UMTA-78-38

REPORT NO. UMTA-MA-06-0025-78-8

PREDICTION AND CONTROL OF NOISE AND VIBRATION IN RAIL TRANSIT SYSTEMS

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Research and Special Programs Administration
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SEPTEMBER 1978

FINAL REPORT

Prepared for

U.S. DEPARTMENT OF TRANSPORTATION
URBAN MASS TRANSPORTATION ADMINISTRATION
Office of Technology Development and Deployment
Office of Rail and Construction Technology
Washington DC 20590

Technical Report Documentation Page

1. Report No. UMTA-MA-06-0025-78-8	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle PREDICTION AND CONTROL OF NOISE AND VIBRATION IN RAIL TRANSIT SYSTEMS		5. Report Date SEPTEMBER 1978
		6. Performing Organization Code DTS-331
7. Author(s) Leonard G. Kurzweil and Robert Lotz		8. Performing Organization Report No. DOT-TSC-UMTA-78-38
9. Performing Organization Name and Address U.S. Department of Transportation Research and Special Programs Administration Transportation Systems Center Cambridge MA 02142		10. Work Unit No. (TRAIS) UM804/R8720
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address U.S. Department of Transportation Urban Mass Transportation Administration Office of Rail and Construction Technology Washington DC 20590		13. Type of Report and Period Covered FINAL REPORT
		14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

This report presents a unified compilation of the techniques, procedures, and data currently available for the assessment and control of rail transportation noise and vibration. The emphasis is on urban rail transit systems, however data on intercity passenger and freight trains are included. The noise environments treated include community noise and vibration near rail lines, vehicle interior noise for urban and intercity passenger trains, locomotive cab noise, and noise in stations and tunnels. This report is intended to serve as a reference document for transit system engineers and their consultants.

17. Key Words Noise, Vibration, Rail Noise, Rail Transportation Noise, Urban Rail Noise, Ground Vibration, Ground-borne Noise, Community Noise	18. Distribution Statement **DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161**

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	124	

Form DOT F 1700.7 (8-72)

Reproduction of completed page authorized

PREFACE

This report presents a unified and organized compilation of the techniques, procedures, and data currently available for the assessment and control of urban rail noise and vibration. It is intended to serve as a reference source for transit property personnel and their consultants.

The material contained in this report was compiled and organized at the Transportation Systems Center (TSC) under the Urban Rail Noise Abatement Program being sponsored by the Urban Mass Transportation Administration's (UMTA's) Office of Rail and Construction Technology. The Noise Abatement Program is part of a larger effort, the Urban Rail Supporting Technology Program, being managed at TSC for UMTA.

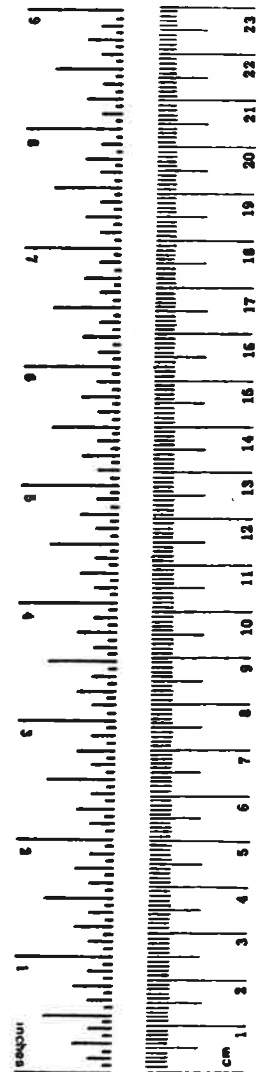
The material provided in this report will form the basis for the development of a Handbook of Urban Rail Noise and Vibration Control.

The efforts of Wesley Cobb, in developing additional material in Section 6 and editing the text, are gratefully acknowledged.

METRIC CONVERSION FACTORS

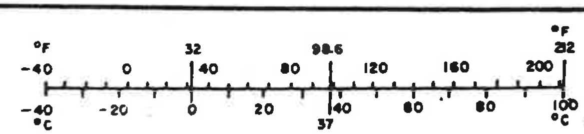
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. INTRODUCTION

This report treats noise prediction and control for metropolitan trains (e.g. subway or trolley) and for intercity passenger and freight trains. The noise environments treated include the following:

(1) train operation in tunnels,
(2) through cuttings,
(3) on surface (at-grade) track,
(4) embankments,
(5) on bridges and elevated structures,
(6) in freight classification yards,
(7) during track maintenance,
(8) and in passenger stations.

For each environment, some or all of the following topics are addressed: measured noise and vibration levels and spectra; prediction and control of noise and vibration; measurement methods; applicable standards, specifications, and criteria. The purpose here is to present a balanced introductory view of noise from rail transportation systems and its control, and to provide references to more specialized material. This report is a synthesis of data and methods from many workers, and it is consistent with past and present practice except where indicated.

For supplementary introductory reading the authors recommend Serendipity, [1970]; EPA, [1976]; Swing & Pies, [1973]; Manning, et al., [1974]. New work, often addressed to the specialist, is reported in the open literature in reports issued by the U.S. Department of Transportation, U.S. Environmental Protection Agency, the Journal of Sound and Vibration, and Proceedings of Inter-Noise and Noise-Con Conferences held annually.

The sections in this report are arranged in several source-path-receiver sequences. The first sequence deals with airborne noise in the community and is covered in the first four sections.

Sources of airborne noise are first presented for an open, at-grade situation where the measurements are normalized to standard measurement distances and other standard conditions (Section 2). Travel on elevated structures yields higher noise levels than travel at-grade. The amount of increase and methods to minimize elevated structure noise are given in Section 3. Section 4 shows how the normalized levels for at-grade and elevated sources can be used to predict wayside noise of trains in the open for a range of observer distances, ground covers, sound barriers, and train lengths. The wayside noise estimated in Section 4 provides the data needed in Section 5 to calculate community noise ratings, thus completing the source-path-receiver chain for air-borne noise to the wayside community.

Each of the remaining sections is self-contained and treats the full source-path-receiver chain for the following situations: groundborne noise to the community (Section 6), noise to patrons and employees in stations and tunnels (Section 7), and noise to passengers in railcars (Section 8). Comprehensive noise control on a system-wide basis is briefly discussed in Section 9.

2. NOISE SOURCES

Rail system noise results from operation of railcars (both self-propelled and locomotive-pulled), locomotives, warning signals, freight classification yards, and construction and maintenance equipment. The major noise sources are wheel-rail noise and propulsion system noise from railcars and locomotives.

Rail system noise sources are characterized by sound pressure level measurements made at standard distances from the track and at standard microphone heights. Sound power level and directionality, often used to characterize noise sources, are difficult to measure for such large sources as locomotives and railcars, and sound power data are rare [Nimura, et al., 1975].

Figure 2-1 shows the A-weighted sound pressure level for a passing train of self-propelled electric railcars. The level fluctuates about an average maximum (indicated by the horizontal line in Figure 2-1) as the train passes by. This pattern is typical of noise measured at 15 to 30 m (50 to 100 ft) from self-propelled cars including trams, rapid transit, commuter, and intercity railcars. In contrast, for locomotive-hauled trains (Figure 2-2), the locomotive often stands out as a distinct maximum followed by the railcar noise. The discussion below on railcars and locomotives refers to such maximum noise levels, to the corresponding spectra, and to the effects of design and operating parameters on these quantities. Except where indicated, discussions emphasize U.S. systems and pertain to trains in the open, on tie and stone ballast at-grade track.

2.1 RAILCARS

2.1.1 Noise Levels and Spectra for Railcars Traveling on Tangent (Straight) Track

To permit comparisons among data and to present standardized conditions for estimating noise, measured data are normalized

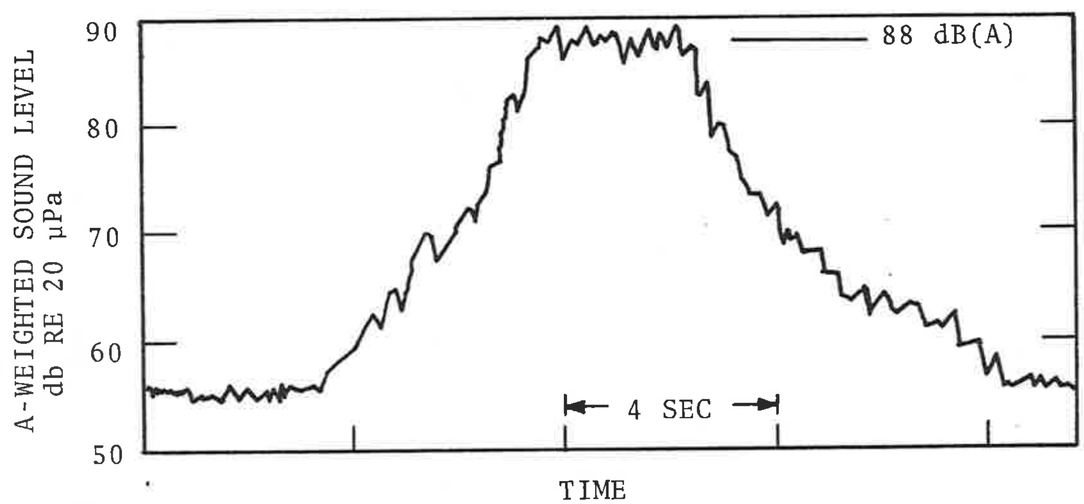


Figure 2-1. Time history of A-weighted sound level during pass-by of six, 85-foot long, self-propelled electric railcars at 30m (100 ft) traveling 177 km/h (110 mph). From [Rickley, et al., 1973; Fig. A-4(c)].

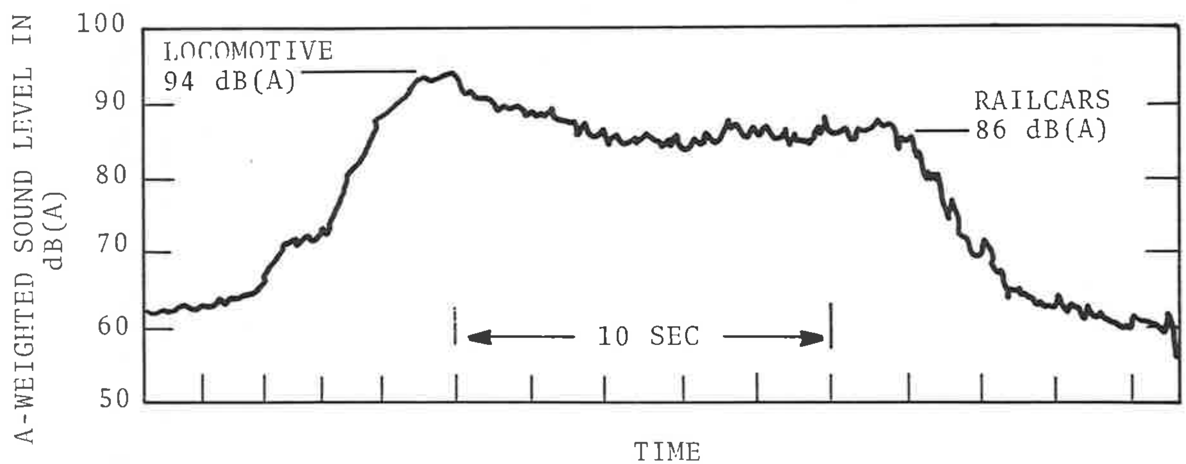


Figure 2-2. Time history of A-weighted sound level during pass-by of a locomotive-hauled passenger train at 30m (100 ft) traveling at 114 km/h (71 mph). After [Rickley, et al., 1973; Fig. C-13(c)].

here to a long train and a standard measurement distance of 30 m (100 ft) to the track centerline. A train is considered long when its length exceeds three times the measurement distance. The method of normalizing and of estimating noise under other conditions are described in Section 4.

When normalized A-weighted source levels for a variety of passenger cars, traveling on continuous welded rail*, are plotted versus speed, the trend line through the center of the data is given by the relation [Lotz, 1977]:

$$L_A(30 \text{ m}) = 74 + 30 \log_{10} V/V_o \text{ dB(A)} \qquad (2.1)$$

where $L_A(30 \text{ m})$ is the distance normalized A-weighted sound level, V is the railcar speed in km/h (or mph) and V_o is a reference speed of 60 km/h (37 mph). The data used in obtaining Equation 2.1 are from more than 50 measurements including both unpowered and electric self-propelled cars at more than 20 sites. Ninety percent of the data lie within ± 6 dB of the line defined by Equation 2.1. Much of the scatter in the data at a given speed is due to differences in the wheels and rails.

Equation 2.1 implies an increase of 9 dBA per doubling of speed. Although this is a typical speed dependence, some systems show clearly different dependences. For example, the Shinkansen trains of Japan show an increase of 6 dBA per doubling of speed [Arai & Ban, 1975; Lotz, 1977].

Measurements of the normalized A-weighted sound levels for passenger cars on jointed rail (or for freight cars in the U.S.A. on both jointed and welded rail) show that more than 90 percent of the data lie within ± 6 dB of the relationship:

*Rail is commonly made in lengths of 11.9 m (39 ft). The rails are then joined together either by using bolted joint bars or by welding. The passing of train wheels over loose joints results in the commonly heard "clickey-clack."

$$L_A(30\ \mathrm{m}) = 81 + 30 \log_{10} V/V_o \ \mathrm{dB(A)} \tag{2.2}$$

These data include measurements of 85 trains from more than 15 sites.

The relative octave band spectra for three types of railcar are shown in Figure 2-3. Relative spectra are obtained by subtracting the A-weighted sound level for a given pass-by noise measurement from the (unweighted) octave band sound levels of the same measurement. For each type of railcar the average spectrum is based on data from welded and jointed rail and a range of speeds and wheel conditions. The relative octave band data emphasize U.S. railcars and show no consistent trend with speed. This is in contrast with some data for European trains which show a spectral shift toward higher frequencies as speed increases. The spectrum for modern electrically powered vehicles covers all three uses: tram, rapid transit, and intercity. Systematic spectral differences due to these different uses are too small to warrant separate consideration. Likewise, no consistent differences were found between welded and jointed rail relative spectra.

2.1.2 Wheel-Rail Noise Generation on Tangent Track and Its Control

The dominant noise source for railcars over most of their speed range is wheel-rail interaction. Excitation of wheel-rail noise on tangent track generally is attributed to rail and wheel surface roughness. A variety of conditions and their effects on wheel-rail noise are summarized in Table 2-1. Railcars traveling on smooth wheels and smooth continuous welded (jointless) rail emit a steady wideband noise, sometimes called rolling noise. Smoothness of wheels and rails are maintained by special wheel truing machines and rail grinding trains. New rail is often ground after it is layed to remove mill scale. Measurements of rail and wheel roughness show it to be random under most situations as the surfaces roughen with wear. When the roughness is

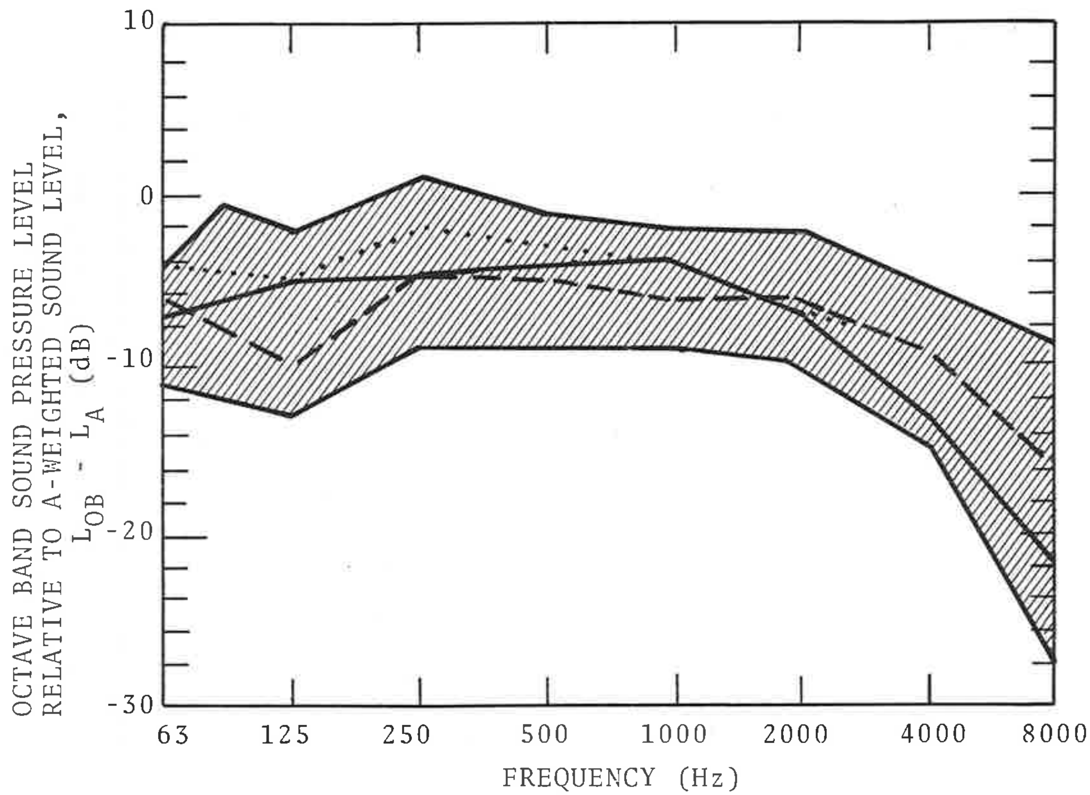


Figure 2-3. Octave band relative spectra for railcars in the open.

——— Electrically powered passenger cars - average of 10 spectra

----- Unpowered passenger cars - average 7 spectra

····· Freight cars - average of 8 spectra

///// 90% of all data

TABLE 2-1. Effects of type and condition of wheels and rails on wayside-noise - tangent (straight) track. [Lotz, 1977]

Noisier Condition/Quieter Condition	Typical Difference (and range)
1. Rough welded rail/Ground smooth welded rail	4 (3-6) dB(A)
2. Rough wheels/Trued smooth wheels	5 (3-6 dB(A)
3. Corrugated rail/Uncorrugated rail	10 (5-15) dB(A)
4. Wheels with flats/Trued smooth wheels	12 (7-15) dB(A)
5. Jointed rail/Welded rail	
(a) Passenger Cars	7 (4-10) dB(A)
(b) Freight, mainline track	2 (0-3) dB(A)
(c) Freight, low speed track	6 (4-8) dB(A)
6. Switch/No Switch	6 (5-8) dB(A)
7. Ordinary wheels/Wheels with viscoelastic damping treatment	1 (0-2) dB(A)
8. Ordinary wheels/Wheels with snap-ring damping treatment	1 (0-1) dB(A)
9. Ordinary wheels/Wheels with resiliently mounted rims	2 (0-3) dB(A)

permitted to develop, noise increases (Table 2-1, items 1 and 2). For reasons not fully understood, corrugations sometimes develop on the running surface of the rail as it wears, often on curves or near stations. A wheel which travels over such a surface is subject to nearly periodic excitation, which results in substantially higher sound level than during travel over uncorrugated rail (Table 2-1, item 3).

When wheels develop flat spots (Table 2-1, item 4), or when railcars run over discontinuities in the rail surface, for example, at rail joints or at special track work such as switches (Table 2-1, items 5 and 6), wheel-rail noise is dominated by a succession of broadband <u>impact sounds</u> typically of rise time 0.01 sec and duration 0.05 sec (within 10 dB of the maximum).

Good design and maintenance practices avoid this impact noise in urban and suburban transit applications except at occassional special trackwork. Joint noise is avoided by using continuous welded rail. In those systems using jointed rail, joints should be maintained smooth and tight. Wheel flats are minimized by brake systems which prevent sliding of wheels, and wheels should be trued smooth as soon as possible if flats do form.

Several design features show promise for still quieter systems than those used in computing Equation 2.1 and 2.2. These include: wheels with high damping or resilience (Table 2-1, items 7 to 9), disk brakes rather than wheel-tread brakes (reported reduction 10 dB), and very resilient primary suspension rather than moderate resilience [Wilson, 1977; a 30 to 1 reduction in the journal sleeve stiffness resulted in a 3-5 dBA reduction in the wayside noise level].

2.1.3 Estimating Railcar Noise on Tangent Tracks

For average situations, Equations 2.1 and 2.2 can be used directly to estimate A-weighted sound levels of long trains at an observer distance of 30 m (100 ft). Adjustments to other train lengths and observer distances are discussed in a later section.

For predicting the effect of wheel-rail condition on the wheel-rail noise it is convenient to start with the levels of a well designed and maintained system (typically 6 dBA below the levels given by Equations 2.1 and 2.2) and adjust these levels by adding the corresponding difference in level given in Table 2-1 for a noiser condition. Few experiments [Saurenmen & Holowaty, 1977; Holowaty, et al., 1976] have been designed to determine the net effect of combined conditions--for example, the difference in sound level between flatted ordinary wheels on rough welded rail vs. smooth, resilient wheels on smooth welded rail--with all other conditions held constant. Experience suggests using only the largest single effect when estimating [Swing & Pies, 1973; Manning, et al., 1974]; for the present example this would give a typical value of 12 dB corresponding to Table 2-1, item 4. If this approach is followed, a typical A-weighted sound level 30 m (100 ft) from a long train of electrically powered passenger cars traveling at 40 km/h (25 mph) on flatted ordinary wheels and rough welded rail would be estimated as:

$$L_A(30\text{ m}) = \underbrace{(74-6) + 30\log\frac{40}{60}}_{\substack{\text{Quiet system at 40 km/h}\\ \text{(25 mph)}}} + \underbrace{12}_{\substack{\text{Largest}\\ \text{effect}}}$$

$$= 68 - 5 + 12 = 75\text{ dBA}$$

Adding the value of $L_A(30\text{ m})$ computed above to the average relative spectrum for electric passenger cars (given in Figure 2-3) the following estimated spectrum is obtained:

Octave Band Center Frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Octave Band Sound Pressure Level, dB re 20 μPa	68	70	70	71	71	68	62	53

2.1.4 Noise Levels and Spectra for Railcars Traveling on Curved Track

Unless preventive measures are taken, the dominant noise for railcars traveling on curves with radii less than about 100 m (328 ft) is a squeal or screech consisting of one or more pure tones (discrete frequencies). Wheel-rail noise consisting of pure tones may also be generated on larger radius curves. The average A-weighted sound level for curve squeal on urban transit systems is 89 dBA, with a range of ± 10 dBA for individual events. This refers to data normalized to a distance of 30 m (100 ft) assuming spherical spreading. As shown in Figure 2-4, the distinguishing feature of the spectrum of such a squeal is the dominance by inharmonic discrete frequency components. Spectrum shapes vary considerably - often even at a single curve for cars of the same fleet, or for the same car on different days. Rolling and impact noise (see above) sets a lower limit for noise on curves.

2.1.5 Wheel-Rail Noise Generation on Curved Track and Its Control

Curve squeal is generated through a process similar to chalk squeal on a blackboard -- when a train negotiates a curve, wheels slide on the rail (in addition to rolling). There are several reasons why sliding occurs: (1) A conventional two-axle truck holds its axles parallel, like a pram or stroller (in contrast with a child's wagon which has one steered axle). Curve negotiation thus forces wheels to slide 90° to the direction of rolling. (2) In a curve, the outside wheel must traverse a longer path than the inside wheel, but the wheels <u>roll</u> the same distances because they are attached to the same axle. The difference is made up by wheel sliding parallel to the direction of rolling. (3) Severe braking or acceleration also can cause sliding along the rail, even on tangent track. (4) The wheel flange will rub against the rail if the wheel is skewed sufficiently to it; such skewing can occur on sharp curves. (5) A restraining rail is sometimes installed on curves next to the inside rail. It assists in guiding the railcar around the curve by pressing toward the curve center on the sides of the wheels; rubbing ensues.

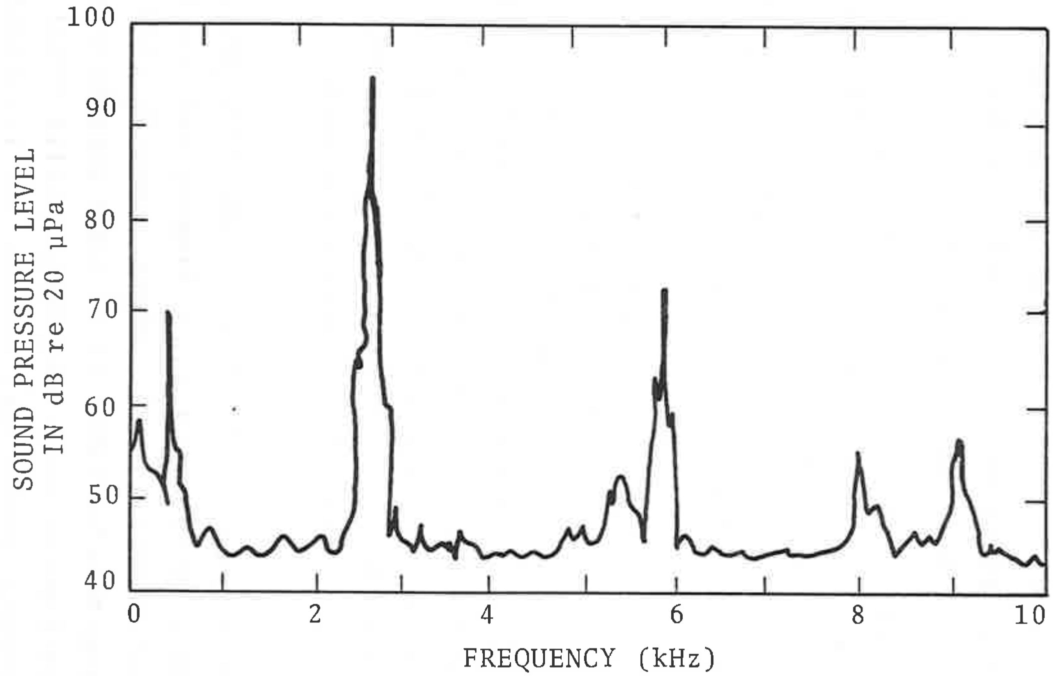


Figure 2-4. Squeal spectrum from an urban rail transit car on a curve of radius 43m (140 ft). After [Kohler, 1975].

Under all these conditions, conventional all-metal wheels may squeal. Factors which affect squeal generation include curve radius, vehicle speed, truck geometry and rigidity, wheel damping, and contact surface frictional characteristics. The dominant discrete frequencies often correspond approximately to natural frequencies measured on the wheel, free of the rail. The generally accepted explanation of curve squeal is that a stick-slip instability from the enforced sliding results in large vibration amplitudes for one or more natural modes of the wheel--especially modes with predominantly out-of-plane motion. Sound is radiated from the vibrating surfaces [Remington, et al., 1975].

To control curve squeal it is often most practical to damp the wheel vibration. Approaches available for this, in order of increasing complexity and cost, are: (a) Snap-ring damping [Kohler, 1975]. Existing or new wheels can be modified by cutting a groove inside the rim fillet and snapping an open steel ring into the groove. This provides about 6 to 18 dB reduction of discrete frequencies above 1000 Hz but has little effect at lower frequencies. (b) Constrained layer damping [Kirscher, 1972]. This treatment may be applied to existing or new wheels and virtually eliminates squeal. (c) Resilient Wheels. This name is applied to wheels having elastomeric materials between the steel rim and the web. Several designs are available which virtually eliminate squeal. The higher first cost of this approach is partly offset by the need to replace only the removable rim when the wheel has become worn.

In North America some objections, aside from cost, have been raised against constrained layer damping and resilient wheels. They are principally uncertainties about compatibility with system operation and long-term reliability of such composite wheels, as compared to unmodified or snap-ring damped solid metal wheels. However, many European urban rail systems use resilient wheels successfully, and in-service tests have been undertaken in the United States to clarify these issues [Saurenman & Holowaty, 1977].

Curve noise control methods other than wheel damping include: (a) not constructing turns of radius less than about 100-150 m (330-500 ft), (b) using trucks (sometimes called radial trucks) which, rather than holding the axles parallel, permit them to lie along curve radii, (c) modifying the frictional character of the rubbing interfaces, e.g., lubricating the wheel flange, the side of the rail head, or the top of the rail head.

2.1.6 Railcar Noise From Sources Other Than Wheel-Rail Interaction

For electrically-driven railcars the major source of noise (other than that resulting from wheel-rail interactions) is that from propulsion equipment, which includes the traction motor, reduction gears, and the traction motor air-cooling system. Another important noise source, especially for stopped trains, is the forced air ventilation system, or air conditioning system.

Aerodynamic noise due to vehicle forward motion does not contribute appreciably to wayside noise from railcars at speeds less than about 240 km/h (149 mph); at higher speeds it may be an important factor. [King & Bechert, 1976; King, 1977]

For third rail power systems, power pickup noise (resulting from contact between the "shoe" or "paddle" and the third rail) usually is masked by noise generated by the wheels and rails. Trolley cars which are powered by overhead wires have a separate discernable source of noise resulting from contact between the trolley and the overhead wire.

2.2 LOCOMOTIVES

With few exceptions, locomotives in commercial use are driven by electric traction motors coupled to the axles. Electric power is provided by diesel or gas turbine generators in the locomotive or by overhead electric power lines.

2.2.1 Noise Levels and Spectra for Locomotives

Figure 2-5 shows the range and distribution of locomotive A-weighted sound levels 30 m (100 ft) from the track centerline as the locomotives pass the point of measurement. These data cover a wide range of speeds, grades, and types of tracks. For diesel-electric locomotives 90 percent of the maximum sound levels at a distance of 30 m (100 ft) are between 87 and 96 dBA. At that distance noise from this type of locomotive generally exceeds railcar noise, even at speeds as high as 129 km/h (80 mph).

Exhaust noise is the dominant source. It depends on engine load and, in contrast with wheel-rail noise, is independent of speed [Close & Atkinson, 1973]. Mufflers are available which reduce exhaust noise by about 10 dBA. When installed, however, the overall effect on diesel-electric locomotive noise ranges only from 4 to 8 dB reduction, indicating the presence of other sources. These include engine casing vibration and cooling fan aerodynamic sources.

Electric and turbine locomotives average 6 to 7 dB quieter than unmuffled diesel-electric locomotives. The octave band spectra for these two locomotive types relative to maximum A-weighted sound level are given in Figure 2-6. Note that above 125 Hz the two spectra agree within 2 dB.

2.2.2 Warning Signals: Horns and Whistles

Typically, horns and whistles produce A-weighted sound levels of about 105 dBA at 30 m (100 ft) ahead of the locomotive, with levels of about 5 to 10 dB less to the sides of the track.

2.3 FREIGHT CLASSIFICATION YARDS

Freight classification yards receive incoming trains of cars bound for various destinations and reassemble them into outbound trains of cars with like destinations. In addition, locomotives and cars are maintained and sometimes stored in these yards. These classification, storage, and maintenance operations produce

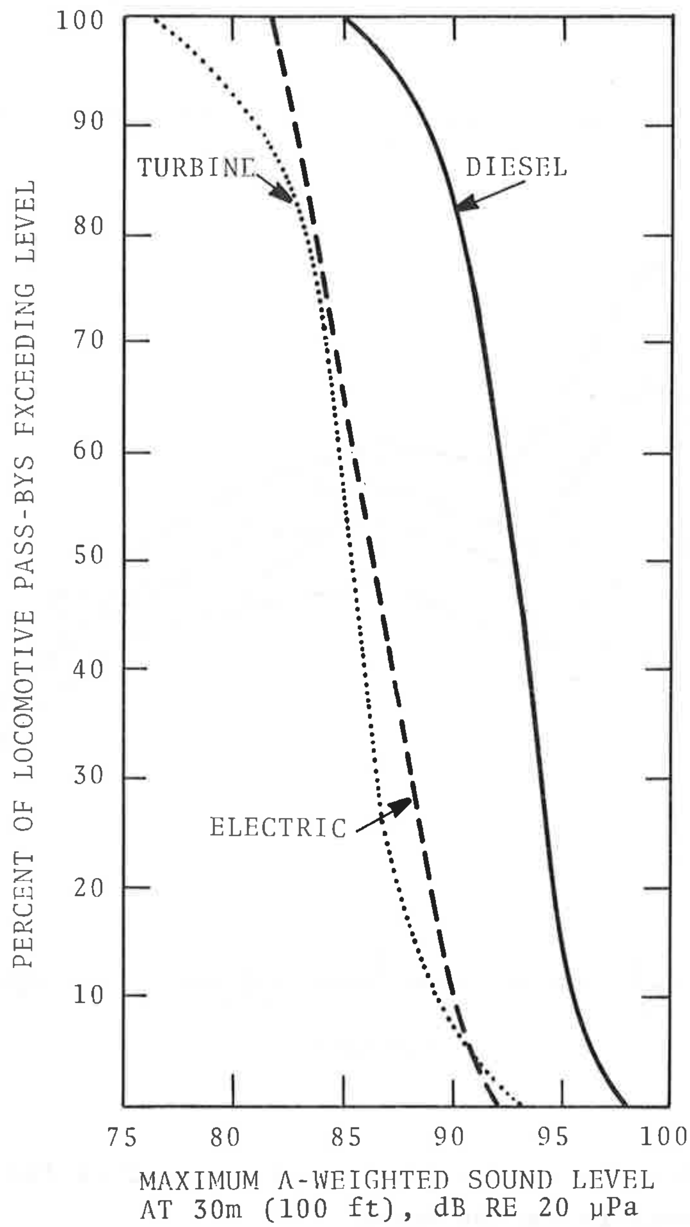


Figure 2-5. Distribution of A-weighted sound levels for three types of locomotives in the U.S. for a variety of sites, train types, number of locomotives per train, and train speeds. After [Close & Atkinson, 1973].

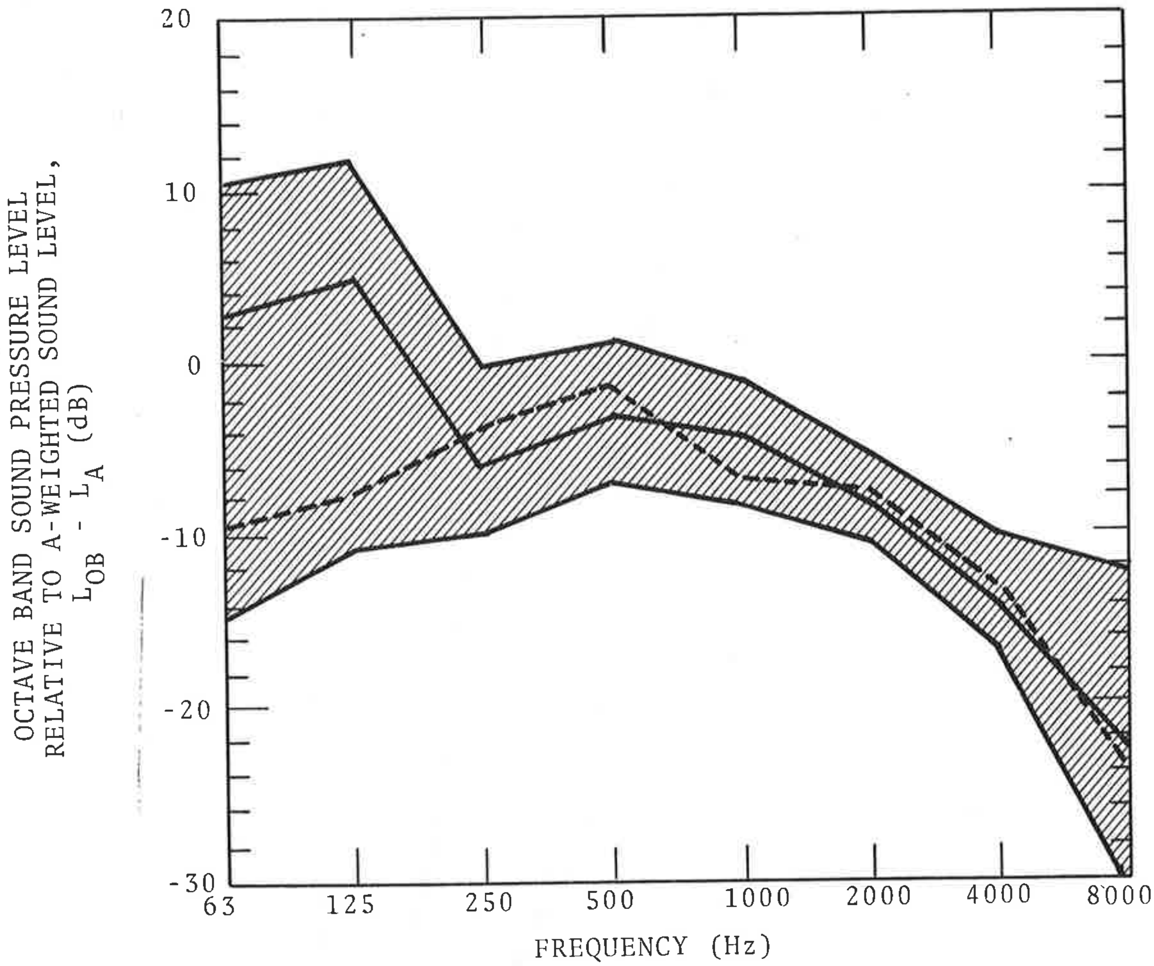


Figure 2-6. Average octave band relative spectra for locomotives in the open:
——— Diesel-electric locomotives;
- - - - - - Electric locomotive;
▨ Range of all available data.

a variety of noises which differ widely in intensity, number, location, and duration depending on yard design and operation. A clear step-by-step aggregation procedure [Swing & Pies, 1973] exists which permits L_{dn} contours (see Section 4) to be plotted if information is available concerning the yard layout, operations, and conditions of sound propagation. In contrast, the data given in Table 2-2 are for noise levels due to the operation of specific individual items of yard equipment.

One method of forming outbound trains is to use a "switcher" locomotive to push a group of cars to the top of a man-made hump. A car or group of cars is released and coasts down the hump, through a series of switches and onto one of many classification tracks. Along the way remote controlled active retarders (beams--two to each rail--which squeeze the car wheels between them) slow the car to the correct speed for its weight and required coasting distance.

The first car on a particular track is stopped at the far end by a spring-loaded or weight-loaded inert retarder. Subsequent cars couple automatically on impact with the cars already on the track, producing a major source of noise. Such impact noise has a duration of about 1 to 3 sec (during which time the A-weighted sound level is within 20 dB of its maximum). Similar impact sounds are generated after the locomotive begins to pull a completed train, removing any slack between cars. The range of levels is shown in Table 2-2. Relative spectra show the typically broadband nature of this source and are given in Figure 2-7.

After the train is made up it is pulled out through the inert retarders, freeing that track. Noise from active and inert retarders is generally considered by the community to be the severest noise in classification yards. The range of levels is shown in Table 2-2. As shown in Figure 2-8, the sound has narrow band components with one or more 1/3-octave bands between 2000 and 4000 Hz dominating the spectrum. Significant factors in

TABLE 2-2. Freight classification yard noise sources

A-Weighted Sound Level at 30 m (100 ft)* for a Single Unit or Event

Source	Typical or Average	Range Including at least 90% of data
Switcher locomotives:		
working [Swing & Pies, 1973; Swing & Inman, 1974]	79	75-90
idling [Swing & Inman, 1974]	63	-
Road locomotives idling [Swing & Inman, 1974]	69	-
Car coupling [Swing & Pies, 1973; Swing & Inman, 1974; EPA, 1976] (Including coupling impact and slack run-in and runout)	100	69-112
Retarder [Rickley, et. al., 1974; Close & Atkinson, 1973; EPA, 1976]	100	74-118
Mechanical Refrigerator Cars [Swing & Pies, 1973; Swing & Inman, 1974]		64-74
Locomotive Engine load cell tests [Swing & Pies, 1973; Swing & Inman, 1974]	92	-
Public Address Loudspeakers [Swing & Pies, 1973] (In the direction pointed)	-	90-96
Diesel-hydraulic equipment for handling "piggy-back" containers and trailers [Rickley, et. al, 1974]	-	78-82

*Normalized to 30 m (100 ft) on the basis of -6 dB per doubling of distance.

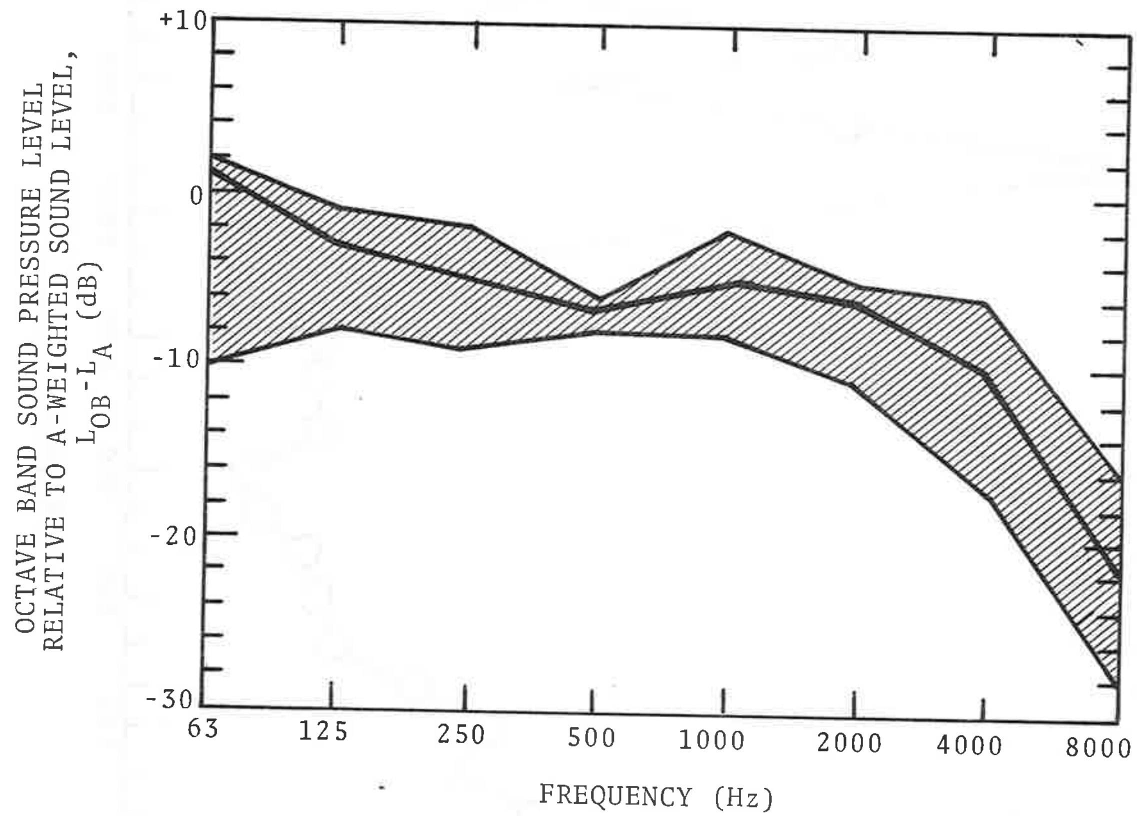


Figure 2-7. Octave band relative spectrum of freight car coupling and slack run-out.

——— Average of 4 spectra

Range

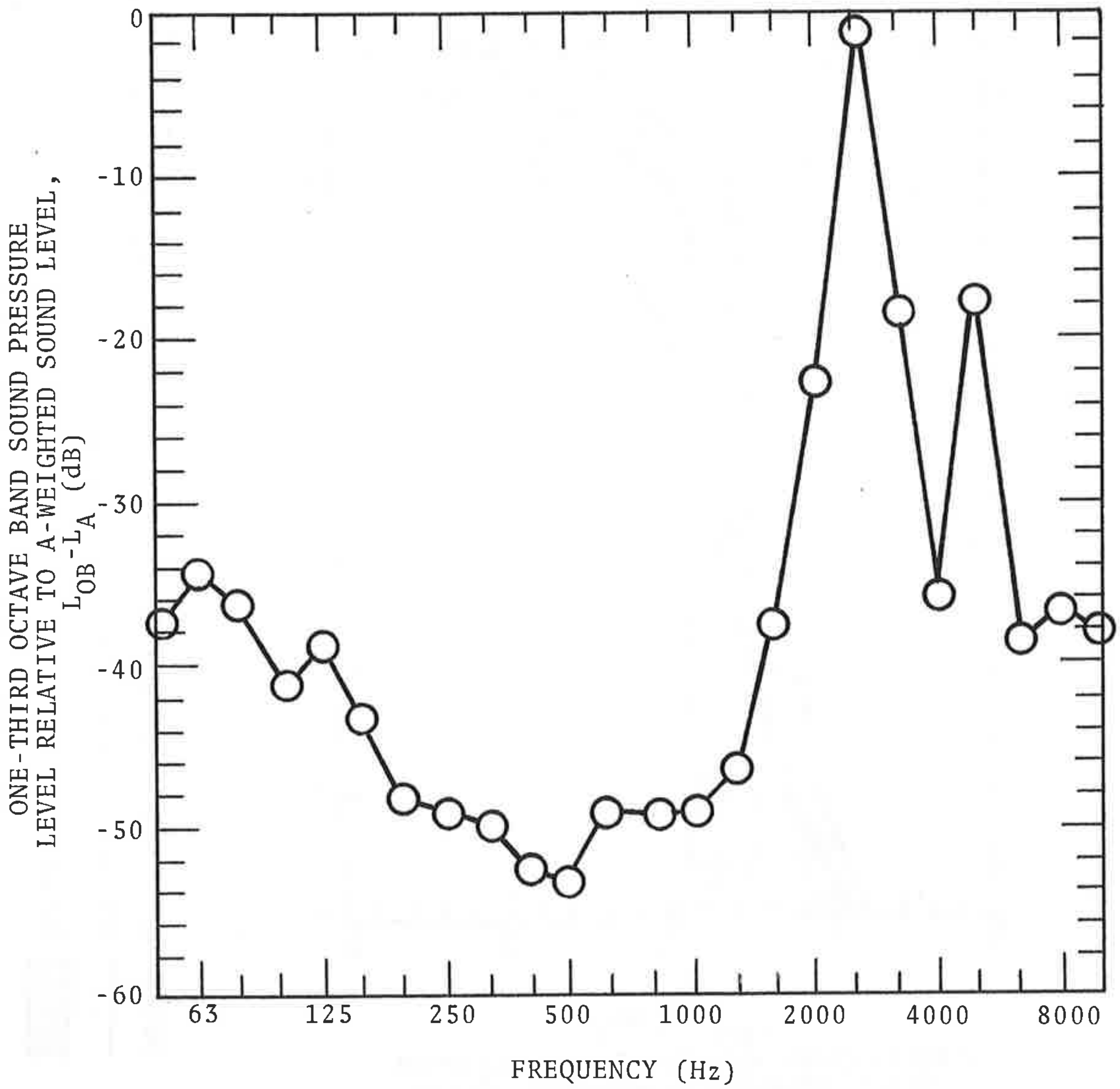


Figure 2-8. One-third octave band relative spectrum typical of retarder squeal. From [Rickley, et al., 1974; Fig. B-15 (b)].

determining whether retarder squeal will occur, and its level if it does, are believed to inlcude car speed, retardation force applied, wheel construction, presence of lubrication, condition of wheels and retarder, [Swing & Inman, 1974; Swing & Pies, 1973] but apparently not car weight [Bender, et al., 1974a]. The mechanism is believed to be a wheel stick-slip excitation or a retarder beam resonance. Some designs of inert retarders may be manually opened to permit free passage of wheels, thus avoiding squeal when the cars are pulled through.

Mechanical refrigerator cars are usually parked together with the diesel engines running constantly to maintain refrigeration. Idling locomotives are also parked in groups. To estimate sound levels at distances greater than any dimension of a cluster of N similar vehicles, one may calculate as though the sound were entirely from the cluster center due to a single vehicle having 10 Log N dBA greater level than in Table 2-2.

2.4 NOISE DUE TO CONSTRUCTION AND MAINTENANCE OF WAY OPERATIONS

Construction of new railroads and rail transit systems requires diesel, hydraulic, and pneumatic equipment typical of other heavy construction. Noise data on this equipment and on specialized equipment for track construction may be found in Lotz, [1977] and in EPA, [1971]. Little data is available on the noise levels of track maintenance equipment. Nighttime maintenance-of-way has sometimes caused strong community action. Several railroads and urban transit systems have found that advising the community of maintenance schedules in advance results in fewer complaints.

2.5 RUBBER-TIRED TRANSIT SYSTEMS

Trains of railcar-size passenger vehicles with rubber tires operate in urban transit service on exclusive rights-of-way in a number of cities, e.g., Montreal, Sapporo, Mexico City, and Paris. Smaller, lighter, rubber-tired transit vehicles carry passengers in several U.S. cities, e.g , at airports and tourist attractions [Lea Transit Compendium, 1974]. Rubber-tired systems do not ordinarily squeal. Otherwise noise levels for rubber-tired and for steel-wheel-on-welded-rail systems in the open are about the same when comparisons are made under equivalent conditions, e.g., speed, train length, and distance-to-track.

For straight track operation below 64 km/h (40 mph), the maximum speed for which data is available, rubber-tired systems produce normalized* wayside noise levels within ±6 dB of Equation 2.1, which was obtained for steel wheel system operation on continuous welded rail. Normalized noise levels in this range are also consistent with those from new truck tires and rubber-tired electric trolley buses. The octave band relative spectrum for rubber-tired transit is shown in Figure 2-9.

For stationary rubber-tired and steel-welded vehicles, noise levels are similar because the electrical and mechanical equipments are similar.

2.6 WAYSIDE NOISE MEASUREMENTS AND CRITERIA

Wayside measurements of rail system noise are usually made for one of the following reasons: (a) to check compliance with noise emission regulations or purchase specifications, (b) to determine community noise exposure from an existing or proposed rail system, or (c) to diagnose a noise problem and evaluate the acoustic performance of various railcar and track designs.

In regulations for interstate freight and passenger rail systems, the U.S. Environmental Protection Agency (EPA) has established measurement criteria and noise standards [EPA, 1976]. A microphone is located in an open area 30 m (100 ft) from the

*Normalized to a long train at 30 m (100 ft) on the basis of a simple (monopole) line source, which is appropriate for tire noise. Use of the dipole line source model described in Section 4 does not change the conclusions.

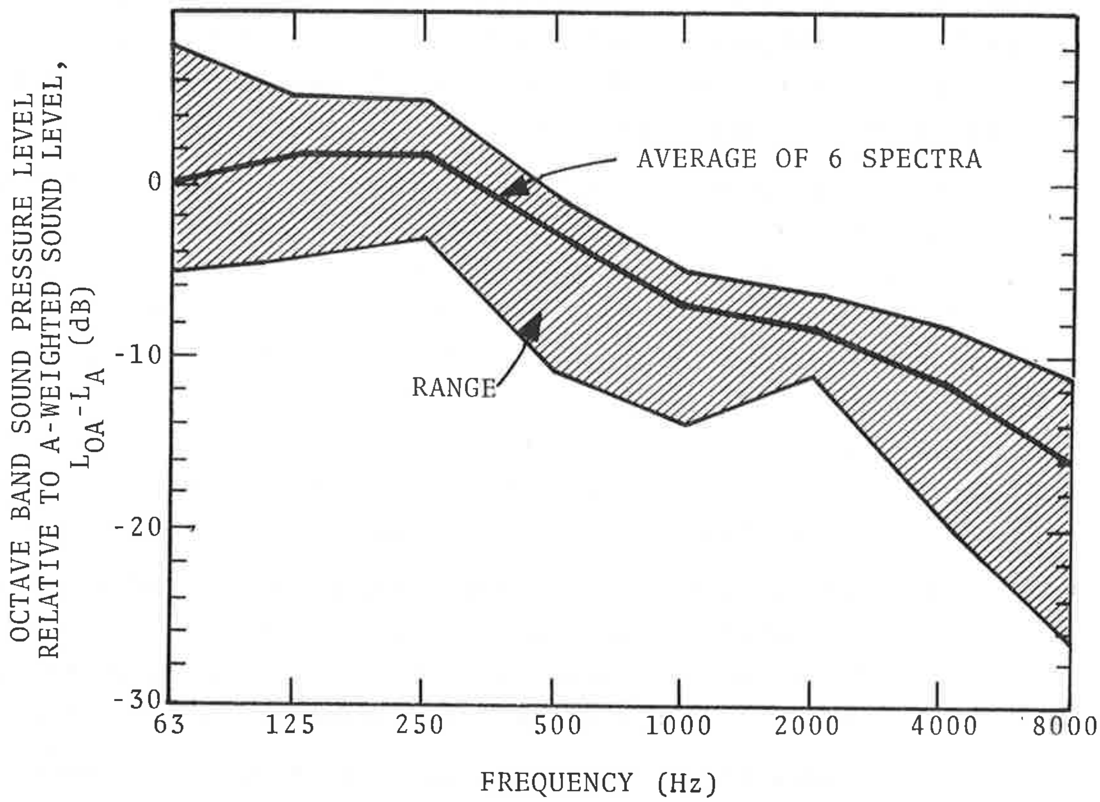


Figure 2-9. Octave band relative spectra for rubber-tired transit systems in the open.

track centerline and at a height of 1.2 m (4 ft); sound level readings are taken for locomotives and railcars in motion, and for stationary locomotives. Specifications governing purchases or renovations of locomotives or self-propelled diesel rail cars should take these standards into account.

For purchases of new electric and diesel-electric locomotives, and for auxiliary power cars which contain diesel-powered generators the U.S. National Railroad Passenger Corporation (Amtrak) specifies A-weighted sound levels at 30 m (100 ft) for stationary vehicles: at idle 67 dBA is permitted, and under full power and load 87 dBA is permitted [Amtrak, 1974, 1975, 1976].

Purchase specifications often give the limit for wayside noise from urban rail passenger cars.* Three measurement conditions are used to check compliance of the first few cars manufactured, and should be used also to check sample cars taken from those manufactured later in the production run. They are:

(1) Prior to installation on the car, traction motors, propulsion gearing, and auxiliary equipments are operated individually in test rigs to simulate actual service.

(2) After installation these components are operated individually and in combination while the car is stationary. This includes propulsion system operation up to full speed with the car wheels jacked-up clear of the rail.

Under these two conditions measurements are made of the maximum A-weighted sound levels at a distance of 4.6 m (15 ft). At rotational speeds equivalent to 97 to 129 km/h (60 to 80 mph), specifications prohibit A-weighted sound levels exceeding 82 to 90 dBA for propulsion components, 65 to 68 dBA for other components. For a complete stationary car, maximum noise at a distance of 15 m (50 ft) often is specified as well.

*The numerical values given in this discussion are identical to those in APTA, [1976] and consistent with modern design practice in the United States.

(3) Complete cars, usually operated as two-car trains in the open on tie-and ballast tangent track, are measured at 15 m (50 ft) from the track centerline for speeds up to their design maximum. Typical specifications for two-car train operation on welded rail are presented below.

km/h	(mph)	Sound Level, dB re 20 μPa
0	(0)	60
32	(20)	68
64	(40)	76
97	(60)	80
129	(80)	84

A penalty of 3 to 5 dB is generally applied if discrete frequency noise is present. To estimate the A-weighted sound level of a long train of cars at 30 m (100 ft) from the sound level of a 2-car train at 15 m (50 ft), subtract 2 to 3 dB from the latter.

For both EPA and transit compliance measurement, a type II Sound Level Meter set on A scale, calibrated within two years, is required, and a wind screen should cover the microphone. The fast dynamic meter response is used except for steady noise from stationary transit cars which calls for slow response. Microphone elevation is 1.2 m (4 ft) above ground under the EPA criteria, and at axle height or 1.2 m (whichever is higher) under the American Public Transit Association (APTA) guidelines [APTA, 1976]. The International Standards Organization (ISO) and the International Union of Public Transport (UITP) also have standards [UITP, 1972a; ISO, 1975]. The reader should consult the referenced documents for detailed information and valuable suggestions.

Measurements made for the purpose of estimating community noise from rail lines (see Sections 4 and 5) are usually made at or normalized to 15 to 30 m (50 to 100 ft) distances. When a definite quantitative relation can be established between vehicle interior and exterior noise, on various track structures, it may be possible to use interior measurements as a basis for estimating wayside noise [BB & N, 1976].

3. NOISE RADIATION FROM ELEVATED STRUCTURES

The term elevated structure includes both rail-transit elevated guideways and railroad bridges. When trains run on such elevated structures, the wayside A-weighted sound levels can be as much as 20 dBA greater than corresponding levels when the train runs on track at-grade (see Table 3-1). This increase is primarily due to the secondary radiation emitted from the vibrating components of the elevated structure.

3.1 NOISE LEVELS AND SPECTRA

The noise level near an elevated structure depends on numerous parameters including: rail vehicle type and condition, train speed, train or bridge length, method of joining rail (welded or bolted joints) and joint condition, type of track system (wood ties with or without ballast, concrete slab, or steel plate deck), and the type of supporting structure. Extensive measurements of noise near a variety of elevated structures have been conducted [ORE, 1971; Manning, et al., 1975a; JNR, 1975]. Figure 3-1 presents the arithmetic mean and range of speed-adjusted sideline noise levels for various elevated structure configurations.

For speeds, V, in the range 20 to 120 km/h (12 to 75 mph), the A-weighted sound level, L_A (30 m), 30 m (100 ft) from a long train and structure, is given by:

$$L_A \text{ (30 m)} = L_A' + 30 \log_{10} \frac{V}{V_o} \text{, dB(A)} \tag{3.1}$$

and the normalized sound pressure level, L_p, is given by:

$$L_p \text{ (30 m)} = L_p' + 20 \log_{10} \frac{V}{V_o} \text{, dB} \tag{3.2}$$

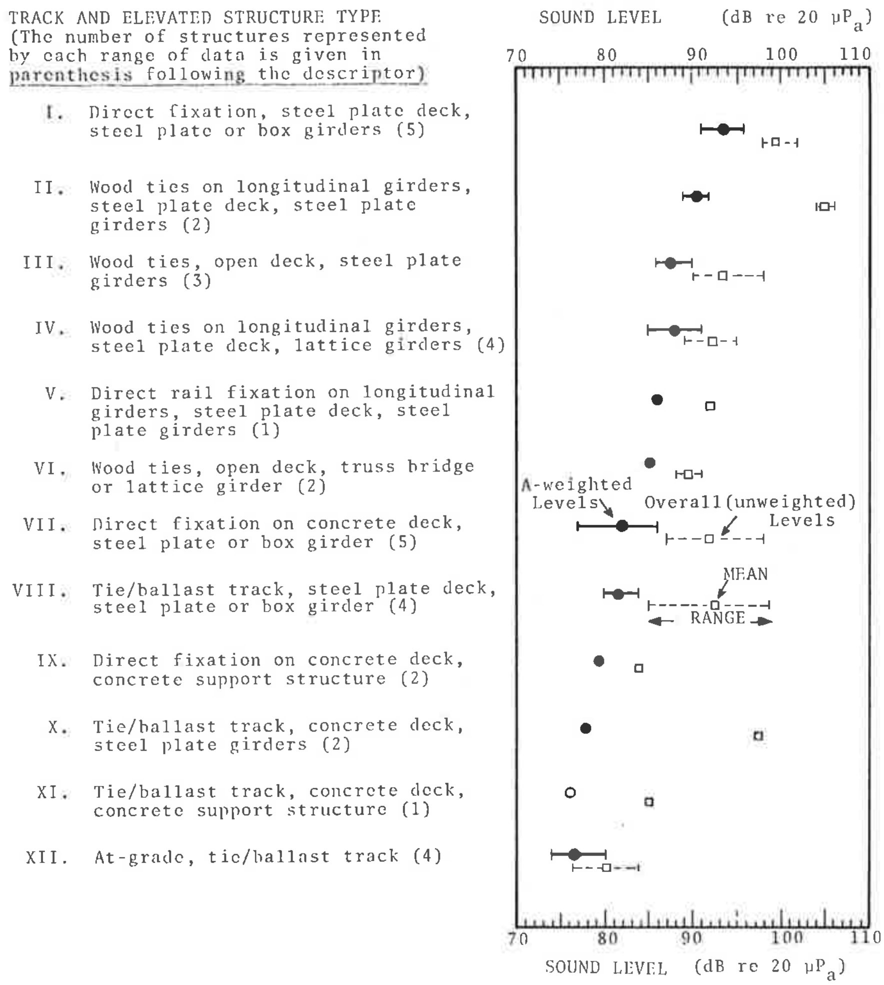


Figure 3-1. Range of speed-adjusted normalized A-weighted (L'_A) and overall (L'_p) sound pressure levels for various track and elevated structure configurations. All data are normalized for a long train pass-by at 60 km/h (37 mph) and a measurement distance of 30m (100 ft) from the track, 1.2 to 1.6m (4 to 5.25 ft) above the ground. |———| Range of A-weighted sound levels, dB(A); |-----| Range of overall (unweighted) sound pressure levels, dB; ●, □ Arithmetic average of levels.

where V_o is the reference train speed of 60 km/h (37 mph); V is the actual train speed in km/h or mph; L'_A and L'_P are the normalized levels at the reference train speed and are given in Figure 3-1. The difference in the speed dependence of L_A and L_P for an elevated structure indicates a spectral shift to higher frequencies as speed increases.

Table 3-1 summarizes the differences in wayside sound levels between railcar operation on a given class of elevated structure and on track at grade. It can be used in conjunction with the material in the preceding section to estimate levels for all types of elevated structures and to estimate the effects of wheel and rail noise control treatments given in Table 2-1, on elevated structure noise.

Figure 3-2 shows the average of the relative spectra for railroad bridge structures and for urban rail transit elevated structures with trains passing at 60 km/h (37 mph). Structure types II and X (see Figure 3-1) have relative spectra which fall near the upper bound of the range shown in Figure 3-2 at frequencies below 500 Hz. Structure types IV, VI, and IX have spectra falling in the lower portion of the range, below 500 Hz. The shape of these spectra changes with speed.

3.2 NOISE CONTROL TECHNIQUES

When a train traverses an elevated structure, the interaction between the wheels and rails excites both the vehicle components (car body, trucks, and wheels) and the track and support structure components (rails, structure deck, supporting girders, and attached surfaces such as side walls and footways) setting them into vibration. The contribution of each of these vibrating components to the total radiated noise depends on the type of elevated structure.

Treatments for controlling elevated structure noise generally fall into one or more of the following categories [Kurzweil, 1977]; reduction of the noise output of the sources, vibration isolation, vibration damping, addition of mass, acoustic isolation, acoustic absorption, and minimizing the area of the

TABLE 3-1. Estimates of increase to A-weighted sound levels for various untreated* elevated structures compared to at-grade operation

Type of Structure	Increase***
Concrete or Composite (Concrete Deck, Steel Girders) Structure with Ballasted Track (XI, X)**	0-5 dBA
Concrete or Composite Structure with Ballastless Track (IX, VII)	5-10
Steel Structure with Ballasted Track (VIII)	5-10
Steel Structure with Ballastless Tie Deck (II-IV, VI) or Direct Fixation on Rail Bearers (V)	10-15
Steel Structure with Direct Fixation on Steel Plate Deck (I)	15-20

*By applying noise control treatments to these structures, the increase in levels from at-grade operation can be reduced.

**Roman numerals refer to structure groups from Figure 3-1.

***At 30 m (100 ft) from the track centerline, 1.2 to 1.6 m (4 to 5.25 ft) above the ground.

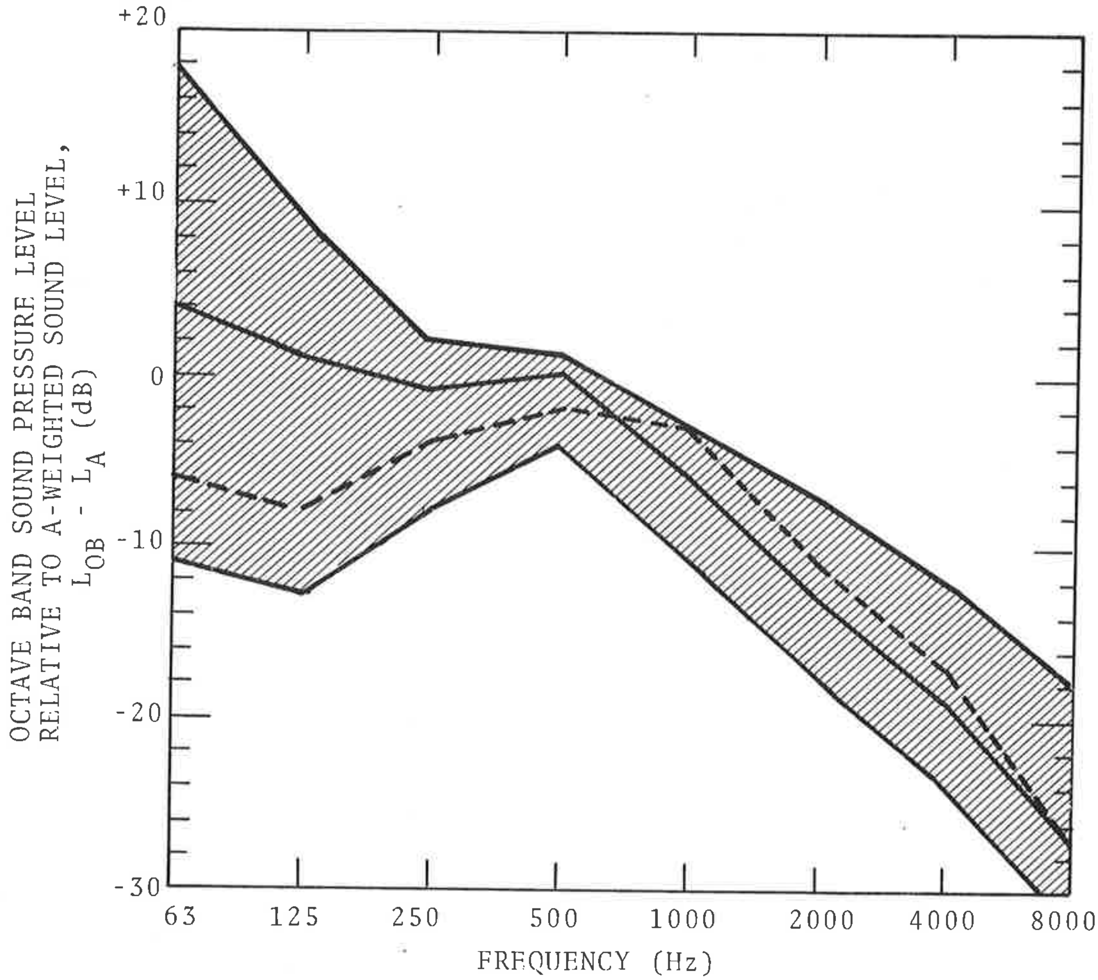


Figure 3-2. Average relative spectrum of sideline noise from elevated structures for trains passing at 60 km/hr (37 mph). The octave-band level minus the A-weighted sound level is plotted as a function of octave-band center frequency. ———, Average for 20 European railroad bridges; – – – – Average for 5 U.S. urban rail transit structures; ///// Range, includes 95% of all the data.

radiating surface. The addition of ballast, for example, may provide added mass, vibration damping, and acoustic absorption. The results of actual noise control applications of various treatments on a variety of existing elevated structures are summarized in Table 3-2 and are discussed below.

3.2.1 Source Reduction

The same treatments described above for reducing wheel-rail noise (see Table 2-1) generally are applicable for reducing the source of elevated structure noise.

3.2.2 Vibration Isolation

Vibration isolation is the dominant effect of the following treatments.

Resilient Rail Fasteners, Elastometric Pads (see Table 3-2, Structures Ib and IIIc). The reduction of structure-borne noise below 100 Hz by the use of elastic materials in track structures is impractical [Kurze, 1973]. For effective use of elastic track supports (without additional mass), the spring elements must compress at least 2 mm (0.08 in.) as the train passes over the track and must also compress considerably more than the deflection of the individual bridge components [Heckl, 1971]. This latter requirement usually is not met in light, steel elevated structures, or bridges. The internal damping of elastometric pads also provides damping to the rails.

Ballast Mats (see Table 3-2, Structures VIII and XI). Ballast mats are rubber mats, approximately 25 mm (1 in.) thick, which are placed under the ballast on ballasted elevated structures. They attenuate structure-borne noise from these structures and also reduce pulverization of the ballast, thus decreasing track maintenance costs [Sato, et al., 1974; Tajima & Kiura, 1974].

Resilient Rails (see Table 3-2, Structure IIIb). Field experiments [Satoh, et al., 1972] show that resilient rails reduce both wayside noise and structural vibration levels on a steel elevated structure. Such rails isolate vibration between the rail head and rail foot and also increases rail damping, which reduces the length of rail that vibrates and radiates.

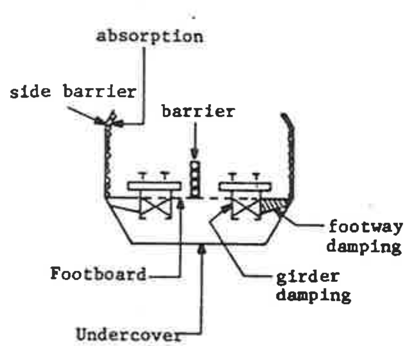
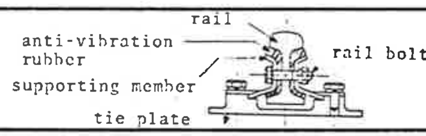
Isolation Between Structural Components. Isolation between structural components includes elastomeric pads between ties and girders (see Table 3-2, Structures IIIc and VI), isolation of concrete slab decks from steel supporting girders, isolation of non-load-carrying components such as footways or conduits, and isolation of certain noise control treatments such as barriers and undercovers (see Table 3-2, Structures IIIa, VI, and VIII).

3.2.3 Vibration Damping

When vibrational energy cannot be isolated from a given component, the vibration amplitude may be reduced by increasing the damping of the vibrating component. Treatments which take advantage of this effect include: resilient rails (see above), viscoelastic pads under rails and ties (see previous discussion of resilient rail pads and isolation between structural components), ballast (see Table 3-2, Structure Ia), and damping of structurally supporting and non-supporting members (see Table 3-2, Structures IIIa, VIIa, VIIb, and VIII).

Untreated steel bridges usually have loss factors on the order of 0.005. This must be increased to 0.05 to have a significant effect on the radiated noise level [Kurzweil, 1977]; it may be achieved either through use of constrained layer damping or addition of sand or ballast material. The feasibility of using the latter depends on whether the structure can tolerate the resulting weight increase.

TABLE 3-2. Summary of results from noise control applications on elevated railway and tracked transit structures

UNTREATED STRUCTURE TYPE (a)	DESCRIPTION OF NOISE CONTROL TREATMENTS	ΔL_A (b) dBA	COMMENTS
I a	Replacement of direct rail fixation by wood tie/ballast track.	13	Maximum change occurred between 125-1000 Hz [ORE,1966]
I b	1) Replacement of resilient rail fasterners with continuous elastic support under rail. 2) Replacement of resilient rail fasteners with "elastic ties" between rail and deck.	-7 0	The approximate stiffness per unit rail length of each fastening is 3.5 x 10^7, 1.5 x 10^8, and 2.5 x 10^7 N/m^2 (5100, 22000, and 3600 psi) for the original fastener, treatment 1, and treatment 2, respectively [ORE,1969&1971]
III a	 1) Sidewall (no absorption) + footboard below ties 2) Treatment 1 + damping of footway and girders 3) Treatment 2 + barrier between tracks + absorption on barrier and sidewalls + resiliently hung, high transmission loss undercover.	7 9 19	After treatment 3, the radiation is due to vibration of the sidewalls. This points up the need for isolating the walls from the structure [JNR,1975]
III b	 Replacement of standard rail with "resilient" rail	4-6 (c)	Noise spectra indicate little or no difference below 500 Hz [Satoh, et al, 1972]

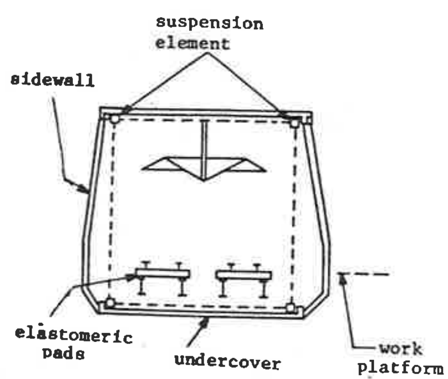
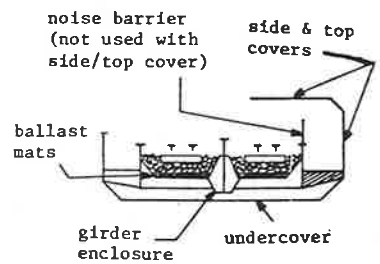
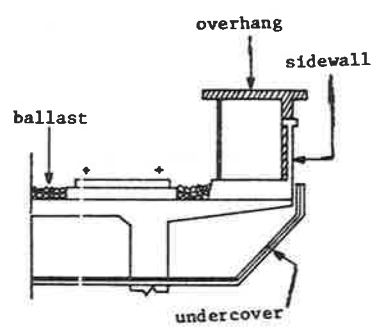
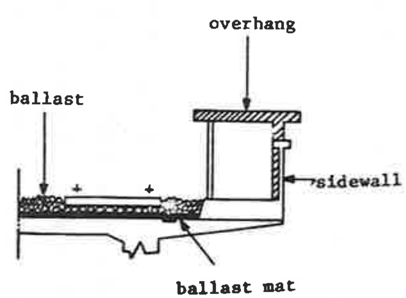
III c	1. Rubber pads between ties and girders	5	Treatment 1 uses 8mm (0.3 in.) thick rubber pads with ribbed profile [Berglund 1972] Treatment 2 uses two rubber pads separated by a steel plate Both methods have no effect below 400 Hz. [Lindholm, 1972]
	2. Rubber pads between rails and ties	5	
VI	 1) Work platform installed	3	Treatment 6 is usually impractical because of cost, increased difficulty of track maintenance, additional weight requirements for existing structures, signal jamming, and safety of running trains. [JNR, 1975]
	2) 1 + Elastomer impregnated pads under ties	3	
	3) 2 + undercover + 2m (6.5 ft) absorptive sidewalls	13	
	4) 3 with sidewalls raised to 4m (13 ft)	18	
	5) 3 with sidewalls raised to 6m (19.5 ft)	18	
	6) 5 with cover over top (complete enclosure)	28	
	7) 6 with work platform removed.	27	
VII a	1) Sidewalls (non-absorbing) + girder damping	4	Damping of girder resulted in 15 dBA reduction in girder acceleration level. A rubber asphalt damping material was used [JNR, 1975]
	2) 1 + absorptive material added to track and sidewall + damping of footway	10	

TABLE 3-2. Summary of results from noise control applications on elevated railway and tracked transit structures (Cont.)

UNTREATED STRUCTURE TYPE (a)	DESCRIPTION OF NOISE CONTROL TREATMENTS		ΔL_A (b) dBA	COMMENTS
VII b		1) Girder damping, applied in 3 layers, 3.2mm (0.125 in.) each 2) 1 + increased thickness of damping material to 16mm (0.675 in.) at selected locations.	3 11	Tests were performed on an 26 m (85 ft) long girder suspended on piers. The structure was excited by means of a pneumatic vibrator. The measured reductions therefore represent only the change in the structure-borne sound. If wheel/rail noise were present, the overall reduction might be less [Kirshner,1965;PBTB,1968]
VIII		1) 25 mm (1 in) thick ballast mats 2) 1 + noise barrier on side main girder 3) 2 + damping of side main girder and footway + enclosure of center girder below track 4) 3 + resiliently hung undercover + partial enclosure of side and top.	8 13 13 16	Treatment 1 results in a reduction of 22 dBA in the floor plate acceleration level and 10 dBA in the side girder acceleration level [JNR,1975]
IX a		1) 1.9m (6.2 ft) high sidewall (no absorption) 2) 1 + 2m (6.6 ft) absorptive overhang 3) 2 + undercover with interior absorption 4) 3 + ballast layer of .3m (1 ft) on slab	7 10 11 11	The TL of the undercover was given as 34 dB at 500 Hz. The major reduction for this structure occurred with treatments 1 and 2 indicating that the sideline noise levels are dominated by the wheel/rail and vehicle noise [JNR,1975]

IX b	1.2m (4 ft) high barrier with absorptive treatment on inner surface.	9	The barrier placement left a 0.2m (8 in.) gap between the barrier and the car body [PBTB,1968]
XI	 1) Non-absorbing sidewall, 1.9m (6.2 ft) high 2) 1 + ballast mat 3) 2.4 m (7.9 ft) high absorptive sidewall + ballast mat 4) 3 + 1.5m (4.9 ft) absorptive overhang 5) 3 + 2 m (6.6 ft) absorptive overhang	7 10 11 12 12	Because of the inherent mass and damping provided by the ballast and the concrete structure, the effect of the ballast mat (which reduces the structure-borne noise) is much less pronounced here than for structure VIII above. [JNR,1975]

(a) Refer to Figure 3-1 for description of structure types (identified by Roman numerals).

(b) $\Delta L_A = L_A$ (untreated structure) - L_A (treated structure). ΔL_A based on measurements 15 to 30m (50 to 100 ft) from the structure, and 1.2 to 1.6m (4 to 5.25 ft) above the ground, except as noted.

(c) This represents the range of reductions at several measurement points near the structure (including above, to the side, and below).

3.2.4 Addition of Mass

For a fixed amount of vibrational energy in a structure, increasing the mass results in a decrease in the vibration level, and hence, in the radiated sound power. This method of noise control can be accomplished by the use of ballast (see Table 3-2, Structures Ia, and IXa) or heavy isolated concrete decks. Additional mass is most effective when it is installed as close as possible to the excitation point, and when it is heavier than the moving masses. For steel structures the additional weight requires more expensive constructions.

3.2.5 Acoustic Isolation

The "shielding" of an observer from noise radiating from wheels, rails, ties or slab decks, girders, columns, etc., is only effective when the lines-of-sight from the primary radiating surfaces to the observer are blocked, for example, by barriers or partial enclosures (see Table 3-2, Structures IIIa, VI, VIIa, VIII, IXa, IXb, and XI). The use of sound absorptive materials on the source side of the barriers (i.e., facing the train) may be necessary to prevent an increase in noise in the train. It may also be necessary to provide resilient isolation between the structure and the barrier in order to prevent the barrier itself from becoming a major radiating surface. A method for predicting the attenuation of railway noise by barriers is presented in the next section.

3.2.6 Sound Absorption

Treatments which provide sound absorption include ballast or other absorptive material on the track (see Table 3-2, Structures Ia, VIIa, and IXa) and sound absorptive material on barrier surfaces (see above discussion on barriers).

The sound absorption coefficient of ballast depends mainly on the ballast depth, usually increasing with the increasing depth. For a layer 0.3 m (1 ft) deep, the effective sound absorption coefficient varies between 0.4 and 0.8 in the frequency range from 200 to 5000 Hz [Bolourchi, 1975].

Practical problems associated with many sound absorptive materials other than ballast include loss of effectiveness with contamination by dirt, oil or grease, difficulty in cleaning, and durability.

3.2.7 Minimizing the Area of the Radiating Surface

To minimize the radiation of noise, radiating surfaces should be kept as small as possible. Two-track bridges should be separate in the middle so that as a train passes on one side, only that half of the bridge acts as a noise source. The surface area of attached members (such as conduits and footways) should be kept to a minimum. Thus, a lattice girder, or truss bridge usually radiates less noise than all-steel bridges using plate girders (see Figure 3-1).

3.3 MEASUREMENT OF ELEVATED STRUCTURE NOISE

Recommended positions for the measurement of noise for rail vehicles on elevated structures are given in References [ISO, 1975; UITP, 1972a; ORE, 1969].

4. PREDICTING NOISE SOME DISTANCE FROM A PASSING TRAIN

Methods for predicting the maximum sound levels at 30 m (100 ft) from long trains of various types, traveling on at-grade and elevated track, are given in Sections 2 and 3. In this section methods are given for extrapolation to other distances for various length trains, for consideration of the effects of barriers, and for estimation of the duration of the pass-by noise. Noise propagation from railway yards and stationary facilities are not specifically addressed here, but have been treated in the literature [EPA, 1976; Swing & Pies, 1973; Bender, et al., 1974b].

The maximum A-weighted sound level L_A at a distance, d, from the track, due to a train of similar rail cars is:

$$L_A = L_A\ (30\ \text{m}) - C_s - G_{g+a} - C_b,\ \text{dBA} \qquad (4.1)$$

where

L_A (30 m) = the level at 30 m (100 ft) from a long train of these same railcars (a train is considered long when its length is at least three times the measurement distance) traveling on track of the desired type and condition (see Sections 2 and 3).

C_s = attenuation factor due to geometrical spreading of sound energy (also accounts for the actual train length).

C_{g+a} = excess attentuation due to both sound propagation near ground and absorption of sound energy by air.

C_b = excess attenuation due to shielding of the noise by barriers, cuttings, embankments, or houses.

For locomotives, and other localized noise sources (e.g., see Table 2-2), Equation 4-1 can be used with L_A (30 m) being the sound level 30 m (100 ft) from a single locomotive or other noise source (see Section 2).

4.1 GEOMETRICAL SPREADING, C_s

For the purpose of predicting the variation in noise levels with distance from a train of similar railcars several analytical models have been developed [Serendipity, 1970; Peters, 1974; JNR, 1973; Manning, et al., 1974]. The geometrical spreading correction, C_s, shown in Figure 4-1 assumes a line source model with dipole directivity. This graph may be used for distances from the track greater than $\frac{2}{\pi}$ times the average truck-to-truck spacing, $\bar{\ell}_t$. For typical trolley and rapid transit cars, $\frac{2}{\pi}\,\bar{\ell}_t$ = 5.5 m (18 ft) on the average (range 4.6-7.3 m, 15-24 ft); for long intercity passenger cars $\frac{2}{\pi}\,\bar{\ell}_t$ = 8.3 m (27 ft). For distances closer than $2\bar{\ell}_t/\pi$, the noise level increases by 6 dB per halving of distance.

Figure 4-1 also can be used to estimate the effect of geometrical spreading of noise from an elevated steel structure if 15 m (50 ft) is added to the length of the train to account for radiation due to vibrations of the structure beyond the ends of the trains. If the length of the train exceeds that of the bridge span, the latter should be used.

For a single locomotive the sound level decreases 6 dB per doubling of distance from the source for distances greater than 7.5 m (25ft) (based on data from Remington and Rudd, 1976). Thus, the spreading loss for a single locomotive is given by

$$C_s = 20\log(d/30) \quad d > 7.5 \text{ m (25 ft)} \qquad (4.2)$$

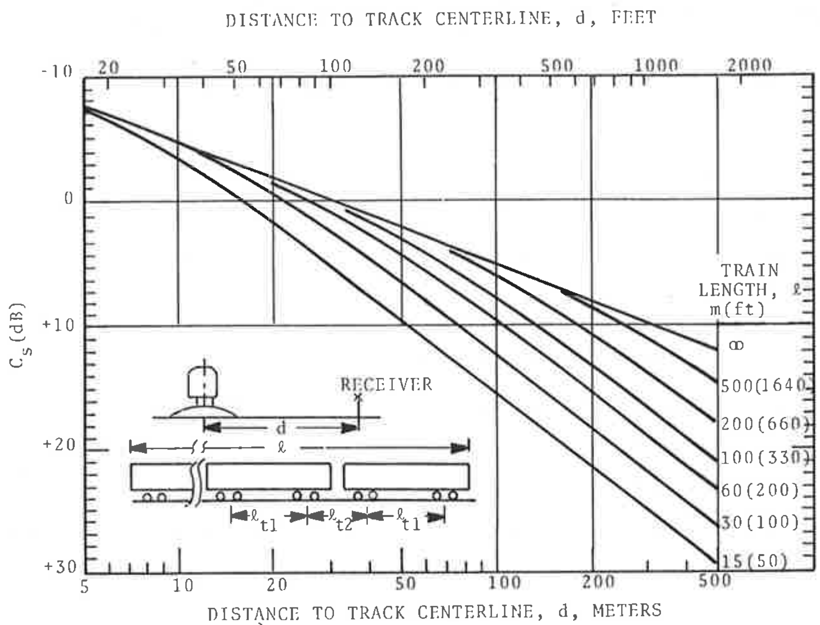


Figure 4-1. Attenuation due to geometrical spreading, C_s in Eq. 4.1. This figure gives the difference in level between an infinitely long train at 30m (100 ft), L_A(30m), and a train of specified length, ℓ, at distances $d > 2\bar{\ell}_t/\pi$ (where $\bar{\ell}_t$ is the average truck center spacing, $\bar{\ell}_t=(\ell_{t1}+\ell_{t2})/2$). For $d < 2\ell_t/\pi$, the noise level increases 6 dB per halving of distance.

To find the difference in levels, L_2-L_1, (due to geometrical spreading), between a train of length ℓ_1 at distance d_1 and a train of the same type and condition but of length ℓ_2 at a distance d_2:

Step 1. Read value of C_{s1} (= $L_{ref}-L_1$) at $d=d_1$ on the curve for $\ell=\ell_1$.

Step 2. Read value of C_{s2} (= $L_{ref}-L_2$) at $d=d_2$ on the curve for $\ell=\ell_2$.

Step 3. $L_2-L_1 = C_{s1}-C_{s2}$.

4.2 GROUND ATTENUATION AND AIR ABSORPTION C_{g+a}

Factors affecting the attenuation of sound propagation over flat ground include ground cover, temperature gradients, wind gradients, atmospheric turbulence, and height of source and receiver above ground. Absorption of sound energy by the air depends upon temperature and humidity as well as frequency and distance.

Figure 4-2 presents estimates of the total excess attenuation, C_{g+a}, due to both effects for railcars, diesel locomotives, and elevated structure noise spectra. These estimates are conservative; actual attenuation will often be greater. They are applicable for temperatures and humidities satisfying the relation, $4000 \leq (1.8C + 32)\,H \leq 8000$, where C is temperature in °C (note that the expression in parenthesis is simply the temperature in °F), and H is relative humidity in percent. Ground interference effects, which occasionally provide increased attenuation for frequencies between 125 and 500 Hz, are not accounted for in Figure 4-2. For noise sources other than those represented in Figure 4-2 or under other environmental conditions the reader may calculate C_g and C_a separately from the material in References Beranek, [1971], FAA, [1976].

Figure 4-3 shows excellent agreement between measured data for attenuation [Tubby, 1974] and the predicted attenuation including geometrical spreading, ground attenuation, and air absorption based on Figures 4-1 and 4-2.

4.3 BARRIER ATTENUATION C_b

Barrier corrections can be used to estimate the sound attenuation for a train traveling behind walls and hills, through cuts, and in some cases on embankments and elevated structures. Figure 4-4 defines the path difference δ for various configurations. The source location is at the axle height on the track centerline

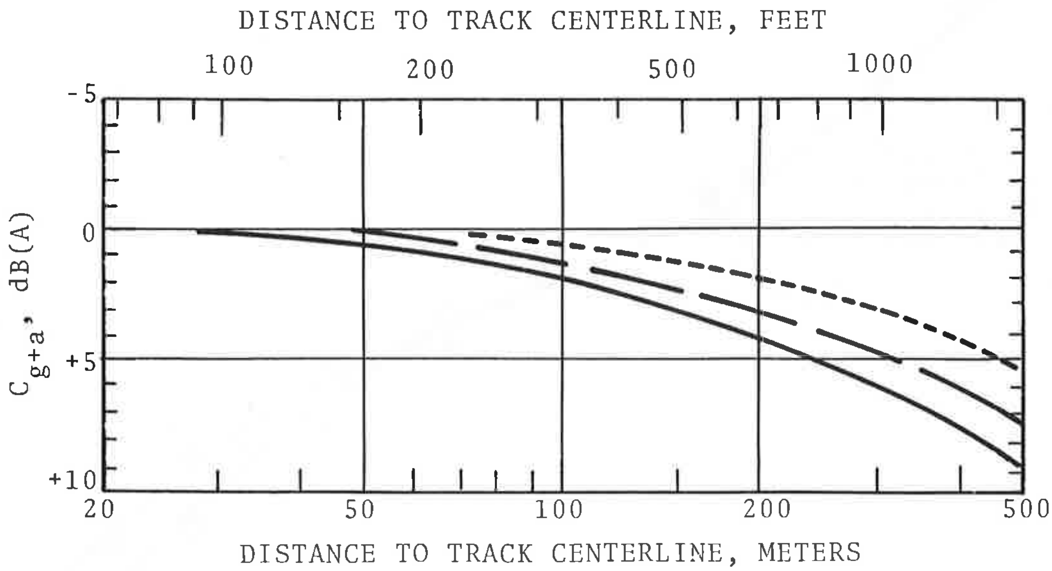


Figure 4-2. Excess attenuation C_{g+a} due to both ground effects and air absorption. The attenuation values are for positions downwind of the track, wind speeds between 3 and 50 km/hr (2 and 30 mph), ground cover less than .6m (2 ft) high, and for combinations of temperatures (°F) and relative humidities (%) whose product is greater than 4000°F%. Most other conditions will lead to higher attenuations.

——— Railcars (passenger, freight, transit) and electric locomotives

— — Diesel locomotives

------- Railroad bridges

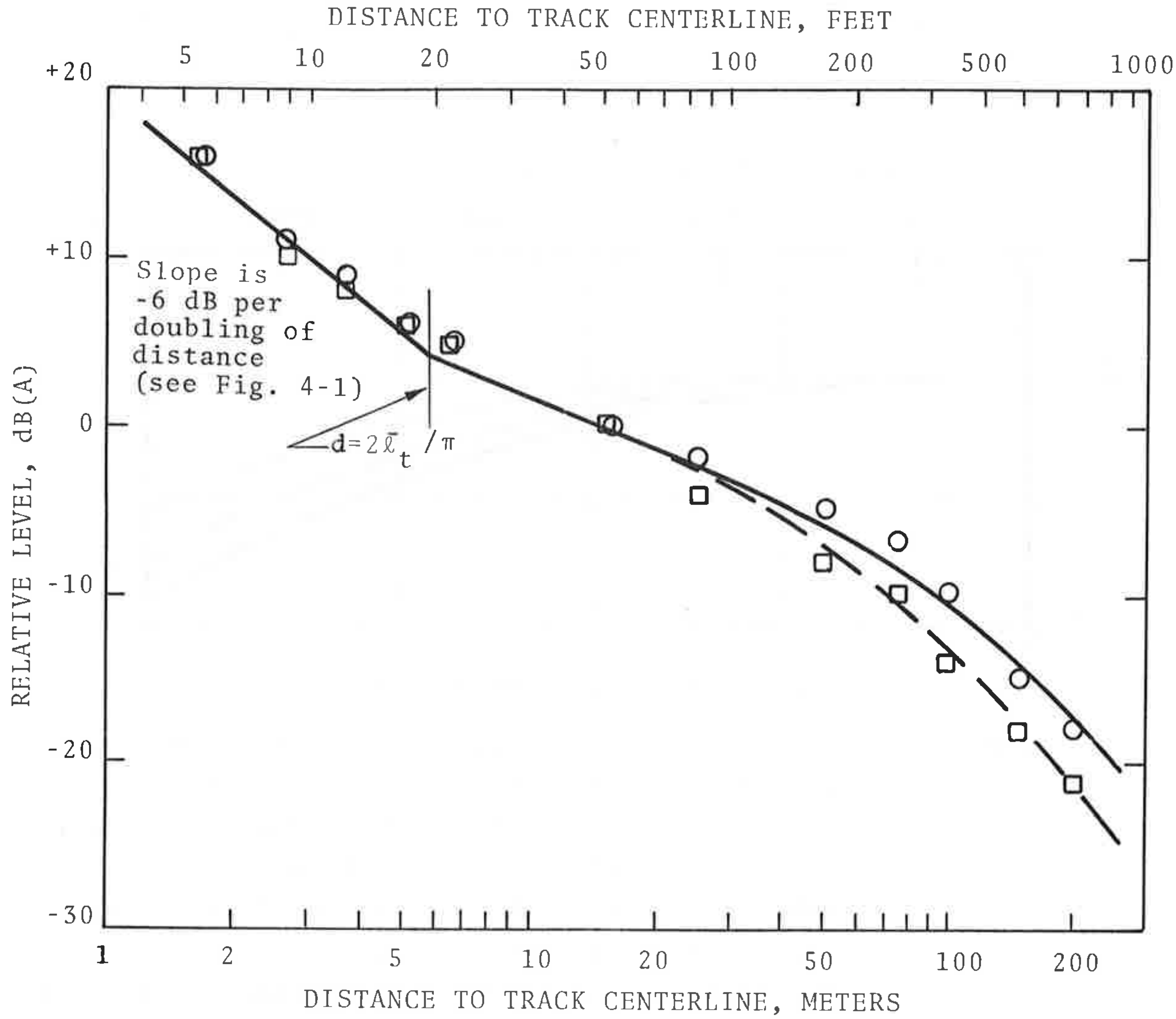
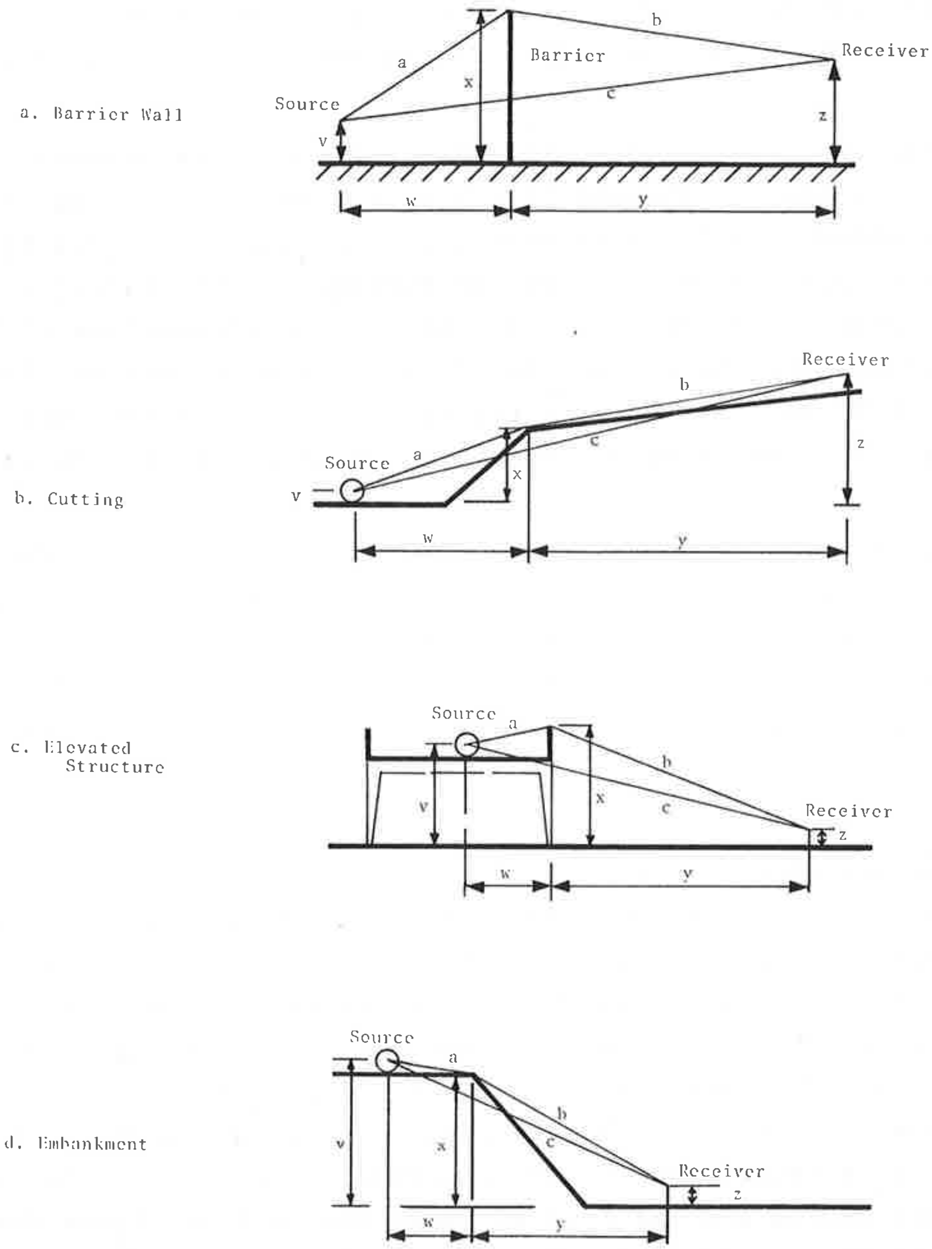


Figure 4-3. Total attenuation of A-weighted levels with distance from track centerline; theory versus experiment. Measured data [Tubby, 1974]; o, 12-car passenger train (ℓ = 240m); □ , 4-car passenger train (ℓ = 80m). Predicted curves based on measured levels at 15m (50 ft) and Figures 4-1 and 4-2: ———— = 240m; — — — ℓ = 80m.



Path difference, $\delta = a+b-c = \sqrt{w^2 + (x-v)^2} + \sqrt{y^2 + (z-x)^2} - \sqrt{(w+y)^2 + (z-v)^2}$

Figure 4-4. Definition of path differences for various barrier configurations. Adapted from [Gill, 1975].

for wheel-rail dominated noise and on top of the locomotive [~4.9 m (16 ft) above the rail surface] for diesel locomotive noise.

Using the theoretical solution for the barrier attenuation of sound from a long incoherent line source [Beranek, 1971 pg. 178], the A-weighted barrier correction term, C_b, was determined as a function of path difference for the average of the railcar noise spectra shown in Figure 2-2. For the barrier attenuation of locomotive noise, the theoretical solution for a point source [Beranek, 1971 pg. 178] was applied to a typical diesel locomotive noise spectrum. The resulting barrier attenuation curves are shown in Figure 4-5.

Note that the resulting insertion loss for a given path difference is greater for diesel locomotives than for railcars. However, because the effective source height for diesel locomotives is typically 4.4 m (14.5 ft) higher than that for railcars, diesel locomotives require a substantially higher barrier to achieve the same path difference.

4.3.1 Sound Barriers Walls

Sound barriers can provide significant reductions in wayside noise levels. When the side of the barrier facing the train is lined with acoustically absorptive (non-reflecting) material, the reduction in sound level can be taken directly from Figure 4-5 with the path difference defined as in Figure 4-4(a). For a reflecting barrier with a height greater than that given in Figure 4-6, noise is transmitted to the observer by two paths - one from the actual source and one from an image source (see Figure 4-6). To compute the wayside noise level, calculations must be carried out for each path and the resulting computed levels "added" according to Figure 4-7.

A number of field tests have been performed to determine the effect of both absorptive [Lang, 1976a; Willenbrink, 1975] and reflective [Lang, 1976a; Hemsworth, 1975] sound barrier walls on noise radiation from trains. In Figure 4-8, measured data are

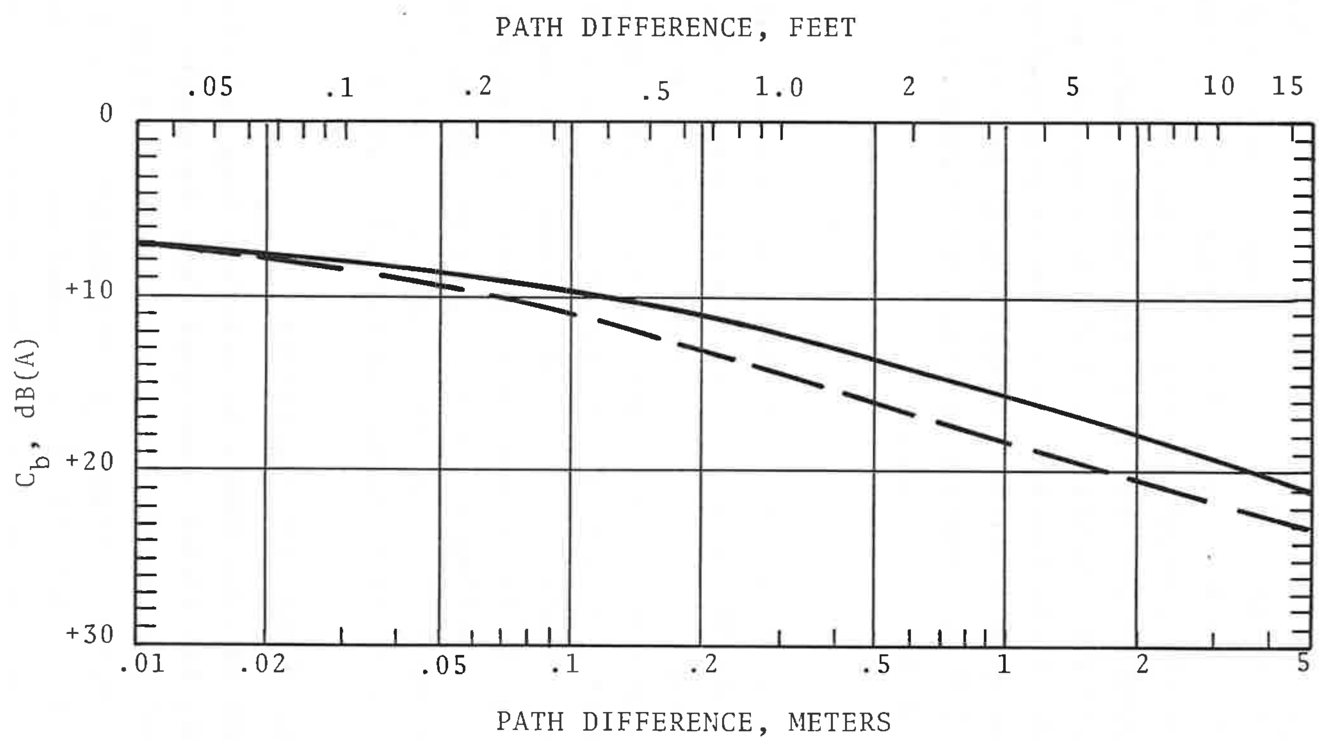


Figure 4-5. Barrier attenuation, C_b in Eq. (4-1), as a function of path difference, δ in Fig. 4-4.

——— Railcars (passenger, freight, transit) and electric locomotives

— — — Diesel locomotives

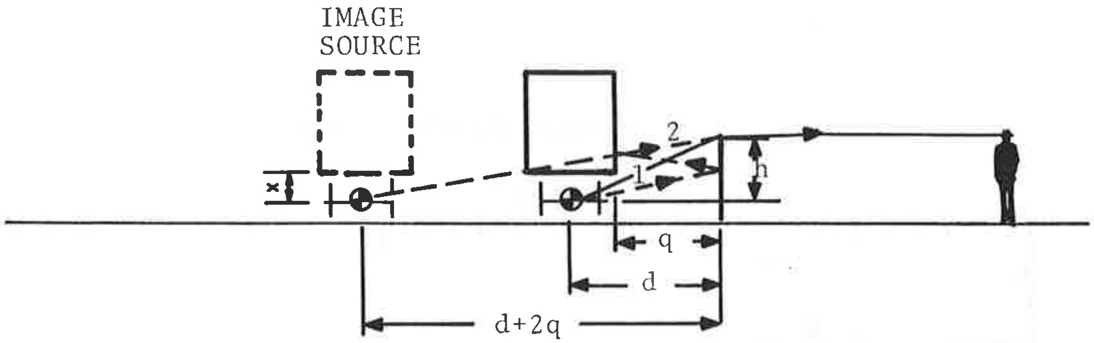
compared with the theory for railcar noise attenuation by barriers. When appropriate (see previous discussion and Figure 4-6) the affect of the image source, due to reflection off the barrier and the side of the car, has been mathematically removed from the experimental data in order to provide a correct comparison with the theory. The solid curve in Figure 4-8 is that given in Figure 4-5 and is based on the average spectrum from Figure 2-3. The dashed curve is based on the average of the actual spectra of the railcars used in the experiments. As might be expected, an improved prediction of the barrier correction is obtained by using the actual spectrum when it differs significantly from the average spectra in Figure 2-3 (particularly at the high frequencies, as was the case for the data in Figure 4-8).

4.3.2 Cuttings

The barrier corrections given in Figure 4-5 may be used for cuttings when the path difference is defined as shown in Figure 4-4(b). The corrections give the approximate difference in level between sound propagation over the cutting and direct propagation from source to receiver (distance c in Figure 4-4(b) over flat terrain with similar ground cover as the cutting. Measured cutting attenuation data is compared with theory in Figure 4-8. When the walls of the cutting are vertical, it may be necessary to account for the presence of sound energy that is reflected from the far wall of the cutting [Manning and Kurzweil, 1974].

4.3.3 Embankments and Elevated Structures

When trains are on embankments or concrete (non-radiating) elevated structures, the direct sound path from the wheels and rails may be blocked for locations close to the base of the embankment or structure (see Figures 4-4(c) and (d)). For distances further from the track, where the direct sound path is not blocked, little or no difference from at-grade operation has been found. Swing & Pies [1973] found that embankments up to 18 m (60 ft) above ground level resulted in the same noise levels as at-grade track for distances greater than 61 m (200 ft). A major effect



In order for reflected path (path 2) to exist:

$$h \geq x \left[\frac{d+2q}{d+q} \right]$$

Figure 4-6. Geometric criterion and location for image source due to reflection from barrier to carbody and back over top of barrier.

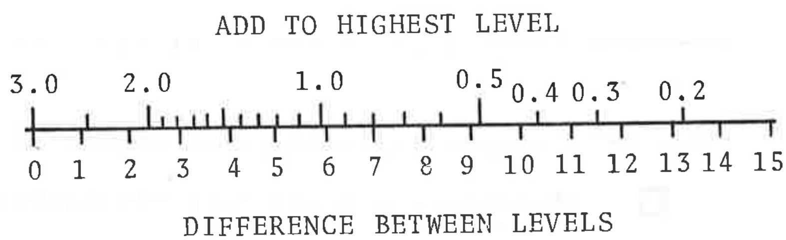


Figure 4-7. Nomograph for addition of noise levels in decibels.

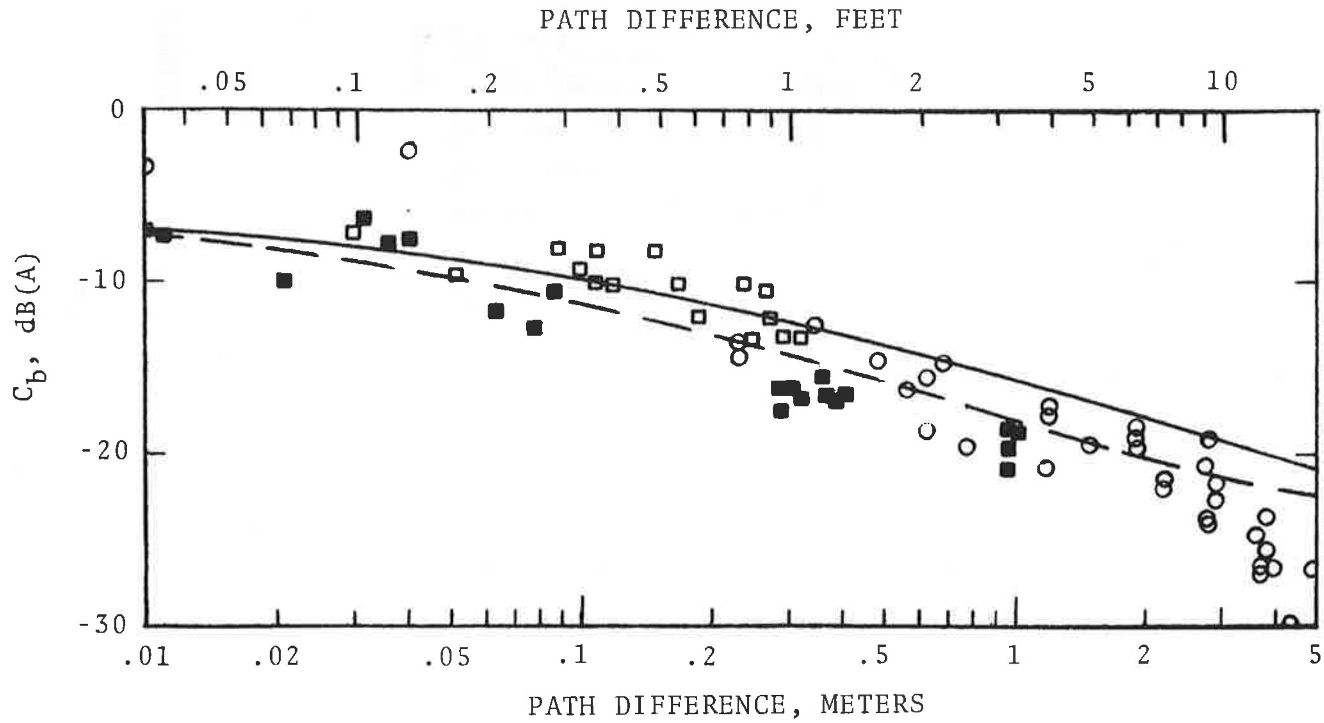


Figure 4-8. Comparison of measured and predicted (incoherent line source model) barrier corrections.

——— Theory using average railcar spectrum (Fig. 2-3)

— — Theory using average of actual test spectra

□ Absorptive barrier data [Willenbrink, 1975]

■ Reflective Barrier Data [Hemsworth, 1977], corrected for reflection

○ Cutting data [Glaretas, 1977]

of embankments is to reduce the path differences caused by houses (acting as barriers) close to the track, thus reducing the attenuation which the houses might have provided.

4.3.4 Propagation in Urban Areas

Data taken to determine the effect of one and two rows of attached two-story houses on railway noise have shown that the barrier theory used to generate Figure 4-5 does not strictly apply. An empirical relation based on data form Gill [1975] can be used to estimate the excess attenuation as a function of path difference for one row of attached houses for δ in the range 0.5 m (1.6 ft) < δ < 5 m (16 ft):

$$C_b \text{ (row of attached houses)} = \begin{cases} 10.2 + \delta \text{ (m)} \\ 10.2 + 3.3\ \delta \text{ (ft)} \end{cases} \qquad (4.3)$$

The values in Table 4-1 can be used when simple estimates of excess attenuation of railcar noise by various types of housing are desired.

4.4 DURATION OF TRAIN PASS-BY NOISE

Two measures of duration which have been used for computing various community noise ratings are the time during which the noise levels are within 5 and 10 dB of the maximum level, T_5 and T_{10}, respectively. Based on the theory for a dipole line source, approximations for T_5 and T_{10} are:

$$T_5 = \frac{\ell}{V}\left(1 + 1.2\frac{d}{\ell}\right) \text{ sec} \qquad (4.4)$$

$$T_{10} = \frac{\ell}{V}\left(1 + 2.3\frac{d}{\ell}\right) \text{ sec} \qquad (4.5)$$

d = distance from the track, m (ft)

ℓ = train length, m (ft)

V = train speed, m/s (ft/s)

The above equations agree with available data for distances up to $d = 2\ell$.

From the theory for a monopole point source, the effective duration T_{EL} (approximately equal to T_5), for the pass-by of a single locomotive is given by:

$$T_5 \text{ (loco)} \approx T_{EL} = \pi d/V \text{ sec} \tag{4.6}$$

TABLE 4-1. Excess attenuation by houses [Adapted from Fields, et al., 1976]

Type of House(s)	C_b, dB(A)
One row of detached or semi-detached two-story houses	8
Two or more rows of attached, two-story row houses (150 m long)	13
One row of attached, two-story row houses (300 m long)	15
Two or more rows of attached, two-story row houses (300 m long)	17

5. COMMUNITY RESPONSE TO NOISE

The noise rating procedures commonly used for judging community response to other types of transportation noise may be applied to rail transportation noise [EPA, 1973]. Peak sound levels as well as the number of times a train passes affect annoyance, thus supporting the use of noise rating measures such as L_{eq} (24 hr), L_{dn}, and CNEL (Community Noise Equivalent Level). These measures are defined below for rail transportation noise.

In addition to the noise produced by rail systems, an individual's annoyance can be affected by his attitude toward the neighborhood, the railway, and noise in general as well as by community factors such as ambient noise levels, time elapsed since the railway was built, and the visibility of the railway line [Fields, 1977]. Studies in France [Aubrée, 1973] and Japan [Nimura, et al., 1973] indicate that the correction factors used by EPA to normalize L_{dn} [EPA, 1974] for various communities and seasonals effects, can also be used to account for some of the community variables when rating annoyance due to train noise.

5.1 SINGLE-EVENT NOISE EXPOSURE LEVEL (SENEL)

The Single-Event Noise Exposure Level (SENEL) is often used as the basis for computing various noise exposure indices such as L_{eq}, L_{dn}, and CNEL. The total SENEL for a train, L_{AX}(train), is the energy sum of the railcar and locomotive generated SENEL's L_{AX}(car) and L_{AX}(loco) respectively:

$$L_{AX}(\text{train}) = 10 \log \left(10^{L_{AX}(\text{car})/10} + 10^{L_{AX}(\text{loco})/10}\right) \quad (5.1)$$

The railcar component is approximately

$$L_{AX}(\text{car}) \approx L_{AC} + 10 \log T_{EC} \quad (5.2)$$

where L_{AC} is the maximum A-weighted sound level resulting from car-generated (wheel-rail dominated) noise; T_{EC} is the effective duration, given in McShane, et al. [1975] and Wolfe, et al. [1976] is equal to T_5 (Eq. 4.4), and in EPA [1974] as 1/2 T_{10} (Eq. 4.5). For a passing rapid transit train, T_5 provides the better estimate.

The locomotive component L_{AX} (loco) due to a train traveling at speed V with n_L locomotives (including helper engines) is

$$L_{AX}(\text{loco}) \approx L_{AL} + 10 \log T_{EL} + 10 \log n_L \tag{5.3}$$

where L_{AL} and T_{EL} are the maximum A-weighted pass-by sound pressure level and effective duration (see Eq. 4.6) for a single locomotive.

5.2 NOISE RATING MEASURES

For any given type of train operation L_{eq} (24 hr), L_{dn}, and CNEL may be computed by the equation

$$\left.\begin{matrix} L_{eq}\ (24\ \text{hr}) \\ L_{dn} \\ \text{CNEL} \end{matrix}\right\} = L_{AX}(\text{train}) + 10 \log N - 49 \tag{5.4}$$

where L_{AX}(train) is the energy average L_{AX}(train) for the selected train type

$$N = \left\{\begin{matrix} N_d + N_e + N_n, & \text{for computing } L_{eq}\ (24\ \text{hr}) \\ N_d + N_e + 10N_n, & \text{for computing } L_{dn} \\ N_d + 3N_e + 10N_n, & \text{for computing CNEL} \end{matrix}\right.$$

and N_d, N_e, and N_n are the number of operations of the train type during the day (7AM-7PM), evening (7PM-10PM), and night (10PM-7AM), respectively.

The overall L_{eq}, L_{dn}, or CNEL for all train type operations is the energy sum of the respective levels (in decibels) from each train category.

5.3 ACCEPTABLE LEVELS FOR TRAIN NOISE

Guidelines for the maximum acceptable sound levels from urban rail systems, which have been developed by the North American urban rail transit systems, are shown in Table 5-1(A). These guidelines are based both on desired noise goals and the practicality of achieving various noise levels. The Japanese guidelines (Table 5-1(B)), which apply to commuter rail noise, are somewhat more stringent (about 5 dBA).

TABLE 5-1. Acceptable levels for airborne noise from train operations for dwellings and commercial buildings

A. U.S. SYSTEMS [APTA, 1976]

Area Category	Maximum A-Weighted Sound Level (for single event) Single Family Dwellings	Multi-Family Dwellings	Commercial Buildings
I. Low-density residential	70 dBA	75 dBA	80 dBA
II. Average urban residential	75	75	80
III. High-density urban residential	75	80	85
IV. Commercial	80	80	85
V. Industrial (or equivalent)	80	85	85

B. JAPANESE SYSTEMS [Hashimoto, 1975]

I. Residential areas	70
II. Commercial and industrial areas	75

6. GROUND-BORNE NOISE AND VIBRATION FROM RAIL SYSTEMS

The important aspects of ground-borne noise transmission from rail systems to nearby buildings are depicted in Figure 6-1. Vibration is generated by track-train interaction and transmitted to the tunnel structure and then to the surrounding soil or, in the case of at-grade track, directly from the track to the soil. The vibration propagates through the soil to adjacent buildings, resulting in vibration of the floors and walls and secondary radiation of noise into the rooms. The secondary radiation, due to structure-borne noise from a passing train, usually can be heard before the vibration levels are high enough to be felt.

6.1 GROUND-BORNE NOISE FROM TUNNELS - A SIMPLE ESTIMATE

Accurate prediction of the structure-borne noise and vibration in buildings near subway tunnels is not possible. A simple, approximate relation for the A-weighted sound level L_{AR}, in cellar rooms between 1 and 20 m (3 and 65 ft) from a subway wall is [Lang, 1971]:

$$L_{AR} = 59-20 \log \frac{R}{R_o} \pm 10 \text{ dBA} \qquad (6.1)$$

where R is the distance in meters or feet from the tunnel wall to the building wall and R_o a reference distance of 1 m (3.28 ft). The data upon which Equation 6.1 is based represents a wide range of vehicle types, conditions, and speeds; track types and conditions; tunnel and building constructions; and soil types. In general, systems with poor wheel and rail conditions and stiff track fastening designs will produce higher levels than systems with smooth wheels and rails, tie-and-ballast trackbed, or resilient direct fixation fasteners. Also, for a given subway system, given soil conditions and distance from building to subway tunnel,

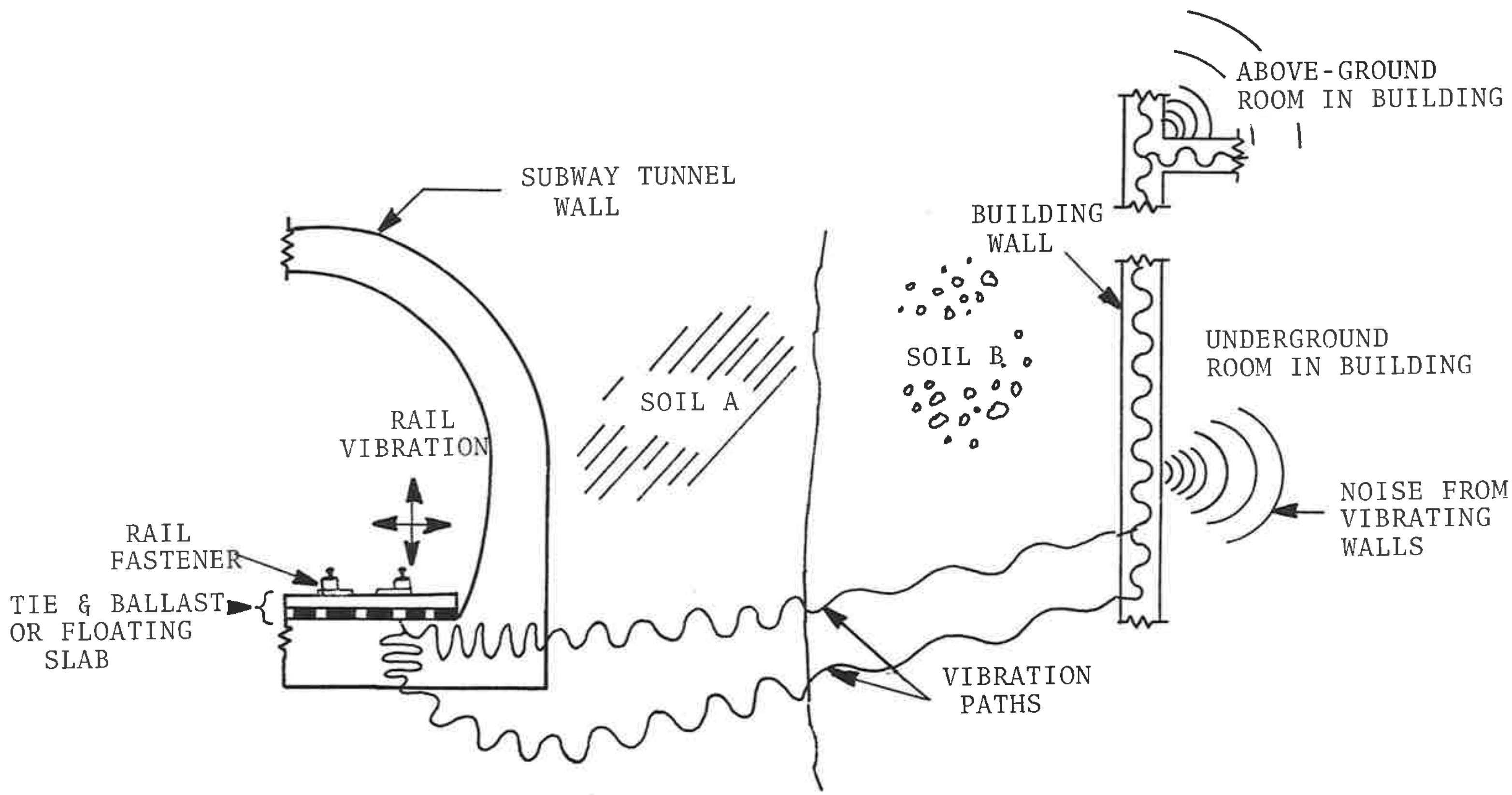


Figure 6-1. Schematic view of propagation of subway vibrations into buildings. After [Unger and Bender, 1975]

levels in lightweight frame buildings and buildings on slab foundations will be higher than those in masonry buildings on piles or spread footings.

6.2 GROUND-BORNE NOISE FROM TUNNELS - PARAMETRIC EFFECTS

In contrast with the simple estimate above, the following material takes into account the dynamic properties of the intervening structures and soils through which the vibration propagates. This permits estimation of changes in noise and vibration for various design and operating alternatives. The vibration acceleration level, L_a, discussed below refers to the maximum level during a train pass-by and is defined as

$$L_a = 20 \log \left(\frac{\text{rms octave band acceleration}}{10^{-6} \text{ g(rms)}} \right) \text{dB.}$$

The floor vibration level L_a (room), due to train-generated ground-borne vibration in a room of a building near a subway can be estimated from:

$$L_{a(\text{room})} = L_a \text{ (tunnel wall)} - C_{gt} - C_{cl} - C_{bt} \qquad (6.2)$$

$$\text{dB re } 10^{-6} \text{ g (rms)}$$

where

L_a (tunnel wall) = octave band acceleration on the wall of a subway tunnel during a train pass-by.

C_{gt} = the vibration attenuation due to propagation through the ground.

C_{cl} = the vibration attenuation (coupling loss) between the ground and the building

C_{bt} = the vibration attentuation due to propagation in the building.

6.2.1 Tunnel Wall Vibration Levels, L_a(tunnel wall)

Figure 6-2 shows tunnel wall acceleration spectra representing a range of levels measured in earth-based concrete tunnel structures for both ballasted and direct fixation rail fastening systems (not floating slab track). The data are for vehicles traveling at 60 km/h (37 mph) on continuous welded rail. In general, lower levels are for systems with smoothly ground wheels and rails, and ballasted trackbed for fairly resilient direct fixation rail fasteners.

The levels in rock-based tunnels are typically 6 dB lower than those shown in Figure 6-2. In both earth- and rock-based tunnels, tunnel floor vibration levels average 7 dB greater than the wall levels; however, measured differences between floor and wall levels have ranged from 2 to 15 dB.

6.2.2 Factors Influencing Tunnel Vibration Levels

Parameters which affect tunnel vibration and hence structure-borne noise in nearby buildings include:*

Train Speed. For speeds in the range 24-113 km/h (15-70 mph), a doubling of train speed results in a 4 to 6 dB increase in tunnel and ground vibration levels.

Axle Load. A doubling of the axle load results in a 2 to 4 dB increase in tunnel wall vibration levels, independent of train speed and track design [Hauck, et al., 1972]

Carbody Suspension. Tunnel vibration levels are independent of the type of carbody suspension system for the suspensions used in modern rail transit vehicles [Stuber, 1975a].

*Except where indicated, the effects are uniform over the frequency range shown in Figure 6-2.

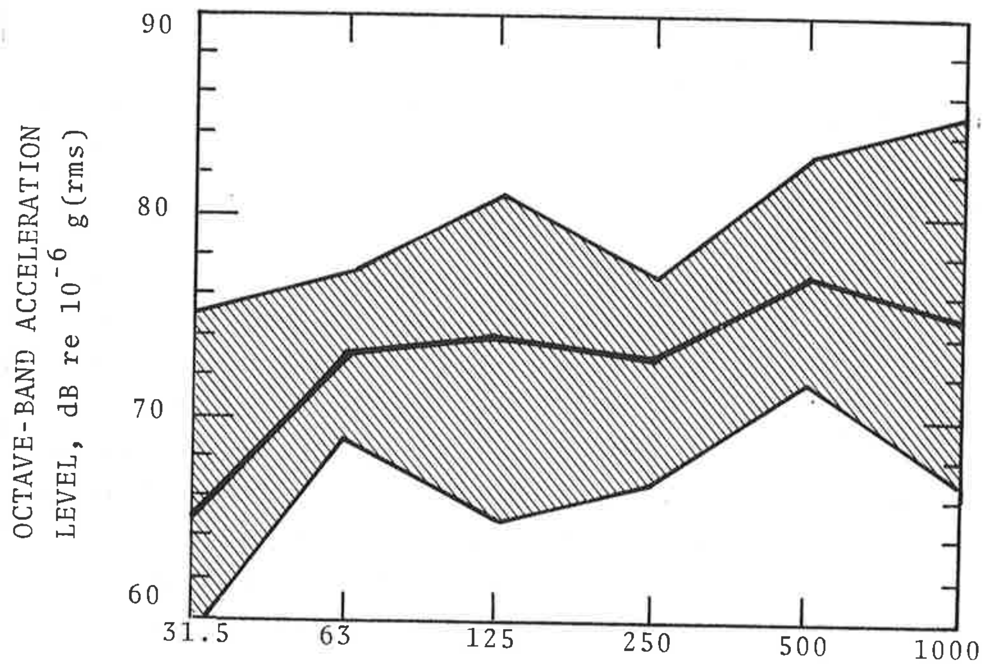


Figure 6-2. Tunnel wall vibration spectra, L_a (tunnel wall) in Eq. 6.2, for concrete earth-based tunnels. Trains traveling at 60 km/h (37 mph) on continuously welded rail.

——— Average of 15 spectra

▨ Range includes at least 90% of the data

Resilient Wheels. Tunnel vibration levels in the frequency range 40 to 250 Hz can be reduced about 4 dB by the use of resilient wheels [TTC, 1976].

Unsprung Mass. A halving of the unsprung mass results in a 6 dB reduction in ground vibration level [TTC, 1976]. The unsprung mass is the portion of the truck mass that is not isolated from the rail. It may include wheels and axles, and, unless isolated, the gear box assembly and traction motors.

Wheel and Rail Conditions. Poor maintenance of wheels and rails can result in levels significantly higher than for systems with smooth wheel and rail running surfaces and well-balanced wheels. Wheel flats, loose rail joints, or corrugated rail can increase vibration levels by 10 to 20 dB. Wheel or rail roughness on jointless uncorrugated rail and in the absence of wheel flats can result in 3-10 dB increase in the tunnel vibration levels [Wilson & Murray, 1974; Hauck, et al., 1973]. Roughness (without large corrugations) does not increase levels significantly on jointed track or when wheel flats are present [Manning, et al., 1974].

Special Trackwork. Subway structure and ground vibration levels increase 10 to 15 dB at track crossovers and turnouts. This increase has little effect on the spectrum and is independent of structure type [Wilson, 1971a].

Resilient Rail Fasteners (Including Resiliently Supported Ties). The rail support modulus K (equal to fastener stiffness divided by fastener spacing) directly affects the vibration transmission to the tunnel structure. For support moduli below about 1.4×10^8 N/m^2 (20,000 $lb/in.^2$), the tunnel vibration levels are proportional to 20 log(K) for frequencies above about 50 Hz [Bender, et al., 1969; Paolillo, 1973; Wilson & Murray, 1974; TTC, 1976]. Resilient fastener design considerations are discussed in detail in WIA, [1974], and Bender, [1974a]. When track stiffness is such that the deflection of the rail under the static train load is between 2 mm (0.08 in.) and 5 mm (0.2 in.), good vibration isolation, improved ride comfort, and reduced wear on train and track systems will result [Lang, 1976b].

Floating Slabs. A method for achieving substantial reductions in vibrations transmitted to tunnel structures is by the use of resiliently supported trackbed slabs (so-called "floating slabs")--either continuous cast-in-place slabs up to about 21 m (70 ft) long, or a series of precast slabs 0.7 to 1.5 m (2 to 5 ft) long [Wilson, 1970; Wilson & Murray, 1974; Bender, 1974b; Manning, et al., 1975b; TTC, 1976]. Measured reduction of tunnel vibration due to floating slabs of several designs is shown in Figure 6-3.

Ballast Depth. Variation in ballast depth under the tie between 30 and 70 cm (1 to 2.3 ft) has no effect on tunnel wall vibration levels [Stuber, 1975a].

Ballast Mats. Rubber mats placed between the ballast and the tunnel invert reduce tunnel vibration levels. The results of two applications of rubber mats are shown in Figure 6-4. In the Tokyo subway the deflection of the rail under a static train load amounted to about 1.2 mm (0.05 in.) for the ballasted track without mat, and 3 mm (0.12 in.) with the ballast mat in place [Fujiwara, et al., 1974].

Tunnel Construction. The type of tunnel structure and its mass influence the tunnel and ground-borne vibration levels. Table 6-1 indicates the relative levels to be expected for various structure types for earth-based subways.

Tunnel wall thickness has a significant effect on tunnel vibration levels. A doubling of the average wall thickness using the same material can lead to reductions in the wall vibration levels of 5 to 18 dBA [Stuber, 1975a; Hauck, et al., 1972 & 1973; Kazamaki & Watanabe, 1975; Koch, 1974].

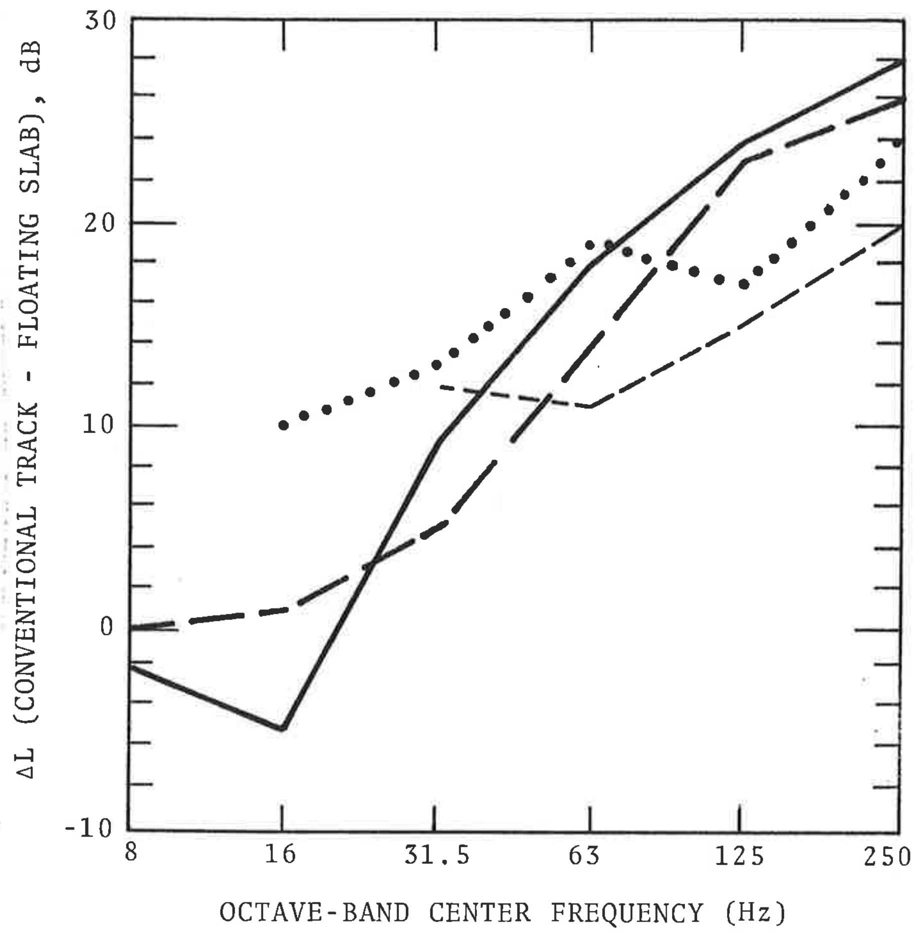


Figure 6-3. Difference in tunnel vibration levels between conventional track and floating slab track.

———— New York (NYCTA) continuous concrete slab, support frequency $f_o \approx 16$ Hz. [Manning, et al., 1975b]

— — — Washington, D.C. (WMATA) continuous concrete slab, $f_o \approx 16$ Hz. [Wilson, 1976]

--------- Cologne continuous concrete trough containing a conventional tie/ballast track, $f_o \approx 10$ Hz. [Hauck, et al., 1972]

.... Frankfort discontinuous precast concrete "slablets", $f_o \approx 10$ Hz. [Hauck, et al., 1973]

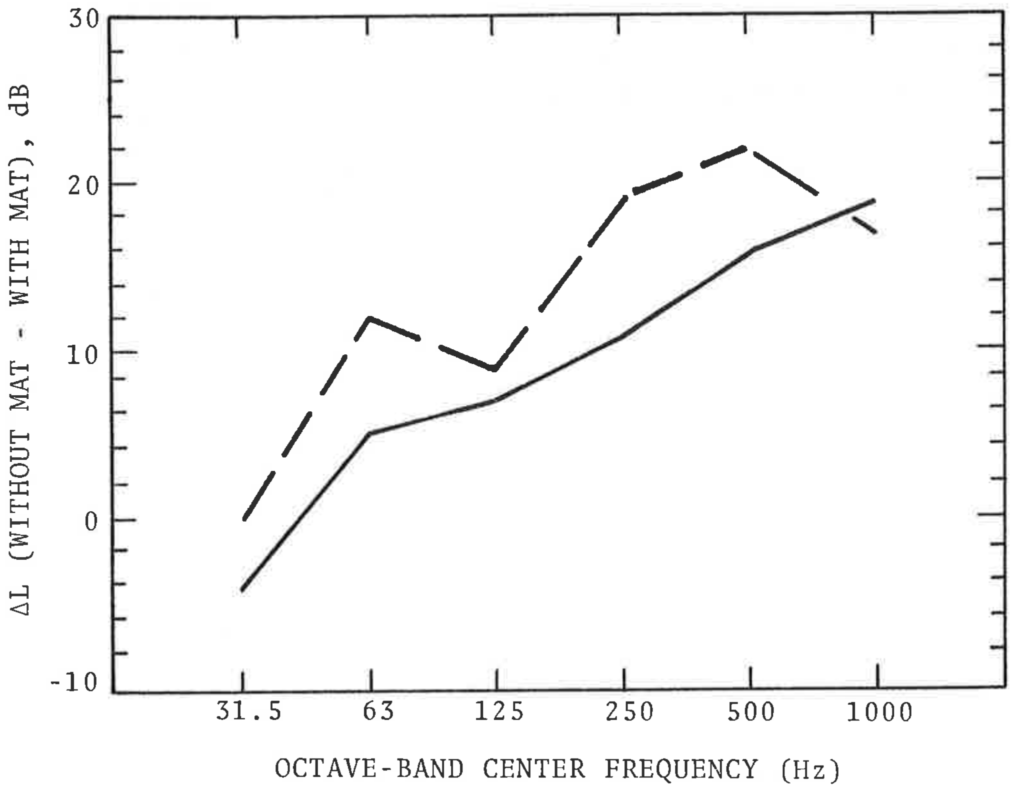


Figure 6-4. Difference in tunnel vibration levels between ballasted track without ballast mats and with ballast mats.

——— Triple rubber mat, average reduction for floor and wall [Hauck, et al., 1973]

— — — 2-layer mat, made from the tread section of used rubber tires, level reduction on tunnel wall [Fujiwara, et al., 1974]

TABLE 6-1. Effect of subway structure type on tunnel vibration levels for earth-based subways [Wilson, 1971a]

Subway Structure	Relative Vibration Level, dB
Cast Iron or Steel Single Tunnel	+4
Concrete Single Tunnel or Box	+2
Double Box	0
Triple Box	-2
Station	-4

6.2.3 Vibration Attenuation in the Ground, C_{gt}

Vibration propagation in soil depends on vibration frequency, soil type and strata interfaces, water content, and distance [Wilson, 1971a; Ungar & Bender, 1975; TTC, 1976; Manning, et al., 1974].

Figure 6-5 (adapted from Wilson, [1971a]; similar to the adaptation used in Manning, et al., [1974] shows the attenuation in ground vibration levels C_{gt} (soil), due to propagation in an "average" soil, as a function of frequency for various distances from the wall of an earth-based tunnel. To obtain an estimate of the vibration spectrum in soil at a given distance from the tunnel wall, add the correction from Figure 6-5 to the octave-band levels selected from Figure 6-2.

In rock the attenuation of vibration is small and due mainly to geometric spreading of the vibration. Geometric attenuation for distances up to about one half the train length is given by Ungar & Bender, [1975] as:

$$C_{gt}(\text{rock}) = 10 \log_{10} \frac{R_o + X}{R_o} \text{ dB} \tag{6.3}$$

where R_o denotes the distance from the tunnel center to the outer wall surface and X the distance from that surface to an observation point.

6.2.4 Coupling Loss Between Ground and Building, C_{cl}

The difference between the vibration level of the soil and that of the building support structure is defined as the coupling loss. For lightweight frame buildings and slab foundations on grade the coupling loss, C_{cl}, is 0 dB; for heavy masonry buildings on spread footings or piles, $C_{cl} = 15 \pm 5$ dB [Wilson, 1971a]. If a building or its footings are very close to (or attached to) the

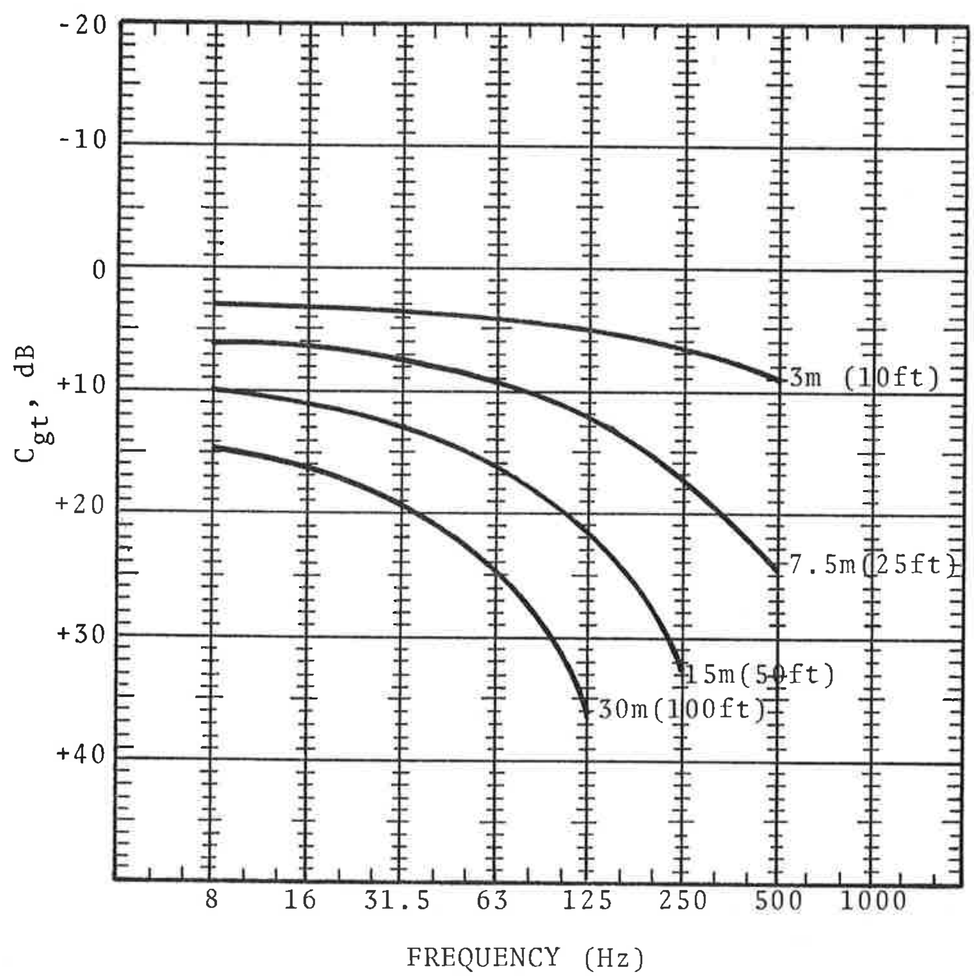


Figure 6-5. Vibration attenuation due to propagation through the ground (C_{gt} in Eq. 6.2). [Adapted from Wilson, 1971a]

tunnel structure, use of resilient material between the building and the structure can introduce a coupling loss of 10 to 20 dB, otherwise the coupling loss is zero. In such cases resilient isolation pads between the building columns and footings or support piles may also be necessary [WIA, 1971]. When the building is supported directly on rock, $C_{cl} = 0$.

6.2.5 Vibration Attenuation in a Building, C_{bt}

Estimates of the decrease in vibration levels (and hence sound pressure levels) with increasing height in heavy (masonry) buildings range from 1 dB/floor [Lang, 1976b] to 3-4 dB/floor [Ungar & Bender, 1975; Wilson 1971a]. A value of 3 dB/floor, starting at the ground level, appears to be typical, although for lightweight constructions no decrease with height is found. In some cases, vibration levels on the upper floors of lightweight buildings can be amplified due to floor resonances.

6.2.6 Resulting Structure-Borne Noise in Buildings

Vibration of the floors and walls of a room cause air-borne noise to be radiated into it. Equation 6.4 provides a reasonable estimate of the relation between the octave-band room floor acceleration level [L_a (room) in Equation 6.2] and the resulting octave-band sound pressure level in the room, L_p (room):

$$L_p \text{ (room)} = L_a \text{ (room)} - 20 \log_{10} f + 37, \text{ dB} \qquad (6.4)$$

where f is the octave-band center frequency. More detailed discussion of the relationship between L_a (room) and L_p (room) is given in references TTC, [1976]; Manning, et al., [1974]; Bender, et al., [1969]. The most significant octave-bands for A-weighted noise in buildings from ground-transmitted vibrations due to subway trains are those centered at 31.5, 63, and 125 Hz.

6.3 GROUND-BORNE VIBRATION FROM SURFACE (NON-TUNNEL) TRACK

6.3.1 Ground Vibration Levels Near At-Grade Track

Figure 6-6 shows vertical and lateral octave-band ground acceleration levels at a distance of 7.5 m (25 ft) from the center-line of track at grade level. Data from locomotives, passenger trains, freight cars, and rapid transit trains traveling on both continuously welded rail and bolted jointed rail have been normalized to a train speed of 60 km/h (37 mph) assuming a 6 dB increase in vibration level per doubling of train speed. This assumption is based on a typical speed dependence for rapid transit trains in tunnels. The overall acceleration levels at 7.5 m (25 ft) for freight cars (for which spectra were not available) are typically 6 dB below those for locomotives [Rickley, et al., 1973]. Data from light rail (tram) vehicles [Wilson, 1971b] measured on a <u>paved</u> <u>roadway</u> about 7.5 m (25 ft) from the track centerline yield vertical octave-band levels ranging from the same level in the 16 Hz band to 28 dB greater in the 250 Hz band than the levels shown in Figure 6-6. There higher levels are most likely due to lower vibration attenuation by the pavement (compared to the ground) rather than being due to differences between the rail vehicles involved.

6.3.2 Ground Vibration Levels Near Elevated and Depressed Track

Ground vibration levels near track on embankments and in cuttings are approximately the same as for track at grade. The levels near elevated guideway structures [Kobayashi, 1974] are also about the same at 7.5 m (25 ft) as for track at grade.

6.3.3 Vibration Propagation from Surface Track

The reported results of attenuation of vibration levels along the surface of the soil do not fall into any simple pattern [Manning, et al., 1974]. For rough estimating purposes, the vibration attenuation in the dominant frequency bands (from 16 to 125 Hz), an attenuation rate of 4 dB per doubling of distance beyond 7.5 m (25 ft) can be used for track at-grade, on embankments, and in cuttings. This attenuation is based in part

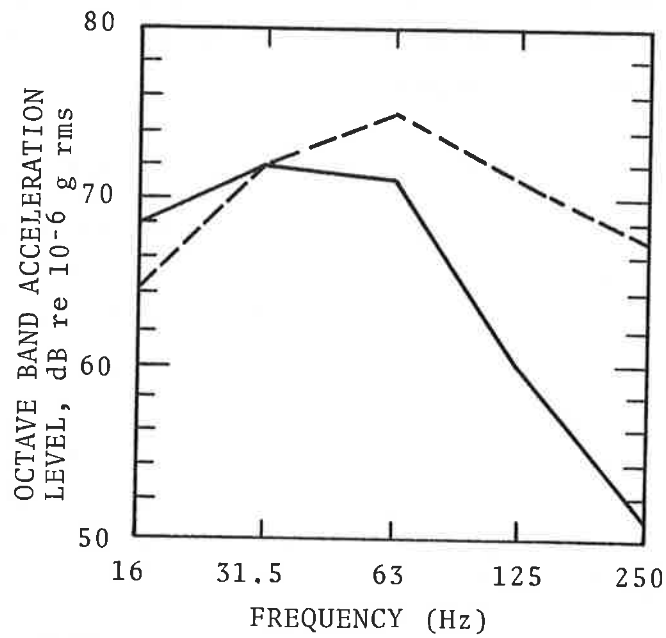


Figure 6-6. Ground surface acceleration spectra near at-grade track. Data taken 7.5 m (25 ft) from at-grade tie and ballast track, normalized to a train speed of 60 km/hr (37 mph). All data falls within ±6 dB of the spectra shown except in the 250 Hz band.

———————Vertical, average of 15 spectra

---------Horizontal (perpendicular to track direction), average of 14 spectra.

upon median ground vibration levels from Kobayashi, [1974] for distances up to 50 m (164 ft). Data from Shibata & Matsuhisa, [1975] also exhibits an attenuation rate of about 4 dB per distance doubling. For track on elevated guideways, the attenuation rate out to about 30 m (100 ft) is 7 dB per doubling of distance, probably due to point source excitation of the ground by column footings. Beyond 30 m (100 ft), the attenuation rate becomes 4 dB per distance doubling.

6.4 GROUND-BORNE NOISE AND VIBRATION CRITERIA AND MEASUREMENTS

Ground-borne noise criteria are presented in Table 6-2 in terms of the maximum A-weighted sound levels in buildings. These criteria were taken from guidelines specifically designed for new urban rail systems [APTA, 1976].

Standards for human exposure to vibration and vibration criteria for structural damage to buildings are shown in Figure 6-7. These standards are based on 1 Hz bandwidth. To convert the standard shown for human vibration to other bandwidths see, for example, Beranek [1971, p. 43].

The standard shown for human vibration exposure is the International Standards Organization (ISO) combined (vertical and horizontal motion) standard for continuous or intermittent vibration. To comply with the standard, neither vertical not horizontal vibration levels may exceed these limits. This combined standard is included here because it is recommended "for preliminary investigations to decide whether a vibration problem exists" (especially when the direction of vibration is not specified) [ISO, 1977]. When estimating the possibility of damage to buildings, the ISO suggests that peak velocity measurements be taken because velocity, as opposed to acceleration or displacement, is most easily related to damage. In the case of continuous or intermittent floor vibration, the peak vertical velocity at the location of highest amplitude should be determined (usually at the midspan of the floor). The levels shown in Figure 6-7 are the ISO damage criteria for continuous floor vibration. For intermittent

TABLE 6-2. Criteria and design goals for maximum ground-borne noise from train operations [APTA, 1976]

A. RESIDENCES AND BUILDINGS WITH SLEEPING AREAS

Community Area Category		Maximum Single Event Ground-borne Noise Level		
		Single Family Dwellings	Multi-Family Dwellings	Hotel/Motel Buildings
I	Low Density Residential	30 dBA	35 dBA	40 dBA
II	Average Residential	35	40	45
III	High Density Residential	35	40	45
IV	Commercial	40	45	50
V	Industrial/Highway	40	45	55

B. SPECIAL FUNCTION BUILDINGS

Type of Building or Room	Ground-borne Passby Noise Design Goal
Concert Halls and TV Studios	25 dBA
Auditoriums and Music Rooms	30 dBA
Churches and Theatres	35 dBA
Hospital Sleeping Rooms	35-40 dBA
Courtrooms	35 dBA
Schools and Libraries	40 dBA
University Buildings	35-40 dBA
Offices	35-40 dBA
Commercial Buildings	45-55 dBA

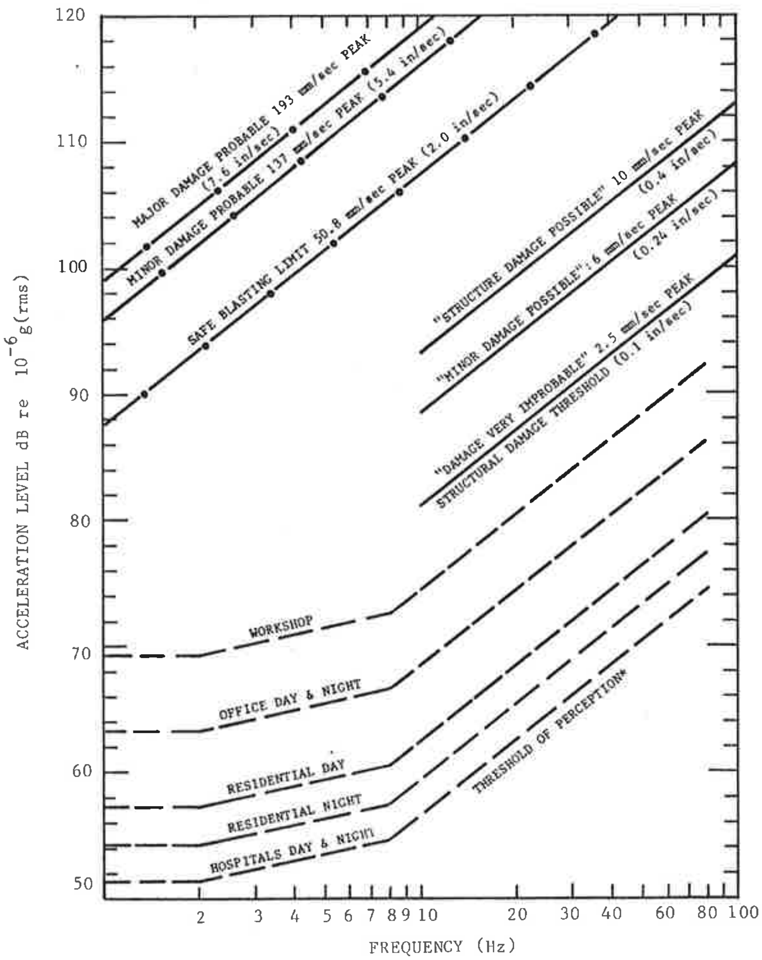


Figure 6-7. Vibration criteria for human response and structural damage to buildings

— — — ISO combined (3 axes) standard for vibration and shock limits for occupants in buildings [ISO, 1977]

——— ISO evaluation of damage associated with continuous building floor vibration [ISO, 1976]

—·— U.S. Bureau of mines blasting criteria [Nicholls, et al., 1971]

*Approximation of ISO perception threshold for floor vibration [ISO, 1976] vibration such as a train pass-by, the velocity values should be doubled (i.e., acceleration levels raised 6 dB) because intermittent vibration is not as damaging.

Present measurement practice is to measure the vibration levels perpendicular to both the tunnel floor and the tunnel wall and to measure the vertical vibration of the rail (as an indication of the uniformity of vehicles producing the vibration). On the ground surface above, tunnel measurements are generally made above either the structure centerline or track centerline.

For surface track, common measurement locations used in North America are 7.5 and 15 m (25 and 50 ft) from the track centerline, whereas in Germany [VDI, 1973], the same distances are used but are measured from the edge of the roadbed.

For measurements on the soil surface, VDI [1973] recommends attaching the vibration pickup to a pointed angle iron (45 mm x 45 mm x 5 mm thick - 1.8 in. x 1.8 in. x 0.2 in.) which is driven at least 0.3 m (1 ft) deep into the soil and does not extend more than 5 cm (2 in.) above the soil surface.

In reporting measurement results, a complete description of the tunnel, track, ground surface, and adjacent building configuration should be given. Specification of soil geology is also useful.

7. STATION AND TUNNEL NOISE IN URBAN RAIL SYSTEMS

7.1 STATION PLATFORM NOISE LEVELS AND SPECTRA

Figure 7-1 shows a typical recording of the A-weighted sound level measured on the platform as a rapid-transit train enters and leaves the station. Noise sources during train entry and departure include wheel-rail interaction, mechanical brakes, impulsive air release from the brake system, door operation, air conditioning, and train auxiliary equipment. The ambient sound level is determined by other sources such as air-handling systems, escalators, and road traffic.

Figure 7-2 shows the platform maximum noise levels due to trains for stations of various constructions. The highest noise level situations are those where older cars run non-stop on jointed rail through underground stations lacking sound absorptive treatment. Local operations (i.e., all trains stop at the station), modern cars, welded rail, modern underground station design with noise control features each contribute to a reduction in sound level. The quietest stations are above ground with tie and ballast track and are away (or protected) from the ambient noise, due to highways for example.

Figure 7-3 shows the average values of relative spectra of local and express trains in a variety of underground stations. For stations having curved track in their vicinity, wheel squeal can alter the sound level and spectral shape significantly, e.g., Figure 2-4.

7.2 NOISE LEVELS AND SPECTRA IN TUNNELS

In untreated tunnels, the A-weighted sound levels <u>alongside</u> moving trains are in the range:

$$L_A = 102 + 30 \log_{10}\left(\frac{V}{V_o}\right) \pm 6 \text{ dB (A)} \qquad (7.1)$$

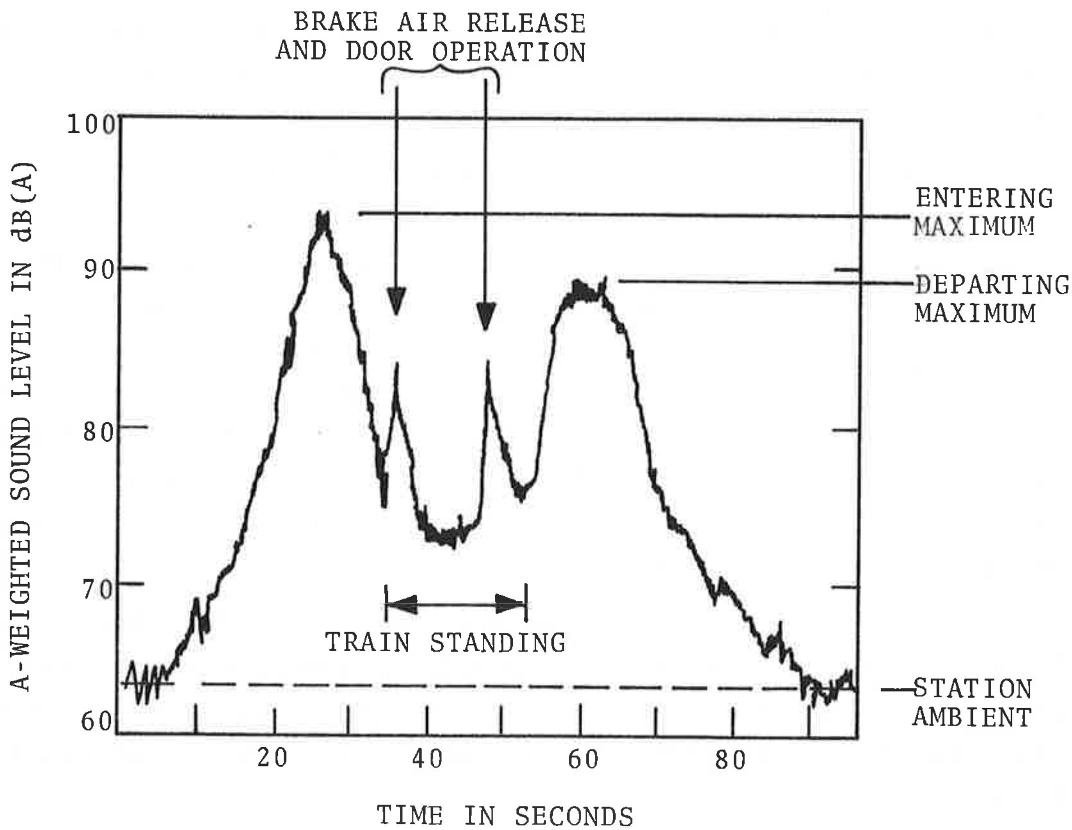


Figure 7-1. Noise on a station platform vs. time as a rapid transit train enters and leaves the station.

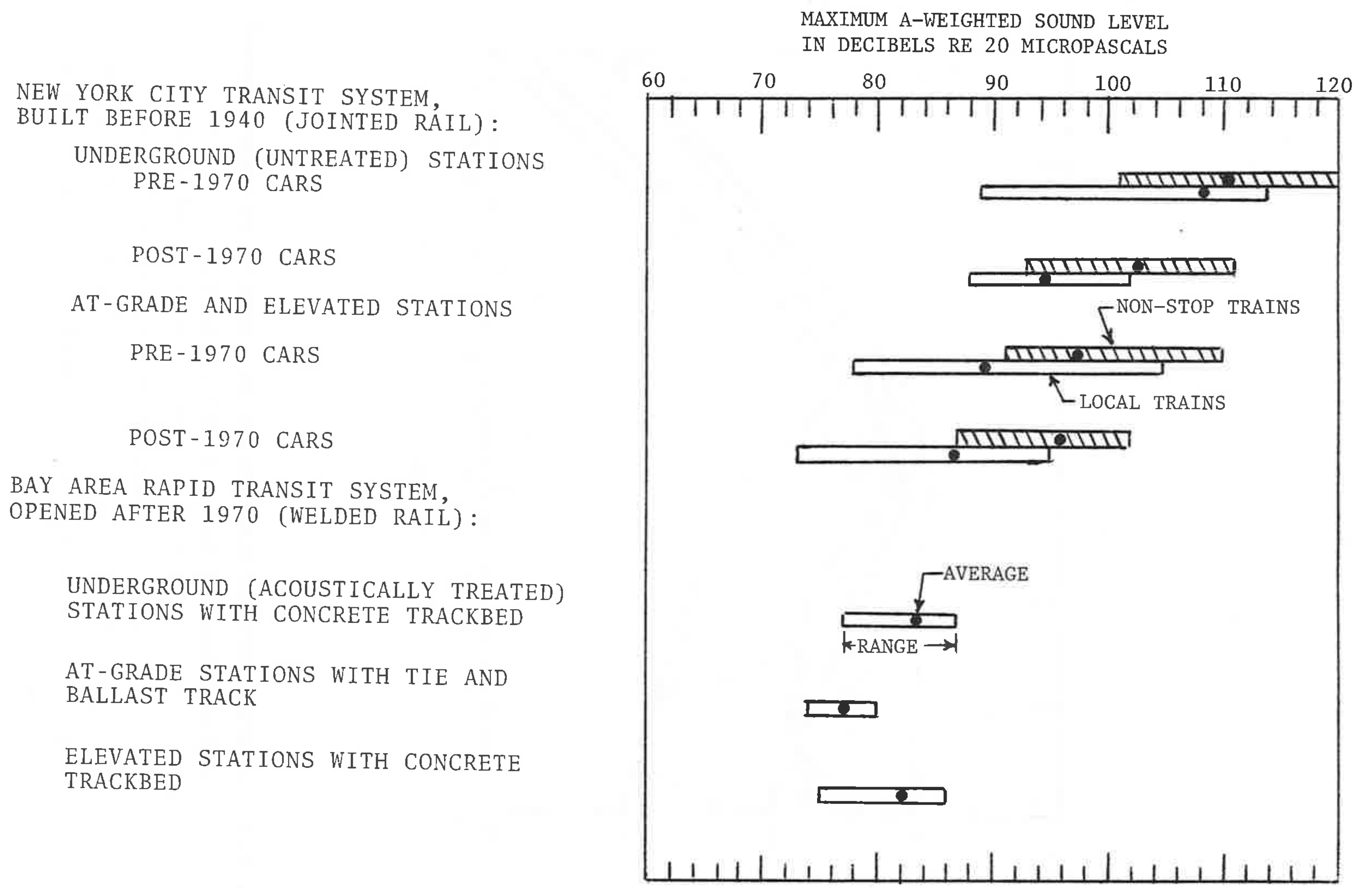


Figure 7-2. Averages and ranges of typical station platform A-weighted sound levels. [Wolfe, et al., 1976; McShane, et al., 1975]

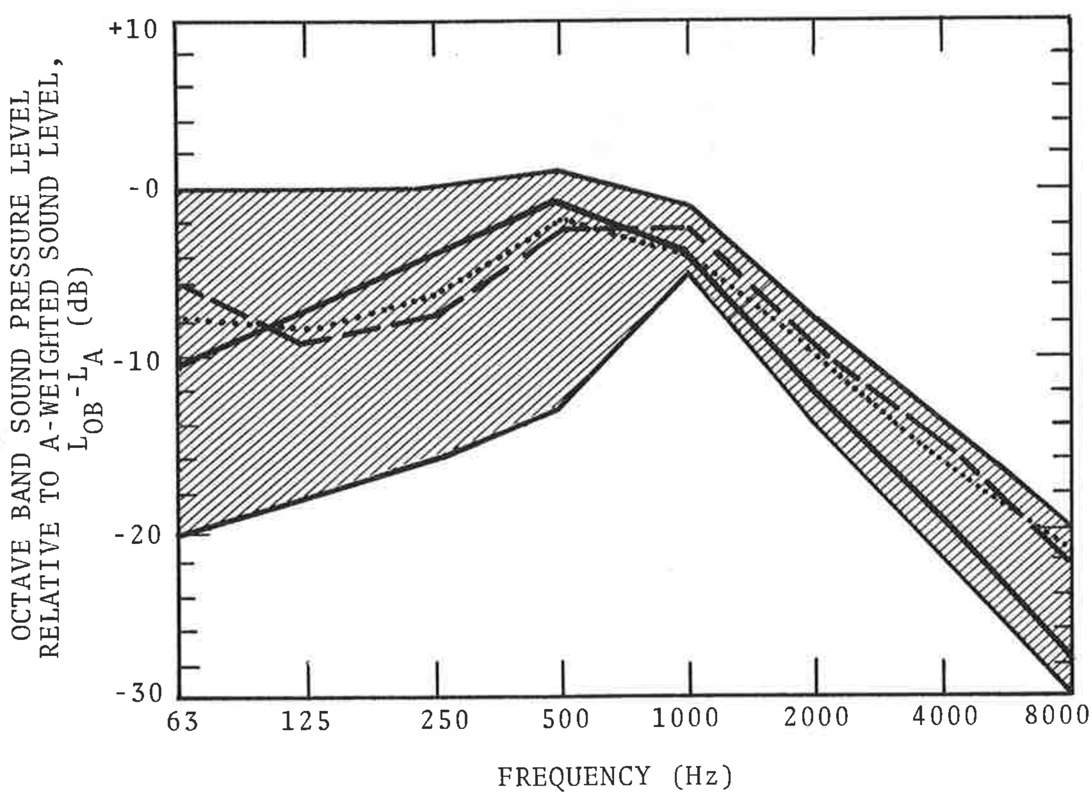


Figure 7-3. Average octave band relative spectra in underground rapid transit stations and tunnels:
—— —— Untreated stations, above platform;
———— Stations having absorptive treatment;
--------- Untreated tunnels alongside and between cars. Crosshatched region includes 90 per cent of all data.

where V is the train speed in km/h (or mph) and V_o is 60 km/h (37 mph). Equation 7.1 is based on data from four transit systems. Jointed rail and/or ballastless construction yield levels near the top of the range, and smooth welded rail in ballasted trackbed yields levels near the bottom. A-weighted sound levels under moving trains in tunnels are 5 to 10 dB (A) higher than the levels alongside the trains.

Average relative spectra for trains in tunnels are included in Figure 7-3.

7.3 CONTROL OF NOISE IN STATIONS AND TUNNELS

To achieve acceptable sound levels in stations it is usually necessary to control wheel-rail noise at the source (see Table 2-1) and to minimize the build-up of reflected (reverberant) airborne noise by sound absorptive treatments. In certain cases low barriers are used in conjunction with the absorptive treatments.

As a rule, untreated underground stations, having tie and ballast track, have a sound level 5 to 15 dB(A) lower than similar stations with non-ballasted track, because of sound absorption provided by the ballast. An absorptive ceiling over the station platform results in a reduction in A-weighted sound level of 5 to 10 dB(A) on the train platform (for otherwise untreated stations). Suitable locations for absorptive treatment are shown in Figure 7-4.

7.3.1 Design Calculations

On the platform or in the tunnel outside the train the noise level reduction (NR) for reverberant sound, which is provided by sound absorptive treatment, can be estimated from the classical diffuse reverberant field equation:

$$NR = 10 \log_{10} \frac{a_a}{a_b} \doteq 10 \log_{10} \frac{T_b}{T_a}, \text{ dB for } \frac{a_a}{a_b} \leq 10. \qquad (7.2)$$

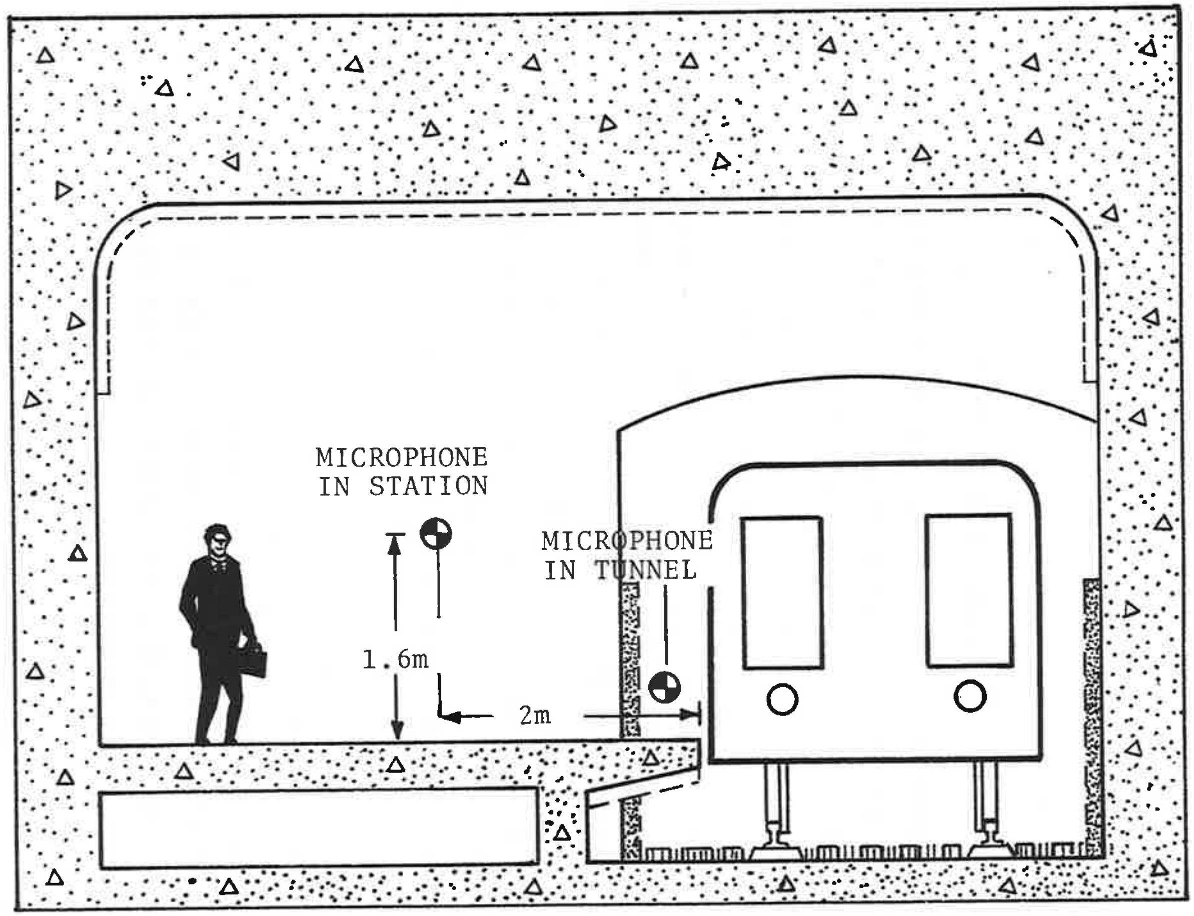


Figure 7-4. Locations for acoustical treatments in subway stations tunnels. – – – – – ___ Suitable locations in stations; ...In tunnels; Location for stone ballast; Preferred measurement locations in in station and tunnel.

This level reduction applies at a given frequency; a_b and a_a are the total absorption at the specified frequencies before and after treatment, expressed in Sabins; and T_b and T_a are the reverberation times before and after treatment, respectively. The reverberation time in stations can be estimated from classical theory as [Beranek, 1971, p. 238]

$$T = \begin{cases} 0.16\ V_s/a \text{ (SI units)} \\ 0.049\ V_s/a \text{ (English units)} \end{cases} \tag{7.3}$$

where V_s is the volume of the station. When laboratory values are used for calculating total absorption, "a", the equation generally overpredicts reverberation times slightly.

Equation 7.2 tends to underestimate noise reduction due to absorptive treatments near the wheels and rails. This results from the fact that the space under the railcar is partially enclosed by the station and tunnel structure. In addition, Equations 7.2 and 7.3 do not account for the duct-like attenuation of sound with distance along the station or tunnel length.

Reverberation times averaged over the four octave bands from 250 to 2000 Hz, range from about 1 sec in acoustically treated stations to greater than 6 sec for untreated stations.

7.3.2 Selection of Materials

Sound absorptive treatments for stations must generally be visually attractive, reflective of light, vandal resistant, cleanable, fire resistant, easily maintainable, reasonably inexpensive, undamaged by water, and compatible with normal inspection and maintenance. They must hold firm against air currents or over-pressures due to train motion (this requires careful design of suspended ceilings). Sound absorptive treatments for tunnels must have all but the first two attributes, with general ruggedness substituted for vandal resistance. Some materials which meet these requirements are discussed on page 90.

Sound absorption provided by ballasted track has an acoustic advantage over many other sound absorptive treatments because the stone ballast is close to the sources of noise [Bolourchi, 1975; Heller & Bender, 1973]. Furthermore, ballast is not damaged by workmen or their equipment.

Certain spray-on materials of mineral fiber are suitable for use on tunnel walls [Wilson, 1971c]. Treatment heights are generally between 1.8 and 3 m (6 and 10 ft). Such materials are also suitable at station locations where they cannot be reached by patrons. Whether specific treatments will function successfully in a particular underground rail system depends on the safety standards and cleaning methods of that system as well as on the material properties. The Toronto Transit System, for example, which has extensive and successful tunnel treatment, uses only products passing its tests for acoustic performance, washability, fire resistance, and durability.

Glass or mineral fiber blankets or boards, which generally require thicker treatment than spray-on materials, may be used in tunnels and stations provided sufficient clearance exists [Wilson, 1971c]. The blankets should be sealed in a thin plastic film (such as polyethylene) to keep out water, to prevent erosion, and to avoid accumulation of dirt in the material. Surface ruggedness and visual acceptability may be achieved by enclosing the sealed blankets behind performated metal or other stiff, perforated material.

7.4 NOISE CRITERIA AND MEASUREMENTS: STATION AND TUNNELS

The objective of noise control in stations is to provide a comfortable environment in which waiting passengers may converse and understand the public address system. In tunnels the objective is to provide a sound field external to the rail car which will, in conjunction with sound insulation of the body of the car, structure-borne noise isolation, interior sources, and car interior absorption result in an acceptable interior environment. A secondary purpose for tunnel treatment can be to control train noise propagation along the tunnel to station areas.

For trains entering or leaving above-ground rapid transit stations a maximum A-weighted sound level of 75-80 dBA* is considered acceptable for tie and ballast track (levels to 85 dBA are acceptable if the trackbed is concrete). An A-weighted sound level of 80 to 85 dBA is considered acceptable in underground stations. For express trains which run through a station without stopping, a maximum A-weighted sound level of 85 dB is considered acceptable. Lower levels, though desirable in all cases, may be disproportionately costly. Sound levels up to 68 dBA and up to 55 dBA anywhere on the platform are acceptable for stopped trains and station ventilation systems, respectively.

The usual location for measuring noise in a station if a single microphone is used is 1.6 m (5.25 ft) above the platform, 2 m (6.6 ft) back from the platform edge (Figure 7-4) or halfway to the back wall, whichever is less, and at the mid-point along the platform length. Additional microphones may be used at either end of the station.

In order to achieve acceptible levels of station platform noise, sufficient absorption is recommended to obtain station reverberation times of 1.2 to 1.6 sec in mid-frequencies.

Train noise in tunnels is often measured from fixed microphones between the rails under the train, and if space permits, alongside it (Figure 7-4). Microphones on trains are often mounted in the truck area or, for rapid transit trains, outside between the cars, 1.2 m (4 ft) above floor height. In all cases steps must be taken to ensure data is not contaminated by wind noise or microphonics.

*Except where indicated, all references to criteria and present practice are consistent with APTA, [1976] and apply to new construction. In modification of existing facilities and equipment these criteria sometimes cannot be met for reasonable cost.

8. NOISE AND VIBRATION IN RAIL VEHICLES

8.1 NOISE LEVELS AND SPECTRA

8.1.1 Rapid Transit and Streetcars

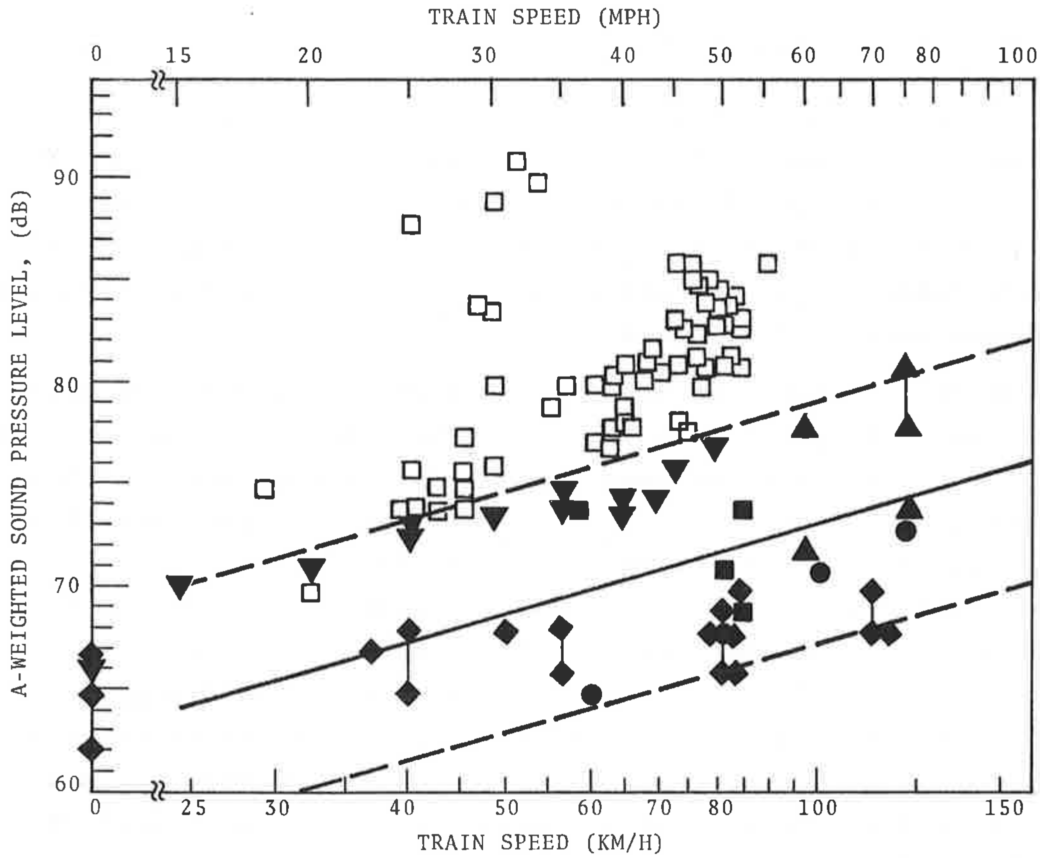
A-weighted sound levels inside metropolitan trains range from about 65 to about 105 dB(A) during normal operations in the open and in tunnels [EPA, 1975; Davies & Zubkoff, 1964; Ungar, 1974]. The wide range is due to differences in track and car construction and maintenance, and in speed.

Figure 8-1 shows the interior A-weighted sound levels [measured at the car center, 1.2 m (4 ft) above the floor] as a function of speed for a variety of metropolitan trains in the open. Within a given car, sound levels are typically 3 to 4 dB higher over the trucks compared to car center. For many older cars (represented by open squares) the interior noise is dominated by wheel-rail or propulsion system noise emanating from the under-car region. For these cars, sound levels increase approximately as 30 $\log_{10}$ (velocity). In newer cars (represented by solid symbols in the figure) insulation from the under-car noise sources is greater and sound levels depend on other sources, including the air distribution system. For modern cars on tie-and-ballast track in the open, the A-weighted sound level is,

$$L_A = 70 + 15 \log_{10} \left(\frac{V}{V_o}\right) \pm 6 \text{ dB(A)} \qquad (8.1)$$

where V is the speed in km/h (or mph) and V_o is a reference speed of 60 km/h (37 mph). Noise levels inside a car are typically 7 dB(A) higher (range 2 to 11) in a tunnel than when in the open at the same speed.

The U.S. Department of Transportation's State-of-the-Art Cars (SOAC) running in demonstration service on one existing rapid transit system produced interior A-weighted sound levels of 68 dB(A) in the open compared to 75 dB(A) for cars which ordinarily



Older or Non-Sealed Cars, Elevated Track and Tie & Ballast Track at Grade

□ Data from four systems in USA

Newer Air Conditioned Cars, Tie & Ballast Track

◆ State-of-the-Art Cars, Transportation Test Center, Pueblo, Colorado.

▼ Light Rail Vehicle (trolley), Boston.

● Model 420 S-Bahn Cars, Germany.

■ #1 South Shore Rapid Transit Cars ("Silverbirds"), Boston.

▲ San Francisco Bay Area Rapid Transit (BART) and Philadelphia-Camden Port Authority Transit Corp. (PATCO) Cars.

Figure 8-1. Interior A-weighted sound levels of steel-wheeled metropolitan trains in the open vs. speed.

operate on that line. In a tunnel of another, older system, SOAC's interior sound levels were 74 dB(A) compared to 102 dB(A) for the ordinary cars [Boeing Vertol, 1975]. This demonstrates that the technology for providing acoustically comfortable urban railcars is currently available.

The average octave band relative spectrum for interior noise in modern metropolitan railcars is shown in Figure 8-2.

8.1.2 Intercity Passenger Trains

A-weighted sound levels range from about 56 to 81 dB(A) for trains in the U.S.A. and Europe at speeds up to 200 km/h (124 mph [EPA, 1975; Stuber, 1975b; Eade, 1977; Bray, 1974]. Sound levels inside passenger trains increase approximately as 15 log V. Spectra for several cars traveling 120 to 145 km/h (75 to 90 mph) are shown in Figure 8-3.

8.1.3 Locomotive Cabs and Engine Rooms

Measurements of steady A-weighted sound levels in cabs of U.S. diesel locomotives at full load and throttle and speeds to 80 km/h (50 mph) average 91 dB(A) (range 89-93) with windows closed. Sound levels of 122 to 124 dB(A) were measured in the engine rooms. The corresponding A-weighted sound levels were about 20 dB(A) lower in both locations with throttle at idle. The C-weighted sound levels were typically 10 dB higher than the A-weighted levels. Engine and exhaust are the dominant noise sources. However, in one set of cab noise measurements at 245 km/h (152 mph), [Eade, 1977] air flow over the outside of a cab produced a steady interior sound level of 92 dB(A).

Cab transient noises include: whistle, 98 to 108 dB(A) with windows closed and open respectively; air brake application, 98 to 105 dB(A) for normal application, up to 125 dB(A) for emergencies; overspeed alarm, up to 120 dB(A); and two-way radio, 93 to 103 dB(A).

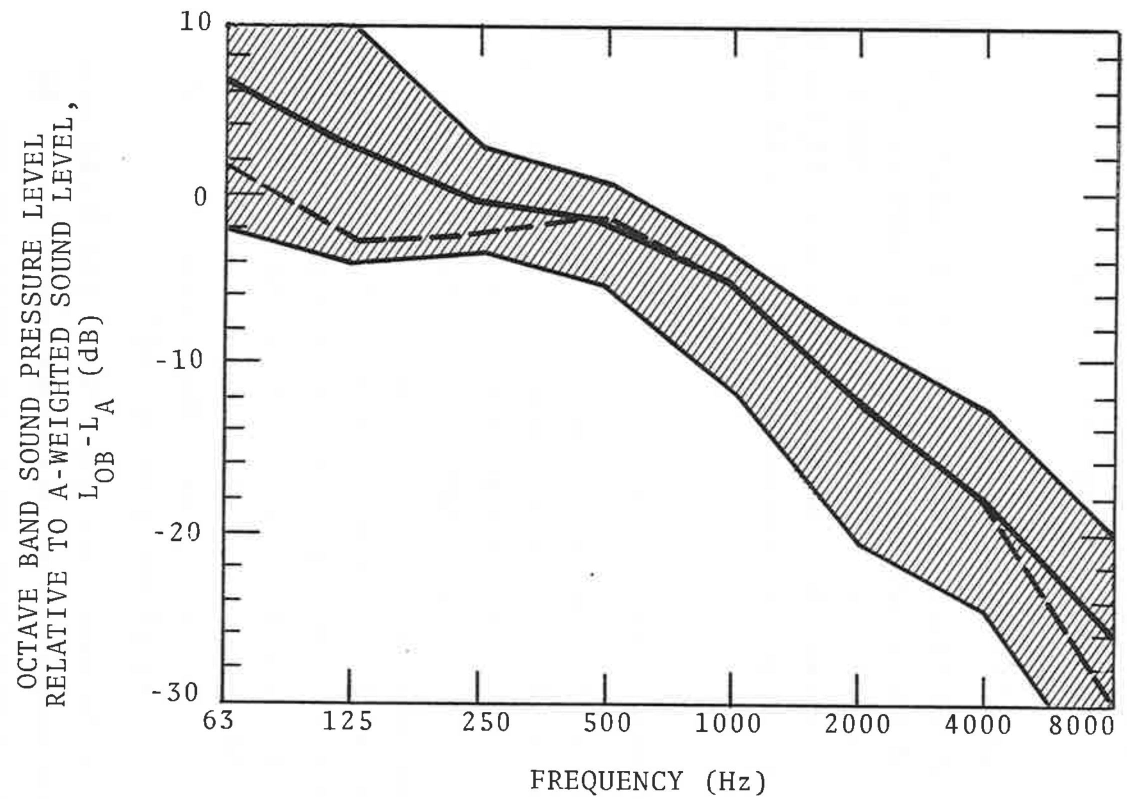


Figure 8-2. Octave band relative spectra for noise inside North American metropolitan trains.

——— Average of 19 measurements of modern sealed vehicles in the open and in tunnels, and 4 measurements of older vehicles in the open.

– – – Average of 3 older vehicles in tunnels.

Range of all data.

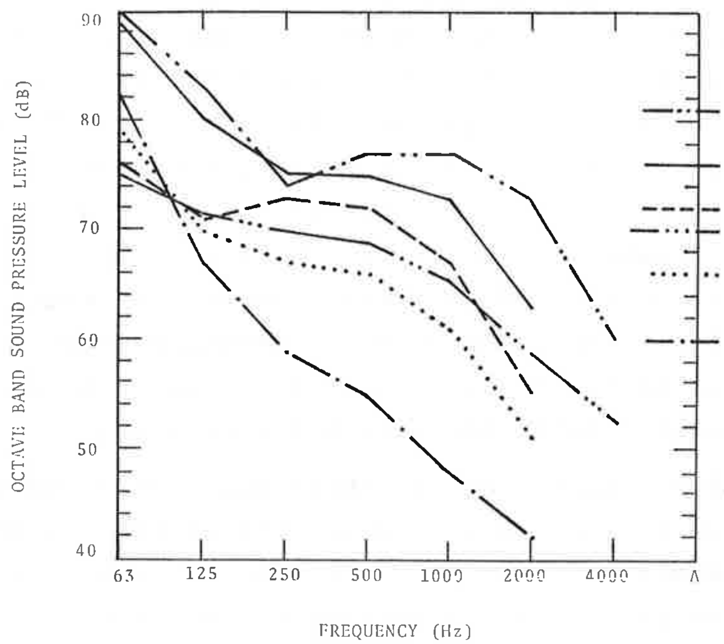


Figure 8-3. Passenger car interior noise spectra.

Metropolitan cars:

—···— Modern sealed North American cars at 40 to 129 km/h (25 to 80 mph) in the open and in tunnels, average octave band <u>relative</u> spectrum (from Fig. 8-2) plus 70 dB(A).

Intercity cars:

—··— British MkI 2nd class coach (older coach) 145 km/h (90 mph) [Eade, 1977].

——— U.S.A. high level coach (older coach) 127 km/h (70 mph), and

— — — French turbotrain coach in U.S.A. 127 km/h (79 mph) both at highway crossings.

······· German 2nd class coach 120 km/h (75 mph).

—·— British MkIII 2nd class coach 145 km/h (90 mph) [Eade, 1977].

8.2 CONTROL OF NOISE AND VIBRATION IN VEHICLES

Most of the important noise and vibration sources have already been discussed. These include exhaust and engine casing for diesel locomotives, wheel-rail interaction, the propulsion system, and under-car auxiliary equipment for railcars. Pressure fluctuations on the rail-car shell due to turbulent flow of air can also be an important concern for internal noise at speeds as low as 160 km/h (100 mph) [Bickerstaffe, et al., 1975] even though wayside noise is unaffected by airflow sources at these speeds. Turbulent flow noise is reduced by smoothing vehicle contours. Noise control at the source should be pursued in conjunction with methods discussed below for path noise control.

Figure 8-4 illustrates the three ways noise and vibration can reach occupants of passenger cars: (1) external airborne sound and flow turbulence cause pressure fluctuations on the car shell components which transmit a portion to the interior; (2) structure-borne vibration is transmitted from wheels, motors, and under car equipment along solid paths to interior surfaces which then vibrate and radiate noise inside; and (3) sound is generated within the car itself. Any one of these paths or interior sources can dominate depending upon car design [Ungar, 1974; Eade, 1977; Spencer 1978; Boeing Vertol, 1974].

In general, noise and vibration tests with a prototype railcar operating up to design speeds are recommended for noise control design and development.

8.2.1 Control of Airborne Noise

To understand the important aspects of car design for the control of airborne noise transmitted from the car exterior, it is useful to begin by considering the sound field immediately outside a moving railcar. The strength of the field generally varies from a maximum under the car in the vicinity of the trucks, to a minimum above the car. Octave band data for noise under metropolitan cars in the open and in tunnels are shown in

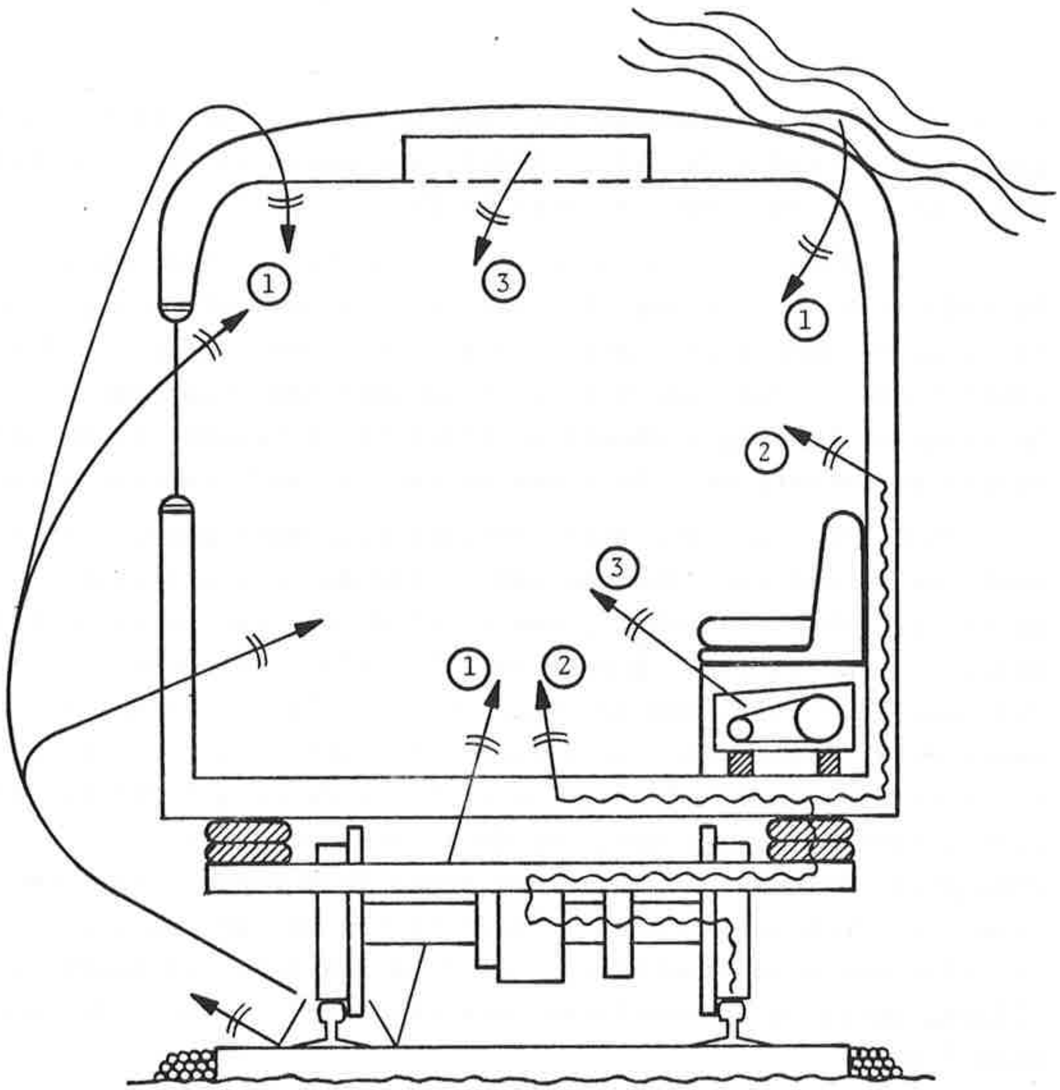


Figure 8-4. Noise and vibration paths to railcar interior. 1. External airborne noise transmitted through car shell. 2. Structure borne noise and vibration. 3. Airborne noise from interior sources. Further discussion is available on sources in Ungar, [1974], and Eade, [1977]; and construction details in Boeing Vertol, [1974], and Spencer, [1978]

Figure 8-5. Corresponding data for intercity cars in the open is given in Newland & Cassidy, [1975], and noise at the side and above cars in the open is shown in Figure 8-6.

Noise levels inside a car due to airborne sound outside can be reduced by increasing the transmission loss of walls, floors, etc., or by adding absorption to the interior surfaces. The noise control design must satisfy any load carrying requirements imposed by structural design, should be moderate in weight, flame and vandal resistant, easy to clean or repair, and visually acceptable.

In modern railcars these requirements have generally led to door, wall, and roof designs comprising damped, airtight exterior sheeting, fibrous blankets, and metal or plastic interior trim panels [Spencer, 1978; Boeing Vertol, 1974]. Windows in walls and doors are fully sealed with rubber gaskets. Either two glass panes or a single-pane three-layer laminate is used. The laminate consists of a layer of glass on either side of a layer of viscoelastic material. Air handling ducts are generally lined with absorptive material to attenuate noise from outside or from blowers. Even small holes or gaps to the ouside can increase interior noise substantially. After a few years of service, sliding doors on metropolitan cars frequently leave small gaps when closed.

For locomotives the principles are the same, but the higher noise levels in the engine compartment adjacent to the cab has led to double wall construction in some circumstances [Willenbrink, 1973]. In German diesel locomotives having carefully designed acoustic and structural isolation, cab sound levels are 72 to 85 dBA under full throttle and load.

8.2.2 Control of Structure-Borne Noise and Vibration

The most important sources of structure-borne noise are wheel-rail interaction, truck-mounted equipment, and under-car auxiliary equipment. Control of vibration along structural paths is accomplished by resilient isolation and damping.

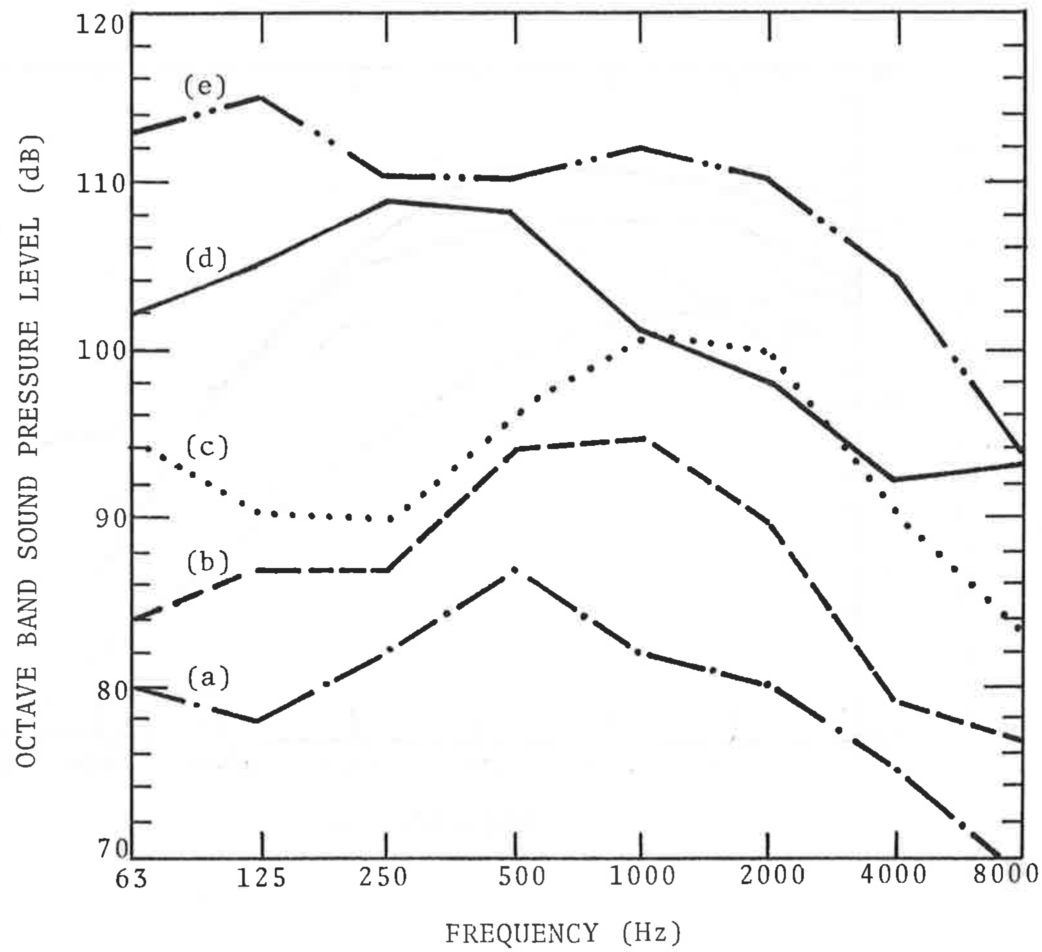


Figure 8-5. Noise under railcars at different speeds.
(a) SOAC in the open on tie and ballast track, 0 km/h, all auxiliary equipment operating.
(b) SOAC, 40 km/h (25 mph).
(c) SOAC, 80 km/h (50 mph)
(d) New York City transit car in all- concrete subway, 43 km/h (27 mph).
(e) BART test car in the open on tie and ballast track, 129 km/h (80 mph).

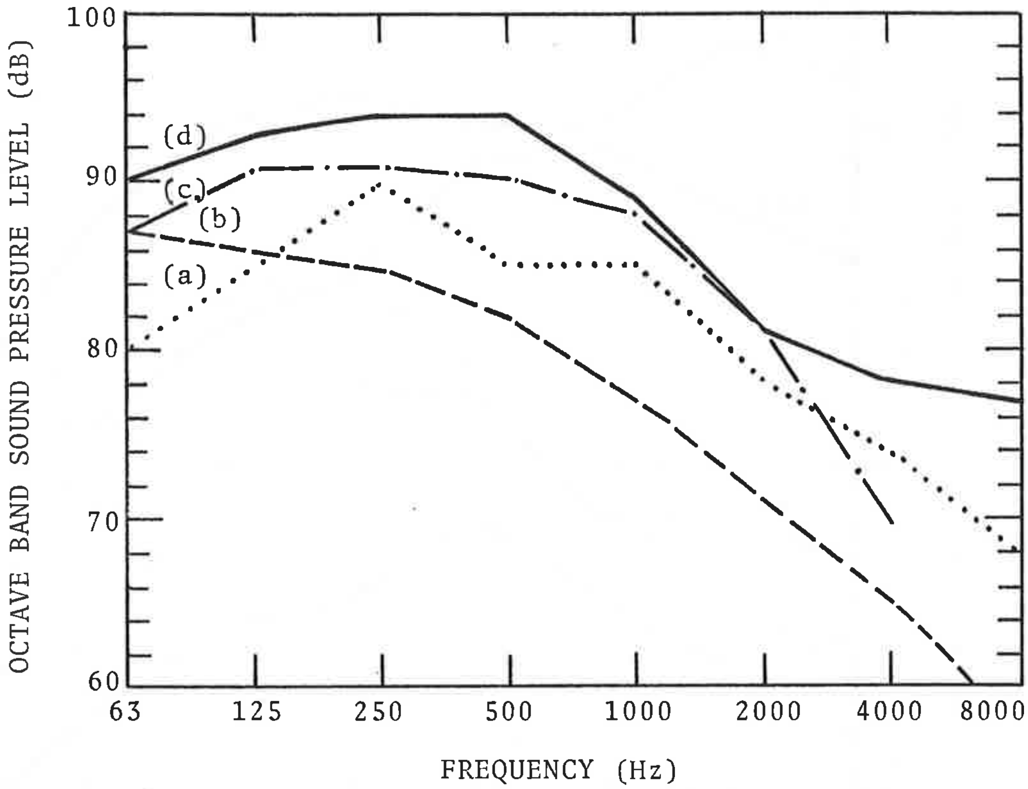


Figure 8-6. Noise alongside, above and between railcars in the open.

(a) Boston transit car, welded rail, 80 km/h (50 mph), between cars.
(b) SOAC, 40 km/h (25 mph), alongside.
(c) British intercity car, 160 km/h (100 mph), alongside and above.
(d) New York City transit car, 70 km/h (42 mph), between cars.

Vibration in the range from about 0.3 to 30 Hz is perceived by passengers as whole body vibration. For moving railcars, whole body vibration is largely determined by basic truck design and wheel maintenance, track design and maintenance, and car body structural mass and stiffness [Newland & Cassidy, 1975]. Above 30 Hz, passengers feel local vibration (on hand holds, floor, car seats) and hear noise due to structure-borne vibration.

Most modern metropolitan transit cars incorporate rubber-like materials in the axle suspension, and many (particularly in Europe) have resilient wheels as well. The Hamburg metro cars, for example, use both methods to isolate wheel-rail vibration from the truck.

8.3 INTERIOR CRITERIA AND MEASUREMENTS

The objective of noise and vibration control for metropolitan railcars interiors is to provide an acoustic environment in which passengers are comfortable and can converse or read without strain. Intercity passengers expect, in addition, to be able to write or sleep if they wish. In locomotive cabs hearing conservation and job performance are prime considerations; the latter requires communication by radio, perception of sound signals, and avoidance of noise-induced fatigue.

8.3.1 Railcars

Criteria for the maximum A-weighted sound levels in modern metropolitan trains operating in the open on new tie-and-ballast track range between 68 and 75 dB(A). For example, APTA, [1976] suggests a maximum of 70 dB(A) at 1.2 m (4 ft) above the floor on the centerline of a complete car, empty of passengers. Sound levels 4 and 10 dB(A) higher are suggested as limits when cars travel over concrete roadbed (e.g., slab deck elevated structures) and in tunnels, respectively. For stationary trains in the open the limit for steady sound level is 68 dB(A). For modern intercity trains in the open, interior sound levels between 60 and 70 dB(A) are specified. Adequate speech privacy in non-compartment cars requires sound levels not less than about 60 dB(A). Measurements should be made according to standard procedures [ISO, 1975a; UITP, 1972b; APTA, 1976].

8.3.2 Locomotives

The U.S. National Railroad Passenger Corporation, Amtrak, specifies cab interior A-weighted sound limits of 84 dB(A) for newly purchased diesel locomotives under full throttle and load at 80 km/h (50 mph) with all auxiliaries operating and windows closed. For electric locomotives the maximum is 78 dB(A). These specifications do not apply to transient events such as horn, brakes, or to overspeed alarm.

9. SYSTEM-WIDE CONSIDERATIONS

9.1 THE DESIGN PROBLEM

Whereas previous sections have dealt with various parts of rail systems individually, this section discusses the planning and design of noise and vibration control from the viewpoint of overall system performance. The discussion applies mainly to metropolitan systems.

System noise control begins with the selection of noise and vibration criteria for stations, car interiors, and wayside communities. Often during planning or early design, environmental commitments are made to those potentially affected by the new or renewed system. These commitments vary along the right-of-way depending upon land use, prior exposure, and perhaps rider expectations. The designer's goal is to meet all noise and vibration criteria everywhere on the system for minimum possible cost.

Metropolitan rail transit differs from other transportation modes in that a single organization, the transit authority, controls both vehicle and guideway designs and maintenance. This makes it possible to trade off fixed treatments which affect local conditions (e.g., wayside barriers or tunnel absorptive linings) against car-borne treatments which affect noise and vibration over the entire system (e.g., resilient wheels or greater car body noise insulation) [Lotz & Kurzweil, 1973]. A complete, systematic cost and acoustic performance trade-off study probably requires computer assistance [Lotz, 1976] because the number of possible treatment combinations exceeds manual capabilities. Less complete manual methods are usually used, see for example Ungar, [1974].

9.2 DESIGN DEVELOPMENT

Basic decisions, related to ways and structures, must be made on the lengths and locations of subway, at grade, and elevated track sections. Secondary decisions must also be made on such things as the use of direct fixation track versus tie-and-ballast track in tunnels, and on elevated structures and on the use of floating slabs, tunnel and station absorption, ballast mats, and barrier walls.

With respect to rail car acoustic design the transit system has essentially two alternatives: (a) specifying noise and vibration performance in the purchase documents to guarantee performance under standard conditions on the completed track; or (b) specifying materials and designs which prescribes how the railcar should be built, but not what acoustic performance it must have.

In practice, specifications tend to combine these apparently disparate alternatives. But with the increase of environmental commitments, U.S. car purchases since about 1970 have increasingly emphasized comprehensive performance specifications covering interior and wayside noise, and interior vibration. This places the responsibility for noise control design and development clearly with the car builder. The builder typically fabricates only the car body, but installs in or on it all other components which he purchases through subcontract.

The quantitative design development approach outlined below has produced several modern metropolitan transit car models, each of which met its noise and vibration performance specification. Although less quantitative approaches based more on experience may also succeed, some specifications actually require an analysis and verification of the type outlined. Guidelines are presented sequentially, but executed iteratively:

(a) Include a bench specification on noise and vibration for all potentially noisy components in subcontract documents. (This is in addition to prime contract requirements on equipment after mounting.) Visit subcontractor plants to insure the intent and importance are understood.

(b) Develop a mathematical model from measured or estimated sound power levels of each component including wheel-rail and propulsion system noise. Take into account any ducting or enclosures planned. Estimate wayside levels. Plan for enclosure or relocation of troublesome components at this time.

(c) Compute transmission loss requirements from (b) unless they are already specified separately. Perform transmission loss, damping, and possibly absorption tests on sample, structurally complete body panels unless such data is already available. These tests are not substitutes for acceptance transmission loss tests on the completed car, however.

(d) Within constraints of cleanability and vandalism resistance, maximize absorption in the car.

(e) Avoid and eliminate flanking paths for both structure-borne and airborne sound. Check design of equipment mounts, air comfort ducts, cable and pipe hangers, and eliminate gaps and holes in the car shell.

(f) Conduct noise and vibration tests on the first few production cars to locate design oversights or shop practices that defeat the design.

A complete, quantitative evaluation has been made of the transportation, economic, and environmental impacts of one new metropolitan system, the Bay Area Rapid Transit (BART), after normal operations began and new travel patterns had established themselves [Bart, 1975; BB&N, 1976]. Measured wayside noise from trains in normal service was found to be equal or slightly less then predictions made six years earlier on the basis of prototype cars operating on a limited length of test track.

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