CHAPTER 9 WAYSIDE TREATMENTS

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CHAPTER 9

WAYSIDE TREATMENTS

9.1 INTRODUCTION

Wayside treatments and structure treatments are applied separately of the vehicle, and trackwork, and include sound barriers, open cuts, tunnel wall sound absorption, station treatment, and receiver sound insulation. Each of these treatments is discussed below in respective sections.

9.2 SOUND BARRIER

A sound barrier is any solid obstruction which blocks the line-of-sight from a sound source to a receiver, and is one of the most effective noise control treatments after other treatments such as rail grinding and wheel truing, resilient wheels, and lubrication, are included in the design. Well-designed vehicle and track systems nevertheless produce wayside noise, and, because of close proximity of residences or other sensitive receivers, a sound barrier may be required to provide additional noise reduction over that provided by onboard and trackwork treatments, as well as provide a visual barrier between residential properties and a transportation corridor. Examples of sound barriers include concrete block or powder-coated sheet metal walls erected at the right-of-way line, or, preferably, at about 9 to 10 ft from the track center, and aerial structure mounted panels. Existing topography, earth berms, depressed rail corridor cuts, the edge of an aerial structure, and buildings can act as sound barriers. Barriers are commonly bare, without sound absorption; these are the least costly and most easily maintained. Absorptive barriers are used in special situations where barrier height must be limited, or where the geometry of the vehicle and barrier is such that sound absorption might be effective in controlling reflections, such as on an aerial structure. Sound barriers are most effective if placed close to the track or close to the receiver.

9.2.1 Source-Receiver Path

The sound attenuation provided by a barrier is not generally significant unless the sound source is blocked from the receiver's view. Once the direct path from source to receiver is blocked, the only remaining sound paths are

- Over the top of the barrier,
- Directly through the barrier, or
- A reflected path over the barrier.

The amount of sound energy that passes over the barrier can be reduced by increasing barrier height and length. The sound that passes directly through the barrier can be reduced to a sufficient level with essentially any wall material that has the structural integrity to stand by itself. As a general rule, a surface density of 4 lb/ft^2 is sufficient (*1*). Barrier design should minimize reflected sound from surfaces that direct sound over the barrier.

9.2.2 Sound Barrier Attenuation Prediction

Figure 9-1 shows the barrier source-path-receiver relationship. The wheels and rail are the sound source, the direct path to the receiver is indicated as *C*, and the diffracted path over the barrier is indicated as *A* and *B*.

The acoustical performance of a barrier is represented by its insertion loss for each octave band frequency. The insertion loss, at a given receptor location, is the difference in the octave band sound pressure levels before and after the barrier is "inserted" (constructed):

$$IL_{\text{barrier}} = L_{p(\text{before})} - L_{p(\text{after})}\ \text{dB}$$

This definition of barrier performance avoids the ambiguity which arises because the barrier, besides introducing attenuation due to diffraction, also reduces the attenuation due to the ground by increasing the height of the ray path above the ground. The effect of ground attenuation is often ignored for distances less than 100 ft between track center and receiver to obtain a conservative estimate of insertion loss. The following section uses the terms "insertion loss" and "attenuation" interchangeably, even though "insertion loss" has the more precise meaning.

Predictions of attenuation achieved with a barrier are based on the path length difference between the direct path and the diffracted path, assuming a characteristic spectrum of noise. Detailed calculations are made for each octave or 1/3-octave band, based on the Fresnel Number associated with the band center frequency and path length difference. Given the source, barrier, and receiver geometry illustrated in Figure 9-1, the path length difference is

$$D = A + B - C \qquad \text{Eq. 9.1}$$

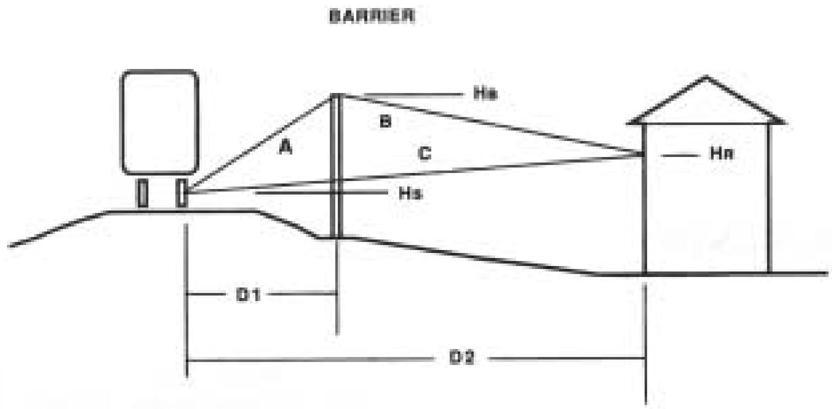

FIGURE 9-1 SOUND BARRIER SOURCE-PATH-RECEIVER RELATIONSHIP

where

$A = ((H_B - H_S)^2 + D_1^2)^{1/2}$
$B = ((H_B - H_R)^2 + (D_2 - D_1)^2)^{1/2}$
$C = ((H_S - H_R)^2 + D_2^2)^{1/2}$
H_S = Source Height
H_B = Barrier Height
H_R = Receiver Height

The distance, $(A + B)$, is the shortest path over the barrier's top edge from the source to the receiver and C is the direct distance through the barrier from the source to the receiver. For rail transit systems, the source is usually assumed to be at axle height (1.5 ft above the rail) at the near rail or the track center line, though detailed octave band calculations could consider source heights as a function of octave band.

The following formula for theoretical point-source barrier insertion loss (*2*) has been used in many sound barrier prediction methodologies:

$$IL = (20 \text{ Log } (2\text{p}N)^{1/2}/\tan h\ (2\text{p}N)^{1/2}) + 5 \text{ dB for } N \geq -0.2, 0 \text{ otherwise} \quad \text{Eq. 9.2}$$

where

$N = \geq \pm 2/\lambda\ (P)$
λ = wavelength of sound, *m*
d = straight-line distance between source and receiver, *m*
P = Path-length difference
+sign = receiver in the shadow zone
–sign = receiver in the bright zone

For an incoherent line source, such as a moving train, the barrier insertion loss becomes (*3*)

$$IL = -10 \log \int 1/\Delta\alpha 10^{-(IL_{point}/10)} \quad \text{Eq. 9.3}$$

where
$\Delta\alpha$ is the aspect angle of the closest part of the line source.

The barrier insertion loss for an incoherent line source includes the effect of directivity of sound radiation. Rail transit noise radiation has been shown to follow a dipole radiation pattern (*4*,*5*). The effect of the dipole radiation pattern is to reduce the contribution of noise from portions of the track at large angle of incidence relative to the track, compared with the contribution from portions of the track immediately opposite the receiver.

For a given path length difference, the barrier insertion loss is a function of frequency. Barriers are more effective at controlling high-frequency, short-wavelength sound than low-frequency, long-wavelength sound. Based on a typical frequency spectrum for rail transit noise, equations can be developed to calculate insertion loss as a function of path length difference specifically for transit systems.

Figure 9-2 presents curves that illustrate insertion loss based on various assumptions. The top curve is a theoretical curve based on Equations 9.2 and 9.3, ignoring the effect of directivity. The theory predicts 5 dBA attenuation even for very small path length differences; however, this cannot be depended upon for field installations, and is usually ignored in practical situations. For example, the 5 dB attenuation at grazing incidence is due to interruption of the line of site between the reflected image source on a hard surface and the receiver. For typical transit systems, there is usually no reflected image due to absorption by existing ballast or ground cover. Similarly, transit systems on aerial structures do not produce a significant virtual image source due to shielding afforded by the edge of the structure. An elevated receiver, looking down on an aerial structure with direct fixation track, would observe a virtual image of wheel rail noise reflected from the aerial structure deck. Similarly, a virtual image would exist for embedded track, for which the 5 dB attenuation at grazing incidence would apply. The following equation is suggested for use with non-absorptive transit barriers (*6*).

$$IL = \text{MIN}[12 \text{ or } 5.3 \log(P) + 6.7] \quad \text{Eq. 9.4}$$

Figure 9-2 presents examples of measured barrier insertion loss for several field installations. One example is the

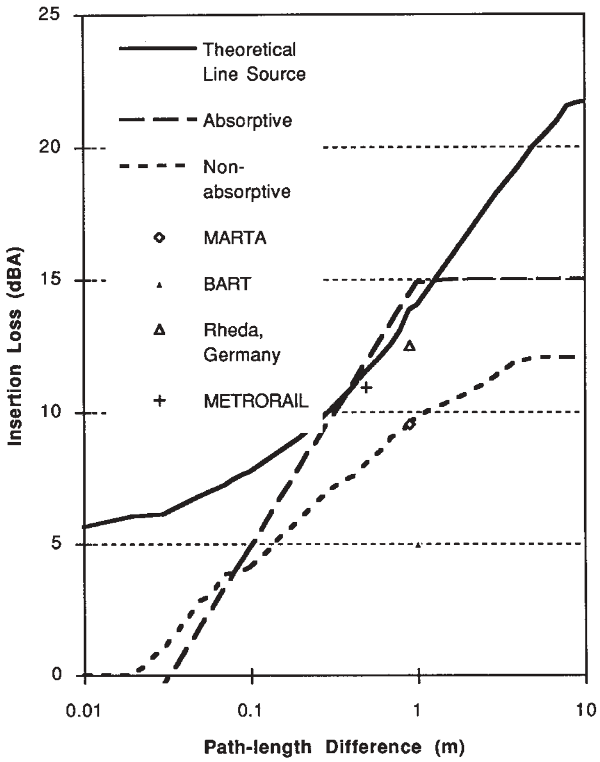

FIGURE 9-2 SOUND BARRIER INSERTION LOSS FOR RAIL TRANSIT SYSTEMS

measured insertion loss for barriers on a MARTA elevated structure. As indicated in Figure 9-2, the theoretical insertion loss curve overestimates the insertion loss achieved with the MARTA aerial structure barriers. Equation 9.4 was developed to reflect the attenuation measured at MARTA. Using this equation will result in a conservative barrier design which will be effective over a wide range of meteorological conditions. Since wind, thermal gradients, and other conditions can significantly change the performance of a barrier, the theoretical curve defined by Equations 9.1 and 9.2 should not be used directly for design purposes.

9.2.3 Absorptive Sound Barriers

The following equation has been developed to account for the improved insertion loss provided by barriers with sound absorptive material on the side of the barrier facing the tracks (*7*):

$$IL = \text{MIN}[15 \text{ or } 10 \log (P) + 9.7] \qquad \text{Eq. 9.5}$$

As can be seen in Figure 9-2, the insertion loss calculated from Equation 9.5 approaches the theoretical line source curve for path length differences between 0.3 and 1 m. However, experimental data do not indicate that absorption is as effective as predicted by Equation 9.5. On ballast-and-tie track, for example, there is already substantial absorption provided by the ballast, and the additional absorption applied to the barrier may have a limited effect. On aerial structures with direct fixation track and closed decks, addition of absorption to the trackside surface of the aerial structure sound barriers is usually assumed to provide about 3 dB additional insertion loss.

Transit system barrier insertion loss predictions become more complicated in actual barrier designs. For example, the equations given above are for infinitely long barriers; not for finite length barriers. Because noise radiation from a rail transit vehicle has a dipole radiation pattern, the prediction of barrier performance with the infinite barrier assumption is usually adequate for maximum sound levels. However, if energy equivalent noise levels are to be considered, as recommended by the Federal Transit Administration (*8*), then the overall passby signature must be considered, which places greater demand on barrier calculations.

Some designers have attempted to use the FHWA STAMINA 2.0 computer model for rail transit noise prediction, employing *ad hoc* noise emission levels developed for transit vehicles. STAMINA assumes a monopole radiation pattern in design, so that it is not entirely appropriate, though the results are reasonably accurate within a few decibels and adequate for design. STAMINA may over-predict noise levels to some extent.

9.2.4 Practical Design Considerations

Despite the problems associated with applying complex theoretical models of sound barrier insertion loss, adequate barrier design can be developed following relatively simple design principles. A general rule of thumb is that 5 dBA attenuation is relatively easy to obtain; 10 dBA attenuation can be achieved with careful attention to the barrier design; and 15 dBA is usually the physical limit in field installations of barriers, although it is difficult to achieve with practical designs. Barrier theory indicates that attenuations up to 25 or 30 dBA can be achieved, and laboratory experiments have generally proved the validity of the theory. However, the theory does not include the effects of temperature variations, air turbulence, ground effects, reflections, and incoherent line sources, all of which act to reduce the effectiveness of barriers. The following are design guidelines:

- The barrier must break the line-of-sight path between the noise source and the receiver and block all possible sound propagation paths from the source to the receiver.
- Open areas in the barrier, such as maintenance access ports, for example, should be kept as small as possible. They provide flanking paths for the sound to "short circuit" the barrier.
- The barrier should be constructed of a material that is sufficiently heavy to control the transmission of sound

through the barrier. In most cases, virtually any material that is sufficiently strong to withstand wind loads and provide structural support will also be sufficiently massive to control sound transmission through the barrier.
- Barriers must block both the direct path and any reflected paths between the source and the receiver.
- The most effective location for barriers is either close to the noise source or close to the receiver.

In almost all cases, these basic guidelines will result in a barrier with 5 to 8 dBA attenuation. To obtain greater attenuation, more care must be taken in the design of the barrier.

9.2.5 Case Studies

Shown in Figure 9-2 are the attenuations that were achieved with barriers on aerial structures at BART and MARTA. As discussed previously, the nonabsorptive insertion loss curve was developed to reflect the MARTA insertion loss data. In contrast, the BART barriers are 4 to 5 dBA less effective than predicted by the nonabsorptive insertion loss curve, due to a gap between halves of the elevated structure and, to a lesser extent, a gap at the bottom of the barrier. Measurements of the BART barriers with the gap at the bottom plugged by rubber gasketing material reduced the wayside noise levels further by 1 to 2 dBA, illustrating the importance of designing barriers with minimal openings.

Sound insertion loss measurements were conducted for barriers constructed on an elevated structure of the Miami Dade County Metrorail system. (*9*) The goal of the insertion loss measurements was to determine the effect of sound absorption material on the side of the barrier facing the tracks, and the effect of filling a 1- to 2-in. gap between barrier panels at each pier and a 1-in. gap at the junction of the barrier and the guideway. The measurements indicated insertion losses of 8 dBA with no sound absorption treatment, 9 dBA with barrier/guideway gap sealed, 10 dBA with barrier/guideway and barrier panel gaps sealed, and 11 dBA with all gaps sealed and sound absorption treatment. That is, the sound absorption treatment provided approximately 1 dBA of benefit at rail height. The measurements also indicated 3 dBA of benefit at a receiver height higher than the top of the barrier, suggesting that sound absorption material is only of significant benefit in the case where receptor locations are above the barrier top. Recent discussions with the Miami Metrorail staff indicate that these barriers were dismantled because of the degradation due to weather of the sheet metal barrier material and absorptive material. The Metrorail insertion loss data for absorptive barriers are shown in Figure 9-2 and are in close agreement with the absorptive barrier design curve.

A rail barrier insertion loss measurement program in Germany was conducted with intercity trains to determine the effects of various barrier top configurations (*10*). The barrier tops included a standard vertical wall, a slant-top, and a y-top. For a 2-m high absorptive barrier 3.8 m from the track centerline, 13 dBA attenuation is reported. This value is also shown on Figure 9-2 for comparison with the absorptive barrier design curve. For a slant-top absorptive barrier, 14 dBA attenuation is reported. The y-top absorptive barrier achieved 16 dBA attenuation. Barrier insertion loss generally increases with increasing width or complexity of the barrier top. However, the cost of a barrier with a complex top is generally equivalent to a taller conventional barrier with equivalent insertion loss. One may conjecture that the improved performance of the y-top barrier is due to locating the crown of the barrier closer to the source, thus increasing the source-receiver path-length difference.

Another example of a nonstandard barrier top design is cylindrical absorber-topped barrier. The barrier consists of a vertical wall with a cylinder of sound absorptive material along the top. This type of barrier is being used in Japan to shield residents from highway noise. Insertion loss measurements show that these barriers can be 1.5 to 2 m lower than an acoustically equivalent conventional barrier. The cost of these barriers is expected to be approximately the same as a higher conventional barrier with equivalent insertion loss.

9.2.6 Source Height

Source heights can be considered in detailed calculations of sound barrier insertion losses for individual 1/3- or 1/1-octave bands, using the Fresnel Number in the calculations, and more complex formulas than those provided for single-number (for example, A-weighted sound level) determinations. However, source heights are difficult to define, though there are some isolated literature which shed light on source height as a function of frequency.

Barsikov et al. (1987) has measured source heights for trued wheels on smooth ground rail to be about 0.2 m above the top of rail for frequencies in excess of about 1,000 Hz for high-speed trains. With a damping treatment applied to the face of the wheel center to damp resonances in the wheel, the source heights are reduced to the top-of-rail, consistent with assuming that the rail is the dominant radiator. Beguet et al. (1988) provided data on source heights for low speed trains of 60 and 100 km/hr. At 250 Hz, the source location is prominent at the center of the wheel, though one must remember that the wavelength of sound at 250 Hz is about 4 ft, larger than the diameter of the wheel, so that localizing the source at the center of the wheel may not be subject to interpretation. At the 500 and 1,000 Hz octaves, the source appears to be concentrated at the wheel/rail contact point. At 2,000 Hz, the source appears to be distributed over the surface of the wheel, and is believed to be related to the modal behavior of the wheel.

Sources were investigated at the Deutsche Bundesbahn in Europe (*11*) and the results of these studies indicate

- The main sound source is the wheel disc at frequencies above 1,600 Hz.
- Between 500 and 800 Hz, the truck frame appears to be most important. (This may not be directly applicable to rail transit systems at lower speeds than those of the Deutsche Bundesbahn which produce substantial air turbulence in the region of the truck).
- The rail does not appear to be a significant source of noise over the important frequency range of noise radiation, though when the ties and ballast are included, the rails may dominate at frequencies less than 250 Hz.
- The car body is not a significant source of noise.

Tests at the SNCF indicate that the running gear, as apposed to the rail and car body, is the major source of noise in the 500, 1,000, and 2,000 Hz octave bands for various railroad vehicles (not transit). The definition of running gear was not provided, but is assumed to include the wheels, gearbox, and motor (*12*). These data suggest that a reasonable source height for barrier calculations is that of the axle center.

For practical design calculations, the source should be assumed to be at the axle elevation, over the near rail. This will produce conservative estimates of insertion loss, and will result in over design of the barrier by, at most, about 1.5 ft. However, the assumption of source at axle height for wheel squeal is not necessarily conservative, because much of the noise energy is radiated by the wheel.

9.2.7 Berms

Berms are an attractive alternative to sound barrier walls. Further, berms appear to provide additional sound absorption over walls of equivalent height. The reason is conjectured to be the sound absorption afforded by the ground cover, which is usually grass, ivy, or some other foliage providing resistance to erosion. However, the width of the berm crown also increases the path length difference, which may have some effect. The California Department of Transportation assigns an additional 3 dB to the noise reduction afforded by an equivalent height wall. For prediction purposes, the conservative approach would be to ignore the additional insertion loss due to absorption. Regardless of the noise reduction performance of berms *vis-a-vis* walls, berms are often more visually attractive than walls. Berms require greater footprints, however, to maintain slope stability, though certain "geo-cloths" or retaining structures can be employed. The smaller footprint of a wall relative to that of a berm often makes the wall the more practical alternative.

9.3 SUBWAY WALL TREATMENT

Vehicle interior noise is an important factor in wheel/rail noise control, particularly in subways. Noise reductions can be readily achieved in subways by treating the subway walls and ceiling with sound absorbing materials. The only systems where this has been done extensively are the Boston MBTA Red Line and the Toronto Transit Commission, where 1 to 2 in. of spray-on cementitious acoustical treatment were applied. Without treatment, the only absorption available is that due to the concrete subway walls, ballast (if part of the trackwork), the vehicle, and radiation losses up and down the tunnel away from the train. Subways with ballasted track would not benefit from subway wall treatment as much as those with concrete inverts and direct fixation track, because the ballast provides some sound absorption.

The noise reduction can be estimated on the basis of the effective length of the train which may be considered as a sound source for a particular receiving vehicle. That is, with treatment, the most significant sound source is the vehicle in which a receiver is riding. Without treatment, the receiving vehicle and 2 to 4 more vehicles may be considered as sources. Thus, treatment may reduce the effective source length of the train to that of a single vehicle. For a 10-car train, the noise reduction provided by the treatment would be on the order of 7 to 10 dB, while for a single-car train, the noise reduction would be less, because much of the noise energy is radiated up and down the tunnel away from the vehicle. Still, the noise reduction achieved with subway treatment might be on the order of 3 dB for a single vehicle.

The cost for subway wall cementitious sound absorption is on the order of $7 to $10 per square ft, depending on quantity, labor rates, and ease of installation. As a practical matter, only the upper half of the subway wall and all of the ceiling may be treated, so that the cost per lineal foot will be about $180 to $260 per lineal foot.

9.4 ACOUSTICAL TREATMENT OF STATIONS

Transit system designers have often used acoustically reflective materials, such as painted concrete or ceramic tile, on all surfaces of train platform areas, for durability, abuse resistance, and ease of cleaning. With these materials, train noise is not dissipated, resulting in a reverberant and noisy space. Wheel/rail noise may be reduced in transit system stations by applying sound absorbing materials to exposed surfaces. Acoustical treatment of the walls and ceilings prevents excessive build-up of reverberant sound energy, substantially reduces train, ventilation equipment, and crowd noise, and greatly improves the intelligibility of public address systems, an important factor in station design. Most new subway stations in the United States are acoustically treated, examples of which include stations at BART, WMATA Metro, MARTA, Baltimore Metro, NFTA (Buffalo), TTC, MTA NYCT, Chicago CTA, and LA Metro.

The type and placement of acoustical lining determine treatment effectiveness. There is a wide assortment of acoustically absorptive materials, and the choice of the appropriate material is based on the amount of required absorption, architectural considerations, ability to withstand train movement induced pressure transient loading and buffeting in stations, resistance to mechanical abuse, safety considerations such as flame resistance, cost, and other considerations. In most cases glass fiber products are the most economical treatment. However, there are many other products that should be considered, such as spray on cementitious sound absorption.

Barriers may be used between the tracks to block sound from trains passing through stations. This type of treatment has been used in New York, though there are concerns regarding safety. As a rule, this type of treatment would be less needed if the trainway ceiling and station walls and ceiling were treated with acoustical absorption, and if the rails and wheels were maintained in good condition.

9.4.1 Design Guidelines

The APTA Guidelines include design goals for maximum levels of station platform noise, and are summarized in Table 9-1. As long as the noise created by the trains is consistent with the APTA Guidelines for wayside passby noise, then following the guidelines for treatment of walls and ceilings in platform areas as listed in Table 9-2 will ensure that the design goals for station noise levels are achieved.

The design guidelines in Table 9-2 are based on an efficient use of materials. The recommended sound absorption treatment will control reverberation and train noise efficiently. Further noise and reverberation control is possible by using greater amounts of treatment, but doubling the amounts would have only a small additional effect on the acoustical environment, and would not justify the added cost. Thus, the use of sound absorbing materials is to some extent governed by the law of diminishing returns; beyond a certain point additional treatment becomes uneconomical and inefficient, and other noise control procedures should be considered.

9.4.2 Materials

A number of treatment configurations are available for the ceilings and walls of the train rooms. Glass fiber is one of the most efficient and inexpensive sound absorbing materials available. Absorption coefficients for various thicknesses of glass fiber board are presented in Table 9-3. Table 9-4 indicates some of the representative sound absorbing materials used for treatment of stations. Materials equivalent to the glass fiber products listed in Table 9-4 should be given equal consideration. The last two materials listed in Table 9-4 are appropriate only where flammable materials are acceptable.

TABLE 9-1 DESIGN GOALS FOR PLATFORM MAXIMUM NOISE LEVELS

CONDITION	LEVEL - DBA
Trains Entering or Leaving	80-85
Trains Passing Through Station	85
Trains Stationary	68

TABLE 9-2 DESIGN CRITERIA FOR ACOUSTICAL TREATMENT OF STATION PLATFORM AREAS TO CONTROL TRAIN NOISE

Maximum Reverberation Time	500 Hz	1.5 sec.
Treatment Area	Wall and Ceiling	35% (1)
	Under Platform Wall and Overhang	100%
Ceiling and Wall Treatment Properties	Minimum Absorption Coefficient at 500 Hz	0.6
	NRC	0.6
Under Platform Treatment Properties: Minimum Absorption Coefficient (3 to 4 In. Thickness)	250 Hz	0.4
	500 Hz	0.65

TABLE 9-3 TYPICAL SOUND ABSORPTION COEFFICIENTS TO BE EXPECTED FROM GLASS-FIBER SOUND ABSORBING MATERIALS MOUNTED DIRECTLY AGAINST A CONCRETE SURFACE

FREQUENCY - HZ	125	250	500	1,000	2,000
1 In. Thick Glass Fiber	0.08	0.30	0.65	0.80	0.85
2 In. Thick Glass Fiber	0.20	0.55	0.80	0.95	0.90
3 In. Thick Glass Fiber	0.45	0.80	0.90	0.95	0.90

TABLE 9-4 SOUND ABSORBING MATERIALS FOR CONSIDERATION FOR ACOUSTICAL TREATMENT OF STATIONS

MATERIAL		APPROXIMATE ABSORPTION COEFFICIENT WITH RIGID BACKING	
		250 HZ	500 HZ
4 In. Thick Geocoustic Block 1 x 1.5 Ft Slotted	Unspaced	1.0	1.06
	Spaced 2 In. Both Directions	0.90	1.06
	Spaced 6 In. Both Directions	0.60	0.66
4 In. Thick Geocoustic Block 1 x 1.5 Ft Perforated	Unspaced	0.79	0.84
	Spaced 2 In. Both Directions	0.82	0.94
	Spaced 6 In. Both Directions	0.53	0.59
0.842 In. Thick Geocoustic block 1 Ft x 1.5 Ft Perforated	Unspaced	0.79	0.73
	Spaced 2 In. Both Directions	0.74	0.71
	Spaced 8 In. Both Directions	0.42	0.60
2 In. Thick Glass Wool of 2 to 6 pcf wrapped with glass cloth		0.60	0.80
2 In. Thick Owens-Corning Aeroflex Duct Liner 3 pcf or Type 702 board faced with vinyl or neoprene		0.55	0.80
2 In. Thick Owens-Corning Type 703, 704, or 705 Board Faced with Glass Cloth		0.55	0.85

9.4.3 Mounting

Sectioned or continuous panels (consisting of either a metal or plastic slit-and-slat system or a perforated metal facing) with fiberglass or cellular glass blocks between the facing and the concrete surface are appropriate for treating flat, continuous surfaces and platform or mezzanine ceiling areas. In trainway areas, if a continuous panel system is assembled such that there exists an air gap between the back of the panels and the concrete backing (panels furred away from the walls), or a suspended acoustical tile ceiling is used, gaps or openings must be provided around the panel edges or elsewhere to permit free air flow to the region behind the panel. If pressure equalization provisions are not provided, the loading due to air pressure transients can eventually cause fatigue failure of the fastenings, allowing the panels to come loose from the mounting surface and fall, possibly injuring personnel and patrons. Trainway acoustical treatment in station areas should be designed to withstand air pressure transient loadings of about 15 psf.

Panels with perforated metal or slit-and-slat facings — in underplatform, ceiling, and wall installations — should have a dimpled screen placed between the metal facing and the face of the acoustic blanket to establish an airspace of about $^1/_2$-in. thickness between the perforated facing and the blanket or glass-cloth bag. This airspace serves two purposes: (1) it allows the sound waves to diffuse over the entire face of the acoustic material, thereby assuring full efficiency as a sound absorber; and (2) it allows free airflow for pressure equalization, thus preventing loading of the facing by air pressure transients produced by the train.

Note that several combinations of spaced and unspaced Geocoustic Blocks are listed. The absorption coefficients for the spaced configurations are based on the gross area of the treatment, i.e., the block area plus the area of the spaces between blocks. Use of spaced configurations can result in material economy, but to avoid loss of low-frequency absorption, the 4-in. thick units should be spaced not more than 6 in. apart and the 2-in. thick units not more than 4 in. apart. For lowest cost and nonflammability, Geocoustic Blocks should be specified unpainted, and without surface coating or wrapping.

9.4.4 Treatment Locations

Provision of sound absorbing materials is appropriate in subway stations at underplatform areas, platform walls and ceilings, and in enclosed concourse spaces such as fare collection areas, stairs, escalators and corridors. Similarly, enclosed areas of above-grade stations should have ceiling- and wall-mounted absorption treatment to create an attractive acoustic environment for transit patrons. Fortunately, the platform areas of most surface stations are only partially enclosed, thus reducing reverberation and noise, and additional acoustical treatment to control reverberation is generally not needed in surface station platform areas or other spaces with large openings to the outdoors. However, sound absorption should be considered for the underside of canopies to control reflected train noise beneath the canopy. The absorption can be particularly important when the station platform is located in a highway median and patrons are exposed to high levels of highway traffic noise (*13*).

9.4.5 Example of Typical Station Treatment

Figure 9-3 presents an example of an acoustically treated subway station. The figure illustrates the use of a platform

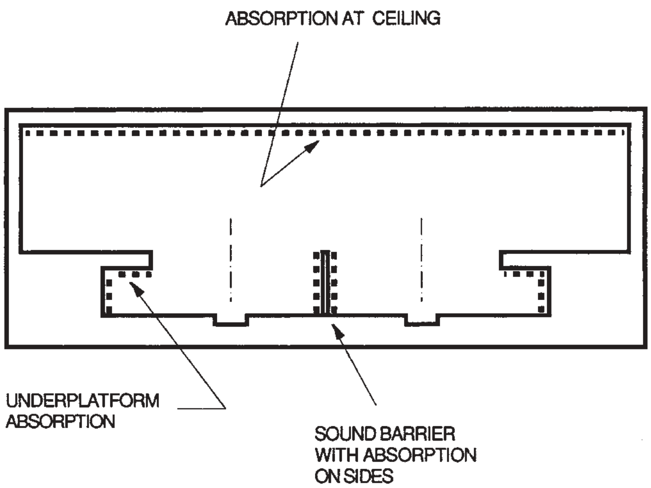

FIGURE 9-3 ACOUSTICAL TREATMENT OF STATION PLATFORM AREAS

height sound barrier to control train noise and appropriate locations for acoustical absorption. The noise sources on a transit car are primarily located in the confined space beneath the transit cars. Sound absorbing materials located on the trackbed or on the walls of the underplatform areas effectively absorb sound energy close to the source, and reduce the level of train noise on the station platform. The underplatform acts as an acoustically lined plenum when the train is in place, and is thus very effective in controlling noise, especially with between-track platforms. For double track configurations with platforms on either side of the tracks, the plenum noise reduction is only effective for noise produced by the wheels and rails located adjacent to the platform.

9.4.6 Sound Barriers

On side platform stations, further reductions can be achieved by using absorptive sound barriers to block noise from far track trains. Barriers only need to be as high as the platform level to achieve significant reductions of train noise, because wheel/rail noise originates beneath the cars. Sound absorption should be provided on both sides of the barrier where direct fixation track is employed. Without sound absorption, there would be little reduction of noise. For ballasted track, the ballast provides substantial absorption, and there is no need for absorption to be applied to the barrier. A platform height barrier between the near and far tracks of a side platform station can reduce sound levels on the platform by as much as 10 dBA (*14*). The actual amount of reduction is dependent on the design of the barrier and the measurement location. The greatest reduction occurs on the far platform, where the wheels and rail would otherwise be in full view of patrons, but there is also some reduction on the near platform.

9.4.7 Treatment Effectiveness

Figure 9-4 indicates reverberation times measured in WMATA Metro and BART subway stations, before and after installation of acoustical treatment on ceiling and underplatform overhang surfaces. The reverberation times measured in treated BART and WMATA stations are typically 1.3 to 1.5 sec at 500 Hz, as compared with 7 to 9 sec for untreated stations. Train noise levels in acoustically treated stations are much more acceptable than those found in older systems with completely untreated, highly reverberant stations.

Figure 9-5 shows noise levels on the platforms for trains passing by at 40 mph at several subway stations. The noise levels at BART and WMATA platforms are in the range of 87 to 89 dBA. Noise levels in untreated Chicago CTA stations, under similar operating conditions and using similar trains, are as high as 108 dBA on the platform of stations with concrete trackbed and 93 dBA on the platform of stations with ballast-and-tie tracks. The 15 dBA difference due to the ballast confirms that the ballast provides a significant amount of sound absorption which both absorbs sound at the source and reduces the reverberant sound energy build-up.

Noise measurements inside WMATA Metro cars indicate that acoustical treatment of subway stations can substantially reduce car interior noise levels (*15*). Figure 9-6 shows the results of these measurements. In a box structure with no sound absorption treatment, the interior noise level for a 2-car train operating at 40 mph was 79 dBA, whereas in passing through an acoustically treated station the interior level was 68 dBA. The same type of measurement indicated 64 dBA for at-grade ballast-and-tie stations, where no reflective sound impinges on the transit car.

Figure 9-7 presents typical noise levels, measured in TTC tunnel stations having sound absorption treatment on the underplatform overhang surfaces only (an insufficient amount to control reverberation and allow intelligibility of the public address system), and in a station in which the entire ceiling, as well as the underplatform, has been treated (*16*). The range shows the typical maximum levels that occur on the station platforms as trains arrive and depart. The sound absorption on the ceiling in this case is provided mainly by a suspended acoustical tile ceiling, an arrangement which gives nearly uniform absorption and noise reduction over the entire frequency range relevant to wheel/rail noise. The effective noise reduction is very dramatic—about 13 dBA.

Figure 9-8 compares noise levels observed in two BART stations: one with underplatform overhang and ceiling treatment, and the second with the ceiling treatment only (*17*). Both stations had sufficient acoustical treatment to reduce the reverberation time to about the same range, i.e., about 1.3 sec at 500 Hz. However, they were different in that the underplatform surfaces at the Lake Merritt Station had a complete and continuous treatment of 4-in. thick glasswool with a sheet plastic cover, while the 19th Street Station, at the

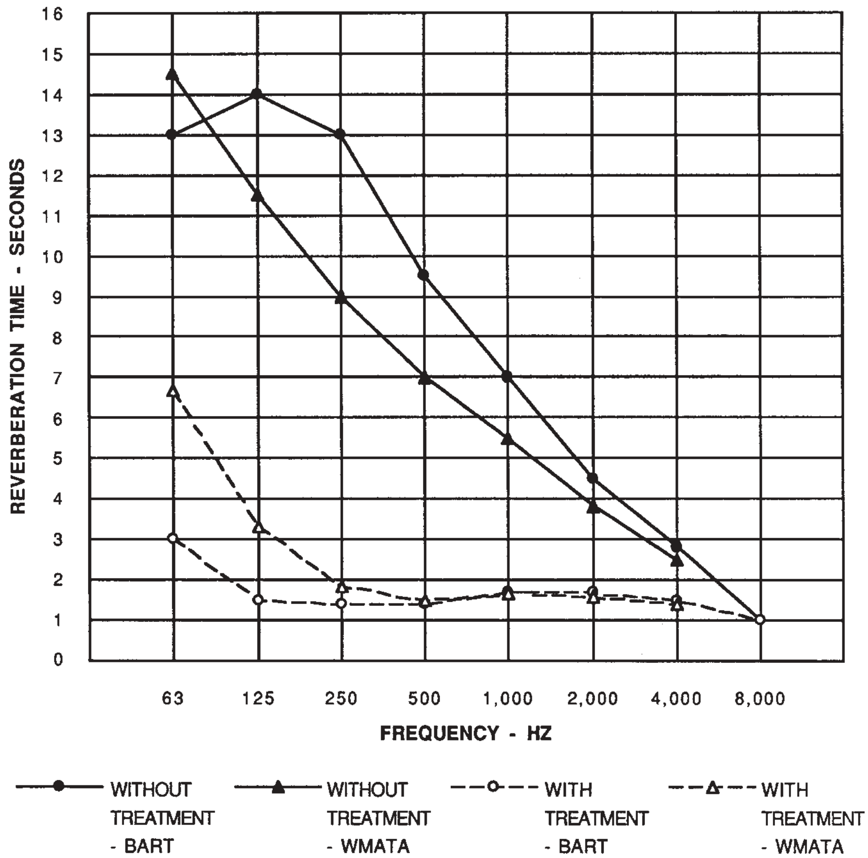

FIGURE 9-4 REVERBERATION TIMES FOR TREATED AND UNTREATED STATIONS

time of measurement, had almost no acoustical treatment under the platform edge; only one row of acoustical tile units spaced at about 0.6 m on center.

Figure 9-8 shows the dramatic effects of treating the relatively small area under the platform (*18*). In the 19th Street Station, where the underplatform treatment was omitted, the average noise level was about 5 dBA greater; in the middle and low frequencies, the difference in noise level was 5 to 8 dB, illustrating the importance of proper placement of the sound absorbing material. To obtain full benefit from acoustical treatment, continuous treatment must be placed in the underplatform overhang area, including the underside of the overhang and the rear wall beneath the overhang.

9.4.8 Underplatform Treatment

For underplatform overhang treatment, a recommended material assembly is a 3-in. to 4-in. thickness of nonflammable glasswool with an appropriate cover of glass fiber cloth or nonflammable plastic film of not more than 0.004 in. thickness, and a facing of expanded metal or hardware cloth. Cellular glass blocks of 2- to 4-in. thickness are a recommended alternative for underplatform overhang treatment.

The material should be mounted to cover as much of the underplatform area as possible. At stations with significant platform overhangs, sound absorbing material should be placed on the underside of the overhang surface as well as the vertical wall. The minimum treatment for the underplatform area is a 2.5-ft wide strip of continuous treatment on the vertical rear wall surface and complete coverage of the underside of the platform overhang.

If glass fiber wrapped in glass cloth is used for the underplatform treatment, the panels should be held in place with either an expanded metal facing, hardware cloth facing, or perforated metal facing. For center platform stations, expanded metal or hardware cloth is the most economical material since the material is not visible to patrons. For a side platform station, where the material is visible to patrons on the opposite platform, a better appearance can be obtained with perforated metal facing. Perforated metal or slit-and-slat facings should have open areas of at least 10% ($^1/_8$-in. diameter holes at $^3/_8$-in. center-to-center) or, preferably, 20% of the total area. Either expanded or perforated metal facings can be attached to the underplatform surfaces with simple metal brackets. The sound absorbing materials and retention hardware must be able to withstand high pressure wash and other cleaning methods that might be employed in subway environments.

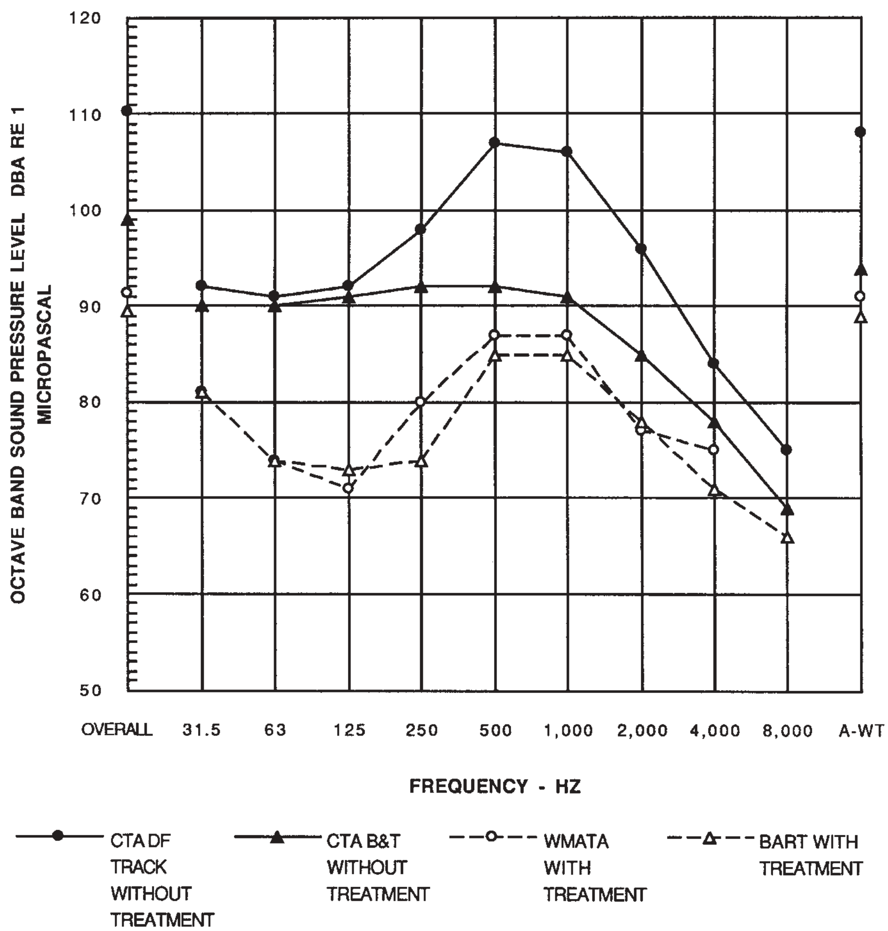

FIGURE 9-5 SUBWAY STATION PLATFORM NOISE LEVELS WITH TRAINS PASSING THROUGH AT 40 MPH

9.4.9 Mezzanine, Entrance, and Corridor Treatments

The design recommended for platform and mezzanine ceilings includes a 2-in. thick layer of glasswool with appropriate covering, and either perforated sheet metal or slit-and-slat configuration facings. A treatment with 1-in. thick glasswool is sufficient in other areas of the stations, such as at entrances, corridors, etc.

A basic panel system for ceilings and walls in the mezzanine and corridor areas can provide acoustical absorption very simply. The panel may be of perforated metal, a slit-and-slat configuration of boards or metal, or some form of architectural trim. The latter design should have at least 20% open area and no bars or sections with width greater than 3 in. between the openings. Such an arrangement will provide a completely transparent acoustical face. Acoustical material can then be located $^{1}/_{2}$ in. to 6 in. behind the face; cellular glass blocks or nonflammable glasswool of 2-in. thickness would be suitable materials. Also, as for mezzanine areas, panels of glasswool blankets with perforated metal facing can be used. A consideration when designing treatment for mezzanine areas is that, depending on station configuration and ventilation, they may be subject to dust (including brake dust) from the trains.

For areas other than platforms and mezzanines, ordinary acoustical tile or panels of $^{3}/_{4}$-in. to 1-in. thickness are appropriate. These materials—which may be of compressed glasswool or other appropriate fire resistant cellular material—can be of the type with painted or vinyl facing.

At corridors and entrances, the sound absorption treatment can be the same as that described above for platforms and mezzanines, or it can consist of an application of $^{3}/_{4}$-in to 1-in. thick acoustical tile, acoustical ceiling board, or cellular glass blocks. Another choice might be a sound absorption assembly, such as perforated sheet metal with fiberglass blankets behind the sheet metal facing. The absorption coefficient should be at least equal to the value listed in Table 9-2 for each type of space, taking the type of mounting into account.

A suitable covering for any side wall treatment is perforated sheet metal with at least 20% open area. Perforation patterns, such as $^{1}/_{16}$-in. diameter holes staggered at $^{7}/_{64}$-in. center, $^{1}/_{8}$-inch diameter holes at $^{3}/_{16}$-in. centers, or $^{3}/_{16}$-in. diameter holes at $^{5}/_{16}$-in. centers, provide adequate open area.

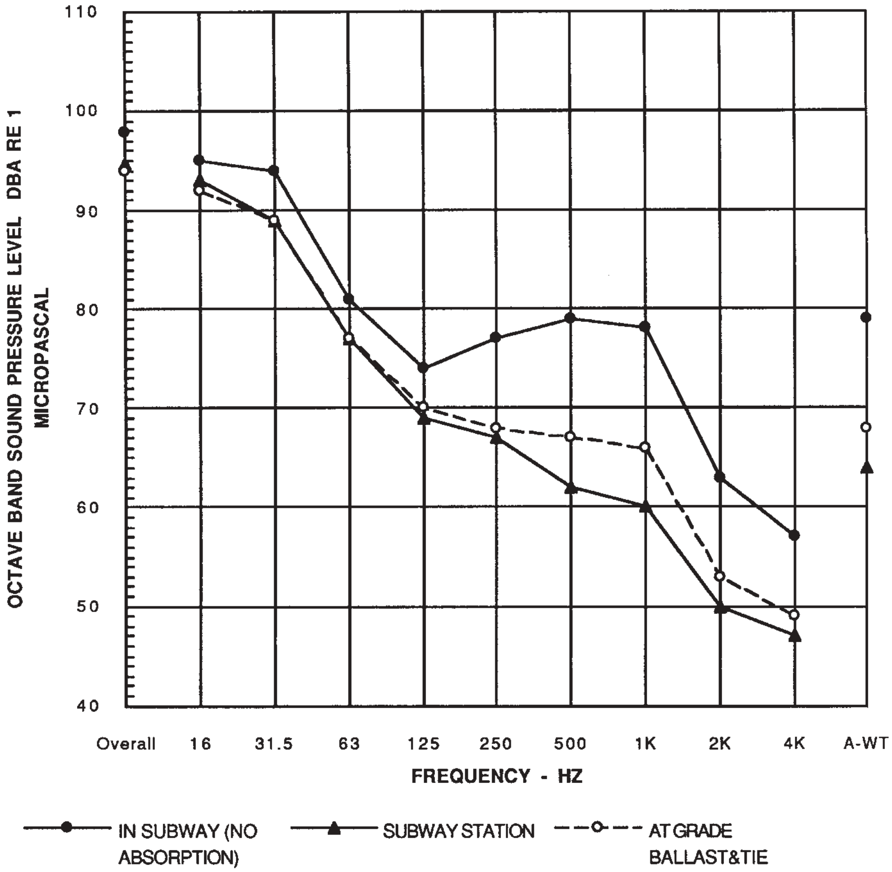

FIGURE 9-6 WMATA METRO CAR INTERIOR NOISE LEVELS, 2-CAR TRAIN AT 40 MPH

There are, of course, other combinations of equivalent performance.

9.4.10 Other Design Considerations

The acoustical design of the station should consider other sources of sound, such as highway and crowd noise, and intelligibility of the public address system. These are discussed in the *Handbook of Urban Rail Noise and Vibration Control* (*19*).

9.5 FAN AND VENT SHAFT TREATMENT

This section addresses the control of wheel/rail noise transmited from underground subways to the street above through fans and vent shafts. Noise control treatment provided for controlling ancillary equipment noise, such as fan noise, necessarily will control train noise. Where communities are located near shaft openings, noise control may be required to prevent an increase or at most allow a minor increase in ambient levels in the community.

The APTA Guidelines provide design guidelines for ancillary equipment noise levels. These guidelines are for maximum noise levels at 50 ft from the facility, at the setback line of the nearest building, or at the nearest occupied area, whichever is nearest to the source. “Transient noise” refers to intermittent noise, such as that caused by trains. The criteria cover a 25 dBA range with the appropriate level depending upon the type of neighborhood in which the facility is located; the criteria are least restrictive for industrial areas and the most restrictive for low density residential areas. Fan and vent shafts located in industrial areas generally do not require any noise control treatments. However, when fan and vent shafts are located in a low density residential area, particular care must be taken to ensure that the noise levels are not excessive.

In most situations, designing transit systems to meet the APTA Guidelines will prevent community annoyance with transit noise. However, local noise ordinances may apply at specific locations, in which case they may take precedence over the transit system design criteria.

9.5.1 Procedures for Attenuating Vent Shaft Noise

Figure 9-9 illustrates the cross-section of a typical fan/vent shaft configuration, and the paths by which noise travels to

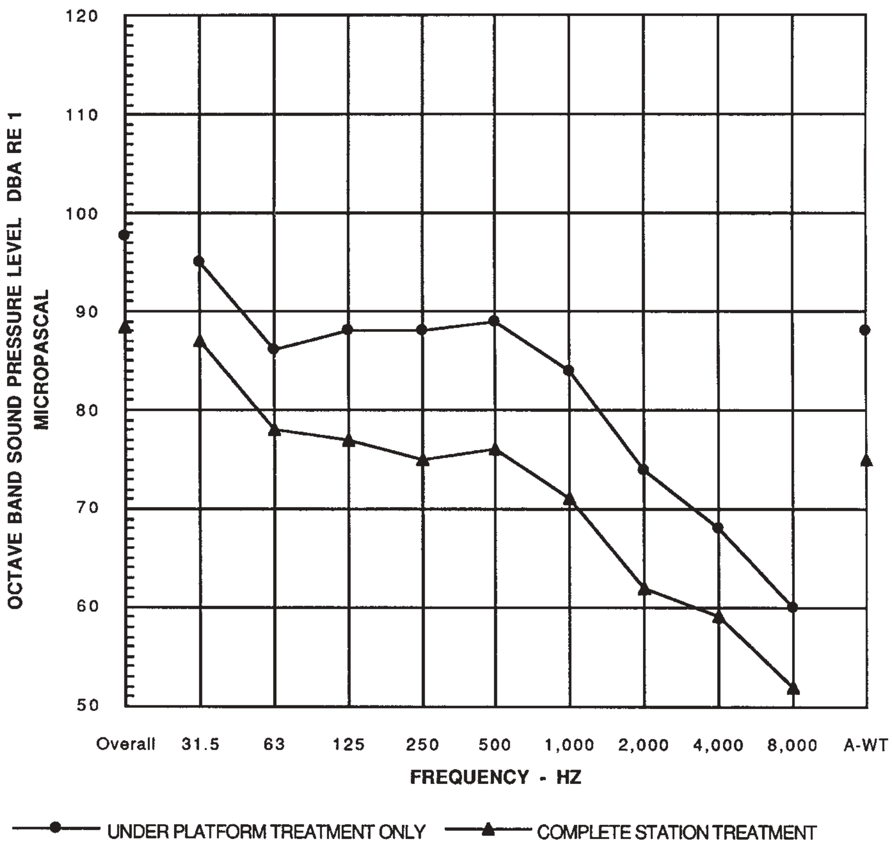

FIGURE 9-7 TYPICAL MAXIMUM PLATFORM NOISE LEVELS OF TTC TUNNEL STATIONS WITH TRAINS ENTERING AND LEAVING

patrons and adjacent communities. Noise from trains and ancillary equipment travels into fan shafts and plenums and radiates outdoors. Without acoustical treatment, the noise can be intrusive in nearby residential or other sensitive areas.

The following basic procedures are available for controlling vent shaft noise:

- Apply acoustical absorption material to surfaces of shafts, tunnel walls, and subway ceilings near vent shafts.
- Attach silencers to fans.

The noise reduction achieved by lining a fan or vent shaft depends on the placement, area to be covered, and type of material used. Noise control provisions such as an inline silencer incorporated to control fan noise will necessarily control train passby noise. Most of the discussion of ventilation shaft noise treatment for wheel/rail noise will thus focus on treatment of wall surfaces with sound absorbing materials. Information on fan shaft treatments to control fan noise may be obtained from the *Handbook of Urban Rail Noise and Vibration Control* (*20*).

Acoustical absorption in air handling systems can be very effective for controlling wheel/rail noise propagated through the duct system. Attenuation is maximized when treatment is applied to the surfaces on which the sound energy impinges directly, such as at bends in the shafts. The following discussion is largely based on the ASHRAE Guide which contains more detailed information on the control of noise in ventilating systems (*21*).

There are three types of areas on which absorption lining can be applied:

- Straight sections of duct,
- Bends in ducts, or
- Plenums.

The least effective treatment is the lining of the walls of straight sections of ducts. Because ducts in vent and fan shafts are usually large, typically 100 sq ft, large amounts of treatment are needed to achieve significant attenuation. Lining bends and plenums, however, can be very effective and economical for reducing noise. Since fan and vent shafts are large, they are most appropriately analyzed as plenums rather than as ducts.

9.5.1.1 Straight Ducts

The exact mathematics of sound propagation in lined ducts are complex. Design calculations are generally based on an empirical formula:

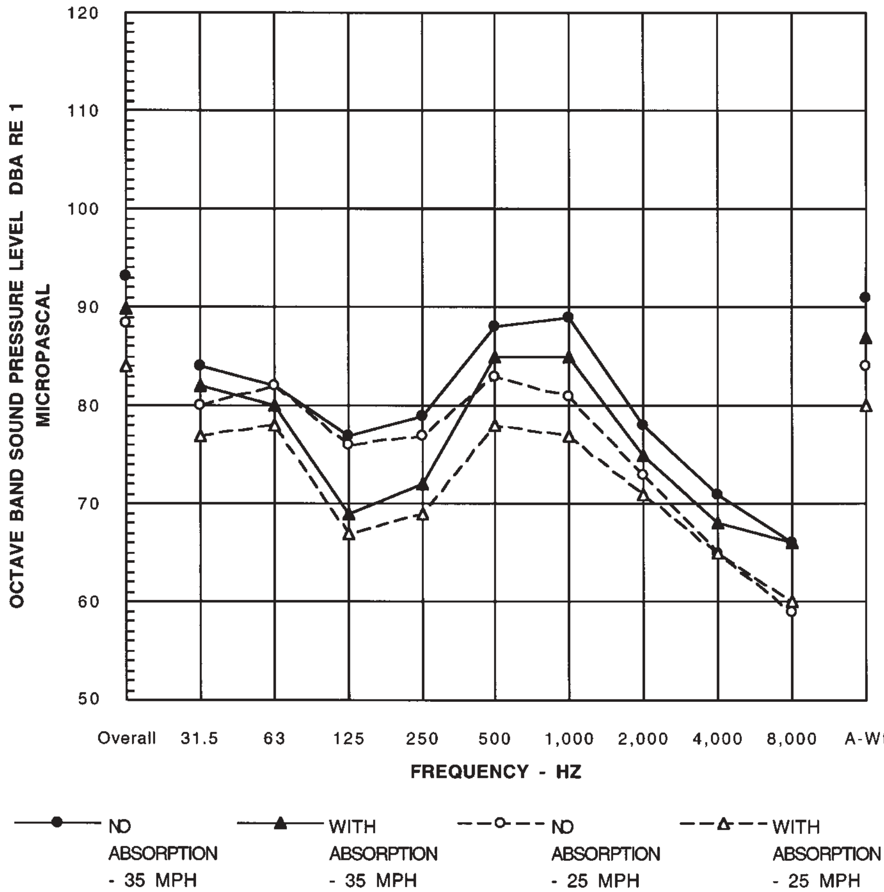

FIGURE 9-8 NOISE LEVELS AT BART SUBWAY STATIONS WITH AND WITHOUT UNDER PLATFORM SOUND ABSORPTION (CEILINGS AND WALLS TREATED IN BOTH CASES)

$$\text{Attenuation (dB)} = 1.05\ (dP/A)\ a^{1.4}$$

where

d = length of lining
P = duct perimeter
A = duct area
a = average absorption coefficient (a function of frequency)

The values for d, P, and A can be specified in any consistent unit system.

This formula does not account for line-of-site propagation, which limits attenuation at high frequencies. For a duct with a minimum cross-section dimension of about l m (3 ft), the maximum attenuation in a straight lined duct is 10 dB in the 2,000 Hz octave band. The attenuation in the 1,000 Hz octave band will be approximately midway between 10 dB and the value calculated from the equation. The frequency above which the 10 dB limit applies is inversely proportional to the shortest dimension of the duct. For a large vent shaft with a minimum dimension of about 10 ft, the 10 dB limit applies to frequencies above approximately 600 Hz. Note that for a 10-sq ft-shaft of cross-sectional area 100-sq ft, lined with 2-in. thick glass-fiber board with an acoustical absorption coefficient of 0.8 in the 500 Hz octave, the formula requires lining nearly 30 ft of shaft to obtain a 10 dB noise reduction. This length, usually not available in typical ventilation shafts, indicates why lining lengths of straight shafts is usually impractical.

9.5.1.2 Lined Bends

Acoustical treatment of ventilation shaft walls and ceilings before and after bends is very effective in reducing noise. The sketch in Figure 9-10 indicates the most effective locations for placing sound absorbing material. The definition of shaft depths, d and D, and shaft width, W, as they apply to sound attenuation treatment in a fan or vent shaft, are also indicated in the figure. Note that the sketch shows acoustical material on only the sides *normal* to the plane of the bend. Additional material on the sides parallel to the plane of the bend would contribute to the total sound attenuation; but such placement is inefficient because the added material acts only as lining in a straight duct.

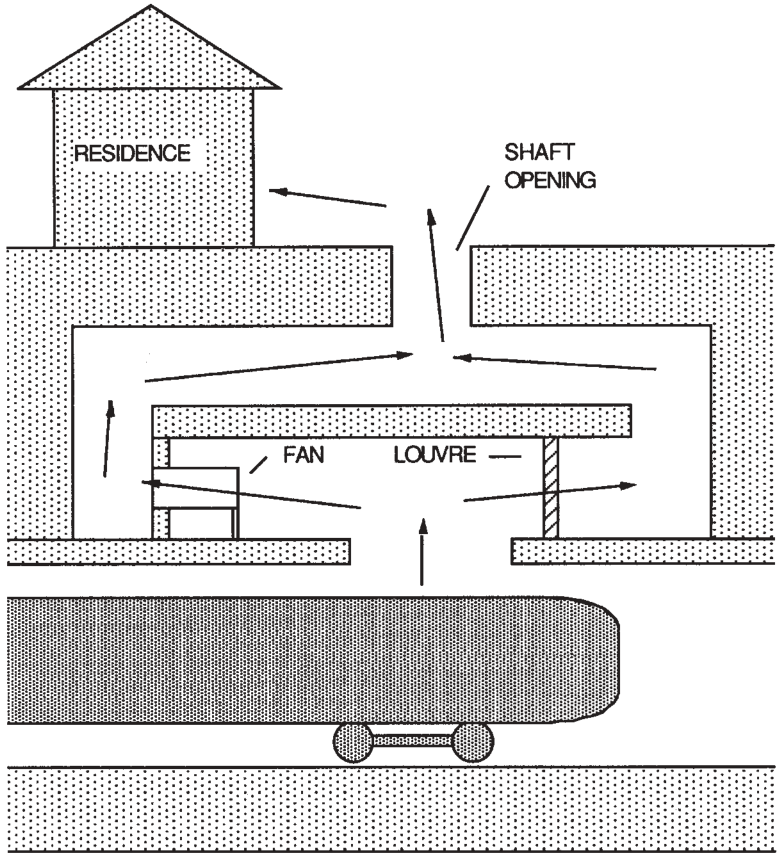

FIGURE 9-9 SKETCH OF SOURCES AND PATHS OF FAN AND VENT SHAFT WHEEL/RAIL NOISE

The maximum attenuation from acoustical absorption treatment placed before or after a right angle bend is accomplished by lining surfaces over the following distances:

- Two shaft depths before or after the bend, if a high absorption lining is used ("thick" treatment),
- Three shaft depths before or after the bend, if a low absorption treatment ("thin" treatment), such as directly applied spray-on materials, is used.

Extending the lining for additional lengths does not appreciably increase sound attenuation. The sides of the shaft parallel to the bend should be lined only if shaft width and depth are comparable. In practice, lining the wall and ceiling areas of a shaft is practical, but lining the floor is not. This restriction means that in some cases, only one side of the duct shaft can be lined, either just before or just after a bend.

Table 9-5 indicates the approximate amount of attenuation that can be achieved by applying either "thick" or "thin" sound absorbing treatment at shaft bends. The "thick" treatment would typically consist of 2-in. thick glass fiber or a spray-on material on metal lath backed by a 1-in. air space. A "thin" treatment would consist of directly applied spray-on material $^5/_8$- to $^3/_4$-in. thick without air space. Note that 1-in thick glass wool, applied directly to the concrete surface, would provide absorption and attenuation about halfway between the "thick" and "thin" treatments as described earlier.

The data in Table 9-5 illustrate the effects of lining straight portions of the shafts, as well as the effects of interaction at a bend, and are intended for use with rectangular shafts of a width greater than 2D. For shafts of width less than 2D, the sides parallel to the plane of the bend should also be lined. The results apply to round shafts also if the lining on one half the circumference is considered equal to lining on one side of a rectangular shaft, and the lining around the entire circumference equals the lining shown on two sides. In shafts where the available length of straight duct to be lined before or after a bend is less than 1.5D, attenuation will be less than that given in Table 9-5; in these shafts, other methods of achieving noise reduction may be appropriate.

The attenuations listed in Table 9-5 are equally valid for small, medium, or large shafts. This generalization is possible because shafts are usually designed so that the lining area, as defined by the table, is approximately proportional to the shaft size. For full 180 deg bends in the shaft, the attenuation

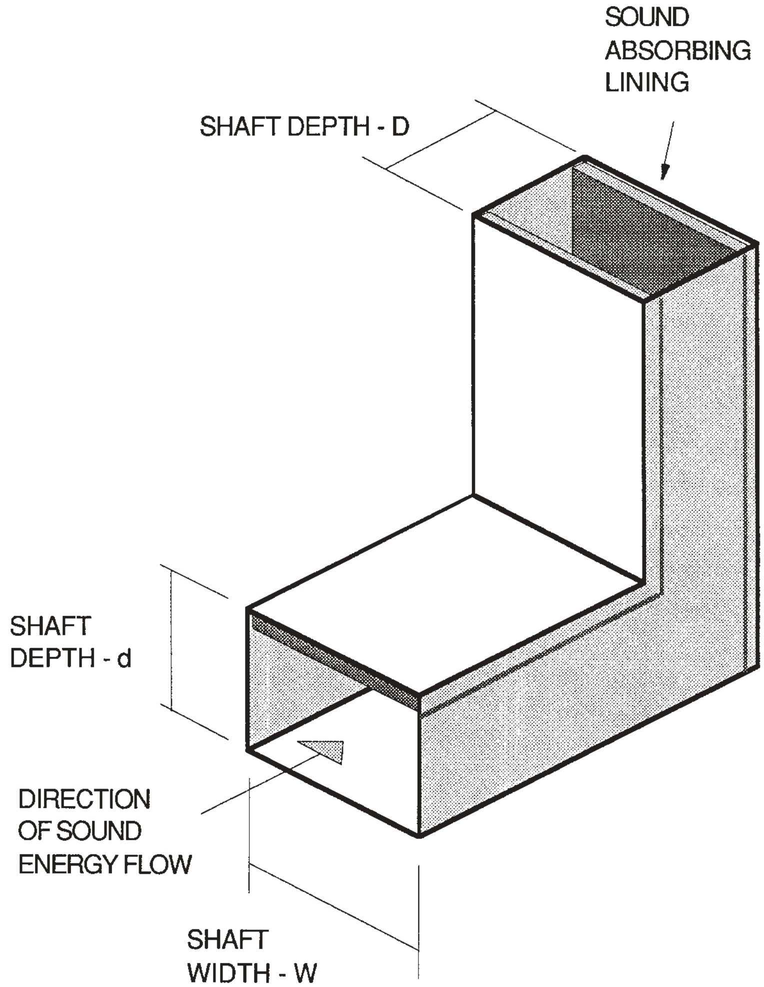

FIGURE 9-10 RIGHT-ANGLE BEND IN DUCT

TABLE 9-5 APPROXIMATE ATTENUATION OF SOUND ACHIEVED BY LINING RIGHT ANGLE BENDS IN FAN AND VENT SHAFTS

LINING LOCATION	SIDES TREATED	LINING AREA		ATTENUATION - dBA	
		THICK TREATMENT[1]	THIN TREATMENT[2]	500 HZ	1,000 HZ
AHEAD OF BEND	1	2dW	3dW	2	3
	2	(4d+D)W	(6d+D)W	6	8
AFTER BEND	1	(2D+d)W	(3D+d)W	5	6
	2	(4D+d)W	(6D+d)W	8	10
AHEAD AND AFTER BEND	1 + 1	(2D+3d)W	(3D+4d)W	7	10
	1 + 2	(4D+3d)W	(6D+4d)W	10	14
	2 + 2	(3D+5d)W	(4D+7d)W	11	16

1 "Thick Treatment" consists of 2 in. thick glass fiber or spray-on material on metal laths. Absorption coefficients at 500 Hz and 1,000 Hz of 0.8 to 0.9.

2 "Thin Treatment" consists of 5/8 to 3/4 inch thick spray-on material with absorption coefficients of at least 0.5 at 500 Hz and 0.7 at 1,000 Hz.

obtained by lining the shaft either before or after the bend is approximately 1.5 times that listed in Table 9-5.

The following example illustrates the use of Table 9-5 to estimate the reduction caused by lining a vent shaft. Figure 9-11A indicates a shaft bend, similar to many found in both station and line vent shafts. In this situation, several circumstances prevent the maximum utilization of a lined bend. It is generally impractical to line the floor of a shaft or any area open to the weather; the only remaining area available for treatment is the ceiling before the bend. The 18-ft length of thick treatment on the ceiling, indicated in Figure 9-11A, will give a reduction of only about 3 dB. However, if the shaft is rearranged, as shown in Figure 9-11B, the bend can be lined much more effectively, and the total attenuation will be about 15 dB. The attenuation achieved by lining one side before and one side after the bend is about 10 dB, and the extra attenuation of a 180 deg bend, compared to a 90 deg bend, is 1.5 times 10 dB. Clearly, such alterations will complicate the airflow, and increased flow resistance must be considered.

Vent and fan shaft designs often fail to allow sufficient area for acoustical treatment, and some of the potential attenuation that might be provided by a lined bend cannot be obtained. Although the design attenuation may not be achieved, lining a bend will be more effective than lining a straight shaft with identical treatment. Using half the recommended length of lining on a bend will result in approximately half the attenuation.

9.5.1.3 Plenums

Analyzing the section of shaft illustrated in Figure 9-12 as a large bend, with thick absorption material on the two sides before the bend, gives 8 dB attenuation at 500 and 1,000 Hz. According to the estimates of treatment area given in Table 9-5, 910 sq ft of treatment would be needed. However, the shaft section is more accurately modeled as a plenum than as a lined bend. The formula for noise reduction obtained by lining a plenum is

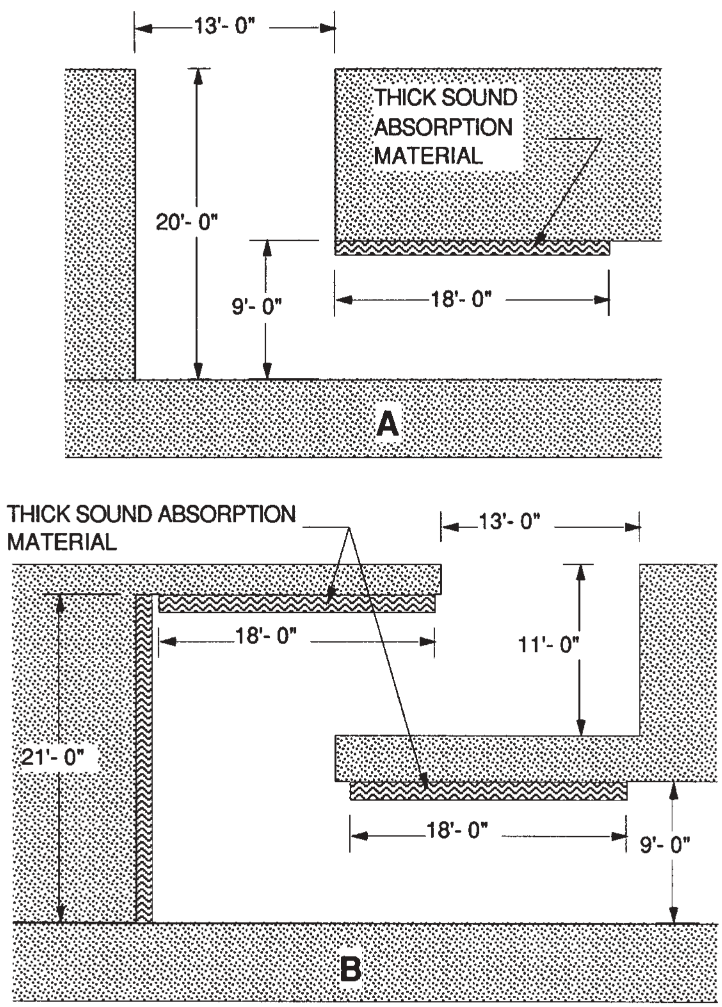

FIGURE 9-11 TYPICAL BEND IN SHAFT

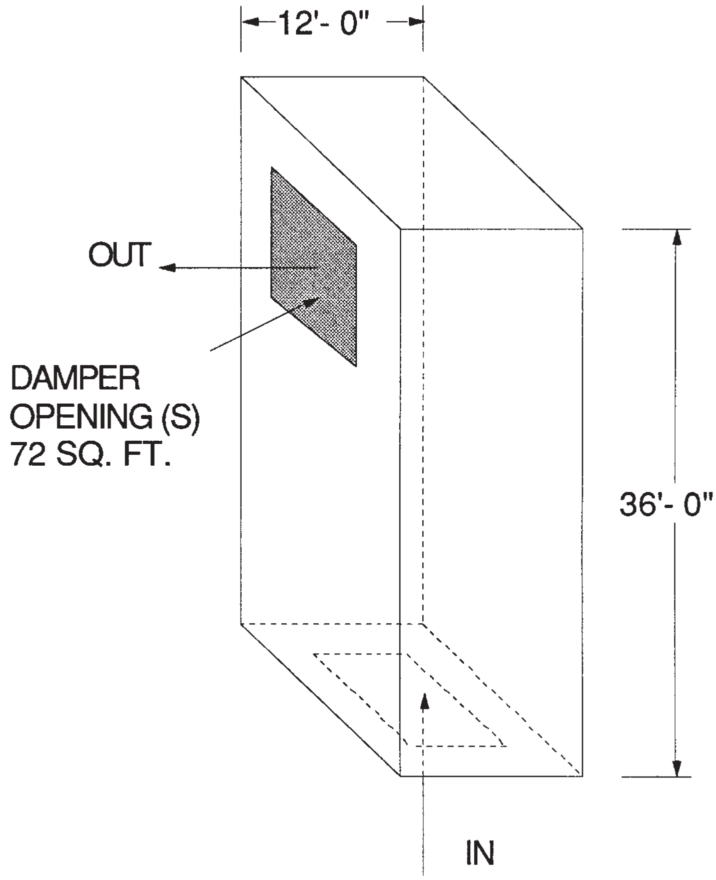

FIGURE 9-12 TYPICAL PLENUM CONFIGURATION

$$\text{Attenuation (dB)} = 10\log\left[S\left(\frac{\cos(\theta)}{2\pi q^2} + \frac{1-\alpha}{a}\right)\right]$$

where

S = area of outlet, m^2
α = average random-incidence absorption coefficient of plenum, dimensionless
a = total absorption in plenum chamber in sabins, equal to area of treatment times absorption coefficient, m^2
q = diagonal distance between inlet and outlet, m
θ = angle between inlet and outlet, degrees

In the design sketched in Figure 9-12, assuming 910 sq ft of absorptive lining with an absorption coefficient of 0.9, the reduction is 12 dB; 4 dB more than the lined bend estimate. Doubling the treatment area increases the reduction to 15 dB. This formula for attenuation by a lined plenum is accurate within a few decibels at high frequencies where the acoustic wavelength is less than the plenum dimensions. At lower frequencies, the equation is conservative; actual attenuation sometimes exceeds the value calculated by the above equation by 5 to 10 dB, due to sound reflections caused by the expansion at the plenum inlet. The effect is not unlike a tuned muffler.

Whether a section of shaft should be considered as a plenum or a duct is not obvious. Conceptually, a plenum is a large chamber, connected by small openings to two or more ducts. The problem is complicated by the large cross-section area of most fan and vent shafts. A short section, in which the connecting ducts contain significantly smaller cross-sectional areas than the shaft section under analysis, may generally be modeled as a plenum. When the cross-section area is constant through the shaft and the connecting ducts, the lined duct model or lined bend model is more appropriate. There are no simple guidelines to provide for all possible situations. Each shaft configuration must be considered carefully to determine whether the plenum model or the duct model is more appropriate.

9.5.1.4 Vent Shaft Entrances

The subway wall and ceiling at the entrance to the vent shaft is an effective place to add sound absorption. When vent shaft entrances are at the ends of stations, placing sound absorbing materials on the subway walls and ceilings near them provides multiple benefits, including reduction of noise radiated out of vent shafts by as much as 7 dB, platform noise caused by trains approaching the station, and fan noise transmitted to the stations via the tunnels. Placing special sound absorbing treatment at vent shaft entrances is most effective when the subways do not normally have such treatment. In treated subways, additional treatment in the transition section of the vent shaft is usually only mildly effective; the amount of improvement depends on specific design details.

The added attenuation achieved with a thick absorption treatment on the tunnel walls and ceilings is 5 to 7 dBA, compared to no lining at all. If a thin treatment is used, the expected noise reduction is 3 to 5 dBA at 500 and 1,000 Hz. These figures assume that the walls and ceiling of the subway are lined for 50 ft on each side of the vent shaft entrance. Train noise radiated from the vent shaft can be further reduced by lining the shaft with acoustical absorption, as is discussed in the preceding section.

9.5.1.5 Fan Attenuators

In-line duct silencers installed to reduce fan noise in communities will necessarily reduce wheel/rail noise transmitted through the fan shaft to the community. Attenuators placed to reduce station noise will have no effect on wheel/rail noise transmitted to the station platform area, because the train will be on the same side of the attenuator as the station platform. Many types of silencers are available for installation on ventilation fans. Prefabricated attenuators are available in both rectangular and cylindrical shapes, and either a conical or a round-to-rectangular transition section can be used to couple the silencer to the fan.

In general, the selection of a fan attenuator depends on the amount of attenuation required, which will be determined primarily by the fan sound power, rather than wheel/rail noise. If all required noise reduction is to be provided by the silencer, a unit is needed that provides sufficient attenuation in the 500 Hz and 1,000 Hz octaves. A number of factors should be considered when selecting an appropriate attenuator. The first is whether a rectangular or a cylindrical unit is needed, a choice that will probably be decided on the basis of length and convenience of installation. In general, rectangular units are shorter, and often less expensive, than cylindrical ones. Rectangular attenuators are typically supplied as modules in lengths of 3 ft, 5 ft, 7 ft, or 10 ft. The cross-sectional area depends on the limitations on head loss and face velocity of the airflow through the unit. For subway ventilation fans, round units that give sufficiently low head loss vary from 10 to 16 ft in length. The round units are therefore less desirable, in view of probable space limitations.

After the configuration to be used is selected, the maximum permissible head loss determines the size and type of unit. Rectangular units can provide sufficient noise reduction with satisfactorily low head loss if the appropriate cross-sectional area is selected. For example, rectangular units, ranging from 25 sq ft to 100 sq ft in total cross-sectional area can be obtained with appropriate attenuation ratings and head loss in the range of 25 to 75 Pa (0.1 in. to 0.3 in. of water) at 60,000 cfm airflow. Rectangular units are probably a better choice, since they are available in relatively short lengths and offer good noise attenuation. A 3-ft length can provide attenuation at 500 Hz in the range of 12 to 26 dB, depending on the head loss rating. At 500 Hz, a 5-ft long unit can provide attenuation in the range of 17 to 37 dB, and a 7-ft long unit can provide attenuation in the range of 23 to 46 dB. The head loss for 60 in. diameter round silencers which could be directly attached to the fans varies from 25 to 100 Pa (0.1 in. to 0.4 in. of water), at 60,000 cfm airflow. Attenuations provided by these silencers vary from 10 to 34 dB. However, the minimum available length is 10 ft, and some units are as long as 15 ft.

The noise reduction rating for a selected attenuator under the design air flow face velocity and volume must be used in design. Although not a general practice, some attenuator catalogs still present sound attenuation data under static conditions. When air is flowing in the same direction as the sound propagation, attenuation will be less than it is under static conditions by a few decibels, depending on flow velocity or volume.

9.5.1.6 Fan Rooms

When a fan room acts as an intake or discharge plenum, significant noise reduction is possible through lining the fan room with acoustical treatment. However, when axial fans are installed within the ductwork, sound-absorbing material in the fan rooms will not reduce the transmission of train noise to the street.

9.5.2 Fan and Vent Shaft Noise Prediction

Train noise radiated from fan and vent shafts can cause annoyance in the community. The noise reaching a receiver position at the surface is dependent upon the sound power emission of the train, which is speed dependent; the sound power transmitted from the subway into the shaft; the attenuation of sound energy as it travels up the shaft to the surface, and, finally, the distance and angle of the receiver relative to the shaft opening. Figure 9-13 illustrates levels of vent shaft noise calculated at the WMATA Metro system.

The sound power transmitted from a subway tunnel into a vent shaft can be approximated as

$$L_W = L_p + 10\ \mathrm{Log}(A_s) - 6\ \mathrm{dB}$$

where

L_p = sound level in the tunnel when the train is passing the vent shaft

A_s = area of shaft opening, in m^2

L_W = sound power level transmitted into the shaft, dB re 10^{-12} watts

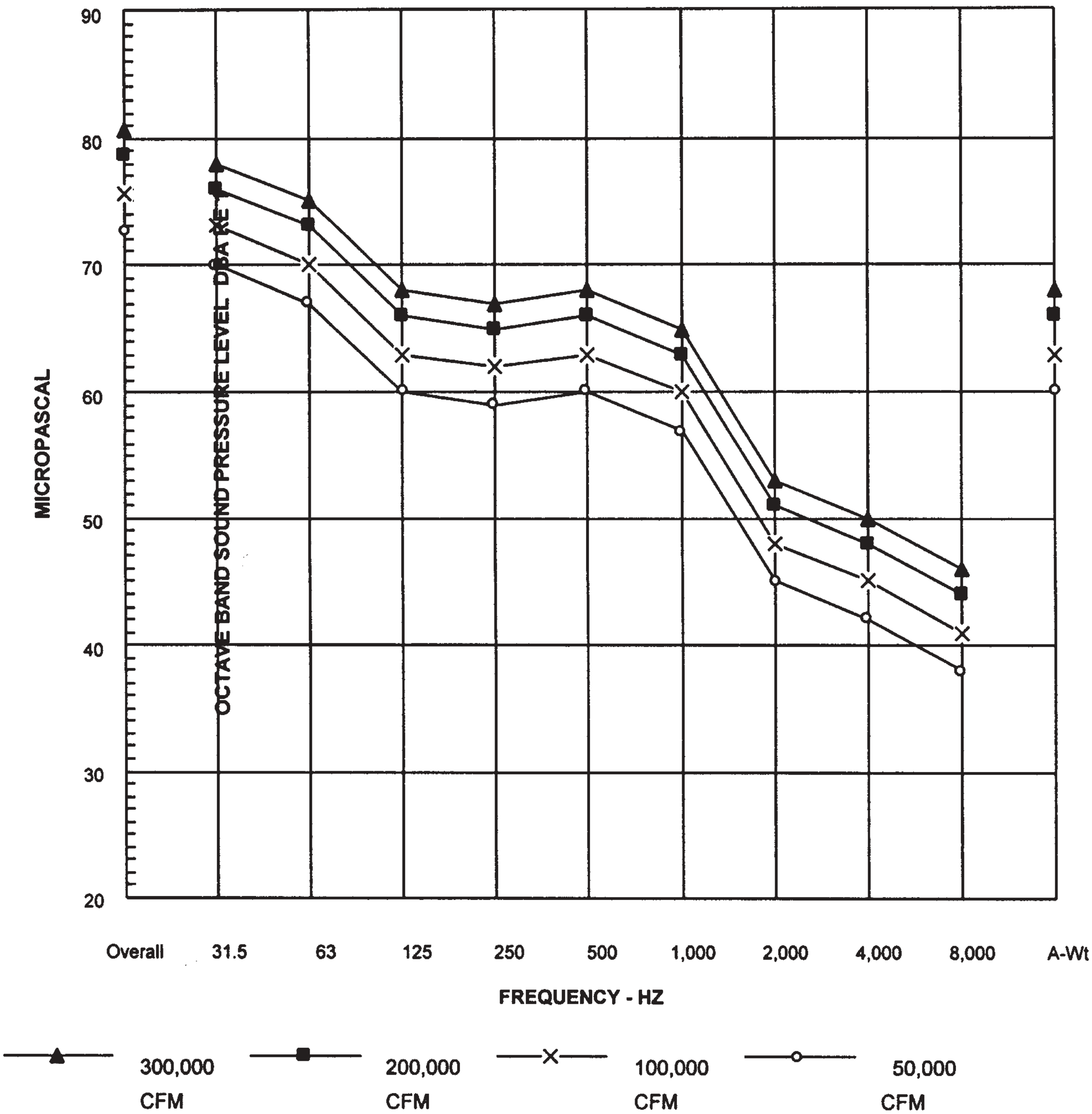

FIGURE 9-13 WMATA DESIGN CURVES FOR MAXIMUM EXPECTED TRAIN NOISE AT 30 FEET FROM SHAFT GRATINGS

The sound level reaching a receiver near the shaft opening and the required sound attenuation can then be estimated in the same manner as for fan noise. The sound level at receiver locations is given by

$$L_{pq} = L_{WR} + DI_q - 20\ \text{Log(r)} - 8\ \text{dB}$$

where

L_{pq} = Sound pressure level measured at distance r and angle q from the source, dB re 1 micro-Pascal.

Di_q = Directivity Index

r = Distance from the shaft center

L_{WR} = Radiated sound power

The radiated sound power is simply the input sound power level, L_W, reduced by the shaft attenuation. In an untreated concrete shaft, a maximum of 2 to 3 dB attenuation may exist, but most of the available sound power is radiated to the community.

Measured directivity factors are presented in Figure 9-14. The directivity index increases as the angle above the horizon increases. From these results, the noise level at a receiver can be obtained.

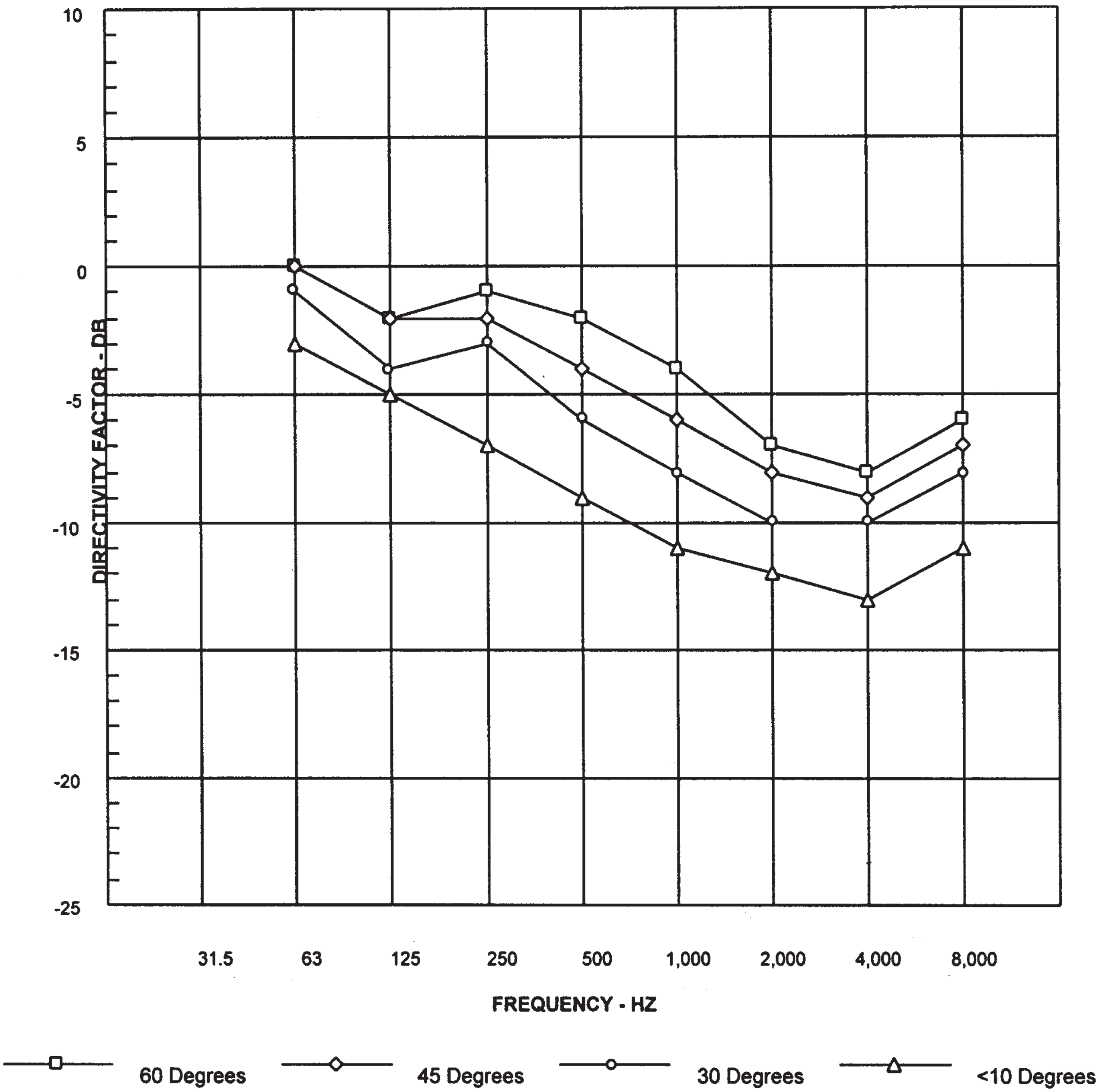

FIGURE 9-14 MEASURED DIRECTIVITY FACTORS FOR VARIOUS ANGLES ABOVE THE HORIZONTAL LINE

9.5.3 Sound Absorbing Materials

Sound absorbing materials used for treating vent and fan shaft surfaces must be durable and economical, and must provide extremely efficient sound absorption in the frequency range covered by the 500 and 1,000 Hz octave bands. The efficiency of sound absorbing materials is measured in terms of the Noise Reduction Coefficient (NRC), which is the arithmetic average of the absorption coefficient at 250, 500, 1,000, and 2,000 Hz. However, because wheel/rail noise is most significant at 500 and 1,000 Hz., the absorption coefficients at these frequencies should be used. Most of the concern is over rolling noise. If wheel squeal is of concern, then absorption coefficients at 2,000 and 4,000 Hz should also be considered, though materials exhibiting good absorption properties at 500 and 1,000 Hz will normally be very effective at higher frequencies. Materials of similar NRC's may perform differently from one another at specific frequencies.

Three kinds of sound absorbing materials are available for treating fan and vent shaft walls and ceilings:

- Spray-on materials cementitious materials, such as those appropriate for use on subway walls.

- Conventional glass-fiber boards or blankets, mechanically attached to the fan and vent shaft interior surfaces.
- Cellular glass blocks, mechanically attached or adhered to walls.

9.5.3.1 Spray-On Materials

Spray-on materials are the easiest to install, and may be cheaper than glass-fiber materials. The number of satisfactory spray-on products is much more limited than glass-fiber blanket or board materials. Many spray-on materials either provide very little sound absorption, or are not durable enough for use in subway fan and vent shaft installations. Those considered effective based on reported transit agency experience include, but are not limited to

Cafco "Sound Shield 85," supplied by the United States Mineral Products Company, Stanhope, New Jersey.
"Kilnoise Acoustic Plaster," supplied by Pfizer Minerals, Pigments & Metals Division, New York.
"Pyrok," supplied by Pyrok Company, New York.
"Pyro-Spray," supplied by Baldwin-Ehret-Hill, Inc., Trenton, New Jersey.

All of these materials have similar absorption characteristics when applied in thicknesses of $^5/_8$ to $^3/_4$ in. When properly installed, all are durable enough to withstand repeated cleaning or washing with water spray. The installation procedures must be clearly defined and monitored to ensure a durable application. Improper installation may result in inadequate acoustical performance and poor adhesion to surfaces.

Spray-on material applied to fan and vent shaft surfaces in thicknesses of $^3/_8$ in. to $^3/_4$ in. will provide absorption coefficients of 0.45 to 0.55 at 500 Hz and 0.70 to 0.80 at 1,000 Hz. Reported absorption coefficients for 1-in. thick treatment applied directly to a concrete surface include 0.24 and 0.42 at 500 and 1,000 Hz, respectively. Evaluation of the initial installations at WMATA subways confirmed that these figures are reasonably accurate. The measured values for 1-in. thick treatment were 0.17 and 0.39 at 500 and 1,000 Hz, respectively (*22*). Although these figures are less than those reported by the manufacturer for laboratory test results, they are still sufficient to provide significant noise reduction.

9.5.3.2 Glass-Fiber Boards and Blankets

Glass-fiber boards or blankets provide the highest sound absorption coefficient, and, therefore, the highest sound absorption for the amount of area covered. A wide range of glass-fiber blanket and board materials will satisfactorily control fan and vent shaft noise. The materials should have a density of 1.5 to 6 pcf, with or without sprayed on vinyl or neoprene protective coating. The most economical and appropriate material is glass-fiber duct liner, as used in ventilation system ducts. This material is generally available in 1-in. thickness; two layers can be used to obtain a 2-in. thickness. Materials for sound absorption treatment include Owens-Corning Fiberglass rigid or semirigid board. Johns-Manville and Certain-Teed/St. Gobain (Gustin-Bacon) supply similar material, with equivalent mechanical and acoustical characteristics.

Where mechanical protection of the material is necessary, the installation may include an outer covering of acoustically transparent hardware cloth or expanded metal. Dust or dirt collecting on the surface of the glass fiber will not significantly affect its sound absorption characteristics, although dust can be a fire or smoke hazard. Water has no permanent degrading effect on the sound absorbing ability of glass fiber, but absorption is reduced while the material is wet. Over the course of time, the detergents used in tunnel washing may leave an accumulation of residue, the effects of which are not yet known.

Glass-fiber material can be kept from collecting dirt and absorbing water by enclosing it in an envelope of Tedlar of thickness no greater than 0.003 in. Any plastic film with a thickness of 0.004 in. or less is acoustically satisfactory if its weight is less than about 0.3 oz/sq ft. The selection of plastic film must be based on the life expectancy of the tunnel and the fire resistance of the material. In many outdoor environments, mylar or polyethylene film is used. However, when the fire resistance capacity of these materials is unacceptable, a polyamide film such as DuPont Kapton or any other fire-resistant material should be considered.

A number of procedures can be used to attach glass-fiber boards and blankets to concrete surfaces. In machinery rooms and concrete ducts, "Stic Klips" (similar to large headed nails) can be used, either cast in the concrete or fastened to the concrete with cement or epoxy. The glass-fiber material is impaled on the rod, a washer is placed over the rod, and the rod is bent over the washer to retain the material and any added protective covering. Wood or metal furring strips, attached to the concrete surface, can be used with mechanical fastening to support and retain the glass fiber. An "acoustical stud" or similar shaped metal extrusion can be used to attach 2 in. thick material to concrete walls, without using fasteners that penetrate the glass fiber or protective coating. Such mountings are especially convenient when a waterproof covering is to be used.

Fire safety is a major concern when specifying acoustical treatments for subway applications. A glass fiber with no binder is necessary to achieve an incombustible product. However, very few products are manufactured without binders because glass fiber tends to lose fibers when there is no binder material. Temp-Mat, from Pittsburgh Corning, is one example of a glass-fiber product held together by a mechanical felting process that contains no binder. It does contain a small amount of residual oil, used in the manufacturing process, which can be baked out by the manufacturer. It has a density of 11.25 pcf in the 1-in. thickness, which is two or three times the density normally recommended for glass-fiber acoustical absorption. The density of Temp-Mat makes it cost more than lower density materials, but its acoustical performance will be as satisfactory as lower density 6 pcf glass fiber of the same thickness.

9.5.3.3 Cellular Glass Blocks

Geocoustic Blocks, a proprietary product of Pittsburgh Corning, have been used successfully in a number of subway applications. They are made of rigid porous glass foam in thicknesses of 2 and 4 in., and are slotted to increase the effective surface area and absorption. The manufacturer states, and tests confirm, that these blocks have absorption coefficients above 0.90 at 500 and l,000 Hz. Geocoustic Blocks have the significant advantage of being completely inorganic and incombustible. However, they have the disadvantage of shedding small glass granules. The manufacturer has experimented with a spray-on coating to control the shedding problem, but this adds an organic component to the material that might increase fire-related problems. The Geocoustic Blocks are no longer regularly supplied, though they might be obtained provided sufficient quantities are ordered.

9.6 RECEIVER TREATMENTS

Receiver treatments include improving the sound insulation characteristics of homes or other sensitive structures, usually by retrofitting glazing and doors with acoustically rated units. Receiver treatment has become popular for aircraft noise control, where interior Day Night Levels (L_{dn}) may exceed 45 dBA. Receiver treatment should be considered as the last approach for controlling wheel rail noise, since there are no established criteria for interior noise from rail transit systems, and the interior noise limit of L_{dn} 45 recommended by the Environmental Protection Agency for community noise exposure is not likely to be exceeded by wheel rail noise, provided that the system is in good condition and that receivers are not located within a few feet of the system. A second reason for avoiding receiver treatment is that during the process of retrofitting windows and doors, pest damage and/or code violations may be encountered, the correction of which may require substantially greater funds than needed just for the retrofit. In effect, the transit system may become responsible for the overall improvement of the structure. Nevertheless, the experience obtained with aircraft noise insulation projects may serve as a valuable guide. For those pathological situations where a receiver is located within a feet of the transit system, or where rail corrugation cannot be controlled adequately, and a sound barrier is impractical, treatment of windows and doors remains as a viable option, unless they are already acoustically effective.

The best approach is to replace, where appropriate, an existing window with a complete unit consisting of glass, mullion, weather stripping, and miscellaneous hardware. In newer homes built to thermal insulation code requirements in colder climates, the windows may already be sufficiently effective in controlling interior noise. Often, storm windows and doors are added to existing windows and doors, respectively, to improve the sound insulation properties of these components. The selection of windows or doors should be based on the spectrum and level of wheel/rail noise. Typically, a door or window with a Sound Transmission Class (STC) of 30 should be adequate. Higher STC ratings may be needed if sound levels are particularly high, or involve pure tones due to rail corrugation, or impact noise from special trackwork.

Treatment of residential receivers may require permanent closure of operable windows. This may, in turn, require provision of mechanical ventilation or air conditioning systems, an expense which may greatly increase the cost of mitigation. Further, if not properly selected, mechanical ventilation and air conditioning systems may produce their own noise impacts.

Contrary to popular opinion, typical or common thermal insulating glass consisting of two identical glass layers separated by a $^1/_4$- or $^1/_2$-in. air space provides poor sound insulation compared to acoustically rated glass. A better approach is to use single layer laminated glass, thermally insulating glass with dissimilar glass thicknesses, or ideally, thermally insulating glass with at least one laminated glass pane. In spite of the poor performance of common thermally insulating glass, the requirements of the mullions and framing for limiting air intrusion are acoustically beneficial, and such installations will generally be superior to simple casement or sash windows regardless of glass characteristics. Again, sealing air leaks in windows or doors is always acoustically beneficial.

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CHAPTER 10: RAIL CORRUGATION CONTROL

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RAIL CORRUGATION CONTROL

10.1 INTRODUCTION

Rail corrugation causes some of the most serious wheel/rail noise problems experienced by transit systems. The noise is irritating, and can be painful and possibly harmful to patrons in subways. At the wayside, wheel/rail noise due to corrugated rail contains a distinctive tonal character, and the noise may be heard and identified at many miles from the track, especially under certain wind and temperature conditions. While noise from corrugated rail is particularly severe and may be a source of significant community reaction, it is also relatively easily controlled by rail grinding. The only, and not insignificant, limitations are available funding for rail grinding equipment and materials and personnel, and track time for grinding.

Presented below are discussions of the corrugation and possible treatment methods, based on the literature, anecdotal information, and the experience of the authors. Substantial research has been conducted within the last 10 to 15 years which appears to be culminating in a relatively clear picture of various corrugation processes. Particularly relevant is a recent survey of corrugation formation at rail transit systems (*1*), which provides a comprehensive discussion of rail corrugation formation and track configurations which may contribute to corrugation. A rather thorough review of the literature and state-of-the-art in methods of control has been provided by Grassie and Kalousek (*2*), who indicate that concern over rail corrugation has existed since the beginning of railroading. In 1961, British Rail had compiled a list of references from the year 1904, and the level of interest and research in the area has not abated.

10.2 CLASSIFICATION OF CORRUGATION

Rail corrugation is defined as a periodic longitudinal rail head profile brought about by either wear or plastic flow of the rail head running surface. Rail corrugation consists of three basic wavelength regimes:

1. "Roaring Rail" corrugations with wavelengths of 1 to 3 in. These are usually associated with light axle load operations such as at transit systems and commuter/passenger operations. The term "short pitch" is synonymous with roaring rail. At BART, roaring rail is also described as "howling rail."
2. Intermediate Wave Corrugations with wavelengths of 6 to 18 in. These are usually associated with heavy-axle-load freight operations and are most frequently found on the low rail of curves.
3. Long-Wave Corrugations with wavelengths greater than 24 in. These are usually associated with high speed passenger operations, but can be found on other low speed track as well.

The above classifications pertain to the phenomenological characteristic which is most readily observed, namely, corrugation wavelength.

Rail corrugation has been classified by Grassie and Kalousek (*3*) into six categories, including (1) heavy haul, (2) light rail, (3) booted sleeper, (4) contact fatigue, (5) rutting, and (6) roaring rail. The various classifications are listed in Table 10-1.

10.2.1 Roaring Rail, or Short Wavelength Corrugation

Roaring rail or short-pitch corrugation often produces severe noise at moderate to high vehicle speeds, and is often responsible for community noise complaints. Not only is the noise from short-pitch corrugation higher than from rail in smooth condition, but it has a tonal character centered at about 500 to 800 Hz, and is much more easily detected and difficult to ignore than normal rolling noise.

One of the most comprehensive discussions of short wavelength corrugation is provided by C. O. Frederick, Chief Civil Engineer of British Railways Board (*4*). Several observations are made by Frederick, drawing on the literature and his own work for high speed rail operations in Britain:

1. Corrugation exhibits severe plastic deformation at the crests, and only mild plastic deformation in the troughs.
2. Corrugation is essentially stationary in position. The peaks and troughs do not translate as the wear progresses. The entire rail loses material, and the rate of loss appears to be greater at the trough than at the crest, which process produces the corrugation. For very small wave amplitudes, the rate of corrugation deepening is proportional to the existing depth (leading to an exponential rate of growth with time).

TABLE 10-1 CLASSIFICATION OF RAIL CORRUGATION (3)

	Classification	Wavelength - mm -	Cause	Effect
1	Heavy Haul	200-300	P2 Resonance	Plastic flow In Troughs
2	Light Rail	500-1500	P2 Resonance	Plastic Bending
3	Booted Sleeper (Stedef, LVT)	45 - 60 (RATP)	Sleeper Resonance	Wear of Troughs from lateral oscillation
		51-57 (Baltimore)	Flexural Resonance of wheel set	Oscillation, plastic flow of peaks
4	Contact Fatigue	150-450	P2 Resonance lateral	Rolling contact fatigue
5	Rutting	50 (light rail)	Torsional resonance of wheel set	Wear of troughs from longitudinal slip oscillation
		200 (RATP)	Peak vertical dynamic force	
		150-450 (FAST)	P2 Resonance	
6	Roaring Rails	25-80	Unknown	Wear of Troughs from longitudinal slip

3. Corrugation waves at the two rails do not appear to be correlated with distance down the track, and do not form an absolutely periodic pattern. (Correlation of corrugation between rails has been observed elsewhere.)
4. Corrugation appears first at the running edge of concrete ties, and not at all on paved track with continuous rail support, indicating that the corrugation is somehow related to the geometry of the rail support (the support stiffness is also different for paved continuous support versus concrete ballast and tie track).
5. Open hearth steel is less prone to corrugate than Acid Bessemer steels. Steels with low resistance to plastic yield form corrugations very slowly, and steels with high resistance to plastic yield form corrugations very rapidly, though an increase in wear resistance helps to slow the rate of corrugation formation.
6. Grinding rails smooth immediately after installation in the track substantially delays corrugation formation. Corrugation reforms quickly after grinding rail that has already been corrugated, and this may be due to residual corrugation. (Residual periodic work hardening would be another factor.)

Moreover, corrugation wavelengths are observed to be largely independent of train speed, indicating that a wavelength fixing mechanism rather than a mechanical resonance frequency fixing mechanism is at work. A geometric fixing mechanism for wavelengths supports the theory of rail corrugation caused by stick-slip or roll-slip. Wear appears as the principal corrugation mechanism, rather than plastic deformation.

Frederick's observations of short-pitch corrugation phenomena and sensitivity to treatment appear to be supported by others. Following below are discussions of the various parameters that may affect short-pitch corrugation.

10.2.1.1 Relation to Rail Support Stiffness

Short-pitch corrugation is often present at track with relatively stiff rail supports and where rail grinding is not performed regularly. Examples include corrugation at the TTC Scarborough Line with Pandrol plates and Landis elastomer pads. A recent survey of transit system corrugation lists elastomeric direct fixation track as one of several types of track which exhibit a corrugation rate of 40 to 50% of track (*5*). Although not identified, support stiffness may be the principal parameter responsible for corrugation at direct fixation track.

Older designs of resilient direct fixation fasteners at U.S. transit systems tend to have relatively high stiffness. New transit system track, such as the new BART extensions, WMATA Metro, and Los Angeles Metro, are employing fasteners of generally lower stiffness. In fact, the tendency in direct fixation track design has been toward using progressively softer direct fixation fasteners with each new transit construction. The influence of soft track support stiffnesses has not yet been reflected in surveys of corrugation conducted to date. Daniels indicates a need to evaluate the influence of track stiffness on rail corrugation formation (*6*).

Daniels indicates that the rail support modulus did not have a conclusive effect on ballast-and-tie curved track at FAST during heavy railroad tests (*7*). Where substantial rail corrugation occurs on direct fixation track, relatively stiff track fasteners are involved, though insufficient data are available to draw definite conclusions. Stiff track support is conjectured by some authors to exacerbate the "pinned-pinned" resonance related to the discrete track supports, and reduction of support stiffness might be beneficial in reducing this possible interaction (*8*).

Reduction of tie-saver pad stiffness to remove coincidence between the rail/tie vertical anti-resonance and wheelset 2nd order bending and torsional vibration modes appears to have been effective at the RATP in reducing or eliminating booted sleeper rail corrugation at 380-m radius curves. This is perhaps the most dramatic demonstration of the rail corrugation control effectiveness by reducing rail support stiffness. A study of two-block tie corrugation at the Baltimore Metro system suggests that reduction of the tie-saver pad stiffness by 50% would be sufficient to control corrugation at curved track (*9*). However, work by Grassie et al. indicates that reduction of tie pad stiffness at British Rail may not be beneficial, as discussed above (*10*).

Concrete tie track exhibits corrugation in the range of 6 to 12 in., compared to wood tie and ballast track, where the predominant corrugation wavelength is 12 to 24 in. Computer modeling indicates that contact patch forces on concrete tie track can be reduced dramatically in the range of 200 Hz if relatively soft (1,000,000 lb/in.) pads are used in lieu of stiff (8,000,000 lb/in.) pads (*11*).

10.2.1.2 Wheel/Rail Dynamic Forces

Corrugations (as well as the other rail surface defects) will generate dynamic wheel/rail interaction forces that are directly dependent on the magnitude of the defect and the speed of the vehicle. The periodic nature of rail corrugation produces an increasing dynamic wheel/rail force of the nature shown in Figure 10-1. As can be seen in this figure, these forces build up to maximum value very quickly, typically within 50 milliseconds (which corresponds to less than 3 ft for a vehicle travelling at 40 mph).

Analysis of these dynamic forces in the vertical plane results in the relationship presented in Figure 10-2 which presents wheel/rail contact force as a function of corrugation depth and frequency (which in turn is a function of vehicle speed and corrugation wavelength). The transit system's combination of speed and wavelengths (1 to 3 in.) generate corrugation frequencies of 150 Hz and greater (see Figure 10-3). While this does not appear to generate significant dynamic activity for corrugation depths of 0.005 in., deeper corrugations do generate increased dynamic load levels at these high frequencies.

10.2.1.3 Stick-Slip Wear Mechanism

Grassie and Kalousek indicate that roaring rail corrugation is occasioned by high contact forces at the peaks of the corrugation and low contact forces resulting in slip at the troughs. Stick-slip wear as a corrugating mechanism is also supported by Grassie et al. in a mathematical and experimental study of short-pitch corrugation (*12*). Creep is predicted at corrugation valleys due to an out-of-phase relationship between wheel/rail contact force and rail profile over a

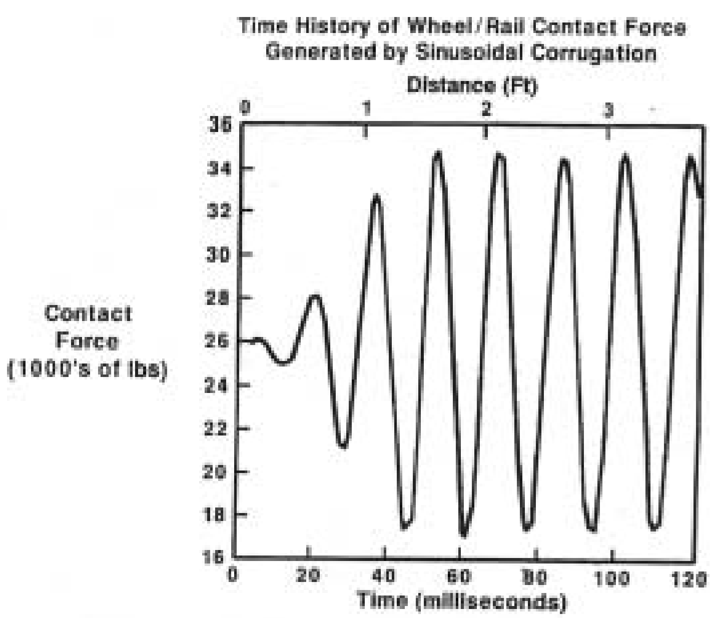

FIGURE 10-1 WHEEL/RAIL CONTACT FORCE VERSUS TIME

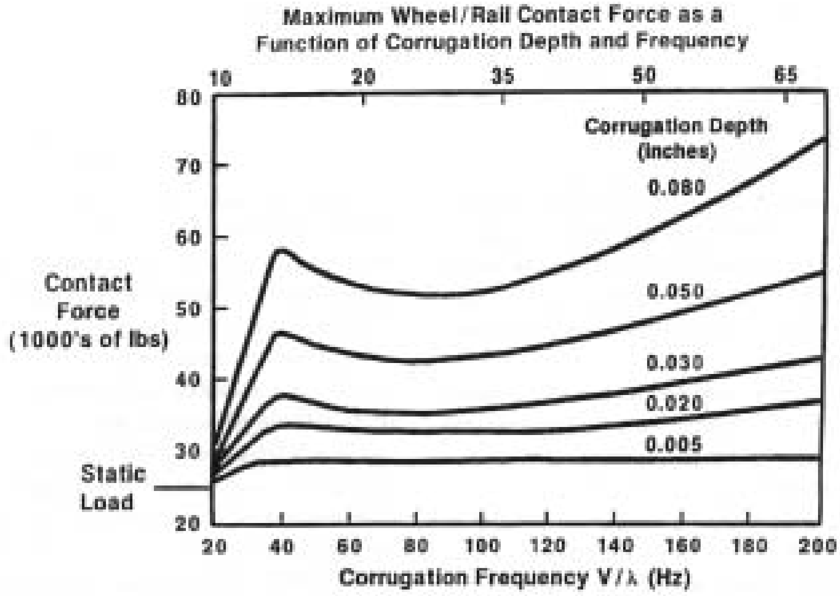

FIGURE 10-2 WHEEL/RAIL FORCE VERSUS CORRUGATION AMPLITUDE AND FREQUENCY

broad frequency range above roughly 500 Hz, not necessarily involving a resonance condition. The contact force is at a minimum at corrugation troughs, allowing longitudinal (and lateral) slip to occur, and thus wear. The results are consistent with wear patterns observed at short-pitch corrugated tracks. These authors further indicate that rail pad resonance does not appear to contribute to short-pitch corrugation, and that reduction of rail pad stiffness will not significantly reduce corrugation rates. This conclusion differs from that of Daniels, who has suggested reducing pad stiffness at Baltimore MTA to reduce corrugation in connection with the two-block booted ties (*13*).

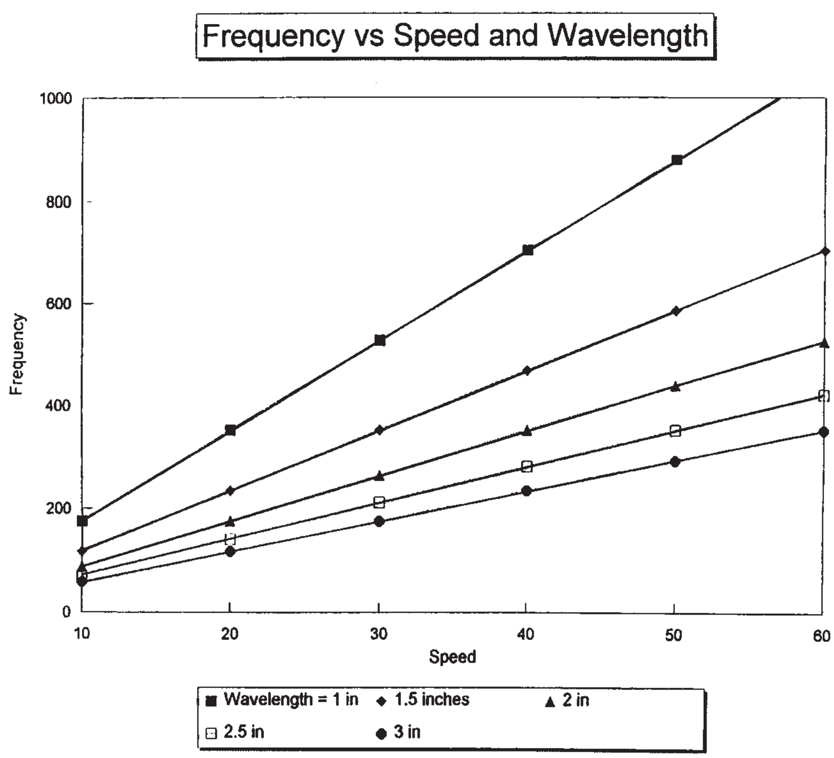

FIGURE 10-3 EFFECT OF TRAIN SPEED AND CORRUGATION WAVELENGTH ON CORRUGATION FREQUENCY

"Roll-slip" behavior (*14*) has been identified by Kalousek and Johnson at the Vancouver Skytrain (*15*), and they have further identified a possible relationship between spin-creep and rail corrugation, brought on by wide contact band, high tread conicity, and high wheel/rail conformity due to tread wear. High wheel/rail conformity evidently leads to high-frequency local oscillations at about 800 Hz involving rotation and translation of the rail, producing short-wavelength corrugations. These are "aggravated" by a small amount of lateral slip (*16*). However, at BART, substantial short-pitch corrugation occurs on tangent track with cylindrical wheels, suggesting that wheel tread conicity is not necessary for generation of corrugation.

In any rolling motion, creep must occur due to the finite size of the contact patch. The creep may be occasioned by roll-slip behavior, due to decreasing friction with increasing creep. Roll-slip behavior may be determined primarily by geometrical characteristics such as contact patch dimensions, because short-pitch corrugation often has a wavelength that is relatively independent of vehicle speed, as noted by Frederick. If the corrugation wavelength was directly proportional to train speed, then mechanical resonances would be expected to be the sole cause of the corrugation. Of course, constant wavelength as a function of speed is not necessarily the rule, either.

Kalousek indicates that not all wheels nor all sections of track are subject to stick-slip oscillation and corrugation. Further, "saturated" creepage is required to generate stick-slip oscillations, and only each twentieth to fiftieth wheelset would be sufficiently misaligned or have other geometrical inconsistency leading to corrugation. Further, Kalousek indicates that asymmetrically worn wheelsets, or misaligned wheelsets due to inadequate maintenance of the track or vehicle, are prone to excite stick-slip oscillations: "Wheelsets with new wheels or turned wheels, installed in well aligned trucks and negotiating tangent track cannot and will not produce stick-slip oscillation" (*17*).

10.2.1.4 Rail Steels

Grassie and Kalousek indicate that short wavelength corrugation appears to be more extensive with acid Bessemer than with open hearth rail steels, and appears to be moderate with oxygen rail steels. (This may be a reiteration of Frederick's comments.) Interestingly, rail hardness does not appear to affect short-pitch corrugation rates, at least at tangent track. However, Daniels reports that corrugation rates at curved track with hardened or alloy steel rail are roughly half those of carbon steel rail (*18*).

10.2.1.5 Rail Support Spacing

The corrugation amplitude appears to be modulated at a spatial wavelength corresponding to tie spacing, with the more severe corrugation on the approach to the ties than between in the case of resiliently supported ties (*19*). This effect may be related to so-called "pinned-pinned" resonances of the rail at frequencies of about 350 and 800 Hz. The discrete track supports tend to inhibit motion of the rail, thus creating the pinned-pinned resonance. The associated wavelength of rail vibration is twice the support spacing, considering the rail to be simply supported at the fasteners, hence the term "pinned-pinned." Theoretical studies support this mode of vibration in modulating the amplitude of corrugation (*20*).

The pinned-pinned resonance frequency for vertical motion of the rail is between 500 and 750 Hz for fastener spacings on the order of 30 to 36 in., based on a model of beam bending and accounting for rotary inertia and transverse shear. This is *precisely* the range of corrugation frequencies observed at many transit systems. By reducing the fastener spacing to 24 in., the pinned-pinned mode frequency would be increased to the order of 1,000 Hz, sufficiently high to produce a wavelength of about twice the contact patch length, or about 1¼ in. for 70 mph trains. Corrugation waves less than twice the contact patch length might reasonably be expected to be smoothed by the contact patch, though this has not been demonstrated. However, corrugation wavelengths of less than 1 in. or 1¼ in. do not appear to exist. Note that this type of corrugation generation is at odds with the notion of geometrical causes producing stick-slip, so that considerable study is required before excessive support spacing can be blamed for rail corrugation.

Models predict reduced corrugation rates for wear-induced corrugation at tangent track, e.g., "roaring rail," with reduced rail support modulus and tie spacing. Again, the fixative agents appear to be the rail/tie and the "pinned-pinned" anti-resonance frequencies. Reduction of tie pad stiffness from 770 MN/m to 280 MN/m lowered a 400 Hz anti-resonance of the tie and rail and effectively removed predicted rail corrugation with a wavelength of 9 cm. However, the pinned-pinned mode remains, producing a predicted corrugation at about 3 cm wavelength. Reducing the tie spacing from 600 mm to 400 mm and reducing the pad stiffness to compensate for closer tie spacing raises the pinned-pinned mode anti-resonance from about 1,200 Hz to 2,000 Hz, at which frequency the predicted corrugation wavelength is reduced to 1.7 cm, sufficient, possibly, to be suppressed by the contact patch size (or filter.) No corroborating field data are available (*21*).

10.2.2 Rutting Corrugation

Rutting corrugation, the 5th form of corrugation identified by Grassie, is produced by windup and torsional resonance of the axle. Grassie and Kalousek indicate that rutting corrugation may occur at curves, braking sites and termini, and where both wheel sets are constrained by coupling to the same truck motor, as with mono-motor trucks. Cures for rutting corrugation include lubrication of the high rail gauge corner, use of hard rails, possibly use of a friction modifier, and avoidance of mono-motor trucks (*22*).

A particularly interesting study of torsional resonance induced corrugation indicates that two corrugation wavelengths may be generated, one exactly twice the other (*23*). This phenomenon was observed in the FAST test data collected at the Pueblo, Colorado, Transportation Test Center (*24*). The mechanism evidently involves cyclic slip in both the forward and reverse longitudinal directions, and applies mainly to curves.

10.2.3 Booted Sleeper

This type of corrugation is related to a resonance condition between the rail and resiliently supported concrete ties, controlled by the high stiffness of the resilient "tie-saver" pad. A coincidence between the rail/tie resonance, axle bending mode, and wheel torsional resonance is a possible cause, leading to corrugation frequencies on the order of 250 to 400 Hz, well within the range of significance to wheel/rail noise (*25*). A similar relationship between rail and concrete tie was investigated at the Baltimore MTA system (*26*). This type of corrugation is evidently unique to this two-block tie system. Reduction of pad stiffness (by a substantial amount) evidently eliminates this form of corrugation at the RATP (*27*), and a similar modification was proposed for the Baltimore MTA by Daniels.

Booted sleeper corrugation, if significant, is particularly important, given the popularity of the two-block tie system as a vibration isolation provision. Installations include MARTA, Baltimore MTA, Tri-Met, the Channel Tunnel, Paris Metro, and elsewhere. Although the two-block tie system provides vibration isolation, corrugation would negate any isolation benefit. The two-block tie concept is not necessarily conducive to corrugation, but certain parameters, such as rail pad stiffness, may be responsible.

10.2.4 Resilient Wheels and Rail Corrugation

A seventh category of rail corrugation may exist. Corrugation of tangent track has been experienced at many light rail systems using resilient wheels. Examples include the Portland Tri-Met, PA Transit in Pittsburgh, Sacramento RTD Metro, the Los Angeles Blue Line ballast-and-tie and direct fixation aerial structure track, Santa Clara Transit Agency, and others. The Greater Cleveland RTA has reported ripple corrugation at station stops for light rail vehicles with resilient wheels, but no corresponding corrugation with heavy rail vehicles with solid wheels (survey questionnaire return). The nature of the corrugation is similar to that of tangent track short-pitch corrugation, and may in fact have identical causes unrelated to wheel resilience.

A periodic pattern has been observed at the rail head at Tri-Met within a few weeks after rail grinding, the pattern being revealed by a variation in moisture at the top-of-rail after passage of a transit train on a damp day (*28*). The cause of the pattern is not known. The Tri-Met vehicles use Bochum 54 resilient wheels. The periodic pattern is indicative of periodic motion of the wheel tread, which might result in corrugation over a sufficient period of time.

Corrugation at light rail systems using resilient wheels shows up on both embedded and ballast-and-tie track. Anecdotal evidence suggests that particularly sensitive types of embedded track include urethane embedded track (Portland Tri-Met, Calgary), suggesting that a very high rail support modulus may be a contributing factor, possibly in combination with soft steel girder rail. Some corrugation is also observed at the concrete embedded track in Santa Clara, though reports are mixed. No corrugation appears at the resilient embedded track employed at the Los Angeles Blue Line. Infrequent rail grinding at many light rail systems may be a principal factor in corrugation growth rates at these systems. Further investigation of this phenomenon is necessary to determine the actual influence, if any, of resilient wheels on rail corrugation.

10.3 TREATMENT OF RAIL CORRUGATION

A number of treatments have been proposed for controlling rail corrugation, depending on the type of corrugating mechanism involved.

Short wavelength corrugation was controlled at the Vancouver Skytrain by the following measures (*29*):

1. Reduction of truck axle misalignment
2. Profile grinding to reduce contact patch width and move the wear strip laterally over different sections of track to avoid rutting of the wheel tread. Reduction of contact patch width is believed to reduce spin-slip motion of the tire. Variation of contact position at the rail head is employed to spread the wear of the tire, further inhibiting formation of tire surface concavity which would increase wheel/rail conformity and thus increase spin-slip.
3. Use of a friction modifier to make the dynamic friction coefficient greater than the static friction coefficient.

The Vancouver system uses a steerable truck and small diameter wheels. However, these procedures should be applicable to conventional heavy and light rail systems. One of the main points of the Vancouver experience is that a combination of treatments or procedures may be necessary to control rail corrugation.

10.3.1 Rail Grinding

Rail grinding is *the most effective control procedure* for rail corrugation and wheel/rail noise in general. Other methods are unpredictable at best, with no guarantee that their implementation will reduce or prevent corrugation. Most modern rail transit systems are actively engaged in rail grinding programs. The RATP regularly grinds corrugated sections of track to control ground vibration (*30*), which should result in lower levels of wayside and vehicle interior noise.

The Toronto Transit Commission and Chicago CTA regularly grind rail with block grinders to control wheel rail noise and rail corrugation. Examples include the TTC Scarborough SRT, main subway, and light rail systems (survey questionnaire response). Chicago CTA regularly grinds track during revenue hours. BART is engaged in regular rail grinding with a rotary vertical axis grinding machine manufactured by Pandrol-Jackson. WMATA grinds tangent and curved track with a LORAM 24-stone grinder annually or whenever corrugations reach a depth of 0.010 in. Recently, the Los Angeles County Metropolitan Transit Authority has purchased and begun using a Fairmont Tamper vertical axis profile grinder with great success in reducing wayside noise. Rail grinders are increasingly recognized as being essential in any noise control program.

10.3.1.1 Preventive Grinding

Initial rail grinding prior to startup is of great importance to maintain low dynamic contact forces and increase the time before corrugation appears. Rail corrugation was delayed by a period of five years by preventive rail grinding during a study by British Rail (*31*). The SNCF uses preventive grinding to remove the decarbonized surface layer of the rail head by grinding new rail immediately after laying, taking off 0.30 mm of rail head metal (*32*). The procedure reduces head checking and fatigue cracking, improves rolling conditions and weld performance, reduces energy consumption, and improves ride comfort.

10.3.1.2 Profile Grinding

Profile grinding to optimize contact patch dimensions, while providing good ride quality characteristics, is desirable to define the contact strip edge and reduce roughness due to variation in rail head ball radius.

10.3.1.3 Rail Grinding Frequency

Finish grinding on a periodic basis controls the formation of corrugation. As pointed out by Frederick, corrugation growth rates are largely exponential in nature. Waiting too long before rail grinding, perhaps until corrugations are "visible," may allow large corrugations to occur prior to the time grinding can be performed. Thus, grinding is desirable when corrugation development is still on the low end of the "exponential curve." Grinding off corrugations early maintains an intact work hardened running surface, and "the heat generated by rail grinding slows down the development of corrugations because it reduces internal stresses in the rail head" (*33*).

An optimum grinding interval can be defined. Assuming that the corrugation growth is exponential in time, and that the corrugations are entirely removed by grinding, the optimum grinding interval, or *corrugation time constant*, is the time required to grow corrugation amplitudes by 170%. Grinding at longer intervals to completely remove corrugation will result in greater rate of metal removal. Grinding at shorter intervals will result in unnecessarily high expense as well as greater metal removal. Determination of corrugation growth rate is thus of great importance.

The growth rate can be determined by monitoring at various sections of track. The growth rate should be determined by measuring visible growth rates over a defined time interval. Assuming the growth of corrugations is exponential, the amplitude, a, of corrugation varies as

$$a = a_0 \exp(t/\tau)$$

where a_0 is an initial roughness, and τ is the corrugation growth time constant, which is also the optimal grinding interval. If the amplitudes a_1 and a_2 are measured at two respective times t_1 and t_2, then the corrugation time constant, τ, is given as

$$\mathrm{t} = (t_2 - t_1)/\ln(a_2/a_1)$$

Subsequent grinding at the optimal time interval may, conceivably, prevent formation of visible corrugation, and waiting for visable corrugation to appear before before grinding may be counterproductive.

10.3.1.4 Grinding Depth

The amount of metal removed per pass using an optimum grinding interval might be very slight, much less than normally removed by conventional grinding, and would depend on the residual roughness left after grinding. Rail corrugation will return faster with a rough grind than with a smooth grind, assuming that the wheels are perfectly smooth. If the peak-to-peak amplitude a_0 is the initial roughness of the rail immediately after grinding, then the optimum metal removal would be $2.7 \times a_0$. A reasonable estimate for a_0 is between 300 micro-in. and 1,000 micro-in. for a 1/3-octave band centered at wavenumber 3.7 radians per inch, corresponding to 500 Hz at 50 mph train speed, or a wavelength of 1.7 in. The lower end is for block-ground rail, and the upper end is for MBTA revenue service rail (*34*). The lower end is probably appropriate for vertical axis rotary stone grinders, except for the grinding pattern that can be introduced into the rail head by the grinding stones, which may be less than an inch in wavelength. This information would suggest that an optimum metal removal per pass would be on the order of 1,000 micro-in., or 1 mil. Because corrugation amplitudes of 1 mil are on the verge of significance, 1 mil would likely be a reasonable amount of metal to be removed per grinding pass.

Again, the minimum grinding interval would be equal to the corrugation growth time constant. If the optimal grinding interval were 1 year, the total metal removed over the course of 30 years would be 30 mils, in addition to normal wear of the rail. This is not a great amount of metal. To contrast this with grinding on a 5-year basis, the amount of metal that would have to be removed every 5 years would be 15 mils, assuming an initial roughness of 300 micro-in. peak-to-peak, and an exponential growth time constant of 1 year. Over a 30-year period, 90 mils of metal would be removed. This is still not a great amount, but the calculation indicates the advantage of using an optimum grinding interval. Second, throughout much of the 30-year period, corrugation noise would be prevalent on a 5-year grinding interval, but non-existent on a 1-year basis. Not only is noise reduced by frequent grinding, but metal removal is minimized.

Grinding on a 5-year grinding interval may also require multiple passes to remove defects and corrugation, followed by profile grinding to re-profile the rail head. Grinding on a 1-year basis would require, perhaps, a single pass per year with a multiple stone grinder to dress the rail without re-profiling.

Data concerning the adequacy of the above approach in defining an optimum grinding interval and metal removal per grind have not been obtained, and research is needed in this area. In practice, removing greater than 1 mil of material may be necessary because of the type of grinding machine involved, in which case the grinding interval might be extended, though at the price of increased noise and metal removal.

Corrugation with amplitude as high as 0.008 in. (0.2 mm) will have already formed hardened martensitic peaks which will require several passes for removal. For rail grinding to be maximally effective, the corrugation must be over-ground by about 0.002 in. (0.05 mm) to remove periodic strength variation between peaks and troughs (*35*). In contrast, research by Daniels at FAST suggests that grinding should remove visible corrugation, but no more, to prevent rapid return of corrugation at curved track, based on rapid recurrence of rail corrugation after over-grinding by 0.005 in., a relatively small amount. No explanation has been proffered, though the results apply to freight environments (*36*). Tri-Met has experienced a return of rail corrugation a relatively short time after grinding to remove corrugation with a horizontal axis grinder which does not re-contour the rail head, and may have left unnecessary roughness in the rail head, leading to rapid return of corrugation. The information obtained suggests that once corrugation is established, the cost for corrugation removal and subsequent control in both labor, machines, and rail material removal are expected to be high until periodic internal stresses are controlled.

10.3.1.5 Profile

If the experience at Vancouver Skytrain is a guide, the grinding profile should be such that the contact patch is essentially oval or circular. That is, the longitudinal dimension of the patch should be similar to the transverse dimension, to reduce unnecessary conformity between the wheel and rail, and thus reduce the propensity for spin-slip of conical wheels. Smith and Kalousek suggest that the rolling radius difference from one side of the contact patch to the other should not exceed 0.5% of the wheel diameter (typically 0.015 in.), which limits conicity and transverse radii of the wheel tire and rail (*37*). Reduction of the transverse contact patch dimension may result in a slight increase of high frequency noise due to reduction of the contact patch filtering effect, though this slight increase in noise may be minor in comparison to controlling corrugation and "roaring rail." Further, Smith and Kalousek indicate that higher noise levels would occur with a wider contact patch, which increases the contact patch stiffness, contrary to the theory advanced by Remington (*38*).

Reducing the contact patch transverse dimension may be desirable to control roll-slip behavior of cylindrical wheels as well as conical wheels. If substantial conformity exists between the wheel and rail, due to wear induced concavity of the wheel profile, not only might ride quality suffer, but there would necessarily be creep at the outer edges of the contact patch, due to larger wheel diameter at the outer edges of the contact zone relative to the center of the contact patch zone. This type of creep is referred to as Heathcote slip (*39*). In this case, roll-slip behavior might also occur due to a negative friction versus creep curve slope. Reduction of the contact patch transverse dimension, and thus wheel/rail conformity, would reduce creep at the lateral edges of the contact patch.

10.3.1.6 Variation of Contact Patch Location

The rail at the Vancouver LRT system was ground to vary the location of the contact patch on the rail head, and thus distribute wear of the tire over a broad running zone to reduce the formation of a rut in the tire, and thus reduce conformity between the rail and tire, and, thus, spin-slip corrugation (*40*). This approach was employed with steerable trucks of small wheel diameter, and may not be representative of heavy and light rail transit systems.

10.3.2 Lubrication

Lubrication of the high rail has long been known to control corrugation at curved track. However, long wavelength corrugation at short radius curves is not necessarily a wheel rail noise problem, due to the much lower corrugation frequency.

10.3.2.1 Onboard Solid-Stick Flange Lubrication

Metro-Dade County Transit Agency employs a solid-stick lubricant manufactured by Phymet, Inc., applied with on-board spring loaded applicators to control corrugation (*41*). Flange lubricators used on 75% to 100% of the wheels appear to reduce corrugation growth rates substantially compared to wayside lubrication (*42*).

10.3.2.2 Onboard Solid-Stick Friction Modifiers

Friction modifiers reduce negative damping associated with stick-slip oscillation produced by a negative slope of the friction versus creep curve, and may be effective in reducing or eliminating short wavelength corrugation, as suggested by Kalousek and Johnson for the Vancouver Skytrain (*43*). Several systems are experimenting with dry-stick friction lubricants applied to wheel treads. BART, in particular, has experimented with the Kelsan high positive friction (HPF) friction modifier at the Hayward test track, though the test was not long enough to assess reduction of corrugation rate. Other systems that have experimented with or are using the treatment include the Portland Tri-Met, Los Angeles Blue Line and Green Line, Sacramento RTD Metro, and Metro-Rey in Monterey, Mexico (*44*). WMATA reports that it uses a Century Oil LCF and KLS Lubriquip Glidemaster solid-stick friction modifiers applied to the wheel flanges. Tests were "discontinued due to noise created by material used. Mounting rattles and material causes squeals" (*45*). The Los Angeles Blue Line is using the HPF and LCF flange lubricant.

The effectiveness of friction modifiers has been investigated at Sacramento and at the Los Angeles Blue Line. The results of these inspections and measurements are that corrugation is not prevented, nor are sound levels appreciably reduced relative to those at other systems. Corrugation is evident at the Sacramento RTD within 2 years after grinding with a horizontal axis grinder, with attendant roaring rail at short sections of track. Incipient corrugation is observable at the Los Angeles Blue Line 1½ years after grinding, though there may be confusion with a grinding pattern which may have survived over this period of time. Inspection of the aerial structure direct fixation track at the Los Angeles Blue Line reveals corrugation and attendant noise. The Vancouver Skytrain which reports control of rail corrugation by using friction modifiers is also engaged in aggressive rail grinding. The various light rail systems are developing considerable experience with dry-stick friction modifiers, and long-term performance data should be obtainable soon.

10.3.3 Hardfacing

Orgo Thermite provides materials and services for hardfacing, using a proprietary Riflex alloy which has exceptional hardness characteristics. The material is deposited as a weldment into a routed groove in the rail head. The manufacturer indicates that the treatment is effective because of the hardness of the material. A limitation of Riflex hardfacing is that it may be used only with low carbon steel rail, so that it may not be useable with RE 115 lb/yd rail. This should be checked with the manufacturer.

10.3.4 Rail Selection

Corrugation rates are dependent on the type of rail material and method of manufacture. As discussed above, corrugation rates are evidently higher with Bessemer than with Open Hearth steels (*46*). Bessemer steels typically have about 1% manganese and 0.6% carbon alloy, while the Open Hearth steel rails have about 0.6 to 0.9% manganese and 0.7 to 0.9% carbon. Daniels indicates that at FAST, the highest corrugation growth rates were obtained with standard carbon steels of Brinell hardness 270 (*47*). Alloy or heat-treated rails with Brinell hardness between 320 and 360 produced lower rates of corrugation than the standard rails. Heat-treated alloy steels with Brinell hardness in excess of 360 were most resistant to corrugation (*48*). The information suggests that open hearth steels with hardness in the range of 320 to 360 Brinell would be the best choice for corrugation resistant rails.

10.3.5 Rail Support Modulus

Grassie and Kalousek suggest using soft resilient direct fixation fasteners to reduce wheel/rail contact forces, and reduce possible interaction with the "pinned-pinned" resonance due to tie spacing (*49*). The worst condition would be rigid supports, which would enforce a complex pinned-pinned mode of vibration at typically 500 to 800 Hz for fastener spacings of 36 and 30 in., respectively. There would exist various pass- and stop-bands for vibration transmission up and down the rail. (A comparison of the input mechanical impedance of the rail head at a location between adjacent fastener supports and over a fastener is presented in Chapter 4 of this manual.) At the other extreme is a completely resilient fastener, which would eliminate resonance of the rail on the fastener as well as the pinned-pinned mode, by decoupling the rail from the trackbed. An unsupported rail would appear as a damped and compliant beam, which might not support formation of rail corrugation. As is well known in the dynamics of beams on elastic foundation, the mechanical input impedance of the beam at frequencies well above the beam-on-foundation resonance frequency is out of phase by 45 deg due to bending wave propagation (*50*); the result might be a substantial damping effect on wheel and rail vibration. Making the fastener as soft as practical would appear to be appropriate.

Whether or not the pinned-pinned mode is a factor, avoiding a very stiff rail support appears to be desirable to keep the rail on fastener resonance frequency below the range of corrugation frequencies. For fasteners with dynamic stiffness of 2,000,000 lbs/in. at 30-in. spacing, the rail-on-fastener resonance is about 450 Hz. Fasteners of dynamic stiffness on the order of 500,000 lbs/in. would have a rail-on-fastener resonance frequency of about 225 Hz, which is below the range of concern. However, rail corrugation occurs at BART with fastener stiffnesses on the order of 400,000 to 500,000 lbs/in., so that simply reducing fastener stiffness to 500,000 lbs/in. is not the answer.

Recent fasteners procured for BART have a dynamic stiffness on the order of 150,000 lbs/in., giving a rail-on-fastener resonance frequency of about 110 Hz, well below the short-pitch corrugation frequency range. Only time will reveal if using these softer fasteners will reduce corrugation. Other modern transit systems are incorporating relatively soft direct fixation fasteners with stiffness on the order of 100,000 lb/in. or less. Examples include WMATA and LACMTA, where current resilient direct fixation track fastener stiffnesses are on the order of 60,000 to 120,000 lb/in. static and 85 to 170,000 lb/in. dynamic. As experience is gained with these new fasteners, additional evidence will be obtained concerning the effect of track support stiffness on rail corrugation generation and control.

10.3.6 Trackwork Resonances

There are a number of resonances in the track and wheels which may contribute to rail corrugation. These include the pinned-pinned mode resonance of the rail at about 500 to 800 Hz, fastener top plate bending at about 500 Hz and higher frequencies, a radial anti-resonance of the wheel which may occur at about 500 to 1,000 Hz, and lateral bending of the wheel at about 500 Hz. These frequencies are all comparable with short-pitch corrugation frequencies. Without further information concerning the interaction of these vibration modes and anti-resonances, prudent engineering would include separating these resonances in the interest of reducing their possible interaction. Changing the wheel vibration characteristics is an unlikely avenue, though damping or vibration absorbers might be explored to reduce the effects of resonance and anti-resonances. Increasing the pinned-pinned mode resonance frequency of the rails by decreasing fastener spacing appears to be attractive. Increasing the top plate bending stiffness of resilient direct fixation fasteners to raise their bending resonance frequency in excess of 1,000 Hz is also attractive.

10.3.6.1 Rail Support Spacing

As discussed above, evidence is being collected which suggests that the pinned-pinned mode of rail vibration associated with the finite rail support spacing contributes to rail corrugation. Pinned-pinned modal resonance frequencies for vertical vibration are plotted as a function of fastener spacing in Figure 10-4, based on a bending wave dispersion equation with the effects of rotary inertia and transverse shear included (*51*). As illustrated in Figure 10-4, a fastener spacing of 30 or 36 in. results in pinned mode resonance frequencies on the order of 500 to 750 Hz, the range of typical corrugation frequencies observed at BART with 36 in. fastener spacing. Further, corrugation amplitudes have been correlated with fastener location, suggesting that the fastener separation does have an effect of some kind.

If the pinned-pinned mode is indeed contributing to rail corrugation, then a possible solution is to reduce the support spacing and/or increase the bending modulus of the rail to increase the pinned-pinned modal frequency sufficiently that

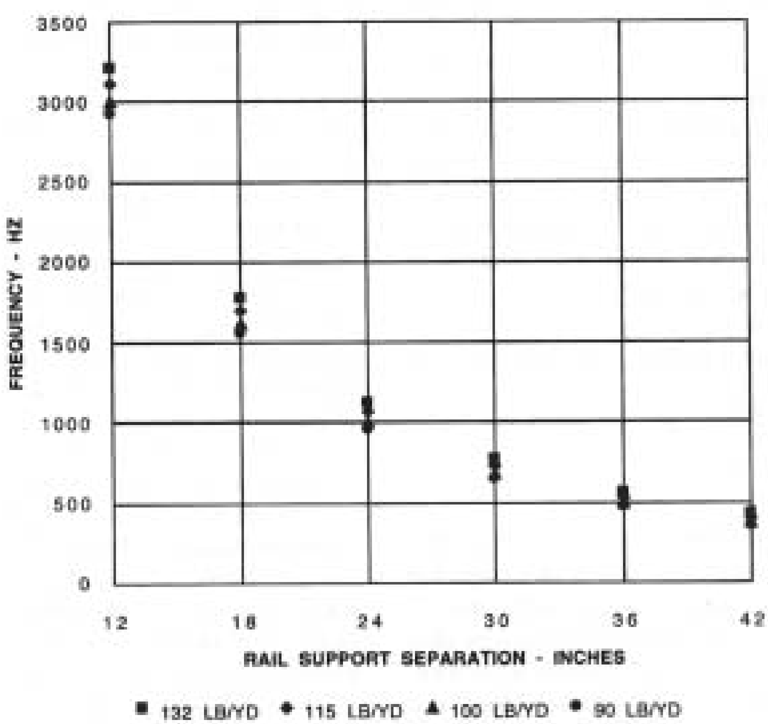

FIGURE 10-4 VERTICAL PINNED-PINNED RESONANCE FREQUENCY VS. RAIL SUPPORT SEPARATION FOR VARIOUS RAILS

the associated corrugation wavelength is less than the contact patch dimension. In this case, corrugations will tend to be "ironed out" or worn away with time. For 70 mph trains with contact patch length of ⅝ in., the design resonance 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 track and is not practical. From a theoretical perspective, the pinned-pinned mode of vibration should not contribute to corrugation if the associated wavelength is less than one half the contact patch length, corresponding to a minimum frequency of about 1,000 Hz, and a minimum fastener spacing of 24 in., though this needs to be verified. Interestingly, the tie spacing of about 24 in. employed at typical ballast-and tie railroad track satisfies this criterion.

Increasing the rail section does not appear to greatly increase the pinned-pinned modal frequency, as illustrated in Figure 10-4. The pinned-pinned resonance frequency does increase with bending modulus, however, so that heavier rails would be less prone to this possible corrugation mechanism than lighter rail.

An argument against the pinned-pinned resonance influence on rail corrugation is that short-pitch corrugation often exhibits a roughly constant wavelength as a function of train speed, as discussed above, implying a geometrical rather than a mechanical resonance as the cause of the corrugation. There are, however, insufficient data available to exclude the pinned-pinned resonance from the list of possible corrugation wavelength fixing mechanisms.

10.3.6.2 Direct Fixation Fastener Mechanical Impedance

A direct fixation fastener does not appear as a pure spring to the rail, but rather as a complex multi-degree-of-freedom

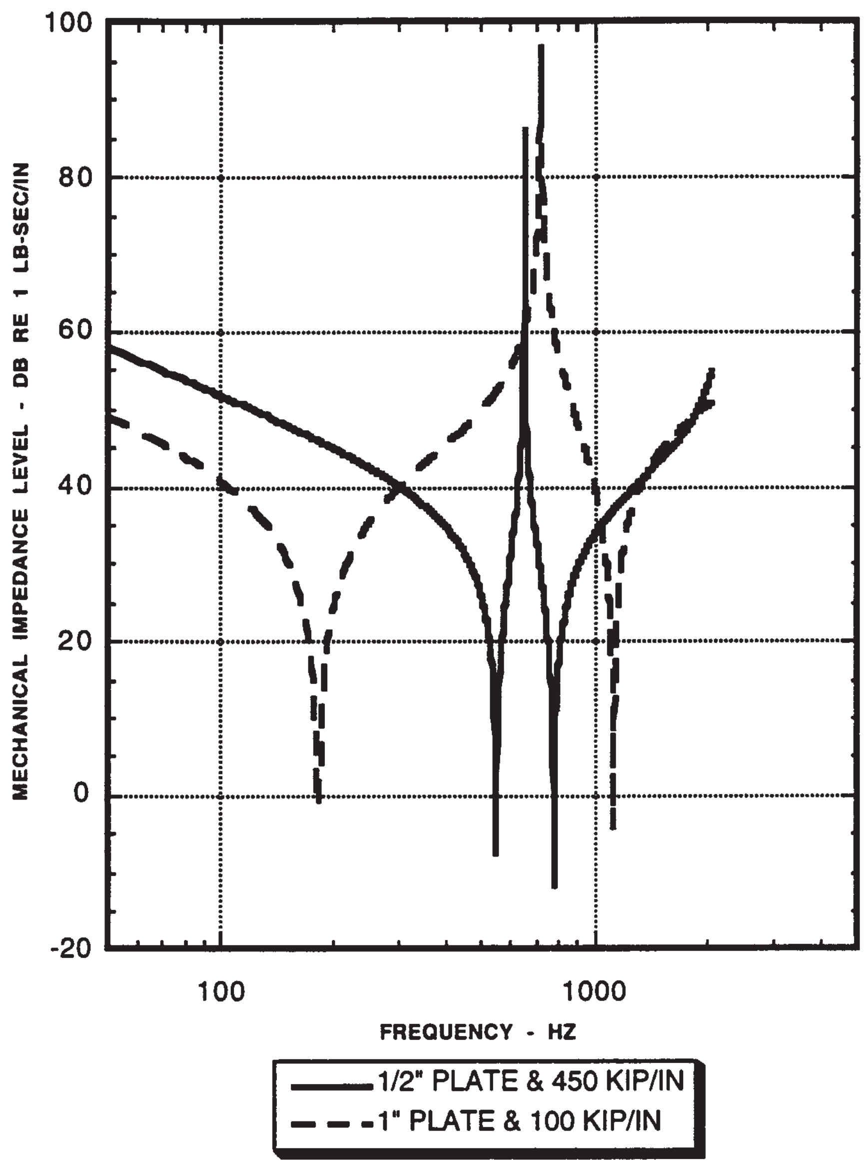

FIGURE 10-5. RAIL FASTENER INPUT MECHANICAL IMPEDANCE.

mechanical element, even when considering only vertical motion. There are two resonances that are particularly interesting. One is the top plate-on-elastomer spring resonance, and the other is the bending resonance of the top plate. The first of these can be thought of as a single-degree-of-freedom oscillator with mass equal to the top plate mass and the spring equal to the stiffness. The second is influenced by the vertical stiffness per unit area of the elastomer. Figure 10-5 illustrates the theoretical input mechanical impedances of two idealized fasteners measuring 8 in. wide by 14 in. long. One of these has a 0.5-in.-thick steel plate supported on ideal springs distributed beneath the fastener with a total dynamic stiffness 450,000 lbs/in. These are typical parameters for relatively stiff fasteners. The second has a 1-in.-thick steel plate supported by ideal springs with total stiffness of 100,000 lbs/in. The fundamental resonance frequency of the top plate vibrating vertically on the elastomer spring is identified by the first dip in the input mechanical impedance curves, which occur at about 550 Hz and 200 Hz for the first and second design configurations, respectively. The effect of bending in the fastener produces an anti-resonance above this resonance, at which frequency the rail would experience an abnormally high reaction to vertical motion. A high reaction by the fastener will tend to "pin" the rail at this frequency, exacerbating the "pinned-pinned" mode, which might contribute to rail corrugation. At higher frequencies a dip occurs in the mechanical input impedance due to a resonance condition of the plate at about 780 Hz and above 1,000 Hz for the first and second design configurations, respectively. The sharp peaks and troughs would normally be smoothed out by damping in the elastomer.

The anti-resonance and resonance frequencies of actual direct fixation fasteners under load are about 500 to 800 Hz, very close to corrugation frequencies for short pitch corrugation. No clear causative relation has been identified between these frequencies and rail corrugation, but prudent design would suggest that they be moved away from corrugation frequencies and modal resonances of the wheels. This can be achieved by thickening the top plate to raise the top plate resonance frequency in excess of 1,000 Hz. Such a fastener has been developed for the BART extensions, and performance of this fastener may be monitored to determine its effectiveness. However, as indicated by Figure 10-5, simply increasing the thickness of the top plate without reducing its mass does not raise the anti-resonance frequency above 1,000 Hz. A ribbed top plate would be most attractive for this purpose. The Cologne Egg type of fastener is of such a design that provides a high top-plate bending stiffness without too severe a mass increase, though the performance characteristics of the Cologne Egg have not been evaluated for this manual.

A most desirable design goal would be to increase the top plate resonance and anti-resonance frequencies such that the associated corrugation wavelength at design train speed would be less than the contact patch length, thus smoothing out incipient corrugation. For 70 mph trains with contact patch of about ⅝ in., the corresponding frequency would be about 2,000 Hz, a frequency which may drive the cost and thickness of the top plate. A lower frequency may be adequate. If the associated corrugation wavelength need only be less than twice the contact patch length, then the top plate bending frequency should be greater than 1,000 Hz.

Achieving sufficient top plate stiffness requires a slightly thicker fastener than typical to allow for a thick top plate. A minimum of 2 in. should be made available between the rail base and invert. No other limitation is apparent. An additional advantage is that the elastomer strain can be kept to a practical minimum under static load with a 2-in.-thick fastener, and there should be increased reliability due to the increased strength of the fastener top plate, with less working of the rail clip due to top plate flexure under static load. Thickening the top plate will increase the cost of top plate castings, though the increase should be relatively small in comparison to the overall cost of the fastener. The fastener procurement specification for the BART extensions provides for an overall fastener thickness of 2 in.

10.3.7 Speed Variation

Corrugations at rail transit systems may be exacerbated by uniformity in train speed, vehicle type, and direction. Varying train speed may be beneficial in reducing corrugation growth rates (*52*).

10.3.8 Super-Elevation

Negative super-elevation imbalance appears to produce higher corrugation rates than no or positive imbalance (*53*). Super elevation imbalance is defined as the degree of uncompensated imbalance. Thus, a positive super-elevation imbalance exists where the super-elevation is insufficient to counter centripetal forces. A negative super-elevation imbalance exists where the super-elevation is more than adequate to overcome centripetal forces, resulting in a higher vertical load on the low rail than on the high rail. In this case, the low rail is most subject to corrugation. Low rail corrugation is also reported by BART in the survey questionnaire response. Evidently, avoiding negative super-elevation imbalance appears to help control corrugation rates.

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Abbreviations used without definitions in TRB publications:

AASHO	American Association of State Highway Officials
AASHTO	American Association of State Highway and Transportation Officials
APTA	American Public Transit Association
ASCE	American Society of Civil Engineers
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
FAA	Federal Aviation Administration
FHWA	Federal Highway Administration
FRA	Federal Railroad Administration
FTA	Federal Transit Administration
IEEE	Institute of Electrical and Electronics Engineers
ITE	Institute of Transportation Engineers
NCHRP	National Cooperative Highway Research Program
NCTRP	National Cooperative Transit Research and Development Program
NHTSA	National Highway Traffic Safety Administration
SAE	Society of Automotive Engineers
TCRP	Transit Cooperative Research Program
TRB	Transportation Research Board
U.S.DOT	United States Department of Transportation