direct fixation track is presented in Figure 4–14. In this case, the existence of corrugation is revealed by the spectral peak at the corrugation frequency, which occurs at about 500 Hz. A second harmonic is also visible at 1,000 Hz. In this example, the trains had aluminum centered wheels, and were typically four to seven cars in length. Also, at this location, train speeds were uniform at about 54 mph, regulated under computer control. Thus, the corrugation wavelength was clearly defined, not being destroyed by random train speed variation.

Narrow band analyses of noise may reveal corrugation wavelengths or other periodic patterns in the rail running surface. For example, noise collected at multiple train speeds for operation on the same corrugated rail will show a linear variation of spectral peak frequencies with train speed. At high speeds, contact separation might occur, and the discrete frequency component due to rail corrugation may broaden, for which a less easily identified pure tone component might be expected. Narrowband analyses of car interior noise with substantial wheel/rail howl are presented in Figures 4–15 and

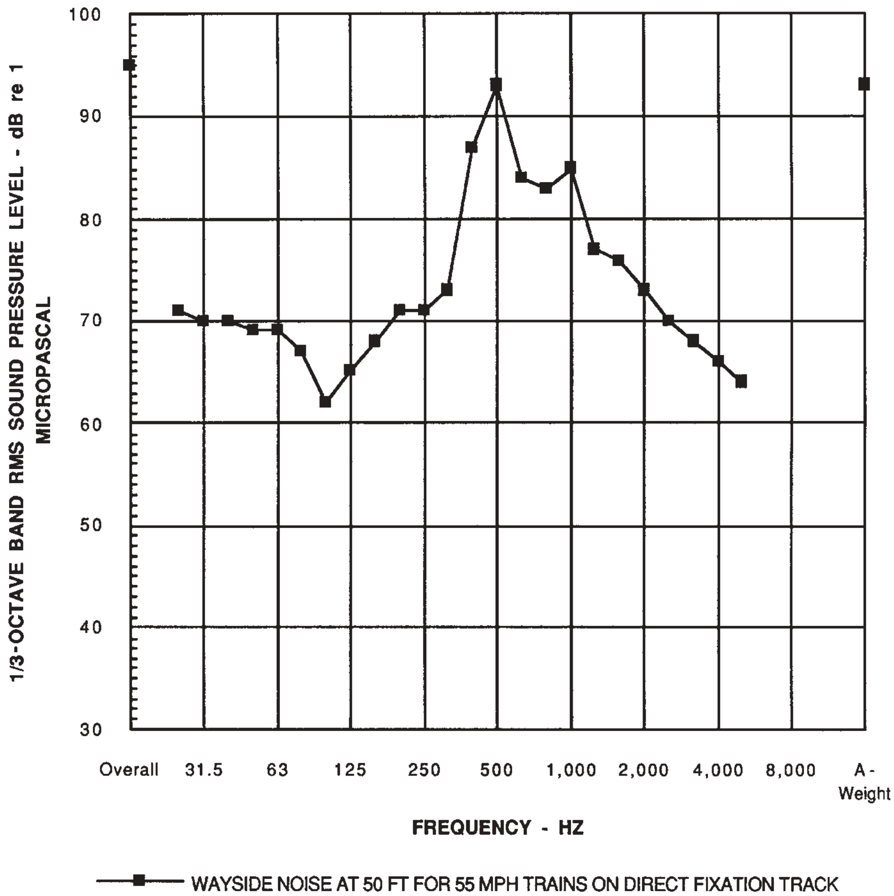



**FIGURE 4-14 ONE-THIRD OCTAVE SPECTRUM OF NOISE DUE TO RAIL CORRUGATION**

4–16 for train speeds 40 and 66 mph, respectively. The data were recorded at a severely corrugated section of track and at a section of recently ground track with grinding pattern in the BART transbay tube. The corrugation frequency is clearly observable at about 400 and 600 Hz in the 66 mph data, while at 40 mph, the peaks are shifted downward to 250 and 375 Hz. These peaks are absent in the data shown for ground rail. (However, a grinding pattern caused peaks to occur at about 1,400 and 900 Hz, respectively, at 66 and 40 mph.)

A mechanical resonance, such as that produced by resonance of the wheel set or track, will produce noise with spectral components whose frequencies do not vary with train speed. Thus, if spectral analyses for various speeds indicate no change in the peak frequency with varying train speed, the cause of the noise is likely not rail corrugation or any other periodic pattern in the rail. On the other hand, mechanical resonances will be excited by rail corrugation, and the amplitude of the resulting noise may vary substantially, depending on the relationship between corrugation frequency and the mechanical resonance frequency.

Machine bluing, magnaflux dye, or lamp black spread over the rail running surface may be helpful in identifying corrugation waves. The corrugation wavelength may be measured and compared with noise spectra obtained at the same location to determine if corrugation is a significant source of the noise. A rail profilometer can be used for evaluation of wear rates and grinding effectiveness.

### *4.2.4.2 Wheel/Rail Interaction at Corrugated Rail*

The interaction between the wheel and rail in the presence of corrugation may be considerably more complex than that described by the standard linear model used to describe normal rolling noise. One principal reason is that contact separation may occur as a result of the inability of the wheel to follow the corrugated rail's vertical profile. For example, the force required to displace the center of gravity of an 800-lb wheel by 0.005 in. is about 10,000 lb, roughly similar to the wheel static load, assuming the wheel to be rigid. Contact patch stiffness, rail compliance, and wheel resonances will modify this estimate. Higher amplitude corrugation would produce higher dynamic loads, except that the static load places a limit on the maximum dynamic load that can occur during an unloading cycle. Thus, there must be a contact separation at higher amplitudes. Corrugations of even moderate amplitude can be expected to produce contact separation, or

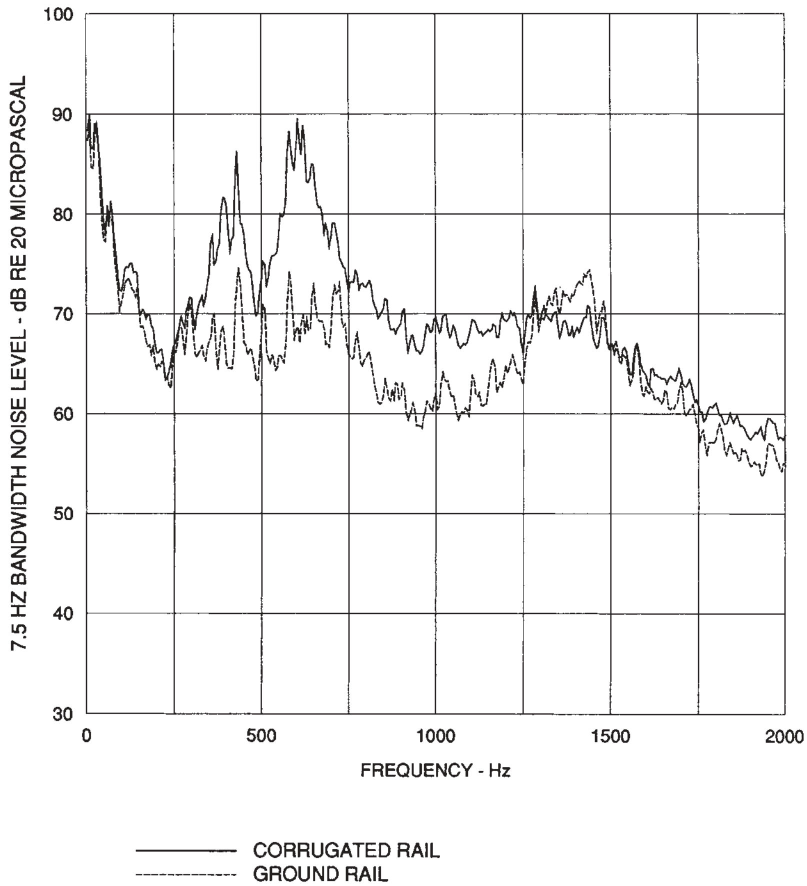



**FIGURE 4-15 INTERIOR NOISE AT 66 MPH ON CORRUGATED RAIL (TRANSBAY TUBE)**

at least an increased propensity for lateral dynamic slip between the wheel and rail due to unloading of the contact patch. Periodic lateral slip can then lead to additional periodic wear and corrugation, producing regenerative wear.

Figure 4–17 illustrates 1/3-octave band noise levels with and without corrugation. In these cases, a corrugation amplitude of merely 0.002 in. produces noise at a discrete frequency in the 630 Hz octave band roughly 8 to 10 dB higher relative to neighboring bands. A larger amplitude of 0.004 in. produces a broader spectral peak between 500 and 1,000 Hz. The actual conditions under which these data were taken are not known, but the broader bandwidth associated with corrugation amplitude of 0.004 in. versus an amplitude of 0.002 in. suggests that the noise generation process might be different between these two sets of data. Contact separation is a possible mechanism by which this might be brought about. More importantly, the data indicate that only a relatively small corrugation amplitude is sufficient to produce a substantial increase in wayside noise levels.

In the presence of lateral slip or contact separation, the equality of lateral dynamic displacement of the wheel and rail is relaxed. A practical result is that the wheel's lateral vibration modes may be easily excited, with the result that the ratio of noise energy radiated by the wheel relative to that radiated by the rail may be more with corrugated rail than with uncorrugated rail.

Modeling of wheel/rail noise in the presence of contact separation or lateral dynamic slip is complicated by the nonlinearity of the contact vertical load versus vertical displacement, and lateral slip versus vertical displacement. A realistic theoretical model requires time-domain integration of the equations of motion, and simple modeling with the usual linear theories may be inadequate. For example, accounting for the nonlinearity of the Hertzian contact patch stiffness as a function of dynamic load indicates that dynamic contact forces may be higher than those predicted on the basis of constant contact patch stiffness, exceeding the wheel static load for rail roughness amplitudes on the order of 0.001 in. and trains traveling at close 80 or 90 mph, further encouraging lateral slip during unloading (*21*), which has been identified with the corrugation process (*22*).

To summarize, rail corrugation is more difficult to control at rail transit systems than at railroads because of lighter contact patch loads at rail transit systems. The uniformity of transit vehicle types and speeds prevents randomization of wheel/rail force signatures, which simply exacerbates the corrugation process. Thus, for rail transit systems with lightly loaded rails, the importance of maintaining rail smoothness is greater than at heavy freight systems. Rail cor-

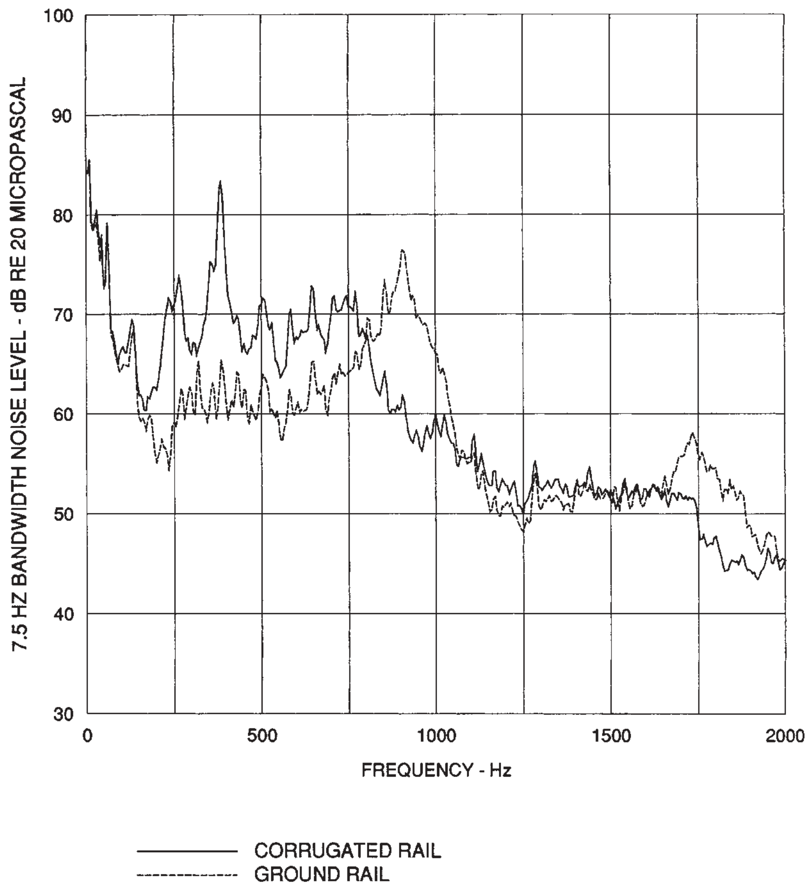



**FIGURE 4-16 INTERIOR NOISE AT 40 MPH WITH CORRUGATED RAIL (BART TRANSBAY TUBE)**

rugation is the principal cause of excessive noise levels at many transit systems, and controlling rail corrugation is key to rail transit system noise control.

Rail corrugation processes and methods of control are discussed in greater detail in the chapter on rail corrugation.

## 4.3 CURVING NOISE

Curving noise includes both normal rolling noise, which also occurs at tangent track, and noise unique to curving, resulting form lateral slip of the wheel tread across the rail head. Noise due to lateral slip is often manifested as an intense, sustained squeal, caused by the negative damping associated with the friction versus creep characteristic. Normal rolling noise will not be significant at short radius curves, especially in the presence of wheel squeal, due to low train speed, and at high-speed curves, rolling noise may be treated in much the same manner as for tangent track. This section is directed entirely to curving noise due to lateral slip generated noise.

There appear to be two types of curving noise. The first and most prevalent is wheel squeal cause by sustained nonlinear lateral oscillation of the wheel. The second, less obvious, and less common, is wheel howl, which may be due to the resonant but unsaturated response of the wheel to dynamic lateral creep forces. There are reasons to believe that these types of curving noise are separate phenomena: (1) they do not sound the same; (2) wheel howl at curves increases with train speed, while conventional wheel squeal may disappear with sufficiently high train speed; and (3) wheel/rail howl may be closely associated with short pitch corrugation at curves. Of these two types of curving noise, wheel squeal is the more prevalent, while wheel howl may be limited to lightly damped aluminum centered wheels such as used at BART. In fact, wheel howl may be unique to BART, because BART is the only transit system employing rigid aluminum centered wheels.

### 4.3.1 Wheel Squeal

Wheel squeal is one of the most noticeable types of noise produced by rail transit systems, and can be very significant at both short and long radius curves. At embedded track curves of light rail systems, pedestrians and patrons may be in close proximity to the vehicle and thus be subjected to high squeal noise. An example is the Government Center Curve at the Boston Green Line, where transit patrons are necessarily within a few ft of the track within an enclosed underground area, without the benefit of shielding by a platform. At older transit systems where windows may be left open during passage through subways, squeal

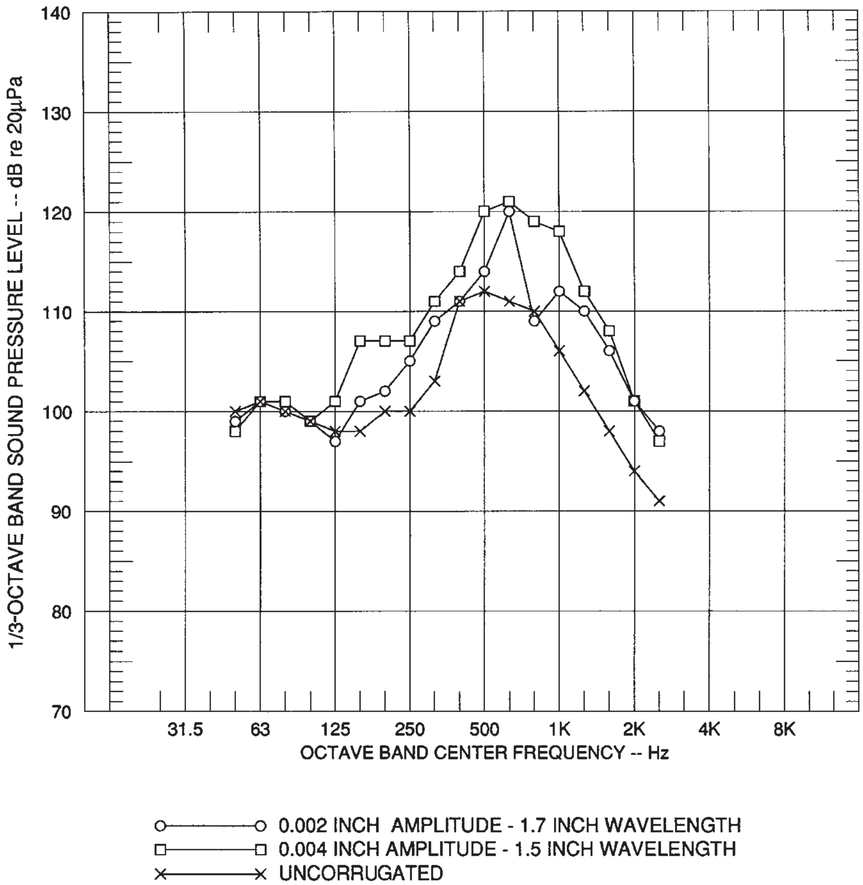



**FIGURE 4-17 WAYSIDE NOISE LEVELS DUE TO RAIL CORRUGATION (SPENO INTERNATIONAL)**

can be discomforting to patrons inside the vehicle, especially if no substantial sound absorption exists in the subway. Even in modern transit vehicles, inadequate door seals may allow exposure of patrons to high levels of squeal noise in tunnels.

Wheel squeal may be intermittent due to varying tribological properties, varying amounts of rail surface contaminants, or curving dynamics of the vehicle and rail. An example of the time dependence of wheel squeal is provided in Figure 4–18. In this case, the squeal is produced by a Portland Tri-Met vehicle with resilient wheels on an 87-ft radius embedded track curve. The squeal noise level varies considerably over time. On damp mornings, wheel squeal may be nonexistent for all or most of the curve negotiation. The maximum level of squeal noise is relatively insensitive to duration and frequency of occurrence.

Maximum and energy-averaged 1/3-octave band spectra of the wheel squeal illustrated in Figure 4–18 for the Tri-Met vehicle are provided in Figure 4–19. These data include fundamental squeal frequencies in the 500, 1,250, and 3,150 Hz 1/3-octave bands. The high levels of noise at discrete squeal frequencies indicate the high level of perceptibility and potential for annoyance. The maximum levels shown are the maximum observed 1/3-octave noise levels occurring during curve negotiation, and may not have occurred simultaneously. The lower levels are the energy-averaged, or rms, sound levels occurring during curving. In this example, there exists considerable disparity between the maximum sound level and energy-averaged sound level, caused by the intermittency of the squeal.

Figure 4–20 illustrates the narrowband frequency spectrum of wheel squeal recorded at BART during the Car 107 tests in 1971 at a 540-ft radius curve. These data include sustained squeal at frequencies above 1,000 Hz and low train speed of 18 mph. At a train speed of 35 mph, the squeal above about 1,000 Hz is absent. The inhibition of squeal at higher train speeds is a feature of wheel squeal that is due to reduced negative damping at higher lateral slip velocities, as discussed below.

#### *4.3.1.1 Causes of Wheel Squeal*

Three types of stick-slip motion have been postulated for producing wheel squeal noise:

1) Longitudinal stick-slip,
2) Flange contact with the gauge face, and
3) Stick-slip due to lateral creep across the rail head caused by nonzero angle of attack of the wheel, in turn produced by the finite wheel base of the truck.

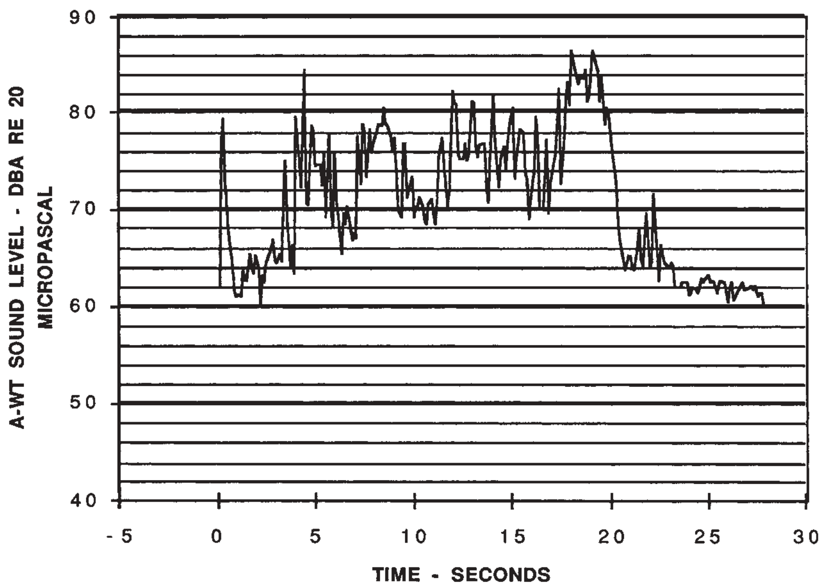



FIGURE 4-18 WHEEL SQUEAL FROM ARTICULATED LIGHT RAIL VEHICLE WITH RESILIENT BOCHUM WHEELS ON EMBEDDED TRACK WITH 87-FOOT RADIUS CURVE

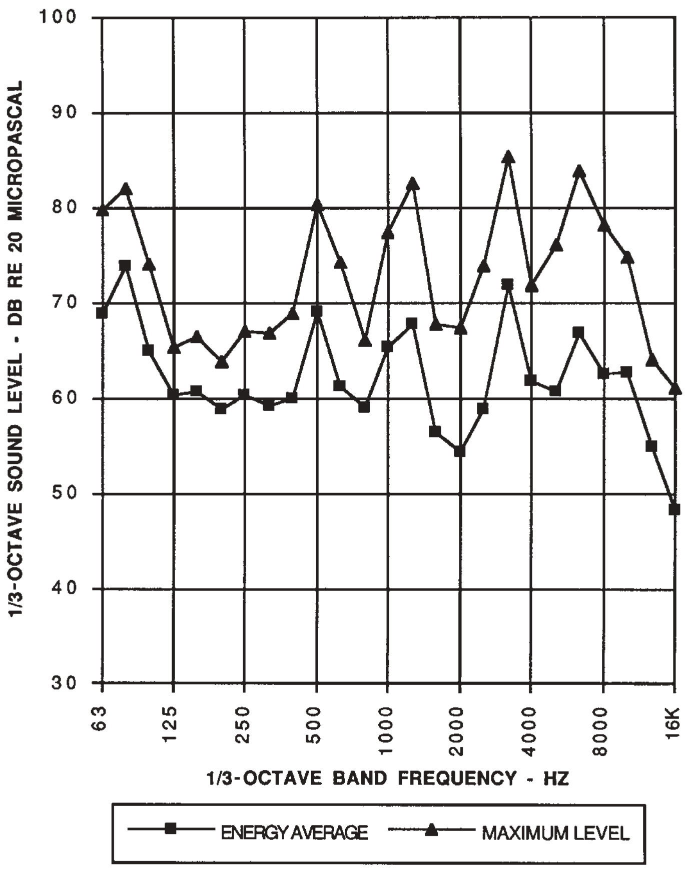



FIGURE 4-19 WHEEL SQUEAL AT PORTLAND TRI-MET WITH RESILIENT BOCHUM WHEELS ON URETHANE EMBEDDED TRACK WITH 87-FOOT RADIUS CURVE

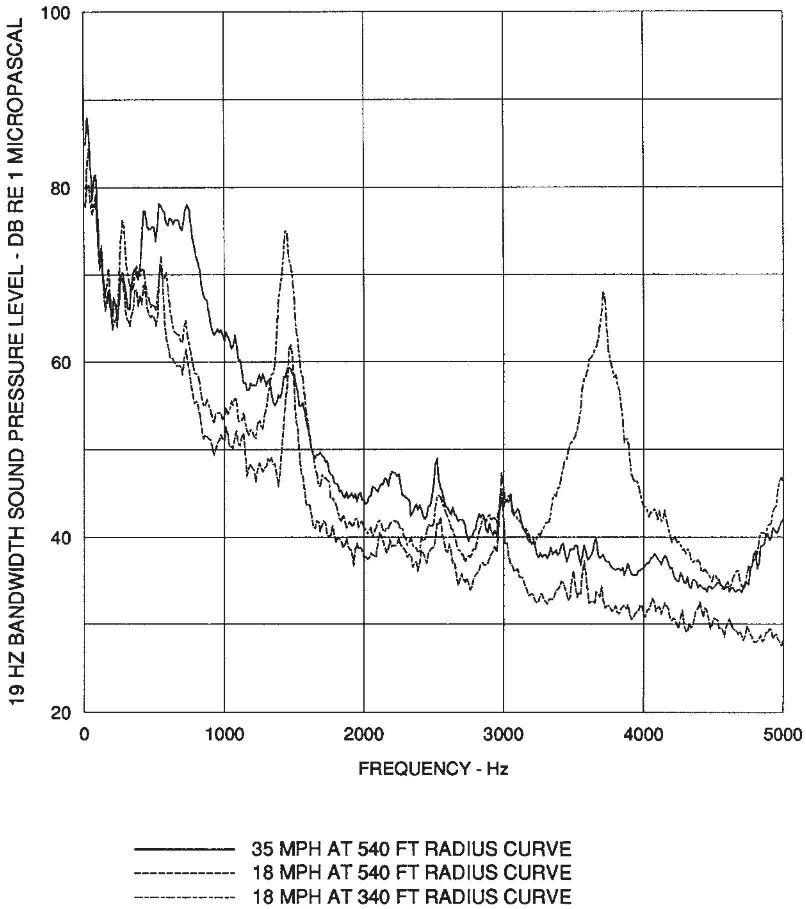



**FIGURE 4-20 CURVING NOISE IN BART CAR 107 WITH ALUMINUM CENTERED WHEELS ON GROUND RAIL (1971)**

Longitudinal stick-slip is due to the different translation velocities between the high and low rail wheels. Wheel taper is sufficient to compensate for differential slip at curves in excess of about 2,000-ft radius, though shorter radii may be accommodated by profile grinding of the rail head. Further, Rudd reports that elastic compression of the inner wheel and extension of the outer wheel tread under torque can compensate for the wheel differential velocities, and, further, that trucks with independently driven wheels also squeal (*23*). Therefore, longitudinal slip is not considered to be a cause of wheel squeal.

Flange rubbing is due to contact between the flange and high rail and has been considered by many as a cause of wheel squeal. However, lubrication of the flange does not always eliminate wheel squeal, and (according to Rudd) Stappenbeck reports that the low rail wheel has been identified as the source of wheel squeal, which does not ordinarily undergo flange rubbing (*24*). These observations suggest that flange contact is not a significant cause of squeal. However observations of noise at the MBTA Green Line indicate that substantial squeal is produced by the unrestrained high rail wheels at a short radius curve, and little or none by the low rail restrained wheels. Substantial flange rubbing occurs at the high rail leading wheel, suggesting that flange rubbing contributes to squeal. Further, it is not clear whether the leading or trailing wheel is the source, and the trailing wheel flange does not contact the gauge face. Other observations at the MBTA Red Line indicate that squeal occurs at points of repeated flange contact with the high rail, though, in this case, determining whether the high or low rail wheels are the sources of squeal was not possible. Flange lubrication is used to reduce rail wear and noise at curves, but the noise reduction observed with flange lubrication may be due to migration of lubricant to the rail running surface.

Lateral slip of the tread running surface across the rail head is *the most probable cause* of wheel squeal, and is most tractable from a theoretical point of view. Even though flange contact is often observed at sections of track where squeal occurs, such as at the MBTA, the squeal may be due entirely to lateral slip across the rail head. The lateral slip theory of wheel squeal is discussed more fully below.

*4.3.1.2 Lateral Slip Model*

Figure 4–21 illustrates the geometry of curve negotiation by a transit vehicle truck as described in the literature. As illustrated, lateral slip across the rail head is necessitated by the finite wheel base, *B*, of the truck, and the radius of curvature of the rail, where no longitudinal compliance exists in the axle suspension. However, this curving diagram is not realistic of actual performance. Figure 4–22 illustrates the crabbing of a truck under actual conditions (*25*). In this case, the leading axle of the truck rides toward the outside of the curve, limited only by flange contact of the high rail wheel against the gauge face of the rail. The trailing axle, however, travels between the high and low rail, and the low rail wheel flange may, in fact, be in contact with the low rail gauge face. The result is a reduction of creep angle at the trailing axle, but an increase of creep angle at the leading axle. Gauge widening would only increase the actual creep angle under these circumstances. Moreover, with severe creep angle, flange rubbing at the high rail occurs, contributing to wear. This crabbing condition is particularly observable at the MBTA Green Line Government Center Station curve.

The theory of stick-slip oscillation is presented below in some detail in an attempt to clarify the literature. Figure 4–23 is a schematic of the wheel and rail system, for which various parameters are defined. The most useful way of thinking about squeal is to consider the problem from the center of gravity of the wheel, and think of the rail as slipping beneath the tire in the lateral direction at an average slip velocity, *V*. The deflection of the tire at the contact point, relative to the center of the tire, is represented by *U*. The velocity of the tire relative to the center of the wheel is then $dU/dt$. Then, the total creep velocity of the tire relative to the rail head is $V - dU/dt$. If the rolling velocity is defined as *S*, then the total creep is defined as

$$\xi = V - \frac{dU}{dt}$$

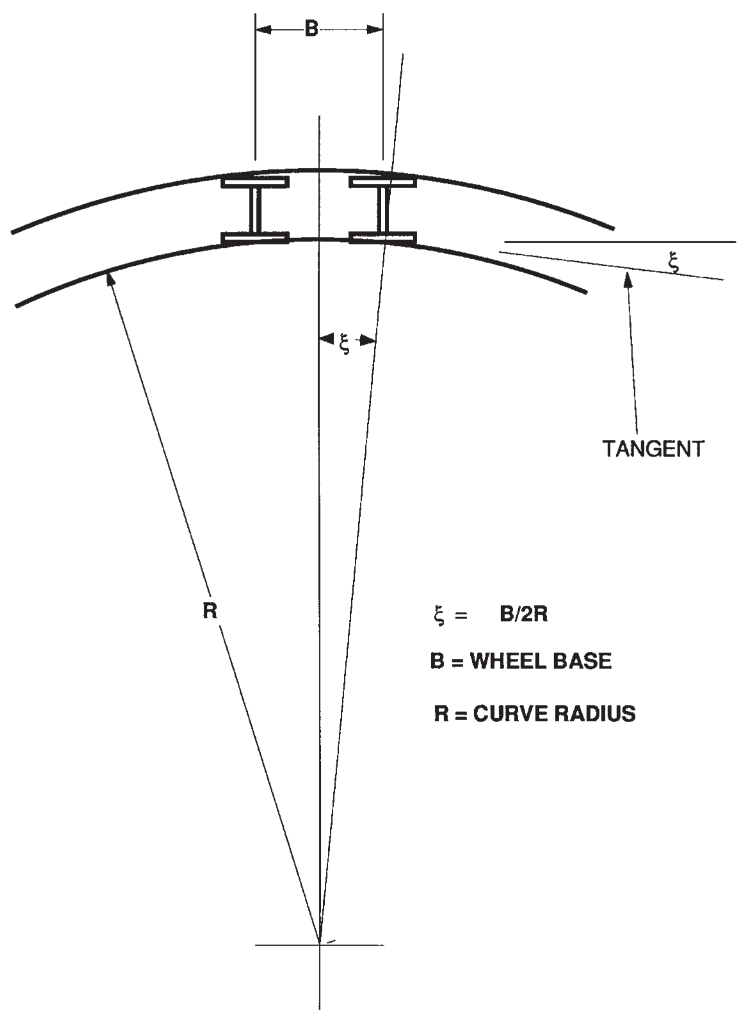



**FIGURE 4-21 GEOMETRY OF CURVE NEGOTIATION AND LATERAL SLIP**

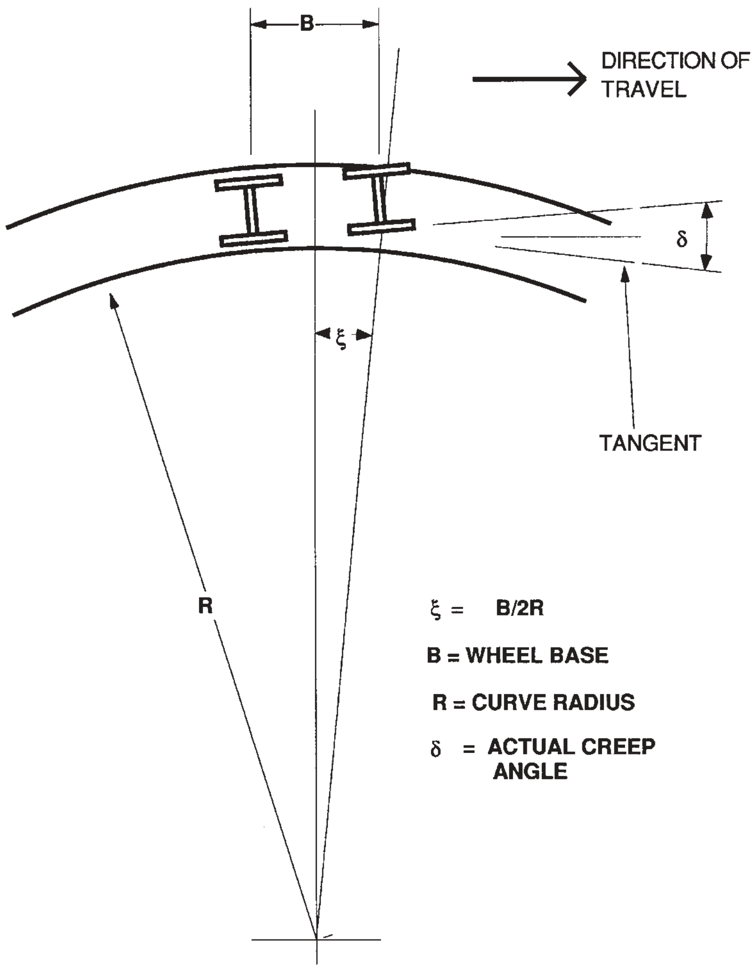



**FIGURE 4-22 TRUCK CRABBING UNDER ACTUAL CONDITIONS**

and the average creep for a lateral slip velocity, $V$, is

$$\xi_0 = \frac{V}{S}$$

The friction coefficient, μ, is assumed to be entirely a function of the total creep, ξ.

The equation of motion is obtained by setting the acceleration of the wheel tread equal to the sum of the forces acting on the tread, which includes the forces due to the stiffness of the wheel, the internal damping of the wheel, and the friction force, $\mu W$, where $W$ is the vertical contact load

$$m\frac{d^2U}{dt} = -kU - c\frac{dU}{dt} + \mu W$$

The friction coefficient can be expanded about the average creep, $\xi_0$:

$$\mu = \mu_0(\xi_0) + \left.\frac{d\mu}{d\xi}\right|_{\xi_0}(\xi - \xi_0) + \ \ldots$$

Now define the slope of the friction-creep function, μ, as

$$v = \left.\frac{d\mu}{d\xi}\right|_{\xi_0}$$

Defining the deflection of the tread as the sum of a steady state deflection, $U_0 = \mu_0 W$, and a dynamic component, $u$, so that $U = U_0 + u$, the dynamic component of the creep can be expressed as

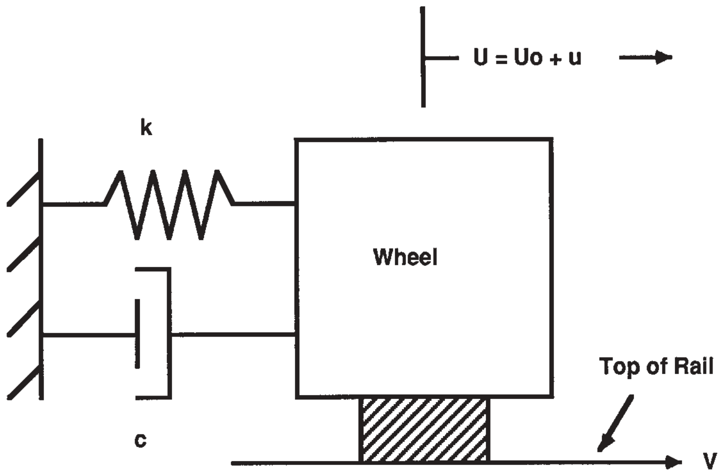
$$\xi - \xi_0 = -\frac{1}{S}\frac{dU}{dt} = -\frac{1}{S}\frac{du}{dt}$$

Then, the equation of motion becomes

$$m\frac{d^2u}{dt^2} + \left[c + v\frac{W}{S}\right]\frac{du}{dt} + ku = 0$$

The solution of this equation is

$$u = u_0 e^{\pm i\left[\frac{k}{m} - \frac{1}{4m}\left(c+v\frac{W}{S}\right)\right]^{1/2}} e^{-\frac{1}{2m}\left(c+v\frac{W}{S}\right)}$$




| | |
|---|---|
| Deflection: | U |
| Spring return force: | -kU |
| Internal damping force: | -cdU/dt |
| Friction force: | $-\mu$ W |
| Friction Coefficient: | $\mu(\xi)$ |
| Lateral Creep: | $\xi$ = (V-dU/dt)/S |
| Average Creep Velocity: | V |
| Rolling Velocity: | S |
| Normal load: | W |

FIGURE 4-23 MODEL OF WHEEL SQUEAL GENERATION

Thus, exponential growth of the solution occurs if the slope, $\nu$, of the friction-creep curve is sufficiently negative to overcome the internal dissipation, $c$.

*4.3.1.3 Loss Factor Due to Stick-Slip*

The internal damping and negative damping due to stick-slip are described by Rudd in terms of loss factors (*26*).

The combined loss factor for the wheel vibration in the axial direction is

$$\eta = \eta_{int} - \eta_{ss}$$

where

$\eta_{int}$ = internal loss factor
= $c/m\omega$
$\eta_{ss}$ = Loss factor due to stick-slip
= $-\nu W/m\omega S$
$m$ = modal mass
$\omega$ = radian frequency
= $(k/m)^{1/2}$

If the slope, $\nu$, of the friction coefficient versus lateral creep, is negative, then the loss factor due to stick-slip is positive. If the loss factor due to stick-slip is large enough, the combined loss factors due to stick-slip and internal energy dissipation will be negative, resulting in amplification of vibration at a modal frequency, as described above. The modal mass is assumed to be about ⅓ of the wheel mass, given in a consistent system of units.

The loss factor due to stick-slip is inversely proportional to the rolling velocity of the wheel, because the creep function is independent of rolling velocity (the creep is equal to the ratio of the slip velocity and rolling velocity). Thus, wheel squeal may be expected to cease above a certain velocity speed where the loss factor due to stick-slip is not sufficient to overcome the loss factor due to internal damping. The form of the friction versus creep used by Rudd is illustrated in Figure 4–24 and is given by

$$\mu = \mu_{\max} \frac{\xi}{\xi_m} \exp(1 - \xi/\xi_m)$$

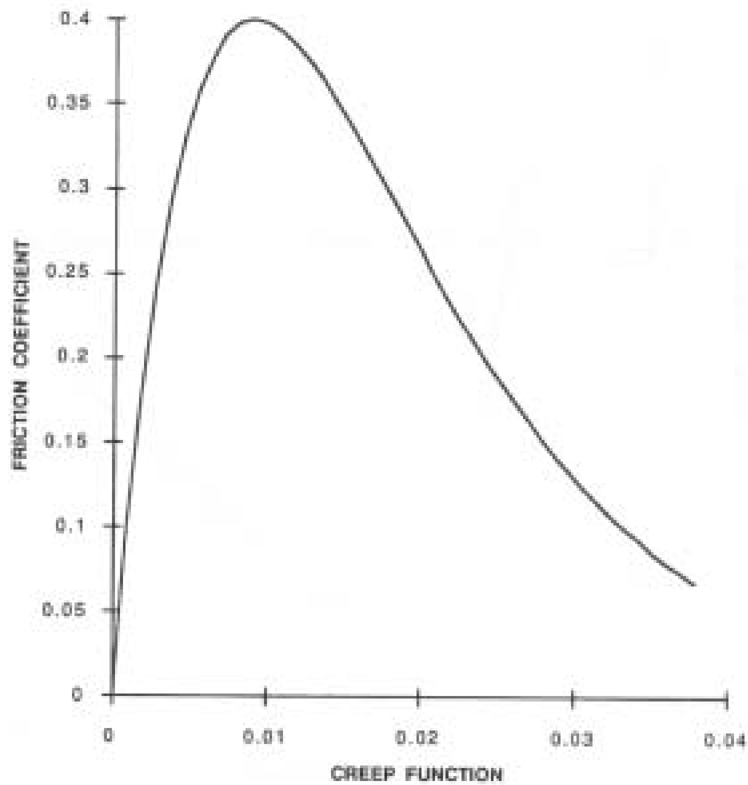



FIGURE 4-24 FRICTION VERSUS CREEP

Here, $\xi_m$, is the value of $\xi$ at maximum friction coefficient, $\mu_{max}$. At a creep angle of about 0.009 radians (0.5 deg), the friction coefficient reaches a maximum value of about 0.4. The corresponding slope of the friction versus creep curve is illustrated in Figure 4–25. Above the maximum friction point, the slope becomes negative, and unstable oscillations may occur, though the occurrence still depends on the magnitude of the slope. Below the maximum friction point the curve has positive slope, and thus would not produce stick-slip oscillation.

For a wheel base of 7.5 ft, squeal would not be expected for curve radii greater than 410 to 830 ft, the lower limit being achieved when there is no gauge relief. As illustrated above, gauge widening increases the creep angle for the same radius of curvature. A typical assumption is that squeal does not occur for curves with radii greater than about 700 ft, corresponding to a creep rate given by

$$\xi = 0.7B/R$$

where
$B$ = wheel base
$R$ = curve radius

*4.3.1.4 Effect of Internal Loss Factor*

The internal loss factor required to overcome the negative friction due to stick-slip can be estimated by equating the loss factor due to internal damping to the absolute value of the loss factor for stick-slip. In this case, the loss factor varies as a function of curve radius due to varying crab angle, or creep, and train speed, as illustrated in Figure 4–26. Two curves are shown for vehicle lateral accelerations of 3% and 15% of earth's gravitational acceleration. The lateral acceleration is determined from the forward velocity of the vehicle and curve radius. For example, the typical speed of a BART rail transit vehicle in a 540-ft radius curve is about 35 mph, corresponding to a lateral acceleration of about 15% of gravity. At high train speed and lateral acceleration, the minimum loss factor due to stick-slip is about 20%. At low speed, corresponding to lateral acceleration of 3%, the loss factor is considerably higher.

In practice, wheel squeal is usually inhibited by wheels with lower loss factors than necessary to compensate for the theoretical negative damping effect. Loss factors for resilient and damped wheels which have been shown to eliminate or at least partially reduce squeal are provided in Table 4–1. Immediately apparent is that the loss factors of these wheels are much less than predicted to inhibit stick-slip oscillation (illustrating the complexity and difficulty in achieving a satisfactory theory of wheel squeal phenomena.) Deficiencies in the model likely concern the friction-versus-creep representation.

*4.3.1.5 Friction*

Meteorological conditions affect the generation of squeal. In wet weather, for example, wheel squeal occurrence may

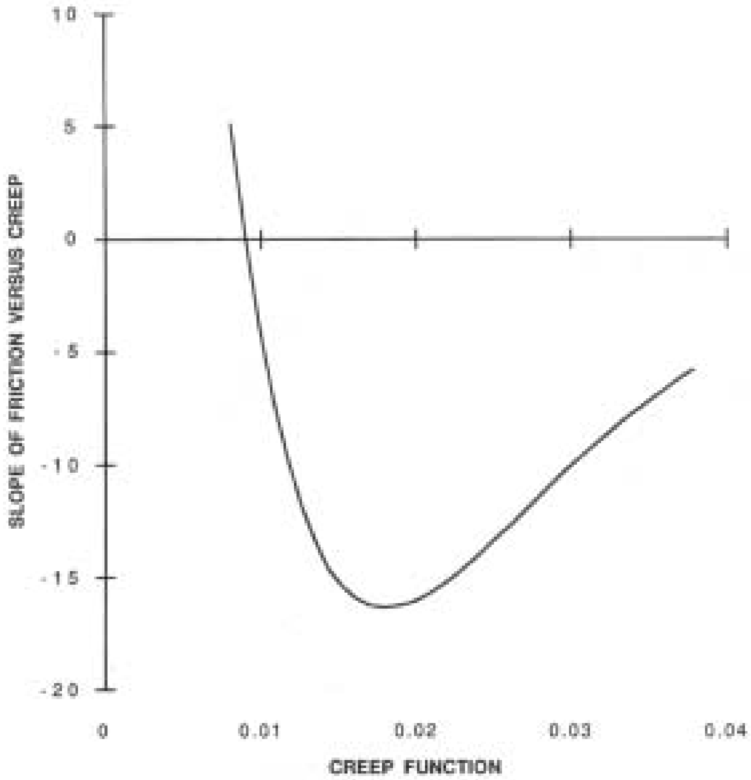



FIGURE 4-25 SLOPE OF FRICTION VERSUS CREEP CURVE

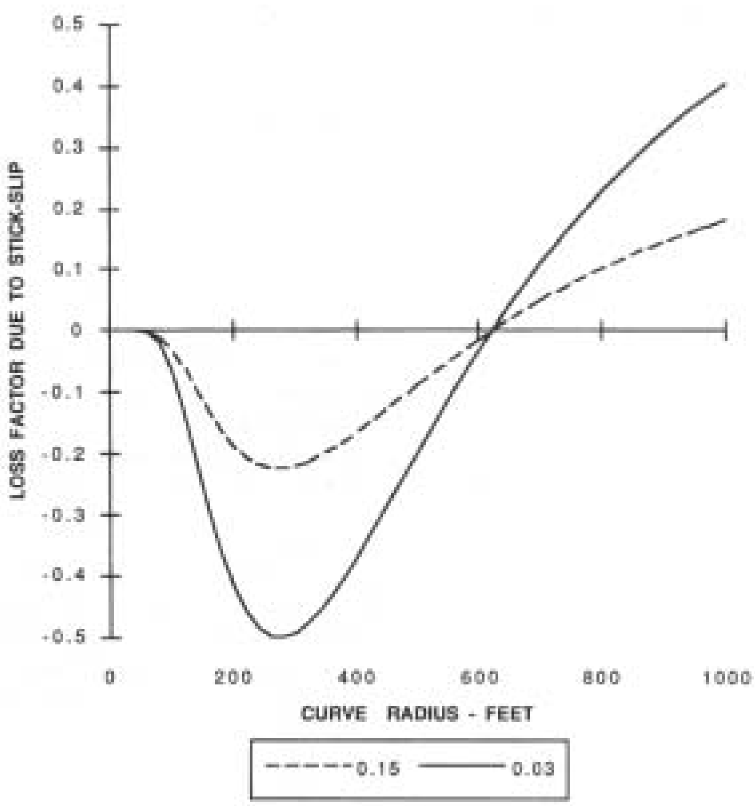



FIGURE 4-26 LOSS FACTOR AS A FUNCTION OF CURVE RADIUS FOR TWO LATERAL ACCELERATIONS

TABLE 4-1 LOSS FACTORS FOR RESILIENT AND DAMPED WHEELS

| Wheel Type | Frequency - Hz | | |
|---|---|---|---|
| | 500 | 1000 | 2000 |
| Bochum | .012 | .014 | .0072 |
| SAB | .0026 | .0016 | .0009 |
| Acousta-Flex | .012 | .0051 | .0034 |
| Ring damped | .0009 | .0004 | .0019 |

Reference: Remington, JSV 116(2), 1985

be greatly reduced, due to the change in friction characteristics caused by moisture. Water sprays are known to control squeal at curves, and lubrication of the flanges is often claimed to reduce squeal. In this latter respect, however, flange lubrication may only be effective to the extent that the lubricant migrates onto the rail head if squeal is due entirely to stick-slip between the tread and rail running surface.

The coefficient of friction varies substantially as a function of slip velocity for contacting metal surfaces of identical materials. Under vacuum conditions, steel or iron surfaces without contamination or oxidation will weld together if not maintained in relative motion, while contamination and oxidation prevent welding from occurring under normal atmospheric conditions (*27*). Regardless of contaminants, the static friction is normally greater than the dynamic friction coefficient for steel-steel contact. Under lateral creep in the presence of longitudinal rolling contact, the friction versus creep curve differs from one in which only sliding contact in a single direction is involved. Again, as discussed above, the friction versus creep curve for lateral creep in the presence of rolling contact is assumed to be smooth through zero lateral creep, reaches maximum at some finite creep, and thereafter decreases monotonously with increasing creep, producing the negative slope of friction versus creep discussed above.

Modification of the friction-creep curve is an attractive approach to noise control. Dry-stick friction modifiers applied to the wheel tread, and, thus, the rail running surface, improve adhesion and flatten the friction-creep curve, thereby reducing or eliminating the negative damping effect. A second possible means of modifying the friction versus creep curve is to employ metals of dissimilar metallurgical composition. An intriguing possibility is to employ alloy steel rails with sufficiently different metallurgical properties than those of the steel wheel treads. No data collection efforts nor tests have been conducted to investigate this approach, though some rail head inlays are reputed to reduce wheel squeal at curves. Nitinol (Nickel-Titanium) wheel treads have been considered for reduction of wheel squeal and improvement of traction, though no full-scale tests have been conducted.

### 4.3.2 Wheel/Rail Howl

Wheel/rail howl is another type of tonal noise produced by curving vehicles. The howl is contained within a frequency band ranging from about 250 to 750 Hz. An example of wheel/rail howl for BART Car 107 on ground rail is also included in Figure 4–20. On systems with steel centered wheels and tires, wheel howl is believed to be less of a problem, or nonexistent, because of a higher damping ratio for solid steel wheels. Because BART is the only system employing aluminum centered wheels, wheel howl may be unique to BART. The wheel howl appears to increase with train speed, especially in the example provided in Figure 4–20. Wheel squeal, on the other hand, may actually disappear with sufficient speed. Thus, the phenomenon of wheel/rail howl appears to be distinct from squeal.

#### *4.3.2.1 Mechanism of Wheel/Rail Howl*

Wheel/rail howl appears to be controlled by the fundamental lateral mode of oscillation of the wheel and low damping ratio for aluminum centered wheels with steel tires. The phenomenon probably involves lateral slip of the tire across the rail running surface, but is possibly not a sustained oscillation of the type associated with well-developed squeal, as evidenced by the larger bandwidth of the noise compared with that of wheel squeal. To the extent that howl is presumed to involve lateral slip, the process probably involves negative damping as does wheel squeal, for which the treatment would be essentially the same. That is, flattening the friction-creep curve or increasing the internal damping of the wheels should reduce the incidence of wheel howl at curves, as supported by the Car 107 tests at BART with the resilient Bochum wheel and a damped wheel (*28*).

#### *4.3.2.2 Relation to Rail Corrugation*

The howl appears to grow with time after rail grinding at large radius curves due to emergence of rail corrugation, and the corrugation formation is believed to be directly related to the vibration producing the howl. Thus, a regenerative effect may occur, where lateral oscillation over a broad frequency band encompassing the wheel's fundamental modal frequency produces rail corrugation with the same corrugation frequency. The corrugation further excites the wheel, increasing the amplitude of vibration of the wheel and radiated noise.

The process appears to occur at large as well as small radius curves. The howling noise also appears at tangent track, usually associated with corrugation. One might conclude, therefore, that, at least at BART, the wheel's dynamic properties which contribute to howl at curves are also related to formation of corrugation at tangents and moderately curved track. That is, the wheel howl at curves is controlled by resonances in the wheel which are also active at tangent track in the formation of short-pitch corrugation through dynamic lateral slip.

Fortunately, most rail transit systems do not employ composite aluminum centered wheels, and the problem may be limited to those systems that do. The most attractive mitigation measure is to use wheels with higher damping ratio, such as solid steel wheels. Wheels with tapered treads and sufficient longitudinal compliance in the primary suspension will tend to self-steer, align the axle with the curve radius, and, thus, reduce or eliminate lateral slip and howl. Wheel vibration absorbers, damped wheels, and resilient wheels, also reduce or eliminate wheel howl.

## 4.4 SPECIAL TRACKWORK

Special trackwork includes switches and crossover diamonds. Impact noise generation occurs as the wheel traverses a switch frog or crossover diamond gap. A second source of impact noise is the vertical ramp at the switch points, which typically exhibits a lot of damage because of wheel impact forces. Impact noise from special trackwork is very noticeable to wayside receivers and transit patrons, with A-weighted maximum noise levels roughly 7 to 10 dBA (fast sound level meter response) greater than levels at tangent track with ground rail. Special trackwork noise will generally be greater than that due to rail joints, because of the much larger gap that must be traversed. The noise radiation characteristics from special trackwork are significantly different from those of a long train on ground rail. For one, special trackwork noise radiation is primarily confined to the vicinity of the frog and switch points. Secondly, at large distances from very long trains, the wheel/rail noise radiated from the tangent portions of the track may be comparable with or greater than that attributable to the switch frog and points.

The theory of impact noise generation at special trackwork frogs is represented by that for level rail joints with finite gap. The radiating surfaces of impact noise at special trackwork have not been studied in detail, and one must assume that the wheel, frog, and ties are all significant radiators. On concrete bridge decks, deck radiated noise may also be significant. If steel girders are included, substantial structure radiated noise may be expected at low frequencies in the range of 50 to 500 Hz. There is also the possibility of truck frame or other component radiated noise, though this has not been documented.

Special frogs such as moveable point frogs and spring frogs have been developed to reduce or eliminate impact forces and noise. These are discussed further in the chapter on trackwork treatments.

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## CHAPTER 5 SELECTION OF NOISE CONTROL TREATMENT

CHAPTER 5

# SELECTION OF NOISE CONTROL TREATMENT

## 5.1 INTRODUCTION

This chapter is a guide for identifying and evaluating noise control treatments with respect to noise reduction effectiveness, site-specific limitations, and cost. Treatments are segregated according to tangent track, curved track, and special trackwork noise control. In each of these categories, treatments are presented for onboard, trackwork, and wayside application. Some unavoidable duplication of discussion is involved, but the intent is to provide a step-by-step approach for identifying appropriate noise control treatments for each type of noise control problem. The chapter on generation of wheel/rail noise provides a theoretical and symptomatic description of various types of wheel/rail noise.

The block diagram presented in Figure 5–1 illustrates the relationship of various categories of wheel/rail noise: (1) rolling noise at tangent track, (2) curving noise, and (3) special trackwork noise. Rolling noise at tangent and moderately curved track without squeal is most representative of conditions used for qualification testing of transit vehicles. Normal rolling noise with smooth rails and trued wheels, excessive rolling noise resulting from excessive random rail and wheel roughness, impact noise resulting from rail and wheel imperfections and joints, and noise resulting from short-pitch rail corrugation are normally associated with tangent track noise. Curving noise primarily involves wheel squeal and, perhaps, wheel howl, in addition to rolling noise. The discussions concerning tangent track noise should be referred to when dealing with rolling noise at curved track. Noise from special trackwork includes impact noise generated by wheels traversing gaps in trackwork components, specifically switch frogs and crossovers, and the associated treatments include special trackwork components that minimize this type of noise.

The first step in selecting noise control treatments is to identify the type of noise by the measurement of noise levels, spectral analysis, observation, and listening. For example, there should be no difficulty distinguishing between squeal and rolling noise. However, distinguishing between wheel howl and corrugated track noise may be very difficult without inspecting the track to determine whether corrugation is present. Similarly, abnormally high rolling noise may be difficult to identify without assessing wheel and rail condition; rough wheels or rails directly produce high levels of noise. Once the particular type of track and associated noise are identified, the user may concentrate on the options listed under the general headings of onboard, trackwork, and wayside treatments, which are discussed and compared in tables. Detailed discussions of onboard, trackwork, and wayside treatment options are provided in later chapters.

Any treatment selected for noise control should be carefully reviewed by the transit system engineering staff for cost, practicality, and safety. Further, for noise control treatments that have not received widespread application in the United States, or treatments involving custom fabrication or modification of vehicles, prototype tests should be conducted to better determine actual noise reductions which may be reasonably expected and to identify limitations in application.

Representative order of magnitude costs are provided for a comparison of various noise control treatments. These costs are listed to aid in the selection process, and the user should verify costs with suppliers and contractors before selecting or rejecting a treatment. Further, non-noise related costs of the treatment should be considered as well, because there may be ancillary benefits which may mitigate the cost of the treatment. The estimated costs are not vendor or contractor bid prices; actual prices must be obtained directly from suppliers.

## 5.2 TANGENT TRACK NOISE

Tangent track noise includes (1) normal rolling noise, (2) excessive rolling noise, (3) impact noise, and (4) corrugated rail noise. The selection of a noise control treatment depends on the type of noise. For example, damped wheels are effective in controlling squeal, but historically have produced little reduction of tangent track rolling noise at U.S. transit systems. As another example, *normal* rolling noise at tangent track would not be reduced significantly by rail grinding or wheel truing because the rails and wheels already would be in good condition (though some minor noise reduction might still be expected). Therefore, the user must identify the type of noise before deciding on a treatment scenario.

### 5.2.1 Normal Rolling Noise

Normal rolling noise occurs for smooth ground rail with optimum rail and wheel profiles. The rail will appear smooth and free of spalls, pits, shelling, and corrugation. The con-

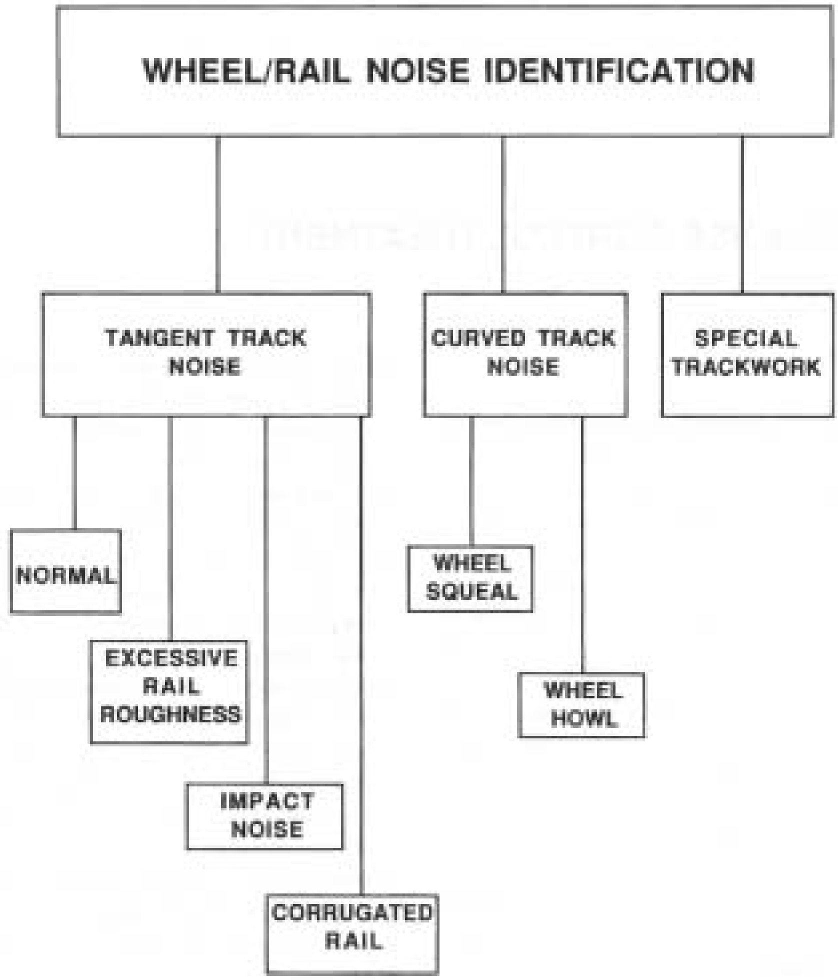


FIGURE 5-1 CATEGORIZATION OF WHEEL/RAIL NOISE

tact patch width will be uniform in width, without plastic flow at the edges, and will be about ½ to ¾ in. A straightedge placed longitudinally along the rail running surface will indicate a uniform rail height profile when backlit. There should be a minimum of flange contact with the gauge face, and there should be no two-point contact wear patterns on the rail head. The contact wear strip ideally should be centered on the rail head, over the stem, or, at most, centered ½ in. to either side of the rail center if variation of contact location is desired to avoid tread rutting. The ball of the rail should be radiused, and the wear pattern should not encounter the edge of the radiused portion of the ball. The wheel tread will be smooth, without pits, spalls, polygonalization, or other imperfections, and the tread profile will not be worn significantly.

The maximum passby level generally will be consistent with the maximum A-weighted noise levels indicated in Figure 5–2. The levels shown are for eight-car rapid transit trains, such as those at BART and WMATA, with solid steel or aluminum centered wheels; lower levels of noise may be expected with shorter trains. The passby noise will appear to be uniform from one vehicle to the next and will not exhibit harsh pure tones or "roar" resulting from corrugation or other imperfections. The passby noise level signature will vary smoothly with time, with each wheel set contributing a similar amount of noise energy. In contrast, a single flatted wheel will produce as much as a 7 to 10 dB higher noise level in the open on at-grade or aerial structure track than each of the remaining wheels and will be clearly identifiable during passage. Rail grinding and wheel truing will not greatly reduce normal rolling noise levels because the running condition of the rail and wheels already should be good. There may be some optimization of rail head contour and wheel profile that may reduce noise, but this likely will be limited to a few decibels or less.

Table 5–1 lists various options available for controlling normal rolling noise. Expected noise reductions, costs, and site-specific limitations also are listed. The options available for noise control are not extensive and are primarily concentrated in the area of wayside and onboard applications.

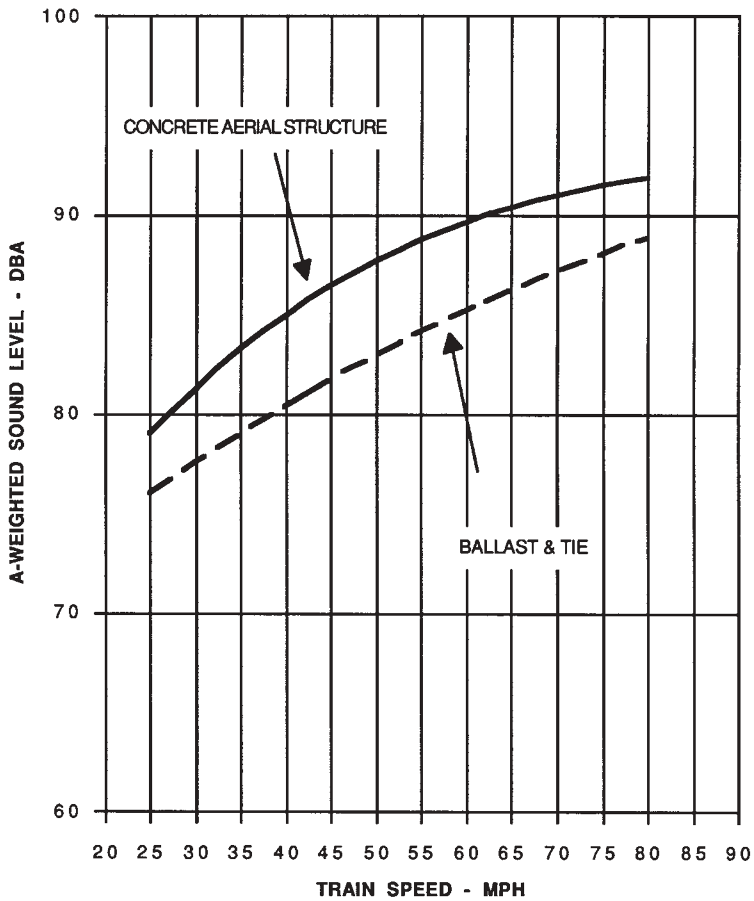


**FIGURE 5-2 MAXIMUM PASSBY LEVELS FOR 8-CAR TRAIN AT 50 FEET FROM TRACK CENTER WITH GROUND RAIL AND TRUED WHEELS**

*5.2.1.1 Onboard Treatments*

The onboard treatment options available for controlling normal rolling noise are limited primarily to vehicle skirts, undercar sound absorption, enhancement of car body sound transmission loss, and, to a lesser extent, resilient wheels, though the latter option's noise reduction is limited to 1 to 2 dB. Damped wheels are not considered to be effective because the maximum A-weighted noise reduction observed for typical transit application has been about 0 to 1 dBA. Similarly, although onboard dry-stick lubricants have been promoted by various manufacturers as an effective reducer of rolling noise, data collected at BART do not demonstrate this and thus are not included here.

*Vehicle Skirts*. Vehicle skirts located about the trucks may reduce wayside noise by up to 2 dB if combined with sound absorption treatment applied to the interior surfaces of the skirts. The skirts must deflect and absorb wheel radiated noise and may be most effective in controlling squeal as opposed to rolling noise. Skirts should be less effective on ballast-and-tie track than on direct fixation track because of the absorption provided by the ballast. Skirts are likely to be ineffective in reducing noise radiated by the rails. Costs for skirts are estimated to be between $5,000 and $10,000 per vehicle. Limitations of vehicle skirts concern clearance for the third rail, and there may be an impact on vehicle maintenance because they limit access to the area of the truck. Systems currently using skirts include the Denver and Portland light rail systems, the latter with the procurement of low floor height vehicles. Vehicle skirts would be effective systemwide.

*Undercar Absorption.* Undercar sound absorption may yield limited interior and exterior noise reductions, on the order of 2 to 3 dB, if applied to the underside of the floor over the truck. Attractive features of undercar sound absorption are the fact that (1) it is reasonably inexpensive and (2) it would be effective systemwide. However, there may not be sufficient free area under the car to treat, and the treatment may interfere with vehicle maintenance. Costs for treatment are estimated to be about $10 per square foot, or about $3,500 per vehicle, if 50% of the entire underside of the floor is treated. These costs could be considerably higher if installation is difficult or if complex attachment and retention methods are required.

TABLE 5-1 NOISE CONTROL OPTIONS FOR NORMAL ROLLING NOISE AT TANGENT TRACK

| LOCATION | TREATMENT OPTION | NOISE REDUCTION - dB - | COST | SITE SPECIFIC LIMITATIONS |
|---|---|---|---|---|
| Onboard | Undercar Absorption | 3 | $3,500/vehicle | May not be effective for ballasted track |
| | Resilient Wheels | 1 to 2 | $2,400 to $3,000/wheel | May not be appropriate for tread braked systems |
| | Skirts | 1 to 2 | $5,000 to $10,000 per vehicle | Third rail clearance. Should be combined with undercar sound absorption |
| | Undercar sound absorption | 2 to 3 | $3,500/vehicle | Applicable to interior noise. Application will be limited by undercar equipment, but should be provided over trucks |
| | Car Body Insulation | 0 to 5 dB | Original equipment | Includes floors, windows, door seals. No site-specific limitations |
| Track-work | Trackbed Absorption | 5 | $10/sf | Ineffective for ballasted track |
| | Between track barriers | 3 to 5 | $50 to $100/tr-ft | Must be combined with train-way ceiling absorption |
| | Rail Vibration Absorbers | 1 to 2 | $500/ft | Requires clearance between rail and invert. Concern over falling from aerial structures. |
| Wayside | Sound Barriers | 5 to 10 | $15 - $20/sf | Adverse topography. May be adverse visual characteristics |
| | Absorptive Barriers | 7 to 12 | $25/sf | Less effective for ballast and tie track |
| | Earth Berms | 7 to 12 | Site dependent | Attractive for landscaping. |
| | Depressed Grade | 5 to 10 | Site dependent | |
| | Receiver Treatment | 0 to 10 | $5,000 to $10,000/receiver | May require forced ventilation. May encounter structural deficiencies, code violations, and pest damage |
| | Enclosure | 10 dB | Unknown | Impact on fire control, evacuation, and ventilation |
| | Subway wall treatment | 5 | $7-10/sf | |
| | Station treatment | 5 to 10 | $7 to $10/sf | |
| | Fan and Vent Shaft acoustical absorption | 5 to 15 | $7-10/sf | |

*Resilient Wheels.* Resilient wheels may provide about a 1- to 2-dB reduction in wayside and car interior noise. However, their use would likely be driven by the need to control wheel squeal at curves, where they are most effective. Resilient wheels, as a rule, should not be used solely to control rolling noise. Another benefit of resilient wheels is reduced truck shock loading. Costs for resilient wheels, which vary by manufacturer and by application, range between $2,000 and $3,000 per wheel, considerably higher than the cost of about $400 to $700 for solid steel wheels.

*Car Body Sound Insulation.* Car body sound insulation, which is necessary for reducing car interior noise, is normally a provision in standard car procurements. Car body sound insulation is controlled by the car body shell, floor, windows, doors, and connections between the trucks and vehicle body. Effective car body designs include a composite double layer shell and liner with glass fiber sound absorption, a composite floor with a resilient floor covering, acoustically rated glass windows, and effective door seals. Car interior noise control for normal rolling conditions is best achieved by specifying car interior noise levels for new car procurements. Example specifications are provided in the APTA Design Guidelines. Older vehicles may have to wait for rehabilitation before they can be treated. Shop treatments include repair or replacement of door seals.

*5.2.1.2 Trackwork Treatments*

Trackwork treatments include sound absorption at the track level, perhaps between the rails; rail vibration absorbers; low-height barriers between tracks; and any other measure for which the track maintenance department would be responsible. A brief discussion of each of these treatments follows, and more detailed discussions are provided in Chapter 8.

*Trackbed Absorption.* Trackbed absorption is effective for direct fixation track with concrete inverts or slabs, such as at concrete aerial structures. Noise levels at ballast-and-tie track are normally 4 to 5 dB lower than at aerial structure and concrete slab track with direct fixation fasteners, ostensibly because of the sound absorption provided by the ballast. Additional trackbed sound absorption would be ineffective at ballast-and-tie track. There may be substantial maintenance problems associated with sound absorption treatments positioned beneath the train in exposed situations. Such problems may involve the ability to inspect and maintain track components. Debris may accumulate beneath the absorption, making cleaning of the invert difficult. The treatment would be effective for station platform areas and areas where debris would not accumulate. The absorption must be protected from tunnel washing machines and other maintenance equipment which might otherwise damage the treatment.

Candidate treatments include (1) Tedlar-encased glass-fiber board of density 3 psf, protected by perforated sheet metal or fiber reinforced panels; (2) spray-on cementitious sound absorption; and (3) ballast. In the case of ballast, electrical insulation may be compromised if the ballast extends to the top plate of the fastener. A variant of trackbed absorption is underplatform absorption for stations. There usually is a recess under the platform edge which is called the "suicide pit," and underplatform absorption placed against the far wall of this recess and the underside of the platform overhang provides a particularly effective means for controlling station platform noise levels in subways. This is discussed in greater detail with respect to wayside treatment.

*Rail Vibration Absorbers*. Rail vibration absorbers are an intriguing noise reduction treatment. Vibration absorbers are spring-mass systems with damping incorporated into the spring to absorb and dissipate vibration energy. They are attached to the rail with clamps, without contacting the invert or ballast. Vibration absorbers may be tuned by the absorber manufacturer to optimize dissipation of rail vibration energy into heat over a particular range of frequencies and may be particularly desirable at locations where a sound barrier would be impractical and the needed noise reduction is on the order of a few decibels. The unit cost for rail vibration absorbers is expected to be on the order of $50 to $100 per absorber. Assuming that one absorber is applied in every space between fasteners at both rails, and that the fastener spacing is 30 in., the cost per track foot would be on the order of $40 to $80. Placement of absorbers at every other or every third fastener spacing may be possible, and tests should be conducted to determine effectiveness. The rail vibration absorber may be particularly effective in reducing the pinned-pinned mode response of the rail associated with fastener pitch.

*Between-Track Barriers*. Barriers positioned between tracks can reduce station platform noise levels. Both sides of the barrier should be lined with sound absorbing material, such as 2 in. of glass fiber of weight 3 pcf. Cementitious panels with sound absorbing properties have been proposed. Barrier height should extend to the floor level of the transit vehicle. There is a safety issue concerning entrapment of track inspection personnel or patrons caught in the trainway.

*Resilient Rail Fasteners.* Resilient fasteners are not normally considered a treatment for wheel/rail noise. They are designed to reduce low-frequency groundborne or structure-borne noise above about 30 Hz and can be effective in reducing wayside noise radiated from steel elevated structures and aerial structures with steel box girders. Included in this category are the Stedef and Sonneville twin booted tie systems. Resilient fasteners with elastomer springs have been proposed for reducing wheel/rail noise radiation by using the damping properties of the elastomer. Further, adding bonded resilient fasteners to wood tie elevated structures may reduce secondary impact noise radiation caused by an otherwise loose system consisting of tie plates and cut-spikes. The resilient fasteners provide rail support without looseness and, therefore, may reduce noise related to impact between the rail and tie plate. However, the damping provided by the elastomer may be the principal noise reducing agent, because the tight rail support with damping would be effective in reducing vibration transmission along the rail which otherwise might be free to vibrate. Systems that use concrete ties with spring clips may not benefit from the use of resilient fasteners, because the spring clips already eliminate any looseness between the rail and tie.

#### *5.2.1.3 Wayside Treatments*

Wayside treatments for normal rolling noise include sound barriers, absorptive sound barriers, and receiver sound insulation. These are the most effective treatments available for reducing normal rolling noise at wayside receivers. Although not listed, shifting the alignment is always a possibility, though this option is possible primarily for new construction in which such an option can be exercised at the design stage. Alignment shifting is not considered an option for existing systems because of the severe cost and disruption of service. Also included are station and subway treatments for controlling noise received by the transit patron and ventilation and fan shaft treatments for controlling noise radiated to nearby residences. Brief descriptions of these treatments follow; see Chapter 9 for more detailed discussion.

*Sound Barrier Walls*. Sound barrier walls are the most effective treatment for controlling normal rolling noise in wayside areas. Sound barrier walls may be treated with sound absorbing materials to enhance their effectiveness, though at considerable cost. The site-specific limitations of sound barriers include (1) lack of sufficient access for wayside maintenance vehicles at at-grade track, (2) lack of sufficient distance from track center to guarantee safety of individuals that might be trapped between the track and wall, (3) source and receiver elevations that may not be appropriate for achieving effective noise reduction for a practical barrier height, (4) high sound barriers that may be unattractive or undesirable, and (5) high wind loads, steep grades, and poor soils that may require substantial foundations. The cost of a typical sound barrier wall in 1994 was about $15 to $20 per square foot, though costs as high as $40 per square foot have been encountered for masonry walls. Thus, a typical 8-ft-high sound barrier wall could cost about $160 per lineal foot of track. There may be additional costs for landscaping, maintenance, and additional architectural features. Where transit systems pass through residential neighborhoods within 35 ft of residential structures, center barriers may be useful to control far-track noise at second-story receivers. Assuming that a center barrier would be about 6 ft high, the cost may be about $120 per lineal foot, bringing the total cost for barriers to $280 per track foot. Barrier costs may vary considerably from one locale to another and may be driven by aesthetic considerations and topography; therefore, barrier costs could be as high as $40 per square foot when landscaping and drainage are included. Nevertheless, sound barriers can provide a visual as well as physical separation between a wayside residential community and a transit corridor, which may be highly desirable to the community.

*Berms*. Berms are another form of sound barrier which are visually attractive, provide a safety barrier against derailment, and are believed to offer more noise reduction than a simple nonabsorptive solid wall of the same height. Berms can be attractively landscaped to add a natural look to the ground between source and receiver. Reinforcement can be incorporated into the berm to allow high aspect ratios where insufficient space exists for the customary 1 in 2 slope of the berm.

*Absorptive Sound Barriers*. The track side of sound barrier walls may be treated with sound absorbing materials to improve barrier performance and reduce or eliminate reflections. Absorptive sound barriers are most effective at concrete invert track with direct fixation; the noise reduction obtained at ballast-and-tie track may be limited by the existing absorption provided by the ballast if the barrier is close to the vehicle. A careful analysis should be performed to determine actual effectiveness, and prototype testing is desirable. There are two conditions to consider: (1) a barrier at a large distance from the track and (2) a barrier very close to the track.

In the first case, sound absorption is expected to improve the insertion loss of simple barriers, though the improvement may be limited to 2 to 3 dB. In the second case, multiple reflections between the barrier face and vehicle body will degrade barrier insertion loss. Sound absorbing material applied to the face of the barrier facing the train will reduce or eliminate multiple reflections and improve barrier performance. Such treatment is particularly attractive for bridges, viaducts, and aerial structures where barrier weight is a critical design factor and where structures with direct fixation track would otherwise provide little sound absorption.

Absorptive barriers may be useful in situations in which high barriers are needed on either side of the track or tracks and where there is a need to reduce the reflection of sound from the more distant barrier. Sound absorption applied to the face of the barrier would cost about $10 per square foot, which is added to the initial cost for the barrier of about $15 to $20 per square foot. Savings can be obtained for absorptive barriers constructed of sheet metal and 4 in. of glass fiber, powder coated to provide an attractive finish and protection against the weather and oxidation. An advantage of the sheet metal barrier is its low weight, which may be attractive for bridges and aerial structures. A “living wall,” a type of retained earth barrier with very steep or vertical sides which support plant growth, may be attractive. Only the upper 3 to 5 ft of the barrier may need to be treated if reflections are not of concern. A variant of this approach is the addition of an absorptive crown to the barrier, which would increase barrier height, though less than that required for a nonabsorptive barrier. Examples of this last approach have been used in Japan.

*Aerial Structure Barriers.* Barriers are commonly mounted on the edge of aerial structures just outside the dynamic clearance envelope of the train. Examples of such installations include those at BART and WMATA. The height of the barrier may be limited to that of the vehicle floor to allow passenger rescue over the top of the barrier. The barrier may be treated with spray-on cementitious sound absorption materials to enhance the performance of the wall. The expected noise reduction typically is 7 to 10 dB, especially if the space between the track can be closed off, making the absorptive barrier one of the most effective treatments available for controlling wayside noise. Addition of a center barrier would further reduce noise from far-track trains, though such a use may be prevented if a center walkway is needed or if entrapment is a problem. A center wall is not normally employed on double track aerial structures.

*Enclosures.* Aerial structure track has been enclosed to reduce wayside noise in Hong Kong. This approach requires consideration of fire control, evacuation, and ventilation, as with any subway. Costs for track enclosures are not known, but are likely to be substantial. Sound absorption can be applied to the interior surfaces of the enclosure to further reduce wayside noise and to reduce car interior noise. Sound absorption would be particularly important if there are substantial openings in the enclosure and in the aerial structure between the tracks.

*Depressed Grade or Open Cut*. Depression of the track in an open cut provides significant noise reduction in a manner analogous to that of high barriers on either side of the track. Treating the walls of the open cut with sound absorbing material is very beneficial in reducing or eliminating multiple reflections, particularly for elevated receivers which may look down on the open cut. Without sound absorption, the sound eventually would be radiated upward and away, possibly toward sensitive receivers. Ballasted track in open cuts produce lower noise levels than direct fixation track, thus reducing the need for additional absorption on the walls of the cut. Costs for open cuts relative to at-grade track have not been determined. Use of an open cut may be driven by the desirability of a grade separation for traffic flow or aesthetic considerations.

*Station Treatment.* Station treatment includes application of sound absorption to underplatform surfaces, station ceilings, and station walls. Effective underplatform treatments include spray-on cementitious sound absorbing materials and 3-pcf glass fiber encased in 3-mil-thick Tedlar plastic and perforated fiber reinforced plastic sheet or sheet metal. The rear walls of suicide pits and the underside of the platform can be treated, producing very effective results. Station walls and ceilings may be treated with acoustical ceiling panels, subject to provisions for air pressure transient loading. In the trainway, ceiling and wall treatments should be mounted flush against the ceiling without air gap to avoid stresses induced by dynamic air pressure loading or buffeting as the train enters and leaves the station. Ceilings and walls also may be treated with spray-on cementitious sound absorbing materials. The spray-on cementitious treatments can be applied in an architecturally appealing manner, and substantial experience has been gained with the application of these treatments. Costs for station treatment are difficult to assess, but are likely to be on the order of $7 to $10 per square foot, depending on labor rates.

*Subway Wall Treatment*. In subways with direct fixation track, treating the upper half of the subway walls and the entire ceiling with sound absorbing materials will reduce car interior noise. The treatment is especially desirable where vehicle windows are often left open for ventilation or where there is substantial sound transmission through the car body or doors. Subway wall treatments also reduce station platform noise levels caused by approaching trains and subway ventilation fans. Ballast provides substantial sound absorption; therefore, the addition of tunnel wall and ceiling absorption in tunnels with ballasted track will have much less effect than in tunnels with direct fixation track. An example of extensive tunnel wall treatment with cementitious sound absorption includes the MBTA.

Subway wall treatments consisting of spray-on cementitious sound absorbing treatment are practical and effective. Alternative treatments include 3-pcf glass-fiber board protected by a 3-mil-thick plastic film with a perforated sheet metal or fiberboard cover. Sound absorbing porous glass blocks also have been used, though these tend to be more costly than cementitious absorption and are difficult to procure. Costs for subway wall treatment with cementitious spray-on treatments are on the order of $7 to $10 per square foot. Treatments with 3-pcf glass fiber may be on the order of $10 per square foot, and encasement in Tedlar and perforated sheet metal facings will increase the cost. Cost estimates should be obtained and carefully reviewed.

*Fan and Vent Shaft Treatment*. Wheel/rail noise emanating from fan and vent shafts in normally quiet residential areas can be controlled with sound absorbing materials applied to the walls and ceilings of the shafts. For fan shafts, in-line sound attenuators may be employed to reduce both train and fan noise, the latter being the most objectionable. Effective treatments include 1-in.-thick, spray-on cementitious sound absorption material, sound absorbing porous glass foam block, and 1-in.-thick, 3-pcf glass-fiber board with perforated cover. Typical costs for treatment are about $10 per square foot. Details of shaft treatment are provided in Chapter 9.

*Receiver Treatments.* Receiver treatments include replacement of single pane fenestration with acoustically rated fenestration, weather stripping, and provision of forced air ventilation. Receiver treatments may not be practical if the structure is already weatherproofed and has thermally insulating glass. The building construction should be inspected to determine if there would be any benefit achieved by treating a receiver directly. Treating a residence, which involves the residents in the design process, may require relocation of the residents during treatment. Further, receiver treatments may involve bringing an existing structure up to local code requirements or may require replacement of pest damaged portions of the structure. Receiver treatments do not reduce exterior noise, where residents may spend varying amounts of time. Thus, receiver treatments are best avoided if a sound barrier or other treatment can be employed successfully. Treatment of upper story receivers may still be necessary near track where sound barriers would otherwise be unreasonably high. Thus, a transit system designer may explore a combination of barrier and receiver treatments to optimize overall performance and cost.

### 5.2.2 Excessive Rolling Noise

Excessive rolling noise resulting from random roughness is caused by rough rails and wheels, but the rail is without identifiable rail corrugation, joints, or other large imperfections in the running surface. Excessive rolling noise would normally arise after a period of no rail grinding or wheel truing, and rail condition should be visibly deteriorated with pits, fatigue cracking, and gauge corner metal plastic flow. Excessive rolling noise may occur without obvious visible defects, resulting simply from high amplitude random roughness, flat rail head, improper cant, and rail grinding pattern. Excessive rolling noise also may exist despite rail grinding, where the rail grinding is minimal or does not provide a smooth, uniform contact wear pattern edge definition.

Candidate treatments for controlling excessive rolling noise are listed in Table 5–2. Discussion of these treatments with respect to onboard, trackwork, and wayside application follows. All of the treatments for normal rolling noise are applicable to excessive rolling noise, though treatments for normal rolling noise usually should not be applied unless needed after the treatments identified in Table 5–2 are considered. The treatments listed in the table are effective, particularly against the wheel and rail conditions which produce excessive noise, and should be explored before resorting to treatments for normal rolling noise.

#### *5.2.2.1 Trackwork Treatments*

The principal trackwork treatment for excessive rolling noise is rail grinding. The trackwork treatments indicated for treating normal rolling noise also are applicable, but should be considered only after grinding the rail.

**TABLE 5-2 NOISE CONTROL TREATMENTS FOR EXCESSIVE ROLLING NOISE WITHOUT CORRUGATION**

| TREATMENT LOCATION | TYPE OF TREATMENT | NOISE REDUCTION (dB) | PROBABLE COST | COMMENTS |
|---|---|---|---|---|
| Trackwork | Rail Grinding (In combination with wheel truing) | 7 to 10 | \$5/trk-ft/yr[1] | |
| | Defect Welding and Grinding | 5 | \$200/defect | |
| | Joint Maintenance and tightening | 5 | \$6/joint/yr[1] | |
| | Joint Welding | 5 | \$600/joint[1] | |
| | Bonded elastomer resilient rail fasteners | 5 to 10 | \$50/fastener | Reduces structure radiated noise and impact noise generated by loose fixation |
| | Treatments listed above for normal rolling noise control | | | |
| Onboard | Wheel truing (in combination with rail grinding) | 7 to 10 | \$60/wheelset[1] | Cost of wheel truing machine is \$600,000 to \$1,200,000 |
| | Slip-slide control | | \$5,000-\$10,000 /vehicle[2] | Reduces incidence of flats |
| | Treatments listed for normal rolling noise | | | |
| Wayside | Treatments listed for controlling normal rolling noise | | | |

1 L.R. Kurzweil et al., **Noise Assessment and Abatement in Rapid Transit Systems** (1974) pg. 3-10

2 Recent bid spreads for retrofit at rail transit systems

*Rail Grinding.* Rail grinding in combination with wheel truing is the most effective means of controlling noise caused by excessive rail roughness. Rail grinding should be optimized to reduce fatigue and wear of the running surface, which would lead to excessive roughness and noise. There is disagreement in the literature concerning the best profile for obtaining the least noise. Widening the contact patch has been conjectured to reduce net contact dynamic forces by averaging the rail and wheel roughness over the contact patch area. On the other hand, increased rail conformity increases spin-creep, possibly contributing to rail corrugation, wear, and, thus, noise. At present, "squaring up" the contact patch appears to be a reasonable approach for controlling wayside noise, subject to further study.

Increased conformity resulting from wheel tread wear can be controlled with wheel truing. If wheel truing is insufficiently frequent such that concavity is prevalent in the tread profile, there may be other problems besides spin-creep, such as poor ride quality and steering. In all cases, the track should be ground and the wheels trued to prevent two-point contact on the running surface of the rail. Rail grinding may actually increase wheel/rail noise if grinding introduces a periodic grinding pattern in the rail head, a condition which should be avoided, because there is no guarantee that the pattern will be entirely worn away with time. Visual evidence of the grinding pattern may disappear with time, but undulation and residual hardness variation may persist.

Grinding equipment must be selected and maintained in good condition to avoid tool chatter and debris accumulation in the grinding wheels, which can lead to excessively deep grinding patterns. The wavelength or pitch of any grinding pattern should be reduced to less than the contact patch longitudinal dimension by reducing grinding train speed. Patterns in narrow ($\frac{1}{16}$ to $\frac{1}{8}$ in. wide) grinding facets, produced by multiple stone grinders and multiple passes to shape the head, will be averaged over the contact width, thus reducing noise. Wide grinding facets, on the other hand, will prevent such averaging. Site-specific limitations for rail grinding include (1) lack of clearance in tunnels for the grinding machine, (2) lack of track access because of conflict with revenue operation, and (3) inability to grind certain kinds of track, such as embedded curves.

### *5.2.2.2 Onboard Treatments*

Wheel truing, the principal onboard treatment for controlling excessive rolling noise, should be considered before resorting to treatments identified for normal rolling noise.

*Wheel Truing.* Wheel truing, the most effective onboard treatment for controlling excessive rolling noise caused by wheel roughness, may be considered a necessary part of a vehicle maintenance program. Systems which do not have effective wheel truing programs probably experience abnormally high rolling noise. Current information indicates that there is little difference between the type of wheel truing machine used (e.g., milling or lathe) and the resulting noise level.

Noise reductions on the order 7 to 10 dB may be expected for initially rough wheels if the rail also is ground, though actual noise reductions will depend on the state of roughness of both the rails and wheels. Truing severely polygonalized wheels or wheels with extensive and severe wheel flats may result in much greater rolling noise reductions. Wheel truing without an effective rail grinding program may not achieve the lowest noise levels possible, because a rough or severely corrugated rail may partially or completely mask the noise control benefits of wheel truing. However, even without rail grinding, wheel truing will help maintain a quiet system and should be performed.

There are no apparent site-specific limitations for wheel truing, other than availability of facilities. Some transit engineers have expressed a concern over the ability to true resilient wheels, which suggests that the wheel truing machine manufacturer should warrant that its equipment is capable of truing resilient wheels. However, transit systems such as the Portland Tri-Met and Los Angeles Blue Line regularly true resilient wheels with no reported difficulty.

The cost of wheel truing includes the cost of the wheel truing machine and its maintenance, materials such as cutting tools, and labor. The cost of a typical truing machine is on the order of $1 million, with the milling type of machine being the less expensive (and less accurate) than the lathe type. Selection of a wheel truing machine is important. Monomotor trucks do not allow slip between front and rear axles; therefore, wheel diameters must be maintained to close tolerances. The truing tolerance of milling- and lathe-type truing machines should be carefully reviewed before selection if monomotor trucks are involved.

Another consideration is the amount of tread metal removal required to clean up the flange. BART has indicated informally that the milling machine requires more metal removal per truing operation than the lathe machine. Thus, the upfront cost savings realized by purchasing a milling machine instead of a lathe machine may be more than canceled by additional costs for wheel replacement. Metal removal and overall operating costs savings should be carefully reviewed with the truing machine manufacturer before selecting a particular machine. Finally, the wheel truing machine operator should be a qualified machinist capable of precise milling and cutting operations, and willing to focus on the truing process.

#### *5.2.2.3 Wayside Treatments*

Wayside treatments for controlling normal rolling noise also are effective in controlling excessive rolling noise. Where rail grinding cannot be performed because of (1) geometric limitations, (2) an inadequate grinding budget, or (3) lack of equipment, the remaining choices may be provision of a sound barrier wall or berm, receiver treatment with acoustically rated windows and provision of forced air ventilation, or enclosure of the track. These treatments are likely to be more costly than procurement of necessary rail grinding and wheel truing equipment if a large number of wayside receivers are involved. Further, more expensive wayside treatments generally would be required without rail grinding and wheel truing than if rails and wheels are maintained in good condition.

### 5.2.3 Impact Noise Control

Impact noise may be the most significant source of noise at transit systems where rail grinding and wheel truing are not performed or are performed on an infrequent basis. The causes of impact noise include chips, spalls, burns, rail joints, and excessive curvature of the rail surface in the longitudinal direction. These causes are described in Chapter 4. Remedies for impact noise follow. A summary of impact noise control treatments is presented in Table 5–3. These treatments are applicable to rolling noise at both tangent and curved track. Treatments for impact noise caused by special trackwork are discussed in a separate section that follows.

#### *5.2.3.1 Trackwork Treatments*

Treatments for impact noise include rail grinding, defect welding and grinding, joint maintenance, field welding of joints, and elimination of loose track supports. These should be considered before resorting to trackwork treatments identified for normal rolling noise.

*Rail Grinding*. Rail grinding is by far the most effective means of controlling impact noise caused by rail defects. The rail grinding machine should be able to control rail height uniformity over a length of about 6 ft to eliminate impact noise caused by excessive rail height curvature in the longitudinal direction. The horizontal axis single-stone grinder may not be appropriate for controlling rail height curvature unless provision is made for controlling stone height relative to reference points before and after the stone. The grinder should remove enough metal to eliminate fatigue cracks, pits, spalls, chips, and burns. Overgrinding may be desirable to reduce stress concentrations and hardness variation. After an initial deep grind to remove defects, the rail head should be recontoured to maintain proper contact patch width and location. After this process is complete, the rail should be regularly ground to maintain a smooth running surface.

*Defect Welding and Grinding.* This procedure involves deposition of weldment to an engine burn, using an electric arc, and grinding the weldment to achieve a smooth running surface. Field welding costs are estimated to be $200 per defect, assuming 2 hours per defect with two track maintenance workers at $50 per hour. Site-specific limitations may involve weldability of alloy steels and track access. The rails should be inspected before treatment to determine if other more serious defects exist, such as fatigue fracture of the body of the rail, in which case rail replacement may be the

TABLE 5-3 TREATMENTS FOR IMPACT NOISE DUE TO RAIL DEFECTS

| TREATMENT LOCATION | TYPE | NOISE RED'N - dBA - | COST - $ - | COMMENT |
|---|---|---|---|---|
| Trackwork | Rail Grinding | 7 to 10 | $1,000 to $7,000/ track-mile | Must be done in conjunction with wheel truing. Cost based on labor and overhead for 2 persons at $2,000/day and grinding 2 miles of track per shift. Costs vary substantially. |
| | Defect welding and grinding | 0 to 3 | $200/defect | Noise reductions depend on number of defects. Costs are subject to local labor rates and field conditions. |
| | Joint Maintenance | 2 to 3 | $200 to 400/joint | Primarily relevant to older transit systems with steel elevated structures. |
| | Field welding of joints | 5 | $600/joint | Ancillary cost benefits in reduced maintenance. |
| | Eliminate rail support looseness | 5 | $66 to $100/tr-ft | Achieved with resilient direct fixation fasteners or concrete ties with spring clips. Primarily relevant to steel elevated aerial structures. |
| Onboard | Wheel truing | 7 to 10 | $60/ wheelset[1] | Single most important treatment, because wheel flats are most significant cause of impact noise. |
| | Slip-slide control | 7 to 10 | $5,000 to $10,000/ vehicle | Reduces flat occurrence by about 50%, and thus reduces wheel truing costs proportionately. |
| Wayside | Treatments for normal and excessive rolling noise | | | |

[1] L. G. Kurzweil et al., Noise Assessment and Abatement in Rapid Transit Systems (1974).

best choice. Chips and spalls should be ground out without welding by moving the contact strip and cleaning up the defect. If the chips and spalls are not too deep, they may be taken out by a thorough grinding of the rail with a rail grinding train as discussed previously.

*Joint Maintenance.* Impact noise is generated at rail joint gaps and elevation discontinuities. Large joint gaps create more noise than short joint gaps. Misalignment of the running surface elevation will result in impact noise. Further, the ends of the rail may require weldment deposition and grinding to repair end-batter. Thus, joint maintenance includes tightening rail joints to remove or reduce gaps, aligning running surface elevations, and repairing battered ends. Properly tightened joints and tie plates will reduce impact noise to manageable levels. The cost of joint maintenance is roughly $5 to $10 per foot without welding, based on data collected in 1980 and adjusted for producer price index changes.

*Field Welding of Joints.* Field welding of joints, or replacement of the rail with continuous welded rail, eliminates impact noise at joints and results in an overall reduction of maintenance effort. Field welding, or use of continuous welded rail, may not be practical on aerial structures, where thermal expansion and contraction may place high loads on aerial structure components. Examples include the MTA NYCT steel elevated structures, some of which may be more than 100 years old. Modern reinforced concrete aerial structures normally are capable of carrying continuous welded rail, though provisions are made in fastener design to accommodate thermal expansion and contraction. The cost of field welding of joints is about $600 per joint, based on data collected in 1980 and subsequent changes in the producer price index. After welding, the weldment must be ground to provide a uniform rail height across the joint.

*Elimination of Track Support Looseness.* Standard wood ties and tie plates retained with cut spikes allow vertical looseness which may promote impact noise caused by hammering of the rail and tie plate. This may be particularly significant at steel elevated structures such as at MTA NYCT and CTA, where impact forces generate vibration in the steel structure, which then radiates noise to the wayside. Replacement of tie plates with bonded resilient fasteners eliminates impact noise between these loose components, and the elastomer fasteners add damping to the rail which helps to reduce the effective noise-radiating length of the rail and, thus, noise. Resilient fasteners also may help to reduce noise radiated by steel box girders supporting aerial structures or noise from older steel elevated structures. Tests at MTA NYCT indicate that the greatest noise reductions were achieved with natural rubber fasteners with stiffness on the order of 100,000 lb/in. and less, lateral stiffness measured at the top plate of between 30,000 and 60,000 lb/in., and top plate resonance in excess of 800 Hz. Resilient fasteners with very low lateral compliance are less desirable than those which provide some isolation. Costs for bonded resilient fasteners are in the range of $50 to $80 per fastener, which, for an 18-in. tie spacing on an elevated structure, translates to $66 to $100 per track foot. Fastener manufacturing is very competitive; therefore, costs for a specific design characteristic vary little from one manufacturer to the next.

*5.2.3.2 Onboard Treatments*

Wheel truing and slip-slide control are the principal onboard treatments available for controlling impact noise, and these should be considered before employing treatments identified for normal rolling noise.

*Wheel Truing.* As with rail grinding, wheel truing is the most effective means of controlling impact noise produced by wheel flats, chips, spalls, and other defects in the tread running surface. Noise reductions on the order of 10 dB may be expected where no truing had been performed before, provided that the rail is sufficiently ground and maintained. There also is an improvement in the qualitative perception of the noise. After truing, and assuming the rail is smoothly ground with no defects or joints, the wheel/rail noise should have the sound of a smooth running bearing, without harshness or audible impacts. In fact, there may be some difficulty distinguishing between wheel/rail noise and propulsion system noise.

*Slip-Slide Control.* Slip-slide control, standard on most new transit vehicles, is an electromechanical servo-controlled system which limits wheel slip during acceleration and sliding during braking and reduces the occurrence of wheel flats and burns. Braking pressures and motor torques are modulated to equilibrate wheel set rotational velocities. Wheel flats will still occur with a slip-slide system; thus the need for wheel truing is not eliminated. However, the truing interval can be lengthened, and lengthening the truing interval reduces truing costs. Roughly, a 50% reduction of wheel flat occurrence may be expected under normal conditions, which would translate into a 50% reduction of wheel truing periodic costs. Further, slip-slide control improves traction and braking during wet weather, providing ancillary benefits in addition to noise control which might justify its cost regardless of noise reduction benefits. The cost of slip-slide control is difficult to determine because many vehicles come standard with such control, but should range between $5,000 and $10,000 per vehicle. The cost should be balanced against the savings in reduced wheel truing and extended wheel life.

*5.2.3.3 Wayside Treatments*

Wayside noise control treatments identified previously with respect to normal rolling noise are effective for controlling impact noise. However, the trackwork and onboard treatments normally should be considered before engaging in wayside treatments. Only if normal rolling noise levels are in excess of criteria should barriers be considered as a noise control treatment. An exception to this might be a situation in which for reasons of geometry, clearance, or lack of funds, rail grinding and wheel truing cannot be performed.

### 5.2.4 Corrugated Rail Noise

Noise caused by rail corrugation is perhaps the most objectionable type of wheel/rail noise occurring at tangent or moderately curved track, and one of the most difficult to control. The harsh tonal character of corrugation noise makes it one of the most easily heard and identifiable types of community noise, often affecting large areas. Rail corrugation noise can be painful to transit system patrons and interferes with conversation, and many complaints concerning excessive noise from rail transit systems are directly related to rail corrugation. Descriptive terms for noise caused by rail corrugation are “roaring rail” or “wheel/rail howl.” Roaring rail or wheel/rail howl at severely corrugated track may be a special type of periodic impact noise resulting from loss of contact between the wheel and rail. An inspection of the theory of impact-generated noise for smooth rail undulations reveals that typical corrugation amplitudes are sufficient to produce contact separation.

The noise control provisions appropriate for corrugated rail noise are presented in Table 5–4. Again, rail grinding is the most effective method of treating the symptoms of rail corrugation, but it may not remove the conditions which lead to or promote corrugation. Many other treatments for which there is insufficient information concerning effectiveness are indicated, and they are included for consideration and further evaluation by the user.

*5.2.4.1 Trackwork Treatments*

Treatments for rail corrugation are limited primarily to aggressive rail grinding. A second direct treatment is hardfacing. Additional trackwork design provisions, subject to field evaluation, which might be effective in controlling rail corrugation include reduced track support stiffness, reduced rail support separation or pitch, stiffened top plate, and vibration absorbers.

*Aggressive Rail Grinding*. The most effective approach to controlling recurrent or chronic rail corrugation is to grind the rail running surface, using an aggressive rail grinding program optimized to minimize long-term material loss and cost. Assuming that the corrugation growth rate is exponential, at least during the early stages of corrugation when the amplitude is not sufficient to produce contact patch separation, and that sufficient metal is removed to eliminate corrugation without overgrinding, an optimum grinding interval can be approximated as the time interval required for corrugation to grow by about 170% (based on an exponential growth rate). Both longer and shorter grinding intervals will result in higher rates of metal removal and thus reduced rail life. A longer grinding interval will increase noise exposure; a shorter grinding interval will maintain lower levels of noise, but increase grinding costs.

The corrugation growth time may not be sufficient for significant corrugation to appear after rail grinding, and once corrugation amplitudes develop to a point that corrugation noise is audible, the corrugation growth may already exceed the above criterion. In this regard, the rail should be ground often enough to avoid visible corrugation growth in excess

**TABLE 5-4 TREATMENTS FOR NOISE DUE TO RAIL CORRUGATION**

| TREATMENT LOCATION | TYPE | NOISE RED'N - dB - | COST | COMMENT |
|---|---|---|---|---|
| Onboard | Wheel truing | NA | $60 /wheelset[1] | Believed to reduce spin-slip corrugation at Vancouver |
| | Friction modifier | NA | $1,400 /veh/yr[2] | Believed to be effective at Vancouver, but data inconclusive. Costs based on two axles treated per vehicle. |
| | Damped Wheels | NA | $500 /wheel[2] | Effectiveness unknown, but should be effective to extent that wheel resonances influence corrugation. Little direct reduction of wheel/rail noise. |
| | Conical Treads | NA | $0 | Effectiveness Unknown, but promotes steering without lateral slip on tangent track. |
| Trackwork | Aggressive Rail Grinding | 7 - 10 | $1,000 to $7,000 /track-mile | Definitely very effective in reducing noise and controlling corrugation. Cost based on labor and overhead for 2 persons at $2,000/day and grinding 2 miles of track per shift. Costs vary substantially. |
| | Reduced rail support stiffness | NA | $0 | Believed to be effective in reducing corrugation rate, but would not reduce noise directly |
| | Reduced rail support separation to 24 inches | NA | $50/trk-ft | Unknown effectiveness, but would separate the pinned-pinned mode resonance frequency from the corrugation frequency and thus might reduce corrugation rates. |
| | Increase top plate bending stiffness | NA | Negligible | Unknown effectiveness, but would avoid coincidence of top plated bending with the pinned-pinned mode rail resonance. |
| | Hardfacing | 7 - 10 | $15/rail-ft | Controls corrugation by providing hard running surface |
| Wayside | Treatments for normal and excessive rolling noise | | | |

1 L. G. Kurzweil et al., **Noise Assessment and Abatement in Rapid Transit Systems** (1974) Pg. 3-10.

2 Current estimates

NA Not applicable or not available

of random rail roughness and to avoid audible corrugation noise. The optimum grinding interval, thus, is difficult to define, and some experimentation and careful monitoring of growth rates may be required. Profile grinding to minimize spin-creep or spin-slip is employed at the Vancouver Skytrain to reduce rail corrugation. The maximum facet width is $\frac{1}{16}$ in., and several passes are made to produce a radiused ball and limit contact width to the order of $\frac{5}{8}$ in.

Costs for rail grinding include the capital cost of the rail grinder, fuel and grinding materials, and personnel which may involve a grinder operator, flagger, and supervisor. Dust collection equipment in the form of a vehicle-mounted vacuum cleaner also may be needed, producing an additional cost.

*Hardfacing.* Hardfacing with a very hard rail head inlay has been incorporated at European transit systems to control rail corrugation, especially at light rail or streetcar systems. The effectiveness of the treatment in controlling rail corrugation at U.S. light and heavy rail transit systems has not been demonstrated. Further, the hardfacing material is not recommended by the manufacturer for high carbon steel rail, common in North America. Treatment of short sections subject to unusually severe corrugation may be appropriate, and rails may be supplied with hardfacing. The cost for hardfacing rail is estimated to be about $15 to $36 per lineal foot.

*Alloy and Hardened Rail.* Alloy rail, such as chromium vanadium, has been considered for controlling corrugation at curves in heavy freight railroads. Its usefulness at transit systems for controlling short-pitch corrugation has not been determined. Alloy rail with greater hardness and wear characteristics might be considered to be less prone to corrugation than standard carbon steel rail, though exactly the opposite has been observed. There may be a reduction of weldability with hardened or alloy steels. Alloy and hardened rail should not be considered for rail corrugation control without careful evaluation and consultation with a metallurgist.

*Track Support Stiffness.* Rail corrugation growth appears to be most prevalent at stiff direction fixation track where the rail support modulus is in excess of perhaps 10,000 lb/in. per inch of rail, though no clear quantitative relation has been identified between track stiffness and rail corrugation growth. Many other factors must be considered. Anecdotal evidence suggests that a stiffness reduction may be beneficial in reducing corrugation growth rates. With stiff track supports providing a dynamic rail support modulus of about 10,000 to 15,000 lb/in.

per inch of rail, the rail-on-fastener resonance frequency is on the order of 175 to 200 Hz. A fastener consisting of a steel tie plate and thin elastomer pad is very stiff, on the order of 1 to 2 million lb/in., giving a rail support modulus of 30,000 to 60,000 lb/in. per inch of rail. Under these conditions, the vertical rail on fastener resonance frequency is on the order of 350 Hz or higher, at the lower end of the range of corrugation frequencies, suggesting a possible interaction between corrugation and vertical rail resonance.

A benefit of soft rail supports is that the rail is less affected by the fastener pitch, which controls the pinned-pinned resonance frequency of the rail. Another benefit is that at corrugation frequencies, soft fasteners decouple the rail from the concrete invert so that the rail is essentially an infinite beam and, thus, has an input mechanical impedance with equal real and imaginary parts. (The real part is caused by energy transmission along the rail.) Thus, there is a natural damping mechanism that may reduce local resonances in the track and wheels at corrugation frequencies, a mechanism that is negated if the rail support modulus is too high.

Although low stiffness fasteners appear to be attractive, the dynamic interaction between the rail and wheel is very complex, and a careful analysis, testing, and evaluation should be conducted before committing to a wholesale replacement of fasteners to control rail corrugation. Fortunately, BART, WMATA, and the LACMTA have fasteners of widely varying stiffnesses; therefore, answers concerning differences between corrugation growth rates for various fastener stiffnesses should be obtainable by monitoring these systems.

The cost of a soft fastener may range from about $50 per unit to perhaps $80 per unit. The cost of a bonded resilient fastener is normally higher than the cost of a fastener consisting of a rolled steel tie plate, neoprene elastomer pad, and clips, a fastener which has a stiffness on the order of 1 to 2 million lb/in., an order of magnitude greater than that of modern bonded direct fixation fasteners.

*Rail Support Separation*. A fastener spacing of 30 to 36 in. results in a pinned-pinned mode resonance frequency on the order of 500 to 750 Hz, the range of typical corrugation frequencies observed at systems such as BART. There is, therefore, potential for interaction of the pinned-pinned mode with wheel set anti-resonances in this frequency range. Further, corrugation amplitudes have been correlated with fastener location, suggesting that the fastener location does have an effect of some kind. If the pinned-pinned mode is indeed contributing to rail corrugation, a possible solution is to reduce the support spacing and increase the pinned-pinned modal frequency sufficiently so that the associated corrugation wavelength is less than the contact patch longitudinal dimension. In this case, corrugations caused by a pinned-pinned mode might be expected to be "ironed out" or worn away with time.

For 70 mph trains with a contact patch length of ⅝ in., the design resonance ideally should be higher than 2,000 Hz, suggesting that the fastener separation should be less than 16 in., a separation which would double the cost of current direct fixation fasteners. However, the ironing out effect might be achieved if the associated wavelength is less than half the contact patch length, corresponding to a minimum frequency of about 1,000 Hz and a maximum fastener spacing of 24 in. (The tie spacing of 18 to 24 in. employed at conventional ballast-and-tie track, interestingly, satisfies this criterion.) Finally, increasing the rail section does not appear to greatly increase the pinned-pinned modal frequency for practical rail sections.

*Direct Fixation Fastener Top Plate Bending Resonance*. The top plate bending resonance frequencies of direct fixation fasteners under load are about 600 Hz, very close to corrugation frequencies for short-pitch corrugation. No clear causative relationship has been identified between rail corrugation and top plate resonance frequency, but prudent design practice would suggest that the resonance frequencies in the track support components be kept away from corrugation frequencies, the first pinned-pinned rail vibration mode, and the first anti-resonance frequency of the wheels. This can be achieved by thickening the top plate to raise its resonance frequency in excess of 1,000 Hz, which requires fasteners that are slightly thicker than what is typical of bonded direct fixation fasteners. A minimum of 2 in. should be provided between the rail base and invert. There should be increased reliability because of the increased strength of the fastener top plate, with less working of the rail clip resulting from top plate flexure under static load. Less working of the clip will result in less fretting corrosion and clip failures. Thickening the top plate will increase the cost of top plate castings, though the increase should be relatively small in comparison with the overall cost of the fastener.

#### *5.2.4.2 Onboard Treatments*

Onboard treatments do not reduce noise caused by rail corrugation significantly, but may help to control rail corrugation rates. The onboard treatments that follow are subject to testing and careful analysis; they are not proven approaches to corrugation control. However, to the extent that corrugation is intimately related to vibration and dynamic interaction, the treatments can be expected to influence the corrugation process.

*Wheel Profile.* The profile of the wheel's running surface directly affects interaction between the rail head and wheel. High wheel/rail conformity has been identified as a contributing factor in rail corrugation at the Vancouver Skytrain, resulting from increased spin-slip (*1*). Frequent wheel truing to maintain a conical tread taper without concavity would reduce wheel/rail conformity and resulting spin-creep torques. Wheel truing in combination with rail grinding is performed at Vancouver for this purpose.

*Dry-Stick Friction Modifiers.* Onboard lubrication of the tread with dry-stick friction modifiers to reduce or eliminate negative damping associated with stick-slip or roll-slip of the wheel/rail contact has been claimed to reduce corrugation growth rates at the Vancouver Skytrain (*2*) and is very attractive from a theoretical standpoint. Further, there is some improvement of adhesion reported by the Sacramento RTD. However, long-term data on corrugation growth reduction are only now being developed by several light rail transit systems using the treatment, and observations at Sacramento RTD indicate that corrugation is not inhibited (*3*). The cost of onboard dry-stick lubrication is estimated to be about $1,500 per vehicle per year, assuming that two axle sets of each vehicle are treated.

Comparing this cost with the cost of rail grinding and other treatments is difficult because train headways and consist lengths must be considered. Onboard lubrication is employed in conjunction with careful and aggressive rail grinding at the Vancouver Skytrain. Only anecdotal information has been obtained to indicate that dry-stick lubrication reduces the need for grinding. Further, at the Sacramento RTD, where dry-stick lubrication is used, rail corrugation was observed within 2 years after grinding, and corrugation has been observed at the Los Angeles Blue Line at an aerial structure with direct fixation track.

*Damped Wheels.* Damping of the wheel to reduce its response at corrugation frequencies is particularly attractive, though no data have been obtained indicating a reduction of corrugation rate with wheel damping. In any case, addition of damping to wheels should not increase the rail corrugation rate. A careful evaluation of damping should be conducted before employing damped wheels strictly for corrugation control. For instance, resilient wheels provide some additional damping, but has not prevented the occurrence of rail corrugation at the Portland, Sacramento, and Long Beach systems. The decision to use damped wheels, therefore, should be predicated on controlling squeal and other operational factors.

*Conical Wheel Tapers.* Crabbing of the truck induces lateral slip and possibly stick-slip oscillation at the wheel's fundamental lateral resonance frequency. Although this is most often associated with squeal at curves, such a mechanism may be responsible for rail corrugation at BART, which employs aluminum centered wheels with low damping ratios and cylindrical tread profiles. (BART is one of the few systems, perhaps the only system aside from the Chicago CTA, that employs a cylindrical tread profile.) The cylindrical wheel profile does not promote self-steering of the truck; therefore, the truck may not align itself along the center of the track. A conical wheel taper promotes centering of the truck and axles. Therefore, a possible approach for reducing rail corrugation at systems using cylindrical wheels is to change to a tapered wheel profile.

Infrequently trued wheels will lose their profile and, in extreme cases, develop conformal contact between the rail and wheel or develop a two-point contact with the rail. Therefore, adopting a conical tread profile to control lateral slip would be useless unless truing is done frequently enough to prevent adverse contact conditions. Hunting of the truck is likely to increase with increasing conicity, and care must be taken in selecting rail and wheel profiles. Spin-creep is increased with increased wheel taper and thus may contribute to rail spin-creep corrugation, though the spin-creep torques would be maintained at minimal levels if the contact patch width is kept to ⅝ in. or so. Further, there are many systems using conical wheels that experience rail corrugation; therefore, there is no guarantee that conical wheels will inhibit rail corrugation. The decision to employ a conical wheel, therefore, should be considered carefully, in relation to other factors such as ride quality. The cost of wheel profiling is negligible and, if successful, switching to a cylindrical profile may be one of the least costly corrugation control treatments available.

#### *5.2.4.3 Wayside Treatments*

The treatments identified for controlling normal rolling noise also are applicable to controlling noise from corrugated rail. However, corrugated rail produces higher levels of noise and a peaked frequency spectrum that is more detectible to the average receiver and thus may be more objectionable than normal rolling noise. Wayside treatments such as berms, barriers, receiver modifications, or tunnel wall sound absorption designed to control normal rolling noise would probably not be adequate to control noise from rail corrugation and, as a rule, trackwork or onboard treatments that reduce or eliminate corrugation are preferable to wayside treatments. If corrugation cannot be controlled, wayside treatments remain as an option.

## 5.3 CURVING NOISE CONTROL

As discussed previously, wheel/rail noise at curved track may differ considerably from such noise at tangent track and may include a combination of normal and excessive rolling noise, impact noise, noise resulting from corrugation, wheel squeal resulting from stick-slip oscillation, and wheel howl. Wheel squeal is the most common form of curving noise, caused by stick-slip oscillation during lateral slip of the tread over the rail head, and may be excruciating to patrons and pedestrians. Wheel howl at curves may be related to oscillation at the wheel's lateral resonance on the axle, caused by lateral slip during curving. At short radius curves where train speeds may be limited to 20 mph, rolling noise may be insignificant relative to wheel squeal. At curved track, normal rolling noise, excessive rolling noise resulting from roughness and corrugation, and impact noise resulting from rail defects and undulation are similar to those at tangent track. The user is referred to the section on tangent track for discussion of noise not directly related to curving. The discussion of curving noise control that follows focuses on wheel squeal and wheel/rail howl.

### 5.3.1 Wheel Squeal

Treatments for controlling wheel squeal are listed in Table 5–5 with respect to onboard, trackwork, and wayside application. Wayside treatments are primarily limited to sound barriers, receiver modifications, and subway structure sound absorption, which were discussed previously with respect to normal rolling noise. The spectra of squeal and howl, however, differ from those of normal rolling noise, and spectral characteristics should be considered in the design of wayside treatments.

The results of the survey of transit systems and inspection of track reveal a combination of factors which, taken together, appear to control or eliminate wheel squeal at embedded track. These are as follows:

1) Use of resilient wheels,
2) Resiliently supported track with 115 lb/rail,
3) ½-in. gauge widening at curves,
4) ¼ in. wider wheel gauge than standard,
5) Minimum curve radius of 90 ft,
6) Rubberized grade crossing bearing against the rails at both sides,

TABLE 5-5 WHEEL SQUEAL NOISE CONTROL TREATMENTS

| LOCATION | TYPE | NOISE RED. - dB | COST - $ | COMMENTS |
|---|---|---|---|---|
| Onboard | Resilient Wheels | 10 to 20 | $2,000 to 2,300/wheel | Well demonstrated to be effective. |
| | Constrained Layer Damped Wheels | 5 to 15 | $500 to $1200/wheel | Effective |
| | Ring Damped wheels | 5 to 10 | 30$-50$ per wheel | Used by Chicago CTA and MTA NYCT. May be most economical treatment. |
| | Wheel Vibration Absorbers | 5 to 15 | $500 to 700/wheel | Demonstrated effectiveness in trials. |
| | Conical Wheel Taper | Elimination at large radius curves | $0 | Must be in combination with gauge widening or asymmetrical rail profile and flexible truck. |
| | Flexible Primary Suspension | Elimination at large radius curves | Unknown | Conical treads required to induce axle alignment with radius |
| | Steerable Trucks | Elimination at large radius curves | Unknown | Conical treads required to induce axle alignment with radius |
| | Onboard friction modifier | Possible elimination. | $1,400 /vehicle/year | Limited effectiveness below 90 foot radii |
| Track-work | Asymmetrical rail profile | Elimination at large radius curves | Nil | Must be in combination with conical wheels and flexible truck. |
| | Petroleum lubrication | Partially eliminates squeal | $10,000 to $40,000 | Can lubricate flange only, so that effectiveness is limited. |
| | Water spray lubrication | Eliminates squeal | $10,000 to $40,000 per track curve | Not practical in freezing weather, though antifreeze can be used with water. |
| | Maintain constant gauge or gauge narrowing in curves | Reduces truck crabbing and potential for squeal | Nil | Gauge narrowing has been correlated with squeal elimination in conjunction with resilient wheels, HPF friction modifier, and elastomer rail embedments. (See Text) |
| | Restraining rails | Has inconsistent effectiveness | $100 to $200/tr-ft | Reduces truck crabbing and lateral slip, and, thus, squeal. Flange face must be lubricated. |
| | Rail vibration dampers | Unknown | $20 to $50/ft | Reported to be effective in Europe |
| | Rail Inlay - (anti-squeal) | Reduces squeal | | Reduced or eliminated squeal at WMATA for several months |
| Wayside | Sound Barrier walls | 7 to 10 | $20/sf | Does not eliminate squeal |
| | Absorptive Barriers | 9 to 12 | $25/sf | Does not eliminate squeal |
| | Berms | 10 to 13 | NA | Does not eliminate squeal |
| | Subway wall treatment | 5 to 7 | $7-$10/ft$^2$ | Does not eliminate squeal |
| | Receiver Treatments | NA | $5,000 to $10,000 per receiver | Does not eliminate squeal. Noise reduction dependent on construction |

NA Not available or not applicable

7) Onboard dry-stick low coefficient of friction flange lubrication,
9) Onboard dry-stick friction modifier applied to wheel tread,
10) Maintenance of conical wheel tread profile, and
11) Southern California climate.

There is no guarantee this combination of factors eliminates squeal, but this is the result observed at the Los Angeles Blue Line embedded curves in Long Beach, California. The researchers note that Long Beach is close to the ocean, and humidity is a factor in squeal generation. The observations at Long Beach were made during a very sunny day, and the tracks appeared to be very dry.

#### *5.3.1.1 Onboard Treatments*

Onboard treatments for controlling wheel squeal include resilient wheels, constrained layer damping, ring dampers, vibration absorbers, dry-stick lubrication, oil spray lubrication, conical wheel tapers, longitudinal primary suspension compliance, and steerable trucks. Of these, resilient wheels are the most popular and are almost universally used for light rail transit, though squeal is not eliminated at short radius curves under 100 ft. Conical wheel tapers and longitudinal truck compliance might reasonably be considered as standard in truck design, but they are also listed for systems with particularly stiff primary suspensions and cylindrical wheels. Some of the treatments, such as oil spray lubrication and steerable trucks, have received only limited application.

*Resilient Wheels.* Resilient wheels incorporate elastomer springs between the tire and wheel rim to provide compliance between these components. Examples of resilient wheels include the Bochum 54 and 84 wheels, the SAB wheel, and the older PCC wheel. Resilient wheels are fitted with most light rail transit vehicles and are enjoying widespread popularity. They are effective in reducing or eliminating wheel squeal at curves of radii greater than about 100 ft, though squeal may still occur. Examples of systems using resilient wheels include the Portland Tri-Met; Los Angeles Blue Line and Green Line; Pittsburgh; Bi-State Development Agency, St. Louis; San Jose; and Sacramento RTD. No heavy rail rapid transit systems appear to be using resilient wheels within the United States, though BART has a set of Bochum 54 wheels mounted on a single car, a set that has withstood roughly 10 years of service and one rebuild with no adverse consequences other than a problem with shunts, which was corrected.

Vibration dampers may be fitted to the tread of resilient wheels to further improve their squeal reducing characteristics. Regardless of the configuration, resilient wheels are expected to greatly reduce, if not eliminate, wheel squeal occurrence and energy. Electrical arcing may be a problem with internal electrical shunts, involving pitting of the interior surface of the tire tread, thus leading to a stress concentration and possible fatigue failure. No direct evidence of this problem has been obtained, and external shunts with bolted connections would prevent arcing. A potential problem with external shunts may concern tread rotation relative to the rim, resulting in stretching or breakage of the shunts, though this has not been a problem with the single BART vehicle using resilient wheels.

Light rail transit systems have reported no difficulty in truing resilient wheels. The cost of a typical resilient wheel may range between $2,400 and $3,000, which can be compared with a cost of roughly $700 for a standard solid steel wheel. However, after tread condemnation, the treads may be replaced with new treads and rubber springs, either at the factory or at the shop, thus saving the cost of the rim.

*Constrained Layer Damped Wheels*. Constrained layer damping consists of visco-elastic material constrained between the wheel web and a metal plate or ring. Constrained layer damped wheels have been shown to be at least partially effective in reducing squeal at SEPTA, MTA NYCT, and BART. Although damped wheels tend to be less effective than resilient wheels, they may have an advantage over resilient wheels in that monobloc or composite steel and aluminum wheels would not have any difficulty with dynamic alignment and can withstand high cyclic loading without heat buildup. A disadvantage of constrained layer dampers is that they add weight to the wheel, though this may be compensated partially by weight losses due to machining. Constrained layer damping treatments for wheels have not received much attention within the United States, perhaps because of cost, which could be as high as $1,000 per wheel, depending on engineering requirements and quantity. Costs for constrained layer dampers tested at the MTA NYCT in 1984 were on this order, and applying a producer price index ratio of (1:2) would suggest a small quantity cost of about $1,200 per wheel. Costs would be expected to be considerably less with quantity orders.

*Wheel Vibration Absorbers.* Wheel vibration absorbers are tuned and damped spring-mass mechanical systems that are attached to the tire to absorb vibration energy at wheel modal frequencies. The absorbers can be effective over a broad frequency range, though they are most effective if tuned to the modal resonance frequencies of the wheels. A disadvantage is that, similar to the constrained layer damper, they add weight to the wheel. Tuned vibration absorbers have not received widespread use within the United States, though they have been applied to European high speed rail systems. They have been recently tested, with varying degrees of success, most recently at the West Falls Church yard at WMATA. Additional testing is needed at short radius curves of light rail systems.

The cost for tuned vibration absorbers in large quantities may be between $400 and $700 per copy. Part of the cost includes engineering to tune the absorber to the resonance frequencies of the wheel, which may be unique to a given wheel design. Certain "broad band" absorbers would presumably avoid tuning problems, allowing "off-the-shelf" procurement, though they might be less effective. Costs for materials and engineering should be discussed with the manufacturer carefully, and field testing of prototype units is very desirable before large scale procurement. A stress analysis may be advisable to evaluate the effect of the bolt holes on wheel stresses.

*Ring Damped Wheels.* Ring dampers typically consist of a ⅝-in.-diameter carbon steel rod bent into a circle and retained within a groove in the inside edge of the tire. Tests at SEPTA, CTA, and MTA NYCT indicate that ring dampers are effective in reducing wheel squeal. A major attractive feature of the ring damper is that its cost is the lowest of the damping technologies. At CTA, the ring damper is supplied with new wheels for an additional cost of about $30 to $50 per wheel, and little maintenance is required. Concerns have been expressed over the possibility of dirt or steel dust caking in the damping groove and inhibiting damper performance. CTA indicates that this has not been a problem, but such a problem has been observed at SEPTA during in-service tests with carbon steel rings. There is no known instance of the dampers falling from their grooves during service.

*Dry-Stick Friction Modifier*. Dry-stick friction modifiers are applied to the tread running surfaces of one or two axle sets of each vehicle to enhance adhesion and flatten the friction versus creep curve of the wheel and rail running surfaces, thus reducing negative damping and squeal. Dry-stick lubrication, which includes low coefficient of friction flange lubricant applied to the flange throat to reduce friction and flange wear, may help with curving of the truck and reduce lateral slip. Several light rail transit systems are now using onboard friction modifiers and flange lubricant on a regular basis, the most notable being the Sacramento RTD and the Los Angeles Blue Line.

No wheel squeal occurs at the embedded 90- and 100-ft radius curves of the Los Angeles Blue Line, which employs a combination of resilient Bochum 54 wheels and resilient embedded 115 lb/yd rail track with elastomer grade crossing bearing against both sides of the rails. Two axle sets of each vehicle are lubricated with HPF and LCF dry-stick lubricant. Further, the Los Angeles Blue Line vehicles have wheel gauge ⅛ in. wider than standard, which reduces lateral slip in curves.

In contrast to the experience at the Los Angeles Blue Line, significant squeal was observed at the 82-ft radius embedded ballast-and-tie curves of the Sacramento RTD, which also employs resilient Bochum 54 wheels. The Sacramento system uses embedded ballast-and-tie track with a concrete road filler between the rails and asphalt at the field sides. These curves are restrained with lubricant applied to the restrainer, preventing flange contact at the high rail side. In spite of the lack of flange contact, squeal occurs. At the 100-ft radius curve, wheel squeal was much less than at the 82-ft radius curve, occurring at about 500 Hz. Only one axle set, on the idling truck, of the Sacramento RTD articulated vehicle is lubricated.

To be effective, the manufacturers indicate that onboard dry-stick lubricants must be applied at least to every vehicle, and possibly every truck, to deposit sufficient lubricant to the running surface of the rail. The cost of friction modifier varies from system to system. Costs on the order of $1,500 per vehicle per year are estimated for friction modifier applied to two axle sets of each vehicle. Maintenance and product loading costs are minimized with the use of rear loading lubricant holders, requiring only a few minutes per truck to change lubricant sticks.

*Oil Drop.* Portland Tri-Met has experimented with onboard oil dispensers on two vehicles out of a fleet of perhaps 50. This approach did not completely eliminate wheel squeal, and Portland Tri-Met is experimenting with dry-stick lubricants on every vehicle. The cost of oil drop or oil spray lubrication has not been determined, though a commercial lubrication system is available.

*Conical Wheel Taper*. A conical wheel profile reduces the longitudinal slip of the wheel over the rail by providing a rolling radius differential, provided that there is some lateral shifting of contact location brought about by gauge widening and/or asymmetrical rail profile grinding. However, lateral slip resulting from finite truck wheel base will still occur if there is no longitudinal compliance in the truck primary suspension to allow the axles to align themselves parallel with the curve radius. Providing primary suspension longitudinal compliance in combination with wheel taper and asymmetrical profile grinding is expected to reduce squeal. Relative longitudinal motion is not entirely compensated at curves of less than about 1,200- to 2,000-ft radius. Although wheels may be trued to a 1:40 or 1:20 taper, wear will eliminate taper with time, possibly leading to a concave tread. This has been observed at Sacramento RTD tangent track, where the wheel/rail contact was concentrated at the corners of the rail head on tangent track, rather than at the center, with a ball radius of about 7 to 12 in. Further, wheel squeal occurs at Sacramento. At systems with significant flange wear, truing occurs more frequently, so that taper may be preserved.

*Flexible Primary Suspension.* Trucks with compliant longitudinal stiffness in the primary suspension combined with conical wheel tapers promote steering and reduce axle crabbing angle, thus reducing lateral creep and squeal. Crabbing angles up to 5 deg may be expected for 90-ft radius curves; therefore, considerable flexibility must be incorporated into the primary suspension to reduce lateral creep at short radius curves, where lateral slip may never be fully prevented for practical tread profiles. At larger radius curves, the creep angle is lower, requiring less accommodation. At BART, for example, the shortest curve radius is about 360 ft, giving a creep angle of 1.2 deg, requiring on the order of plus or minus 0.05-in. longitudinal compliance at each bushing.

Conical wheels must be used with sufficient gauge widening to induce axle alignment with the curve radius and improve curving performance. Cylindrical wheels would not induce the necessary forces to cause the axles to shift laterally and produce a rolling radius differential and, thus, would not encourage the axles to align themselves parallel with the curve radius at the point of wheel contact. (These comments indicate that the control of wheel squeal requires cooperation between track and vehicle designers and departments responsible for maintenance of these components.)

*Steerable Trucks*. Steerable trucks have been developed for reducing wheel squeal by allowing the axles to align themselves parallel with the curve radius at the point of wheel contact. An example is the UTDC truck employed at the TTC Scarborough Line and the Vancouver Skytrain. Problems occurred with friction between the side bearing pads, which inhibited proper alignment of the axles on tangent track after coming out of curves. Although steerable trucks may eliminate axle crabbing, longitudinal slip must still occur unless compensated with rolling radius differential.

#### *5.3.1.2 Trackwork Treatments*

Trackwork treatments include restraining rails, flange lubrication, asymmetrical rail profile grinding, gauge widening, rail head inlays, hardfacing, rail vibration dampers, and frictionless rail. Of these, flange lubrication is the most common trackwork treatment for controlling wheel squeal. Rail inlays are effective in reducing squeal, but are limited in life. The remaining treatments have been tried at various systems with varying degrees of success, and care should be used in selecting these for incorporation in track as noise control provisions.

*Flange Lubrication*. Flange lubrication with automatic grease lubricators is employed by various systems to control wheel squeal, based on the theory that flange contact with the rail is the principal cause of squeal. Successful examples of this approach include those used by SEPTA. However, flange lubrication also may involve migration of lubricant to the rail running surface, and this limited and inadvertent lubrication may, in fact, be the principal cause of wheel squeal reduction. Excessive migration of lubricant to the rail running surfaces will result in loss of adhesion and braking performance and therefore is to be avoided. For this reason, flange lubrication may not be employed by many systems.

*Water Spray Lubrication*. Water sprays are used at some systems such as the MTA NYCT and TTC to control wheel squeal. The advantage of water spray over petroleum lubrication is that the water evaporates quickly and thus traction is reduced for only a short period of time and for a short distance. Also, water spray systems should pollute soils much less than petroleum systems. Water sprays cannot be used during subfreezing temperatures because of buildup of ice. The TTC incorporates water sprays during the nonfreezing periods of the year and grease lubrication during winter months. Water has a relatively low cost compared with petroleum products. No cost data have been obtained for water spray systems, but these systems should be comparable to, if not less than, costs for grease lubricators.

*Restraining Rail.* A restraining rail can be employed at the low rail to guide the leading axle through the curve and reduce the axle crabbing angle. To the extent that flange contact may contribute to some forms of wheel squeal, preventing flange contact with the high rail would tend to reduce squeal, though the principal benefit would be a reduction of flange and gauge face wear. The trailing axle will tend to ride against the low rail; therefore, an additional restraining rail can be employed on the high rail side to further pull the low rail wheel away from the reduced crab angle.

The restraining rail should be lubricated with grease to prevent squeal from developing as a result of the wheel rubbing against the restraining rail. This should not result in migration of lubricant to the rail running surfaces, because only the back sides of the wheels would be lubricated. Wheel squeal will still occur if the curve radius is short enough, but the restraining rails should prevent unnecessarily large crab angles unless the flange way is too large. Restraining rails used simply to prevent wheel climb-out would also allow high rail wheel flange contact, not necessarily reduced crab angle. Careful adjustment of the restraining rail position is required to control crab angle, and subsequent wear of the restraining rail may require readjustment.

*Asymmetrical Rail Profile Grinding*. Asymmetrical profile grinding to offset the contact patch to the low rail side at both rails in combination with tapered wheel tread profiles reduces longitudinal slip of the wheel set. Further, a compliant primary suspension in the longitudinal direction (parallel with the track), in combination with tapered wheel treads to achieve a rolling radius differential, will allow the high rail wheel to translate along the rail faster than the low rail wheel and align the axle parallel with the curve radius, thus minimizing axle crab angle and lateral creep. Asymmetrical rail profile grinding would further this self-steering process and thus reduce wheel squeal.

*Rail Head Inlay.* Treating the rail running surface with a babbit-like alloy inlay in the rail head can reduce wheel squeal caused by lateral stick-slip across the rail running surface by modifying the friction-creep curve. This treatment was successful at WMATA, though problems were experienced with retention of the material. After a period of several months, the inlay wore away, allowing direct contact between the wheel and carbon steel portions of the rail, and the effectiveness of the treatment at WMATA declined with wear. A variant of the rail head inlay is a resin or plastic that is deposited and bonded into a groove of similar dimensions to that used for hardfacing. No data have been obtained concerning noise reduction effectiveness, though the treatment appears to have been applied in Europe.

*Hardfacing.* Hardfacing with a very hard inlay has been used at SEPTA with mixed results, though the treatment appears to be most desirable for controlling wear at curves. The use of hardfacing inlay at the flange contact face may not produce a reduction of squeal because, as theory suggests, the squeal is likely caused by lateral stick-slip of the wheel tread across the top of the rail. Still, the dissimilar metallurgical properties of the inlay and wheel tread material may modify the friction-creep behavior to have some benefit.

*Rail Vibration Dampers*. Rail vibration dampers have been advertised to reduce wheel squeal. If so, this would be contrary to expectations based on theoretical grounds, because of low participation of the rail in the squeal process relative to the wheel. However, reports provided by one manufacturer are encouraging. Rail vibration dampers consist of elastomer sheets or molded elastomer components that are held against the rail web with spring clips or clamps. They should be easily installed between the ties or fasteners, possibly without raising the rail. The lack of squeal at the Long Beach Blue Line 90- and 100-ft radius curves which have elastomer grade crossing embedded between and about the rails suggests that damping of the rail may be beneficial in reducing squeal, though other factors at Long Beach also may help to reduce squeal.

*Gauge Widening.* Gauge widening is reputed to reduce wheel squeal by increasing the rolling radius differential or by reducing flange rubbing. However, gauge widening promotes truck crabbing, increasing the angle of attack, or creep angle, of the wheel relative to the rail, thus increasing lateral creep and squeal. Thus, gauge widening should not be expected to reduce squeal unless a rolling radius differential can be effected. Further, high crab angle allows flange contact and wear at the high and low rail gauge faces, evidenced by a sharp edge between the gauge face and rail running surface, as has been observed at the MBTA. The Los Angeles Blue Line experiences no wheel squeal at short radius curves with ¼ in. wider wheel gauge and about ½ in. of track gauge widening, resulting in about ¼ in. of gauge widening relative to standard clearance. This system also employs Bochum 54 wheels, onboard flange lubrication and tread friction modifier, resilient embedded track, and elastomer pavement at both sides of the rails, all of which may help to reduce or eliminate wheel squeal. Squeal is observed at the Sacramento RTD 82-ft-radius embedded ballast-and-tie curve with about ¼-in. gauge widening. This curve also has a hand lubricated restraining rail which prevents flange contact at the high rail, though squeal appears to be generated primarily at the high rail, based on aural observation. Gauge widening does not prevent wheel squeal.

*Frictionless Rail.* The MBTA installed a modified 119 lb/yd rail with reduced rail head width to reduce the running surface at a 70-ft-radius single restrained curve on the Green Line. The Green Line vehicles use SAB resilient wheels which, by themselves, help to reduce squeal, and the restraining rail is well lubricated with automatic lubricators. Flange contact is popularly believed to produce squeal, contrary to the most likely theory of lateral slip as the cause of squeal, and the frictionless rail was intended to eliminate flange contact in conjunction with the restraining rail. In this sense, the frictionless rail represents another method of gauge widening.

The MBTA indicated that a noise reduction was obtained with the installation of the frictionless rail, but that wear evidently caused a recurrence of squeal, though the precise cause is uncertain. Visual observations conducted in July 1995 indicate that gauge face wear is occurring at the frictionless rail and, further, that there is substantial visual crabbing of the trucks in the curve. The substantial crabbing leads to high creep angle, or lateral creep, and thus promotes squeal, which appears to be produced by the high rail wheels. Also, the flange contact may be causing squeal, especially because of the large angle of attack, although frictionless rail with low rail head width may reduce or eliminate flange contact, such as what was not observed at the MBTA. Unless a restraining rail is adjusted to control crab angle, the use of frictionless rail will aggravate crabbing and thus squeal. In view of the above, simply using conventional rails with small ball radius and gauge widening would be equivalent to using frictionless rail.

*Curvature Design.* Limiting track radius of curvature to greater than 150 ft for vehicles with resilient wheels and 700 ft for vehicles with the solid wheels with conical treads is probably the most practical wheel squeal noise control provision for new track construction. These appear to be practical limits below which wheel squeal can be expected. In the case of light rail systems where tight curvature is required to negotiate intersections, 100 ft should be a limiting radius, though some limited squeal may be expected. Gauge widening should be avoided if possible. Although large radii are recommended for noise control, the benefits of reduced flange and rail wear should be obvious.

### *5.3.1.3 Wayside Treatments*

As with other forms of wheel/rail noise, wayside treatments consisting of sound barriers, berms, and receiver treatments are applicable to wheel squeal. However, these treatments must be more effective than at tangent track, because of the pure tone character of wheel squeal. The average person is more sensitive to squeal than to broad-band passby noise, and reducing the noise with a barrier or receiver treatment may not be sufficient to prevent adverse community reaction. For this reason, treating the track or vehicle directly to eliminate wheel squeal is the most attractive approach.

*Sound Barrier Design*. Barriers are more effective at wheel squeal frequencies of about 1,400 to 4,000 Hz than they are for normal passby noise which may have a peak frequency in the range of 500 Hz on smooth ground rail. Usually, sound absorption applied to a barrier to control reflection will be more effective at squeal frequencies than at frequencies associated with rolling noise.

*Receiver Treatments.* Acoustically rated glazing must be selected to be effective in the range of 500 to 5,000 Hz. There may be significant dips in the sound transmission loss characteristics of window glazing which coincide with wheel squeal frequencies. For example, conventional thermally insulating glass exhibits a coincidence dip at 4,000 Hz which may coincide with the second mode of wheel tire squeal. Monolithic single-pane glazing exhibits a coincidence dip at 2,500 Hz which should be avoided if squeal frequencies are at that frequency. Single-pane glazing should be ¼-in.-thick laminated glass, and thermally insulating glass should have at least one pane consisting of laminated glass, to avoid problems of this nature.

*Station Treatment.* Wheel squeal is not normally a problem in stations because tangent track is normally used adjacent to the platform. In this case, station treatment need only be applied to control rolling noise (as well as public address system performance and ancillary facility noise). An exception to this is the MBTA Green Line, where a station has a short radius curved section of ballast-and-tie track at patron level. Treatment of station walls and ceilings with sound absorption in such cases is desirable.

*Subway Wall Sound Absorption.* Curves beginning at the end of station platforms may cause considerable squeal which transmits to the station platform area and may be uncomfortable to patrons or interfere with conversation. In this case, sound absorption applied to the upper portion of subway walls and the ceiling in the curved track section may be effective in reducing squeal noise transmission to the station platform area. Car interior noise reductions would also be obtained with subway wall treatment. Car doors are often positioned near or over the trucks, and wheel squeal noise can be easily transmitted to the interior of the vehicle, where it may actually be painful to patrons. Sound absorption placed against the subway wall from floor to ceiling and extending throughout the curve would be particularly effective in controlling this transmission path, even with ballasted track where ballast normally provides some sound absorption. Cementitious spray-on sound absorbing materials are particularly attractive for this purpose, although the most effective treatment would be 2-in.-thick 3-pcf glass fiberboard encased in Tedlar plastic and protected with perforated powder coated metal.

### 5.3.2 Wheel/Rail Howl

Noise at moderate to large radius curves may be greater than at tangent track because of wheel lateral oscillation caused by uncompensated differential wheel velocities and lateral slip. The problem is exacerbated by cylindrical wheels with low damping ratios, such as the BART aluminum centered wheel. Wheel howl is closely related to wheel squeal, but differs in that it is believed by this author to not reach a saturated condition limited by nonlinearity of the wheel/rail interaction and friction-creep curve. Also, wheel howl differs from squeal in that it tends to increase in level with train speed, whereas wheel squeal may actually be inhibited by increasing train speed, and wheel howl is generally of lower level than wheel squeal. Wheel squeal may be viewed as a lightly damped harmonic oscillator excited by random forces produced by lateral slip. The oscillation involves rigid motion of the tire controlled by the bending stiffness of the web and axle. Wheel squeal, on the other hand, occurs at a higher frequency of about 1,400 Hz, related to the first distortional mode of the tire. Wheel howl might also be considered as poorly developed wheel squeal and might transform into saturated stick-slip oscillation at about 500 Hz. Wheel howl, in the absence of corrugation, is not necessarily a serious curving noise problem, though corrugation at curves may increase the howl to unacceptable levels. Finally, the opinion of this author is that wheel howl does not exist or is not sufficiently high in level to be identified at transit systems which use solid steel wheels with tapered wheel treads, by far the most common configuration. The problem appears to be limited to BART with aluminum centered wheels and cylindrical wheels.

Wheel/rail howl may be indistinguishable from corrugation noise, and wheel howl may ultimately be traced to corrugation, though it has been observed after rail grinding. Short pitch corrugation and howl at BART are most prevalent at curves, though they are not limited to curves. Lateral oscillation at about 500 Hz, driven by lateral slip, appears to contribute to corrugation at curves. The condition of the track should be checked prior to investigating possible treatments for wheel/rail howl. If the rails are found to have visible corrugation, not necessarily of high amplitude, the methods discussed for treating rolling noise caused by corrugation should be considered first. That is, the rail should be ground to remove corrugation and restore a smooth running surface. Profile grinding to provide a rolling radius differential is discussed below and should be considered in developing the grinding program. If, after rail grinding is completed, wheel/rail howl persists, the additional treatment approaches listed in Table 5–6 can be considered, many of which are the same as those for wheel squeal and/or rolling noise. The reduction of wheel/rail howl is expected to reduce rail corrugation to the extent that the corrugation is produced by the abrasive motion of lateral vibration of the wheel tread at the howl frequency.

#### *5.3.2.1 Onboard Treatments*

Many of the onboard treatments, such as skirts and undercar absorption, identified for controlling rolling noise may also reduce wheel howl slightly, and the user is referred to the discussion presented above with respect to these treatments. Wheel taper, rail profile, and primary suspension compliance need to be considered as a whole with respect to curving performance and reduction of lateral slip which is the principal cause of wheel howl.

TABLE 5-6 NOISE CONTROL TREATMENTS FOR WHEEL / RAIL HOWL

| LOCATION | TYPE | NOISE REDUCTION - dBA - | COST | COMMENT |
|---|---|---|---|---|
| Onboard | Dry-stick friction modifier | Unknown | $1,400 /vehicle/year | Theoretically attractive, but untested |
| | Resilient wheels | 5 to 15 | $2,400 - $3,000 /wheel | Most effective proven technology |
| | Constrained layer damped wheels | 5 to 10 | $1,000 /wheel | Proven technology |
| | Wheel vibration absorbers | Unknown | $500 to $700/wheel | Not proven, but expected to be effective |
| | Ring dampers | Unknown | $75/wheel | Untested for howl |
| | Conical wheel taper | Unknown | $0 | Reduces lateral and longitudinal slip |
| | Longitudinally compliant primary suspension & steerable truck | Unknown | Unknown | Reduces lateral and longitudinal slip. May have problem with ride quality and truck dynamics |
| | Improved car body sound insulation | Depends on car body design | Unknown | For car interior noise |
| Trackwork treatments | Profile rail grinding | 5 to 7 | | Must be in conjunction with conical tread tapers |
| | Superelevation | Unknown, but expected to be minor | 0 | Theoretically attractive |
| Wayside | Treatments associated with rolling noise control | | | Does not eliminate howl |

*Speed Reduction.* Wheel/rail howl tends to decline or disappear with speed reduction. Therefore, if howl is excessive, speed reduction can be considered as a noise mitigation measure. This is likely to be incompatible with operational objectives for the transit system, but may represent a short-term solution.

*Resilient Wheels*. Resilient wheels have been shown to inhibit wheel howl at curves of 340- and 470-ft radius with smooth ground rail at BART. To the extent that the wheel/rail howl involves lateral vibration of the tire as it undergoes lateral creep or slip across the rail head, a reduction of wear and thus corrugation due to wear might be expected with the use of resilient wheels. However, light rail systems using resilient wheels experience corrugation at both tangent and curved track, indicating that resilient wheels do not prevent corrugation. The noise reduction achieved with resilient wheels is likely to result from its higher lateral compliance relative to solid wheels. The performance is similar to that obtained for wheel squeal and, indeed, the squeal reduction data reported for resilient wheels are based, in part, on wheel howl reduction data for BART. The comments provided in connection with wheel squeal control with respect to cost and limitations of resilient wheels for controlling wheel squeal are relevant here as well.

*Constrained Layer Damped Wheels*. Constrained layer damped wheels have been shown to reduce wheel/rail howl at 500 Hz at curves at BART. The treatment involved a constrained damping treatment adhered to the wheel web. However, most wheel damping systems are optimized for wheel squeal in the higher frequency range of 1,000 to 5,000 Hz; therefore, their vibration reduction effectiveness at the typical howling frequency range of 250 to 1,000 Hz may be marginal. In addition, a constrained layer treatment applied only to the wheel tire would not absorb energy at the web and therefore would be ineffective in controlling wheel howl. The damping treatment must be applied at the location of greatest bending strain, which occurs at the wheel web for howl frequencies. An advantage of damped wheels over resilient wheels is that they have the same rigidity as undamped solid wheels and damped wheels and have no elastomer springs which may generate heat due to hysteresis.

*Wheel Vibration Absorbers.* Wheel vibration absorbers may be more effective than constrained layer damped wheels for reducing wheel howl if the vibration absorber frequency can be tuned to about 500 Hz. This may require absorbers with considerably greater mass than those intended for conventional squeal control. Other than this, the same comments as given above for damped wheels apply. The cost of tuned wheel vibration absorbers is expected to be on the order of $400 to $700 per wheel. Costs might be lessened if ordered in sufficient quantities.

*Dry-Stick Friction Modifier.* Dry-stick lubrication with a friction modifier has been proposed to reduce the tendency for stick-slip and squeal at curves. No data have been obtained identifying the friction modifier as being effective at controlling howl at curves, but the theory of lateral stick-slip would suggest that unsaturated lateral oscillation of the wheel set might be reduced simply by reducing the negative damping associated with the negative slope of the friction-creep curve. Experiments are necessary to quantify any noise reduction that might be obtained prior to selection.

*Wheel Truing Profile.* Providing and maintaining a conical tread profile will encourage the high rail wheel to ride higher, close to the flange, thus allowing it to travel faster around the curve than the low rail wheel without slip. The effect is improved if the rail also is asymmetrically ground to move the contact patch of the high rail close to the gauge side and the contact patch of the low rail close to the field side. These provisions alone are not enough to prevent lateral creep because of the finite wheelbase of the truck, unless there is sufficient longitudinal compliance in the primary suspension to allow the axles to align themselves perpendicular to the tangent to the rail at the point of contact. (This is the motivation for the steerable truck.) Conversely, without a conical tread taper, there would be no force generated to cause the axles to align themselves perpendicularly to the rail, even with a longitudinally compliant truck. (See the discussion in Chapter 4 on wheel/rail noise generation concerning wheel squeal due to lateral stick-slip.)

*Primary Suspension Longitudinal Compliance*. As discussed above, reducing the primary suspension compliance in the longitudinal direction may allow the axles to better align themselves with curve radii and reduce or eliminate lateral slip due to finite truck wheel bases. However, reducing primary stiffness may not be effective with cylindrically tapered wheels. Further, ride quality and truck dynamics could be adversely affected by reducing the longitudinal stiffness. Testing and analysis should be conducted to ensure proper performance of the vehicle with reduced longitudinal stiffness.

#### *5.3.2.2 Trackwork Treatments*

There are few other trackwork treatments that might be applied to control wheel howl without corrugation. Trackwork treatments might include asymmetrical profile grinding and possibly correction of superelevation imbalance. To the extent that pinned-pinned vibration of the rail at about 500 Hz may be modulating lateral wheel/rail forces, reducing the fastener pitch to 24 in. may help to reduce howl, though this is strictly conjecture on the part of this author. Lubrication is definitely not an option.

*Profile Rail Grinding.* Asymmetrical profile grinding on an optimized grinding interval in conjunction with conical wheel tread profiles will provide a rolling radius differential and may promote self-steering of the axle to minimize or eliminate howl in the absence of corrugation at large radius curves. Further, there may be a reduction of any related rail corrugation rates. A programmable profile rail grinder is desirable to simplify set up and speed grinding operations at curves.

*Superelevation*. Superelevation tends to relieve the static load on either the high or low rail, depending on speed and degree of superelevation. Under light wheel load, the wheel tread may more easily slip over the rail when negotiating a curve, with attendant oscillation, than with a heavier wheel load. BART has reported just such an effect with respect to rail corrugation generation. Thus, matching the superelevation to train normal operating train speed would appear to be beneficial. Costs associated with maintaining superelevation balance are not known, but are expected to be part of normal track maintenance.

#### *5.3.2.3 Wayside Treatments*

Wayside noise control treatments such as sound barriers and receiver treatment are all effective in reducing howl noise at curves, just as they are in reducing rolling noise. However, barriers, berms, and receiver treatments would not eliminate the tonal character associated with howl and short-pitch corrugation and, because of the higher amplitude and detectability of howl at curves, may not be sufficient without a substantial increase in height, transmission loss, and so on. Further, wheel/rail howl, especially when associated with rail corrugation, can generate adverse community reaction at relatively large distances from the track and, because of adverse propagation conditions, barriers may not provide the necessary noise reduction to satisfy the community. Treating individual receivers over large distances from the track is uneconomical unless only a few receivers are involved. Thus, wayside treatments should not be the first line of defense in controlling howl due to stick-slip oscillation and/or corrugation. Subway wall treatment will be effective in reducing wheel/rail howl, just as it is in reducing squeal and rolling noise.

## 5.4 SPECIAL TRACKWORK

Special trackwork includes turnouts, crossovers, and switches, which may cause particularly intrusive impact noise. The theory of impact noise is discussed in the preceding chapter on wheel/rail noise generation. Impact noise from special trackwork is usually associated with switch frogs and crossover diamonds. In particular, impact noise is generated as the tire traverses the gap in the frog or diamond and is described by the theory of impact noise for a wheel traversing a gap. Noise control measures are thus directed at reducing or eliminating the frog gap.

### 5.4.1 Special Trackwork Designs for Noise Control

Noise control methods which may be applied to special trackwork are listed in Table 5–7 and include moveable point

**TABLE 5-7 SPECIAL TRACKWORK NOISE CONTROL TREATMENT SELECTION**

| Location | Treatment Type | Noise Reduction - dBA - | Cost | Notes |
|---|---|---|---|---|
| Trackwork | Moveable Point Frogs | 7 | $120,000/turnout | Requires additional signalling |
| | Spring frogs | 5 | $6,000/frog | May produce noise for trains turning out |
| | Flange Bearing | 3 | Nil | Standard Design for embedded girder rail track |
| | Welded "Vee" | 5 | Nil | No significant cost difference relative to standard frog |
| | Floating slab cut | 5 | Unknown | Reduces floating slab rumble in tunnels |
| Onboard | Undercar absorption | 0 to 2 | $3,500 per vehicle | |
| | Soft primary suspension | 0 to 6 | Nil | Any noise reduction depends on design. Noise reduction applies to vehicle interior noise |
| | Resilient wheels | 3 | $2400 to $3,000 / wheel | |
| Wayside | Sound barriers | 5 to 10 | $20/sf | Must account for low frequency impact noise |
| | Receiver sound insulation | 0 to 10 | $5,000 to 10,000 per residence | |
| | Subway wall and ceiling sound absorption | 5 to 7 | $10/sf | |

frogs, spring frogs, and embedded-track flange-bearing frogs. Reduction of impact noise is the result of reduction of impact forces; therefore, a reduction of truck shock and vibration may be expected, which may lead to reduced truck maintenance.

### *5.4.1.1 Moveable Point Frogs*

Moveable point frogs are most suited to high-speed turnouts. They are used in the railroad industry primarily as a means of reducing frog and wheel wear under high load environments. They are particularly effective for noise control because they virtually eliminate the gap associated with normal rail bound manganese frogs. Switches with moveable point frogs require additional signal and control circuitry compared with those with standard frogs; as a result, the total cost of a turnout with moveable point frogs is about $200,000, roughly twice that of turnouts with standard frogs.

### *5.4.1.2 Spring Frogs*

Spring frogs are suitable for low-speed turnouts and crossovers where use is occasional or emergency in nature, because ancillary noise may be produced by the spring frog for other than tangent running trains. Where frequent use of the turnout is expected, the spring frog may not be appropriate because of secondary noise related to spring frog actuation. The cost of a spring frog is roughly $12,000, twice the cost of a standard frog. The incremental cost of a turnout with spring frogs relative to the cost of a turnout with standard frogs is thus relatively small. Spring frogs are not appropriate for speeds greater than about 20 mph.

### *5.4.1.3 Flange-Bearing Frogs*

Flange-bearing frogs are used in embedded track and support the wheel flange as the tread traverses the frog gap. Flange-bearing frogs do not necessarily eliminate impact noise and vibration, and their effectiveness depends on the degree of frog and flange wear. Flange-bearing frogs are not used in high-speed sections of nonembedded track and therefore are not normally considered as a noise control treatment.

### *5.4.1.4 Welded V-Frogs*

Welded V-frogs have been tested at BART with high-speed heavy rail transit vehicles and have been found to be effective in controlling impact noise and vibration for trains operating at 50 to 70 mph.

### *5.4.1.5 Welding and Grinding*

Maintenance of frogs by welding and grinding the frog point results in lower impact forces and thus lower impact noise levels and wheel and frog wear.

*5.4.1.6 Floating Slab Cuts*

Substantial rumble noise is generated by special trackwork located on large continuous floating slabs. The floating slab acts as a sounding board and is particularly efficient in radiating noise. An effective method to control this type of noise is to cut the slab between the tracks. The cut should be 1 in. wide and should be filled with a closed cell foam neoprene or other suitable filler. The practicality of this may be limited by design details, slab reinforcement, trackwork geometry, and so on.

### 5.4.2 Onboard Treatments

There are few onboard treatments available for reducing special trackwork noise. However, some of the treatments identified for controlling rolling noise may be beneficial in reducing impact noise. These are undercar absorption and suspension design, both of which are applicable primarily to car interior noise control.

*5.4.2.1 Undercar Absorption*

Undercar absorption over the truck area should be effective in reducing special trackwork noise in subways with direct fixation fasteners by a few decibels in a manner similar to that expected for normal rolling noise. Sound absorption by ballasted track would obviate the usefulness of undercar absorption.

*5.4.2.2 Suspension Design*

Resilient bushings and springs must be included in the truck design to reduce impact noise transmission into the car body, and soft Chevron suspensions are expected to transmit less impact vibration and noise into the truck and car body than rubber journal bushing suspensions. Primary suspension modifications to existing vehicles are not normally considered for noise control, though experiments have been conducted with the prototype BART vehicle and the MARTA C-Car, the latter with respect to ground vibration reduction. Primary suspension design is a factor in new vehicle procurements, and specifications may be written to achieve desirable limits on car interior noise. Of particular concern are any drag links and traction linkages which may transmit impact vibration into the car body. These must be isolated with elastomer bushings.

*5.4.2.3 Resilient Wheels*

Resilient wheels reduce shock and vibration transmission into the truck and car body and thus may be expected to reduce vehicle interior impact noise at certain frequencies, though the A-weighted noise reduction may be slight. No data have been collected indicating the possible car interior noise reduction obtained with resilient wheels at special trackwork.

### 5.4.3 Wayside Treatments

Wayside treatments for impact noise from special trackwork include provision of sound barriers, receiver treatments, and subway wall sound absorption. These are discussed briefly below, and more detailed discussion may be found with respect to rolling noise control.

*5.4.3.1 Sound Barriers*

Because of the intrusive nature of impact noise at special trackwork, sound barriers may be desirable in residential areas near special trackwork installations not involving moveable point or spring frogs. Barriers would be impractical in street settings. Even with barriers, impact noise from special trackwork is easily detected in the presence of normal background noise, and low-frequency impact noise may penetrate building interiors more readily than the higher frequency components of normal rolling noise. Thus, barriers should be higher than normally required for controlling normal running noise.

*5.4.3.2 Location*

For new systems, every effort should be expended to locate special trackwork away from residential receivers or other sensitive receivers such as parks, schools, libraries, hospitals, theaters, auditoriums, and any other receivers where low ambient sound levels are a prerequisite for use. Relocation of existing special trackwork may not be practical, but should be considered as an option.

*5.4.3.3 Receiver Treatments*

Receiver treatments can be considered in reducing noise from special trackwork. However, the low-frequency energy and impulsive character of special trackwork noise requires that window glazing be more effective than would be considered for normal rolling noise. Laminated glass should be employed in most cases to provide additional noise reduction over that obtained with standard monolithic glass.

*5.4.3.4 Subway Wall Treatment*

Special trackwork noise in tunnels may be reduced by application of sound absorption to subway walls in the vicinity of special trackwork. Sound absorption reduces not only vehicle interior noise, but also station platform noise. Special trackwork is often located at the end of a station platform, and special trackwork noise reduction can improve the overall station noise environment. Costs for subway wall treatment are expected to be on the order of $10 per square foot.

## 5.5 REFERENCES


1. BC Transit, private communication. (1994).
2. BC Transit, private communication. (1994).
3. Nelson, J.T., Wayside Noise Measurements at the Sacramento RTD, Wilson, Ihrig & Associates, Inc., Technical Memorandum, TCRP Project C–3, Unpublished Final Report (June 1995).