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16. Abstract

This handbook is a guide to the prediction and control of all types of urban rail transit noise, ranging from train noise heard by the community at large to noise in maintenance shops heard only by transit employees. The topics covered include: acceptability criteria for transit related noise and vibration; the general characteristics of urban rail noise and vibration; techniques and equipment for measurement of noise and vibration; control of transit vehicle noise and vibration; control of community noise from surface tracks and aerial structures; prediction and control of groundborne noise and vibration; control of noise in transit stations; control of noise from station ancillary equipment such as air-conditioning systems and fan and vent shafts; control of noise around yards and shops; control of wheel squeal noise; and control of pressure transients in subway tunnels. The handbook is primarily intended for transit engineers, although it should also be useful for transit system executives and transit system acoustical consultants. Extensive bibliographic information is provided for readers looking for more detailed information. Also, for those interested in a general overview of urban rail noise and vibration, the executive digest of this handbook is available as a separate document.

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PREFACE

This document presents a comprehensive review of the state-of-the-art in rail transit noise and vibration control in a single document that can be used by transit property engineers, transit planners, and transit consultants. The handbook has been prepared by Wilson, Ihrig & Associates, Inc. (WIA) under contract to the U.S. Department of Transportation. The project is part of the information dissemination activities of the Urban Rail Noise Abatement Program managed by the Transportation Systems Center, Cambridge, Massachusetts, under the sponsorship of the Office of Rail and Construction Technology of the Urban Mass Transportation Administration, Office of Technology Development and Deployment.

The handbook has been prepared with the cooperation and assistance of a number of people. In addition to the authors listed on the title page, invaluable contributions have been made by other WIA staff including Steven L. Wolfe, Armin T. Wright, Thomas A. Mugglestone, and Peter Y.N. Lee. Many useful suggestions have also been provided by the Liaison Board of the American Public Transit Association. Their suggestions were particularly helpful in ensuring that the final document be useful to transit system personnel. Elizabeth Ivey of Smith College also provided valuable review and comments.

The technical effort on this project was coordinated by Michael Dinning and Gordon Plank of TSC and Leonard Kurzweil, formerly of TSC. The authors express their gratitude for their assistance and encouragement during their preparation of the handbook.

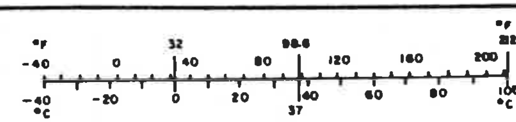
METRIC CONVERSION FACTORS

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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CHAPTER 1

1. INTRODUCTION AND OVERVIEW

1.1 BACKGROUND

During design of new transit facilities it is important to consider the noise and vibration experienced by the adjacent communities, as well as the acoustical environment of the patrons. The acoustical environment created by urban transit systems can be an important source of community discontent, and can have a strong influence on patrons' opinion of the system. This has led to a desire of transit system operators to reduce noise on existing lines, and to minimize patron and community exposure to noise and vibration in new transit facilities.

Recent research in this area has significantly increased the possibilities for transit system engineers to predict and control noise and vibration. Although a large volume of data has accumulated, this handbook is the first effort at summarizing and presenting the material in a single document.

This handbook is a guide for the prediction and control of all types of urban rail transit noise, ranging from train noise heard by the community at large, to the noise in maintenance shops heard only by transit employees.

For the purpose of this document, urban rail transit includes:

- Heavy railcar transit, usually referred to as heavy rail transit. Examples are BART in the San Francisco Bay Area, CTA in Chicago, and NYCTA in New York. The standard steel wheel/steel rail technology is the principal focus of this handbook.
- Light railcar transit, including the traditional trolley systems such as the Muni in San Francisco and the Green Line section of the MBTA in Boston. Also included in this category are systems using light rail vehicle (LRV)

technology, such as the NFTA in Buffalo, the MBTA in Boston, and the Muni in San Francisco.

- Fixed Guideway rubber tire transit. Rubber tire transit refers to rapid transit technology such as the RATP in Paris and the systems in Montreal and Mexico City. It should not be confused with buses or trackless trolleys. With the exception of its rubber tires and concrete guideways, rubber tire transit has much in common with standard steel wheel/steel rail rapid transit.

Many of the noise and vibration control schemes and much of the introductory and fundamental material presented here is equally applicable to all modes of transportation. Therefore, this handbook is also relevant to systems not specifically covered, such as commuter rail, inter-city rail, inner-city people movers, trackless trolleys, freight trains and buses.

Techniques of vibration control are a necessary part of this handbook. Noise and vibration are inherently linked; all noise is created by some type of vibration, and many noise control methods consist of reducing the vibration at the source. The ride quality, or ride comfort, of transit system vehicles also depends on vibration levels; however, ride quality is not closely related to noise control, and as such is only briefly discussed. No attempt has been made to provide guidelines for modifying truck design in order to improve the ride quality.

1.2 HANDBOOK USE AND ORGANIZATION

Although the handbook is primarily intended for transit engineers, it should also be useful for transit system executives; members of the public interested in a general overview of transit noise and vibration; transit system acoustical consultants; and others familiar with acoustics and noise control.

Chapters 1, 2, and 3 present general information: introducing the subject of transit noise and vibration and presenting a general overview (Chapter 1), discussing acceptability criteria for noise and vibration (Chapter 2), and finally, presenting specifics of transit system noise levels (Chapter 3). Chapters 5 through 12 each cover a specific area of transit system noise control; thus the reader who is interested in a specific transit noise control problem can refer directly to the relevant chapter.

One chapter that does not fit readily into the above two categories is Chapter 4, which discusses measurements of noise and vibration. The discussion in Chapter 4 ranges from the proper use of portable sound level meters to the selection of a computerized data analysis system. Chapter 4 is supplemented by a discussion of different measurement instrumentation components in Appendix D. Supplementary information to the handbook is provided by four appendices, and an index.

How the handbook is used will depend upon the specific goals of the reader. Those interested in a general overview of transit system noise and vibration will probably find Chapters 1 and 2, and parts of Chapter 3 sufficient. If, after reading these chapters, more detailed information on a specific subject is needed (e.g., control of groundborne noise and vibration), the Table of Contents or the Index will indicate where it can be found. It is recommended that the reader at least skim the introductory chapters, even if he is familiar with acoustics or transit systems in general. Those not familiar with the field, but who need detailed information nonetheless, should turn to Appendix A, in which fundamental noise and vibration concepts are presented and general control methods discussed.

For further information, the reader can also refer to the references listed at the end of each chapter, most of which include brief annotations. Transit system acoustical consultants, some transit system engineers, and researchers in the field of transit system noise and vibration will find the annotated bibliographies of particular interest. The annotations indicate briefly what is covered by the reference and the degree to which the reference was used in the preparation of this handbook.

1.3 INTRODUCTION TO URBAN RAIL NOISE AND VIBRATION

Rail rapid transit, like all modes of urban transportation, creates noise. Communities along the route may find the noise intrusive; patrons of the system may find it uncomfortable. In extreme cases, transit employees may be exposed to noise levels above the Occupational Safety and Health Administration (OSHA) criteria for acceptable noise exposure. Operations of older transit systems, particularly those with lightweight steel elevated structures, such as NYCTA in New York and CTA in Chicago, can cause very high levels of noise, giving the public the impression that rail rapid transit is inherently a very noisy form of transportation. However, the more recently developed transit systems, such as BART in San Francisco, TTC in Toronto, WMATA in Washington D.C., and MARTA in Atlanta, show that it is possible to build transit systems that are acoustically acceptable to the communities along the transit route. Although it is economically unfeasible (and technically impossible) to build noiseless transit systems, it is possible and practical to keep total community noise exposure from rail transit well below that created by most other modes of transit, such as buses and highways.

1.3.1 Important Acoustical Terms

In this handbook the term "noise" describes most sounds. Sound and noise are usually synonymous, except that "noise " often means unwanted sound. For our purposes, the pejorative connotations of the term are not intended, but there are other areas of potential confusion. "Noise" often describes undesirable electrical signals in, for instance, electronic amplifiers. This usage is covered in Chapter 4. Since we are concerned with acoustics, not electrical interference, the meaning of "noise" should be clear from the context.

The easiest way to understand the relationship between vibration and sound, or noise, is to consider a vibrating object. Visualize a drum. When the drum is struck, the vibration of the tightly stretched membrane creates sound waves in the air radiating from the drum. They are clearly audible, and if the drum is touched the vibration is felt.

Sound is a wave phenomenon. In a manner analogous to light waves radiating from a light bulb, sound radiates from its source. The difference is that sound waves travel much more slowly than light waves, and because of their longer wavelength, can bend more easily around barriers. Vibration can also be a traveling wave phenomenon. For example, earthquake waves are vibration waves that travel through the earth's crust from an earthquake epicenter. Because of the close relationship between sound and vibration, vibration waves traveling through a building or other structure are often called "structureborne noise" (or sound); vibration traveling through the earth is sometimes called "groundborne noise."

Sound waves are small pressure fluctuations traveling through a medium such as air, water, or soil. The two properties that describe sound and vibration waves are frequency and amplitude. Frequency, the number of repetitions per second, which is analogous to pitch, is a concept familiar to most people. It is measured in cycles per second by the internationally accepted unit of Hertz (Hz). Humans can hear sounds over a frequency range of 20 Hz to 20,000 Hz, but are most sensitive to sounds at frequencies in the range of 200 Hz to 5000 Hz. Most sounds contain many frequencies, and only rarely does one encounter sounds of a single frequency such as the sound from a tuning fork. Another, unfortunately common, example of a single frequency noise is the screech sometimes heard when trains go around short radius curves.

The amplitude, or height of a sound wave, is given in terms of the magnitude of the pressure fluctuation in pounds per square inch (The equivalent metric unit is the Pascal). The ear responds to a very wide range of pressure fluctuations, the ratio of the smallest pressure sensed by the human ear to the highest sound pressure tolerable is approximately 1 to 1,000,000. Because of this wide range of intensities, the decibel unit is used to simplify description of sound wave amplitude. A formal, mathematical definition of the decibel is given in Appendix A. It is sufficient to say here that the decibel (dB) scale is logarithmic with the following properties:

1. Sound or vibration amplitudes denoted in decibels are generally called levels, such as the sound pressure level, or SPL, and represent a ratio to a reference amplitude.

2. Using the standard reference and considering a 1000 Hz sound, the threshold of hearing is approximately 0 dB and the onset of pain occurs at about 120 dB.

3. Because the decibel scale is not linear, two sound levels cannot be directly added to obtain the level of the combined sound. (Refer to Appendix A for a discussion of decibel addition.)

4. An increase of approximately 10 dB is perceived by humans as an approximate doubling of the loudness of the sound.

5. A 1 dB difference is discernable by most careful observers, but a change of approximately 3 dB is required before most observers will consider it significant.

Vibration is also commonly described in decibels, but there is no universally recognized weighting scale for vibration as there is for sound. Vibration is sometimes presented in absolute terms (e.g. m/sec^2) instead of decibels.

Hearing is complex and subtle, and though the sound pressure levels of two different noises may be the same, we cannot assume that the two sounds will be judged equally loud, or equally annoying. This is because the ear is not sensitive in the same way to sounds of all frequencies and because people judge sounds according to individual preference. Although there is no way to quantify the fact that people will judge the same sounds differently (the classic example is rock music), several scales have been developed that account for the frequency dependence of the human ear. The most commonly used scale for sound is the A-weighted sound level in decibels, abbreviated as dBA, or sometimes dB(A).

The A-weighted sound level has been internationally accepted as a measure of noise level because it is easy to measure and appears to correlate human annoyance as well as the more complex scales. However, the A-weighted scale does have some deficiencies that should be recognized. Consider two noises that have the same A-weighted sound level -- one that has no identifiable components, such as a water fountain, and one that has identifiable components, such as pure tones, sirens, rattles, clicks, or buzzes. An example of the latter is typewriter noise in an otherwise quiet office. Even though the two sounds have the same A-weighted level, the typewriter sound is usually more annoying than the water fountain. The A-weighted scale provides only frequency weighting and takes no account of the time history of the sound (e.g., the presence of rattles) or the presence of pure tone components (e.g., wheel squeal). Furthermore, A-weighted sound levels often underestimate the effective loudness, and thus the annoyance value, of sounds dominated by low-frequency components such as diesel engine exhaust pulsation. To a significant degree, the A-weighted sound level can be considered simply a direct measure of the relative loudness of sounds, and can be used without a complete understanding of its mathematical derivation.

Figure 1.1 indicates the A-weighted sound levels of some common

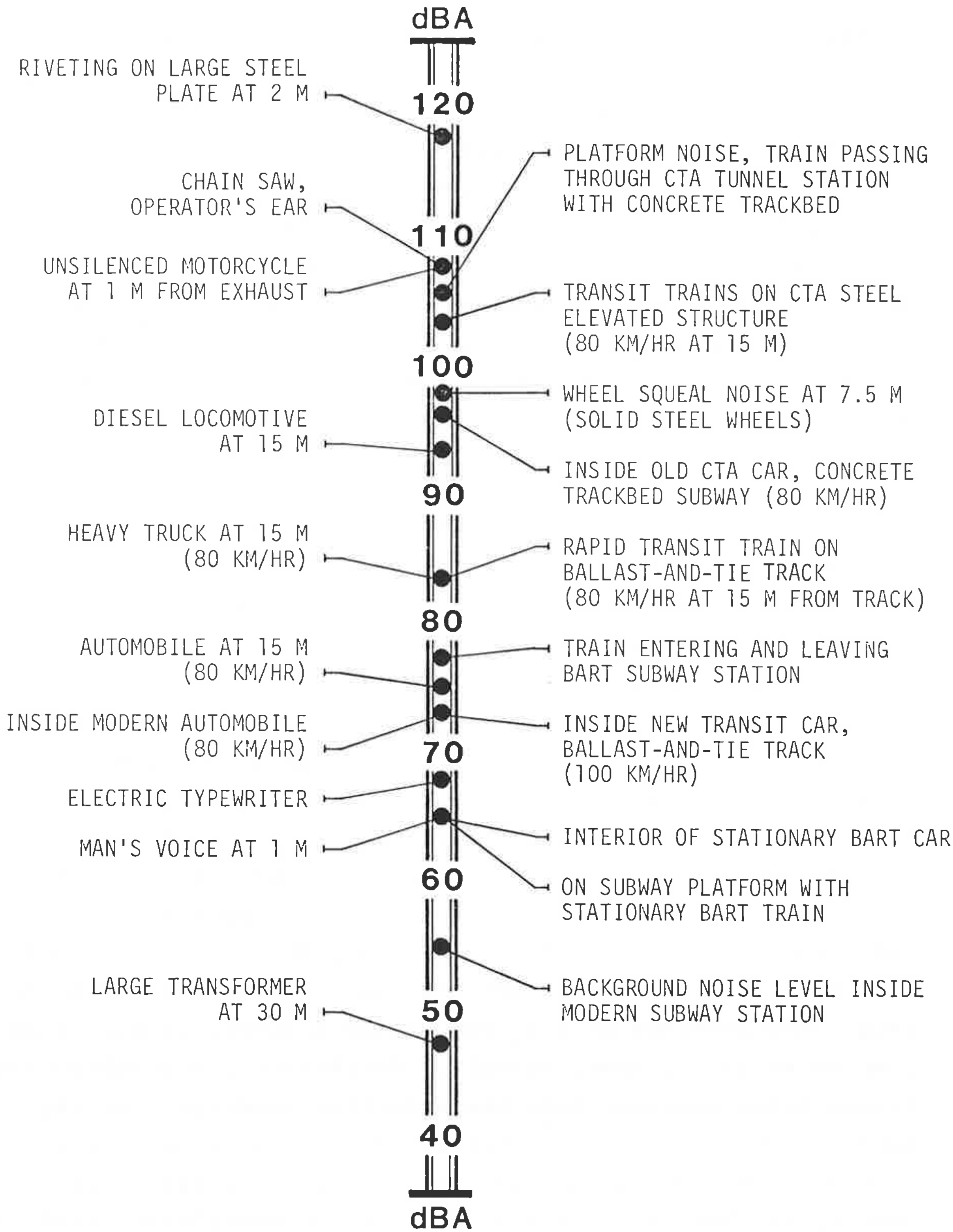


FIGURE 1.1 TYPICAL NOISE LEVELS

noises. Note that the maximum noise level during the passby of a modern rapid transit train is approximately equal to that of a large diesel truck with the train and the truck both at a constant speed of 80 km/hr and a separation distance of 15 m (50 ft). The truck would be significantly louder if it was accelerating.

The A-weighted sound level by itself does not describe the patterns of noise variation over a period of time. Rather, it indicates the amplitude of sound at any specific instant. When used by itself, the A-weighted sound level denotes the sound level at a given moment, a maximum level, or a steady-state level.

When the sound level is constantly changing, it is necessary to account for such variation if the acoustic environment is to be adequately described. Consider the nature of community noise. Short events, such as dogs barking or motorcycles roaring, raise the sound level; normal patterns of human activity throughout the day affect the "average" sound level (when most people are in bed, for instance, community noise drops off dramatically); and long-term trends, such as increases in traffic or a reduction in vehicle noise emission, affect average levels of community noise.

Several scales are used to account for irregularities in community noise. One set of metrics indicates the A-weighted sound level that is exceeded a given percentage of the time. These metrics are usually abbreviated as L_n, where n is the percentage of time. Hence, L_n is the noise level that is exceeded "n" percent of the time. The two most widely used metrics from this group are L_{10} and L_{50}, which represent the noise level exceeded 10% and 50% of the time. L_{10} and L_{50} are useful for describing traffic noise, but they are inadequate for the effects of rapid transit noise.

A second group of metrics is based on the energy average sound levels. These metrics are widely applied to community noises made by heterogeneous sources, including motor vehicles, trains, and airplanes. The most commonly used metrics in this group are the

Equivalent Sound Level (L_{eq}) and the Day-Night Sound Level (L_{dn}). The Equivalent Sound Level of a sound varying over a specific period of time is the steady-state sound having the same A-weighted sound energy as that contained in the time-varying sound. The virtue of the Equivalent Sound Level is that it correlates reasonably well the effects of noise on people, even for wide variations in sound level and time patterns. It is also easily measurable with available equipment.

The Day-Night Sound Level is based on the A-weighted equivalent sound level for a 24-hour period, with an additional 10 dB weighting added to the noise levels during the nighttime hours (10 P.M. to 7 A.M.).

It is important to recognize that L_{eq} and L_{dn} virtually always refer to A-weighted equivalent sound levels. In many cases the units are not explicitly stated or are given as dB instead of dBA. L_{eq} and L_{dn} can always be assumed to be in terms of dBA, unless there is an explanation that they are being used in a nonstandard manner.

So far only very basic acoustical terms have been described. Two more difficult and technical terms are acoustical absorption and sound transmission. Understanding these concepts and the difference between them is fundamental to the control of noise; techniques that use absorption to control noise are ineffectual in controlling sound transmission.

Sound absorption is the absorption of sound energy by a given surface. A perfectly absorptive surface will reflect no sound. Glass wool (fiberglass), mineral wool, and acoustical tiles absorb sound well; nonporous materials with hard surfaces, such as concrete, ceramic tile, and steel, reflect sound and make poor acoustic absorbers. A subway station that has been treated with absorption material is much quieter than an untreated one. In the untreated station, sound energy bounces back and forth between the hard surfaces, with only a slight reduction in amplitude on each bounce. As a result, the noise of trains entering and leaving can build up to an almost unbearable level. In the absorption treated station, sound energy is absorbed after only a few bounces, allowing a minimum of reverberation and creating a quieter area.

Sound transmission, on the other hand, refers to the way sound waves travel from one space to another, through a wall, floor, or other barrier. Materials that successfully reduce sound transmission are generally heavy and relatively nonporous; concrete is a good example. A common means of reducing sound transmission through a wall is to add additional mass to the wall, for example by adding extra layers of sheet rock to a normal wood stud wall. Simply adding a layer of absorptive material to the wall surfaces will have virtually no effect on the amount of sound transmitted through the wall.

1.3.2 Human Response to Noise

Any time the noise environment changes perceptibly, noise impact, and, consequently, complaints from those affected, can be expected. The effects that transit noise can have on humans include:

Physiological Effects: There is a considerable amount of disagreement and discussion among researchers in the field about the hazards of noise to health. Studies have shown sound can cause short-term physiological effects such as raised blood pressure. However, even though many people believe that long-term exposure to noise can contribute to a wide range of physical and mental illnesses, these long-term effects have yet to be thoroughly documented by scientific study. On the other hand, scientific studies have also failed to prove that noise does not contribute to physical and mental illnesses. There is no doubt, however, that the primary physiological effect of excessive levels of noise on humans is temporary or permanent loss of hearing. The effect of high noise levels on hearing loss is well-established, and documented by numerous studies. The OSHA standards for permissible noise exposure (applicable to all United States companies engaged in interstate commerce, and to all work spaces in most states) allow a maximum sound level of 90 dBA for 8 hours per day exposure. As the sound level increases above 90 dBA, the permissible exposure time decreases. The maximum level to which a worker can be exposed for any length of time, without ear protection, is 115 dBA. Some of the older U.S. transit systems sometimes expose communities along the transit corridors, as well as patrons and employees, to intermittent noise levels in excess of 90 dBA. In extreme cases hearing could be damaged after long-term exposure.

Speech Interference: One of the most common and annoying effects of environmental noise is interference with speech. Radio and television are affected and conversation suffers. A primary goal of car interior noise specifications is to allow conversation between passengers with normal vocal effort.

Sleep Interference: Noise often keeps people from going to sleep, and loud noises wake them up. Many researchers believe that there is a direct connection between health and noise-disturbed sleep. Quantitative relations between noise level and sleep disturbance have not yet been derived, but it is undeniable that transit system noise loud enough to interfere with sleep will contribute to community dissatisfaction.

Annoyance: By far the most common effect of environmental noise is annoyance. Noise annoyance cannot always be directly connected to such quantifiable responses as sleep or speech interference. A number of social surveys have attempted to determine how noise relates to annoyance, and to predict the degree of annoyance that will result when a community is exposed to specific levels of environmental noise. Evaluations of community annoyance provide the principal basis for most community noise criteria. Research on the subjective acceptability of noise inside passenger vehicles (see Ref. 2.17) indicates that when noise levels reach a point where it is difficult to converse with passengers 1 to 2 m away, the noise will be considered "annoying." Conversely, when the noise level in a public transit vehicle is too low, the passengers lose their sense of privacy, a condition which many people also consider undesirable. Thus, there is no gain in making vehicle interiors quieter than necessary for comfort and ease of conversation between persons next to each other. There is a definite psychological effect of noise in the vehicles; excessive rattles, bangs, screeches, and roars undermine the riders' confidence in the reliability and safety of the vehicle and system.

1.3.3 Noise and Vibration Criteria

In designing new transit facilities or fitting new noise control treatments to already existing facilities, we must decide on acceptable levels of noise and vibration. This is a difficult question, and one that does not have a single answer. Obviously, the outdoor noise levels in the downtown section of a major metropolitan area can be considerably higher than those acceptable in a quiet suburban residential area. The type of area, the human activities on which noise or vibration will intrude, the existing levels of noise and vibration, and the communities' sensitivity to noise and vibration must all be considered when developing noise and vibration criteria.

An increasingly important factor in the development of standards is the effect of local noise ordinances on transit operations. Communities are developing noise ordinances that limit noise levels at property lines to specific amounts. Although federal and state regulations prevent the application of local ordinances to many forms of transportation (e.g., motor vehicles, aircraft), the ordinances are sometimes interpreted as applying to rapid transit trains and almost always apply to transit ancillary facilities.

In the context of this handbook, areas of noise and vibration control can be divided into three basic categories:

1. Employee Noise Exposure. OSHA regulations define the permissible employee noise exposure on the basis of potential hearing impairment. Defining appropriate noise levels in terms of worker comfort and productivity is more complex.

2. Patron Noise and Vibration Exposure. Limiting noise and vibration to comfortable levels in transit cars and stations will contribute significantly to overall patron satisfaction and the general attitude of the patrons toward the transit system. Only in extreme cases is safety and hearing loss a further consideration.

3. Community noise and vibration. The radiation of noise and vibration from transit facilities into the communities along transit routes is an unavoidable consequence of rail transit (or any other form of transportation yet devised). Since it is impossible to reduce noise and vibration to completely imperceptible levels and since there will always be some community complaints about any perceptible noise and vibration levels, the selection of appropriate criteria for community noise and vibration usually involves a careful balance between economic and technical feasibility and community reaction.

The most important criteria are those applied to the community. Very restrictive criteria can be expensive, and in some cases impossible to fulfill. Too lenient standards, on the other hand, can result in considerable community reaction. Acceptable levels of community noise are usually based on surveys of the community, taking possible sleep or speech interference into account. Using this information, one can make reasonably accurate predictions of the response of a community to a new noise source, but it is important to recognize that noise at audible levels has no threshold below which there will be no annoyance. Increasing noise levels naturally increases the number of people annoyed, as well as their level of annoyance; though clearly, a feasible balance between acceptable noise and acceptable population annoyance can never be strictly or scientifically determined. An element of subjective judgement is inevitable.

A list of reference works on human response to noise and vibration, noise and vibration criteria, and community noise ordinances is included at the end of Chapter 2. Two documents of particular significance are:

- Reference 2.1, commonly referred to as the "Levels Document," a document prepared and published by the United States Environmental Protection Agency (EPA). Its purpose is to identify community noise levels "requisite to protect public health and welfare with an adequate margin of safety." The document is not intended to be a regulation, standard, or criterion, but the noise levels identified within it provide very useful reference points and have been the basis of many existing ordinances, regulations, standards, and criteria.
- Reference 2.5, "Guidelines for Preparing Environmental Impact Statements on Noise". These Guidelines are the result of deliberations of the CHABA (Committee on Hearing, Bioacoustics, and Biomechanics, Assembly of Behavioral and Social Sciences), Working Group 69, over a 4-year period. Guidelines are proposed for uniform description and assessment of noise environments potentially requiring an environmental impact statement for noise. At this time, the CHABA document is the most comprehensive guide to preparing the noise section of an environmental impact statement, and the first set of guidelines that have any universal acceptance by both the government and acoustical specialists.

A third document, of particular interest to transit systems planning new facilities, is included as Appendix B to this handbook: "Guidelines for Design of Rapid Transit Facilities," Section 2.8, "Noise and Vibration." This document was prepared by an American Public Transit Association (APTA) Noise and Vibration Subcommittee to provide guidance for transit systems developing noise and vibration criteria and specifications for new facilities. The Guidelines include noise and vibration design goals for transit vehicles and structures. Note that incorporating realistic noise and vibration specifications in the contract documents for new equipment effectively controls the noise and vibration characteristics of the equipment. Often, minor design changes or moderately more expensive ancillary equipment result in significant reductions of noise or vibration. Carefully prepared specifications encourage the inclusion of noise control features in the design; such features might otherwise fall victim to potential cost increases, and never be thoroughly evaluated.

1.3.4 Sources of Transit Noise and Vibration

This section outlines briefly the principal sources of transit system noise and vibration. Most sources are quite obvious, but some may be overlooked until problems arise.

1.3.4.1. Transit Car -- The transit cars themselves are clearly the main originators of rail transit noise and vibration. In a technical sense, transit cars are the source of whatever noise radiates from elevated structures; strictly speaking, the cars are the source of even groundborne noise since the wheels rolling on the rails generate the vibrational energy. However, this discussion will be limited to noise radiated directly from the transit cars and to the vibration of the car itself.

The motion of the wheels on the track, plus the noise generated by propulsion equipment, accounts for most of the noise radiated outwards to the community and inwards to the car passengers.

Auxiliary equipment on transit cars such as air-conditioning systems, ventilating fans, and compressors, can also make noise; the ventilation systems inside transit cars are sometimes as loud or louder than the wheel/rail noise and propulsion equipment noise.

Wheel/Rail Noise: The term "wheel/rail noise" refers to all noise that is created by the wheels rolling on the rails. There are three basic types of wheel/rail noise:

1. Impact noise. A common source is the steel wheels hitting rail joints. The "clickety-clack" associated with trains that run on tracks with jointed rails is impact noise. Rail joints and special trackwork are the principal sources of impact noise on transit systems.
2. Squeal noise. This is the high-pitched screech produced when trains go around short radius curves.
3. Roar noise. This is the normal noise of wheels rolling on straight rails without large discontinuities. Roar noise arises from the small-scale roughness of the wheel and rail surfaces.

Because of clearly identifiable components, impact noise has a more annoying character than roar noise. Replacing jointed rail with continously welded rail provides double benefits: the noise level is reduced and the character of the noise is improved when the "clickety-clack" of the joint impact is removed.

Propulsion Equipment Noise: The propulsion equipment on transit cars can be noisy. Propulsion equipment includes the traction motors, their ventilating systems, and the reduction gears. Many traction motors are self-ventilating, with cooling fans attached to the motor shaft. At high speeds, these fans are often a major source of noise. In some cases the fans are the dominant source of wayside and in-car noise at high speeds. Using a forced ventilation system with separate constant speed fans that blow cooling air through the traction motors helps to make transit cars quieter. However, care must be taken to ensure that the forced ventilation system itself is sufficiently quiet or it may not provide any overall acoustical benefit.

Auxiliary Equipment: "Auxiliary equipment" refers to all equipment on the transit car except propulsion equipment. Heating, ventilating and air-conditioning equipment, braking systems, compressors, motor generators or alternators, choppers, warning horns, door-opening mechanisms are all considered "auxiliary" for present purposes. Since much of this equipment operates whenever a car is in service, even if it is sitting at the station platform, the restrictions placed on auxiliary equipment noise are necessarily tighter than those placed on propulsion equipment or for overall noise levels at operating speeds.

Auxiliary equipment can also be a significant source of car vibration. For example, improperly mounted compressors can vibrate the car body components quite noticeably and sometimes cause discomfort inside the transit car.

1.3.4.2 Elevated Structures -- The noise radiated from lightweight steel elevated structures is a major community problem in many cities with older transit systems. The vibration of the rails transmits downward through the elevated structure, turning it into a large and extremely efficient sounding board. The vibration of the wheels and rails radiates noise and when some of the vibrational energy in the rails is transmitted through the steel panels of the elevated structure, the overall noise level is significantly amplified. Older open-deck or truss structures have little shielding effect on noise.

On modern transit systems, aerial structures and bridges are either built entirely of concrete or have a concrete deck supported by a steel girder. Much less noise is radiated by these types of structures. If properly designed, the overall noise levels adjacent to modern aerial structures compares to that of at-grade, ballast-and-tie track, normally the quietest track structure.

One problem observed in the modern aerial structures, especially composite structures with steel girders, is a tendency for the structures to radiate low-frequency sound. Although the low-frequency noise is not usually a problem outdoors, it can be a problem inside nearby buildings. Most buildings effectively reduce high-frequency noise transmitted from outside to inside, however they deal with low frequencies less efficiently. The result is that the low-frequency noise can become the dominant component of the train noise heard indoors. When low-frequency noise radiation is a problem, it can be controlled with proper treatment of the aerial structures.

1.3.4.3 Groundborne Noise and Vibration -- Groundborne noise and vibration is an environmental problem that has been often overlooked until recently. Forces caused by the wheels rolling on the rails are transmitted through the rail support system into the trackbed of the transit structure, causing movement within the transit structure that radiates vibration into the surrounding soil, and finally to nearby buildings. Occupants of an affected building perceive vibration in two ways. First, the vibration may be felt directly as mechanical motion. Second, the vibration of building components radiates audible airborne sound, identifiable as a low-frequency rumble. This latter, audible perception is more common than the former, kinetic one.

Groundborne noise and vibration are most prevalent near subway tunnels, but problems have arisen with all types of transit structures. One reason that they are less common for at-grade and aerial transit structures is that airborne noise, radiated directly from the transit train, is usually dominant and masks that caused by groundborne vibration. Where the airborne noise level is low or reduced by sound barriers, the effects of groundborne noise and vibration become more noticeable.

People who can feel groundborne vibration from transit trains often fear that their houses will be damaged by the vibration. Although the groundborne vibration is, in virtually all cases, well below criteria for damage risk to residential buildings, transit systems should seek some means of alleviating these fears.

1.3.4.4 Yards and Shops -- Storage and inspection yards and maintenance shops are often a source of irritation to the surrounding community. This is particularly true when the yards and shops are located in residential or semiresidential areas. In storage and inspection yards, the trains themselves make most of the noise that offends the community.

Train noises include normal train noise, as cars are moved from point to point in the yard; noise caused by auxiliary equipment on cars waiting in storage areas; wheel screech, as trains go around short radius curves; clicks and pings as cars go over joints and through switches; air release noises, warning horns, and the coupling and uncoupling of cars. Machines for washing the cars, impact tools and other maintenance machinery, workers shouting, loudspeakers, announcements, all contribute to the yard noise that can be intrusive to a residential area.

1.3.4.5 Station and Line Ancillary Equipment -- The primary sources of noise from ancillary equipment at stations, ventilation shafts, and substations are:

- Large tunnel ventilation or emergency fans, sometimes located at the ends of subway stations and sometimes at subway ventilation shafts.

- The hum from transformers, and the cooling fans for transformers, at electrical substations.

- Various components of ventilation or air-conditioning systems, such as station fans, chillers, compressors, and water towers.

- Train noise transmitted through ventilation shafts.

The placement of the equipment largely determines its potential for causing annoying noise levels. Virtually all mechanical equipment in train stations, if incorrectly designed or installed, can be the source of noise problems. This includes elevators, escalators, and even automatic fare collectors. Surface stations are also subject to outside traffic noise, particularly when they are located in highway medians or adjacent to railroad lines.

Designers should recognize, when locating fan and vent shafts for subways, that the shafts provide paths by which airborne noise reaches the outdoors. If the shafts are improperly placed, the noise from trains in the subway or fans used for ventilation can be quite intrusive in nearby buildings. When a shaft must be placed near buildings, acoustical treatments can be installed that control the fan and train noise radiated from the shafts.

1.3.4.6 <u>Construction Noise and Vibration</u> -- Construction noise should not be overlooked in evaluations of noise caused by new rapid transit projects. Many construction activities can be very noisy, and some, such as blasting and impact pile driving, can create ground vibration levels sufficient to damage nearby buildings. With modern, well-muffled equipment, the noise from construction activities can be considerably lower than most people expect. In most cases, some intrusion from construction noise is unavoidable. However, since the source of the intrusion is temporary, most communities will accept levels of construction noise above the normal ambient noise level provided that control measures are used to keep the noise below appropriate noise limit criteria.

1.4 OVERVIEW OF URBAN RAIL NOISE AND VIBRATION

Since noise and vibration are not directly related to the economics and efficiency of system operations, they are often neglected until problems develop. It is usually more efficient as well as more economical to anticipate acoustical problems and incorporate control solutions in the original transit system design; but when many of the older transit systems were constructed, only cursory consideration, if any, was given to noise and vibration. The results are evident; urban rail transit systems have developed a reputation for exposing both the wayside communities and the transit system patrons to very high levels of noise.

At present, public awareness of all types of environmental pollution, including noise and vibration, is constantly growing. Federal and state agencies require environmental impact reports for virtually all significant urban rail transit projects, and an evaluation of the impact of noise and vibration is an integral part of the environmental review process. New transit facilities are subject to the legitimate environmental concerns of the community, and older transit systems receive a growing number of complaints about noise and demands for changes.

The purpose of this section is to present a general outline of urban rail noise and vibration problems; it includes discussions of problems typically encountered by urban rail transit systems; ways by which potential, or existing, problems can be evaluated; and specific approaches to noise and vibration that should be incorporated in the design and construction phases of new transit facilities and in the operation of existing ones.

1.4.1 Typical Noise and Vibration Problems

A basic premise of noise and vibration control is that for noise or vibration to be a problem it must be heard or felt by someone. Regardless of the intensity of a noise, if no one hears it or feels its vibration there is no need to consider control features. The obvious exception to this general rule is the possibility of damage caused by vibration. With the exception of that caused to buildings during construction, potential damage is usually limited to transit facilities, particularly through fatigue failure of components on transit cars and sometimes way structure components.

Since we are primarily concerned with transit system noise and vibration that affects humans, we can classify the typical problems according to three groups of people affected:

1. Communities in transit corridors. Minimizing intrusion to the adjacent community is usually the highest priority of transit systems.

2. Transit system patrons. Although reliability, convenience, and safety are more important to patrons than exposure to noise and vibration, the acoustical environment has a strong influence on the attitude of patrons toward the system.

3. Transit system employees. Like virtually all other U.S. industries, transit systems must meet government standards setting the maximum noise levels to which employees may be exposed. These criteria are designed to minimize hearing loss and do not consider annoyance or impairment of concentration. Train operators and conductors on most systems, and station attendants on all systems, are rarely exposed to noise levels above the government standards. However, excessive noise exposure is possible for shop and line maintenance personnel.

Transit systems contain a number of potential noise sources, many of which can affect all three of the receiver groups identified above. In order to ensure appropriate acoustical characteristics in new and existing transit facilities, the sources of noise and vibration and their effects on people must be identified.

1.4.1.1 Community Noise and Vibration Intrusion -- Communities are exposed to two forms of intrusion: airborne noise from trains and ancillary equipment, and groundborne noise and vibration. Groundborne noise and vibration is transmitted through the ground, from the transit structure, to nearby buildings. The effect of walls, floors and ceilings vibrating produces a low-frequency rumble; this sometimes causes windows and dishes to rattle, producing more noise (secondary radiation).

Table 1-1 lists the sources of noise, in approximate order of importance, that often affect communities. The most important source is the airborne noise radiated directly from the transit vehicles operating on surface or elevated tracks.

1.4.1.2 Patron Noise and Vibration Exposure -- Patrons are exposed to noise in the station while waiting for trains, as well as while on the trains. They are exposed to significant vibration only while riding on the transit car. Very rarely is vibration in the station area of sufficient magnitude to cause concern. Table 1-2 summarizes the principal sources of patron exposure to noise and vibration.

Modern transit systems, though sometimes causing community intrusion and patron discomfort, rarely produce noise levels high enough to cause hearing damage. However, older systems sometimes expose the public to noise levels which have the potential for causing hearing impairment after long exposure. For example, a recent survey of elevated structure noise in the U. S. estimated that over 100,000 people living near the more than 95 km of elevated structure in New York are exposed to an L_{dn} in the

TABLE 1-1 COMPONENTS OF COMMUNITY NOISE AND VIBRATION

Component	Comments
VEHICLE NOISE	
Wheel/Rail Noise	Noise radiated directly by the wheels and rails. Wheel/rail noise is often broken down into roar noise, impact noise, and wheel squeal noise.
Propulsion Equipment	Noise from traction motors, reduction gears, and motor cooling fans. Usually, propulsion equipment noise is greater than wheel/rail noise at high speeds and comparable at medium speeds.
Auxiliary Equipment	Includes compressors, motor generator or alternator sets, HVAC equipment, braking systems. All add to overall train noise, but it is usually not an important component of community noise except near yards.
ELEVATED STRUCTURE	
Lightweight Steel Structures	Noise created by vibration of the structure. A very severe noise problem exists with many of the older lightweight structures.
All Concrete and Composite Concrete and Steel Aerial Structures	The overall noise level consists of two components, the noise radiated directly from the transit cars and the structure radiated noise. Typically car noise dominates at high frequencies and structure-radiated noise dominates at low frequencies.
GROUNDBORNE NOISE AND VIBRATION	Mechanical vibration of transit structure originating at the wheel/rail interface and transmitted to adjacent buildings. More common for subways but can also be a problem with surface and elevated structures. Airborne noise in subways does not contribute to groundborne noise and vibration.

TABLE 1-1 COMPONENTS OF COMMUNITY NOISE AND VIBRATION (continued)

Component	Comments
CONSTRUCTION ACTIVITIES	Most construction equipment and activities have the potential of creating intrusive community noise and vibration.
VENTILATION SHAFTS	
Fans	The airborne noise of station and tunnel ventilation fans can reach the community through vent shafts and ducts.
Train Noise	The shafts provide an airborne path for train noise to reach the community.
TRACK MAINTENANCE EQUIPMENT	Rail grinding, ballast cleaning, etc. Track maintenance is often performed at night when residential communities are most sensitive to noise.
SUBSTATION TRANSFORMERS AND COOLING FANS	Substation noise is only a problem in the immediate vicinity of the substation.
YARD OPERATIONS AND ACTIVITIES	This includes train movements and equipment, horns, PA systems, trains coupling and uncoupling, maintenance activities.
STATION HVAC SYSTEMS	
Mechanical Equipment	The HVAC equipment noise transmitted to the outdoor can be a community noise problem.
Cooling Towers	Since cooling towers can be relatively noisy and are located outdoors they can be a noise problem.
PA Systems	Public announcements in above-ground stations can produce sound levels intrusive to neighbors.

TABLE 1-2 COMPONENTS OF PATRON NOISE AND VIBRATION EXPOSURE

Component	Comments and Examples
IN-CAR NOISE	
Wheel/Rail Noise	Noise resulting directly from vibration of wheels and rails.
Propulsion Equipment	Traction motors, reduction gears, motor cooling fans.
Auxiliary Equipment	Compressors, HVAC systems, brakes, door opening mechanisms.
Elevated Structures	The noise radiated from the vibration of a lightweight elevated structure can increase the car interior noise 10 to 20 dB.
Subway Tunnels	Not a noise source, however when trains run in acoustically untreated subways, the car interior noise will increase 5 to 15 dBA compared to ballast-and-tie tracks on the surface.
Concrete Trackways	Train noise reflecting off concrete aerial structure decks or at-grade concrete slabs can raise car interior noise 3 to 4 dBA.
Trackway Vibration	Vibration of trackways [e.g., floating slabs] radiate noise that is sometimes audible inside transit cars.
STATION PLATFORM AND OTHER PUBLIC AREAS	
Transit Vehicle	
- Auxiliary Equipment	For stationary and slow moving trains, the dominant noise source is the auxiliary equipment [HVAC systems, compressors, brakes, etc].
- Propulsion Equipment - Wheel/Rail Noise	Because of the relatively slow speed in stations, wheel/rail and propulsion system noises are relatively less important than in most other situations. However, express trains passing through stations can create high levels of propulsion and wheel/rail noise.

TABLE 1-2 COMPONENTS OF PATRON NOISE AND VIBRATION EXPOSURE (continued)

Component	Comments and Examples
Station and Tunnel Ventilation Fans	Fan noise can propagate along tunnels or other airborne paths to station platforms.
HVAC Equipment	Noise from heating, ventilating and air-conditioning equipment can propagate via ducts to public areas.
Mechanical Equipment Rooms	Depending on proximity to platforms and other public areas, mechanical equipment rooms can be significant noise sources, via access doors, louvers, etc.
Road and Railroad Traffic	Noise from traffic can be significant in open stations near busy highways [especially stations in highway medians] and near main line railroads.
Station Vertical Circulation Equipment	Elevators and escalators are potential noise sources in stations.
Fare Collection Systems	Depending on design. There is some chance of noise from fare collection systems.
Crowd Noise	In highly reverberant stations crowd noise can build up to high levels.
PA System Announcements	A system with reasonable intelligibility and volume adjustments is rarely classified as a noise problem. Systems with poor intelligibility can become significant noise problems.

range of 85 to 90 dBA (Ref. 7.1). This exposure is caused by the high levels of wayside noise near steel elevated structures. Allowing for the attenuation as the sound passes through the walls of residential structures, the indoor levels of L_{dn} would be expected to be in the range of 65 to 75 dBA. In an EPA examination of the "levels of environmental noise necessary to protect public health and welfare," it is concluded that an average 24-hour L_{eq} of 70 dBA will produce no more than 5 dB noise-induced hearing damage over a 40-year period (Ref. 2.1). The conclusion is that for the special case of steel elevated structures, the wayside noise levels may be high enough to present at least some potential of noise-induced hearing loss.

1.4.1.3 <u>Employee Noise Exposure</u> -- Employee noise exposure is not usually a major problem for rapid rail transit systems. If the system provides a comfortable acoustical environment for the patrons, then train operators, station attendants, and others in and around the trains are equally unlikely to suffer from excessive noise. The only times transit employee noise exposure might exceed acceptable limits is in the repair shops and during track maintenance. It is important to be aware of employee noise exposure during construction activities. Since the construction is usually by outside contractors, the noise exposure of their employees is not usually the direct responsibility of the transit system. OSHA does inspect construction sites.

Transit employees are exposed to potentially high levels of noise around storage and maintenance yards, and near particularly loud equipment in maintenance shops. Turnaround tracks in regular use that cause high levels of wheel squeal often lead to significant community annoyance and expose employees stationed near the turnaround to high noise levels.

Shop activities also present problems. The repair shops of transit systems have equipment similar to other types of industrial operations. Normal equipment items, such as lathes and milling machines, are noisy, but rarely is the noise intense enough to cause concern. Pneumatic and impact tools, and airbrake test fixtures can create noise in the hearing damage range if they are not fitted with proper mufflers. It is important to be aware of potential problems and to recognize when a particularly noisy piece of equipment will be operating in the shops. Careful scheduling of personnel can prevent excessive exposure. If the noisy operation is performed only on an irregular basis, supplying workers with hearing protection, such as ear plugs or ear muffs, is often enough.

Finally, significant employee noise exposure can occur during various track maintenance activities. Track installation and prefabrication, rail grinding, ballast cleaning and treatment, and the inspection and repair of track are all potentially noisy activities. When maintenance is necessary during revenue operations, noise from the train passbys also produces high noise levels. Noise levels can be high on outside ballast-and-tie track, and in highly reverberant subways the buildup of noise from reflections off the walls can increase the noise levels an additional 8 to 12 dBA.

1.4.2 Evaluation of Noise and Vibration Problems

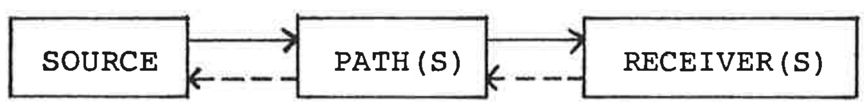
Evaluation of existing or anticipated noise and vibration problems involves three basic steps (Much of the material in this section has been adapted from Ref. 1.1).

1. Consider the noise environment under existing or expected conditions, with and without the problem source. This step will generally require measurement or estimation of the source strengths, plus the level of existing ambient noise or vibration without the problem source. When evaluating anticipated problems, either empirical data or engineering calculations must be used to estimate the characteristics of the noise or vibration source.

2. Determine the acceptable level of noise or vibration. This step involves establishing a criterion that can be used under various conditions and for various purposes. The development of appropriate criteria for transit system noise and vibration is discussed in some detail in Chapter 2.

3. Design noise or vibration control treatments to obtain an acceptable environment. The difference between the levels obtained in step 1 and the acceptability criterion of step 2 is the reduction that must be provided by the control treatment(s).

For convenience, the transmission of sound or vibration from a source to a receiver can be represented by the diagram below.



A source can be any source, or number of sources, of vibratory energy. For example, wheel/rail noise results from eight wheels for each car rolling on the rails. There are also many different paths by which the vibratory energy can be transmitted from the source to the receiver. The receiver can represent a single person, an entire community, or even a piece of machinery sensitive to vibration or noise. The broken lines in the diagram indicate that the source, path and receiver are not always independent; there is frequently interaction among the elements.

In the case of transit noise, it is rarely necessary to consider effects other than those on humans; generally, if there is no human (or sometimes animal) receiver to be affected by the noise, there is no noise impact. In contrast, the major impact of vibration is often fatigue failure of transit car or way structure equipment. Nearby buildings should be evaluated for potential problems from the impact of transit system vibration.

Some of the more common sources of transit system noise and vibration were discussed in Section 1.3.4. Note that this discussion of sources is not exhaustive. Almost any mechanical or electrical equipment can produce noise and vibration impact. It is important, during planning and design stages, to consider potential noise and vibration sources and paths and to take reasonable precautions to minimize their impact. Even equipment not normally associated with noise and vibration should be evaluated, to ensure that unique problems will not develop. Escalators, for instance, deserve scrutiny during the transit system design. Because their drive motors are usually located in wells or basements relatively removed from occupied areas, escalator noise is rarely noticeable. However, some modern escalator designs, such as those installed in the WMATA Metro stations, have motors located directly under the stair track so sound is transmitted from the escalator motors directly into the stations.

1.4.3 Specific Considerations -- New Facilities

Noise and vibration problems that should be considered in the various phases of a new transit system are discussed in this section. Although the discussion is oriented toward new transit systems, or new lines on existing transit systems, it is also valid for existing transit systems that are planning major repairs and upgrading of existing facilities or purchases of equipment, such as new transit cars.

The process from the initial planning of a new transit system to revenue operation has been divided into seven phases. These phases have no distinct boundaries -- as many as seven phases could be initiated simultaneously on one transit system project.

Since the time period between the initial system concept for a new transit system and its actual revenue service operation may be several years (or in some cases decades), some of the phases discussed below will likewise take years to complete.

1.4.3.1 Project Initiation and Planning -- The project initiation phase begins with the recognition of a need for improvements in regional transit facilities, and a decision is made to study various possibilities. Although most people involved in this phase have very definite ideas about what types of transit they prefer, the specific selection of a transit mode is not formally made until Phase 2.

Phase 1 involves developing regional support for transit improvements and arriving at a general consensus about the best transit mode for the region. Until recently, the transit mode chosen was almost always some form of highway. Lately, suggestions for regional transportation improvements almost always include rail transit and/or light rail transit as viable alternatives. Because the activity during this phase is largely political, questions of noise are usually secondary. Usually questions of acoustics first arise in preliminary, qualitative comparisons of different transit modes. The fact that rail transit will generate less noise in a community than either a highway or a bus system, handling the same rider capacity, becomes a factor that should be considered in the decision.

Traditional steel wheel, steel rail rapid transit and rubber tire rapid transit must also be considered. Although many planning groups favor rapid rubber tire transit because they think it produces less noise and vibration, in virtually all cases the patron and community noise exposure from modern steel wheel and rail rapid transit is as low or lower than that from rubber tire systems. Groundborne noise and vibration, however, are lower in rubber tire systems because of the vibration isolation properties of rubber tires.

Decisions made at this point are often based on preconceptions about transit noise. When an important decision regarding the transit system conceptual design is required, and noise or vibration is an important factor, at least a preliminary investigation of other transit system experiences with noise and vibration should be initiated. Often relatively straightforward, inexpensive designs can eliminate what a planning group feels to be a major problem. Some objections may be based on out-of-date information about older rail transit systems. Tremendous improvements in the noise and vibration levels that patrons and the community are exposed to have been made on modern systems; education of the planning groups in this regard is very important.

1.4.3.2 <u>Preliminary Evaluation/Alternatives Analysis</u> -- Once the project has been formally initiated, the preliminary evaluation phase begins. This usually involves development of the basic transit system proposal and various alternatives, and an analysis of their relative costs and benefits. The analysis of alternatives can include an environmental analysis, the preparation of an Environmental Impact Statement, and numerous public hearings.

Vehicle type and technology are generally selected during this phase. The planning committee should consider desirable noise and vibration limits when making its selection. Evaluation of technology takes into account the relative acoustical impacts of various classes of vehicles (e.g., light rail <u>vs</u> heavy rail <u>vs</u> rubber tire vehicles); once the type of car has been selected, vehicular configuration is chosen. Decisions can seriously affect acoustical impact. For instance, the choice between self-ventilation and forced ventilation of the traction motors can have a major impact on the wayside noise levels. This type of detail is not normally considered during this phase, but should be.

In the selection of transit corridors and the preliminary system layout, acoustical impact should be a major consideration. Clearly, the transit structure chosen (e.g., subway or aerial structure) will affect the acoustical environment of the area. But other, less obvious choices can also determine the amount of local acoustical impact. The following items deserve consideration:

- Station locations. If stations are placed in quiet residential areas, the station activities and extra automobile and bus traffic can cause noise impact.

- Alignment. Making maximum use of existing railroad right-of-ways and highway medians can reduce noise impact.

- Subway construction type (for example, deep tunnel vs cut-and-cover).

- Required construction activities. Tunneling, for instance, is quieter than construction of most other subway or surface structures.

- Land use along the proposed transit right-of-way. Obviously, noise will provoke less resentment in a high density industrial area than in a low density residential area.

1.4.3.3 Preliminary Engineering and Design -- Once the basic decisions are made regarding the type of transit system, and the environmental review process is completed, the preliminary engineering and design phase begins. In this stage a final Environmental Impact Statement may be prepared, including a detailed analysis of the noise and vibration impact along the selected transit corridor. Many decisions in this phase should consider noise and vibration; some are very obvious, some less so:

- Noise and vibration design goals must be developed for the transit facilities and equipment.

- The final EIS is submitted using the design goals as the basis for defining potential impacts.

- Decisions on track alignment are implemented, including the location of crossovers. Unusually noisy trackwork (such as crossovers) should be located in areas less sensitive to noise and vibration.

- Sites for yards and shops are determined. Again, sites that are not particularly sensitive to noise are preferred.

- Criteria and/or design directive documents must be drafted covering station acoustics, fan and vent shafts, ancillary equipment, and construction noise and vibration.

- Specifications are set for vehicle noise and vibration performance and reduction features, such as use of car skirts, resilient or damped wheels, and types of primary suspension.

- Aerial structure conceptual designs should be carefully evaluated to ensure that structure-radiated noise will not be a problem.

- Conceptual designs for specific noise and vibration abatement procedures (e.g., floating slab design) are determined.

- Locations where abatement procedures such as floating slabs or sound barriers will be required are determined on a preliminary basis.

- Prototypes of noise and vibration control features are designed and, if possible, tested.

- Unique local conditions that may influence noise or vibration (e.g., soil conditions and buildings that enhance groundborne vibration) must be investigated.

The last two items are often overlooked. When innovative designs are proposed for the transit system, scale model or full-scale tests of the noise and vibration properties should be made. For example, a full-scale model of the floating slab vibration isolation system used at WMATA was tested to ensure that the system would perform as anticipated. Aerial structures are an example of a design requiring special attention. Modern concrete aerial structures rarely create structure-radiated noise problems, but multitrack, lightweight concrete aerial structures can radiate significant levels of noise. Any new aerial structure design should be analyzed for potential problems from noise radiation. If a theoretical evaluation suggests significant structure-radiated noise, a model or prototype test should be performed to positively identify the noise radiation characteristics.

The last recommendation on the list refers specifically to the need for evaluating the wave propagation properties of the soil around proposed subways, and sometimes aerial structure and surface tracks, to discover any potential problems from groundborne noise and vibration. This has not been generally done in the past although recent experience indicates that groundborne noise and vibration is highly dependent on local geological conditions. For example, the Toronto Transit Commission (TTC) received complaints of groundborne noise from residents more than 200 m from the subway structure; whereas the Bay Area Rapid Transit (BART) has received no complaints regarding groundborne noise and vibration. Also, as discussed in Chapter 8, recent studies indicate that the design of the transit car trucks can have a strong influence on the levels of ground vibration. This possibility should be investigated during the preliminary design and engineering phase.

1.4.3.4 Final Design -- Many aspects of noise and vibration must be considered during the final design stage, and necessary abatement procedures should be developed. An acoustical specialist will usually handle technical designs and procedures, but transit personnel should make sure that this work is adequately reviewed. The following are important final design considerations:

- Final alignment. The final alignment, including the plan and profile of the trackway, should recognize potential acoustical impacts. Efforts should be made to keep the subway portions of the system a reasonable distance from buildings. A buffer zone of 30 to 60 m is desirable but certainly not likely in a dense urban setting; the minimum distance should not be less than 12 to 15 m (also not likely in narrow city streets). Crossovers and other special trackwork should be placed in areas that are not sensitive to noise (e.g., highway medians) if at all possible.

- Design of yards and shops. Noise from storage and inspection yards and maintenance shops can be a major community problem. These are often located in residential areas at the end of a transit line, and most communities are not particularly pleased that an "industrial" activity is moving into their neighborhood. Often, noise impact is one of the issues used to express general dissatisfaction. Hence, careful consideration should be given to locating yards and shops. When they must be located in residential areas, the facilities should be carefully located to minimize noise in the adjacent community.

- Fan and vent shafts and substations. Ancillary equipment should be located where they will create a minimum of noise impact. Once the sites have been selected, noise should be evaluated and necessary noise control features specified.

- Specifications for noise and vibration control designs. Once the designs for abatement measures have been approved, specifications for materials and installation should be prepared -- such as specifications for the resilient pads for floating slabs and specifications for acoustical absorption treatment or attenuators to be used in fan and vent shafts.

- Design section analysis. As part of the final design of each design section, thorough acoustical analysis should be made, and where previously determined noise or vibration criteria will be exceeded, control procedures should be specified.

- Station acoustics. When the architectural design of the stations is completed, plans for the interior surface finishes should be reviewed to ensure sufficient acoustical absorption and the appropriate placement of absorptive materials.

- Review of stations' heating, ventilating and air-conditioning (HVAC) design. Although they do not generally constitute a major problem, the mechanical HVAC equipment can create noise problems in stations and the adjacent community. To avoid this, the mechanical equipment design should be reviewed .

- System maintenance. In an operating system, the maintenance of way structures can create serious community noise problems. In the final design stage the potential acoustical impact of anticipated maintenance operations should be considered, and an effort made to develop configurations and procedures that will minimize its effect on the community.

1.4.3.5 <u>Construction</u> -- Once the construction phase of a transit project has been reached, acoustical considerations apply to relatively few areas. During construction, an effort should be made to monitor compliance with the construction noise and vibration specifications and/or local noise ordinances. Normally, if the specifications and the ordinances are observed, there will be few community complaints. Of course, complaints can always occur. Section 1.4.5 suggests strategies for handling such complaints.

Normally, construction contracts include specifications limiting the noise that can be created by the construction activities. However, there is considerable variation in the amount of detail in the specifications and the manner in which compliance with the specification is monitored. Sample noise specifications for construction noise are included in Appendix C.

Contractors who are unfamiliar with the noise control treatments to be installed by the transit facility may sometimes install them incorrectly. To avoid this problem, the installation of special noise and vibration treatments should be carefully inspected to ensure that the installed materials will operate as designed. Recent specifications for construction noise at NYCTA and MBTA have required that the contractor hire an acoustical consultant to monitor the construction noise.

1.4.3.6 <u>Operational Testing</u> -- Once sections of the transit system have been completed and operational testing of trains and other facilities begins, acoustical tests should be performed. There are several reasons for this testing: evaluation of how well facilities have complied with specifications; investigation into unanticipated problems; general retrospective analysis of design decisions and their environmental impact; and development of information that can be used for future designs.

Although it is often very difficult to solve noise and vibration problems after the system is constructed, a testing program that identifies problems before serious community complaints occur will allow time to devise a strategy for handling the complaints. In some cases problems can be rectified before many complaints arise.

Testing to evaluate noise and vibration projections made in the preliminary phases and to measure the effectiveness of the abatement designs is often overlooked. In many cases projections were based on extrapolations of information from similar, or somewhat similar, facilities at other transit systems, or from engineering estimates. Although such projections are usually fairly accurate, important factors can be overlooked, and variations between transit designs can distort projections. An early measurement program will eliminate much uncertainty and, in some cases, will allow design improvements on future sections of the system that reduce noise and vibration and/or costs.

Prior to extensive operational testing, the initial surface roughness of freshly installed rail should be removed by grinding. Before the mill scale and rust have been worked off of new rail, significant increases in both noise and vibration have been observed at several transit systems. Grinding the new rail will eliminate this problem, although once the rail has been in service for some time (approximately 6 months is required), the initial surface roughness will wear off. The trouble is that by this time the adjacent communities may become sensitized to the noise or vibration and be dissatisfied even after the rail has been smoothed.

1.4.3.7 Revenue Service -- Before revenue service begins, damage to adjacent structures caused by construction should be evaluated (and claims settled) so that operating trains are not blamed for the damage.

Once a transit system has reached the stage of revenue service, noise and vibration considerations include evaluation of complaints; anticipation of possible increases in noise and vibration as the facilities deteriorate; scheduling of way structure maintenance to minimize disturbance to noise-sensitive areas; and scheduling of maintenance procedures, such as wheel truing and rail grinding, to avoid excessive noise from rail corrugations or wheel flats. Problems due to wheel flats and rail corrugations do not usually develop until revenue service begins, and both greatly increase wayside noise, groundborne noise and vibration, and in-car noise and vibration. Problems arising from such increases should be anticipated before revenue service begins. Note that the problem can also appear during the operational testing phase, especially during operator training.

Changes in operational procedures often affect noise and vibration levels. Most of these changes will be trivial, but the potential for noise increase should be considered whenever operational changes are contemplated. For example, if tunnel ventilation fans are used on a regular basis instead of the emergency use for which they were designed, they may require acoustical treatment.

1.4.4 Specific Considerations -- Existing Facilities

Existing transit systems are usually well aware of noise and vibration problem areas and have a good idea of the impact created by the system. Specific concerns generally relate to system modifications, and attempts to anticipate increases in noise or vibration due to the deterioration of system components.

Public awareness of environmental issues is growing rapidly; local communities increasingly request that transit systems make efforts to reduce acoustical impact. However, mitigation measures are often prohibitively expensive, and it is often impractical to make significant acoustical improvements in existing transit facilities. If however, replacement or renovation has already been scheduled, acoustical improvements that should particularly be considered include:

- Inclusion of acoustical absorption material in renovated stations;

- Use of welded rail when track must be replaced;

- Use of resilient vibration isolation systems when subway trackways (and surface tracks in special cases) are upgraded;

- Inclusion of noise and vibration specifications in the purchase contracts for new vehicles to ensure that the new cars create less noise and vibration than the old cars;

- Replacing solid steel wheels with damped or resilient wheels that reduce wheel squeal noise;

- Use of vibration isolation or damping treatments on steel elevated structures to reduce structure-radiated noise;

- Use of sound barrier walls along at-grade or aerial track segments to reduce wayside noise;

- Use of absorption material to reduce reflected sound in subways or outdoors.

Particularly severe problems will often require a study of abatement measures and the development of financially viable solutions. Unfortunately, often the only real solution is complete replacement of the offending facility.

Transit systems should cooperate with local communities that are in the process of preparing noise ordinances. As discussed in Chapter 2, noise ordinances interpreted to apply to transit facilities significantly affect the transit system. To avoid any unforeseen problems, the transit system should work with the communities involved to ensure that the ordinance is one that is compatible with the transit system characteristics.

A problem more common to airports than to transit systems is community encroachment upon noise-producing areas of activity. There are a number of cases of residential developments being built at the end of airport runways, and the buyers eventually suing the airport authority for noise pollution damages. Since similar problems can occur near transit systems, it is advantageous to monitor proposals for development near transit facilities to promote compatibility with existing conditions.

1.4.5 Response to Community Complaints

Normally, transit systems are built in corridors of high population density. It is impossible to design new transit systems that are totally inaudible, and a new transit system will usually result in significant adverse noise and vibration for at least a small portion of the population living in the transit corridor. Some complaints must be expected. The manner in which these complaints are anticipated and handled will greatly influence the development of a positive image for the transit system.

The experiences of airports may be useful here. Airports are much noisier than rail rapid transit. Airport noise typically affects entire neighborhoods under approach and take-off patterns, while railway noise is limited to a relatively small (less than 100 m) distance from the railway. Nonetheless, transit systems can use similar methods for alleviating community dissatisfaction. Airport agencies have found that the most effective strategy has been to include the community in decisions regarding noise-producing facilities. Community involvement brings together all the people who can contribute to a solution -- federal agencies, local and state governmental entities, transit authorities, interest groups, interested and affected individuals -- in a decisionmaking process aimed toward reducing or mitigating transportation-related environmental problems. Community involvement can take many forms: public meetings, neighborhood offices, advisory groups, workshops, newsletters, study groups, public hearings. In effective community involvement the various methods are combined into a total program or process designed to ensure that the concerns and needs of all the participants are considered when decisions are made. The environmental impact studies and public hearings that are required for most new transit facilities can provide a good forum for community participation and input.

Transit complaints can be divided into two categories: complaints from patrons about excessive noise and vibration in cars or stations; and complaints regarding excessive noise and vibration in the community at large. Proper design and adequate maintenance will ensure that noise rarely reaches levels sufficient to generate hearing loss or other serious physiological problems. The one possible exception is the pressure transients that are caused by high speed trains entering and exiting tunnels, or by trains passing vent shafts or other trains within tunnels. Although we are not aware of any case where pressure transients have been severe enough to cause hearing damage, it is not uncommon in small bore tunnels with high train speeds for pressure transients to create discomfort and sometimes even pain. Obviously this can lead to complaints. The control of pressure transients is discussed in Chapter 13.

The following attitudes are recommended when dealing with the public or with government officials:

- Be direct about possibilities and clearly state limitations.

- Maintain a willingness to listen, a desire to achieve a real solution, and genuine sympathy with the complainants.

- Develop the communication skills necessary to go beneath the surface of a complaint and expose the real problem. Fears of decreasing property value and property damage from transit-created building vibration are common, and commonly unspoken.

CHAPTER 1 REFERENCES

1.1 Cyril M. Harris (Editor), Handbook of Noise Control, Second Edition, (McGraw-Hill Book Company, NY, 1979).

This widely available book covers a broad spectrum of topics. Each chapter has been prepared by a different author. This is a valuable reference for both the specialist and non-specialist in acoustics, however, when designing specific noise control measures, other references will also be required. Much of the introduction in Chapter 1 of this Handbook is based on material in this reference.

CHAPTER 2

2. ACCEPTABILITY CRITERIA

2.1 NOISE

Defining appropriate acceptability criteria for transit noise requires determining both an appropriate measurement scale and the appropriate levels. The problem of predicting human response to a new source of environmental noise has been investigated in several recent studies. One initial question was the correlation between human annoyance and various descriptors of community noise. It is only in the past 10 to 15 years that the A-weighted level has been generally accepted as an appropriate measure of community noise; changing from A-weighting to other weighting curves (e.g., D-weighting and E-weighting) that more closely approximate the response of the human ear to noise is still being debated. The other weighting curves have been shown to give only marginal improvement of the correlation between human response and sound level, and it is unlikely that rail transit criteria, standards, or measurements will incorporate weighted levels other than A-weighted in the foreseeable future.

The A-weighted sound level is an instantaneous level; when the sound level is constantly changing it is necessary to use statistical levels (e.g., the sound level exceeded a certain percentage of time), an average level, the maximum level, or some other descriptor to characterize the noise environment accurately. The appropriate descriptor often depends on the type of noise source being characterized.

For community noise, the most commonly used descriptors are:

- L_{10} and L_{50}: These descriptors, widely used to characterize traffic noise, are the A-weighted levels which are exceeded 10% and 50% of the time. However, since trains are an intermittent noise source, generally occurring less than 10% of the time, these are very poor descriptors for train noise.

Energy Averaged Sound Levels: Descriptors such as Equivalent Sound Level (L_{eq}), Day-Night Average Sound Level (L_{dn}), and Community Noise Equivalent Level (CNEL) are based on energy averaged sound levels (see definitions in Appendix A and Glossary). L_{eq} includes no penalty factors for the time of day, while L_{dn} includes a 10 dB penalty during nighttime hours, and CNEL a 10 dB penalty in the nighttime hours (10 P.M. to 7 A.M.) and a 5 dB penalty in the evening hours (7 P.M. to 10 P.M.). Descriptors based on energy averaging are appropriate for characterizing a wide variety of noise environments and are highly accepted for describing community noise resulting from road traffic, trains, aircraft, and other sources of community noise. The great advantage of energy averaged descriptors is that they can be used when several different types of noise sources (e.g., train noise and traffic noise) contribute to the noise environment.

Maximum Levels or Peak Levels: The maximum level is generally used to define the characteristics of a specific noise source within the environment and not to describe the noise environment. For a train passby, the quantity will be the average maximum level during the train passby. The terms maximum level and peak level are often used interchangeably. The term peak level can be misleading, since, strictly speaking, it refers to a maximum instantaneous sound level no matter how short its time duration (even if the time duration is so short that the ear cannot respond to the sound). Accurate measurement of the peak level requires an oscilloscope or other special instrumentation, since the hysteresis of a meter will affect the reading. To avoid confusion it is recommended that peak level be used only when referring to the instantaneous maximum sound level that occurs during a specified period of time.

There are a number of other metrics that characterize community noise. However, most of these metrics are significantly more complex than the L_{eq} measures, and have not shown a higher correlation with annoyance.

There are two basic methods by which we can indirectly assess the effects of environmental noise on people. The first is to evaluate community actions with respect to noise (e.g., complaints, legal action); the second is to use social surveys. Figures 2.1 and 2.2 summarize the community reactions to 55 noise problems. The data are for a wide range of noise sources (noise levels are given in terms of L_{dn}). The data for the figures, originally presented in Reference 2.16, have been widely quoted. The basic conclusions of the Environmental Protection Agency (EPA) "Levels Document" are largely based on this data.

Figure 2.1 gives the community reaction in terms of the normalized L_{dn}, and Figure 2.2 presents the same data without normalization. The normalization procedure adds correction factors in 5 dB increments to account for seasonal effects, pre-existing community noise levels, community attitudes, and the pure tone or impulsive character of the noise. Comparing Figures 2.1 and 2.2 shows that the normalization process reduces the spread of the data. The data in Figure 2.1 were normalized to urban residential noise, some previous exposure, no pure tone or impulsive character, and windows partially open. The analysis of the data in Figures 2.1 and 2.2 by Eldred (Ref. 2.16) indicates that metrics such as L_{dn} can reasonably approximate community reaction to an intruding noise source.

In References 2.10 through 2.12, Schultz thoroughly reviews the social surveys of noise annoyance. Eleven social surveys from several countries were reviewed, the noise ratings normalized to L_{dn}, and the scales of community annoyance evaluated to determine the percent "highly annoyed." The results were highly

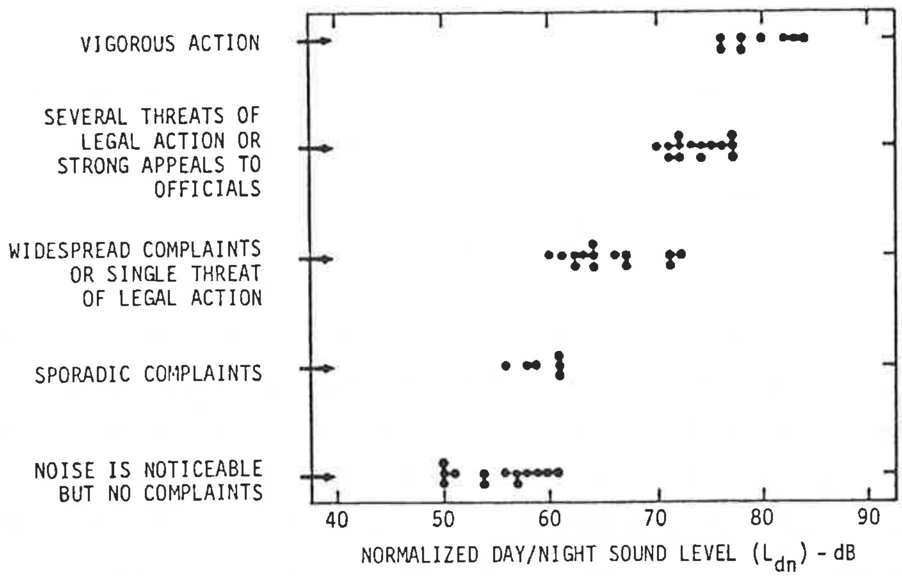


FIGURE 2.1 COMMUNITY REACTION TO INTRUSIVE NOISES AS A FUNCTION OF NORMALIZED L_{dn} (ADAPTED FROM REF. 2.1a)

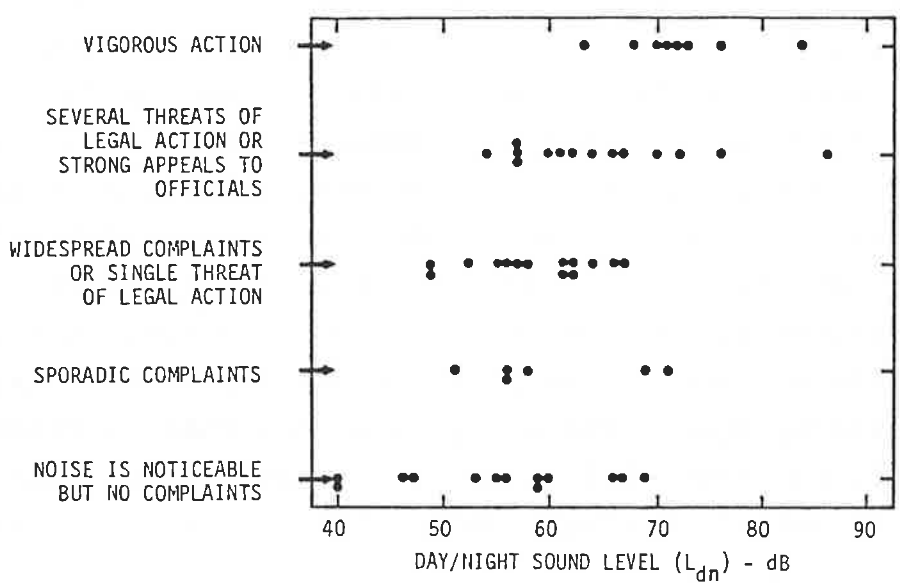


FIGURE 2.2 COMMUNITY REACTION TO INTRUSIVE NOISES AS A FUNCTION OF L_{dn} (ADAPTED FROM REF. 2.1a)

consistent. His basic conclusions follow:

- The combined curve (Fig. 2.3) is currently the best available relationship for predicting community annoyance due to transportation noise of all kinds. More research into how people respond to noise is needed.

- There is no evidence of a "supersensitive" portion of the population who will be annoyed by any noise or a percentage who will not be disturbed by any noise. Schultz concludes that there is always a threshold below which there is no part of the population that is highly annoyed (Ref. 2.12). However, there is also, apparently, no threshold below which all annoyance disappears.

- It has often been claimed that complaints represent only a small proportion of the population that is "highly annoyed." However, Schultz concludes that the percent highly annoyed is comparable with the percent of complaints.

Available social survey data on noise disturbance have been synthesized into a generalized annoyance curve. The derived curve is presented in Figure 2.3. The analysis by which this curve is derived is presented in References 2.5 and 2.12. As found in Reference 2.12, "The data further confirm previous assumptions that the statistical relationship between population annoyance and (energy) average noise level is essentially independent of noise source."

This generalized annoyance function is used in Reference 2.5 to derive a quantitative procedure for assessing noise impact in terms of a "sound level-weighted population." The sound level weighted population is part of the fractional impact method of evaluating noise exposure. For a population exposed to a specified sound level, the "fractional impact" consists of the product of the exposed population and a weighting value based on sound level; the sound level-weighted population is the sum of the fractional

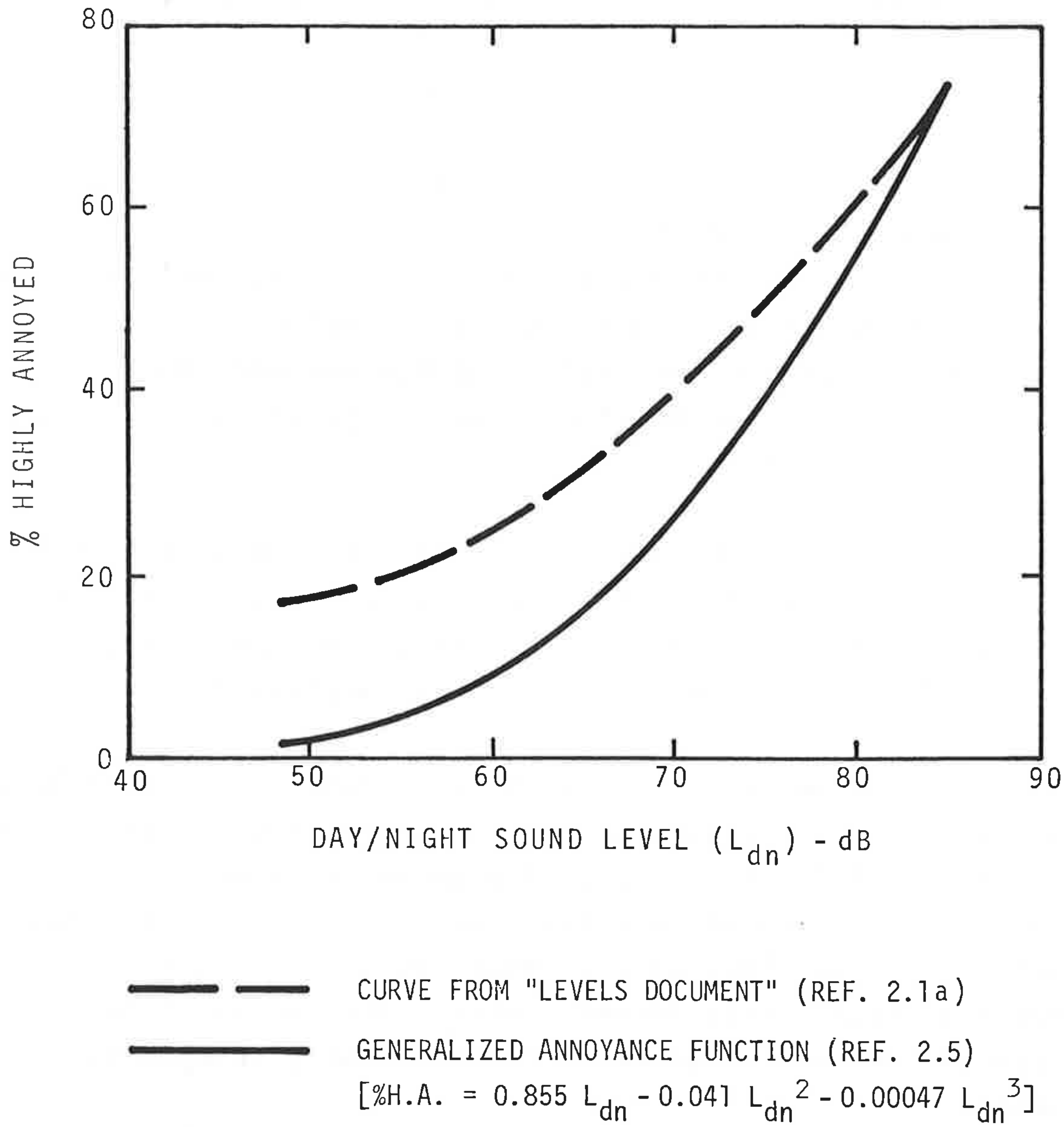


FIGURE 2.3 GENERALIZED ANNOYANCE FUNCTION

impacts. The fractional impact method is convenient for comparing the noise impact of various alternatives (e.g., several different route alignments). Details of the fractional impact method are given in Reference 2.5.

Also shown, in Figure 2.3, is the generalized annoyance curve derived in the "Levels Document." This curve was developed before much of the existing social survey data were available and differs from the more recently derived curve primarily at the low sound levels. However, the data now available tend to support the general conclusions of the "Levels Document," with respect to the noise levels "requisite to protect public health and welfare with an adequate margin of safety." Note that the term "public health and welfare" is an all-encompassing term covering hearing loss, activity interference, and annoyance. Also note that the "adequate margin of safety" used was 5 dBA. That is, appropriate levels were defined that would protect public health and welfare. To provide the margin of safety, these levels were reduced 5 dB to obtain the "acceptable levels."

Tables 2-1 and 2-2, extracted from the "Levels Document," summarize the conclusions of the EPA on noise exposure; basically, a long-term average L_{dn} of 70 dBA will protect most people against noise-induced hearing loss; and L_{dn}'s of 45 dBA indoors, and 55 dBA outdoors, in residential areas are sufficient to protect against interference with speech or other human activity and will protect most of the population against annoyance.

Table 2-2 presents the levels for various activities and locations that are identified by the "Levels Document" as required to protect public health and welfare with an adequate margin of safety. Of particular interest is the level of $L_{eq(24)}$ of 70 dBA, that is identified for "inside transportation." For a typical commute trip on a modern transit system, L_{eq} will be in the range of 70 to 80 dBA. The level depends upon the train speed and the type of way structure; noise levels are highest in subways, and lowest on surface ballast-and-tie tracks. An L_{eq} of 70 dBA could be

TABLE 2-1 SUMMARY OF NOISE LEVELS IDENTIFIED AS REQUISITE TO PROTECT PUBLIC HEALTH AND WELFARE WITH AN ADEQUATE MARGIN OF SAFETY (SEE TABLE 2-2 FOR DETAILED DESCRIPTION) (ADAPTED FROM REF. 2.1a)

Effect	Levels	Areas
Hearing Loss	$L_{eq(24)} < 70$ dB	All areas
Outdoor activity interference and annoyance	$L_{dn} < 55$ dB	Outdoors in residential areas and farms and other outdoor areas where people spend widely varying amounts of time and other places in which quiet is a basis for use.
	$L_{eq(24)} < 55$ dB	Outdoor areas where people spend limited amounts of time, such as school yards, playgrounds, etc.
Indoor activity interference and annoyance	$L_{dn} < 45$ dB	Indoor residential areas
	$L_{eq(24)} < 45$ dB	Other indoor areas with human activities such as schools, etc.

Explanation of Table:

1. Detailed discussions of the terms L_{dn} and L_{eq} appear later in the document. Briefly, $L_{eq(24)}$ represent the sound energy averaged over a 24-hour period while L_{dn} represents the L_{eq} with a 10 dB nighttime weighting.

2. The hearing loss level identified here represents annual averages of the daily level over a period of forty years (These are energy averages, not to be confused with arithmetic averages.)

3. Relationship of an $L_{eq(24)}$ of 70 dB to higher exposure levels.

EPA has determined that for purposes of hearing conservation alone, a level which is protective of that segment of the population at or below the 96th percentile will protect virtually the entire population. This level has been calculated to be an L_{eq} of 70 dB over a 24-hour day.

TABLE 2-2 YEARLY AVERAGE* EQUIVALENT SOUND LEVELS IDENTIFIED AS REQUISITE TO PROTECT THE PUBLIC HEALTH AND WELFARE WITH AN ADEQUATE MARGIN OF SAFETY (REF. 2.1a)

	Measure	Indoor		To Protect Against Both Effects (b)	Outdoor		To Protect Against Both Effects (b)
		Activity Interference	Hearing Loss Consideration		Activity Interference	Hearing Loss Consideration	
Residential with Outside Space and Farm Residences	L_{dn}	45		45	55		55
	$L_{eq(24)}$		70			70	
Residential with No Outside Space	L_{dn}	45		45			
	$L_{eq(24)}$		70				
Commercial	$L_{eq(24)}$	(a)	70	70(c)	(a)	70	70(c)
Inside Transportation	$L_{eq(24)}$	(a)	70	(a)			
Industrial	$L_{eq(24)}$(d)	(a)	70	70(c)	(a)	70	70(c)
Hospitals	L_{dn}	45		45	55		55
	$L_{eq(24)}$		70			70	
Educational	$L_{eq(24)}$	45		45	55		55
	$L_{eq(24)}$(d)		70			70	
Recreational Areas	$L_{eq(24)}$	(a)	70	70(c)	(a)	70	70(c)
Farm Land and General Unpopulated Land	$L_{eq(24)}$				(a)	70	70(c)

Code:

a. Since different types of activities appear to be associated with different levels, identification of a maximum level for activity interference may be difficult except in those circumstances where speech communication is a critical activity (See Figure D-2 for noise levels as a function of distance which allow satisfactory communication.)
b. Based on lowest level.
c. Based only on hearing loss.
d. An $L_{eq(8)}$ of 75 dB may be identified in these situations so long as the exposure over the remaining 16 hours per day is low enough to result in a negligible contribution to the 24-hour average, i.e., no greater than an L_{eq} of 60 dB.

Note: Explanation of identified level for hearing loss: The exposure period which results in hearing loss at the identified level is a period of 40 years.

*Refers to energy rather than arithmetic averages.

achieved on modern transit systems for a commute trip that does not include a significant amount of subway operations. However, for operations in subways with modern lightweight transit cars that have wide doors for easy access, it is impractical to reduce noise levels so that L_{eq} for a typical commute trip will be less than 70 dBA.

2.1.1 Criteria

In developing noise criteria for rail rapid transit systems, it is very important to specify levels that will be acceptable to those exposed to the noise, without specifying levels that are prohibitively expensive or impossible to achieve. The American Public Transit Association (APTA) "Noise and Vibration Guidelines" (Ref. 2.17) recommend noise level goals for most noise control problems that occur on rail transit systems; for easy reference, it is included in full as Appendix B of this handbook. Since the APTA guidelines are quite comprehensive and provide specific sound level recommendations, the discussion in this section is relatively general and does not repeat detailed material.

2.1.1.1 Construction Noise -- Almost any transit system construction produces noise. Most people recognize construction noise as a temporary, intermittent problem, and will accept higher levels of noise than those emanated from permanent sources, such as traffic. However, construction noise remains a major source of community annoyance.

The construction of a rapid transit system, as for any large project, involves machines and procedures that produce intense noise and, occasionally, vibration. Transit system construction includes demolition, blasting, clearing, grading, excavating, pile driving, drilling, handling and placement of materials, erection, and finishing work, and uses all the various machines and procedure associated with these activities. In recent years considerable progress has been made in the control of construction equipment noise through:

- Modifying equipment to reduce noise emissions;

- Modifying construction procedures to reduce noise;

- Basing selection of alternative construction procedures on the resulting noise impact;

- Limiting the noise impact of construction by noise limit standards that were included in the construction contracts or applied by governmental agencies;

- Requiring that the noise levels be monitored during the construction so that any violations can be immediately rectified.

These efforts have been very successful, and new construction projects are often much less noisy than expected.

The three general configurations of transit way structures (subway, aerial, and at-grade) involve different construction techniques and, hence, produce somewhat different noise and vibration levels:

- For at-grade construction, the greatest impact will be caused by demolition; clearing and grading; placement of materials, including any retaining walls and the ballast-and-tie tracks; and finishing activities, such as fencing and landscaping.

- For aerial structure configurations, activities include demolition; ground clearing and grading; erection of foundations, possibly including pile driving; construction of aerial structure columns; erection of girders; and finishing.

- For subway construction, acoustical impacts can be of two different characters. In the areas where tunneling is used, the only impact resulting from construction activities (except at access shafts) will be the groundborne vibration of the excavation process, from either the tunnel-boring machine or blasting. The vehicles removing material may cause additional ground vibration. For cut-and-cover subway, noise impact will occur from ground clearing, excavation, erection, and finishing activities.

Note that construction vibration is covered in Section 2.2.

One of the most effective ways to assure controlled noise and a minimum of acoustical impact from construction is to include noise limit specifications in the construction contract documents. Sample specification for construction noise and vibration -- similar to those used on recent construction projects of NYCTA (New York), WMATA (Washington, D.C.), MARTA (Atlanta) and NFTA (Buffalo) -- is given in Section C.2 of Appendix C. Such noise specifications ensure that contractors use state-of-the-art construction techniques that do not generate excessive noise; ensure that the contractor considers noise impact when placing equipment and scheduling operations; and ensure that the equipment used is properly maintained and has appropriate mufflers. Enforcement of noise limit specifications has been remarkably successful in limiting construction noise.

2.1.1.2 <u>Train Noise</u> -- The majority of transit system noise complaints arise from airborne noise radiated directly from the trains and from the structureborne noise radiated from steel elevated structures. Control of this noise should be a major consideration in the design of new transit facilities. Criteria for acceptable levels of train noise are given in Tables 2-8-C and 2-8-D of the APTA Guidelines (included with this report as Appendix B). Note that the Guidelines employ maximum A-weighted sound levels. T correlate these maximum levels with the expected community response given by the curves in Figures 2.1, 2.2, and 2.3, it is necessary to determine an approximate relationship between maximum A-weighted level (L_{max}) and energy average level, L_{eq}. Based on the mathematical relationship developed by Peterson (Ref. 2.18), the following approximate relationship between L_{max} and L_{eq} can be derived:

$$L_{eq} = L_{max} + 10 \log R(1.5D + d)/v - 30 \qquad (2.1)$$

where: R = number of trains per hour;
D = distance of the receiver from the track centerline in meters;
d = average train length in meters; and
v = train speed in km/hr.

Comparison with measurement data indicates that this formula is generally accurate within ± 1 dBA. It is assumed that the maximum level of train passby noise is known. If L_{max} or speed is significantly different for trains on the near and far track, it is necessary to estimate L_{eq} for the near and far tracks separately and combine the levels to determine the overall level of L_{eq} (An explanation of adding sound levels is given in Appendix A).

Using the formula given above, we can relate the guidelines for maximum airborne noise levels given in Table 2-8-C of the APTA Guidelines to EPA suggested levels (see Table 2-1). The APTA Guidelines allow different maximum levels for different community categories. The most restrictive level is for low density urban residential areas, where the recommended maximum level of train noise is 70 dBA. If the 70 dBA criterion is just met at a location 60 m (197 ft) from the track, with train speeds of 96 km/hr (60 mph) and average train lengths of 91 m (300 ft), then an L_{dn} of 55 dB, for the trains only, would be achieved with an average of 16 trains per hour during the daytime (7:00 A.M. to

10:00 P.M.) and 1.5 trains per hour at night. Reducing the number of trains in the daytime hours would allow increasing night traffic, while still maintaining an L_{dn} of 55 dB.

Generally speaking, the APTA standards for community levels of train noise agree with the goals of the EPA "Levels Document." However, meeting the APTA maximum noise level goals for residential locations will not guarantee that the L_{dn} sound levels identified in the "Levels Document" as "requisite to protect public health and welfare with an adequate margin of safety" will be achieved. Even if the maximum level goals of the APTA Guidelines are achieved, the L_{dn} level will depend on the scheduling of the trains, particularly upon the number of trains in the nighttime hours. Note that social surveys (e.g., Refs. 2.9, 2.12) that have been performed since the "Levels Document" was published do not tend to support the use of a 10 dB penalty during the nighttime hours. Although one might think that people will be much more sensitive to intruding noises at night when they are trying to sleep, the surveys indicate that the increased sensitivity, if it exists at all, is less than originally expected.

At locations where components of the noise are identifiable (due to impacts at jointed track or crossovers, wheel squeal on curves, etc.), the noise will be more annoying than usual. However, at locations where the train noise has no particularly annoying or identifiable components, experience with the sound level guidelines given in Table 2-8-C of the APTA Guidelines has shown that the community will usually accept the resulting noise environment if the single event maximum noise level design goals are met.

2.1.1.3 <u>Ancillary Equipment Noise</u> -- If it is not properly controlled, airborne noise from transit ancillary facilities can be a major source of community annoyance. This topic is discussed in Section 2-8.7.3 of the APTA Guidelines. Following these noise level design goals will minimize the likelihood of complaints about ancillary equipment noise.

2.1.1.4 Station Noise -- Appropriate noise levels for transit stations are discussed and presented in Sections 2-8.6.7 through 2-8.6.9 of the APTA Guidelines. The two major sources of noise in stations are trains and ancillary equipment. Proper acoustical design of transit stations can achieve a comfortable acoustical environment. Trains entering and leaving the stations should not result in noise levels greater than 85 dBA on the platforms. Noise from ancillary equipment should be limited to 55 dBA. In addition, in enclosed stations the reverberation of public areas should be controlled through acoustical treatment. For typical two-track stations, a reverberation time of 1.2 to 1.4 seconds is appropriate. For very large spaces (e.g., unusually high ceilings and multitrack stations) reverberation time in the range of 1.4 to 1.6 seconds is satisfactory. Providing sound absorption materials in sufficient quantity to meet reverberation time criteria reduces speech interference, thus improving intelligibility of public address systems and patrons' ability to converse, as well as controlling train and other noises.

2.1.1.5 Vehicle Noise Standards -- Interior and exterior vehicle noise generally have separate standards, even though the sources are the same. The goal of the noise standards given in the APTA Guidelines is that vehicles be constructed to minimize wayside noise and to maintain an appropriate acoustic environment for the vehicle passengers. The Guidelines recommend specific maximum overall levels at maximum operating speeds (on ballast-and-tie and concrete trackbeds) along with maximum levels for vehicle equipment noise when the vehicle is stationary. These guidelines ensure that appropriate overall levels are maintained and that no specific piece of the vehicle equipment is excessively noisy.

The recommended overall noise levels in the car interior at maximum speed range from a minimum of 70 dBA, on ballast-and-tie track in the open, to a maximum of 80 dBA in subways. Table 2-3 presents the subjective rating of vehicle interior noise proposed by Bryon (Ref. 2.19). Based on the criterion for vehicle interior noise, meeting the APTA Guidelines means that at low speeds on ballast-and-tie trackway, the subjective rating of interior noise should be "quiet" while at high speeds in subway, the level may be "intrusive" but not "annoying". These levels are consistent with the noise levels inside all except the quietest automobiles at highway speeds.

TABLE 2-3 SUBJECTIVE RATING OF NOISE IN PASSENGER VEHICLES (REF. 2.19)

Subjective Rating	Noise Level Not Exceeding
Quiet	67 dBA
Noticeable	73
Intrusive	79
Annoying	85
Very Annoying	91

2.1.1.6 Groundborne Noise -- Setting criteria for groundborne noise is a unique problem, since the noise is caused by the vibration of interior room surfaces. That is, the noise is radiated and is heard only inside buildings. Groundborne noise is generally heard as a low-frequency rumble, in many cases clearly audible, even though the building vibration may be well below the threshold of perceptibility to motion. When the vibration is perceptible, the vibration criteria of Section 2.2 should be applied.

Because groundborne noise is dominated by low-frequency components and may be intrusive even though A-weighted levels are relatively low, the maximum single event design goals for groundborne noise are comparatively low. For most residential structures, the maximum level criterion is 30 to 35 dBA. Achieving the goals of the APTA Guidelines will not keep groundborne noise from being occasionally audible, but it will rarely be intrusive.

2.1.1.7 Shop Noise - OSHA Standards -- As in all industrial environments, transit system employees are sometimes exposed to noise levels that can cause hearing damage. Employee noise exposure is regulated by the Occupational Safety and Health (OSHA) Standards given in Table 2-4, which are applicable to all industries engaged in interstate commerce. Many states have also adopted similar standards for all work places. The OSHA, or similar, standards are now applicable to most industries in the United States.

The OSHA standards are based on the assumption that the average employee will experience no hearing damage at noise levels below 90 dBA. There is a continuing debate between OSHA and the EPA about lowering this threshold to 85 dBA; to date, the controversy has not been resolved. The maximum employee exposure to a level of 90 dBA is 8 hours. For every 5 dBA increase in noise level, the allowable exposure time is cut in half. No exposure to levels above 115 dBA is allowed.

TABLE 2-4 OSHA PERMISSIBLE NOISE EXPOSURE LIMITS

Duration Per Day Hours	Sound Level dBA *Slow* Response
8	90
6	92
4	95
3	97
2	100
1 1/2	102
1	105
1/2	110
1/4 or less	115

When the daily noise exposure is composed of two or more periods of noise exposure of different levels, their combined effect should be considered, rather than the individual effects of each. If the sum of the following fractions $C_1/T_1 + C_2/T_2 + \ldots + C_n/T_n$ exceeds unity, then the mixed exposure should be considered to exceed the limit value. C_n indicates the total time of exposure at a specified noise level, and T_n indicates the total time of exposure at that level permitted during the work day.

Exposure to impulsive or impact noise should not exceed 140 dB peak sound pressure level.

To meet the OSHA standards without designing special noise control features or using hearing protectors, the APTA Guidelines suggest that appropriate noise level limits be included in the specifications for all new shop equipment. The suggested limits are that the noise levels do not exceed 85 dBA at any operators' stations, nor 90 dBA at any point 1 m (3 ft) from the equipment.

2.2 VIBRATION

Vibration, as it affects the community, the transit patrons, and at times the transit facilities, can be a major consideration for rail transit systems. Two categories of vibration are discussed: (1) vehicle vibration as experienced by patrons in the vehicles; and (2) building vibration along the transit corridor, resulting from train operations or construction activities. The vibration created by ancillary equipment, such as ventilation fans, and the fatigue of structural components due to vibration are not covered by this handbook. Note that only in relatively rare situations will vibration from ancillary facilities be a problem.

Design goals for vehicle and structural vibration are generally based on human response to vibration. Research into this field has recently developed some generally acceptable guidelines for evaluation of human exposure to whole body vibration (International Organization for Standardization (ISO) Standard 2631, Ref. 2.4). The ISO Standard is being widely applied to the evaluation of human exposure to vehicle and building vibration. This standard should be consulted by anyone preparing a criterion for human exposure to vibration.

When evaluating environmental vibration, "Guidelines for Preparing Environmental Impact Statements on Noise" (Ref. 2.5) should be consulted. This report suggested that vibration be evaluated as part of the noise section of an Environmental Impact Statement (EIS). This practice is appropriate for transit system EIS's, since many transit activities can cause vibration in adjacent communities. In many cases it is even recommended that measurements of existing vibration be taken as part of the environmental evaluation of potential transit corridors. Such measurements are particularly valuable should there be complaints regarding vibration either during the construction phase or after train operations commence.

2.2.1 Transit Car Vibration

The vibration inside a transit car is a major factor in evaluating overall comfort. There are two major sources for transit car vibration: the rolling of the wheels on the rail and the vibrations created by various propulsion and ancillary equipment systems. The vibration that results from the car moving on the rails falls into the area generally referred to as "ride quality." Ride quality, or ride comfort, is generally determined by the car's suspension system, and the condition and type of trackbed. Ride quality only relates to the motion of the transit car as it moves on the rails and not with other factors that affect passenger comfort. Because ride quality is a form of vibration not directly related to noise, it is beyond the scope of this handbook.

The interior vibration caused by propulsion and auxiliary equipment is normally covered by the car specifications. The recommended specification for car interior vibration in the APTA Guidelines is shown in terms of 1/3 octave band RMS acceleration in Figure 2.4. Also shown in the figure are the specifications for ride quality vibration used by MARTA and WMATA, and the "reduced comfort boundaries" from ISO Standard 2631. The specification for equipment vibration is significantly below the ride quality criterion; the low-frequency vibration limit is also more restrictive and the equipment specification is extended to high-frequency vibration. More restrictive limits must be applied to vibration created by on-car equipment than on ride quality vibration because the equipment vibration is generally continuous,

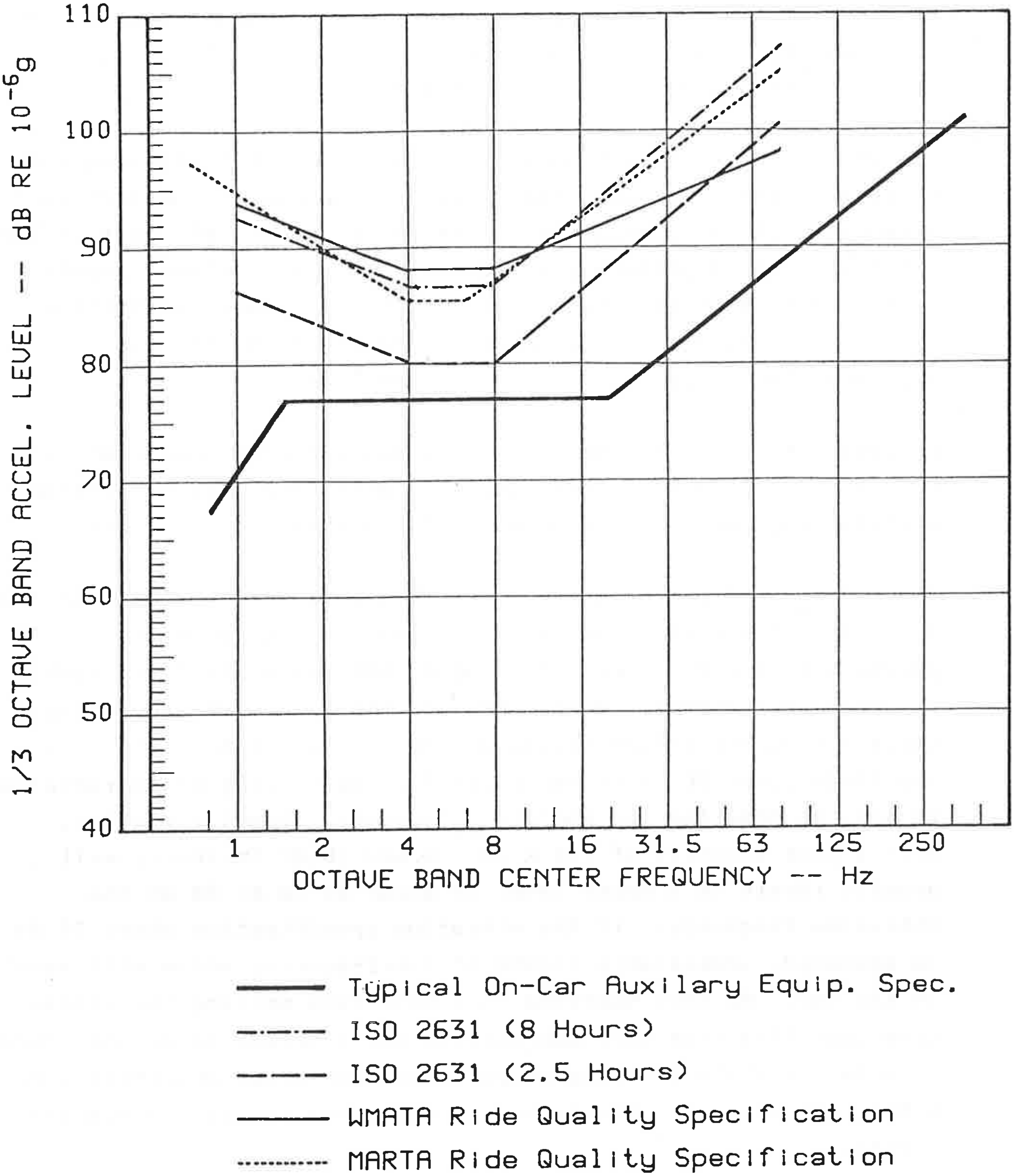


FIGURE 2.4 TYPICAL SPECIFICATION FOR CAR INTERIOR VIBRATION FROM AUXILIARY EQUIPMENT

occurring even when the car is at rest. Should a transit car have anywhere near the same vibration at rest as it has at maximum speed, many people would find the vibration excessive. Further, if the vibration from the on-car equipment is near the ride quality limit, any added vibration due to rolling will increase the overall vibration above the specification.

The vibration specification for on-car equipment is divided into three frequency regions. Below 1.4 Hz, the maximum deflection (zero to peak) is limited to 2.5 mm (0.10 in.). This part of the specification is primarily based on standards of visual impact; when panels or other parts of the car body vibrate in amplitudes greater than 0.10 in., patrons can find it noticeable and disconcerting, especially in stationary cars.

Between 1.4 and 20 Hz the limit is a maximum peak acceleration of 0.01 g. Vibrations of this magnitude will generally be distinctly perceptible, but not uncomfortable to patrons or operations.

Above 20 Hz the criterion requires that peak velocity be less than 7.6×10^{-4} m/sec (0.03 in./sec). This level is above the perception threshold for most people, but below the disagreeable or unpleasant ranges. Vibrations of this magnitude are perceptible but do not reduce patron comfort. The limit on velocity amplitude above 20 Hz is important for controlling noise radiation as well as ensuring the comfort of patrons. Panel vibration, with a peak velocity of 7.6×10^{-4} m/sec (0.03 in./sec), will usually result in a noise level of about 80 to 86 dB at the vibration frequency. If the vibration specification above 20 Hz is exceeded, undesirable levels of low-frequency noise will result in the car. We have observed that BART cars meeting the vibration specifications have appropriate car interior noise and vibration levels while the cars which exceed vibration specifications often have excessive low-frequency noise and annoying vibration levels.

2.2.2 Groundborne Vibration

Groundborne noise and vibration, transmitted from transit structures to adjacent buildings, can be a major source of community annoyance. The problem is usually groundborne noise; the vibration is not often perceptible, although perceptible vibration has occasionally furnished the principal grounds for complaint. Criteria for maximum acceptable levels of groundborne noise, discussed in Section 2.1, are included in the APTA Guidelines. Until recently, however, insufficient data have been available on human responses to building vibration to allow the development of firm criteria for groundborne vibration.

Recently, the new transit systems in Washington, D.C. (WMATA) and Atlanta (MARTA) have initiated revenue service. With these systems, the peak frequencies of both groundborne noise and vibration are in the range of 16 to 60 Hz, and groundborne noise levels have been found to be generally satisfactory or inaudible. However, intrusive vibration has been noticed at several buildings close to subway structures. Evidently, criteria for groundborne noise and vibration should address acceptable levels of both noise and vibration.

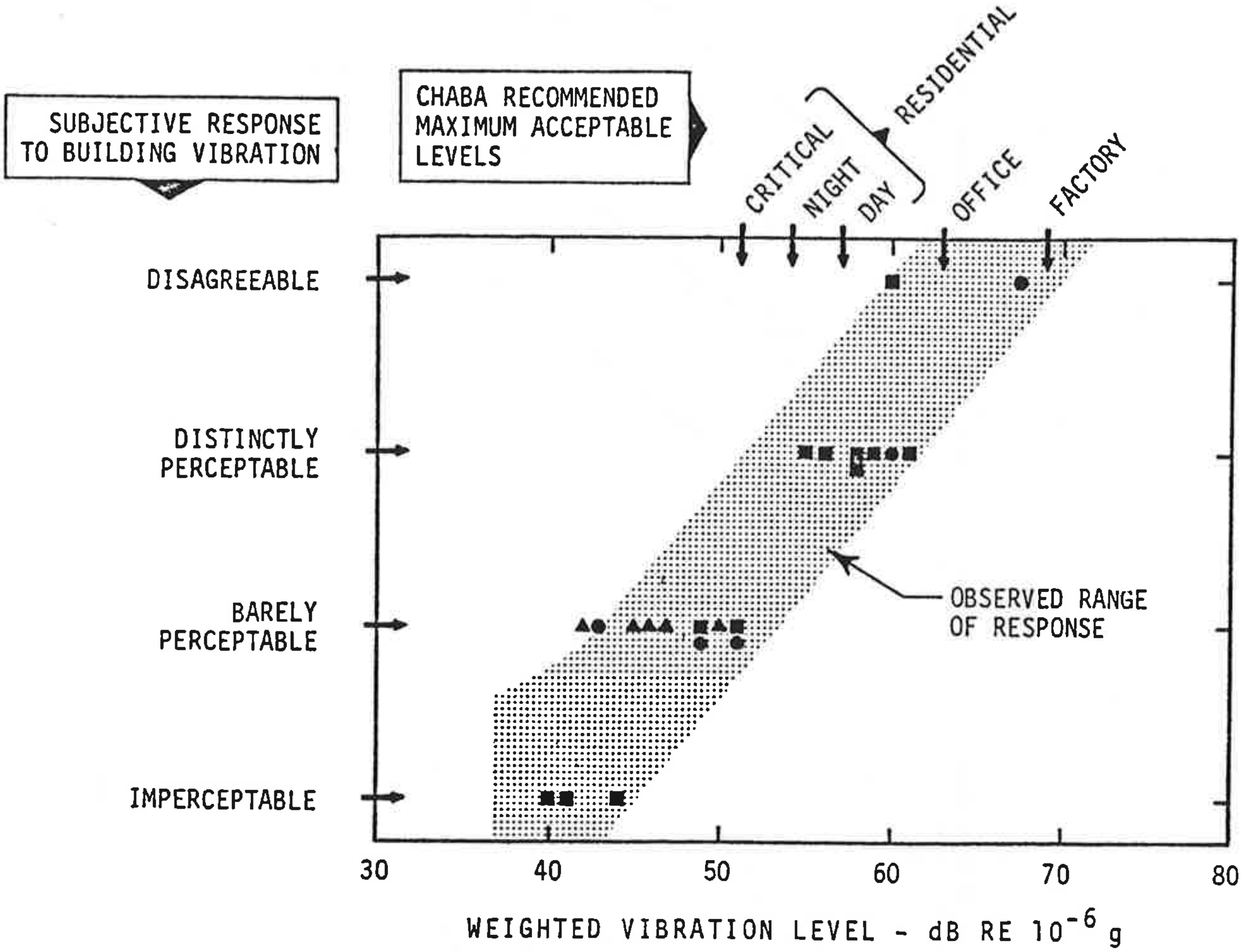
When the original criteria for groundborne noise and vibration were developed, very limited information was available on human responses to building vibration. Indeed, the "level of perception" as defined by various researchers varied over a range of 10 to 20 dB. The recent amendment to ISO Standard 2631 (Ref. 2.4), covering building vibration, and the work of the CHABA Working Group 69 Committee (Ref. 2.5) provide a good basis for developing criteria applicable to building vibration due to transit operations.

The CHABA report recommends use of a weighted vibration level for measuring or evaluating human response to environmental vibration. Although this weighting curve has yet to gain wide acceptance (it is not part of a standardized measurement procedure), it corresponds well to contemporary research on human response to a single-number overall vibration level. Without the weighting, human response must be related to spectral analysis of the vibration, e.g., octave or 1/3 octave band vibration. The weighted vibration level is thus analogous to the use of A-weighting for noise.

Using the CHABA weighting curve, existing data on building vibration at WMATA, MARTA and TTC were reanalyzed. Figure 2.5 presents the weighted vibration levels for measurements of vertical vibration near the center of floors. Using the occupant's response to the vibration and the subjective evaluation of the person performing the measurements, each situation has been classified according to the intrusiveness of the vibration. Although relatively limited information is available at this time, the data clearly indicate that when the weighted vibration level exceeds 50 to 55 dB re 10^{-6} g, complaints will probably occur in residential areas. Of course, the data in Figure 2.5 do not represent the results of a scientific social survey; it is clear that this is an area where further research is required.

Experience with groundborne vibration indicates that when building vibration is perceptible, people worry about damage to the structure (Ref. 2.9). A common criterion is that vibration below 50 mm/sec (2 in./sec) presents little danger of building damage. This figure corresponds approximately to a weighted vibration level of 100 dB. It is very unlikely that levels of groundborne vibration from transit trains could rise to anywhere near this level.

Figure 2.6 provides a comparison of typical levels of ground vibration near transit system subways and various criteria for human and structure exposure to vibration. Also shown in the figure is a curve of approximate residential vibration limits, based on a maximum weighted residential vibration level of 54 dB. When any portion of the 1/3 octave band spectrum of building vibration exceeds this curve, the weighted overall level will

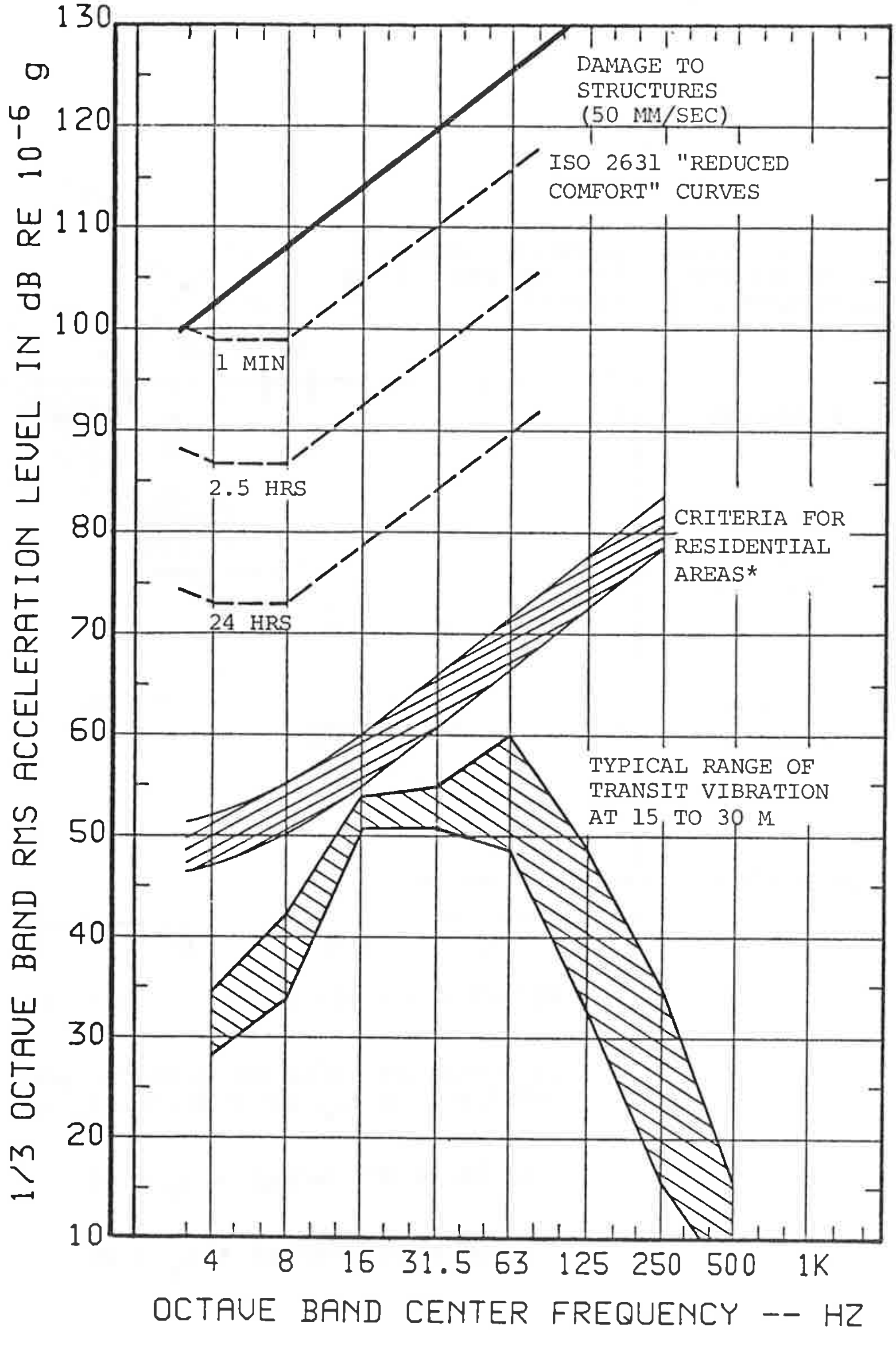


THE APPROXIMATE CONVERSION BETWEEN WEIGHTED VIBRATION LEVEL (L_W) AND VELOCITY LEVEL IS:

$$L_V \text{ (dB re } 10^{-6} \text{ in/sec)} = L_W + 21$$

$$L_V \text{ (dB re } 10^{-8} \text{ in/sec)} = L_W + 29$$

FIGURE 2.5 RANGE OF HUMAN RESPONSE TO BUILDING VIBRATION (ADAPTED FROM REF. 2.24)



*THE CRITERIA CURVE IS BASED ON A WEIGHTED ACCELERATION LEVEL OF 54 dB. THE SPECTRUM SHAPE IS THE INVERSE OF THE CHABA WEIGHTING CURVE. THE LEVEL IS BASED ON THE ASSUMPTION THAT THE ENERGY IS CONCENTRATED IN TWO OCTAVES.

FIGURE 2.6 CRITERIA FOR STRUCTURAL DAMAGE COMPARED TO TYPICAL TRANSIT VIBRATION exceed 54 dB. The curve, therefore, represents an approximate threshold of acceptability inside a residential structure.

The data presented in Figure 2.5 imply clearly that transit structures be designed so that the maximum weighted vibration levels in residential areas do not exceed 55 dB. This recommendation closely corresponds with the CHABA recommended maximum weighted level of 54 dB for residential areas at night. (Note that the CHABA document defines the acceptable acceleration threshold at night in a residential area as a weighted value of 0.005 m/sec^2 (RMS) for any duration of vibration. This translates to 54 dB re 10^{-6} g).

2.2.3 Groundborne Vibration from Construction

Because of the nature of some construction activities, high amplitudes of groundborne vibration may have an impact on neighboring community areas. Blasting and impact pile driving are typically associated with high levels of groundborne vibration, and it is not uncommon for very heavy vehicles and excavation activities to generate groundborne vibration levels noticeable in nearby buildings.

The vibration levels created by the normal movement of vehicles, including graders, loaders, dozers, scrapers, and trucks, are generally of the same order of magnitude as the groundborne vibration created by heavy vehicles running on streets and highways. Groundborne vibrations from such vehicles, even on rough roads, are not often sufficient to be noticeable in adjacent community areas; the vibration is usually imperceptible or only barely perceptible. Therefore, the impact generated by normal vehicle activities at construction sites can generally be assumed to be insignificant.

The most important consideration in devising criteria for acceptable levels of community vibration is the possibility of structural damage. The construction activities with the highest potential for causing structural damage from ground vibration are blasting, pile driving, and tunnel boring. It is widely accepted that 50 mm/sec (2.0 in./sec) is a safe maximum limit for ground-borne vibration.

Published data on ground vibration and damage are reviewed in Reference 2.20, "Blasting Vibrations and Their Effects on Structures." Findings indicate that all the major damage points and 94% of the minor damage points were for vibration above 50 mm/sec. Reference 2.20 recommends 50 mm/sec as a safe vibration limit for blasting vibration, with the qualification that it is a probability criterion; meeting the criterion is not an absolute guarantee that there will be no damage. Note that Reference 2.22 reports a study of vibration during tunneling in which significant settlement of cohesionless soil was caused by vibrations considerably below the 50 mm/sec standard for structural damage. Note that safe limits for blasting vibration are important topics that cannot be adequately covered in this handbook. Further information can be found in References 2.20 to 2.24.

Table 2-5 presents a set of criteria for construction vibration, based on the CHABA report (Ref. 2.5). These are reasonable criteria for preventing structural damage in virtually all circumstances. It is generally necessary to evaluate individually any cases involving a continuous vibration or a large number of impulsive vibrations with magnitudes near the criterion limits. Standards for avoidance of damage are usually based on vibrations that last for a short period of time and are not appropriate for long-term vibrations. Any vibration near the limit for structural damage will be distinctly perceptible and may cause vigorous community complaints.

TABLE 2-5 VIBRATION LIMITS FOR CONSTRUCTION ACTIVITIES (ADAPTED FROM REF. 2.5)

	Ground Vibration Limits	
	Weighted Acceleration[1]	Equivalent Velocity[2]
Normally Safe	1 m/sec	28 mm/sec (1.1 in./sec)
Potentially Unsafe	.5 to 1 m/sec^2	13 to 28 mm/sec (.5 to 1.1 in./sec)
Ancient Monuments and Ruins	.05 m/sec^2	1.3 mm/sec (.05 in./sec)

[1] Acceleration weighted with curve proposed by CHABA report (Ref 2.5)

[2] Above 10 Hz, the velocity in mm/sec is approximately equal to 28.4 times the acceleration in m/sec^2

2.3 LOCAL NOISE ORDINANCES

The recognition of environmental noise as a community problem has grown rapidly over the past decade. Public concern has led to increased governmental activity and the enactment of a large number of local and state noise ordinances. Historically, noise control legislation has largely relied upon subjective evaluation of the nuisance quality of the offending noise. However, ordinances based on subjective evaluations are inherently ambiguous, which makes enforcement very difficult.

To remove this ambiguity, most recent noise legislation specifies numerical noise level limits that are not to be exceeded. The limits usually apply at the property line of the property containing the noise source. Many local noise ordinances are based on model noise ordinances that have been prepared by state agencies and the EPA.

When planning the construction of new transit facilities, it is important to review the impact of any applicable noise ordinances and regulations. Often noise from rapid transit construction or operation is not specifically considered when noise ordinances are developed. As a result, strictly interpreted noise ordinances may require a transit system to meet noise limits that cannot, at present, be achieved. In such cases, the transit systems are being effectively asked to meet noise standards more restrictive than those imposed on motor vehicles. For this reason, community understanding of the technical problems faced by transit designers is important, as is a recognition by the transit authorities of local standards for peace and quiet.

There are four areas of special legal concern to rapid transit systems. These are:

1. Construction noise - The limits that local noise ordinances apply to construction noise are generally realistic and achievable with modern construction techniques and equipment. The contract documents for construction of new transit facilities should include noise limits consistent with the goals of the local noise ordinance.

2. In-Service trains - When transit corridors are located in residential communities, noise levels invariably exceed the usual limits for continuous noise. This is true for most modern types of transportation, including

automobiles, buses, and rail rapid transit. Recognizing this problem, most modern noise ordinances include specific, but separate, limits on motor vehicle traffic. However, if rapid transit was not considered when an ordinance was prepared, it may be that the train noise is exempt; however, it could be that either a modified ordinance must be passed that specifically covers rapid transit, a variance must be obtained, or the transit system must comply with unrealistic noise limits.

3. Track and way structure maintenance - Regular maintenance procedures, such as rail grinding and ballast dressing, can be very noisy operations. Since these operations are often scheduled at night, when residential communities are more sensitive to noise and noise ordinances are more restrictive, the possibility of maintenance operations exceeding applicable noise ordinances should be carefully reviewed.

4. Yard and shop operations - The noises of yards and shops can often exceed legal limits for residential areas. The most common source of noise is trains. Because the trains in many yards often operate on short radius curves, wheel squeal is a common problem. This is a particularly irritating noise that often leads to community annoyance and complaints. In some cases it may be necessary to obtain a variance for the noise from yard and shop operations, but it is better to locate yards away from residential areas when possible. Yards in residential areas usually need noise reduction provisions and/or buffer zones. The noises from the yards and shops of a rapid transit system rarely exceed those of truck or bus staging yards, and with proper location of the yards and shops, it should be possible to obtain whatever variances are necessary.

Because of the broad regional nature of rail rapid transit, transit system operations can extend through several different jurisdictions, each of which may have a noise ordinance. Ordinances including specific noise level limits are usually relatively consistent. However, interpretations vary concerning the applicability of such ordinances to rail transit.

The basic noise limits of several noise ordinances that have been applied to the WMATA system are summarized in Table 2-6. The three ordinances have similar limits for continuous noise, but some important differences are:

- In contrast to the other two ordinances, the Montgomery County Ordinance does not differentiate between day and night or commercial and industrial areas.

- The City of Alexandria Ordinance reduces the acceptable levels by 3 dBA at night for residential, commercial and industrial areas. Reduced nighttime levels are more commonly specified for residential areas only.

Noise ordinances often allow a 5 to 15 dB increase in the noise level limits for noises that are not continuous. This is true of the Montgomery County and City of Alexandria Ordinances, but the District of Columbia has no allowance for intermittent noise. Table 2-6 indicates the Montgomery and Alexandria Ordinances both include a 10 dB allowance for noise sources that operate less than 3 minutes of any 1-hour period.

Table 2-6 also shows the noise level limits that have been applied to WMATA train operations. Both the City of Alexandria and the District of Columbia worked with WMATA to develop reasonable noise level limits to apply to train operations. The limits are specifically included in the ordinances. However, the Montgomery County Ordinance makes no specific mention of transit trains and has been interpreted to mean that the standard noise level limits

TABLE 2-6 NOISE LEVEL LIMITS OF WASHINGTON, D.C. AREA NOISE ORDINANCES (ALL LEVELS IN dBA)

Area Description	Montgomery County	City of Alexandria	District of Columbia
CONTINUOUS NOISE			
Residential	55	55 (day) 52[1] (night)	55 (day) 50 (night)
Commercial	62	60 (day) 57[1] (night)	60
Industrial	62	70 (day) 67[1] (night)	65
INTERMITTENT NOISE[2]			
Residential	65	65 (day) 62[1] (night)	55 (day) 50 (night)
Commercial	72	70 (day) 67[1] (night)	60
Industrial	72	80 (day) 77[1] (night)	65
TRAIN NOISE[3]			
Residential	65[4]	75	75
Commercial	72[4]	80	80
Industrial	72[4]	90	90

[1] Noise level limits apply to Sundays and holidays also.

[2] Sound source operates less than 3 minutes in any 1-hour period.

[3] Not explicitly specified in all of the noise ordinances.

[4] Assuming train noise does not occur for more than 3 minutes in any 1-hour period.

apply to all transit facilities and activities. However, special consideration for other types of transportation, such as motor vehicles, is included in the ordinance. Table 2-6 shows that the resulting noise limits of the Montgomery County Ordinance applicable to transit train operations are much more restrictive than those of the City of Alexandria and the District of Columbia.

2.4 PREPARATION OF NOISE AND VIBRATION SPECIFICATIONS

A comprehensive program for transit system noise and vibration control must include the development of appropriate criteria and specifications for new facilities and equipment. Documents for virtually all major purchases of rail transit equipment now include specifications on maximum acceptable limits of noise and vibration. In addition, the contract documents for major construction projects include specifications on maximum levels of noise and vibration that can be generated by construction activities.

In developing noise and vibration specifications or criteria, one must compromise between specifying levels that are technically and economically feasible and specifying levels that will minimize the adverse impact of noise on people and structures. To provide some guidance in preparing specifications, see Appendix C of this handbook which includes three sample specifications. The first is the noise and vibration portion of a specification prepared in 1979 for the purchase of the new CTA 2600 series cars. This specification ensured that the contractor would devote particular attention to the noise and vibration so that the car would be comfortable inside for patrons, and that noise and vibration propagated to communities would not exceed that created by existing CTA cars. The specifications included limits on individual components prior to installation on the vehicle, and maximum limits for the complete car (when stationary, on jacks with wheels spinning, and when moving on selected track). Limitations were placed on both the complete car and on its individual components in order to ensure that consideration of noise and vibration be included in all phases of the car's design. It is far more efficient to design a traction motor that will not generate excessive noise than it is to design later modifications such as baffles or enclosures to reduce the noise generated by the motor.

Of particular interest in the specifications for the Chicago cars is the designation of a minimum transmission loss through each characteristic section of the car body. Since most noise inside transit vehicles originates from noise sources exterior to the car, achieving the minimum transmission loss specification is essential if an appropriate acoustical environment is to be maintained inside the vehicle.

The second specification given in Appendix C is the Buffalo Light Rapid Rail Transit system (NFTA) specification for construction noise and vibration. It limits noise and vibration from blasting and restricts noise levels for continuous and intermittent noise in residential, commercial, and industrial areas. This specification is similar to some that have been successfully applied to a number of transit system construction projects. In addition, the specified maximum allowable noise levels are consistent with those of most local noise ordinances.

The final specification given in Appendix C covers mechanical equipment used in transit stations, including noise from elevators and escalators, which are not generally associated with noise problems. The contract documents for the escalators used in the first series of WMATA stations did not include an appropriate noise specification; normally such an oversight would not matter, since escalator drive motors are generally located in areas that are isolated from the escalator track. However, in the WMATA stations the drive motors are directly beneath the escalator track, so that the motor noise is transmitted directly to occupied areas of the stations. This example illustrates the need for having noise specifications that are applicable to all potential sources of transit noise.

CHAPTER 2 REFERENCES

2.1a "Information on Levels of Environmental Noise Requisite to Protect Public Health and Welfare with an Adequate Margin of Safety," Final Report, (commonly referred to as "Levels Document"), 550/9-74-004, (U.S. Environmental Protection Agency, Washington, D.C., March 1974).

This report was developed by the EPA to "identify noise levels consistent with the protection of public health and welfare against hearing loss, annoyance, and activity interference." The information presented is based on "analyses, extrapolations and evaluation of the present state of scientific knowledge." The conclusions are supported by many sources. Many noise ordinances and standards are based on the results of this document.

2.1b "Protective Noise Levels, Condensed Version of EPA Levels Document," 550/9-79-100, (U.S. Environmental Protection Agency, Washington, D.C., November 1978).

As is evident from the title, this is a condensed and simplified version of the "Levels Document." It briefly summarizes the results and conclusions of the "Levels Document" (Reference 2.1a above) in less technical terms.

2.2 R. M. Hanes, "Human Sensitivity to Whole-Body Vibration in Urban Transportation Systems: A Literature Review," NTIS Report PB-192-257, (Urban Mass Transportation Administration, Washington, D.C., May 1970).

This study of the literature on human sensitivity to whole-body vibration was made to find vibration limits for use in urban mass transit systems. The article is primarily concerned with ride quality.

2.3 "Assessment of Noise with Respect to Community Response," ISO Recommendation R1996, (International Organization for Standardization, Switzerland, 1971).

This paper outlines a procedure for estimating community response to intrusive sounds. Estimated community reaction ranges from "no observed reaction" to "vigorous community reaction." The method is derived from a British Standard, though neither Britain nor the United States in fact subscribe to it.

2.4 "Guide to the Evaluation of Human Exposure to Whole-Body Vibration," ISO 236-1978(E), Second Edition, (International Organization for Standardization, Switzerland, January 1978).

"Vibration and Shock Limits for Occupants In Buildings," Addendum 2, (International Organization for Standardization, Switzerland, January 1975).

A Standard prepared by the International Organization for Standardization (ISO) which provides guidelines for human response to vibration under various conditions. These guidelines are widely applied in evaluating vehicle vibration (ride quality) and building vibration.

2.5 H. E. Von Gierke, "Guidelines for Preparing Environmental Impact Statements on Noise," NTIS Report AD-A044384, (Committee on Hearing, Bioacoustics, and Biomechanics; Assembly of Behavioral and Social Sciences (CHABA), National Research Council, Working Group 69, June 1977).

The guidelines proposed in this report are the result of the deliberations of the CHABA Working Group 69 from 1972 to 1976. Guidelines are proposed for the uniform description and assessment of the various environments potentially requiring an Environmental Impact Statement for noise. In

addition to general, audible noise environments, the report covers separately high-energy impulse noise, ultrasound, infrasound, and the environmental impact of structureborne vibration. Of particular interest is the recommendation of a weighted vibration level that can be used for a single number assessment of impact from vibration.

2.6 C. G. Rice, "Development of Cumulative Noise Measure for the Prediction of General Annoyance in an Average Population," Journal of Sound and Vibration, 42(3), (Academic Press Inc., London, England, 1977), pp. 345-364.

Rice reports on a laboratory investigation into the concept of using a unified index for the prediction of annoyance from aircraft and traffic noise. It was found that at equal L_{eq} levels, traffic noise was reported to be significantly more difficult to live with than aircraft noise. Therefore, the use of a normalized L_{eq} as a unifying index is not substantiated.

2.7 T. K. Dempsey, J. D. Leatherwood, and S. A. Clevenson, "Development of Noise and Vibration Ride Comfort Criteria," Journal of Acoustical Society of America, 65(1), (Academic Press Inc., London, England, January 1979), pp. 124-132 .

This article reports a laboratory investigation directed at developing criteria for the prediction of ride quality in a noise-vibration environment. Based on the experimental results, a model of subjective discomfort that accounted for the interdependence of noise and vibration was developed. Although the study was directed towards prediction of discomfort in aircraft, the results may apply to other kinds of transportation.

2.8 M. Vernet, "Effect of Train Noise on Sleep for People Living in Houses Bordering on the Railway Line -- On Site Study." Study made with the financial assistance of the French National Railways (S.N.C.F.), approximate date 1978.

Vernet conducts an experimental study of sleep disturbance by railway traffic and road traffic at two sites. Ten subjects were selected at each site. The results of the study showed that for the same value of L_{eq}, there were three times as many disturbances of sleep from road traffic noise as there were from train noise. This was not because people became more accustomed to train noise, but because the disturbances are related to the number of occurrences of noise; for the sites of this study, there were three times as many trucks as trains. Peak noise levels of less than 56 dBA did not wake people living near the railway line.

2.9a J. M. Fields, J. G. Walker, "Reactions to Railway Noise in Great Britain," Proceedings of Inter-Noise 78, (San Francisco, CA, May 1978), pp. 585-590.

2.9b J. M. Fields, "Railway Noise and Annoyance in Residential Areas," Paper presented at Second Workshop on Railway and Tracked Transit System Noise. (October 19, 1978); also Journal of Sound and Vibration, 66(3), (Academic Press Inc, London, England, October 1979), pp. 445-458.

These papers analyze an extensive national study of railway noise in 1975-76. The Institute for Sound and Vibration Research (ISVR) combined a measurement program with a very complete social survey of nearby residents' response to railway noise. The study provided exhaustive evaluations of these responses. Some of the important findings of the study are: that railway noise is less annoying and creates less disturbance at high levels than road traffic noise at the same level; that 24-hour L_{eq} appears to be a noise descriptor as adequate as any other studied; that no evidence could be found to support nighttime correction factors (such as used in L_{dn}); that vibration is an important source of disturbance associated with railways. It appears that most people who reported vibration as a problem also feared that the vibration might result in some structural damage.

2.10 T. J. Schultz, "Development of an Acoustic Rating Scale for Assessing Annoyance Caused by Wheel/Rail Noise in Urban Mass Transit," Interim Report, UMTA-MA-06-0025-74-2, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., February 1974).

Schultz reviews several studies on the impact of train noise on communities. He recommends the use of maximum A-weighted sound level to assess noise exposure, with an addition of 5 dB when pure tones (e.g., wheel squeal) are present. Schultz recommends that to assess total exposure, an additional term, 10 log (T), be added to the mean maximum noise.

2.11 T. J. Schultz, "Noise Rating Criteria for Elevated Rapid Transit Structures," Interim Report, UMTA-MA-06-0099-79-3, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., August 1978).

This is an extension of reference 2.10, purposing to develop a noise rating criteria for rank-ordering elevated structure types. Schultz concludes that a maximum A-weighted level should be used to evaluate the noise radiation potential of elevated structures and that the day-night average sound level (L_{dn}) should be used to assess the impact of elevated structure noise on the community.

2.12 T. J. Schultz, "Synthesis of Social Surveys on Noise Annoyance," Journal of the Acoustical Society of America, 64(2), (American Institute of Physics, New York, August 1978), pp. 377-405.

The author reviews a number of social surveys concerning the noise annoyance caused by aircraft, street traffic, expressway traffic, and railroads. He uses results from eleven of the surveys to synthesize curves for predicting

community annoyance from transportation noises of all kinds.

2.13 L. M. Vallet, M. Maurin, M. A. Page, B. Favre, G. Pachiaudi, "Annoyance From and Habituation to Road Traffic Noise from Urban Expressways," Journal of Sound and Vibration, 60(3), (Academic Press Inc, London, England, 1978), pp. 423-440.

This study was carried out in ten French towns, and comprises noise measurement and psychosociological surveys of 1000 respondents, for the purpose of estimating the annoyance felt by people living close to the expressway. Over a two-year period, no evidence of habituation to the noise was found. Heavy trucks constituted the major source of annoyance, especially during the evening hours. The authors recommend use of L_{eq} to measure traffic noise between 8:00 A.M. and 8:00 P.M.; for this time period it is suggested that L_{eq} should not exceed 65 dBA. To relieve annoyance caused by heavy trucks in the evening hours (8:00 P.M. to 12:00 P.M.), L_1 is proposed as a suitable measure, with levels not permitted to exceed 70 dBA.

2.14 D. Aubree, "Enquete acoustique et sociologique permettant de definir une echelle de la gene eprouvee par l´homme dans son logement du fait des bruits de train," In French, (Acoustical and Sociological Survey to Define a Scale of Annoyance Felt by People in Their Homes due to the Noise of Railroad Trains), (Centre Scientifique et Technique du Batiment, 4 Avenue du Recteur Poincare, Paris, June 1973).

This is a combined social survey and noise measurement program performed near Paris, France for the purpose of defining a scale of annoyance due to noise from railroad trains. Twenty-four hour L_{eq} was the acoustic measure found to have the highest correlation with annoyance. Using an index of annoyance which included L_{eq}, along with terms to account for exposure of the dwelling to the railroad and certain attitudinal factors, Aubree found a considerable increase in the correlation to the difference between ambient noise level and the maximum noise of train passages; on the contrary, there was a positive correlation with noise, implying greater annoyance with train noise where the noise from the other sources was higher.

2.15 D. G. Stevens, "Developments in Ride Quality Criteria," Noise Control Engineering, 12(11), (Institute of Noise Control Engineering, Poughkeepsie, NY, January 1979), pp. 6-13.

Stevens discussed ongoing research at Langley Field to develop models for evaluating transportation system ride quality. Ride quality criteria for vibration and noise are also presented. This work is the same research project reviewed in Reference 2.7. The author concludes: "Criteria are available for evaluating the effects of simple vibration environments on human response. Criteria for complex vibrations, noise effects, and the combined effects of vibration and noise are not presently available."

2.16 C. M. Eldred, "Community Noise," NTID 300.3, (U.S. Environmental Protection Agency, Washington, D.C., December 3, 1971).

This report provides detailed data and analysis of outdoor noise in a number of different community environments.

2.17 "Noise and Vibration," Guidelines and Principles for Design of Rapid Transit Facilities, Section 2.7, (American Public Association, Washington, DC, 1979).

These Guidelines provide detailed recommendations for rail transit noise and vibration standards, specifications, and criteria. They are reproduced in full as Appendix B of this handbook.

2.18 S. Peters, "The Prediction of Railway Noise Profiles," Journal of Sound and Vibration, 32(1), (Academic Press Inc, London, England, 1974), pp. 87-99.

A method is described for predicting the noise, and decay of the noise, of a passing train. The model, which is based on dipole noise radiation, is shown to give reasonably accurate predictions of A-weighted noise level profiles. More references on this topic are given in L. G., Kurzweil, R. Lotz, "Prediction of Control of Noise and Vibration in Urban Rail Systems," UMTA No. MA-06-0025-78 8 (September 1978).

2.19 M. E. Bryan, "A Tentative Criterion for Acceptable Noise Levels in Passenger Vehicles," Journal of Sound and Vibration, 48(4), (Academic Press Inc, London, England, 1976), pp. 525-535.

Describes the results of measuring noise inside a variety of passenger vehicles, and attempts to define a measure of subjective response. The paper presents a criteria for subjective rating of passenger vehicle A-weighted noise level.

2.20 H. R. Nicholls, C. F. Johnson, W. I. Duval, "Blasting Vibrations and their Effects on Structures," Final Report, NTIS Report PB-231 971, (U.S. Bureau of Mines, Washington, D.C., 1971).

The results of a ten-year Bureau of Mines study of the problems of air blast and ground vibrations are presented. The program included an extensive field study of ground vibrations; establishment of damage criteria and empirically safe blasting limits for residential structures; and evaluation of the problem of human response. Values of 50 mm/sec (2 in./sec) particle velocity and 3500 pa (.5 psi) air blast overpressure are recommended as "safe blasting limits not to be exceeded to preclude damage to residential structures. Low limits are suggested to minimize complaints."

2.21 R. J. Steffens, "Structural Vibration and Damage," (Department of the Environment, Building Research Establishment, Her Majesty's Stationery Office, London, England, 1974).

This report reviews information on the vibration of structures and its effect on the occupants, as well as damage to the buildings. Included is a thorough review of the literature and an extensive biliography (311 references).

2.22 B. M. New, "The Effects of Ground Vibration During Bentonite Shield Tunneling at Warrington," TRRL Laboratory Report 860, (Department of the Environment, Department of Transport, Transport and Road Research Laboratory, Berkshire, England, 1978).

In this study the ground vibration during a tunneling project was monitored and the environmental effects evaluated. The maximum measured ground vibration (in terms of peak particle velocity) was 3.9 mm/sec (.15 in./sec). Vibration from the excavation is the probable cause of the observed ground settlement. Although the vibration is unlikely to have caused direct damage from dynamic stressing, the ground settlement did cause some damage. The soil type in this location was cohesionless drift deposits, prone to settle at relatively low vibration levels.

2.23 T. G. Gutowski, L. E. Wittig, C. J. Dym, "Some Aspects of the Ground Vibration Problem," Noise Control Engineering, 10(3), (Institute of Noise Control Engineering, Poughkeepsie, NY, May-June 1978).

General discussion of measurement of, and criteria for, various types of ground vibration.

2.24 V. Petrucelly, A. Patel, "High Frequency and Low Frequency Blasting Vibration Measurements Near Structures," (Port Authority of New York and New Jersey, August 1977).

A review of the criteria for blasting vibration and a discussion of instrumentation and procedures that the Port Authority of New York and New Jersey used to measure and control the blasting for the extension to a bus terminal.

CHAPTER 3

3. CHARACTERISTICS OF URBAN RAIL NOISE AND VIBRATION

In the planning and design of new transit facilities, it is usually necessary to comprehensively evaluate the noise and vibration impact of all the proposed features. In the past, such appraisals were difficult because of the general lack of data. However, now that a number of the new generation transit facilities are operational, a sizeable body of data has been established from which the noise and vibration characteristics of proposed transit systems can be projected. For example, the BART system designed in the 1960's accumulated a wealth of background information from a test track program. Another example is a national assessment study of seven U.S. transit systems that was performed under UMTA funding (Ref. 3.1). Drawing from what is available, we have tried in this chapter to present information detailed enough to be useful in projecting noise and vibration levels of planned facilities and in predicting the impact of noise and vibration on the communities along the transit route. For the purpose of comparison, data on noise and vibration levels found at older systems are also included. The material is divided into six sections: car interior noise, vehicle vibration, wayside noise, groundborne noise and vibration, and station noise.

The results of the national assessment study (Ref. 3.1) provide some insights regarding the character of rail transit noise and vibration and the exposure of communities and patrons. A summary report (Ref. 3.1h) used the information from the evaluation of the seven different transit districts to develop a composite picture of the in-car, in-station, and wayside noise environments. The results in terms of aggregate distributions of noise levels are summarized in the histograms of Figures 3.1 to 3.3. For the summary, six systems (MBTA -- Boston; SEPTA -- Philadelphia; PATCO -- Philadelphia to Lindenwold, N.J.; BART --

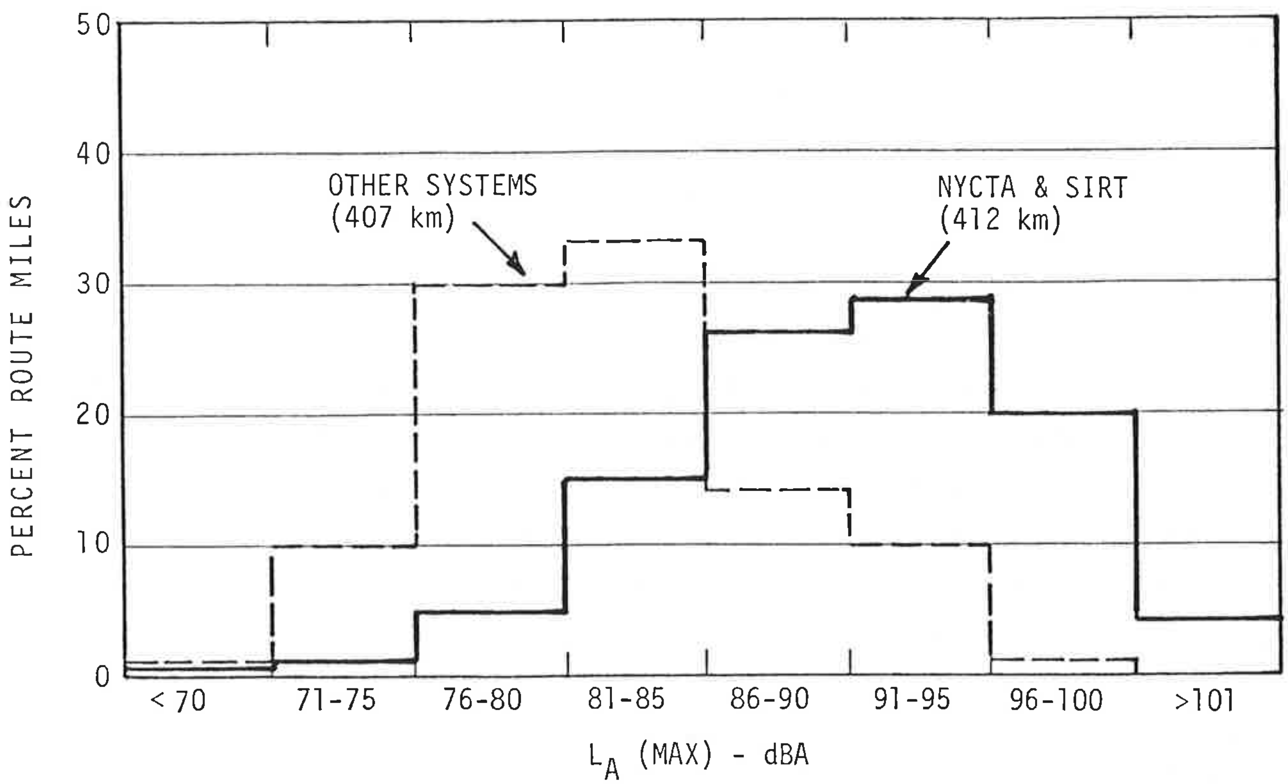


FIGURE 3.1 MAXIMUM IN-CAR NOISE LEVELS (ADAPTED FROM REF. 3.1h)

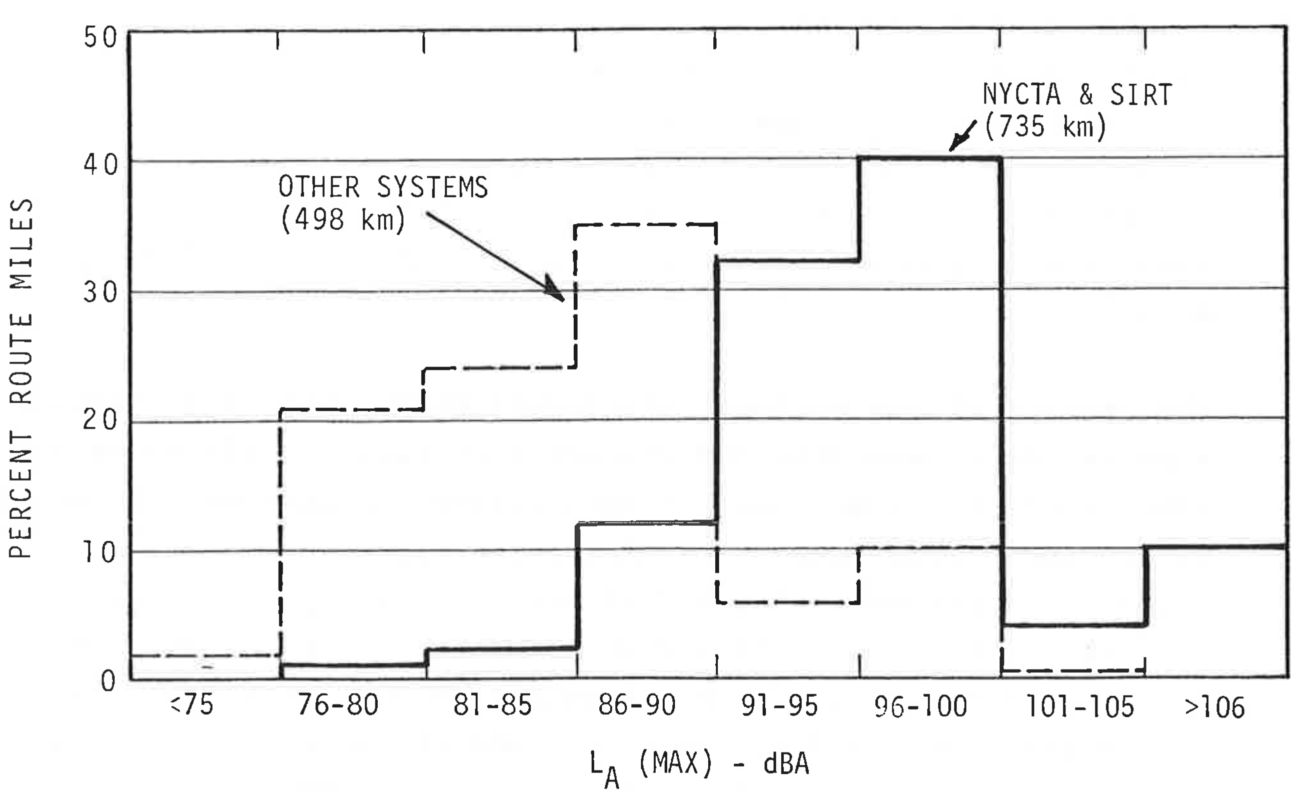


FIGURE 3.2 MAXIMUM IN-STATION NOISE LEVELS (ADAPTED FROM REF. 3.1h)

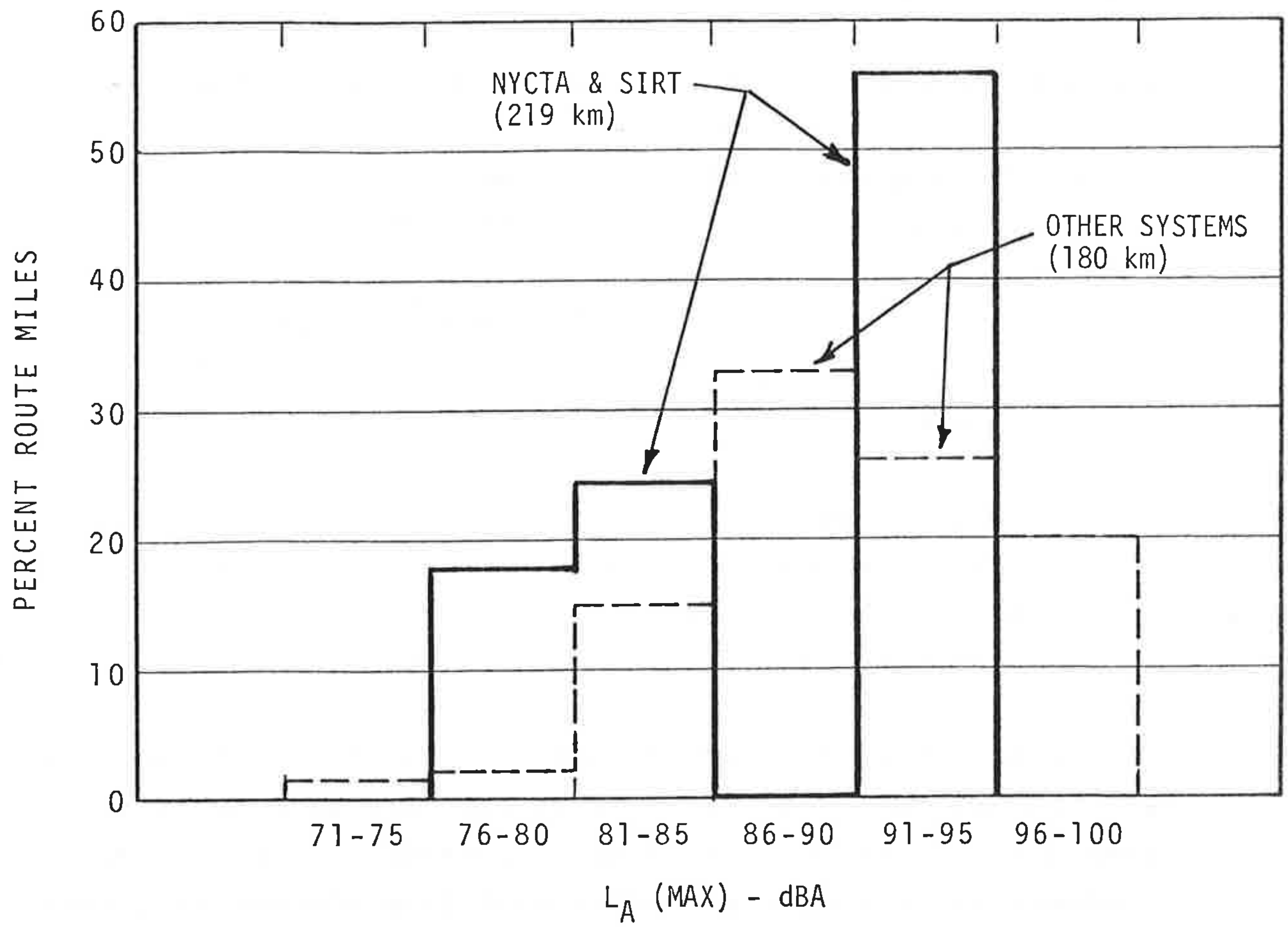


FIGURE 3.3 MAXIMUM WAYSIDE PASSBY NOISE LEVELS (ADAPTED FROM REF. 3.1h)

San Francisco; CTA -- Chicago; RTA -- Cleveland) are presented as one group, and, because of the size, the New York system (NYCTA including the Staten Island Rapid Transit (SIRT) Operating Authority) is presented separately. Note that the Port Authority of New York and New Jersey (PATH) which operates between both states was not included in the assessment.

The data in Figures 3.1, 3.2, and 3.3 indicate that:

- The highest sound levels tend to occur at the oldest facilities, particularly SEPTA, NYCTA, and CTA.

- In-car noise levels in the 86 to 100 dBA range are very common on the New York system and for the older transit cars on steel elevated structures or underground routes.

- Maximum noise levels in the stations on steel elevated structures, and in underground stations tend to be very high. The only exceptions are the BART underground stations, all of which have extensive acoustic treatment.

Although most of the discussions in this chapter focus on steel wheel/steel rail transit systems, they are also applicable, generally, to rubber tire transit systems. Rubber tire, fixed guideway transit systems differ most from conventional rail transit systems in the generation of rolling noise which results from the rubber tires rolling on the fixed concrete trackways, rather than from the rolling of steel wheels on steel rails. At present there are no rubber tire transit systems in the United States, although they have been considered in several cities and have been installed in Montreal, Mexico City and Paris.

As in any transportation system, the noise and vibration from rail transit systems can cause an adverse impact on the community. Fortunately in rail transit systems this impact is limited to a relatively small corridor along the transit way structures. Clearly, one of the most cost effective methods of controlling adverse transit impact is to arrange the system so that no noise and vibration sensitive activity is confined within the impact corridor. Although each transit system will have different sound-radiating characteristics, the major sources of noise and vibration are usually similar; on the same types of way structures and with similar operating conditions, the wayside noise levels from various transit systems will be comparable. It is possible, therefore, to give some general guidelines on the width of the community corridor affected by transit noise, as a function of types of way structure. Table 3-1 presents an ordering, by rank, of nine different classifications of way structures. Also included in Table 3-1 is the approximate impact corridor for each type of way structure (in terms of the separation distance between track centerline and nearest residential buildings). These numbers assume a train speed of approximately 100 km/hr on welded rail with both wheel and rail surfaces in good condition. Jointed rail, wheel flats, rail corrugations, especially loud noise radiated from the traction system, and other factors can result in significant increases in the width of the impact corridor.

As Table 3-1 indicates, the type of way structure selected will strongly influence the distance from the structure at which noise can still be a problem. Given an appropriate choice of way structure, transit systems can be routed through quiet, low density, residential areas with only minimal intrusion. Obviously, the modern transit structures create much less adverse impact than the lightweight steel elevated structures that are common on some of the older transit systems.

Since a number of transit systems have been constructed in the past decade, the necessary information for making planning and design decisions related to noise and vibration can be obtained from existing facilities. The only possible exception is the problem of groundborne noise and vibration, a problem that has received a good deal of attention in the past 10 years and one that new transit systems have largely succeeded in controlling. Experience indicates that local geological conditions can have a large influence on the propagation of groundborne vibration. For example, the BART system in the North Berkeley area and the TTC Yonge Subway Northern Extension (YSNE) are of similar construction; both have sections of the line in residential neighborhoods. At BART there have been no complaints relative to groundborne noise and vibration, but TTC has had many complaints along the YSNE line extending to relatively large distances from the subway. Apparently, the difference is because the soil in Toronto is much stiffer than the soil in the San Francisco Bay Area. Ongoing research should increase the accuracy with which groundborne noise and vibration can be predicted and its adverse impact evaluated.

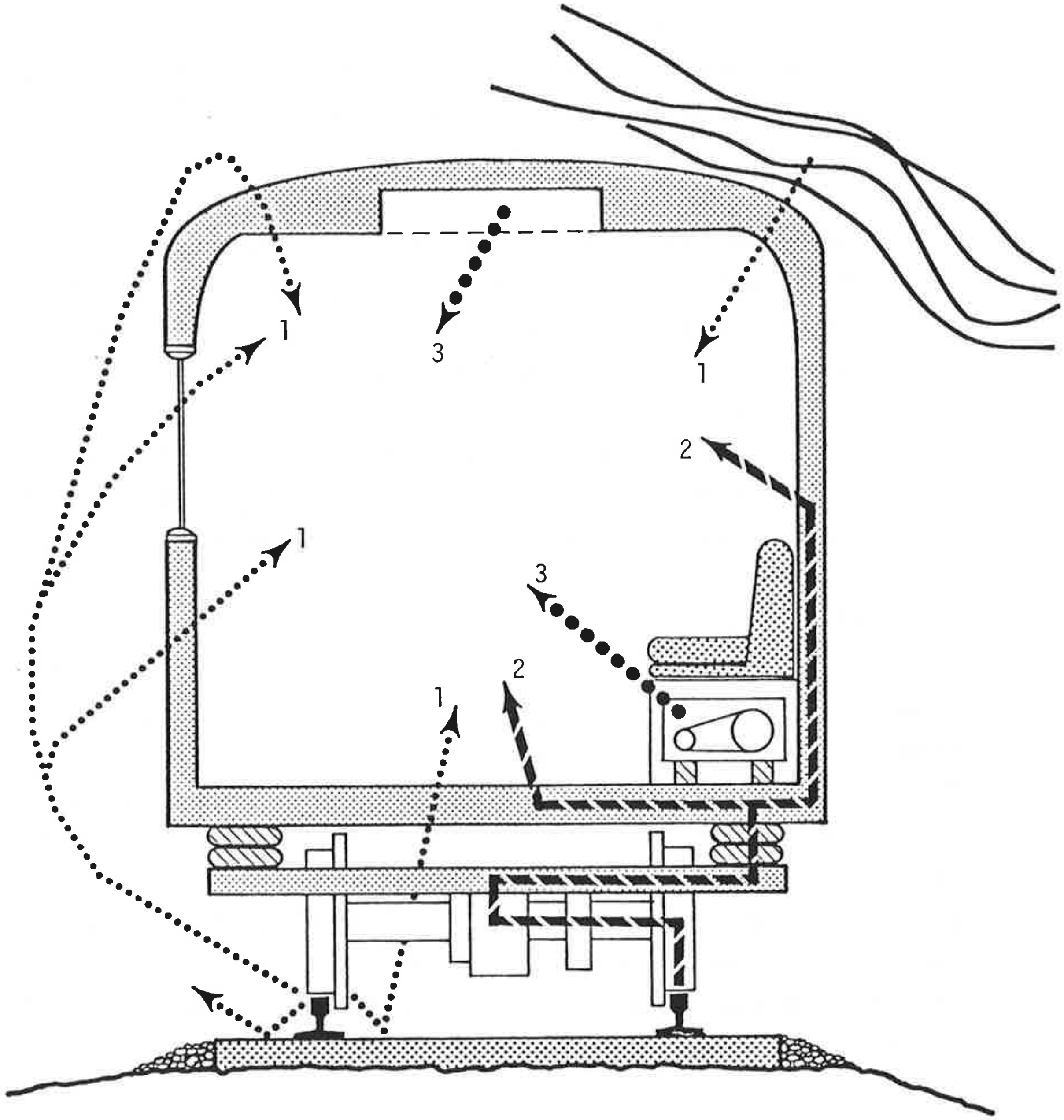
3.1 CAR INTERIOR NOISE

Virtually all noise inside rapid transit cars originates somewhere on the transit vehicle. Figure 3.4 illustrates the three major noise sources on a transit vehicle and the paths by which the noise propagates. Noise levels inside rail transit cars in the U.S. range from about 65 to 105 dBA during normal operation. This wide range of noise levels is a result of the dependence on a number of factors, including:

- Train speed. Car interior sound levels have been found to increase as a function of speed, V, at rates varying from 15 log (V) to 40 log (V).

- Type of way structure. Noise is lowest on at-grade, ballast-and-tie track, and highest during operations on lightweight steel elevated structures and inside subway tunnels that lack acoustical treatment.

- Sound insulation provided by the car body. The amount of



1. EXTERNAL AIRBORNE NOISE TRANSMITTED THROUGH CAR SHELL
2. STRUCTUREBORNE NOISE AND VIBRATION
3. AIRBORNE NOISE FROM INTERIOR SOURCES

FIGURE 3.4 SOURCES AND PATHS OF CAR INTERIOR NOISE sound insulation varies greatly. The newer cars provide significantly more sound insulation than older cars, built without consideration of acoustical design criteria.

- Noise emission characteristics of the mechanical equipment. This includes both the propulsion equipment and the auxiliary equipment, such as the air-conditioning systems, compressors, and motor generator sets.

- Wheel and rail condition. Noise levels have been found to increase as the surface roughness of the wheels and rails increases. When rails develop corrugations or wheels develop flats, the noise level can increase by as much as 10 to 15 dBA.

Figure 3.5 presents measurements of car interior A-weighted sound levels as a function of speed at various transit systems in the United States. Under similar operating conditions, noise levels inside newer transit cars can be as much as 20 dBA lower than the noise levels inside some of the older cars. The variation is a result of the difference in the noise insulating properties of the cars, differences in levels of propulsion and auxiliary equipment noise, and, sometimes, differences in the condition of the wheels and rails.

Figure 3.5 also indicates the degree to which noise levels are dependent on the type of way structure. Compared to operations on welded ballast-and-tie track in the open, at speeds of 40 to 100 km/hr, the increases in car interior noise levels on other types of track structures are approximately:

-	Lightweight steel elevated structure	+8 to 12 dBA
-	Subway tunnel with concrete trackbed without absorptive treatment	+7 to 12 dBA

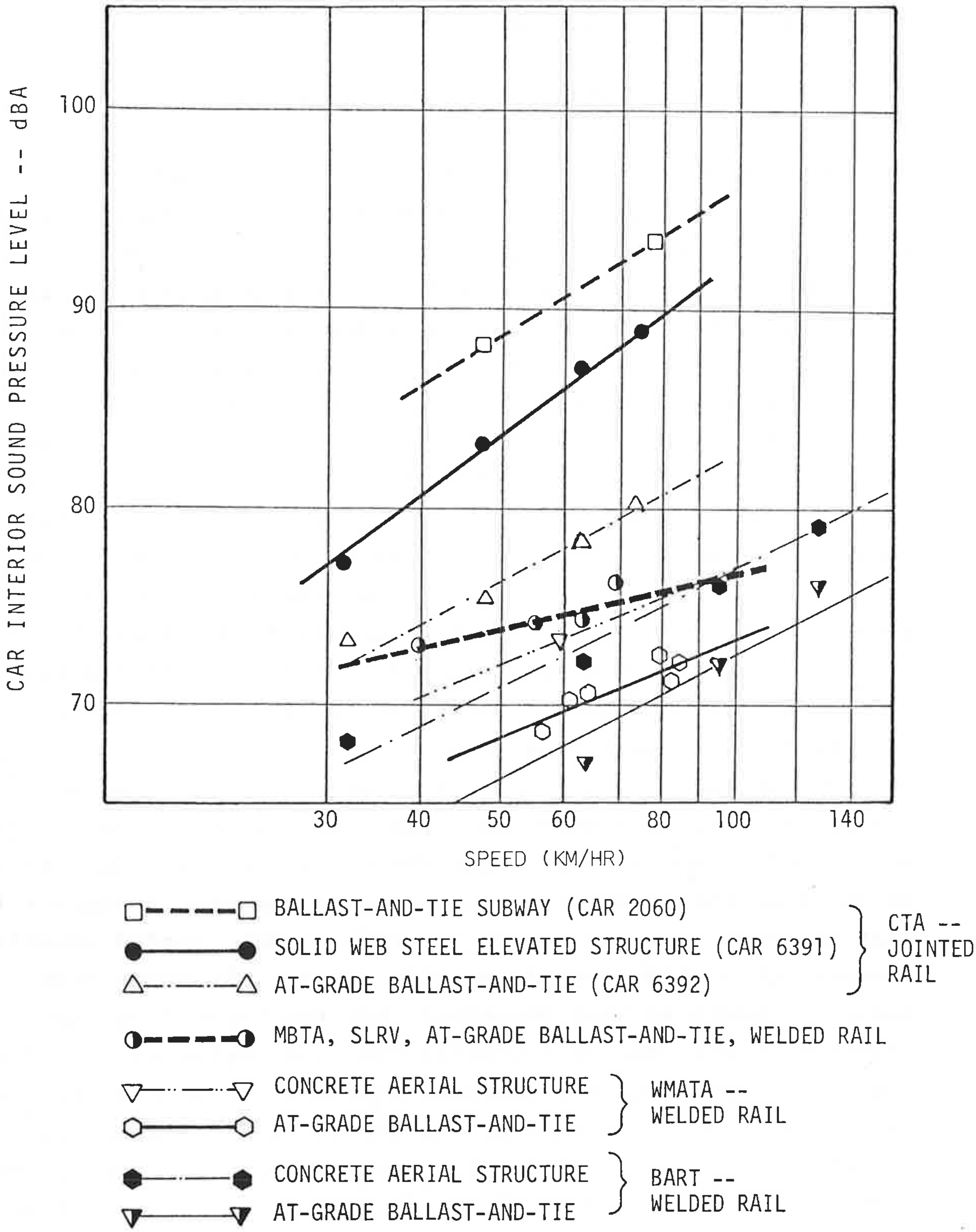


FIGURE 3.5 OVERALL CAR INTERIOR NOISE AS A FUNCTION OF SPEED

- Subway tunnel with ballast-and-tie track or concrete trackbed with absorptive treatment — +5 to 7 dBA

- Concrete trackbed (at-grade or aerial) — +2 to 6 dBA

At operating speeds, the two most important sources of car interior and wayside noise are the propulsion equipment and the wheel/rail noise. These noise sources are generally of the same order of magnitude at high speeds, but the wheel/rail noise is several dBA above the propulsion equipment noise at lower speeds. However, there are cases where the propulsion equipment noise dominates. For example, measurements at SEPTA (Ref. 3.17) and NYCTA (Ref. 3.18) showed that at high speeds the propulsion equipment created as much, or more, noise than the wheel/rail interaction. Figure 3.6 presents measurement data for propulsion equipment noise as a function of train speed. Since it is usually not possible to disengage the propulsion motors from the wheels, wheel/rail noise cannot be measured independently of propulsion equipment noise. However, by supporting the car so that the wheels can spin freely, the level of the propulsion equipment noise can be measured directly. These measurements provide data on the noise emissions of the traction motors, reduction gears, and other components of the propulsion system under no-load conditions. When the wheels spin freely, the noise emission of the propulsion system may be somewhat different from that obtained when the system is under normal, loaded conditions. However, this difference is usually minor. The noise from reduction gears is load-dependent, but the noise from the traction motors usually dominates the gear system noise. Since the loudest sources of propulsion equipment noise are the cooling fan and the turbulent airflow through the motor components (referred to as motor windage), both of which are independent of load, propulsion equipment noise is almost independent of load.

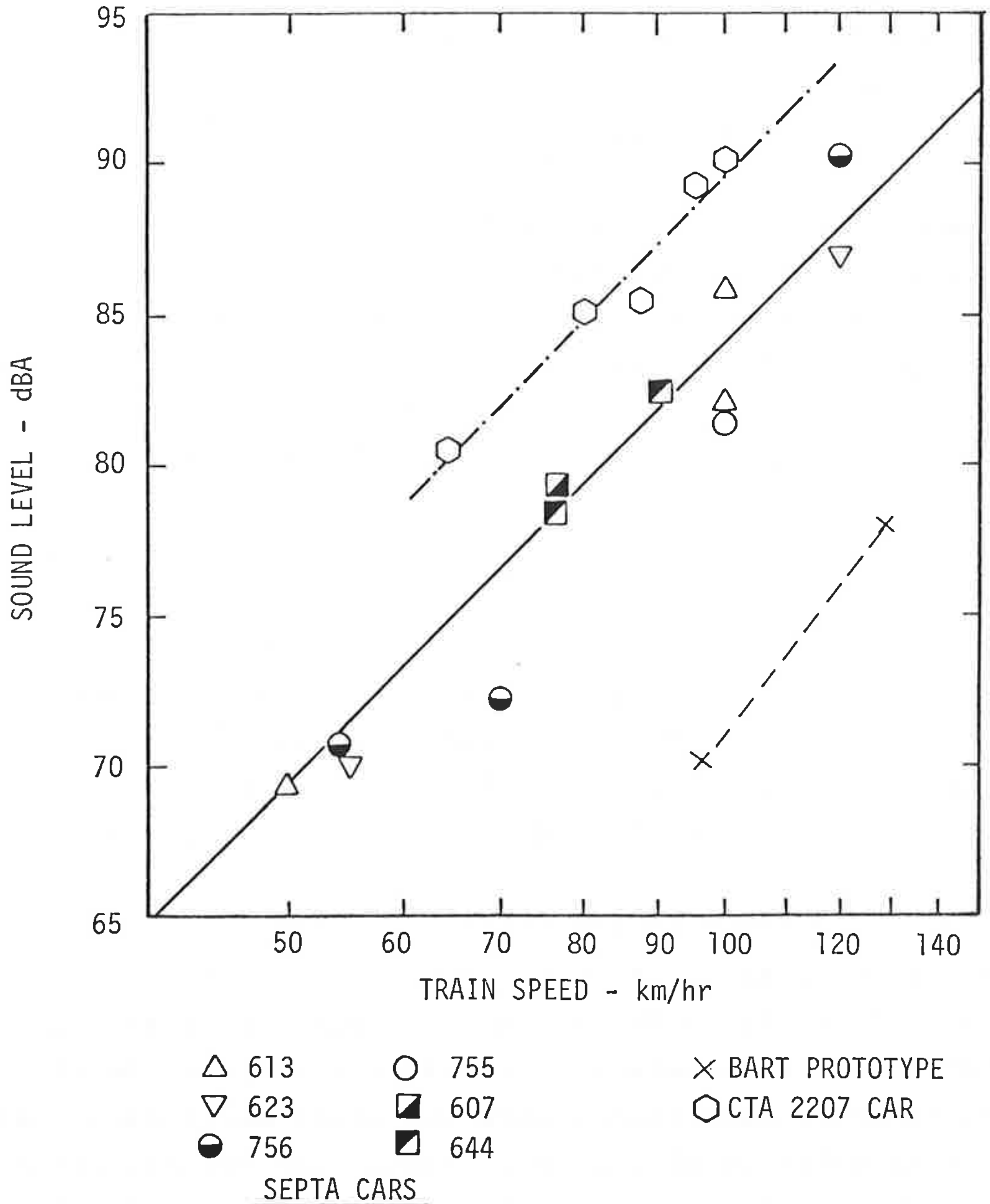


FIGURE 3.6 LEVELS OF PROPULSION EQUIPMENT NOISE AS A FUNCTION OF SPEED - CAR INTERIOR

Note that as shown in Figure 3.5, the noise levels inside the MBTA LRV's are not strongly dependent on train speed. This is a result of auxiliary equipment such as fans and the air-conditioning equipment being as dominant a source of noise as the propulsion equipment and the wheel/rail noise. The noise emissions of most auxiliary equipment are the same when the train is at rest or moving.

The overall interior noise levels in new transit cars, when operating on various types of way structures, are very close to their design goals and are similar to those found on the quietest European systems. In fact, in many cases the noise levels are even lower than those found on rubber tire transit systems. This results in a comfortable acoustical environment for patrons.

Figures 3.7 and 3.8 present octave band charts of noise inside the cars of various transit systems. Figure 3.7 shows the noise levels inside BART and WMATA Metro trains, and indicates the variations in noise levels that occur on different types of way structures. The data also indicate that noise levels inside BART and WMATA Metro cars are quite similar, both in spectrum shape and A-weighted level, under comparable operating conditions.

Figure 3.8 compares the noise level spectra of the BART car interiors with those of the rubber tire and steel wheel transit cars of the Paris Metro, under comparable operating conditions. Note that the levels are generally higher inside the rubber tire Paris Metro cars than inside the steel wheel cars. Since the actual noise level inside a transit car depends strongly on the sound insulating characteristics of the car body, Figure 3.8 does not indicate that steel wheel systems are quieter than rubber tire systems; rather, this indicates that proper design practices will produce comfortable noise levels inside both steel wheel and rubber tire systems. Indeed, the levels will be comparable to those inside luxury class passenger cars. Caution should be exercised with the data presented in Figure 3.8 since the Paris Metro cars are not air-conditioned and have operable windows, while the BART cars are air-conditioned with inoperable windows.

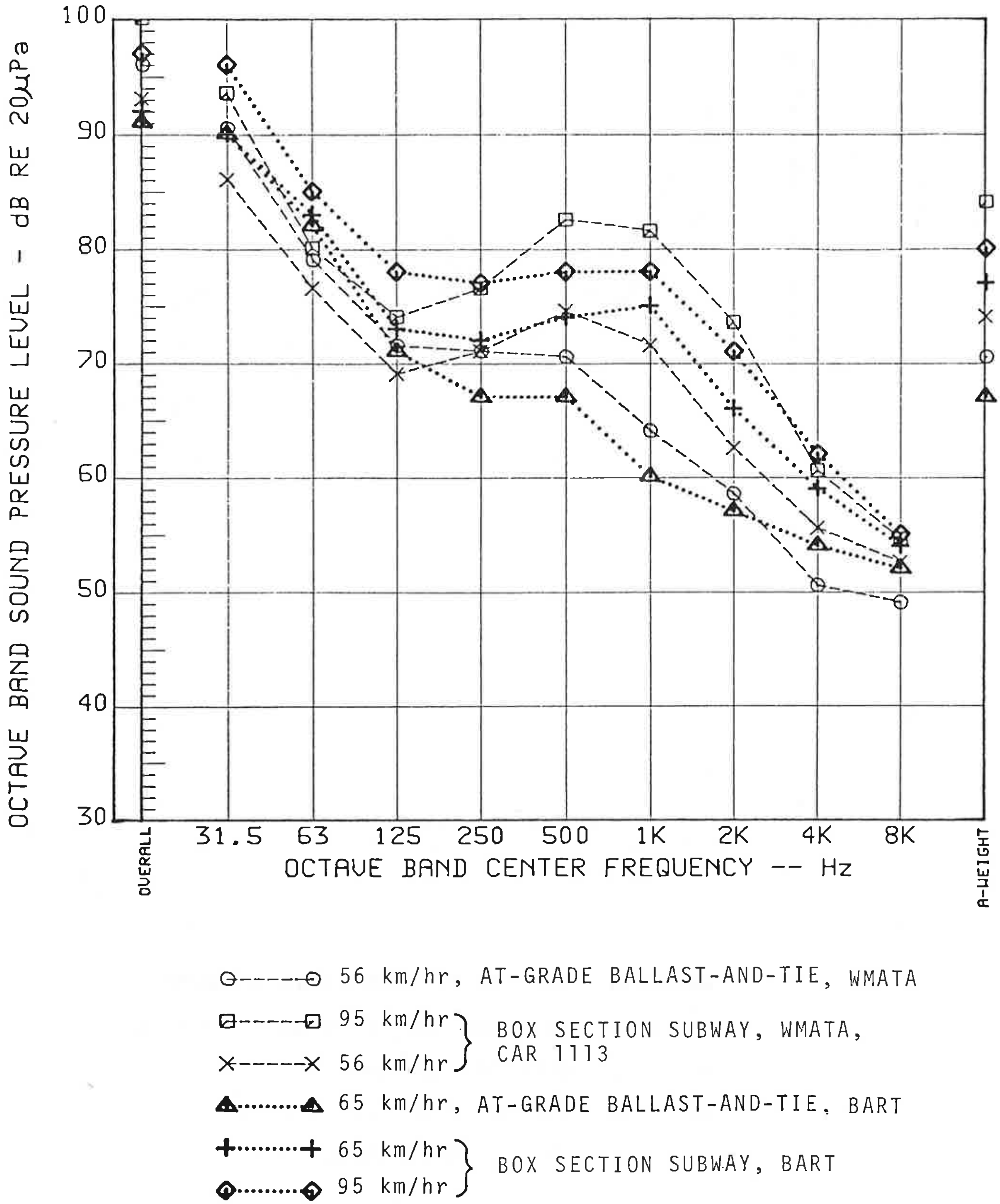


FIGURE 3.7 CAR INTERIOR NOISE SPECTRA

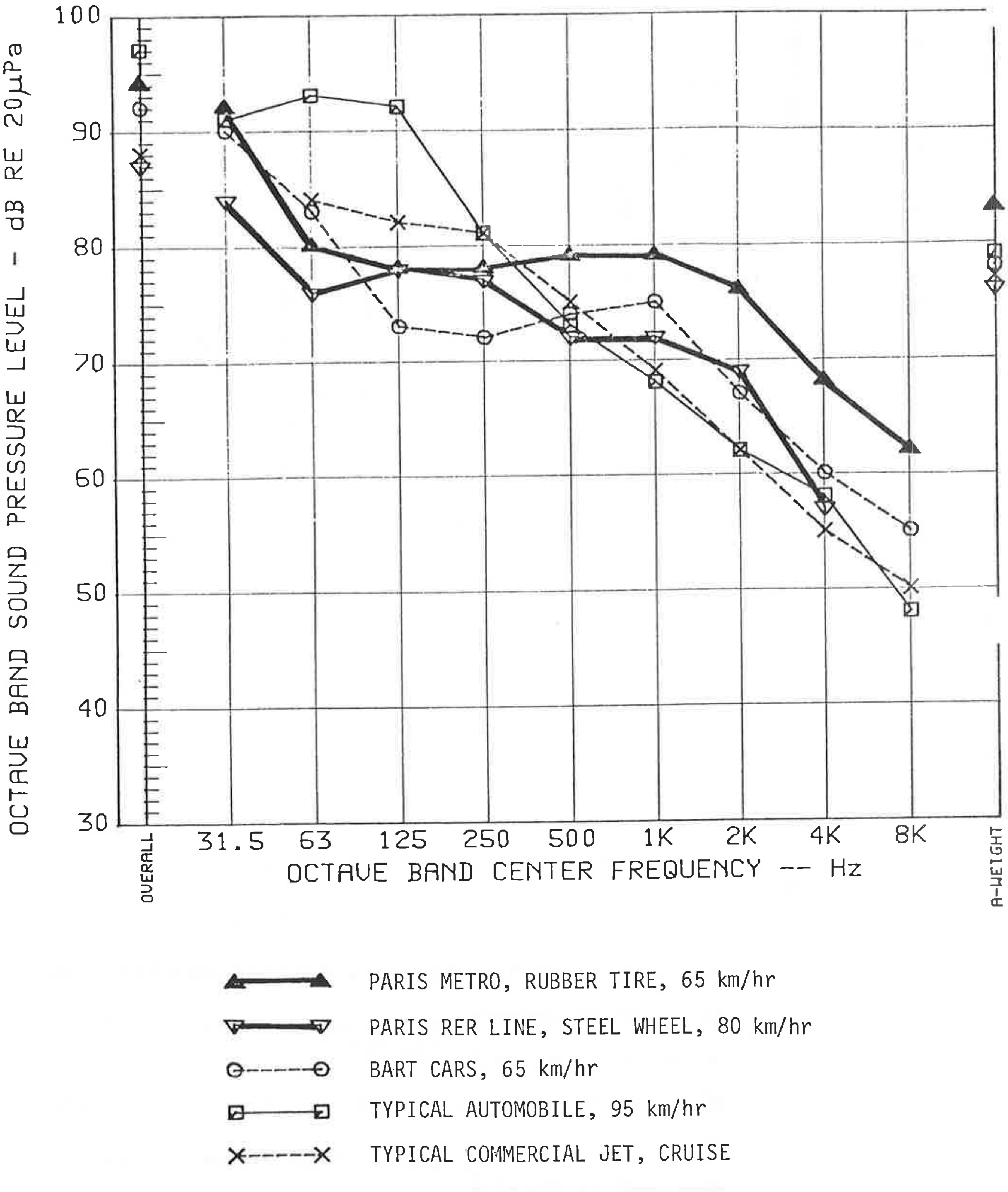


FIGURE 3.8 NOISE INSIDE DIFFERENT PASSENGER VEHICLES

Operable windows are often the weakest link in the sound insulation of the car body.

An important factor to consider when evaluating car interior noise or when developing specifications for car interior noise is the effect of buzzes, rattles, and pure tones. Because of the unique character of such sounds they can be an important source of annoyance even though they do not significantly increase the overall noise level. Typical sources of such noise are choppers, resonantly vibrating trim panels, and rattling of car body or trim components that are not properly clamped. In most cases rattles, buzzes and other unusual noises can be controlled with proper maintenance or minor design modifications.

3.2 VEHICLE VIBRATION

The vibration of the transit vehicle originates from two basic sources: the rolling of the wheels on the rails and the vibration transmitted through the mounts of auxiliary equipment. The vibration from train movement falls into the field of ride quality and is beyond the scope of this handbook; ride quality is strongly dependent on the design of the truck and the condition of the wheels and rails.

The vibration of the auxiliary equipment, however, can greatly influence the comfort of the passenger environment. Since it is generally a continuous vibration that occurs even when the car is at rest, it is related to equipment noise and vibration control, not ride quality. Air compressors, motor generator or alternator sets, ventilation fans, and hydraulic power units are particularly likely to cause excessive vibration.

The vibration levels are strongly dependent upon the configuration used to mount the equipment to the car body and (for rotating equipment) the balance of the equipment. To ensure that the vibration from the auxiliary equipment does not degrade the passenger environment, it is important to include vibration specifications in the purchase documents for any new transit vehicles and in those for any major modifications of existing transit vehicles (e.g., the addition of air-conditioning systems). Recommendations for auxiliary equipment vibration specifications are included in the APTA Guidelines (Appendix B). The preparation of the noise and vibration sections of transit car specifications is discussed in Chapter 5.

Figure 3.9 illustrates specifications for ride quality and auxiliary equipment vibration that have been used by several transit systems. Note that much lower levels of vibration are required for the auxiliary equipment. There are several reasons for this. Since the auxiliary equipment usually runs continuously even when the car is at rest, it would not be appropriate for them to generate vibration with amplitudes comparable to those of a car running at maximum speed. In addition, at lower frequencies the auxiliary equipment specification limits the maximum deflection to 0.10 in. (2.5 mm). The reason for this limit is to ensure that the auxiliary equipment will not cause visible movement of the car body components. An important point to note is that the acceptable vibration limits are sometimes given in terms of rms vibration and other times in terms of peak vibration. In Figure 3.9, the specifications have all been translated approximately to rms acceleration levels, assuming a stationary random vibration. When verifying compliance with a specification for peak vibration amplitude,it is necessary to directly measure the peak level; simply measuring the rms level may be misleading. Measurement of rms and peak vibration is discussed in Chapter 4.

Only limited data are available concerning the vibration of transit cars caused by auxiliary equipment. The most common reason for performing measurements is to determine whether the contract specifications have been met. One relatively complete set of tests was performed by Wilson and Wolfe (Ref. 3.13) on ten BART cars. The primary objective of the tests was to provide

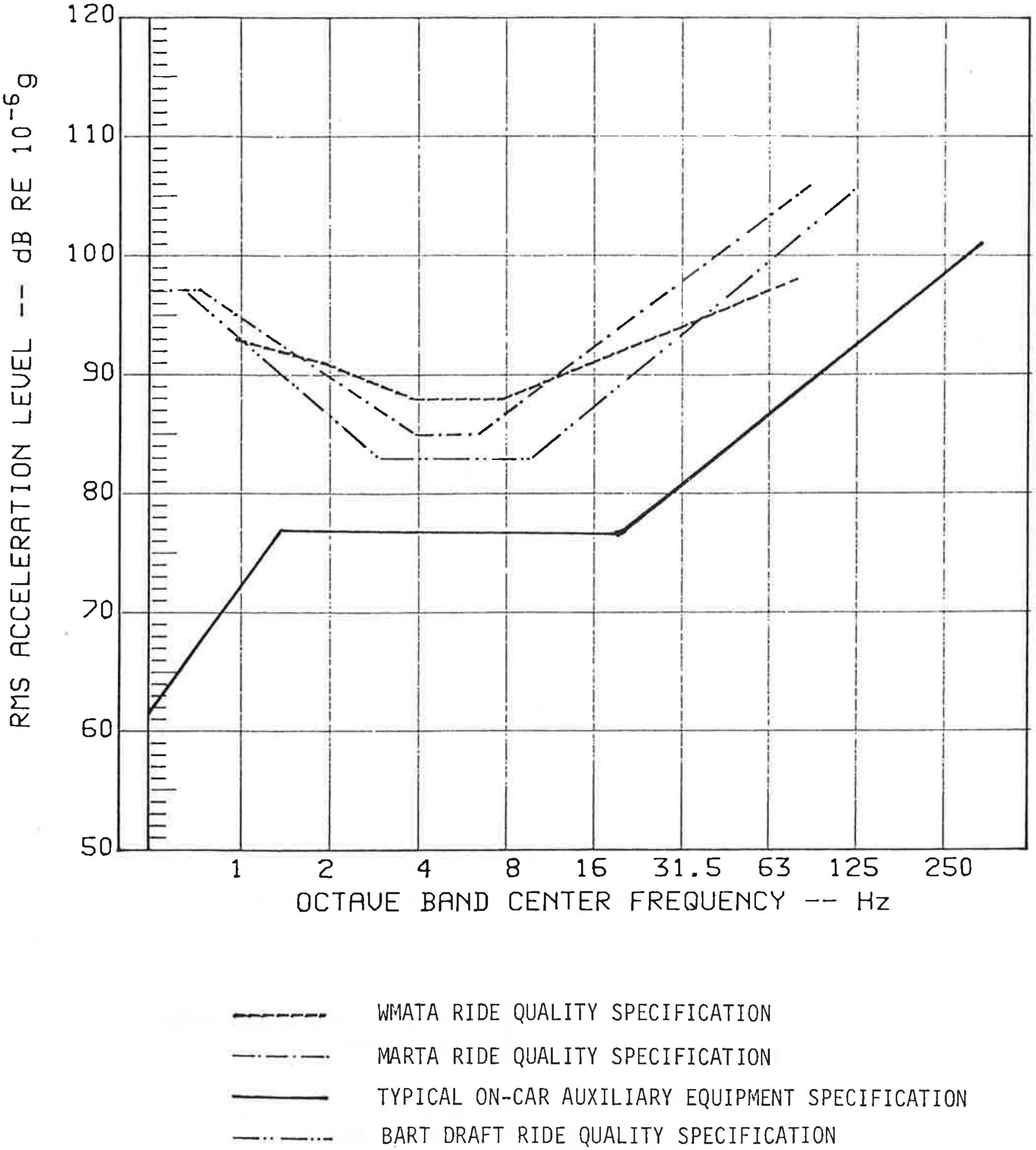


FIGURE 3.9 SPECIFICATIONS FOR CAR VIBRATIONS

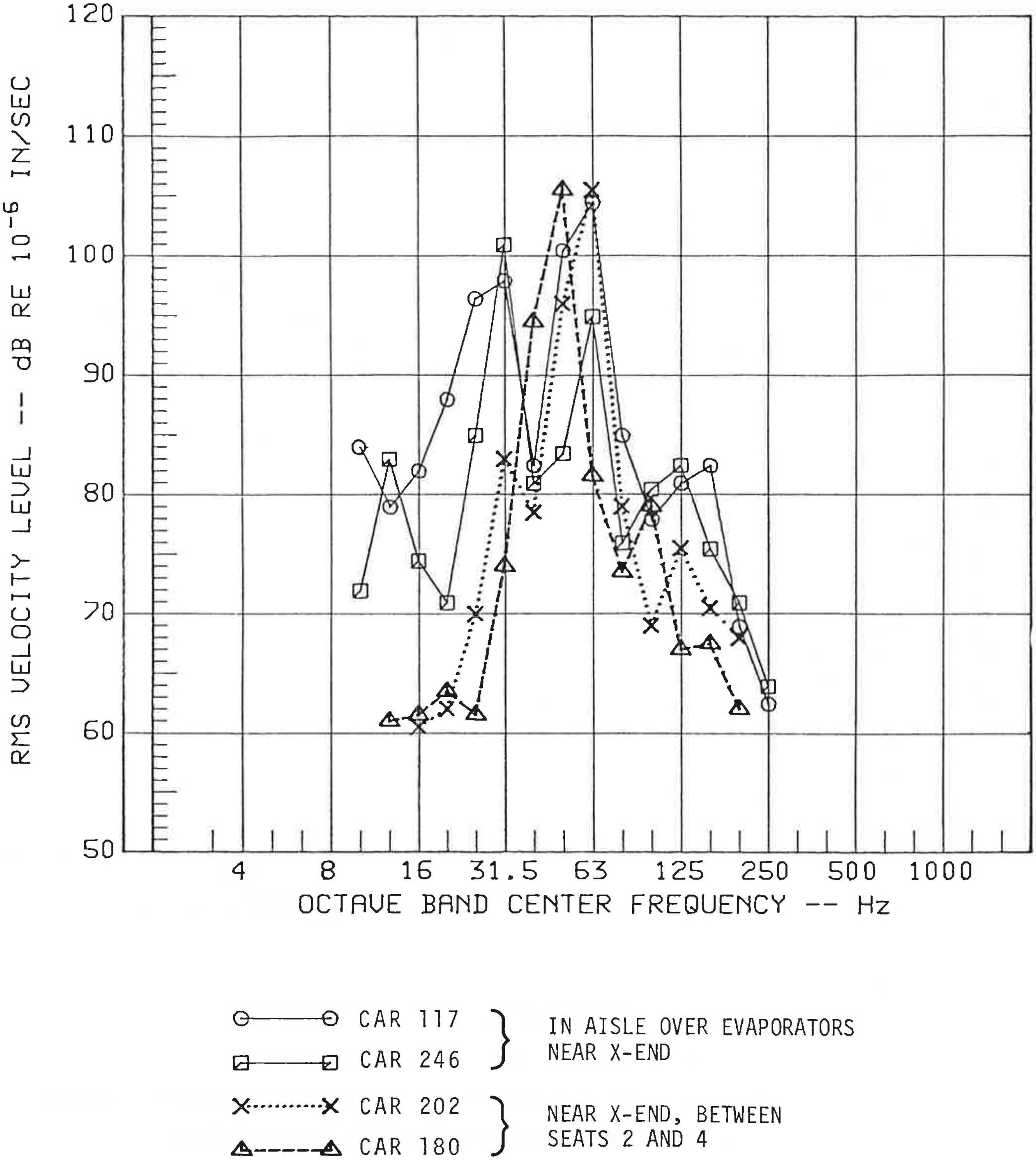


FIGURE 3.10 MEASUREMENTS OF FLOOR VIBRATION OF STATIONARY BART CARS -- ALL AUXILIARY EQUIPMENT OPERATING EXCEPT AIR COMPRESSOR AND HYDRAULIC POWER UNITS

definitive data on the floor vibration levels caused by the underfloor auxiliary equipment on BART cars. Figure 3.10 summarizes some of the results of these measurements. The measurements showed that the tested vehicles failed to meet the specification for vibration velocity above 10 Hz. However, in many cases the specifications were exceeded by only a small amount. Analysis of the floor vibration for the major frequency components showed that for most of the equipment, the predominant frequency of the floor vibration was the rotational frequency of the units. Since the vibration was primarily associated with the rotational frequency, it would be possible to meet the specification by improving the balance of the rotating parts, softening the vibration isolation mounts, and possibly, stiffening the support beams and improving the bearing smoothness.

3.3 WAYSIDE NOISE

Noise radiated from trains and track structures into the adjacent communities is usually the most serious acoustical problem faced by planned or existing rail transit systems. In this section, the sources of noise and techniques for predicting the levels at various distances from the track will be considered. We are concerned only with airborne noise here - noise that travels from the transit structure to receivers in the community by airborne paths. Although this definition includes structure-radiated noise (e.g., elevated structure noise), structure-radiated noise is important enough to deserve a section to itself, and is covered separately in Section 3.4. Another important source of community intrusion by transit operations is groundborne noise and vibration. This topic is also covered separately in Section 3.5.

3.3.1 Components of Wayside Noise

There are four basic sources of wayside noise:

- Wheel/rail noise. This is the noise that is radiated directly from the vibrating wheels and rails.

- Propulsion equipment. This includes noise from traction motors, cooling fans for the traction motors, and the reduction gears.

- Auxiliary equipment. Compressors, motor generators, and ventilation systems can all be noise problems.

- Elevated structure noise. This is the noise radiated by the vibration of the transit structure, excited by the passby of a transit train.

Each of these sources of noise can dominate the overall wayside noise level under specific conditions. Figure 3.11 indicates the manner in which the wheel/rail noise, propulsion equipment (traction motors and reduction gears), and the auxiliary equipment add to the overall noise level. At very low speeds the noise level is determined by the auxiliary equipment. Of course, the noise from most auxiliary equipment is independent of speed. As the train speed, V, increases, the wheel/rail noise increases at a rate of about 30 log V, and the traction motor noise increases at a rate of 40 log V to 60 log V. On the diagram of Figure 3.11, the wheel/rail noise dominates in the speed range of 35 to 50 km/hr. At about 50 km/hr, the wheel/rail noise and the noise of the traction motor fan are equal, and above 50 km/hr, the motor fan noise becomes dominant. Note that noise of the reduction gears, also shown in Figure 3.11, has a speed dependence of approximately 6 to 10 log V, and that at speeds below about 30 km/hr the gear system can be the dominant noise source. While this occurs on many older vehicles, the gear noise at low speed is not dominant on the newer cars with equipment noise limits. Of course at very low speeds, below about 15 km/hr, the auxiliary equipment noise usually dominates. The relative levels of the

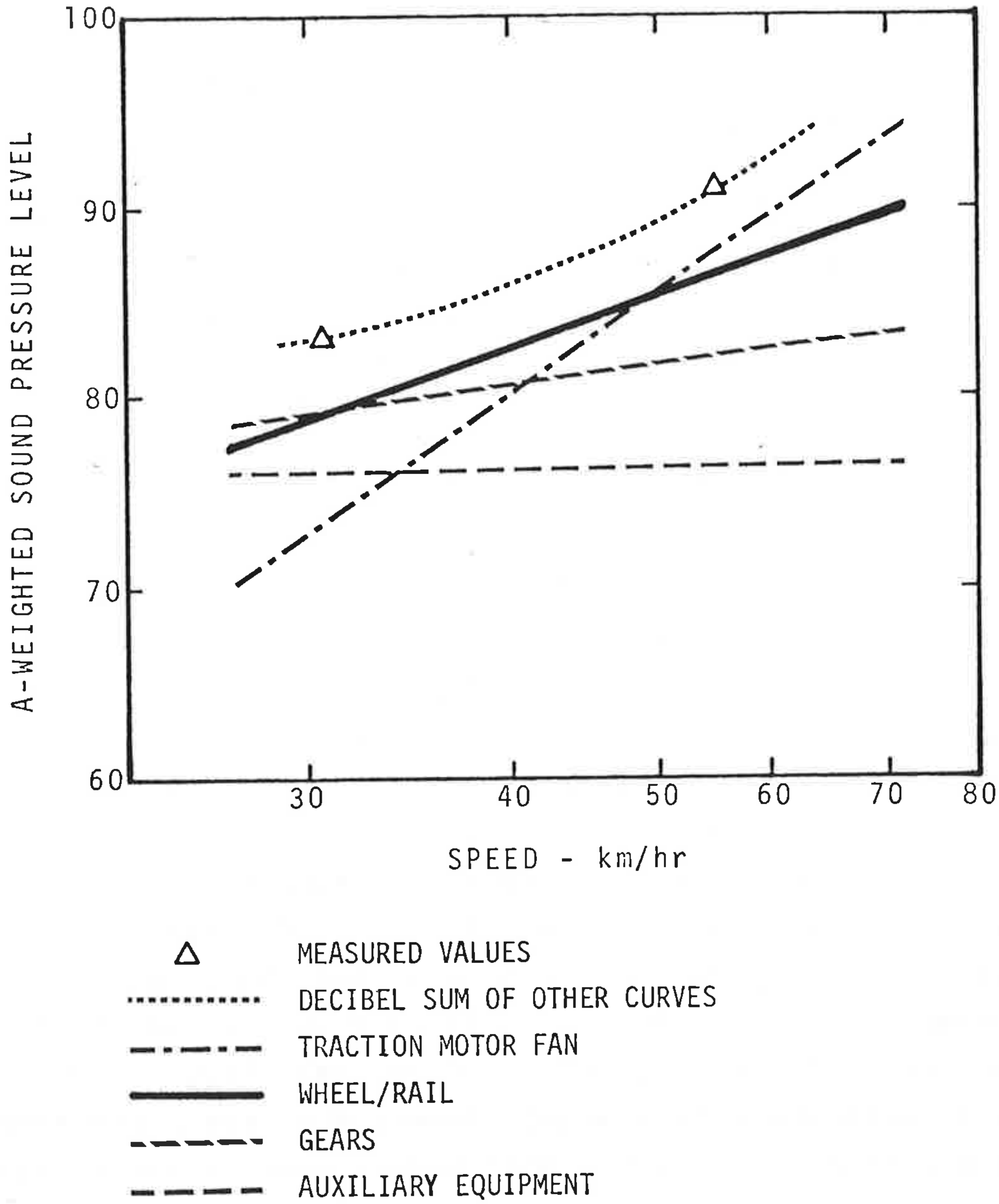


FIGURE 3.11 ESTIMATED RELATIVE CONTRIBUTION OF TRACTION MOTOR NOISE, GEAR NOISE, AND WHEEL/RAIL NOISE AT 3 m WAYSIDE LOCATION AND A BALLAST-AND-TIE TRACKBED (ADAPTED FROM REFERENCE 3.14)

various noise sources associated with transit cars are different on each type of car and can even be a function of the type of way structure.

In addition to the four basic sources listed above, as speeds become high enough, the aerodynamic noise of air passing over the transit car body will become an important component. With the present levels of wheel/rail and propulsion equipment noise, aerodynamic noise is not significant except at very high speeds, e.g., above 250 km/hr (150 mph). At the present state-of-the-art of intracity rapid transit, aerodynamic noise is clearly not a factor. It would only become important if train speeds were increased dramatically, or if methods that dramatically reduced wheel/rail and propulsion equipment noise were implemented.

The levels of wayside noise vary significantly between different transit systems. Modern systems with welded rails, resilient rail fasteners, and wheels and rails in good condition are much quieter than many of the older systems. There can also be significant variations within the same transit system, depending on the wheel and rail condition and on the type of transit car used. Figure 3.12 illustrates the results of measurements that were performed at CTA (Ref. 3.3). Welded rails resulted in significantly lower noise levels than the jointed rails. The noise levels were further reduced with the use of reduced primary spring stiffness via soft axle journal sleeves. On smooth welded rail with the soft journal sleeves the levels are comparable or lower than those on the most modern rapid transit systems. Also shown on the same figure are noise levels measured 15 m from the CTA steel elevated structures. Trains operating on the lightweight steel elevated structures produce the highest levels of wayside noise, and trains on the ballast-and-tie track with smooth rails, the lowest. On ballast-and-tie track the sound levels can be reduced even further by the use of special noise control features, such as sound barrier walls.

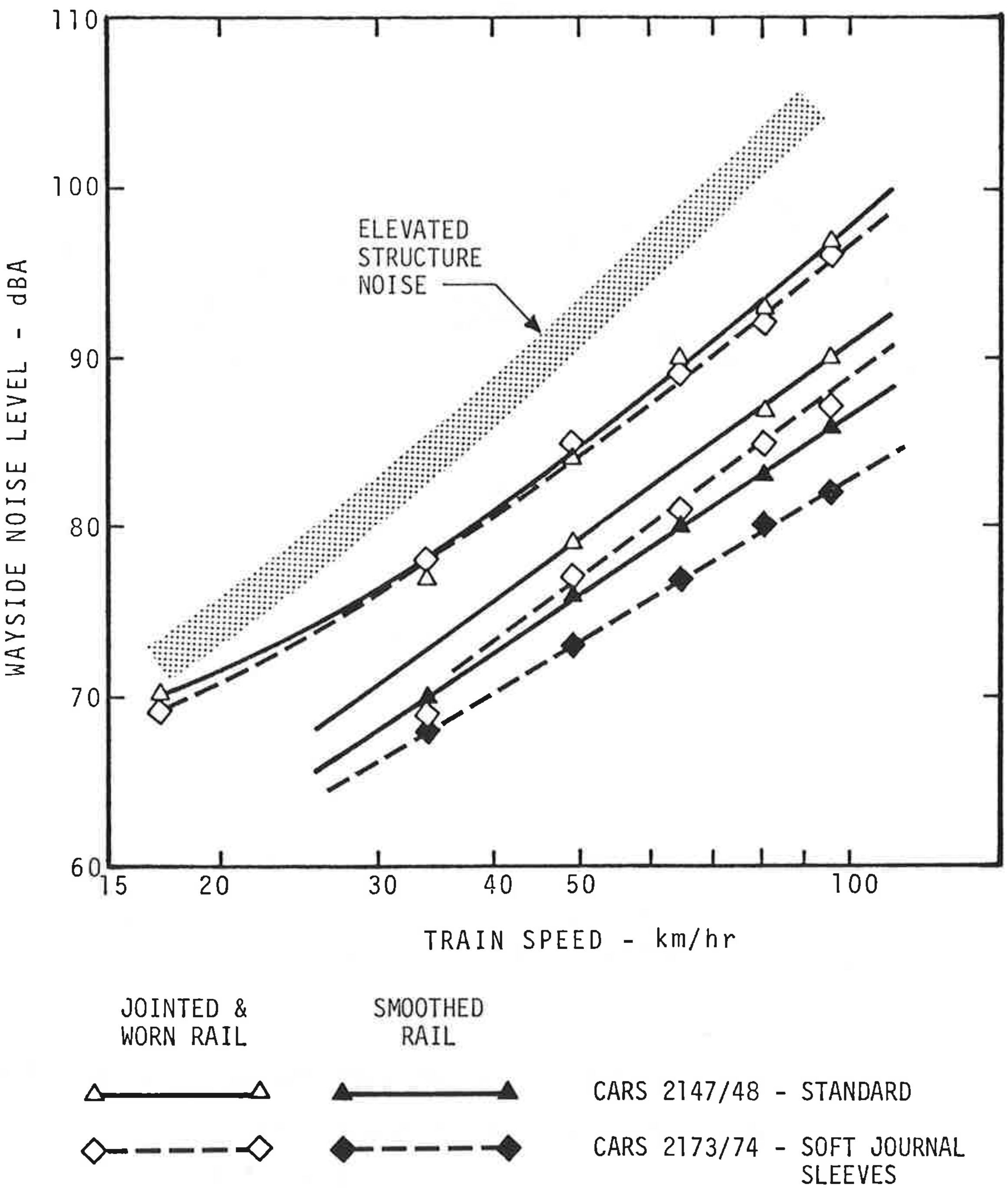


FIGURE 3.12 WAYSIDE NOISE LEVEL AT 15 m FOR THE CTA 2000 SERIES TRANSIT CARS ON VARIOUS RAIL CONFIGURATIONS, 2-CAR TRAINS - BALLAST-AND-TIE TRACK (ADAPTED FROM REF. 3.3)

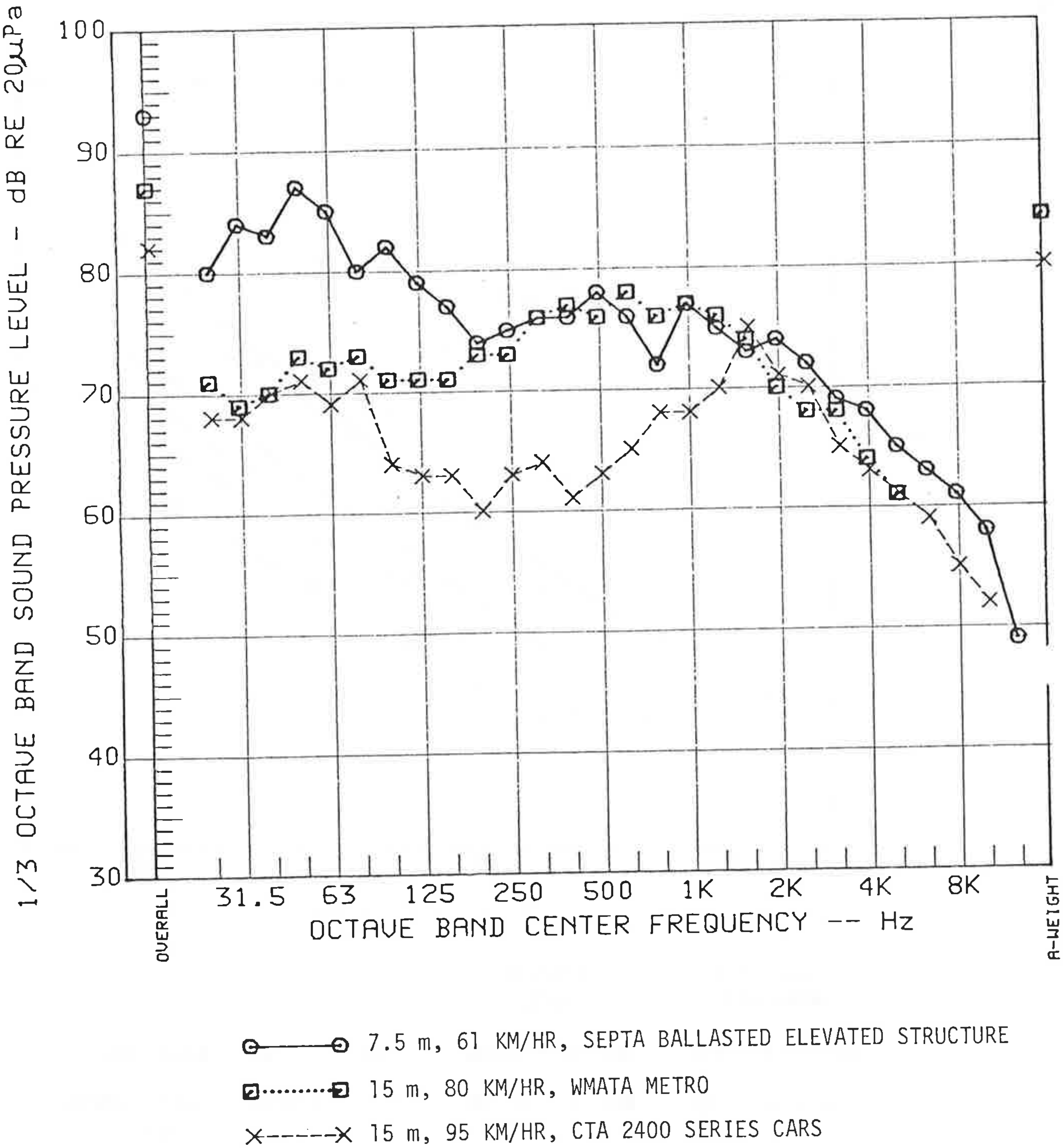


FIGURE 3.13 TYPICAL SPECTRA OF WAYSIDE NOISE, BALLAST-AND-TIE TRACK, WELDED RAIL

Typical 1/3 octave band spectra of wayside noise for several different transit systems are shown in Figure 3.13. With the possible exception of noise near composite, concrete deck, steel girder elevated structures, the spectra of wayside passby noise is usually broadband in nature with the 200 to 2000 Hz frequency range dominating the A-weighted noise levels.

3.3.1.1 Wheel/Rail Noise -- There are three types of wheel/rail noise: (1) roar noise caused by small-scale roughness of wheel- and rail-running surfaces that produces fluctuations in the interaction forces between wheels and rails; (2) impact noise created by discontinuities such as rail joints, wheel flats, or shelled or spalled areas on the wheel- and rail-running surfaces; and (3) squeal noise, produced by stick-slip action between the wheels and rails on short-radius curves.

The level of the roar noise is primarily a function of the roughness of the wheel- and rail-running surface. In general, reducing the wheel and rail roughness will reduce the level of roar noise. However, there appears to be a practical limit beyond which any further smoothing of the wheels and rails will not result in further noise reductions. Rail corrugations can significantly increase the level of roar noise. Rail corrugations have been observed to increase wayside noise levels by up to 10 dBA. As the corrugations grow in size, the mechanism of noise generation will change from roar to impact. Corrugations, or roughnesses similar to corrugations, have been observed on wheel treads as well as rail-running surfaces. This is not a significant problem at any of the North American transit systems. Apparently, there is very little chance of wheel corrugations developing when composition brake shoes are used.

Impact noise can be caused by any discontinuity on the wheel- or rail-running surfaces. Virtually all new rails are continuous welded rails. Although the principal reason for the use of welded rail is the reduction in maintenance costs, significant acoustical benefits are also derived. Figure 3.12 shows that the noise level for jointed rail can be 5 to 10 dBA higher than welded rail. Further, the impulsive character of noise from trains on jointed track (the familiar clickety-clack) makes the noise more noticeable and more annoying to many people.

Even on systems with welded rails, there will be some impact noise at special trackwork, such as crossovers. Since this noise generally cannot be avoided, it is important to select crossover sites carefully and to perform the necessary maintenance to keep the misalignments and wear at frogs and switches to a minimum.

Squeal noise is a feature of rail transit systems that many people find particularly annoying. The slip-stick action between the wheels and the rails on short-radius curves excites natural frequencies of the wheels. At specific frequencies, the squeal can increase the noise level as much as 30 dB. Figure 3.14 illustrates the effects of wheel squeal. The figure shows the time history and 1/3 octave band spectra for two train passbys on a short-radius curve. The first train was equipped with solid steel wheels and created intense levels of squeal as it traversed the curve. The second train was equipped with resilient wheels that had sufficient internal damping to virtually eliminate the squeal noise on the curve. The squeal in this example is made up primarily of high-frequency components, particularly components above 4000 Hz. Below 1000 Hz the noise levels for the two trains were virtually identical.

3.3.1.2 Propulsion Equipment Noise -- Propulsion equipment noise is often dominated by the noise from the traction motor cooling fans. There are two basic configurations available to supply cooling air to the traction motors. The first is self-ventilated traction motors: a fan attached on the shaft of each traction motor forces cooling air over the motor. This method is inherently noisy because of the high rpm of the fan, and the noise is relatively difficult to control. The fan is usually the

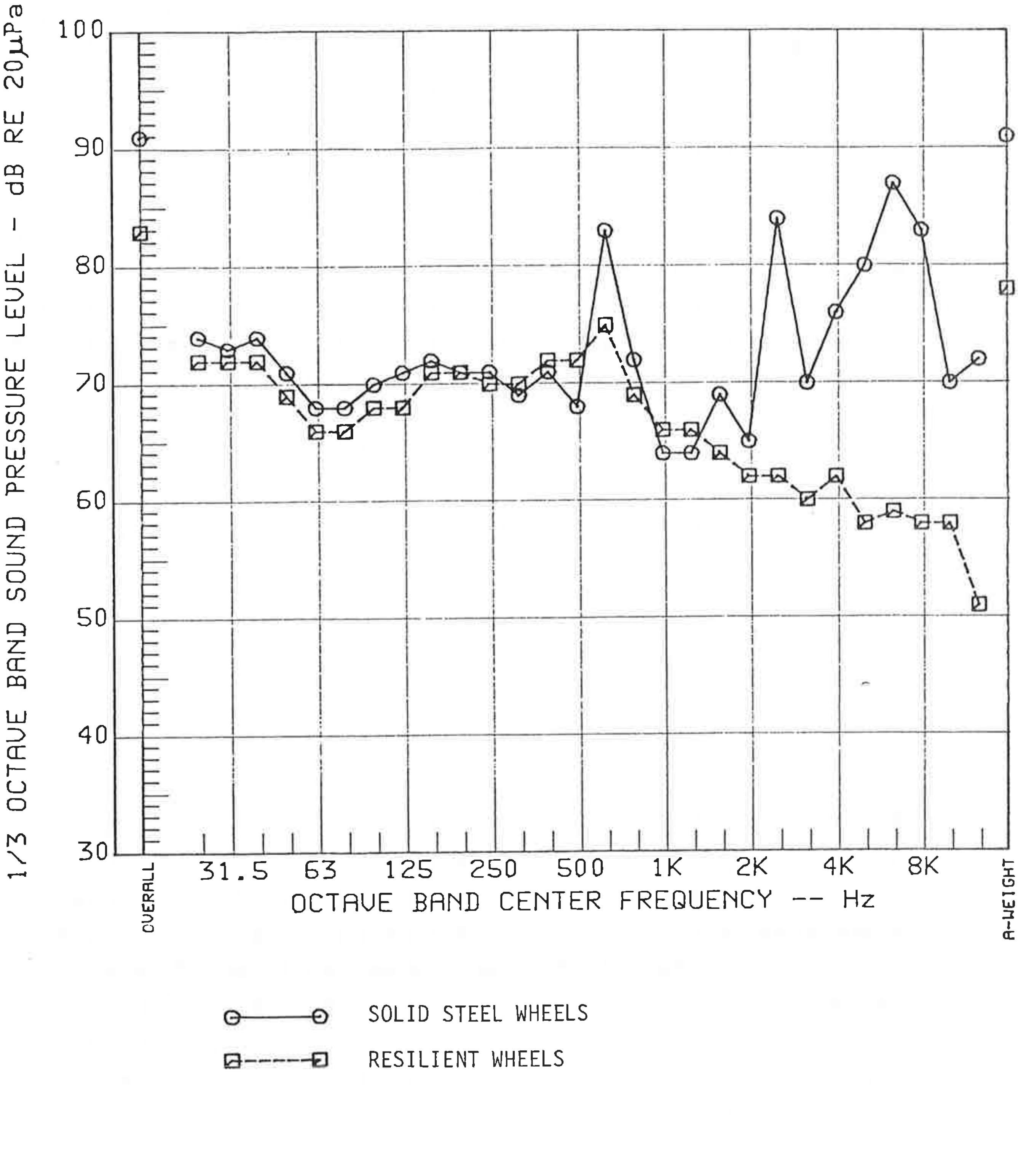


FIGURE 3.14 EXAMPLE OF SQUEAL NOISE ON A SHORT-RADIUS CURVE -- SEPTA 69TH STREET TURNAROUND

dominant source of propulsion equipment noise, the noise level depending upon the train speed. In addition, there is usually a strong tonal component at the blade passage frequency of the fan, which increases the annoyance quality of the overall noise level.

The second method of cooling the traction motor is to use a forced air cooling system. In this system, the cooling fans run at speeds independent of train speed, and since there is some flexibility in selection of fan type and location, straightforward noise control methods can be used to control the cooling system noise. Even with forced air cooling (although inherently quieter than cooling by shaft-mounted fans) the acoustical aspects of the car must be carefully considered to ensure that the cooling system is not a noise problem.

Figure 3.15 presents a summary of propulsion equipment noise measurement results for several different types of transit cars. The data show that the levels of propulsion equipment noise vary over a considerable range.

3.3.2 Prediction of Wayside Noise

It is often necessary to estimate the noise levels that will occur at various distances from a rail transit line. Whenever possible, the best technique is to use a sound level meter to measure the noise directly. Often this is impossible, either because of the number of locations where the sound level must be estimated (for example, when noise exposure contours are being constructed) or because the rail line has not been built. In such cases, whatever information available must be used to estimate the noise levels.

Other chapters of this handbook present information that can be used to estimate the noise levels at standard distances from the track structure. This section gives methods that can be used to extrapolate to other distances and for different lengths of

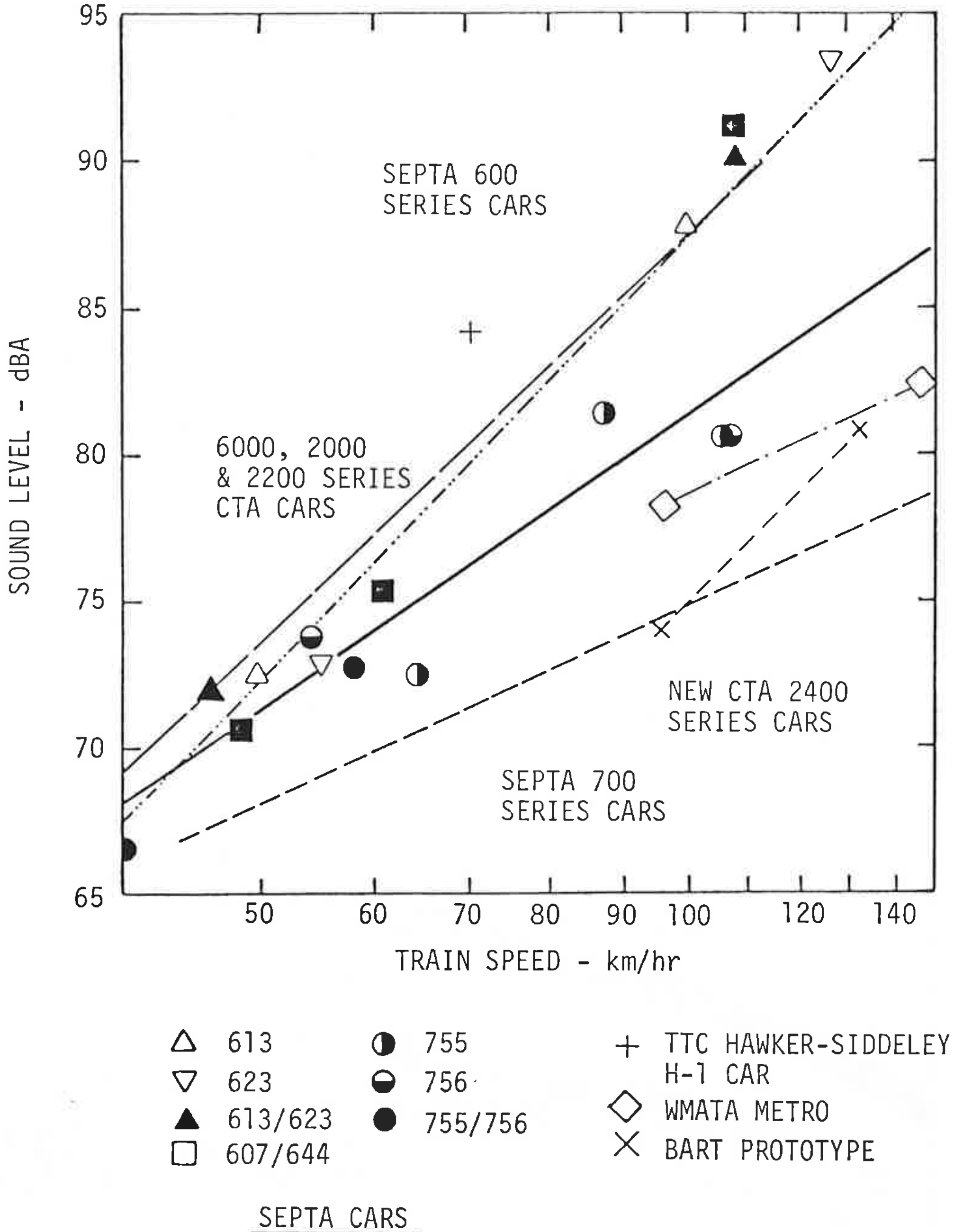


FIGURE 3.15 WAYSIDE NOISE LEVEL FOR PROPULSION EQUIPMENT AS A FUNCTION OF SPEED - 7.5 m FROM TRACK CENTERLINE

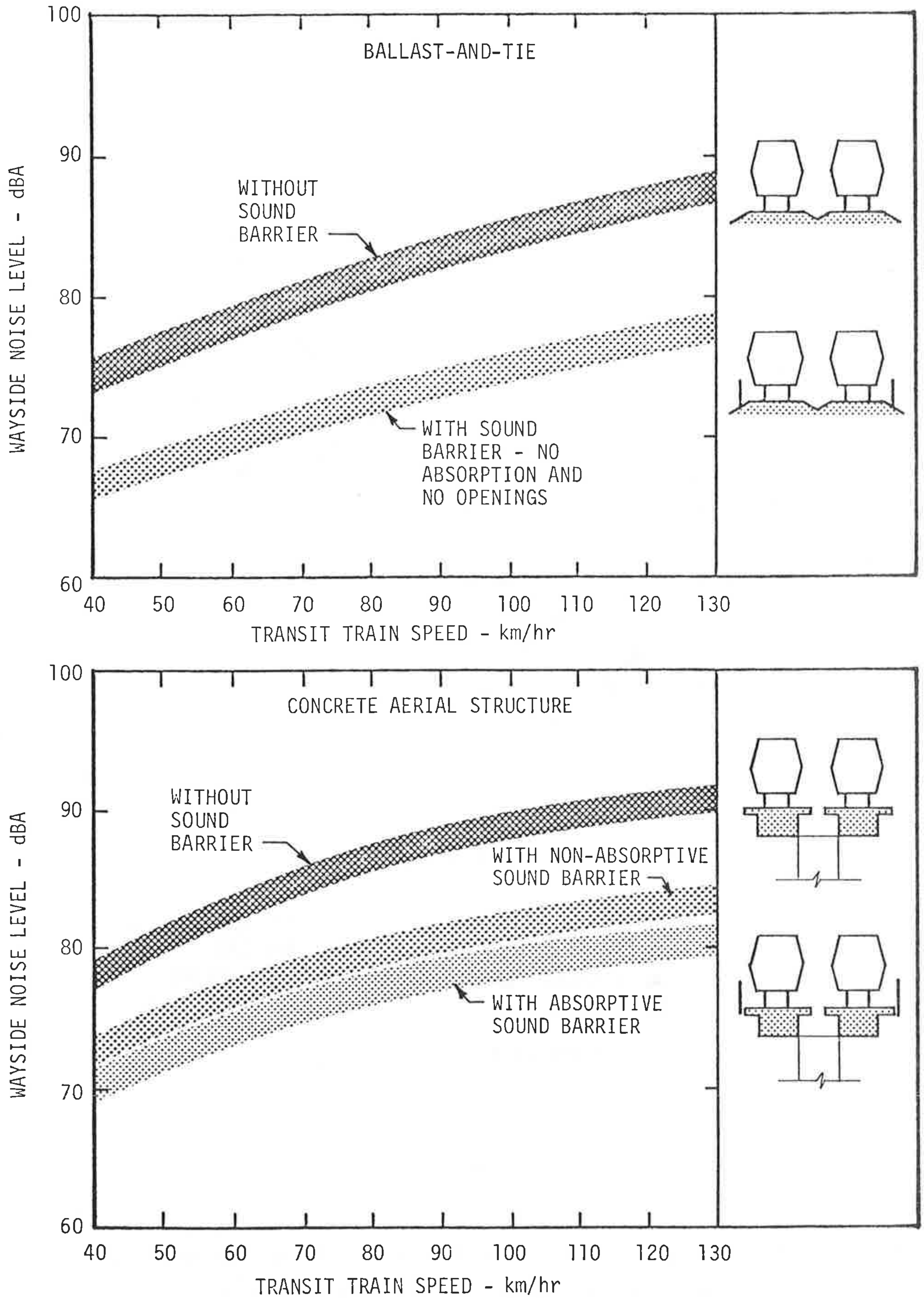


FIGURE 3.16 MAXIMUM WAYSIDE PASSBY NOISE LEVELS EXPECTED AT 15 m FROM TRACK CENTERLINE

trains. Figure 3.16 presents a set of curves that were developed for use in evaluating the environmental impact of the WMATA Metro system (Ref. 3.16a, b, c). The figures show examples of the wayside noise level as a function of train speed at 50 ft (15 m) from the track for aerial and at-grade structures with and without sound barrier walls. When developing a set of curves such as these for a new transit system, it is necessary to consider the noise specifications for the transit cars and equipment on the cars, the noise levels created by existing, similar transit cars, the effects that the transit structures may have on the noise level, and the noise reduction that will be achieved with a specific sound barrier configuration.

Once the maximum level at a specific distance from the train is known, the method presented by Kurzweil (Ref. 3.6 and 3.15) is a good general approach for predicting the noise at various distances from a train. His basic formula in a slightly modified form is:

$$L_A = L_A(15\ \text{m}) - C_s - C_g - C_b \qquad (3.1)$$

where: L_A = the estimated maximum noise level at a specific distance from the tracks.

$L_A(15\ \text{m})$ = the level at 15 m (50 ft) from an 8-car transit train (total train length of 170 m, 560 ft).

C_s = attenuation factor due to geometric spreading of the sound energy; this figure also accounts for differences in train length.

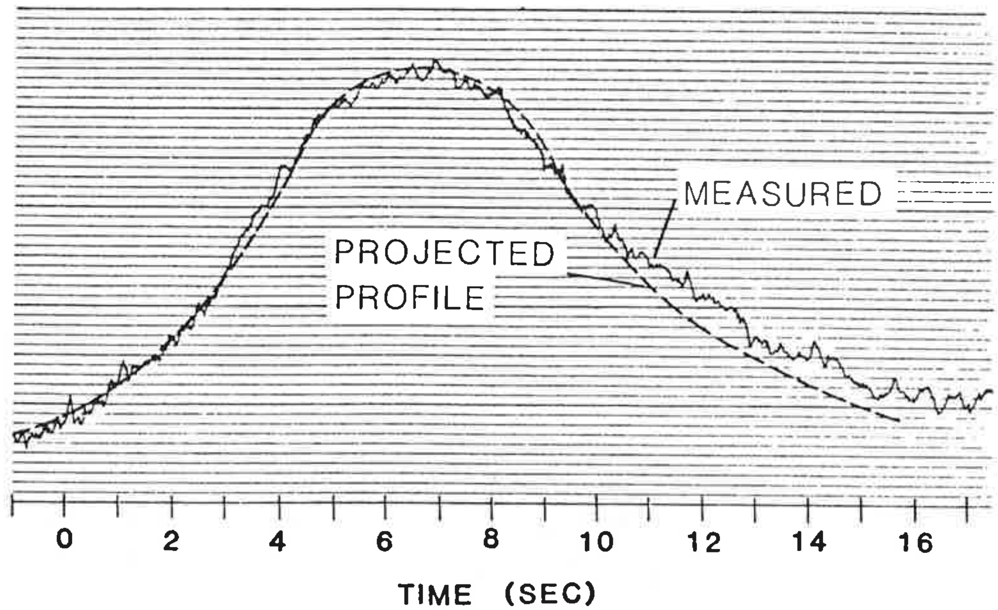
C_g = excess attenuation due to ground effects and air absorption.

C_b = excess attenuation due to obstructions blocking the line-of-sight between the observer and the train. The shielding can be provided by sound barriers, embankments, cuttings, or buildings.

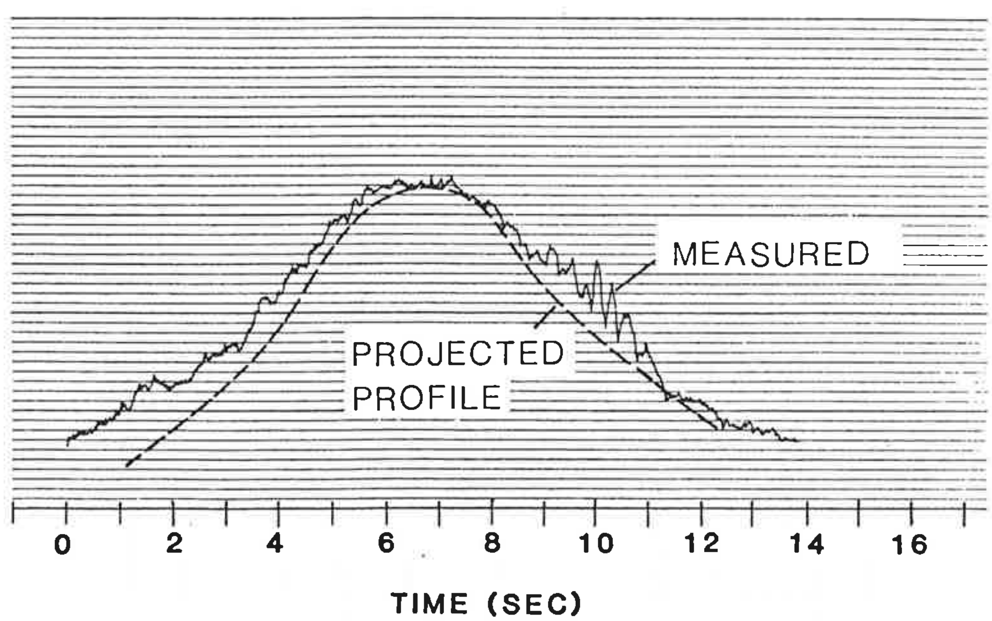
This formula provides a means of estimating the maximum levels at various distances from the track during a train passby. Note that before it can be used, it is necessary to know or estimate the value of L_A (15 m). If values of L_{eq} or L_{dn} must be predicted, the relationship between L_{eq} and L_A can be estimated using Equation 2.1.

3.3.2.1 Geometrical Spreading (C_s) -- Several researchers have investigated methods that can be used to predict the geometrical spreading of train noise. Their general conclusion is that the best approach is to model the train as an incoherent line source, with the directivity characteristics of a dipole. At the distances of interest in most practical situations, the predictions of maximum level are not very sensitive to the model that is used. However, the specific model has a strong influence on the shape of the sound level time history during a train passby. Figure 3.17 illustrates the accuracy with which the dipole line source model of Peters (Ref. 3.7) predicts the time history of a train passby. As is apparent, this model, although not perfect, gives very good predictions of the passby time history. This is a reflection of the fact that trains can be more accurately represented by dipole, or directional source, models than by the simpler monopole, or nondirectional source, models.

Figure 3.18 is a graph that can be used to predict the geometrical spreading factor, C_s. These curves are also based on the Peters' dipole line source model (Ref. 3.7). Close to a train the sound level decreases 3 dB with every doubling of distance from the train. However, when the distance from the train to the receiver is significantly greater than the train length, the train looks like a point source and the sound level decreases 6 dB with every doubling of distance. Theoretically, this chart is not valid at distances closer than $2/\pi$ times the average truck-to-truck spacing. For typical transit cars this distance is 4.5 to 7.5 m (15 ft to 25 ft). At distances closer than this, the line source model breaks down and the passby of each truck creates a peak in the sound level. At these close distances, the peak level increases by 6 dB for each halving of distance. However, human



#1 NEAR TRACK
30 METERS



#2 FAR TRACK
34 METERS

FIGURE 3.17 MEASURED AND PREDICTED PASSBY TIME HISTORY - 5-CAR BART TRAIN AT 110 km/hr (70 MPH)

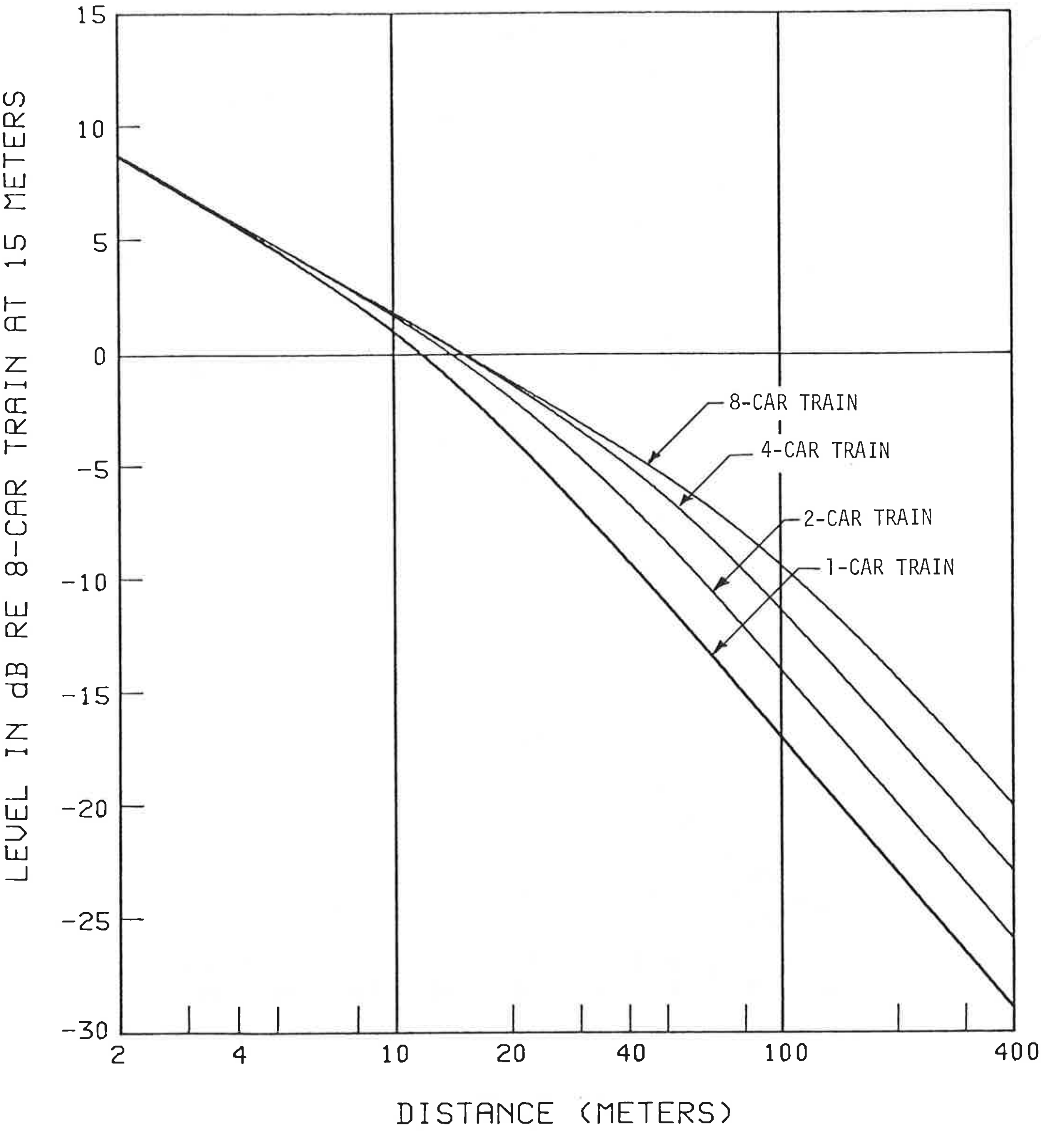


FIGURE 3.18 GEOMETRIC SPREADING FACTOR FOR TRANSIT TRAINS [LEVELS ARE IN dB RELATIVE TO AN 8-CAR TRAIN AT 15 m (50 FT), A TRANSIT VEHICLE IS ASSUMED TO BE 21.3 m (70 FT) LONG]

response is more closely correlated to rms level than to peak level. The curves of Figure 3.18 give reasonably accurate estimates of the rms level during a train passby, even at locations very close to the track.

3.3.2.2 Ground Attenuation and Air Absorption (C_g) -- As the distance from the track increases, the effects of ground attenuation, air absorption, temperature gradients, wind gradients, and atmospheric turbulence become increasingly important. The absorption of the sound energy by the air depends on temperature and humidity as well as on frequency and distance. Although these effects can be very important, they are not usually significant at distances of less than 100 m from the transit structure.

Figure 3.19 presents curves that can be used to estimate the excess attenuation due to ground effects and air absorption, for transit cars at-grade and on railroad bridges (equivalent to aerial structures). According to Kurzweil (Ref. 3.15), these estimates are conservative and actual attenuation will sometimes be greater. The curves are only valid for combinations of temperatures (degree F) and relative humidities (%) whose product is greater than 4000. In addition, ground interference effects that can result in increased attenuation in the frequency range of 125 to 500 Hz are not accounted for in Figure 3.19.

3.3.2.3 Barrier Attenuation (C_b) -- Any time an obstruction blocks the direct line-of-sight between the noise source and the observer, there will be a significant reduction in the observed sound level. Whether the obstruction is a specially built sound barrier wall, an embankment, the edge of a cut, or a building, the attenuation can be estimated using relatively standard methods. The most common method is based on the theoretical relationship between attenuation and path length difference with and without the barrier. Path length difference is defined in Figure 3.20.

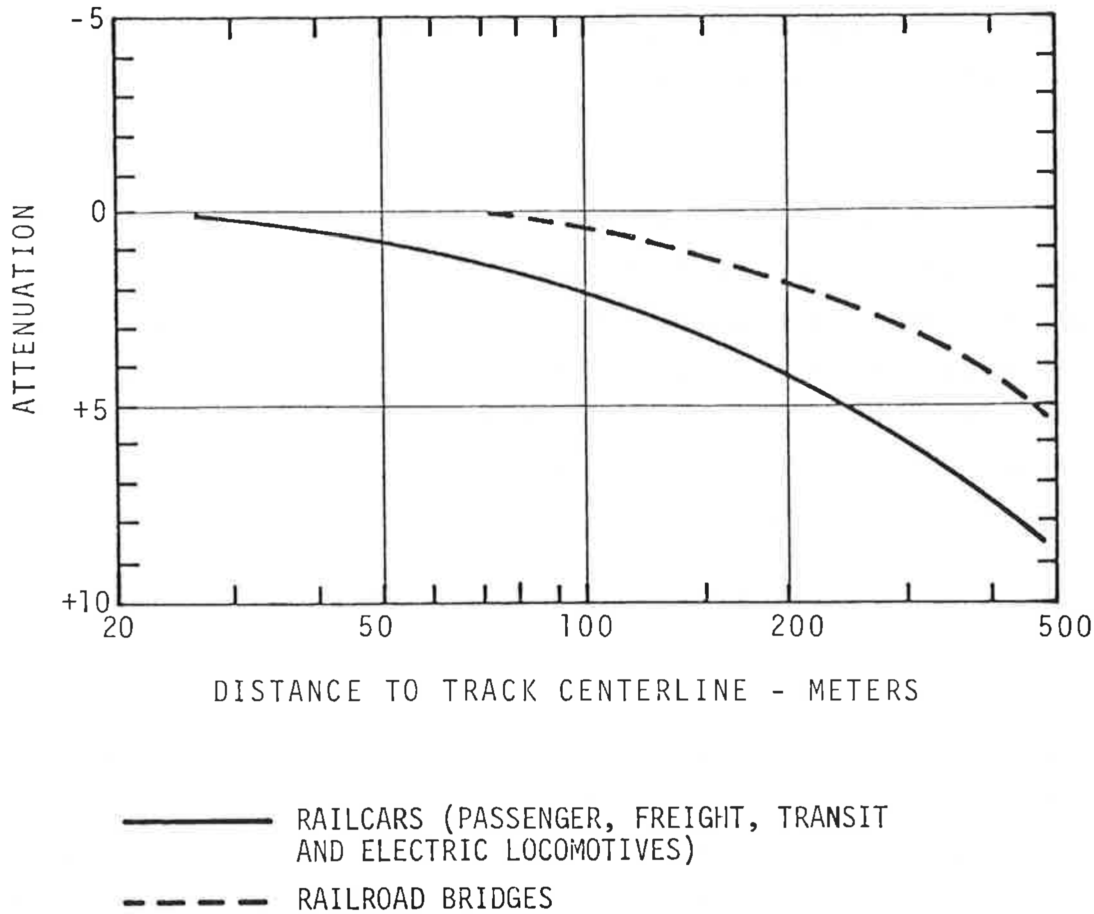
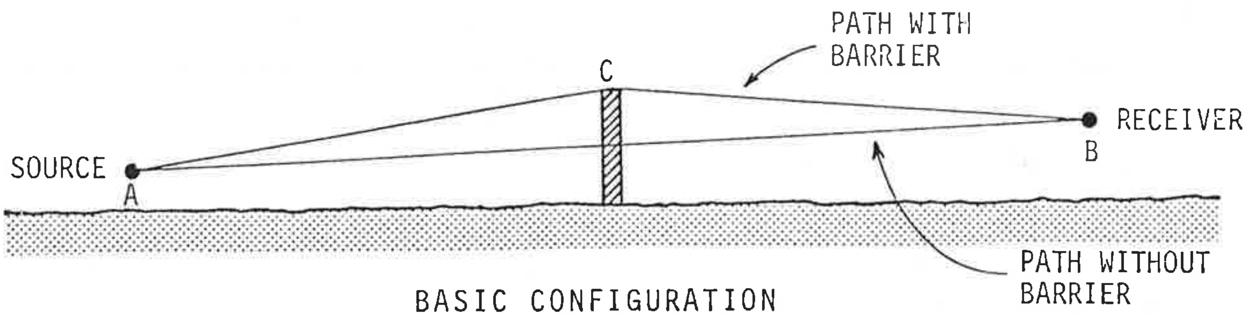
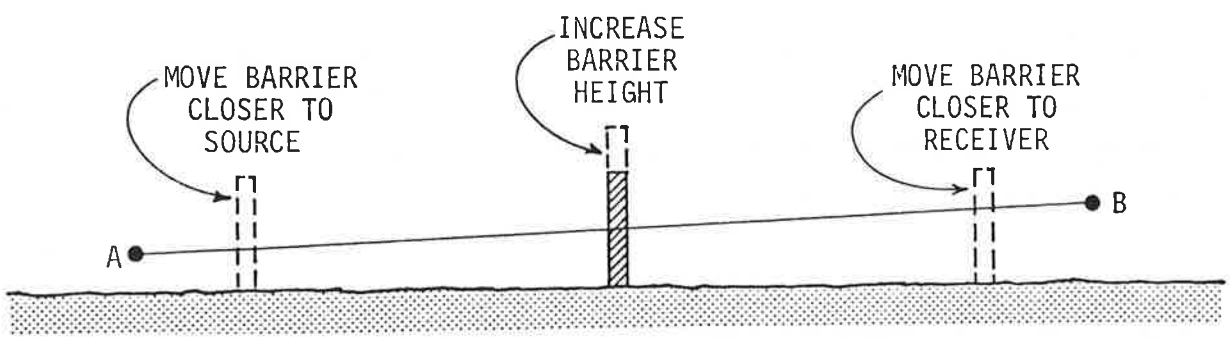


FIGURE 3.19 EXCESS ATTENUATION DUE TO BOTH GROUND EFFECTS AND AIR ABSORPTION. THE ATTENUATION VALUES ARE FOR POSITIONS DOWNWIND OF THE TRACK, WIND SPEEDS BETWEEN 3 AND 50 km/hr (2 AND 30 MPH), GROUND COVER LESS THAN 0.6 m (2 FT) HIGH, AND FOR COMBINATIONS OF TEMPERATURES (°F) AND RELATIVE HUMIDITIES (%) WHOSE PRODUCT IS GREATER THAN 4000°F%. MOST OTHER CONDITIONS WILL LEAD TO HIGHER ATTENUATIONS.

PATH LENGTH DIFFERENCE = AC + CB - AB



BASIC CONFIGURATION



METHODS OF INCREASING BARRIER EFFECTIVENESS

FIGURE 3.20 BARRIER CONFIGURATIONS

Although the same basic method has been shown to be relatively inaccurate for predicting attenuation of highway noise, Tubby (Ref. 3.17) presents data indicating that the predictions are quite good for transit trains.

A general rule of thumb is that any barrier that blocks the line-of-sight will provide 5 to 10 dB attenuation. The attenuation will be greater as the path length difference increases. The path length difference can be made greater by increasing the barrier height, or moving the barrier closer to either the source or the receiver (see Fig. 3.20). Experience has shown that 5 dBA attenuation is relatively easy to obtain with a barrier, 10 dBA can be obtained with a careful design minimizing acoustic leaks that may short circuit the barrier, and 15 dBA attenuation is the practical limit for barrier attenuation in outdoor situations. Although theory and scale model tests indicate that attenuations greater than 15 dBA are possible to achieve with a barrier, practical limitations, the presence of wind and temperature gradients, and other atmospheric effects reduce the real effectiveness of barriers. The result is that 10 to 15 dBA reduction is the maximum that can be expected, with 7 to 10 dBA the typical range for practical barriers.

3.4 ELEVATED STRUCTURE NOISE

Much of rapid transit's reputation as a loud, intrusive form of transportation is the fault of lightweight steel elevated structures. These structures were relatively economical to build, but because of their lightweight, relatively high flexibility, and low-damping characteristics, they cause very high noise levels, both at the wayside and inside the transit cars.

The control of noise from elevated trackways and railway bridges is a major problem faced by rail transit systems. This section presents only an introduction to the general characteristics of elevated structure noise. For a more detailed discussion of the control of elevated structure noise refer to Chapter 7.

None of the designs for elevated trackways on modern transit systems have much in common with the older, lightweight structures. Modern aerial structures and railway bridges are constructed with concrete decks. The rails are attached to the deck with resilient fasteners, or sometimes with ballast-and-ties in a trough on the concrete deck. The concrete deck is supported either by a concrete box girder or a steel girder. The major noise reduction is the result of the high mass and relatively high damping of the concrete deck. Although there are circumstances in which the vibration of the steel girder has caused low-frequency noise radiation problems, relatively straightforward techniques, such as constrained layer damping, have been devised to solve this problem. With appropriate design (e.g., the use of resilient fasteners, welded rail, and, sometimes, sound barrier walls on a concrete aerial structure), it is possible to locate aerial structures in noise-sensitive residential areas without exceeding reasonable noise levels or changing the character of the neighborhood.

The difference in wayside sound levels between the modern concrete aerial structures and the lightweight steel elevated structures is illustrated in Figure 3.21. This figure shows average octave band spectra 15 m (50 ft) from BART concrete aerial structures and from CTA steel elevated structures. Notice that even though the BART trains are traveling 45 km/hr faster than the CTA trains, the noise levels are 10 dBA lower. At locations where the BART structures are equipped with sound barrier walls the noise levels are about 6 dBA lower than those without the barriers. Other more effective barrier designs, such as at MARTA, give 10 dBA reduction. The difference in sound level as a function of speed is illustrated in Figure 3.22.

In most situations the wayside noise levels produced by modern rapid transit systems will be 3 to 4 dBA higher on aerial structures than on sections of at-grade ballast-and-tie track.

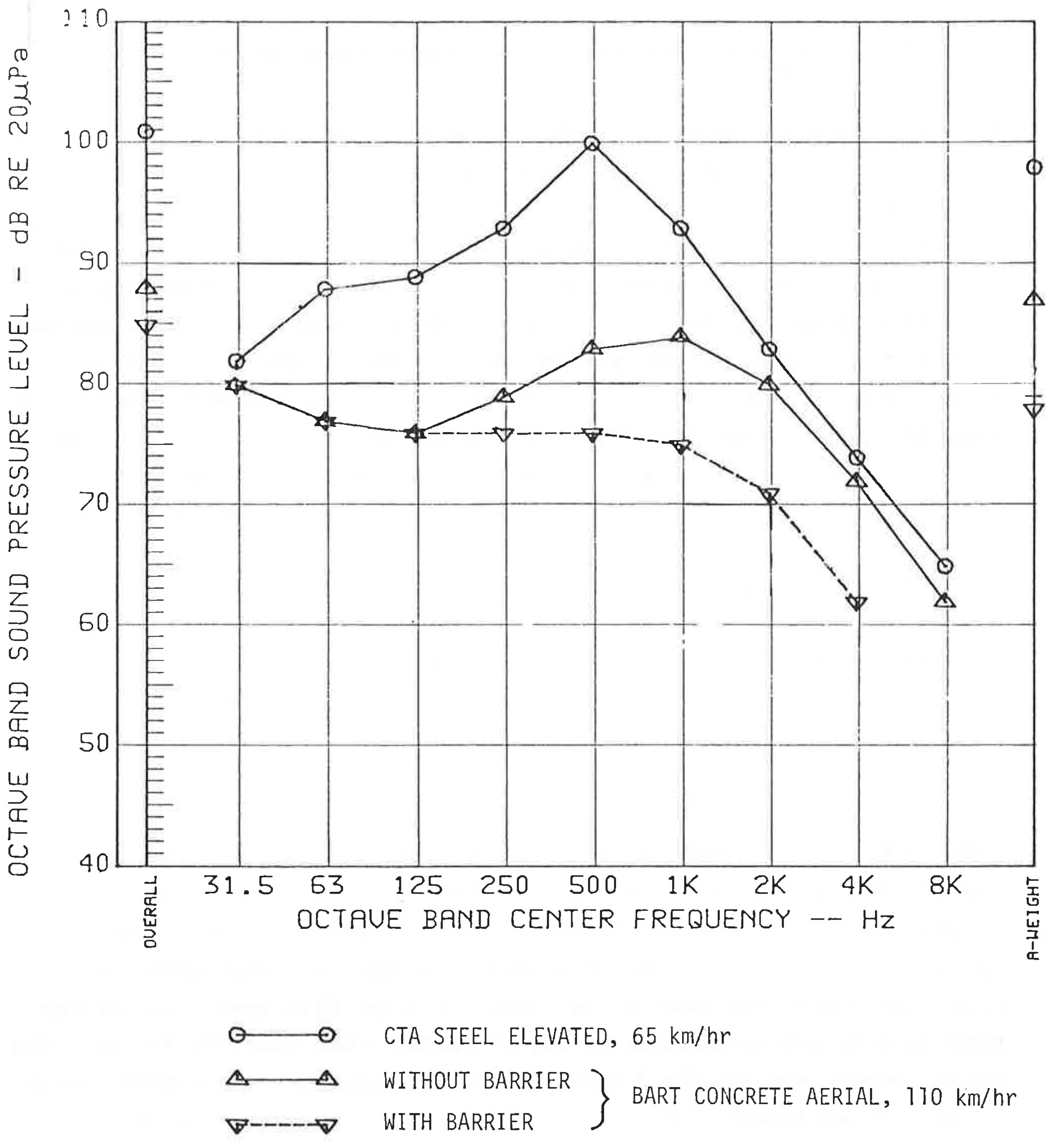


FIGURE 3.21 WAYSIDE NOISE LEVELS AT 15 m FOR 2-CAR TRAINS ON AERIAL STRUCTURES

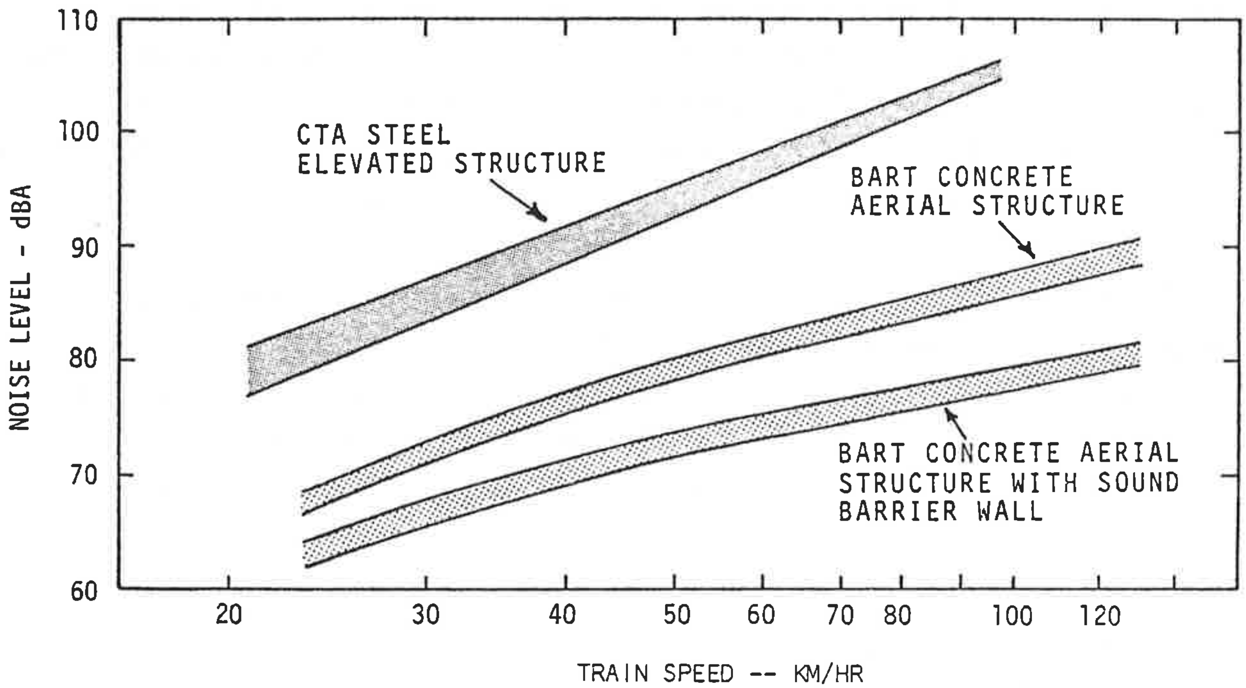


FIGURE 3.22 WAYSIDE NOISE LEVELS AT 15 m AS A FUNCTION OF SPEED FOR 2-CAR TRAINS ON AERIAL STRUCTURES

Structure-radiated noise is virtually never significant at frequencies above about 300 Hz. However, there is a possibility of low-frequency noise radiation, either from the plates of the steel girder or, under very specific conditions, from the concrete deck. Figure 3.23 illustrates the influence this low-frequency radiation can have on the octave band spectra of the wayside noise. At low frequencies, the level is as much as 15 dB higher for the steel girder structure, but, because the A-weighted level is dominated by the mid-frequency components, the A-weighted level is virtually identical for the two structures.

Although the low-frequency radiation does not influence the A-weighted level or the loudness of the noise outdoors, it does change the character of the noise, most noticeably after transmission to the inside of wood frame houses near steel girder structures. The walls of typical houses effectively reduce the mid- and high-frequency sounds, but are not very effective at reducing the low-frequency components. The result is that the train passby noise has the character of a low rumble, which many people find particularly annoying.

3.5 GROUNDBORNE NOISE AND VIBRATION

Problems due to groundborne noise and vibration are most common near subway sections of rapid transit lines, however, occasional problems occur along at-grade or aerial structure track sections. Groundborne noise and vibration are the focus of a considerable amount of research, both here and abroad. One of the problems with evaluating groundborne noise and vibration is that it depends on a number of factors, and this dependency is at best only partially understood. Because of the importance of the problem and the lack of consensus on prediction and control methods, groundborne noise and vibration are discussed in detail in Chapter 8. The chapter also includes two simple prediction procedures that are used for evaluating the impact of new transit facilities.

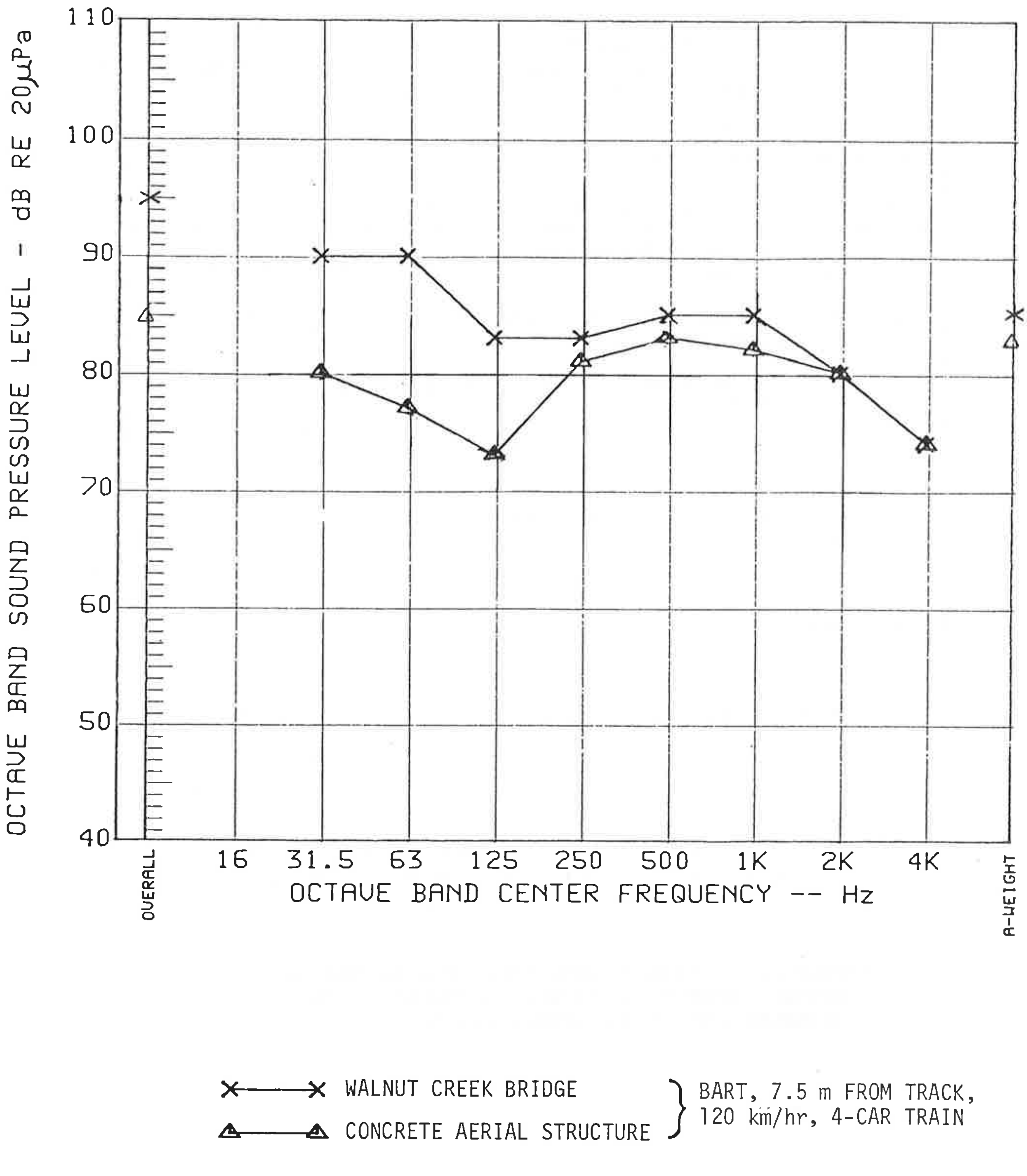


FIGURE 3.23 NOISE LEVEL SPECTRA ADJACENT TO CONCRETE DECK AERIAL STRUCTURES

TABLE 3-1 APPROXIMATE NOISE IMPACT CORRIDORS FOR VARIOUS TRANSIT WAY STRUCTURES - SEPARATION DISTANCE FROM TRACK CENTERLINE TO RESIDENTIAL STRUCTURE

Type of Way Structure	Distance from Track Centerline*
1. Steel Elevated Structure, Lightweight	600 m (or greater)
2. Aerial Structure - Concrete or Composite Steel/Concrete	200 m
3. At-Grade Ballast-and Tie or on Embankment	130 m
4. Aerial Structure with Sound Barrier	60 m
5. Ballast-and-Tie with Sound Barrier	30 to 35 m**
6. Landscaped Cut	30 to 35 m**
7. Retained Cut	25 to 35 m**
8. Subway	25 to 35 m**
9. Subway with Floating Slab	7 to 10 m

*The distance given for subways represents the diagonal distance from the subway structure to the building foundation.

**Representative - geologic conditions can cause large variations in distance for groundborne noise propagation.

Experience with groundborne vibration from rapid transit systems has shown that it is rarely a problem at frequencies above 200 Hz or below 10 Hz. At frequencies above about 30 Hz it is rare for the vibration to be perceptible as motion. Instead there is an audible low-frequency rumble as the trains pass by. Below 30 Hz the human ear is very insensitive to noise and hence, in the 10 to 30 Hz range, vibration perceptible as motion is the main problem. In addition to the perceptible vibration and the radiation of a low-frequency rumbling noise from vibrating walls, floors or ceilings, secondary noise radiation from rattling windows or objects on shelves can occur.

Figure 3.24 presents 1/3 octave band spectra of groundborne vibration from various transit systems. The examples selected for this chart represent similar train speeds and similar tunnel constructions. Significant factors to notice in Figure 3.24 are that the vibration spectra have a relatively narrow peak and that the peak frequency varies between transit systems. The peak frequency appears to depend very heavily on the local geology. This peak frequency is very important. Not only must the frequency be known before an optimum control procedure can be developed, but the frequency determines whether, and to what degree the problem occurs as groundborne noise or as groundborne vibration. Other secondary peaks can be caused by resonances in the track support system or the truck suspension system. As discussed in Chapter 8, these peaks can be a major source of community intrusion.

3.6 STATION NOISE

The noise level at a station platform during a train entrance or exit can be a deafening roar, if the station lacks sufficient noise control treatment. With acoustic treatment, the maximum noise level in a highly reverberant station, which can reach 100 to 105 dBA, can be reduced to 80 to 85 dBA. When the noise of trains passing through stations is reduced to 80 dBA or lower, the

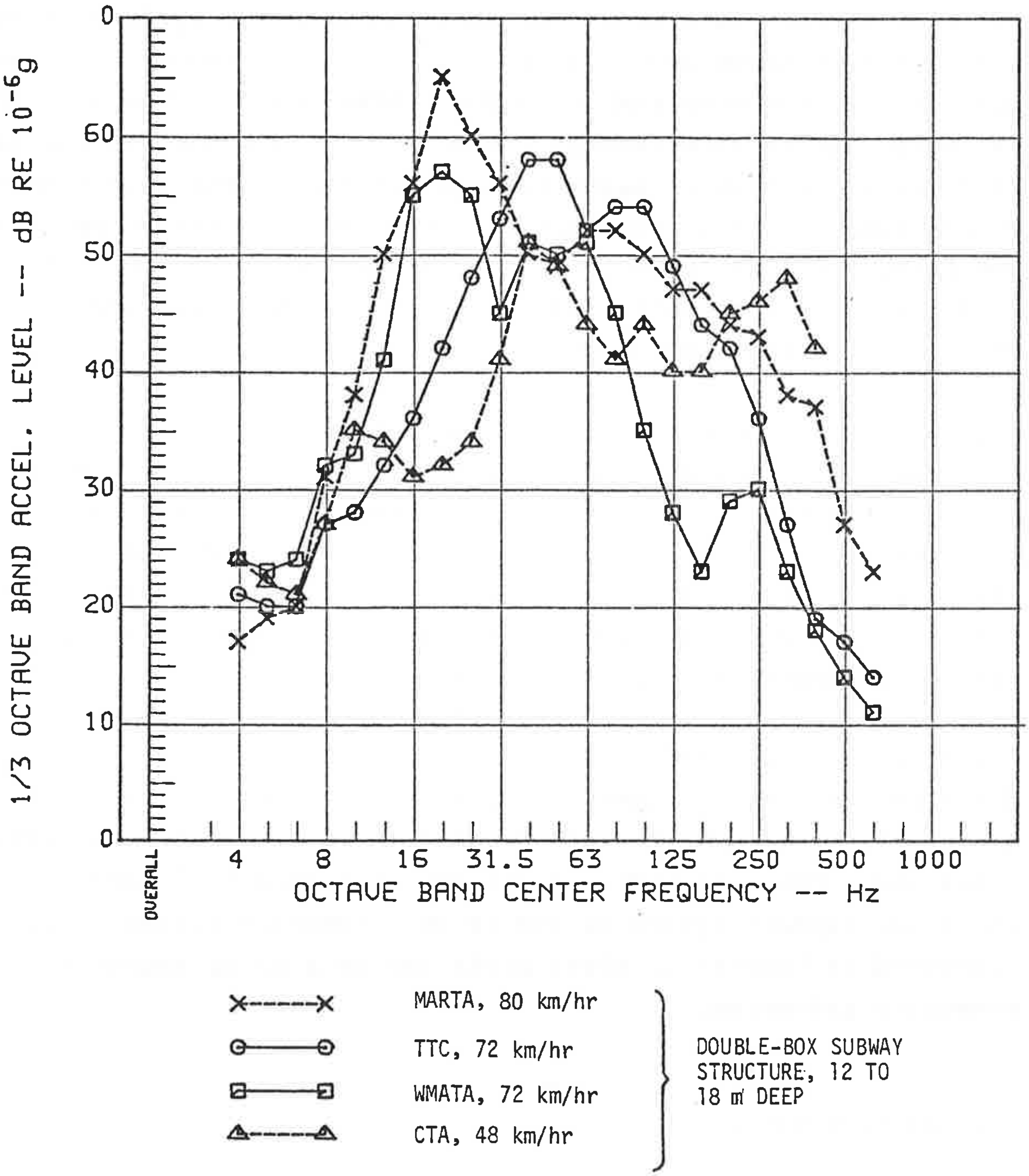


FIGURE 3.24 GROUND SURFACE VIBRATION LEVELS FOR SIMILAR MEASUREMENT CONDITIONS

noise level is low enough that on rare occasions patrons have complained that there is not sufficient warning when a train enters the station.

Table 3-2 presents some data on subway station platform noise levels at several transit systems. These data clearly indicate the effectiveness of acoustical treatment of rail transit stations. Another comparison is given in Figure 3.25. This figure indicates the range of noise levels, in terms of octave band sound pressure level, for selected TTC subway stations with and without acoustical treatment on the ceiling. Again the effectiveness of the acoustical treatment is evident.

TABLE 3-2 SUBWAY STATION PLATFORM NOISE LEVELS

	Passby Speed and Conditions	Measured Noise Levels
BART	0 mph 20 40 Entering & Leaving	63-64 dBA 77 87 75-80
WMATA Metro	0 mph 20 40	65-67 dBA 78-82 87-89
TTC	Entering & Leaving: - No acoustical treatment - With acoustical treatment	 85-91 dBA 70-80
PATCO Lindenwold	Entering & Leaving: - No acoustical treatment	 79-89 dBA
Paris Metro New Steel Wheel	20 mph 40 Entering & Leaving	75-82 dBA 89-91 80-86
Paris Metro Rubber Tire	20 mph 40 Entering & Leaving	73 dBA 86 80-85
Paris RER	Entering & Leaving	70-80 dBA
CTA	Tunnel Stations with Concrete Trackbed Subway Station with Ballast-&-Tie Track	103-110 dBA 90-95 dBA

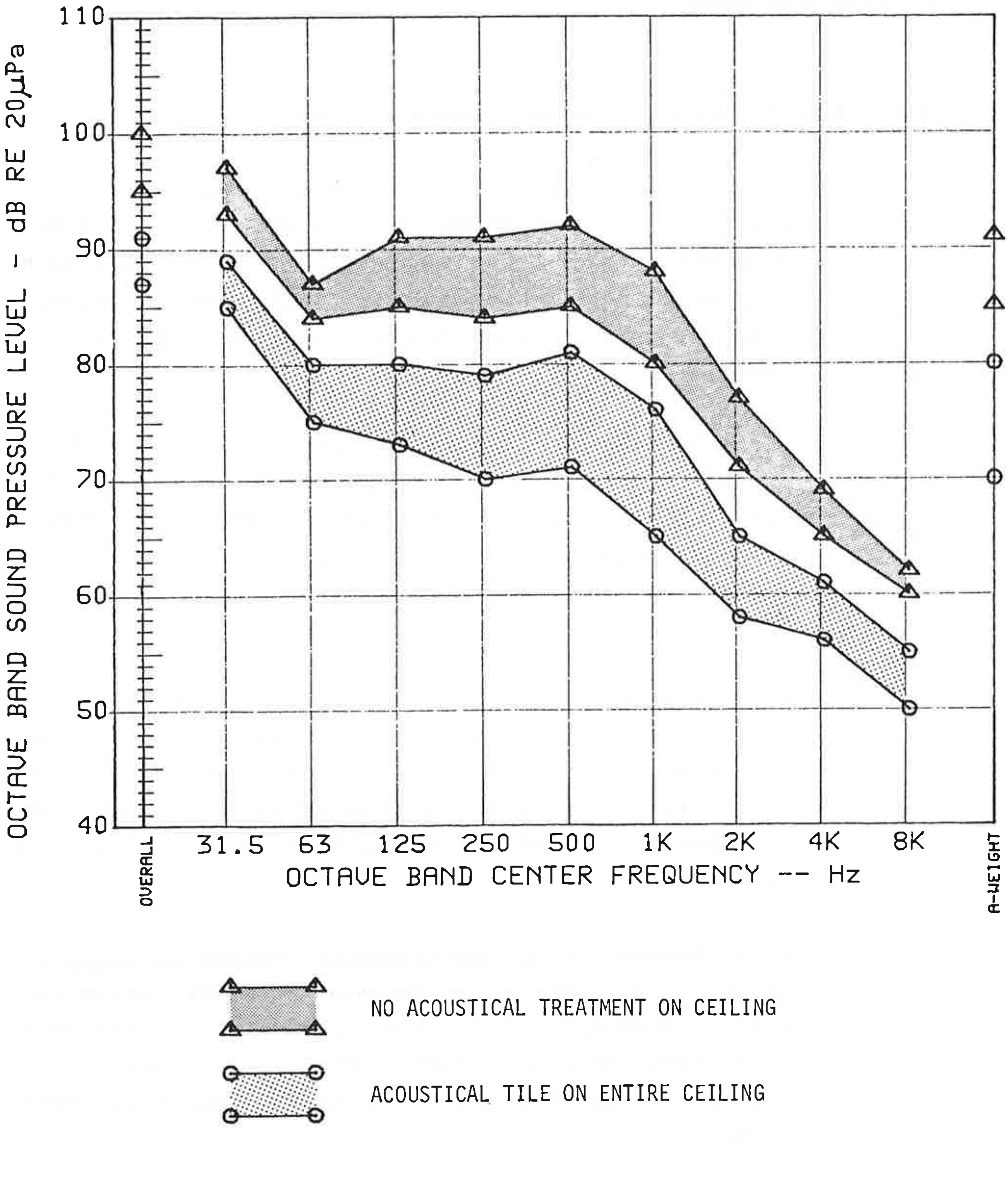


FIGURE 3.25 RANGE OF PLATFORM NOISE LEVELS DUE TO TRAIN OPERATIONS IN THE TTC TUNNEL STATIONS

CHAPTER 3 REFERENCES

3.1 Reports from the Noise Assessment Program

a. L. G. Kurzweil, R. Lotz, E. G. Apgar, "Noise Assessment and Abatement in Rapid Systems--Report on MBTA Pilot Study," Final Report, UMTA-MA-06-0025-74-8, NTIS-PB-238-113, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, September 1974).

b. R. H. Spencer, E. G. Hinterkeuser, "Noise Assessment of the Port Authority Transit Corporation Lindenwold Rail Transit Line," Interim Report, UMTA MA-06-0025-78-9, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., October 1978).

c. R. H. Spencer, E. G. Hinterkeuser, "Noise Assessment of the Southeastern Pennsylvania Transit Authority Heavy Rail Transit System," Interim Report, UMTA-MA-06-0025-78-11, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., October 1978).

d. R. H. Spencer, E. G. Hinterkeuser, "Noise Assessment of the Greater Cleveland Regional Authority Heavy Rail Transit System," Interim Report, UMTA-MA-06-0025-78-12, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., October 1978).

e. S. L. Wolfe, P. Y. N. Lee, H. J. Saurenman, "Noise Assessment of the Bay Area Rapid Transit System," Interim Scientific Report, UMTA-MA-06-0025-78-10, (U.S. Department of Transportation, Urban Mass Transportation

Administration, Washington, D.C., March 1976).

f. W. R. McShane, S. Slutsky, M. F. Huss, "Noise Assessment of the New York City Rail Rapid Transit System," Interim Report, UMTA-MA-06-0025-79-7, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., January 1979).

g. M. L. Silver, R. C. Bachus, R. Priemer, " Noise Assessment of the Chicago Transit Authority Rail Rapid Transit System," Final Report, UMTA-MA-06-0025-79-8, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., January 1979).

h. G. Chisholm, H. Bogan, M. Dinning, M. Primeggia, " National Assessment of Urban Rail Noise," Final Report, UMTA-MA-06-0099-79-2, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., March 1979).

These reports were all produced as part of an UMTA-sponsored project to assess the noise of U.S. urban rail transit systems. Sound level measurements were taken inside the transit car, in stations, and in the communities adjacent to the rail right-of-ways. The reports are a very good source for information on transit system noise levels. Because all of the data was taken in a uniform manner, the information can be used to compare the noise levels found at the various U.S. rail transit systems. The national assessment (Reference 3.1h) uses the data from the other seven reports to develop an overview of the urban rail noise problem and its distribution among the U.S. transit systems.

3.2 R. Lotz, "Railroad and Rail Transit Noise Sources," Journal of Sound and Vibration, 51(3), (Academic Press Inc., London, England, April 8, 1977), pp. 319-336.

The primary purpose of this paper is to review data on locomotive and railcar noise emission and to present the data in a format such that it could be used in making design decisions. Because of the relative wide spread of the available data, it is only suitable for preliminary screening of noise control options or assessing environmental impact; there is too much uncertainty about the effectiveness of many design alternatives to allow using the presented data in making decisions involving capital expenditures.

3.3 G. P. Wilson, "Noise Levels from Operations of CTA Rail Transit Trains," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Chicago Transit Authority, May 1977).

This report is on an extensive study of the noise generated by CTA rail transit vehicles operating on the various types of CTA way structures. Measurements include car interior, wayside, and station platform noise from transit operations on elevated structures, on ballast-and-tie track, and in subways. The measurement program included evaluation of the noise emission characteristics of several types of transit cars and trucks, including cars with standard trucks and cars with overhauled, modified trucks. The modified trucks were tested with standard journal sleeves (creating the effect of primary springing on a truck which does not incorporate primary springs). The testing showed substantial noise reductions with the overhauled and modified trucks relative to standard trucks and even further reductions with the soft journal sleeves. Significant noise reductions were also observed with welded rail compared to jointed rail and in subways with absorptive material on the tunnel surfaces compared to subways with untreated concrete surfaces.

3.4 J. E. Manning, R. G. Cann, J. J. Fredberg, "Prediction and Control of Rail Transit Noise and Vibration -- A State of the Art Assessment," Interim Report, July-December 1973, UMTA-MA-06-0026-74-5, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, April 1974).

A critical review of the current technology (as of 1974) for the prediction and control of urban rail transit noise with primary emphasis on the parameters affecting propagation paths. This is one of the reports prepared as part of the Urban Rail Supporting Technology Program. The report does not concern the development of new technology, instead it summarizes the comprehensive literature search that was performed and presents a critical review of the various prediction and control methods that have been presented in the literature. Although some caution should be exercised when using this document because of advances in the state-of-the-art since the document was prepared, it is a very useful reference for transit system noise and vibration control and can be used to supplement the material of this handbook.

3.5 R. Lotz, L. G. Kurzweil, "Rail Transportation Noise," Chapter 33 in Handbook of Noise Control, Second Edition, Cyril M. Harris, editor, (McGraw-Hill Book Company, New York, 1979).

A concise summary of the noise generated by rail vehicles, including locomotives and railcars (both self-propelled and locomotive pulled). This is a good source for a general introduction to the topic of rail noise and vibration, although it is not sufficiently detailed to use for design purposes. It is essentially a shortened version of Reference 3.15.

3.6 L. G. Kurzweil, W. N. Cobb, R. P. Kendig, "Propagation of Noise from Rail Lines," Internal Report, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, 1979 approximately).

An internal TSC paper on the prediction of wayside train noise, including the effects of geometric spreading, air absorption, ground effects, and barriers.

3.7a. S. Peters, "Prediction of Rail-Wheel Noise from High Speed Trains," Acoustica, 28(6), (S. Hirzel Verlag, Stuttgart, Germany, June 1973), pp. 318-321.

3.7b. S. Peters, "Prediction of Railway Noise Contours," Journal of Sound and Vibration, 32(1), (S. Hirzel Verlag, Stuttgart, Germany, 1974), pp. 87-99.

A method is presented for predicting the peak noise levels from wheel/rail noise and the rise and decay of the noise (noise profile) of a passing train. The dipole line source model was found to give very good agreement with measurements.

3.8 E. J. Rathe, "Railway Noise Propagation," Journal of Sound and Vibration, 51(3), (S. Hirzel Verlag, Stuttgart, Germany, April 8, 1977), pp. 371-388.

The propagation of railway noise is characterized by the typical arrangement of noise sources on a moving train. The rates of sound attenuation with distance are determined for peak levels and for the equivalent steady sound levels in the cases of omnidirectional and directional sources. The means of noise control within the path of sound propagation are discussed.

3.9a. G. P. Wilson, "Rail Transit System Noise Control," Proceedings of Noise-Con 77, (NASA Langley Research Center, Hampton, Virginia, October 17-19, 1977), pp. 247-256.

3.9b. G. P. Wilson, "Progress in Rail Transit System Noise Control," to be published in Noise Control Engineering.

These papers summarize the progress in the control of rail transit noise and vibration. Discussed are specifications for transit cars, the noise levels inside modern and older transit cars, station acoustical treatment, elevated structure noise, and groundborne noise and vibration. Included is a discussion of the design and construction of lightweight floating slabs.

3.10 P. J. Remmington, M. J. Rudd, I. L. Ver, "Wheel/Rail Noise and Vibration," Final Report, UMTA-MA-06-0025-75-10 (Volume 1), and UMTA-MA-06-0025-75-11 (Volume 2), NTIS PB-244-514 and PB-244-515, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., May 1975).

The final report of an UMTA-sponsored program to develop a basic understanding of wheel/rail noise. The report contains analytical formulas for predicting wheel/rail noise and a considerable amount of background material on the mechanism of wheel/rail noise generation.

3.11 H. Meurers, "Gerauschbelastung durch moderne Statbahnen," ("Noise Propagation in Modern Street Railways"), TU: TECHNISCHE UBERWACHUNG; SICHERHEIT ZUVERLASSIGKEIT IN VIRTSCHAFT BETRIEB VERKEHR, 12(1), (January 1971).

Primarily reporting on measurements at the Rotterdam Metro with trains on concrete aerial structures with sound barrier walls. The report includes some averaged 1/3 octave band data.

3.12 "Rapid Transit Railroad Noise," A Report to the City Council Pursuant to Section 5.07 of the New York City Noise Control Code, (New York City Bureau of Noise Abatement, New York, NY, October 1973).

This is a report on subway noise in New York City that was prepared by the New York City Bureau of Noise Abatement. As a part of the New York City Noise Control Code that was enacted in 1972, it was mandated that the Environmental Protection Administrator define allowable sound levels and acoustical performance of new and existing rapid transit railroads. This report presents a summary of the subway system noise survey that was performed, and outlines existing NYCTA noise problems and solutions.

The report contains a considerable body of information on the noise levels of the existing NYCTA transit facilities and approaches that can be taken to reduce the noise. Also included are estimates of the costs of the treatments and specific recommendations of which treatments should be implemented by the transit authority. Because of the implications relative to the large expenditures if specific standards were to be met, the report does not attempt to set acoustical performance standards for the existing subway system.

3.13 G. P. Wilson, S. L. Wolfe, "Bart Car Floor Vibrations," Final Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinkerhoff-Tudor-Bechtel, March 1974).

A series of measurements of the vertical vibration of the

floors of ten BART cars. The measurements were performed with the cars stationary and various auxiliary units operating.

3.14 P. J. Remington, L. E. Wittig, M. M. Myles, "Diagnosis and Control of Noise in Older Rapid Transit Cars," Presented at 96th Meeting of the Acoustical Society of America, (Honolulu, Hawaii, November 1978).

This is a report on a detailed experimental study of the noise sources of existing NYCTA transit cars. The goal of the study was to develop specific recommendations for noise control treatments for the older transit cars on the system. Of particular interest is the evaluation of the relative strengths of the various noise sources on the transit cars.

3.15 L. G. Kurzweil, R. Lotz, "Prediction and Control of Noise and Vibration in Rail Transit Systems," Final Report, UMTA-MA-06-0025-78-8, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., September 1978).

This report is a compilation of much of the available information on rail transit noise and vibration control. Although the emphasis is on urban rail transit, some information on intercity freight and passenger trains is also included. The report has been used as a background document for this handbook and some sections have been adapted into the handbook.

3.16a G. P. Wilson, "Aerial Structure Noise," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for De Leuw Cather & Company and the Washington Metropolitan Area Transit Authority, June 1972).

3.16b G. P. Wilson, "Aerial Structure Sound Barrier Walls," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for the Baltimore Metropolitan Transit Authority, January 1973).

3.16c G. P. Wilson, "BARTD Aerial Structure Sound Barrier Wall Noise Tests," Letter Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinkerhoff-Tudor-Bechtel and the Bay Area Rapid Transit Disrict, February 1973).

These three reports present a large quantity of information on the sound levels from train operation at-grade and on aerial structures. The information is largely based on testing performed at BART, including measurements of the effectiveness of the BART sound barrier walls.

3.17 J. A. Tubby, "Propagation of Railway Noise," M. Sc. Dissertation, (University of Southhampton, Department of Engineering of Applied Science, Southampton, England, 1974).

A report on an experimental study of the propagation of railway noise. A considerable amount of information on the noise level as a function of distance from the track is presented.

CHAPTER 4

4. MEASUREMENT OF NOISE AND VIBRATION

The focus of this chapter is the instrumentation and procedures that should be used to measure transit system noise and vibration. An effort has been made to arrange the contents in an ascending order starting with a general introduction to the measurement of transit noise and vibration, leading into more detailed discussions of instrumentation requirements and procedures for performing noise and vibration measurements. Section 4.1 presents a general introduction describing the reasons for measuring noise and vibration, the different types of noise and vibration, and measurement approaches. In Section 4.2, four levels of measurement and analysis systems are presented, starting with the basic sound level meter and finishing with a comment on computer-based systems, including real-time analyzers. The selection criteria for such instrumentation arrays is discussed in Section 4.3, including such topics as bandwidth and sensitivity requirements. Acquisition of noise and vibration data are discussed in Sections 4.4 and 4.5 respectively, including a description of measurement locations for typical transit system installations. Laboratory procedures are considered in Section 4.6, including a discussion of data presentation and techniques for avoiding electrical interference. Finally, in Section 4.7, various standards and pertinent codes are listed for ready reference. More detailed discussions of specific instrumentation components are presented in Appendix D.

Noise and vibration measurement is a very large topic, one that cannot be completely covered in this handbook. When learning about measurement techniques and procedures, it will be necessary to use other references; several general references are discussed in the annotated bibliography. References 4.1 through 4.4 are particularly valuable. In addition, a review of equipment manuals is extremely important before taking equipment into the field or before using it in the laboratory. The number of instances where improper use of equipment has led to erroneous results would be significantly reduced if more time were taken to read the user manuals.

4.1 BACKGROUND

4.1.1 Situations Requiring Noise Measurements

A variety of situations exist for which measurements may be needed. These include:

Environmental Assessment: For virtually all new transit system construction that may have an impact on the community it is necessary to prepare an EIS (Environmental Impact Statement) that includes an assessment of the noise and vibration impact. Measurements of existing ambient noise and vibration are generally required. The measurements are valuable in determining the change in noise and vibration that the community will experience from the proposed transit facility and for assistance in making design decisions concerning track alignment, required noise and vibration control, condemnation, etc. Should complaints or legal action regarding noise or vibration arise after the new facilities are operational, the documentation of pre-existing noise and vibration can be extremely valuable. Often environmental measurements are also performed during construction to monitor noise and vibration and to ensure that standards for acceptable levels are not exceeded.

Diagnosis: During the construction and operational stages of any rail rapid transit system, various noise and vibration problems will develop for which effective controls must be designed and implemented. These problems may be the result of community complaints, violation of occupational noise standards, or even failure of mechanical or electrical components on transit vehicles resulting from fatigue induced by excessive vibration.

Diagnosis of the problem and design of the necessary control features can sometimes be accomplished with simple measurements such as the A-weighted noise level. However, detailed spectral analysis (octave, 1/3 octave, or narrowband) is often needed for full evaluation of the problem. For analysis of vehicle component fatigue, it may be necessary to measure the vibration amplitude probability density for comparison with fatigue life data.

Evaluation of Abatement Provisions: Once a provision for control of noise or vibration has been developed and a prototype installation completed, measurements should be performed to evaluate the effectiveness of the design. If the measurements can be performed at an early stage in the construction process, necessary design modifications resulting from these tests may be incorporate. A great many measurements performed at transit systems are for the evaluation of noise and vibration control provisions.

Occupational Noise Exposure: Virtually all employees are now covered by state or federal limits on exposure to noise. With simple measurements using a sound level meter, it is often possible to determine if the exposure limits are being exceeded. If the readings indicate that there is a possibility of excessive exposure, the noise "dose" of the affected employees must be determined. Additional measurements will be required to determine effective noise control solutions. When the noise exposure is near or above the acceptable limit, regular audiometric tests may be required to ensure that no employees experience hearing loss.

Evaluation of Compliance with Specifications: Noise and vibration specifications are commonly included in purchase documents for new transit equipment, especially transit cars. The acceptance of the equipment requires verification of compliance, generally under close observation of the contractor and transit system representatives. The measurements required for evaluation of specification compliance are usually relatively simple, as opposed to those which would be performed for diagnosis of vibration problems. Often the specifications are written in a form that allows the determination of compliance without sophisticated measurements, thereby reducing the chance for error and ambiguity, as well as allowing the manufacturer to evaluate prototype designs without performing sophisticated measurements. Prototype equipment should be tested at the earliest possible date to allow design changes and provide reasonable assurance that equipment specifications will be met.

4.1.2 Need for Reliable Data

Measurements of rail transit noise and vibration are performed for many different reasons ranging from diagnosis of a specific noise problem to general research on the nature of transit noise and vibration. It is not uncommon that measurements initially performed to satisfy some limited goal later become useful for far more important reasons. For example, measurements to evaluate the validity of a complaint may be used to help decide between different types of way structures for a new section of transit line. To allow measurement data to be used for more than the specific purpose at hand, it is very important that the test conditions be well documented, that the measurement results be accurate and reliable, and wherever practical, that standard test locations be used to allow direct comparisons between different transit facilities.

4.1.3 General Measurement Approaches

A number of different techniques exist that can be used to measure and analyze noise and vibration data. Indeed, some minor confusion and inconsistency is often encountered when comparing data presented by different researchers. Accordingly, general guidelines are presented below for various categories of noise and vibration that may be encountered at a transit system. In addition, ANSI Standard S1.13-1971 "Methods for the Measurement of Sound Pressure Levels," (Ref. 4.5), should be reviewed and made part of the engineer's library. Many of the considerations given in the above reference may be applied to vibration measurements.

For the purposes of this discussion, the various types of transit system noise and vibration can be divided into four categories: steady-state, non-steady-state, single-event, and impulsive.

Steady-State Noise and Vibration: Steady-state noise and vibration may be periodic and/or random in nature, but the rms amplitude and frequency content are independent of time. The noise from fixed facilities such as the ancillary equipment for a transit station comes the closest to steady-state noise. Noise from a ventilation fan will be essentially time invariant once normal operating speed is reached. Other examples are substation noise and some types of shop noise. Because it does not vary with time, this is the easiest type of noise to measure and analyze. One has the luxury of moving from point to point with a single measurement system and, in many cases, of reading the required data directly from a sound level meter. Reasonably accurate readings can be obtained by taking sufficiently long data samples or by utilizing a long enough integration time, i.e., the "slow meter response."

In many situations, as with noise from a multi-unit fan shaft, the noise level may fluctuate in time in a periodic fashion between a maximum and minimum value as shown in Figure 4.1. This type of noise is still considered as steady-state. For this type, the average level and range should be measured. Ideally, however, the rms level over several level fluctuations should be measured, usually requiring an integrating sound level meter or other more sophisticated instrument.

Non-Steady-State Noise and Vibration: Non-steady-state noise and vibration may be either periodic, random, or transient in nature, but its frequency content and/or amplitude changes significantly over time. Usually, the level variation of non-steady-state noise

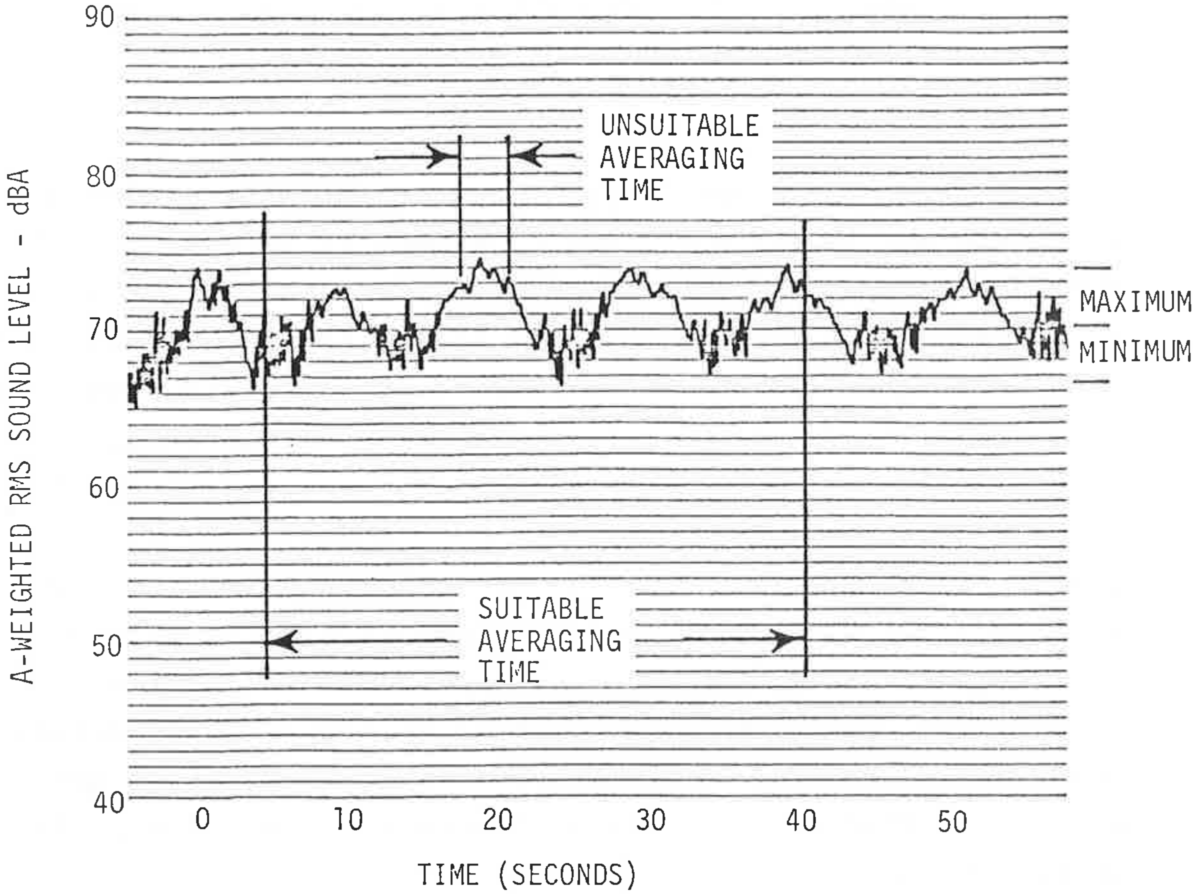


FIGURE 4.1 FLUCTUATING FAN NOISE

is greater than 5 or 10 dB. In general, all noise and vibration may be viewed as non-steady-state if the time period is long enough.

Examples of non-steady-state noise and vibration include community noise, shop noise, groundborne vibration, wheel squeal, and blast noise and vibration. Generally, the approach to characterization of non-steady-state noise and vibration is to measure its statistical properties. Thus, for community noise, the A-weighted rms sound level exceeded n% of the time -- L_n -- is often measured.

The most common statistical levels are L_{90}, L_{50} and L_{10}. Often L_{90} is referred to as the typical background level. For values of n greater than 90 or less than 10, L_n may be affected to some extent by the detector averaging time, e.g., "fast" or "slow" meter response. More detailed analysis sometimes includes a plot of the percentage exceeded, n, as a function of A-weighted sound level, as indicated in Figure 4.2 for general community noise and for community noise along with wayside rapid transit train passby noise. Note that only the levels greater than L_{10} are significantly affected by train passby noise. Generally, L_{50} is unaffected.

Other, equally common, community noise descriptors are the Equivalent Sound Level, L_{eq}, the Day-Night Sound Level, L_{dn}, and the Community Noise Exposure Level, CNEL (For definitions see Appendix A or Glossary). Of these measures, L_{eq} and L_{dn} are most often used, with CNEL giving essentially the same result as L_{dn}. Note that L_{eq} is equivalent to the A-weighted rms sound level computed over the entire period of interest.

Instruments are available for unattended measurement of community noise over periods as long as a week. An alternative, however, is to perform "spot checks" of the community noise environment at various times of the day and to estimate daily figures from these

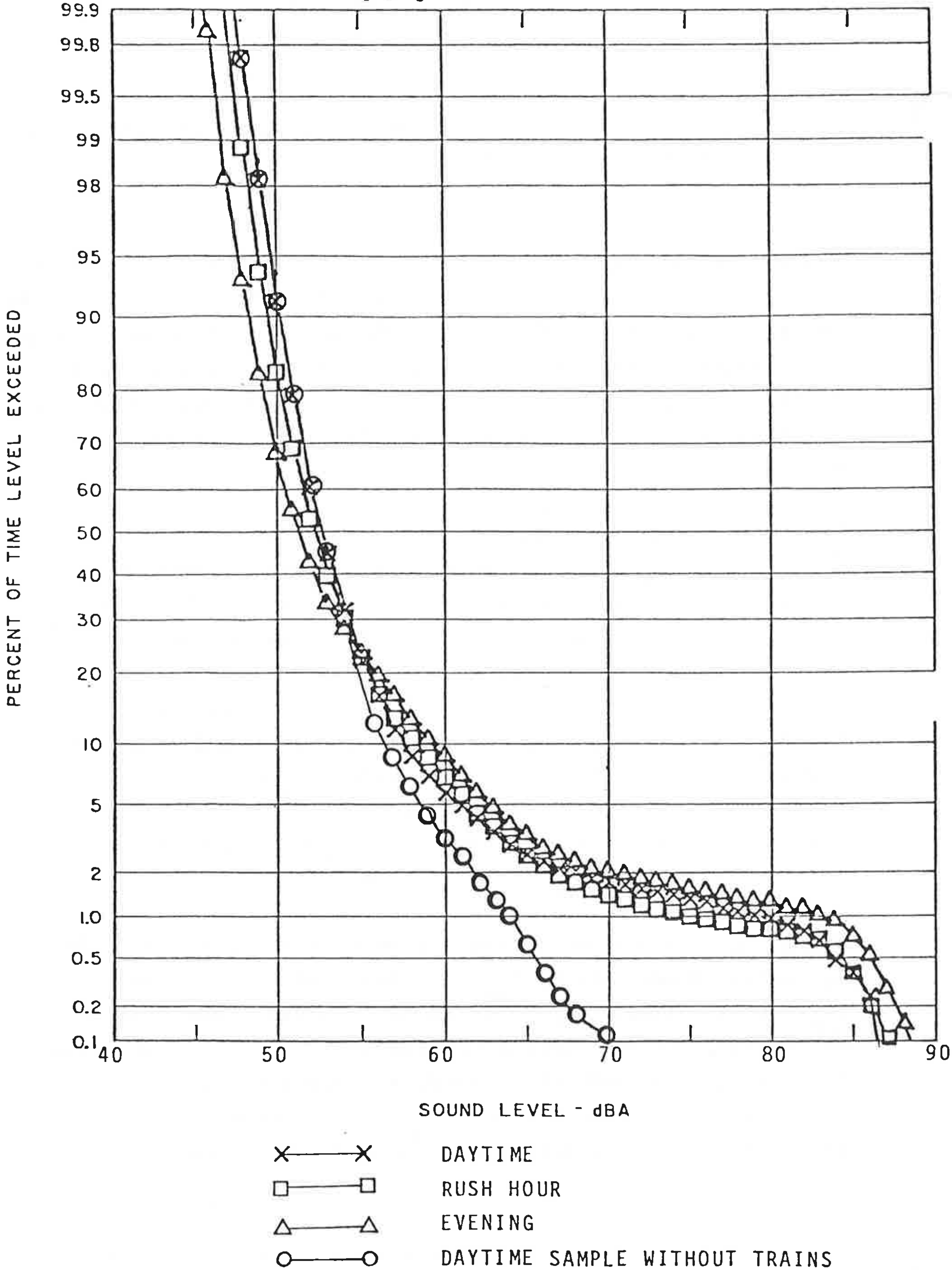


FIGURE 4.2 STATISTICAL ANALYSIS OF COMMUNITY NOISE LEVELS

spot checks. The typical sampling period of a spot check is 10 to 20 minutes. For measurement of L_{eq}, an integrating sound level meter is most appropriate. Often, the same equipment used for unattended 24-hour surveys can also be used for the 10-minute surveys and to provide estimates of L_n as well as L_{eq}.

The community noise descriptors L_n and L_{eq} are sufficiently general in nature that they may be extended to the analysis of groundborne vibration or even 1/3 octave or octave band vibration and noise (without A-weighting). Because of the large amount of data produced during the analysis, octave or 1/3 octave statistical level analysis is only practical if computer-based real-time analysis systems are used.

Note that if the time period is sufficiently small, it is often possible to analyze non-steady-state data as if it is steady-state. The data is considered "quasi-steady-state." A good example is train passbys. Although the noise is a non-steady-state, it is common practice to analyze the noise while the train is passing by the measurement location, assuming steady-state conditions.

Other types of non-steady noise and vibration are single-event noise and vibration, and impulse noise and vibration. These are discussed below under separate headings because of their unique characteristics and specialized measurement approaches.

Single-Event Noise and Vibration: Single-event noise and vibration may be either periodic and/or random but with a temporarily sustained time varying amplitude and frequency content. When analyzing single-event noise, the noise contribution attributable to the specific event is of greatest interest, irrespective of background noise. Since the primary source of transit noise is train passbys, most transit noise falls within this category.

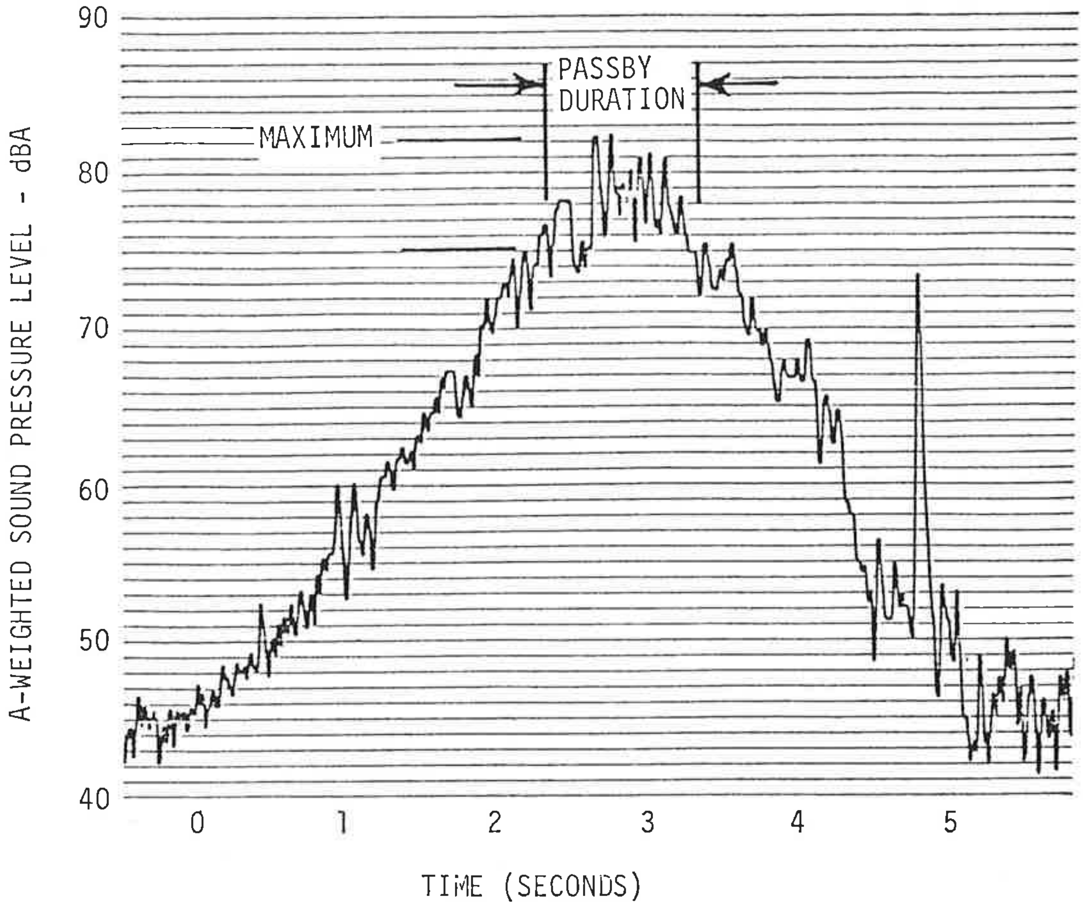


FIGURE 4.3 SINGLE-EVENT NOISE RESULTING FROM TRANSIT TRAIN PASSBY

Figure 4.3 illustrates an example of single-event noise or vibration resulting from passage of a transit train. For this example, the level signature never achieves a steady-state level, but varies over time. One is immediately faced with the problem of how to measure the "level" of this signature. Does one measure the maximum or peak level, or does one measure the average level over a given time period, and if so, what should the time period be? It is recommended that the level be characterized by the rms amplitude over the duration of the train passby only. In practice, the average reading of a precision sound level meter set for the "fast meter response" provides an adequate approximation for the rms noise or vibration level during the passby duration, provided that the level variation is less than about 3 dB.

If the "maximum level" as indicated in Figure 4.3 is measured, the data must be accompanied by the effective response time of the measuring instrument. In the case of a sound level meter, the "fast meter response" is normally used. If, however, the train or the event is long, taking several seconds, then a longer integration can be used, e.g., the "slow meter response," thus making the observation of average maximum level easier with less ambiguity.

In much of the literature, the term "maximum level" is loosely used to refer to the rms noise or vibration level computed over the passby duration, whereas the maximum level indicated in Figure 4.3 is often referred to as the peak level or peak rms level. This latter usage can lead to confusion since the term "peak level" should be used only to refer to the level of the instantaneous maximum magnitude achieved during an event. This confusion may be avoided if both the rms level and measurement duration are specified together, e.g., "75 dBA for the passby duration."

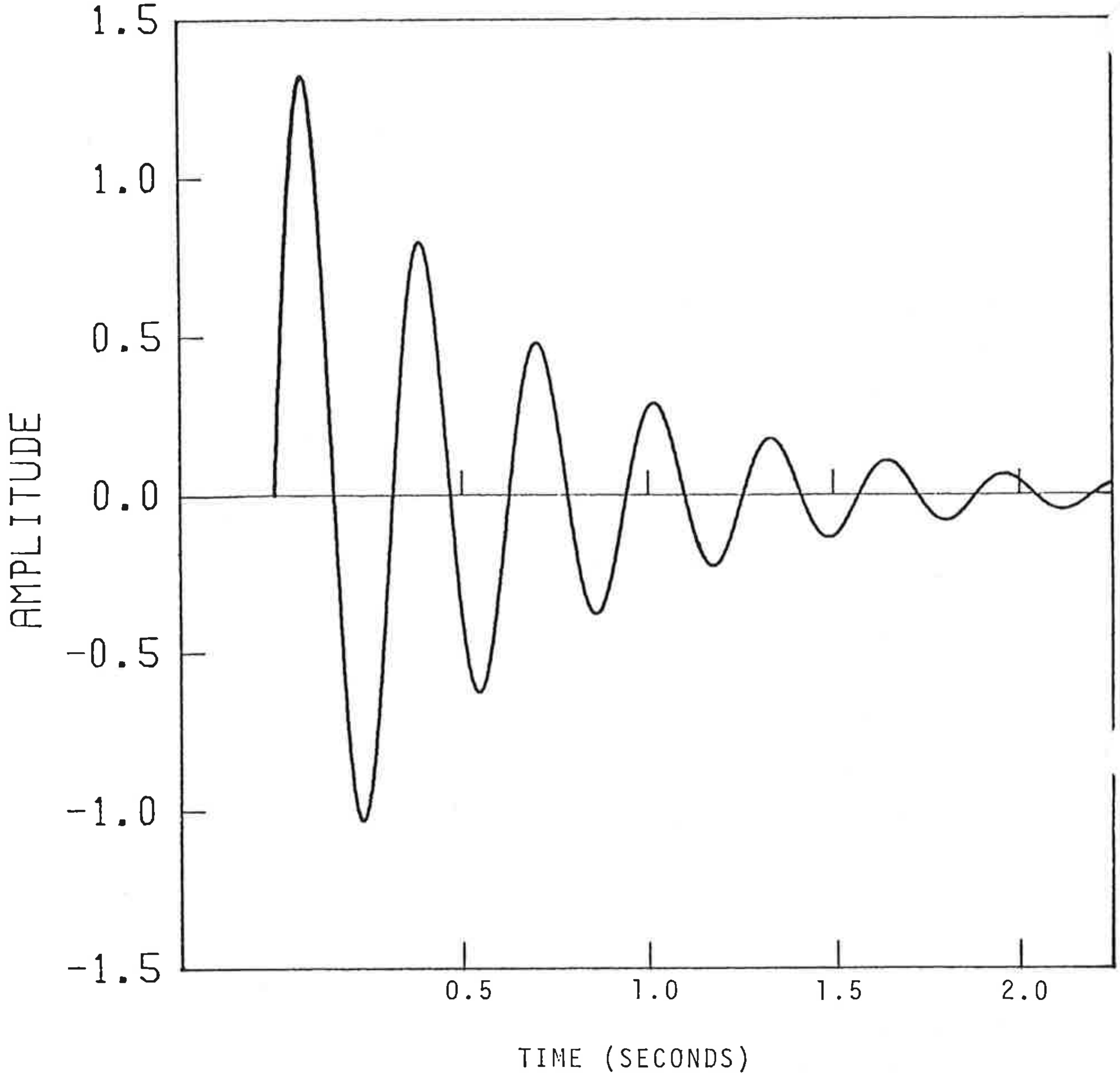


FIGURE 4.4 IMPULSIVE NOISE OR VIBRATION

Impulsive Noise and Vibration: Impulsive noise and vibration is of such short duration, usually only a fraction of a second, that it cannot be considered as steady-state data even for arbitrarily short time periods. Such a transient, in this case a decaying sine wave, is illustrated in Figure 4.4. Examples of such noise and vibration are those produced by blasting, impacts at rail joints, door actuators, etc. Note that repeated impulse noise, such as that produced by wheel flats, can be characterized as steady-state or quasi-steady-state provided that a sufficiently large number of impulses occur within a given time period.

The ideal procedure for measuring impulse noise and vibration, including spectral analysis, is to integrate the square of the amplitude over the entire duration of the impulse. Thus, the result is proportional to the total energy of the impulse or transient at the measurement point. There is no general restriction on use of weighting networks or filters for the measurement of transient noise and vibration. Thus, overall, A, B, or C-weighted or 1/3 octave band impulse pressure levels are all meaningful quantities. The most common example is the analysis of transient phenomena by a Fourier Transform or Spectrum Analyzer which first "captures" the transient and then determines its Fourier components.

In the field of occupational noise exposure, or for general noise measurements which are to be used for predicting effects on humans, impulse noise is generally measured with instruments having a special impulse detector. The measured result is an impulse noise level. Basically, the impulse noise level is the rms noise level determined for a standardized time duration of about 0.035 seconds. For continuous or steady-state noise, the impulse detector will give the same level indication as the standard "fast" or "slow" meter response settings of a sound level meter. Measurements of impulse noise on rapid transit systems, if needed, will probably require an impulse detector.

Impulse or transient noise and vibration is often measured with a peak detector or displayed on an oscilloscope. The peak magnitude of a transient may often be 10 to 30 dB higher than the level measured with a standard sound level meter. In the case of component vibration or shock measurements, the peak magnitude is often of major interest, especially with respect to component failure and fatigue. Because of the extreme shock and vibration environments experienced by transit system components, the ability to measure peak magnitude is of great value.

4.1.4 Specific Measurements

4.1.4.1 Overall or Weighted Levels -- An overall measurement is the measurement of the rms sound pressure level, or vibration acceleration or velocity level, over a specified bandwidth without weighting networks. That is, a flat frequency response over the measurement bandwidth is assumed. For noise measurements, the bandwidth should be from 10 Hz to 12 kHz, and for groundborne vibration, the bandwidth is from about 2.5 to 1000 Hz. For component vibration, the bandwidth should be from about 10 Hz to 10 kHz.

It is often desirable to determine a weighted level. Weighting is most commonly associated with the evaluation of sound levels. The A, B, C, and D-weighting network frequency response curves are defined in Appendix A. Basically, these networks adjust the frequency response of the measurement system to more closely correspond with the response of the human ear to audible noise. Unless otherwise noted, the A-weighting network is almost always used to measure sound levels at transit systems.

No standardization weightings for vibration have been established. A tentative weighting, which approximates the human response to vibration has been proposed by a Committee on Hearing, Bioacoustics and Biomechanics (CHABA) subcommittee (Ref. 2.13). For ground vibration the overall vibration velocity level will give essentially the same results as the weighted level. For ride-quality assessment, where very low-frequency vibration from vehicle motion is encountered, the overall vibration velocity is probably not adequate, in which case a weighting curve such as recommended by CHABA may be most appropriate.

4.1.4.2 Octave Band and 1/3 Octave Band Analysis -- Octave and 1/3 octave band filters are often used to determine the spectral content of noise or vibration. The filter characteristics are specified by ANSI Standards. Octave band and particularly 1/3 octave band analyses have proven to be very practical and useful for the identification and development of solutions to many common noise and vibration problems. Often, specifications for noise and vibration in equipment are presented in terms of octave band or 1/3 octave bands, so that octave band or 1/3 octave band analyses may be necessary to assess equipment noise levels or to comply with procurement specifications.

One-third octave band measurements provide greater detail than octave band measurements and are often sufficient to determine noise or vibration sources, or the effects of changes in various parameters. The effective noise bandwidths of 1/3 octave band filters are often comparable to the response bandwidths of damped, resonant mechanical systems subjected to random vibration forces. Usually, 1/3 octave band measurements are preferable to octave band measurements as the base measurements for transit system noise and vibration. If octave band data are needed, the 1/3 octave band levels may always be summed (logarithmically) to obtain octave band spectra.

4.1.4.3 Narrowband Analysis -- Where identification of noise and vibration sources is required, it is useful to correlate narrowband spectra obtained at various source and receiver locations. Two basic approaches are available. The first approach is to record a sample of noise or vibration data on magnetic tape. Then, form a tape loop and continuously reproduce the sample, while sweeping a narrowband analog-constant-percentage-bandwidth filter over the frequency range and plotting the rms signal output of the filter. A faster technique is real-time analysis, using a constant bandwidth spectrum analyzer. This device usually has about 400 to 1000 frequency bands, or "lines," the bandwidths of which are essentially equal to the frequency range of the analysis, divided by the number of lines. Significantly greater spectral resolution is obtained by this method, and a great deal of time is saved. The constant frequency interval between successive band frequencies is ideally suited to the analysis of complex, periodic noise or vibration with harmonically related frequencies, such as that produced by gear or transformer noise.

4.1.4.4 Fourier Analysis -- Fourier analysis is a more general form of spectral analysis. It can be used to determine the amplitude and relative phase of a waveform as a function of frequency. Fourier analyzers are essentially "hardwired" digital computers, which use the Fast Fourier Transform algorithm to calculate the real and imaginary parts of the Fourier Transform of a short sample of noise or vibration data, often in real time. The Fourier analyzer can compute the spectral density of noise or vibration data by computing the product of the real and imaginary parts of the Fourier transform, thus performing as a constant bandwidth narrowband analyzer. Successive spectral densities of discrete samples of random data may be averaged to determine the spectral density of random noise and vibration data.

Finally, by appropriate energy summation, the spectral density may be used to approximate 1/3 octave or octave band data, although such data will not be identical with data analyzed with 1/3 octave filters meeting ANSI specifications (Ref. 4.6). This is discussed further in Sections 4.1.4.12 and 4.1.4.13, which include modal analysis and random data analysis.

4.1.4.5 Community Noise Measurements -- Accurate description of community noise involves measuring the A-weighted sound level exceeded n% of the time (L_n), the Equivalent Sound Level (L_{eq}), the Day-Night Sound Level (L_{dn}), and the Community Noise Exposure Level (CNEL). Usually L_{10}, L_{50}, and L_{90} are sufficient for community noise statistical analysis. Where federally funded residential housing is involved, comparison of statistical noise data with HUD criteria (Ref. 4.7) may require plotting the percent of time exceeded as a function of sound level. This is often referred to as the percent exceedance as a function of sound level (see Figure 4.2).

The Equivalent Sound Level (L_{eq}) and the Day-Night Sound Level (L_{dn}) are gaining wide acceptance as descriptions of community noise and are commonly used in the environmental assessment of rapid transit systems.

4.1.4.6 Impulse Noise Measurements -- Noise in maintenance shops often includes transient noise, such as that from hammer blows, impact wrenches, or air-handling equipment. Basically, the impulse noise level is the maximum rms noise level, determined for a standardized time duration of about 0.035 seconds. Impulse noise is generally measured with instruments that incorporate a special impulse detector, which, in addition to having a standard rms averaging time of about 0.035 seconds, has a relatively fast rise time. For continuous or steady-state noise, the impulse detector will give the same level indication as the standard "fast" or "slow" meter response settings of a sound level meter. Most measurements of impulse noise on rapid transit systems require an impulse detector.

4.1.4.7 Reverberation Time -- One of the most common acoustic measurements is the measurement of the reverberation time of a decaying sound field within an enclosure. Reverberation time is an important parameter for evaluating station and subway acoustics. The reverberation time is defined as the length of time required for a sound field to decay by 60 dB. A measure of the reverberation time allows a direct estimate of the absorption in a room and hence the degree of noise control provided by the absorption. The relationship between reverberation time and the absorption is:

in MKS units: $T=0.161\ V/(Sa_{sab})$ sec

in English units: $T=0.049\ V(Sa_{sab})$ sec

where V is the volume of the room, S is the total surface area of the room, and a_{sab} is the average Sabine absorption coefficient (Ref. 4.3). A more complete discussion of reverberation time is included in Appendix A.

4.1.4.8 Sound Transmission Loss -- The sound transmission loss of a panel or wall is the difference in decibels between the incident sound intensity and the sound intensity transmitted through the wall. Transmission loss measurement procedures are standardized (Refs. 4.8, 4.9) and require the generation of a random noise field within the transmitting room and the measurement of the resulting reverberant sound field on both sides of the wall under test. For the best representation of a panel, the transmission loss is usually measured in both directions and averaged. The existing absorption in the receiving room space must be determined from reverberation time measurements within the space.

Sound and transmission loss measurements have been performed on transit vehicles to determine the transmission loss characteristic of the car body floor, walls and ceiling. Transmission loss measurements may also be necessary to determine the transmission loss of structure walls designed to enclose particularly noisy equipment, such as a fan, or a chiller pump.

4.1.4.9 Driving Point Impedance -- The driving point impedance can be measured by measuring the driving force and resulting velocity simultaneously, using a shaker, accelerometer, integrator, force transducer, measuring amplifiers, and phase meter (Refs. 4.1, 4.10). This is useful for determining the stiffness requirements of vibration isolators and for characterizing the dynamic response of systems. Although information on driving point impedance can be very valuable, such measurements have rarely been performed at transit systems. They are not readily understood, are tricky to perform, and can often be avoided by relying on general "handbook" approaches.

4.1.4.10 Modal Analysis -- Modal analysis has become popular in the last decade, as a result of the advent of high-speed digital computers with analog-to-digital conversion capability. Briefly, modal analysis is the measurement of the resonance frequencies and amplitude distributions, or mode shapes, of the vibration modes of an object, and is closely related to impedance measurements (Ref. 4.1). Modal analysis can be useful for determining the locations of maximum surface strain or stress caused by vibration; potential failure mechanisms resulting from fatigue or excessive stress can thus be identified. Modal analysis has been performed on the fan blades of a major transit system to identify the cause of their failure. Modal analysis can identify the significant noise-radiating surface vibration modes of specific rail transit system vehicle components, such as wheels and aerial structure surfaces.

Although modal analysis is usually identified with very sophisticated instrumentation, it is possible for mode shapes and frequencies to be measured for relatively simple, beamlike or platelike elements, such as an axle or wheel, simply by sinusoidally exciting the object with a small electromagnetic shaker. The amplitude of vibration is measured at a number of points, with an accelerometer and measuring amplifier. This technique makes use of the fact that when an object is excited at its modal frequency, all of the points on the surface of the object will be vibrating either in phase, or 180 degrees out of phase, so that the complex relationship between the excitation point and the measuring point is no longer important. This process, however, is very time consuming.

One of the major advantages of the computer-based modal analysis system is its ability to present, in a dynamic and graphic fashion, the vibration mode under investigation. These "animated mode shapes" can be quite informative, leading to rapid diagnosis of vibration problems. Computer-based modal analysis also allows for the measurement of the damping factors associated with each particular mode, by evaluating the bandwidth or Q-factor of the mode.

4.1.4.11 Random Data Analysis -- The term "random data analysis" refers to the measurement of power spectral density, cross-power spectral density, and auto- and cross-correlation measurements. This topic is a field in itself and is described in detail in the literature (Refs. 4.1, 4.4, 4.11).

The measurement of these quantities requires sophisticated instrumentation and measurement techniques, usually consisting of an analog-to-digital converter, interfaced to a high-speed mini-computer and large storage medium. However, real-time power-spectral density analyzers, correlation analyzers, and probability amplitude distribution analyzers can all be rented or purchased.

To date, measurements of these quantities have been performed primarily as research (Ref. 4.12) and have not been designed to solve specific noise and vibration control problems on rail transit systems. However, such measurements may prove valuable for the description of component vibration and for reliability testing. The measurement and interpretation of such data require considerable technical expertise and should not be attempted without first considering conventional measurement techniques.

4.2 TYPICAL MEASUREMENT AND ANALYSIS SYSTEMS

The task of purchasing instrumentation for the measurement of rail transit noise and vibration is formidable. Not only must one choose from the wide variety of equipment types available, but between several manufacturers of any given instrument. The purpose of this section is to help organize the options by describing four basic systems for acoustical measurement and analysis. These range from the simplest possible system (a sound level meter), to a sophisticated, computer-based system suitable for detailed measurement and experimental research. Each step beyond the basic sound level meter is a logical extension that will significantly enhance data collection and analysis capabilities. The basic instrumentation arrays discussed are:

1. Basic noise and vibration measurement system: sound level meter;
2. A recording and analysis system: sound level meter, tape recorder, and graphic level recorder;
3. Real-time analysis system: same as (2) with addition of a real-time 1/3 octave band analyzer;
4. Computer system: same as (3) with the interfacing of the real-time analyzer to a computer system.

4.2.1 Basic System

The sound level meter is a self-contained, portable, battery-operated instrument for measuring sound and, with appropriate adaptors, vibration levels. Virtually all sound level meters allow selection of A, B, or C-weighting. Many are also capable of octave or 1/3 octave band analysis. Some of the current sound level meters allow direct measurement of L_{eq} and sound exposure levels for any given time period.

A sound level meter is the only required instrument in a large number of measurement situations. The primary disadvantage is that there is no documentation of the level other than manual tabulation; repeat measurements require re-enactment or re-measurement of the event. With steady-state noise such as fan noise this is not a major disadvantage (except that all the analysis must be performed in the field); however, with transient events, such as train passbys, reading the maximum level directly from a sound level meter requires some discretionary judgement that cannot be checked later.

Specific examples of measurement situations that can be performed with only a sound level meter are:

- estimating the maximum level of train passby noise within plus or minus 1 to 2 dBA,
- measuring levels of steady-state noise from sources such as fans or escalators,
- measuring statistics of community noise using a manual method,
- preliminary evaluation of occupational exposure to shop noise, and
- measuring overall vibration level (or octave band vibration level of steady-state vibration).

A block diagram of a precision sound level meter capable of measuring both noise and vibration is presented in Figure 4.5. Either a microphone or an accelerometer may be used. The transducer signal is first conditioned with an input preamplifier because most transducers are incapable of supplying the necessary signal power to the input attenuator preceding the first amplifier stage, which amplifies the signal to a usable level. The output of the first amplifier may be fed directly to the second stage attenuator, permitting overall measurements, or may be switched to one of the weighting networks or an external filter. Octave band analyzers or 1/3 octave band analyzers may be used as an external filter. If an accelerometer is used to measure vibration, an

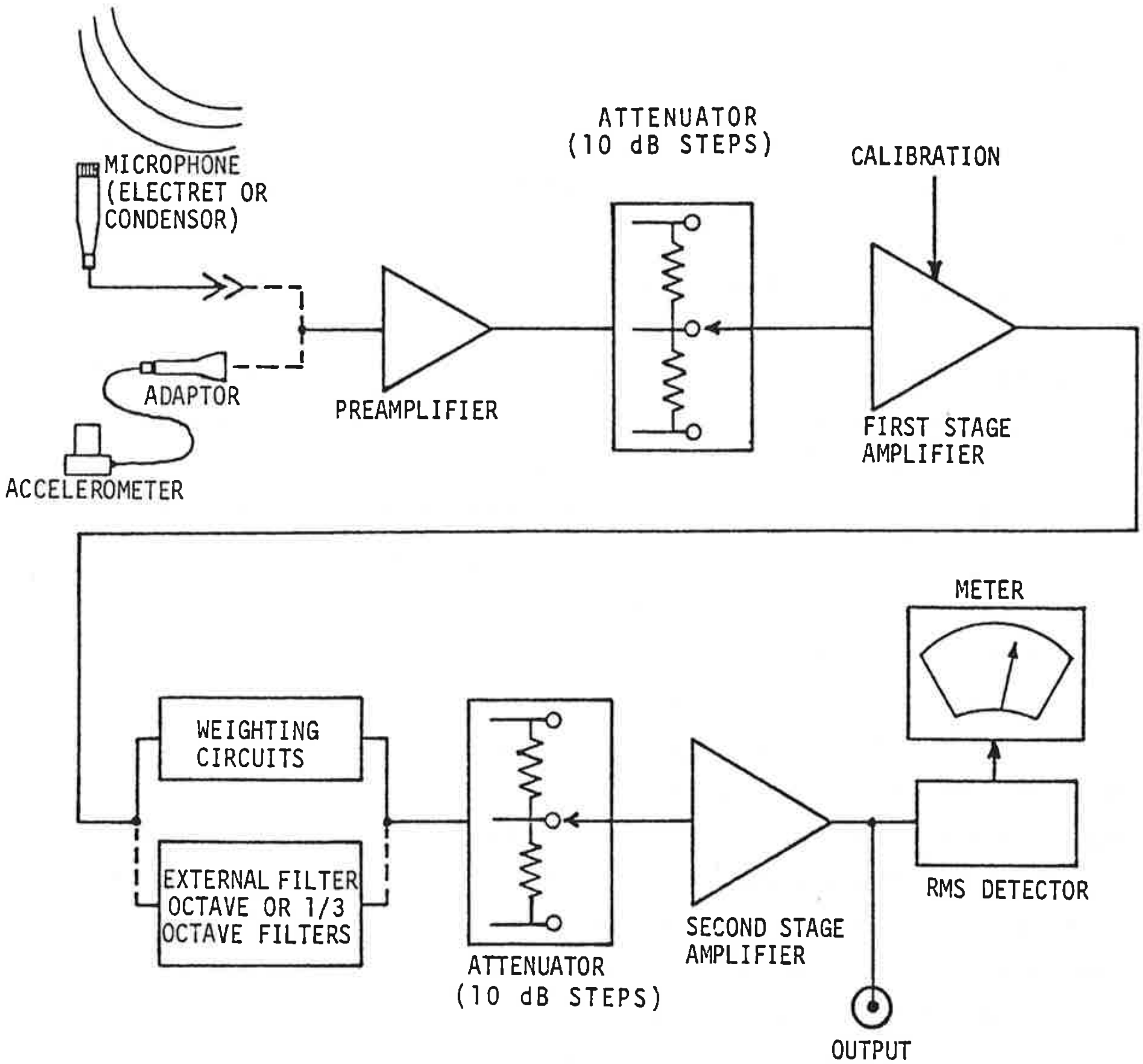


FIGURE 4.5 BLOCK DIAGRAM OF A SOUND LEVEL METER SUITABLE FOR NOISE AND VIBRATION MEASUREMENTS

integrator may be used as an external filter to convert the acceleration signal to a velocity signal. The second stage attenuator and amplifier provide any necessary post-filter gain necessary for detection and display of the result. An analog output signal is usually available for recording or other purposes. Note that on many sound level meters the impact and second stage attenuators are combined on one knob.

Any measuring device is incomplete without a means for calibration. Special handheld calibrators that are placed on the microphone can be used to calibrate sound level meters in the field. Portable battery-operated accelerometer exciters are also available for calibration of accelerometers.

The basic configuration of the sound level meter as shown in Figure 4.5 is similar to almost any type of noise and vibration analysis system, including sophisticated real-time analysis systems, in that the signal is first amplified, filtered, amplified a second time if necessary, and then detected, with the result displayed on an indicating device. The same basic considerations regarding use of the sound level meter attenuators and filters are thus applicable to more sophisticated noise and vibration analysis systems.

Three grades of sound level meters are designated by the American National Standards Institute (Ref. 4.13):

Type 1, Precision
Type 2, General Purpose
Type 3, Survey

The Type 1 precision sound level meter has the closest tolerance, of all three types, being ±1 dB from 50 to 4,000 Hz. Outside of this range the tolerances become slightly greater, owing primarily to limitations of the equipment. For Type 2 and Type 3 sound level meters the tolerances in the mid-audio frequency ranges are about ±2 and ±3 dB, respectively. However, for these latter two types, the tolerances at frequencies below about 100 Hz and above about 2,000 Hz are significantly relaxed. Also, detectors used in Type 2 and Type 3 sound level meters are usually less accurate than in Type 1 sound level meters.

The sound level meter tolerances are determined primarily by the microphone and rms detector characteristics. In addition, the weighting networks, e.g., the A-weighting network, may be increasingly less accurate at frequencies approaching the extreme ends of the audio spectrum.

A Type 1 precision sound level meter is recommended for most rapid transit measurements, although in many cases a Type 2 is adequate. If it becomes necessary to use the measurement results in legal proceedings, it is best that the data be from a Type 1 sound level meter. Another factor is that groundborne noise and vibration are primarily confined to the low-audio and sub-audio frequency range, a range where sound level meter tolerances are not as tight as at the mid-audio frequencies. Only the Type 1 sound level meter preserves close tolerances down to about 10 Hz. The Type 2 and Type 3 sound level meter tolerances are not even defined below 20 Hz.

Most precision sound level meters provide integral or optional external capability for octave band or 1/3 octave band analysis. Generally, an octave band analyzer is sufficient for most measurement purposes, and such an analyzer is recommended for general transit system noise and vibration measurements.

In view of the current emphasis on L_{eq}, L_{dn} and CNEL as community noise descriptors, the selection of an integrating sound level meter may add further flexibility. Units are available which may measure the L_{eq} over durations from a fraction of a minute to as long as 24 hours. The noise exposure level, NEL, can be determined by an L_{eq} measurement over a given time period and adding 10 log(T) to the result, where T is the time duration in seconds. Specifically, the Single-Event Noise Exposure Level, SENEL, may be easily determined for individual train passbys.

4.2.2 Recording and Analysis System

Although a large number of measurements can be performed with only a sound level meter, and as a minimum a sound level meter should be available at all transit systems, the sound level meter by itself is awkward for many of the noise and vibration measurements required at rapid transit systems. A specific example is octave band analysis of train passby noise. Without the ability to record noise samples for repeated reproduction, numerous train passbys would be needed, in order to obtain accurate statistical averages of the levels in all of the octave bands.

Presented in Figure 4.6 is a basic recording and analysis system that will fulfill most transit system needs. The heart of the system is a sound level meter (SLM) with octave and/or 1/3 octave band filters. A battery-operated portable tape recorder is used with the SLM to record the data in the field. Subsequent analysis of the tape recorded signal in the laboratory with the SLM and a graphic level recorder allows display on strip chart records of the overall or A-weighted levels and octave or 1/3 octave band analysis by repeatedly playing back the data and manual stepping of the analyzer bands.

The graphic level recorder allows production of archival strip chart records of the level of noise or vibration recorded on magnetic tape. Thus, the entire time history for recorded data may be easily referenced from the strip chart records. Also, the passby noise level signature as a function of time may be illustrated for publication. Generally, good quality graphic level recorders are capable of a wide variety of tasks and are fundamental instruments in most acoustic laboratories.

The tape recorder indicated in Figure 4.6 should be of laboratory grade and capable of an audio frequency response of about 16 Hz to 10 kHz with a tolerance of about ± 1 dB. Note that if Type 1 precision sound level meter specifications are to be preserved for

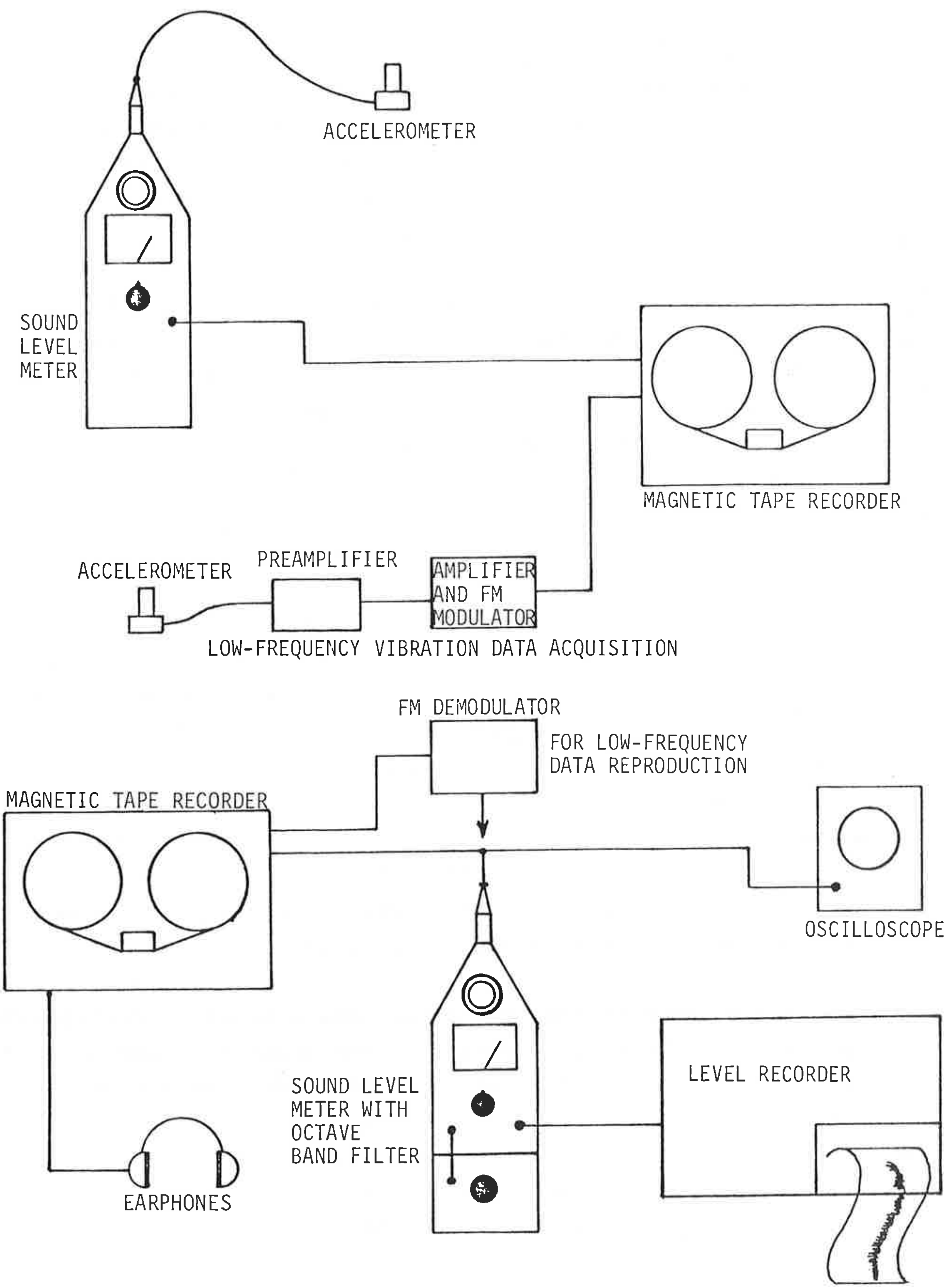


FIGURE 4.6 BASIC NOISE AND VIBRATION DATA ACQUISITION AND ANALYSIS SYSTEM

a given measurement, it may not be possible to include the tape recorder in the measurement chain. Conversely, to meet specifications of Type 2 sound level meters, using a tape recorder with the above tolerance, the sound level meter must meet specifications for Type 1 precision sound level meters.

Recording and reproduction of low-frequency vibration data from 1 Hz to perhaps mid-audio frequencies generally require frequency modulation (FM) recording. This technique is widely used in the field of acoustics and for noise and vibration measurements at transit systems in particular. A particularly good example is a system used by the Toronto Transit Commission's Engineering Department (Ref. 4.14).

The apparatus is indicated in Figure 4.6 which allow FM recording with any standard tape recorder otherwise capable of only direct recording. FM recording is accomplished by recording on magnetic tape a carrier signal whose frequency deviation from an unmodulated center frequency is directly proportional to the instantaneous amplitude of the analog signal. Thus, the low-frequency response of the tape recording is not limited by the tape recorder record-reproduce response, but only by the transducer, amplifier, and modulator/demodulator response. Indeed, one of the features of FM recording is that a very flat amplitude and phase response may be achieved over the recording bandwidth.

The apparatus shown in Figure 4.6 includes a special accelerometer preamplifier, an integrated amplifier and modulator, and a demodulator. The typical sound level meter does not have a flat low-frequency response below 10 or 20 Hz, and is, therefore, not used. Such a system as indicated in Figure 4.6 has been successfully used to record groundborne vibration as low as several micro-g's over a frequency range of 1 to 1000 Hz.

Some tape recorders have integral frequency modulation and demodulation capability. Indeed, multichannel instrumentation recorders may be configured for either direct or FM recording capability as required. However, almost all tape recorders with integral FM recording capability do not provide the necessary preamplification and gain, so that special amplifiers with extended low-frequency responses are required.

For a measurement system which uses a tape recorder in the measurement chain, periodic checking of the record-reproduce response characteristics is essential. For this purpose, a stable audio oscillator or signal generator capable of producing a sine wave signal at frequencies between 10 Hz and 20 kHz is required. The manufacturer of the graphic level recorder usually can supply an oscillator which may be used to produce frequency response curves which are more easily interpreted than tabulated responses at discrete frequencies. Such a system is illustrated in Figure 4.7. In this system the oscillator under control of the level recorder provides a signal of swept frequency which is applied to the device under test. The output of the test device is then amplified and the level recorder displays the output level as a function of frequency on specially printed paper.

The frequency response of any instrument (e.g., amplifiers and filters) may be checked in this manner. Regular evaluation of frequency response characteristics helps to identify malfunctioning equipment and further guarantees reliable and accurate data acquisition and analyses.

The manufacturers' handbooks and catalogs show detailed descriptions of typical frequency response measuring systems and describe their operation fully.

4.2.3 Real-Time Analysis Systems

A real-time analysis system is described in Figure 4.8 and may include any of several types of real-time analyzers, including 1/3 octave, and constant bandwidth narrow band analyzers. A real-time analysis system is useful when a large number of analyses must be

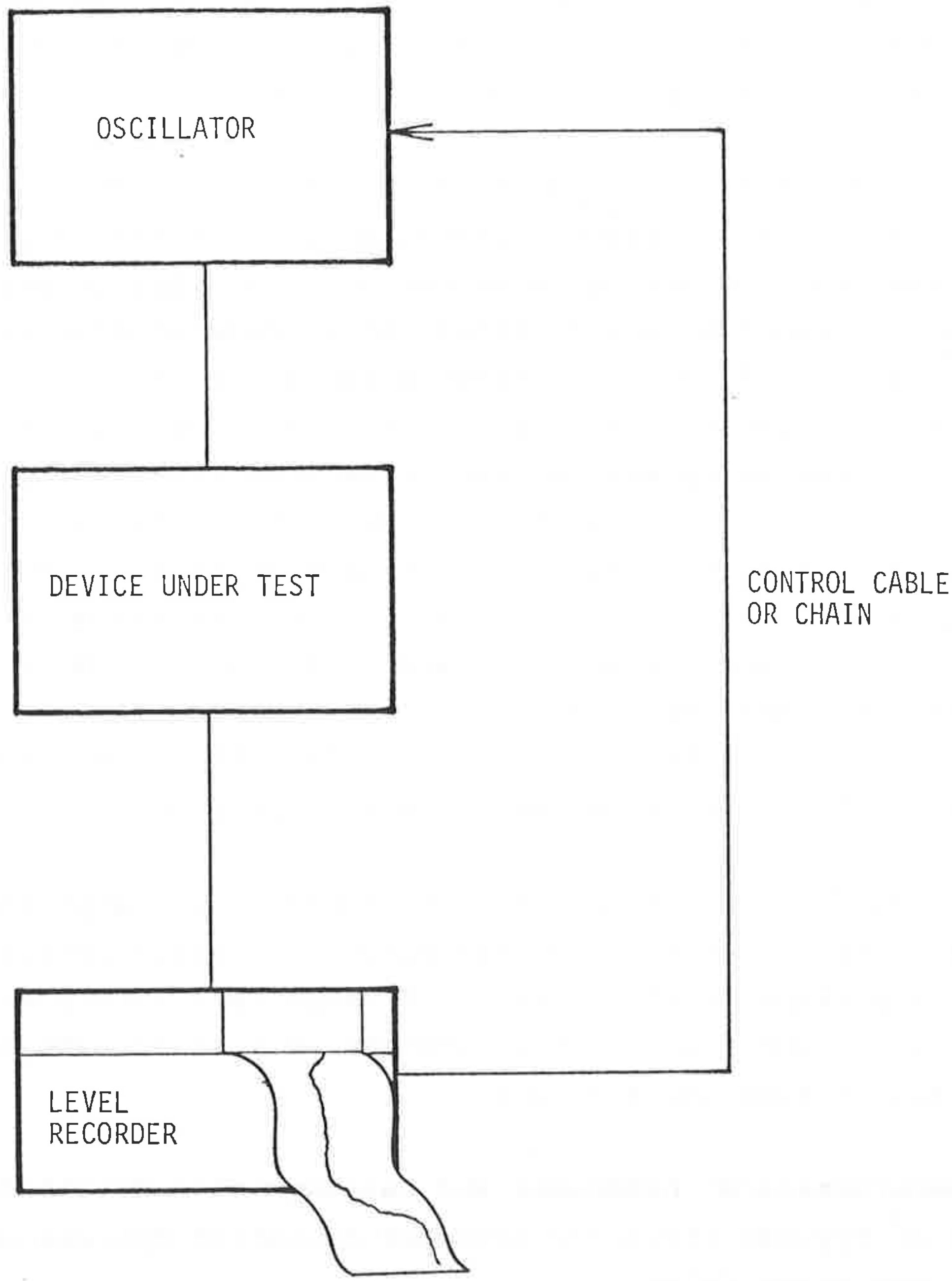


FIGURE 4.7 A SYSTEM FOR MEASURING FREQUENCY RESPONSE

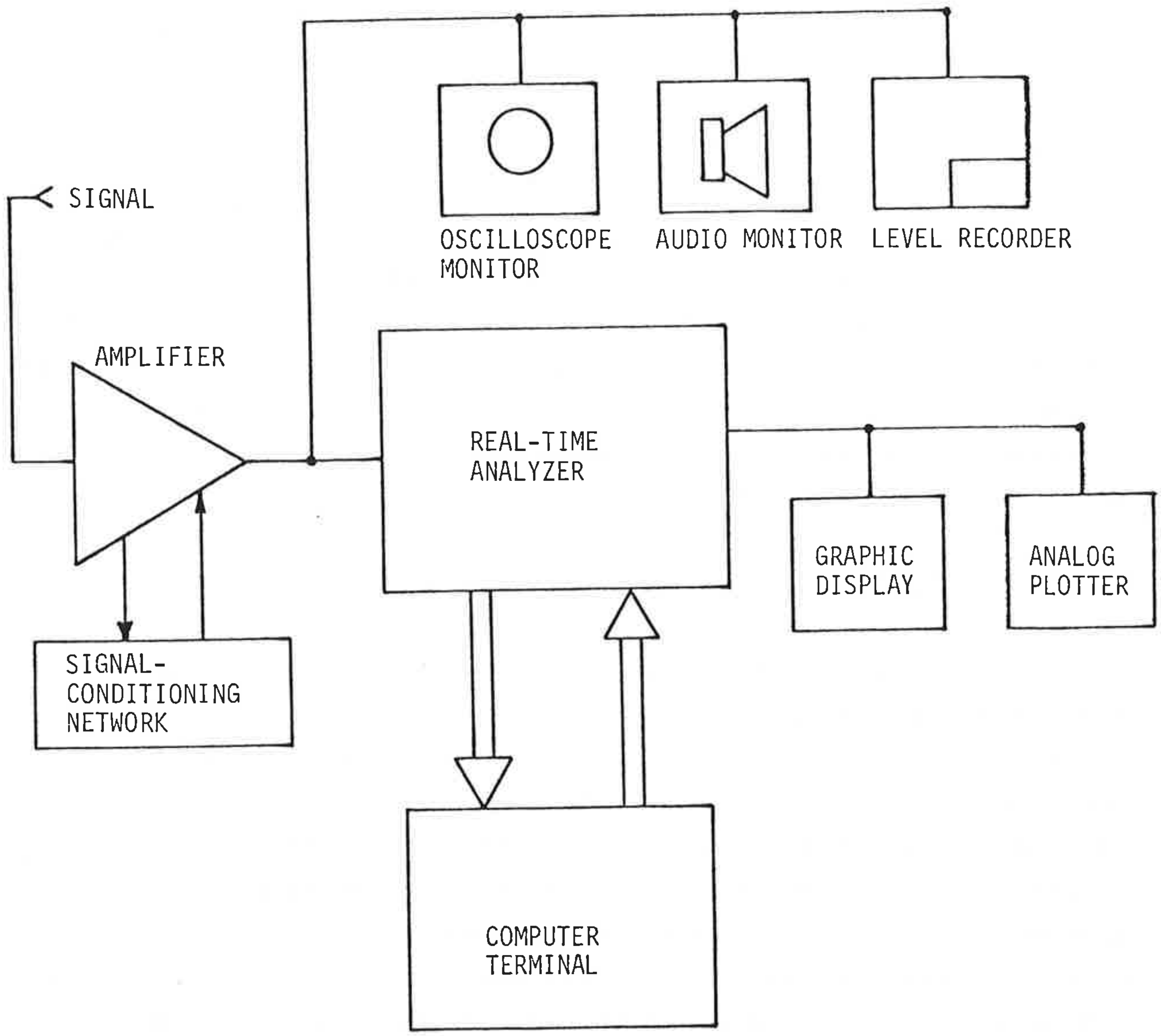


FIGURE 4.8 REAL-TIME ANALYSIS SYSTEM

performed on a more or less regular basis. Furthermore, such a system may be interfaced to a computer such as described under Section 4.2.4 for large-scale data processing and management.

The sound level meter and external high-pass and low-pass filters are used for calibration and signal conditioning. In place of a sound level meter, laboratory measuring amplifiers are available which provide a wide variety of measurement capabilities and a generally broader frequency response. A general purpose measuring amplifier must be used for the analysis of very low-frequency data because the sound level meter response is usually inadequate. External high-pass and/or low-pass filter can be used to exclude unwanted noise or vibration frequencies.

From the output amplifier the signal is sent to the real-time analyzer, a monitoring oscilloscope, level recorder, and audio monitor. Although not shown in Figure 4.8, an additional sound level meter might be used to drive the level recorder, thereby allowing a display of the A-weighted level during analysis. The audio monitor is included to allow continuous audio monitoring of the data. The oscilloscope is used to monitor the peak magnitudes of the signal and check that there are no overloads. The oscilloscope also allows a qualitative visual analysis of the signal. A CRT storage display unit is included for real-time display of the spectral data, and a plotter is included for hard copy. Most real-time analyzers contain the necessary horizontal and vertical analog output signals for CRT and plotter control. A digital printer or terminal may sometimes be connected to the real-time analyzer for printing out levels. Some real-time analyzers combine the function of amplification, real-time analysis, and real-time display of the spectrum. The more sophisticated real-time analyzers allow comparison of two spectra and averaging of spectra.

Generally, two basic types of real-time frequency analyzers are available. The first type is the 1/3 octave real-time analyzer with filter response characteristics conforming to ANSI specifications for 1/3 octave band filter sets (Ref. 4.6). The other type is often described as a Fast Fourier Transform or spectral-density analyzer with constant filter bandwidth characteristics. These latter analyzers may often be fitted with options allowing computation of 1/3 octave spectra from the constant bandwidth spectral densities.

The selection of an appropriate real-time analyzer is often a subject of great deliberation, and a thorough understanding of the capabilities and limitations of the various types of analyzers is necessary before selection may be made. Generally, a constant bandwidth analyzer of any type, fitted with a 1/3 octave band option will not conform to ANSI specifications for 1/3 octave band filter sets. Thus, if 1/3 octave data collected with an analyzer meeting such specifications is of prime consideration, a constant bandwidth analyzer should not be selected. On the other hand, if 1/3 octave data are not particularly important, but detailed spectral or narrowband analysis is, a constant bandwidth spectrum analyzer such as an FFT analyzer will often provide valuable flexibility and versatility. However, for general noise and vibration measurement and analysis, the 1/3 octave representation is most appropriate. Indeed, narrowband constant bandwidth spectra are often much too detailed and add confusion.

4.2.4 Computer Systems

Figure 4.9 is a block diagram of a computer-based system for noise and vibration measurements. Such a system will allow program control of the real-time analyzer, digital processing, and storage of real-time data. The system may be programmed for statistical analyses, Fourier analyses, regression analyses, and numerous other applications. The real-time device may be a real-time analyzer (e.g., 1/3 octave band or constant bandwidth narrow band analyzer) or may be an analog-to-digital (A/D) converter, for conversion of analog signals to digital form for digital processing including Fourier analysis. Often the A/D converter

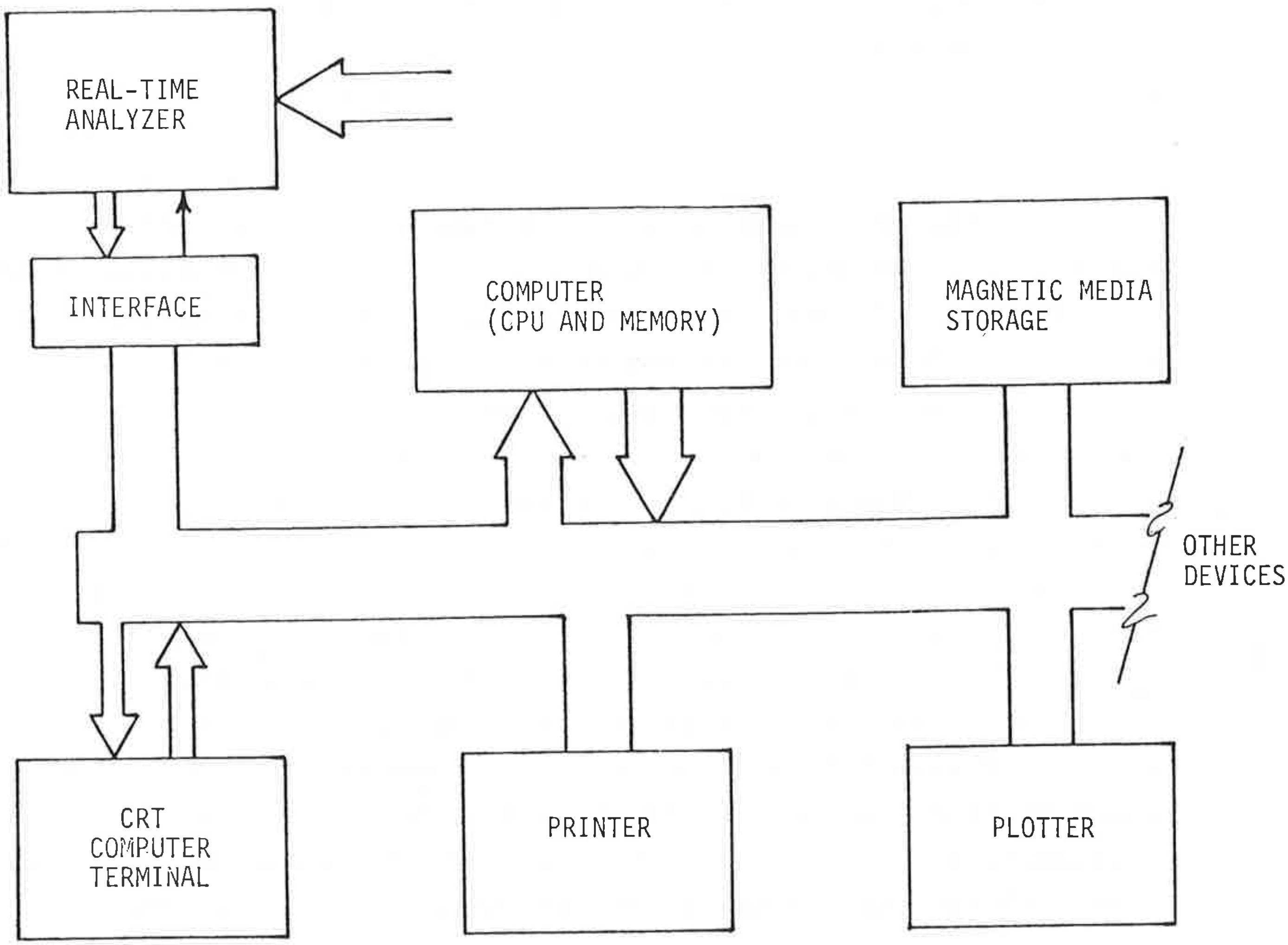


FIGURE 4.9 REAL-TIME ANALYSIS INTERFACED TO DIGITAL COMPUTER

may be included within the computer system main frame.

The interface shown in Figure 4.9 translates the data and control signals between the real-time device and the computer. The interface should be selected with care since the interface determines the transfer speed of data and the control capability of the real-time device.

Auxiliary peripherals included in Figure 4.9 are: a plotter for presentation of graphical data, CRT terminal, and a printer for hard copy of programs and numeric data.

4.3 GENERAL CRITERIA FOR SELECTION OF MEASUREMENT SYSTEMS

A number of important criteria must be established prior to selection of noise and vibration measurement equipment. These criteria concern versatility, bandwidth, dynamic range, sensitivity, filtering requirements, detection, display of data, data processing and storage, and cost and availability.

4.3.1 Versatility

A number of various types of measurements are usually needed for evaluation of noise and vibration parameters at a rapid transit system, ranging from component vibration to community noise analysis. Thus, although a measurement system may satisfy a specific purpose, attention should be focused on other possible uses and future testing requirements. No single instrument can accomplish all functions. However, a carefully chosen instrument can often be used for a variety of measurements without greatly increased cost. An example is the integrating sound level meter or a multipurpose measuring amplifier.

4.3.2 Bandwidth

The overall frequency bandwidth of the entire measurement system from transducer to detector should be greater than or equal to that required by the analysis. The frequency response is affected by the transducer, tape recorder, amplifiers, and detector. Generally, the bandwidth and frequency response of the transducers and mounting configurations used for data acquisition determine the effective bandwidth of the system. For most rapid transit airborne noise measurements , a frequency response of ±1 dB from 10 Hz to 12 kHz is normally adequate. The frequency response for groundborne vibration and noise measurements should be about ±1 dB from 2 Hz to at least 500 Hz. The bandwidth required for the analyses of the shock and vibration environment of components, such as those mounted on the vehicle truck, should be as wide as possible -- typically extending from 1 or 2 Hz to 10 kHz, and should exhibit constant group delay over this range. Without constant group delay, the waveform shape of a vibration shock may be distorted. In most cases, the upper limit of frequency response is determined by the transducer resonance characteristics and mounting configuration.

4.3.3 Dynamic Range, Signal-to-Noise Ratio, and Crest Factor

The dynamic range of a measurement system is the total range of signal voltage capable of being handled by the system. It is usually the sum of the signal-to-noise ratio and the crest factor capability of the instrument. Signal-to-noise ratio is the ratio of the nominal "full-scale" output voltage to the output-rms-noise floor voltage. The crest factor capability is the ratio of its instantaneous peak signal-handling output capability and its nominal full-scale output voltage.

The required dynamic range of measurement instrumentation should be determined from the maximum and minimum signal amplitudes expected for a given gain setting. For most measurements, a dynamic range of 60 dB for a bandwidth of 20 kHz is adequate. Most 1/3 octave band analysis systems have a total dynamic range of approximately 60 to 70 dB, of which only 50 to 60 dB may actually be displayed. Use of a magnetic tape recorder may limit the dynamic range to 50 to 55 dB although 60 dB is achievable with high-quality audio magnetic tape recorders.

4.3.4 Sensitivity

The sensitivity of a transducer is the relation between output voltage and the input parameter. For accelerometers the sensitivity is usually given in terms of the output voltage or electrical charge generated per unit gravitational acceleration, or g. For microphones, the sensitivity is usually given in terms of the output voltage per Pascal, or, equivalently, as a level in dB (re 1 volt per Pascal). The required transducer sensitivity for performing noise or vibration measurements is based on the minimum signal level expected and the electrical noise of the preamplifier, amplifier, and transducer.

Typical vibration transducer sensitivities for accelerometers are about 10 mv/g to 500 mv/g. Accelerometers used for ground vibration measurements should have a sensitivity greater than about 200 mv/g. For transit car component shock and vibration measurements sensitivities of 10 to 50 mv/g are appropriate. Typical microphone sensitivities for community noise analyses or equipment noise measurements should be from about -20 dB to about -40 dB (re 1 volt per Pascal). Where very low levels of noise may be expected, e.g., during the night in residential or rural areas, the sensitivity of the microphone should be as high as possible, typically on the order of -20 dB to -30 dB (re 1 volt per Pascal). The minimum measurable signal for a given transducer is determined by the noise characteristics of the preamplifier and amplifier. Thus, use of a "noisy" preamplifier or amplifier may cancel the benefits which might be realized from a high-sensitivity transducer.

4.3.5 Weighting and Filtering Requirements

The filtering or weighting requirements of an analysis system depends upon the requirements of the measurement project. Typical requirements are A-weighting, octave band, or 1/3 octave band filtering, and perhaps 1/10 octave band filtering. Additionally, an integrator may be required to convert measured acceleration signals to vibration velocity, and possibly to vibration displacement. A variable bandwidth filter may be required to define the overall measurement bandwidth and reduce extraneous effects such as transducer resonances, or wind noise, which can cause high extraneous signal levels detrimental to the analysis of the desired signal.

4.3.6 Detection

Every measurement system requires some means of detection which converts the analog signal into an analog or digital representation of the quantities which describe the analog signal. Types of detection consist of (1) root mean square (rms), (2) peak, (3) absolute average (rectified average), and (4) analog display.

RMS detection requires the squaring, integration, normalization, and square rooting of the analog signal. This may be done by analog modules or by digital detection and digital processing. Before self-contained analog or digital true rms detectors were available, quasi-rms detectors were used, which were essentially complex rectifier circuits usually calibrated to accurately represent the rms amplitude of a sine wave signal. This technique will introduce some error when complex periodic or random signals are analyzed.

In most cases, transit system noise and vibration measurements require rms detection. The rms detectors used in sound level meters usually have provisions for "fast," "slow," and in some cases, "impulse" responses. All good quality real-time analysis systems employ rms detection, or equivalently, energy summation. So-called quasi-rms detectors approximate the rms value of a complex signal waveform, and although adequate in most situations, will introduce some uncertainty in the results. Absolute average detection, an inferior form of detection used in less expensive instrumentation to approximate rms values, should not be used for general noise and vibration measurements.

Peak detectors respond only to the positive peak, negative peak, or maximum of the absolute value of the positive or negative peak of the signal waveform. As such, they can be useful for assessment of physical damage resulting from excessive shock loading, but are not appropriate for more general analyses.

The simplest detection is an analog display of the signal amplitude as a function of time on an oscilloscope or strip chart. In this case, the actual waveform may be inspected for an assessment of crest factor and nature of the signal, e.g., random or periodic. This approach is not practical for general noise and vibration measurements, although special circumstances may require its use.

4.3.7 Processing and Storage

When a large number of 1/3 octave band spectra, or constant bandwidth power spectral densities are obtained, the capability for efficient processing and storage of the data is essential. A reasonable system would consist of a programmable calculator or minicomputer interfaced to the analysis instrumentation. The computer system should be capable of storing the analysis data on magnetic tape or disk and the required storage capacity should be reviewed carefully. Typically, about 3 bytes (24 bits) of storage are required for each number representing a noise or vibration level, in addition to program storage requirements. The computer system should also have a line printer and perhaps a plotter for hard copy.

4.3.8 Cost and Availability

Often the determining factor in the selection of any measurement and analysis system is the available budget and the delivery schedule, which may involve a delay of several months. Rental and leasing organizations supply a wide assortment of instrumentation, ranging from sound level meters to constant bandwidth power spectral density analyzers. If there is no continuing use for the instrumentation, rental or leasing can be the best arrangement. When receiving any instrument, either from a rental organization or a supplier, sufficient time should be allowed for checking-out and calibrating the equipment.

If buying equipment, very careful consideration should be given to future uses. By spending slightly more, it is often possible to greatly increase the number of types of measurements that can be performed. On the other hand, instruments which have a large number of capabilities and functions are often difficult to use. Thus, before spending a great deal of money, the potential user or manager should be thoroughly acquainted with the instrument to be purchased. This is the function of the sales representative, who should be consulted as often as is necessary. A good sales representative can be a very valuable asset, not only to the manufacturer, but to the end user.

4.4 ACOUSTIC DATA ACQUISITION

General methods and instrumentation for the measurement of sound pressure levels are specified by the ANSI (Ref. 4.5). These methods provide a starting point for determining and executing needed measurements of rapid transit system noise. Review of the ANSI methods is a valuable prerequisite for performing such measurements. Other good references are Beranek (Ref. 4.2) and Harris (Ref. 4.3).

4.4.1 Use of the Sound Level Meter

The sound level meter is the most common instrument employed for the measurement of sound. A thorough understanding of its capabilities and limitations will aid acquisition of reliable data. Obviously, the first step when using a sound level meter is to read the "Operator's Instructions."

When making sound level measurements, the optimum orientation of the sound level meter with respect to the sound source depends on the type of microphone being used. The microphone should always be held at arm's length away from the body to minimize the possibility of the operator's body influencing the measurements. If practical, a tripod should be used. Generally, the microphone should be held about 1.6 m above the floor or ground surface. Some standards also indicate 1.2 m or 1.9 m for specific purposes.

The ANSI standard method (Ref. 4.5) requires that the microphone be held so that the sound energy from the principal source arrives from the side. This is practical for stationary sources, but for moving sources, such as a train, the holder must necessarily face the line of travel. In this case, the ANSI method specifies that the sound energy strike the microphone diaphragm at the same angle of incidence relative to the microphone axis of symmetry. This requirement may be met easily by holding the microphone such that its axis is perpendicular to the direction of incidence, e.g., vertically for measurement of noise from at-grade ballast-and-tie track. This is referred to as grazing incidence, which is appropriate for some pressure response microphones. For 1-inch free-field microphones, a random incidence corrector should be employed.

Flexible extenders or goosenecks are available for most sound level meters which allow the microphone to be extended about .3 m away from the sound level meter body, thus reducing effects caused by reflection of sound waves from the sound level meter body. Although more precise measurements of sound can be performed with this type of extender, there can be problems with lack of contact or contamination of the contact between the sound level meter and the extender. Therefore, to increase reliability, the flexible extender may be deleted from a measurement system.

For most measurements of transit system noise, a 1-inch free-field microphone with random incidence corrector is adequate. However, 1/2-inch microphones of the same sensitivity as the 1-inch free-field microphone are available which may be used without random incidence correction. Wind screens should be used at all times, not only to reduce the effects of wind buffeting, but to provide protection for the microphone body.

When making weighted sound level measurements, or using an octave band or 1/3 octave band analyzer, the following procedure should be followed:

1. Check battery condition and calibrate.

2. Set the sound level meter to the "overall" or "linear" position.

3. Reduce the output amplifier gain (10 dB step attenuator) to a minimum. (Note: On some sound level meters the input and output attenuator functions are combined on one knob, and this step is not needed).

4. Increase the input amplifier gain (10 dB step attenuator) until the meter indication is within 10 dB of full scale.

5. Set the sound level meter to the weighted position, e.g.,

A-weighting, or the octave or 1/3 octave band filter position.

6. If the meter indication is within 10 dB of full scale, no further adjustment is required; proceed to step 7. If the meter indication is down 10 dB below full scale, increase the output amplifier gain until the meter is within 10 dB of full scale and proceed to step 7. If the meter indication is higher than full scale, decrease the input amplifier gain until the meter is again within 10 dB of full scale and proceed to step 7.

7. Observe and record the sound level meter reading. Use the "slow" meter response for most types of measurements, e.g., community noise, or steady-state noise. Follow the ANSI procedures outlined in Ref. 4.5. For moving sources and transient noises such as for exterior train noise, or door actuator noise, or wheel squeal, use the "fast" meter response (Ref. 4.15).

The procedure outlined above is designed to avoid overload of either the input amplifier or output amplifier stage of the sound level meter, thereby guaranteeing accurate measurement of weighted or filtered sound levels. Note, only under relatively rare circumstances will a meter indication in excess of the full scale exist during step 6, owing to the nature of most weighting networks and filters. However, with a D-weighting filter and sometimes even with an A-weighting filter, this condition can occur.

4.4.2 Statistical Analyses in the Field Using the Sound Level Meter

Presented below is a technique for performing statistical analysis of noise levels with a sound level meter. This method utilizes a histogram method for obtaining a simplified form of statistical distribution using simple sound level meter measurements. The method has been compared with more precise estimates using computer analysis with very favorable results for typical noise level distributions (Ref. 4.16).

This method involves reading the A-weighted noise levels from a handheld sound level meter at 5-second intervals and placing a mark in the appropriate column or "bin" on a form. Figure 4.10 shows the type of data-taking form which is used. The procedure for using this form is relatively simple and is summarized below:

Data Taking: Read the instantaneous A-weighted sound level on a sound level meter at 5-second intervals and indicate the level in the appropriate column. Generally, a maximum of 150 counts (or until one column is completely filled) is sufficient to define the statistical distribution at a location.

Determining Statistical Distribution: Determine the total number of counts or readings taken (150 in example).

L_{90}: Divide the total counts by 10. This provides the number of counts (15 in example) used to determine the L_{90}.

Starting at the lowest sound level recorded (left hand side), count up the required number of marks (15). When this number is reached, the L_{90} level is in that column.

L_{50}: Divide the total counts by one-half (75 in example).

Start at either end and count in towards the center of the histogram until this number is reached. This column is the L_{50} level.

L_{10}: Follow the same procedure for L_{90}, except that the count is started from the highest sound level recorded (right hand side).

The equivalent sound level, L_{eq}, for the histogram may be calculated as:

$$L_{eq} = 10 \log[(1/N)(n_L \times 10^{L/10})]$$

where, L = the middle sound level of each bin and ranges over all sound levels (bins).

n_L = the number of counts in the bin corresponding to L.

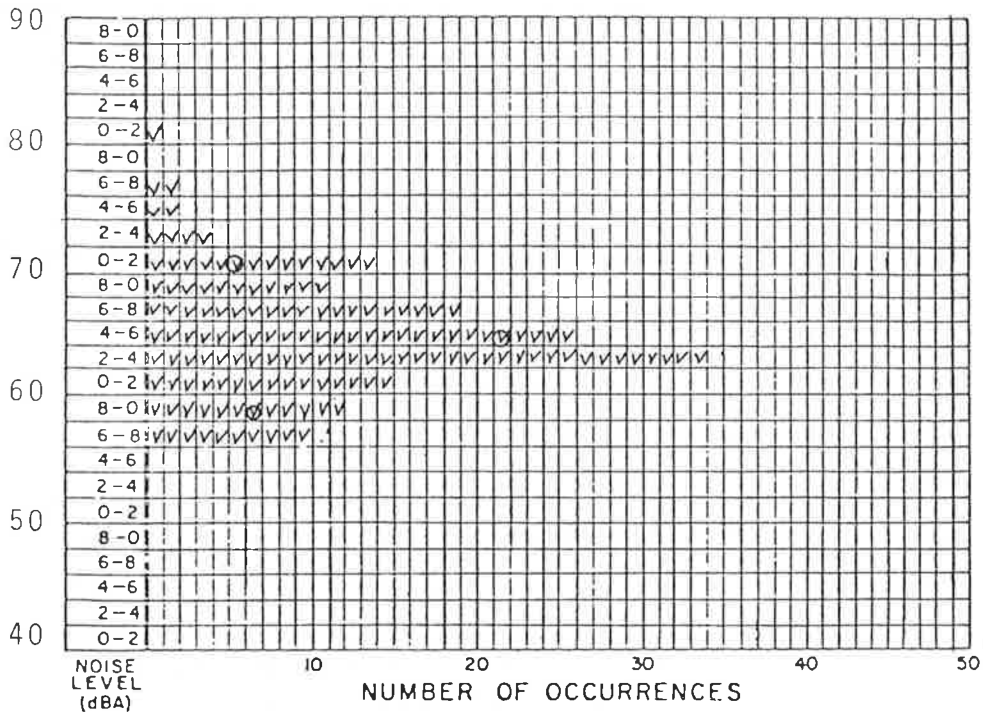
N = the total number of counts.

4.4.3 Measurement Positions

4.4.3.1 Wayside Noise -- Microphone locations for measurement of noise from at-grade or elevated trains are usually specified for various distances from the track centerline and at a particular height above the ground surface, usually at about ear or axle level. The axes of symmetry for the microphones are usually oriented vertically, although this can depend upon the microphone used.

AMBIENT NOISE SURVEY DATA SHEET

POSITION:
ENGINEER:
JOB NO. ________
DAY OF WEEK: ________ DATE: ________ TIME: BEGIN ________ FINISH: ________
CAL: BEGIN ________ FINISH: ________
NOTES AND SKETCH:
SKY: ________
WIND: ________
dBA L_{10}: ________
LIMITS, dBA: ________



TOTAL NUMBER OF SAMPLES = 150

L_{90} = 58 to 60 dBA

L_{50} = 64 to 66 dBA

L_{10} = 70 to 72 dBA

L_{eq} = 68 dBA

FIGURE 4.10 EXAMPLE OF FIELD STATISTICAL ANALYSIS USING A SOUND LEVEL METER (FORM IS FROM REF. 4.16)

ISO International Standard 3095-1975 "Acoustics - Measurement of Noise Emitted by Railbound Vehicles" (Ref. 4.17) recommends a specific measurement procedure for noise emitted by rail vehicles. Specifically, the recommended microphone locations for at-grade track are 7.5 m (25 ft) from the track centerline and 1.2 to 1.6 m above the railhead. Additionally, a second measurement at 3.5 m above the railhead is also recommended, as well as at distances of 25 m, 50 m, and 100 m from the track centerline. Passby noise measurements performed for preparation of the report entitled "Assessment of Urban Rail Noise Climates and Abatement Options," DOT-TSC-850 (Ref. 4.18) were at distances of 25, 50, 100, and 200 ft from the track centerline and approximately 1.6 m above the ground level.

For assessment of noise impact on affected buildings or properties, the measurement locations should be near setback lines of the buildings and/or at the property lines, depending on local code requirements or the specifics of the problem at hand.

In some circumstances it may also be desirable to perform measurements at different heights. An example is a tall building where the lower stories are in the shadow zone of a barrier but the upper stories have direct line-of-sight view of the noise source.

4.4.3.2 Car Exterior Subway Noise -- Measurements within a subway during passage of rail transit vehicles may be required for the assessment of transmission loss requirements for the vehicle car body, or the noise reduction effectiveness of noise control treatments applied to the tunnel or subway walls. Current practice is to place the microphone midway between the subway wall and vehicle and about 1.6 m above the safety walk. An alternative microphone location is directly against the subway wall at about 1.6 m above the safety walk. Because of the small space between the transit cars and the wall, it is usually not necessary to apply any reflection corrections on account of the proximity of the wall. Note that for most measurements close to an acoustically reflective surface in a reverberant field, it is necessary to apply a correction of about -3 dB to the measured value to account for the pressure doubling of the reverberant sound field at the wall. For a free sound field incident perpendicularly to a perfectly reflective wall, the correction is -6 dB.

4.4.3.3 Station Platform Noise -- Microphone locations for the measurement of station platform noise during train operation should be about 1.6 m above the platform and 2 m or one-half of the platform width from the platform edge, whichever is the smaller. Two measurement locations are recommended, one at the middle of a stopped train or middle of the platform and the other near the end of a stopped train but not less than 15 m (50 ft) from the end of the platform. Measurements should be made throughout the entrance, stopping, and exit of the transit train. These measurement locations were used for the preparation of Reference 4.18.

4.4.3.4 Vehicle Interior -- Microphone locations for the measurement of vehicle interior noise should be at about ear level, or 1.2 to 1.6 m above the floor. Historically, two measurement locations have been employed, one above one of the trucks, and the other at the midcar location, both in the aisle area or along the car centerline (Ref. 4.18). ISO Standard 3381-1976 (Ref. 4.19) recommends about five to seven microphone locations within the vehicle to define the interior noise levels. It also recommends that measurements be taken in the center of the driver's compartment at a height of 1.6 m above the floor and at 0.2 m from the operator's ear at ear level as well. This standard should be reviewed prior to planning vehicle interior noise measurements. For multiple car consists, measurements above the first or last truck will not be representative of those above the remaining trucks.

4.4.3.5 Fan and Ventilation Shafts -- Microphone location for the measurement of noise from either fan or vent shafts should be about 7.5 m from the center of the fan or vent shaft grating and about 1.6 m above the ground surface (Ref. 4.20). Alternatively the microphone may be located at the setback lines of affected buildings or at property lines. Background noise levels near the vicinity of the test site should be measured in the absence of fan or train passby noise from the shafts. In all cases local or regional community noise regulations and applicable criteria should be reviewed and measurement locations adjusted to allow direct comparison of measured levels with criteria and regulations.

Measurements of noise levels directly at the opening of a fan or vent shaft will allow an estimate of the output sound power of the shaft when integrated over the shaft opening. The formula to use for frequencies above about 100 Hz is:

$$PWL = SPL + 10 \log(A/2)$$

where: PWL = output sound power level in dB

SPL = sound pressure level in dB measured at the opening

A = cross-sectional area of the opening (m^2)

The factor of 1/2 is included in the area correction to account for the random incidence characteristics of the sound field within the shaft. For shafts with acoustical wall treatment the factor of 1/2 should be deleted. Below about 100 Hz the wavelength of sound becomes comparable with the cross-sectional dimensions of the shaft so that the above formula is no longer correct although it may still provide reasonable results if the factor of 1/2 is deleted. For the most accurate results, the sound pressure level should be measured at a number of locations over the shaft cross-section and energy averaged; however, one or two measurements near the center are usually adequate.

4.4.3.6 At-Grade Substations -- Measurements of substation transformer noise and cooling fan noise should be made at several locations around the substation at about 7.5 m from the substation perimeter. Alternatively, measurements at the setback line of affected structures, or at property lines, may be desirable. Substation noise includes transformer noise which may be barely audible in the presence of relatively high community noise levels. Therefore, background noise levels should be measured in the vicinity of the substation in order to remove the contribution of substation noise from the background measurement. Local or regional community noise regulations and criteria should be reviewed and measurement locations adjusted to allow direct comparison of measured levels with criteria and regulations.

4.4.4 Microphone Calibration

Calibration of measurement microphones should be performed prior to, and after, all measurements. If the measurements extend over a long period of time, calibration of the microphones should be performed approximately every 4 hours. Where extended surveys, e.g., 24-hour surveys, are being performed, or where repeated calibration is infeasible because of the nature of the measurement, sufficiently stable microphones and instrumentation should be used to avoid excessive error caused by drift. For these types of extended measurements, the calibration should be performed at the beginning and the end of the measurement period and the average calibration between the two should be used, unless the drift is from an obvious malfunction. Discrepancies between beginning and ending calibrations on the order of 1/2 to 1 dB are not unusual.

When data are recorded on magnetic tape, a calibration should be recorded on the beginning of each reel of magnetic tape because of the variability of the sensitivity of magnetic tape from roll to roll. Some measurement instrumentation provides internal calibration tones for this purpose so that a complete calibration with a piston phone or acoustic calibrator is not required for each roll of magnetic tape.

Calibration of microphones in the field should be performed with a piston phone or acoustic calibrator and with the protective grid supplied with the microphone in place. After calibration, a random incidence corrector and/or wind screen may be placed on the microphone. Calibration should always be performed with the instruments set to measure the overall sound level. Attenuator settings of the instrumentation used during calibration should be documented and variable gain knobs should be secured before recording the calibration signal. For each calibration tone recorded on magnetic tape, the attenuator settings together with the sound pressure level corresponding with the calibration tone should be announced on the recording.

4.4.5 Subway Pressure Transient Interference with Noise Measurements

Train motion within subways will cause air pressure transient waves with magnitudes relatively high compared with those of sound. The waves are generally at frequencies below 10 Hz. Since the pressure waves can cause momentary overload blocking of sound measurement amplifiers, special precautions are sometimes required when recording sound inside subways. Some noise measurement and recording apparatus contain high-pass wind filters which are effective at reducing the low-frequency signal and preventing blocking. If a high-pass wind filter is not available, the C-weighting of a sound level meter may prevent the blocking. Do not use C-weighting if the frequencies below 50 Hz are important to the analysis. In some cases the only solution to the blocking problem is to construct a custom high-pass filter that can be used as an external filter to the sound level meter or tape recorder, or as a filter in series with the measurement apparatus. Of course, the high-pass filter must be placed in the system prior to the amplifier that is blocking if it is to be effective.

The pressure transients within subways usually are of relatively short duration so that noise levels within the subway or on the vehicle may often be measured in spite of the periodic dropouts of the sound signal by pressure transients.

4.4.6 Tape Recording

A magnetic tape recorder is an important part of any data acquisition system. Good quality recordings of acoustic data are an important form of documentation. However, if appropriate precautions are not taken during the recording process, the data will be subject to error, degradation, or, without adequate documentation of field attenuator settings and amplifier configurations, may be rendered useless.

Generally speaking, when noise data are recorded on magnetic tape in direct mode, the maximum practical recording level should be used, as indicated by the tape recorder's VU meter or peak meter. The "zero VU" or maximum signal point on the meter represents the maximum signal level which may be recorded without significant distortion. In practice, the signal gain may be adjusted in 10 dB steps to within 10 dB below zero VU. Insufficient gain and signal level may result in inadequate signal-to-noise ratio, while gain which produces a signal level in excess of Zero VU may produce excessive distortion. These general considerations hold for noise or vibration data which has most of its energy at frequencies below 2000 to 3000 Hz. However, when measuring mechanical vibration or industrial noise, where very high crest factors may be encountered, and with significant high-frequency energy, the recording level should be reduced to approximately 10 dB below those levels recommended above if the tape recorder has NAB or

CCIR equalization (Audio recorders almost always incorporate such equalization). Instrumentation recorders meeting IRIG specifications do not employ pre-emphasis during the record process. The result is that the maximum recording level may be used regardless of frequency content.

Calibration tones should be recorded on every reel of magnetic tape used for direct recording to account for variation of tape sensitivity from reel to reel. The calibration tone should be at least 10 seconds long to facilitate calibration in the lab and should be recorded at a level within 10 dB below zero VU. The attenuator settings and serial numbers of each instrumentation component and their inter-relationships should be well-documented, either verbally on tape or in writing on a test log form or, preferably, both.

The best form of calibration is by acoustic excitation of the microphone with a known sound pressure. Using such a calibration, all instruments can be adjusted to accurately measure sound level. Reproduction of this calibration signal in the laboratory can then be used to calibrate the analysis instrumentation with very little chance for error.

If possible, verbal announcements should be recorded on a second or third channel to avoid interference with the data. The announcements should be clear and concise but with sufficient detail to identify the measurement location and the gain setting used for each sample of data. Repetitive announcements should be avoided. Where detailed descriptions of the measurement site and measurement instrumentation are required, the description should be written rather than announced on tape.

When individual samples of noise data are taken, the samples should be long enough to allow an accurate indication of the level and frequency analysis in the laboratory. Generally speaking, samples of continuous noise should be a minimum of 5 to 10 seconds. Where varying noise levels are encountered, the sample length should be significantly longer, perhaps as long as several minutes. When community noise levels are recorded for later statistical analysis within the laboratory, the sample should be at least 10 minutes or longer.

Noise measurements in adverse environments are often necessary. Microphones should be provided with wind screens to reduce the buffeting of the microphone. Buffeting produces low-frequency noise which can cause false data and can cause overload blocking of the instrumentation and tape recorder. Some tape recorders are particularly susceptible to wind noise signals; therefore a special high-pass wind filter may be necessary for recording noise data with relatively high gain. A high-pass wind filter is often included with magnetic recorders designed specifically for noise measurements. The frequency response of the high-pass wind filter should be flat within ± 1 or 2 dB down to about 30 to 50 Hz, with a 3 dB down point at 20 Hz or lower. The use of the high-pass wind filter of this type is also effective at controlling the effects of pressure transients within subway tunnels.

During the recording process, earphones should be used to monitor the signal quality of not only the signal being sent to the tape recorder but that which is recorded on the tape. Most tape recorders have a provision for monitoring the recorded signal during the recording process. Failure to monitor the data during the recording process can result in failure to identify errors with the result being poorly recorded data which is of little or no use. The earphones should have very wide frequency response and should be of rugged construction to guarantee reliability in the field.

No weighting networks should be used during the recording process. This allows analyzing the data in a variety of ways in the laboratory.

Note that FM recording, which is sometimes used for acoustic measurements, is discussed in the next section.

4.5 VIBRATION DATA ACQUISITION

In contrast to acoustic data acquisition, the acquisition of vibration data can be more difficult in that mounting of the transducer plays a significant role in the frequency response of the measurement system. Secondly, appropriate measurement locations are not evident because in many cases the vibration can not be heard or felt. In the case of groundborne vibration, unseen subsurface conditions such as layered soil or a perched water table may affect the measurement. Thus, experience in performing vibration measurements is an asset which must be developed at every opportunity. The purpose of this section, however, is to remove some of the mystery from vibration measurements and assure a reasonable chance for success for the novice. Indeed, with suitable attachments to a sound level meter, reliable vibration data may be obtained using some of the techniques outlined above regarding noise measurement. There is no good reason for avoiding vibration measurements simply because they involve vibration transducers instead of microphones.

Very good references are available which describe methods of vibration measurement (see for example Refs. 4.1, 4.3, 4.10). In addition, transducer manufacturers are usually very happy to supply literature and application notes concerning the proper use of accelerometers, amplifiers, and accessories. Indeed, several publish a monthly or quarterly periodical which can be valuable.

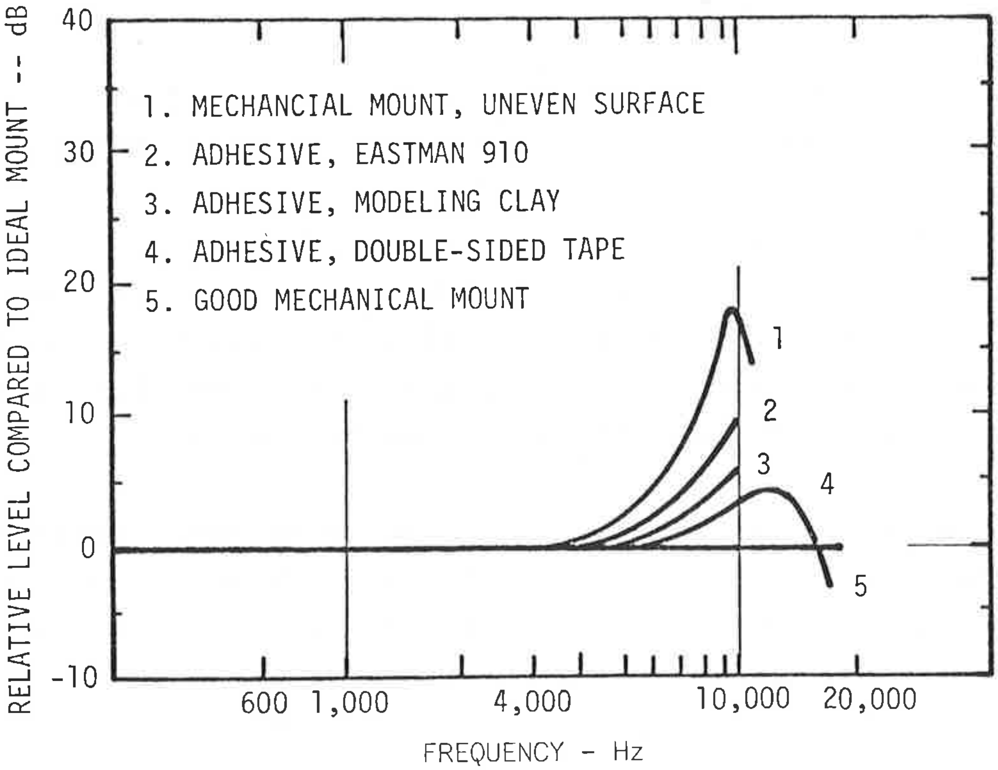
4.5.1 Transducer Mounting

Mounting techniques for accelerometers or velocity transducers are very important with respect to the overall frequency response of the measurement system. Generally speaking, to guarantee a flat frequency response in the desired passband, the mounted resonance frequency of the accelerometer transducer should be about 3 to 5 times higher than the maximum frequency of interest. Figure 4.11 shows the effect of various mounting techniques on the frequency response of accelerometers. Note that mounting an accelerometer on an uneven or dirty surface will greatly reduce the resonance frequency, while use of modeling clay or scientific wax is usually adequate on a smooth, flat surface.

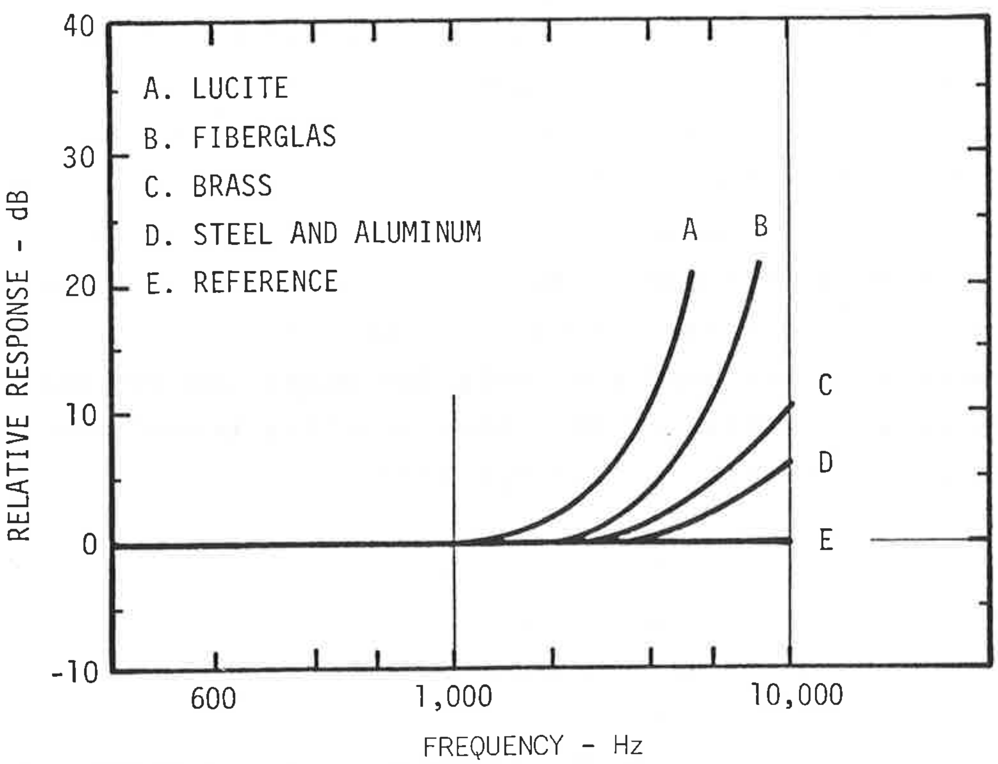
Where very high frequency acceleration data are to be measured, the mounting base must be as rigid as possible, e.g., the accelerometer should be mounted using studs on a machined, polished, and slightly oiled flat surface, or perhaps using a special boss welded or cemented to the surface under test. Special purpose cements such as Eastman 910 or dental cement are often adequate. When studs are used to mount an accelerometer, care should be exercised to prevent overstressing or torquing the accelerometer base which might damage the transducer. For less demanding measurements, scientific wax (beeswax) provides a practical and rapid means of mounting the transducer. Magnet mounts are available which allow attaching transducers to ferrous materials. However, with magnet mounts, the mounted resonance frequency is always greatly reduced.

Ground surface vibration is most commonly measured with an accelerometer adhered to sidewalks or asphalt roadbeds with scientific wax. Sidewalks or asphalt surfaces also provide convenient surfaces for rapidly setting up the field measurement apparatus and performing measurements. A disadvantage is that the paved surface may influence the measured vibration. However, because of the intimate contact of paved surfaces with underlying soil surfaces, the influence of the pavement can generally be assumed to be negligible over the frequency range of interest.

Another common technique for measuring ground surface vibration is to mount the accelerometer on stakes or spikes driven into the soil. The spike should be designed such that the surface area of contact with the soil is maximized and the weight minimized to



MOUNT RESONANCES, 30-GRAM ACCELEROMETER EXCITED AT 1 g



EFFECT OF ADAPTOR BLOCK MATERIALS FOR ACCELEROMETER MOUNTS, 28-GRAM ACCELEROMETER

FIGURE 4.11 EFFECT OF TRANSDUCER MOUNTING [FROM BERANEK, REF. 4.3] avoid decoupling of the transducer at high frequencies. The relative responses and designs of various spikes are discussed by Nolle (Ref. 4.21), who found that aluminum spikes roughly 30 cm long driven "without bounce" into soil provide good response characteristics up to about 200 Hz or higher for most soils. A slight taper in the spike profile is advisable to provide maximum contact with the soil. Good horizontal coupling is also achieved provided that the spike is driven to the hilt.

An alternative to the use of a spike or paved surface is an aluminum disk approximately 15 cm in diameter and 1 to 2 cm thick, which is placed directly on undisturbed, exposed, wet soil to maximize contact (Ref. 4.22). A moderate amount of backfilling of the soil area around the base is advisable. This method is not widely used because there is a good chance of obtaining poor contact with the soil. An alternative is to place the accelerometer on a base of plaster of Paris that has been poured on dampened, undisturbed soil. A 2 to 3 cm diameter circular metal disk may be embedded in the plaster of Paris to provide a mounting surface for the accelerometer. This type of base preparation has the advantage over the 15 cm diameter aluminum disc in that the pouring of the plaster of Paris should improve the contact with the underlying soil. Verhas (Ref. 4.23) has reported a response bandwidth of about 400 Hz using this technique. As generally known, neither of these two mounting techniques are used routinely for measuring vibration at the soil surface. Several of the above mounting techniques have been compared by Gutowski and Dym (Ref. 4.24).

A method often employed for the placement of geophones is to mount the transducer inside a metal container such that the overall density is comparable with that of soil. This combination is then buried. Provided that the wavelength of ground vibration is large relative to the dimensions of the container, very good frequency response characteristics should be achieved. However, the effects of disturbing the soil layer should be considered, as well as the depth of the transducer below the soil surface.

Complications can arise with the transducer connector cables so that precaution must be taken to guard against moisture and dirt intrusion. Because of the very low output impedance of geophones, they are less prone to problems from dirt and moisture and are a good choice for burying.

Often wet concrete surfaces will be encountered when searching for suitable transducer mounting surfaces. This is particularly common in subway tunnels that are below the water table. However, the moisture may be easily dried with a propane torch. A small (2 to 3 cm diameter) metal disk may be epoxied to the concrete surface to provide a convenient mounting surface for magnets, wax, or threaded studs.

When using wax for mounting an accelerometer, adequate holding strength will not be obtained if the mounting surface is dirty or grainy, such as a dirty sidewalk or subway wall surface. Cleaning the area thoroughly with a stiff bristle brush and then repeatedly applying and removing adhesive tape to the surface will often solve the problem. For very cold concrete surfaces, moderately warming the mounting surface with a propane torch will greatly ease the use of wax. However, excessive warming will contribute to pyroelectric noise if measurements are performed immediately (see Appendix D).

Insulating studs should always be used when accelerometers are mounted on electrically active surfaces, such as transit rails. An insulating stud electrically isolates the accelerometer body and reduces electrical interference caused by train operations. Insulating studs typically reduce the mounted resonance frequency of the accelerometer by approximately 15 percent.

4.5.2 Measurement Locations

4.5.2.1 <u>Subway Structures</u> -- A number of vibration measurement locations have been used in subway structures. Some of the more common locations are the safety walk or bench (Ref. 4.25), the invert (Ref. 4.26), and the subway wall (Ref. 4.27). Normally, only the vertical component of safety walk or invert vibration and the horizontal component of wall vibration are measured. Additionally, a vertical wall vibration measurement may be desirable because vertical motion of the subway wall may be particularly efficient at radiating shear waves. Also, the lateral component of invert vibration is sometimes measured for similar reasons.

The Union Internationale Transports Publique (UITP) (Ref. 4.28) has developed a tentative test code for the measurement of subway structure vibration. Basically, the test code specifies that transverse and vertical vibration velocity should be measured at the middle of the invert between crossties or fasteners and at the wall 1.2 m above the top of rail. For comprehensive testing additional transducer arrays should be located 20 m to either side of the above array. Note that center of invert locations are not possible for floating slab installations. Accelerometers may be used with analog integrators to obtain the vibration velocity data. Whenever an integrator is used, the frequency response should be checked to ensure that the system has proper low-frequency response.

4.5.2.2 Aerial Structures -- For characterization of ground-borne vibration from aerial structures a measurement of vertical vibration should be made on the foundation of the aerial structure and at the ground surface 7.5 m and 15 m from the structure. Vibration velocity or vibration acceleration should be measured on steel support girders and panels and on-deck surfaces for comparison with airborne noise levels, in which case the axes of sensitivity should be perpendicular to the radiating surface area.

4.5.2.3 Ground Surface Vibration -- The choice of measurement positions for ground surface vibration is usually predicated on the availability of open areas. Typical locations are 7.5, 15, and 30 m from the near track center line and, for subways, directly over the near track centerline. Normally, only the vertical vibration need be measured since vertical vibration correlates best with building noise caused by groundborne vibration. If horizontal vibration is measured, the axis of sensitivity should be perpendicular to the track. Triaxial measurements are almost never used (Refs. 4.25, 4.26).

4.5.2.4 Groundborne Vibration in Buildings -- Measurement locations on or in building structures usually are on the foundation, walls, floors and possibly ceilings of the structure. Vertical vibration of the foundation is usually measured, however, the transverse component may also be important. Whenever possible, measurements on the walls, floors and ceilings should be at the center of these panels. In most cases, only the vibration perpendicular to the surfaces is measured (Ref. 4.26). Correlation of lateral vibration with perceptibility and noise generated has been poor.

4.5.2.5 Equipment or Component Vibration -- Measurement of vibration of vehicle truck-mounted equipment should be performed at the mounting brackets of the equipment. Triaxial measurements are often desirable. The mass of the mounting bracket or flange should be large in comparison with the vibration transducers.

4.5.2.6 Vehicle Floor Vibration -- Measurements of vehicle floor vibration for the purpose of evaluating patron comfort should be under the seat and in the aisle at locations over the trucks and in the central area of the vehicle. Measurement locations should also be included over air-conditioning and other items of auxiliary equipment. Only the vertical orientation of vibration need be measured (Ref. 4.29). The mass of the accelerometer and any mounting table or bracket should be small compared with the effective mass of the floor panel.

4.5.3 Accelerometer Calibration

An overall calibration of the vibration measurement system should be performed prior to or following a series of tests not only to calibrate the system but to guarantee the proper performance of the measurement system components.

Accelerometers can be calibrated with small, portable, battery-operated exciters specifically designed for calibration of accelerometers in field situations. These exciters often include an adjustment to account for variation in accelerometer mass. Calibration frequencies are usually between 50 and 120 Hz, with acceleration amplitudes of about 1 g. As always, the operator's manual should be reviewed prior to use.

When accelerometer voltage preamplifiers are used, the accelerometer cable used for the data acquisition should be in place during calibration. However, with charge amplifiers, cables of various lengths can normally be substituted without change of sensitivity.

4.5.4 Direct Recording of Vibration Data

There are two general methods of recording vibration data on magnetic tape. The first technique is simply direct recording as is usually used for sound data. The second is frequency modulation recording, or FM recording. A significant disadvantage of direct recording is that the low-frequency response is usually limited to 10 to 20 Hz. With FM recording, response down to 0 Hz can be achieved.

The technique for direct recording of vibration data is basically the same as that used for recording acoustic data. However, where equipment acceleration (e.g., truck vibration) is being recorded, much of the vibration energy may be concentrated at high frequencies, in which case the recorder equalization will cause clipping of the record amplifier. This is particularly important when one considers that an accelerometer resonance is usually above 10 kHz and of very high Q; the resonant amplification can be 20 or 30 dB. This is less of a problem when recording direct with instrumentation recorders meeting IRIG specifications, which do not employ pre-emphasis of the recorded data.

4.5.5 Frequency Modulation (FM) Recording

FM recording is used where a particularly flat frequency response or an extended low- or sub-audio frequency response is required. Generally, groundborne vibration data should always be recorded in the FM mode. In fact, most vibration measurements should use FM recording, unless high-frequency data is of interest.

Most FM recorders are instrumentation recorders meeting IRIG specifications. They typically have four, seven, or fourteen channels. However, portable, battery-operated FM modulators and demodulators are available which may be used to record low-frequency signals (down to zero Hz) on standard audio tape recorders. This latter configuration is particularly useful for field data acquisition where portability is a great asset.

When recording low-frequency vibration in the FM mode for a single event, at least 10 seconds and preferably 20 seconds, of recording should precede the event, especially if the data are to be frequency analyzed. The reason for this is that starting any FM recording causes a pulse or voltage shift during reproduction which excites the analysis filters. For narrowband analysis with bandwidths on the order of 1 Hz, the settling time is several seconds.

During FM recording, the peak signal amplitude should be kept high to maximize the signal-to-noise ratio. In contrast to direct recording, however, little or no headroom or crest factor capability is available with FM recording. Thus, a peak detector or peak reading meter should always be used to monitor the signal amplitude. Some FM instruments use a VU meter for signal monitoring. In this case, the maximum level indication of the VU meter should be no higher than zero VU. The operator should be thoroughly acquainted with the recording characteristics of this equipment and the nature of the data to be able to judge confidently whether or not the FM modulator is being overdriven.

Finally, a calibration signal should be recorded at least once for each instrumentation lash-up and channel. However, when FM recording is used, the tape sensitivity does not affect the overall calibration; thus, recording a single calibration signal is sufficient even though a number of tapes may be recorded. Multiple recordings of calibration signals are still a good practice whenever possible. This will document that instrument sensitivity has not changed and will increase confidence in the accuracy of the data.

4.5.6 Electrical Interference

Electrical interference is a commonly occurring problem in field and laboratory measurement situations. Sources of electrical interference include RF interference from radio and television transmitters, power lines, and collector shoe arcing. If adequate precautions are taken, electrical interference may be reduced to negligible or nonexistent levels.

The major cause of electrical interference is the presence of a ground current loop in the instrumentation setup as illustrated in Figure 4.12. Two ground planes are indicated between which an electrical potential is produced by perhaps a transmitter or collector shoe. The electrical potential produces a current in resistors R_I, R_B, and R_{C2}. The current through resistor R_{C2} produces a voltage between the inputs of the amplifier and is not cancelled by a similar current through R_{C1}. In some situations, a balanced line, as used with dynamic microphones, may be employed to produce equal currents through R_{C1} and R_{C2}, thereby producing little or no net voltage difference between the inputs of the amplifier.

Use of balanced lines is impractical for most measurement situations where only coaxial shielded cable (RG 58) is used with common "BNC" connectors. In these situations, the resistance R_I or R_B should be made as large as possible to reduce the current through R_{C2}. Making R_I or R_B as large as possible is simply to provide insulation between the transducer and measurement surface or between the amplifier and the power or ground plane.

The simplest procedure is to completely isolate the transducer from the mounting surface. In the case of accelerometers, this may be easily achieved with insulating studs and insulated preamplifiers. Accelerometers may also be isolated with dental cement or use of mycolex between the base and measurement surface. The preamps may be isolated from the ground plane 1 by use of paper or cardboard, plastic tape, or sponge rubber. In all cases, jacketed cables should be standard. Some accelerometers are available with built-in electrical insulation. Those units should be selected which minimize the coupling capacity between the piezoelectric element and the mounting surface (Ref. 4.30). When multichannel instrumentation setups are used, the most satisfactory configuration is to provide a single ground at the common main frame which may be a multichannel tape recorder or multichannel amplifier. The transducers, whether microphone or accelerometer, should be electrically isolated from their mounting or supporting surfaces, so that the only common ground is that provided at the main instrumentation (Ref. 4.31). If the measurement circuits cannot be isolated from all but one ground plane, then differential input amplifiers may have to be used (Ref. 4.3).

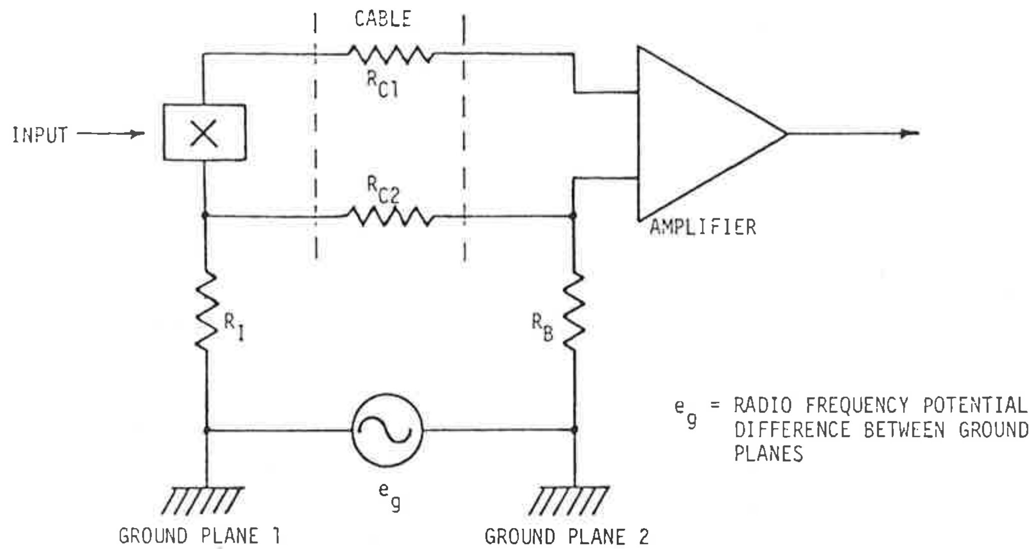


FIGURE 4.12 GROUND CURRENT LOOP -- ELECTRICAL INTERFERENCE

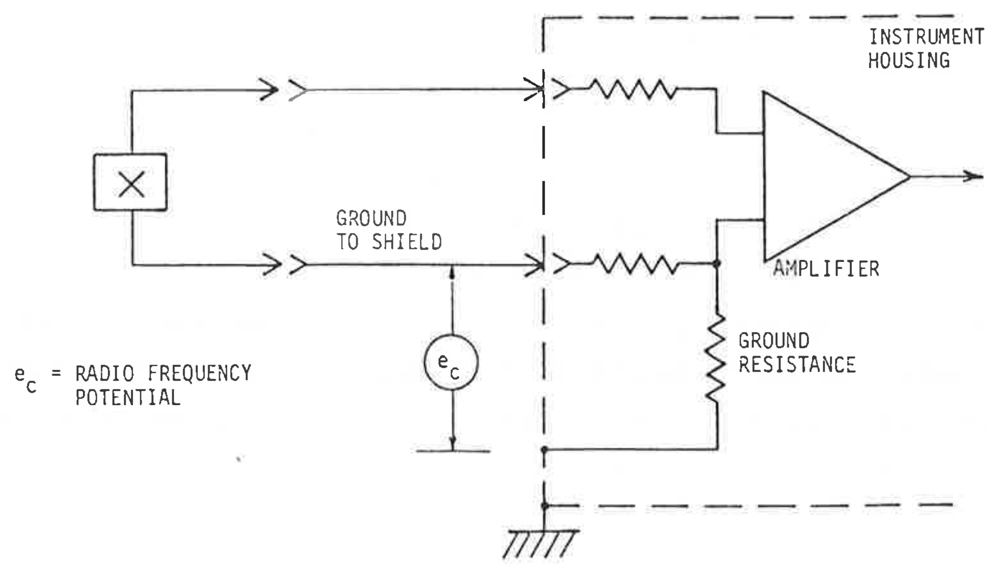


FIGURE 4.13 ELECTRICAL INTERFERENCE DUE TO FINITE RESISTANCE BETWEEN CABLE SHIELD AND CHASSIS

Often, the ground, shield, and transducer housing form one end of an antenna, while the amplifier or perhaps tape recorder forms another end, with a resulting radio frequency potential developing across the input connector resistance. Thus, all connector contacts should be clean and free of oxidation. Connectors with gold-plated contacts are definitely very desirable, since the gold is not susceptible to oxidation and may be cleaned easily. Trichloroethane, alcohol, or distilled water should be used to remove contaminants and grease from connector contact surfaces. Care should be taken that the solvent does not harm connector materials.

One of the possible reasons for electrical interference may be the amplifier design. If the signal ground of the cable is not grounded to the chassis at the input connector, internal resistance of solder joints and conductors can produce radio frequency voltage drops, which are rectified and amplified by the amplifier, as illustrated in Figure 4.13. This problem can often be remedied by grounding the cable shield and signal ground directly to the chassis or rewiring the input connector of the instrument so that the signal and shield grounds are grounded to the chassis <u>at the input connector</u>.

Another form of electrical interference is radio frequency current in the cable shielding, which is most common in very long cables. In general, connecting cables should be maintained as short as possible, and coiling of excess cable should be avoided. If cables must be coiled, the excess cable should be placed in a vehicle or other convenient metallic enclosure to provide shielding from radio frequency radiation from nearby radio or television transmitters.

If very long cables are necessary, a 2-conductor shielded cable or even a fully balanced line with a differential amplifier as a receiver, both with interrupted shields may be necessary, as illustrated in Figure 4.14. The essential idea is to interrupt the cable shield at perhaps 50- or 100-foot intervals, with the shield of each cable section grounded to the ground line at only one end of the cable, e.g., the output connector. All conductors, signals, and grounds are protected by the shield. Note that the transducer housing is grounded not to the shield but to the ground wire in both cases. 3-conductor balanced lines may be most effective with a geophone or velocity pickups, which have very low output impedance. A differential input amplifier works best with the balanced line approach.

Other techniques for reducing electrical interference include raising cables up off the ground by a foot or so. For rapid transit wayside measurements, stray return currents in the ground are common and can induce stray currents in the cabling. This is particularly noticeable with systems using chopper-controlled traction motors.

4.6 LABORATORY AND ANALYSIS PROCEDURES

4.6.1 Calibration of Analysis Systems

An analysis system usually consists of a signal-conditioning amplifier, filters, an analyzer, a level recorder, and possibly other display devices. Prior to analysis, the instruments should be calibrated for the same full scales. This is done by applying an oscillator signal to the input terminal of the main amplifier and adjusting the various gains so that the full-scale sensitivity of all components coincide. At this stage, the overall linearity, signal-to-noise ratio, and frequency response of the analysis system can be easily verified. The main amplifier should be a precision sound level meter or measuring amplifier, with a high quality indicating meter of 1% resolution and a gain adjustment for overall system calibration. The system must then be calibrated for the actual data by connecting the transducers and preamplifiers and by using a microphone calibrator or vibration

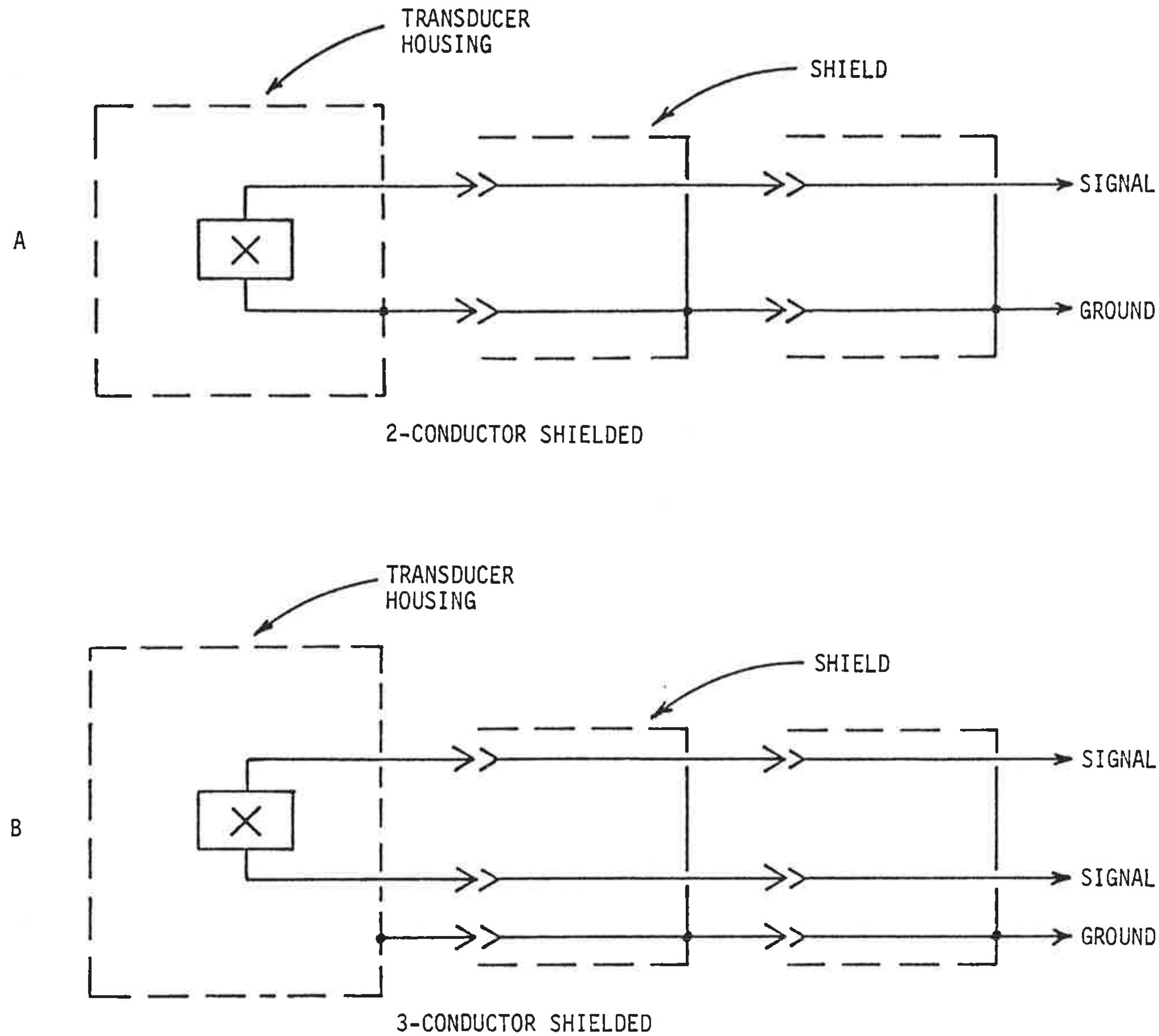


FIGURE 4.14 CABLES WITH INTERRUPTED SHIELDING FOR MINIMIZING GROUND LOOP AND NOISE PICKUP

transducer exciter.

The calibration tones recorded on tape should be used for setting up the system. Calibration is achieved by adjusting the gain on the main amplifier, using the meter of the main amplifier. The main amplifier should be set to "overall," and the output attenuator should normally be set to minimum gain. If an external filter is used with the measuring amplifier, it should normally be switched in after calibration, unless it causes insertion loss. Immediately after calibration, all attenuator settings should be written and variable gain verniers secured to avoid accidentally changing the gain settings.

4.6.2 Tabulation of Noise and Vibration Levels

The simplest type of analysis, which may be accomplished with a minimum of instrumentation, is the observation and tabulation of noise and vibration levels. Tabulation in either the field or the laboratory can be appropriate for surveys of occupational noise levels, maximum A-weighted noise levels for single events, such as train passbys, and the documentation of steady-state noise from fan shafts, substations, or other equipment.

The tabulation format varies according to the problem under consideration. Additional documentation should include such data as:

- Date and Time of Day
- Site Description
- Measurement Locations and Distances
- Serial Numbers and Types of Equipment Under Test
- Operating Conditions
- Block Diagrams of Instrumentation (including make, model number, and serial numbers)
- Weightings Used
- Gain Settings

Documentation should be sufficient to allow for duplication of the test.

4.6.3 Graphic Display of Noise and Vibration Levels

Graphic recording of noise and vibration levels, as a function of time, is the next step in the order of complexity. Such graphic records are normally produced on a graphic level recorder, connected to a sound level meter or other source. The display of the temporal variation of noise and vibration levels allows a visual assessment of the character of single-event noise or vibration, which may be evaluated in terms of maximum level, surrounding background level, and the period of time over which the single-event occurs. The time integral of intensity may also be evaluated by graphic means, which allows the calculation of such energy parameters as NEL or SENEL. Graphic record of overall or weighted noise and vibration is useful for documenting noise and vibration data recorded on magnetic tape. A graphic display of noise and vibration levels also allows the identification of specific events, which might otherwise go unnoticed. For example, groundborne vibration from subway train operation may be readily recognized and distinguished from the background vibration by the maximum level and by the shape of the noise and vibration signature.

Successful use of a graphic level recorder requires knowledge of the recorder characteristics and requires care in setting the various dynamic parameters. The graphic level recorder instruction manuals include discussion of pen writing speed and detector time constants and give guidelines for selecting appropriate settings. There are, however, some specific operating characteristics that are appropriate for most rapid transit-related measurements. Basically, the recorder overall time constant and pen damping should be set to give the fastest pen response possible without significant overshoot, i.e. not more than 1 dB overshoot in response to a step input.

Specifically, for transit system measurements, the detector lower limiting frequency (equivalent to the detector time constant) should be set to 20 Hz for most data. Some workers have indicated that it should be set to 20 Hz for noise data and 2 Hz to 10 Hz for vibration data. However, it is important to recognize that the frequency responses of the three settings are essentially the same: the 10 Hz and 20 Hz settings have flat frequency response down to 3 Hz. The different switch positions actually refer to the averaging time of the detector rather than to the frequency response for the signal being plotted. With the lower limiting frequency set, the damping, or writing speed, and resolution should be set at a combination that does not result in significant overshoot for a step input voltage that is sufficient to give a 10 to 30 dB deflection of the pen. The resolution is usually set to correspond with the full-scale range of the chart paper. However, a preferable setting is one step greater resolution than the chart full-scale range. Using the greater resolution position results in better response from the recorder.

For a Bruel and Kjaer Type 2305 level recorder, a type frequently used for plotting transit train passbys, with the lower limiting frequency set at 20 Hz, the resolution (potentiometer range control) at 40 dB, and writing speed at 200 mm/sec, the maximum pen response without overshoot occurs and the plotted results are the same or very close to those obtained with a digital integrator. Note that these settings also give readings similar to those of a sound level meter set to "fast" response (Ref. 4.13) but do <u>not</u> approximate the ballistics response of the meter. In fact, it is neither possible nor particularly desirable to approximate the sound level meter specification for either "fast" or "slow" response with a graphic level recorder.

There are special cases where the sound level meter type of response can be approximated over a limited range of operation of the level recorder. However, for most measurements these are not the optimum settings and should not be used, as has been recommended by some authors (Ref. 4.32). In general, the level recorder response as described above produces appropriate results and is more desirable than the sound level meter ballistics response, particularly for transient occurrences such as transit train passbys. Use of a damping setting to approximate "slow" response of a sound level meter can result in erroneous (low) readings when measuring a transit train passby because the pen (or the sound level meter indicator) cannot respond with sufficient speed and the event is over before the full level is achieved. With the recorder set for maximum pen response speed without overshoot, this does not happen and the readings obtained are consistent and repeatable for different durations of events and for different sound levels during the events. Such settings usually give a better representation of the measurements than any other type of setting for the level recorder; for example, the results will more closely correspond with digital meters which respond without any inertial effects.

Note that the averaging time, or time constant, of a level recorder does not necessarily have a direct relationship to the writing speed. When slower pen motion or greater damping is desired, care must be taken that the detector time constant is increased at the same time that the writing speed is decreased. Otherwise, the rms level indication may be lower than the actual level. In the case of the Bruel and Kjaer level recorder, longer averaging times, if desired, can be achieved by reducing the lower limiting frequency and then adjusting the writing speed for maximum speed without overshoot. Again, however, remember that either the 20 Hz or 10 Hz setting usually gives appropriate averaging time for transit system type events.

4.6.4 Archival Strip Chart Records of Tape Recorded Data

When noise and vibration data is recorded on magnetic tape, a visual strip chart record of the recorded data should be produced with a graphic level recorder. This continuous record allows the evaluation and identification of noise and vibration data and aids the process of locating specific examples for analysis. When producing the archival record, constant amplifier gain should be used throughout so that changes of field gain will be directly reflected in the record. All verbal announcements should be noted directly on the archival record, together with attenuator settings and dB scales or maximum level on the chart. The recorded calibration tone should be reproduced at the beginning of the tape without the use of any weighting network. An A-weighting network should be used for production of graphic records of community, environmental, or industrial noise to provide direct evaluation of recorded A-weighted noise levels. Graphically recorded vibration data should be unweighted vibration acceleration, or unweighted vibration velocity. However, an octave band filter may be useful for identifying specific events that would otherwise be masked by the background vibration.

4.6.5 Narrowband Analysis with a Tape Loop

Narrowband analyses can be performed with many graphic level recorders and compatible sweeping filters, with the aid of a continuous tape loop. Directions are given in the manuals provided with the instrumentation. However, certain practical points should be followed when performing the analysis. The construction of the tape loop is of particular importance because it directly determines the quality of the analysis. A good general rule is to use high quality studio splicing equipment and to demagnetize the razor blades used for cutting the tape. Use an oblique cut for slicing the magnetic tape for data recorded in the direct mode, and a perpendicular cut for FM recordings. The length of the tape loop should be approximately 10 seconds, but some experimentation may be required to find the best size tape loop for a specific example. Frequencies below the lower frequency range of the narrowband analyzer can be analyzed by running the tape recorder at double speed.

4.6.6 Octave Band Analysis

Octave band analyses may be performed with a level recorder, a sound level meter, octave band filters, and a tape recorder, as discussed in Section 4.2.2. After the system is calibrated, according to Section 4.6.1, reproduce the data once for each filter setting, displaying the filtered signal level on the level recorder. As many as two or three filter traces may be superimposed, using different colored inks for each octave band. For continuous records, or records with many individual samples, a continuous record of the tape recording should be made for each given octave band filter setting. The level recorder paper may be backed-up so that several octave band analyses are presented on the same strip chart record. The octave band levels are read from the chart, and the results tabulated or plotted. Relatively long averaging times and slow writing speeds for the low-frequency octave bands should be used to make the records readable and to accomplish much of the "rms-averaging," which would otherwise have to be done visually.

4.6.7 Real-Time 1/3 Octave Band Analysis

The operator's manual supplied with the analyzer should be reviewed. The manual provides detailed information regarding integration times and corresponding confidence limits as a function of frequency. Generally speaking, the longer the integration time, the more confident and reliable will be the analysis. This is particularly significant at frequencies of 20 Hz or less. A typical real-time 1/3 octave band analysis system is described in Section 4.2.3 and illustrated in Figure 4.8. The system is calibrated in the manner described above.

The analysis system and attenuator settings should be thoroughly documented with block diagrams and tables. Once a system lash-up has been established and verified to be practical and reliable, it should be standardized. There will, however, be exceptions, depending upon equipment availability. A real-time system can be very powerful, but it can also generate a great deal of useless information if not properly used and understood.

When using a graphic level recorder for monitoring and identifying data samples to be analyzed, use a high enough paper speed to determine the precise analysis period needed. Level recorder settings for detector and writing speed should be identical to those used for production of archival strip chart records of recorded data. They should use the same weighting network: e.g., A-weighting for acoustic data. The weighting network should not be in the signal path leading to the real-time analyzer, which should receive unweighted data. An extra sound level meter may be required for this purpose. It is usually best to keep the level recorder full-scale level setting the same as that of the main amplifier and analyzer.

The oscilloscope monitor should be adjusted using a sine wave signal so that the nominal full-scale signal voltage is represented by plus and minus one division, near the centerline of the screen. Plus and minus three divisions will then represent 10 dB over the full-scale signal voltage, the maximum crest factor capability of many real-time analyzers. Signal waveforms, with peaks in excess of plus and minus three divisions, will usually cause overloading of the analyzer detector and should be avoided. The operator's manual should be consulted to determine the actual crest factor capability, and the corresponding signal excursion voltage should be noted on the oscilloscope.

Listening to an audio monitor is one of the best methods of detecting poor quality data caused by electrical interference or extraneous noise or vibration sources. The audio monitor is indispensible, even for vibration data.

The real-time analyzer display is very useful for determining whether or not a particular analysis is successful or consistent with previous analyses. The display can often be used to directly compare two different spectra, for instance the present spectra with a background spectra.

Background spectra should be obtained as often as is necessary to ensure the reliability of the data and should usually be taken with the same gain, or higher, settings as used for the primary samples. Documenting the background level is often overlooked, but is important.

4.6.8 Presentation of Octave and 1/3 Octave Data

The ANSI Preferred Format reporting should be used for 1/3 octave band and octave band data (Ref. 4.5). In this format, 10 dB is represented by a vertical height of 2 cm, and each octave band center frequency range by a horizontal distance of 1.5 cm: each 1/3 octave band is thus separated by .5 cm from the neighboring 1/3 octave band. This format is used internationally and facilitates the comparison of different data. Other vertical scale factors are allowed in the ANSI Preferred Format, but should be avoided. They are not used extensively, and data with a typical measurement error of ±1 dB is most appropriately reported using the scale factor detailed above. Use of the standard format also permits easy comparison of data presented by different researchers.

4.6.9 Statistical Analysis of Noise and Vibration Data

The simplest and least time consuming method for statistical analysis of community noise data is to use a special purpose field analysis system, which may be placed in the field. Such systems provide transduction, amplification, A-weighting, and digital analysis. The results may be tabulated immediately upon completion of the analysis period, which may extend from a fraction of a minute to over 24 hours. The instrument may be reset immediately for the next sample or allowed to continue.

Some units provide printouts at set time intervals without operator intervention, and some even have digital cassette tape recorders for recording the actual noise level at intervals of a second or so. The digitally recorded data may then be analyzed in a variety of ways, using a tape reader and computer.

For overnight, 24-hour monitoring, the noise monitor system should be secured in a strong box with appropriate weatherproofing. The manufacturer should be consulted for precautions against very hot or cold weather, or high humidity. The unit may often be mounted high up on a convenient telephone pole; case-hardened chains and padlocks are always desirable. Good judgement must be exercised in placing the unit,and loss should be anticipated.

Very detailed statistical analysis of noise and vibration data can be accomplished by recording the noise and/or vibration data on tape, and then reproducing it in the laboratory and analyzing it with a level recorder interfaced to a computer. Alternatively, mechanical counters are available which, if attached to the level recorder, will perform these analyses. When a computer is used, a special attachment for the level recorder that converts the pen position to an analog DC voltage is needed. The DC voltage is then digitized by the computer for processing. However, many current measuring amplifier systems provide a DC voltage proportional to the log of the RMS signal amplitude, and this DC voltage may be digitized for analysis by the computer. If a real-time analyzer is interfaced to a computer, statistical level analysis may be performed by octave bands.

A resolution of about 1 dB is more than adequate for most statistical analyses. However, a mechanical counter will produce resolution of 2.5 to 5 dB. In this case, L_n must be estimated by interpolation. Experience has shown that the interpolation will usually give quite accurate results because noise and vibration levels are usually uniformly distributed over a significantly long time period.

Performing spot checks of community noise requires samples of about 10 to 20 minutes in length, and the interval should be 1 or more samples per second. When the measurements are made, idle conversation near the microphone with passers-by should be avoided. Children are irresistably tempted to talk into microphones. When community noise is recorded, the tape recorder can be momentarily halted to delete such extraneous noises. Interruption may not be possible with community noise analyzers unless a pause button is installed.

4.6.10 Reverberation Time

Reverberation times are discussed in Appendix A and Section 4.1.4. Some practical hints for making such measurements follow.

The instrumentation required for measuring reverberation time consists of an amplifier, a 1/3 octave band filter set, and a graphic level recorder. Reverberation time is calculated from the slope of the decay curve of sound level as a function of time, as displayed on the graphic level recorder. The instrumentation manuals supplied with graphic level recorders usually include a detailed discussion of reverberation time measurements. Substantial care should be exercised in the setting of the rms detector time constant, writing speed, and paper speed of the graphic level recorder. The writing speed must be sufficient to exceed the sound level decay rate, as demonstrated by random motions of the pens as the sound decays. A smooth, straight line decay usually indicates the writing speed is too slow.

The reverberation time needs to be analyzed for only each center 1/3 octave of each octave band of interest, which reduces the amount of data reduction required by a factor of 3. Tape recording the decay of the decaying sound field is particularly useful for reverberation time measurements; the decay may be reproduced repetitively, while advancing the 1/3 octave band filter set to the appropriate frequencies. When a tape recorder is employed for recording reverberation time signals, only two or three decays need be recorded for each microphone position.

4.7 STANDARDS AND TEST CODES

There are two recognized types of basic standards. The first type consists of measurement standards that describe the procedures for conducting measurements in a reliable and consistent manner, and the second type covers standards for instrumentation. A test code is a measurement procedure for evaluating the emissions of a specific noise or vibration source (Standards which discuss legal or recommended permissible limits for noise and vibration levels are not discussed in this section).

The term "mandatory standard" refers to those standards which must be met by order of a governmental agency or other body. A "voluntary standard" is one to which there is no legal obligation to comply. Standards promulgated by the American National Standards Institute are voluntary standards. However, voluntary standards may be enforced by legal regulation.

International standards are prepared by organizations consisting of various national standards organizations, such as the American National Standards Institute (ANSI). Examples of international organizations are the International Organization for Standardization (ISO), the International Electro-Technical Commission (IEC), the Commission of the European Committees (European Common Market), the International Organization of Legal Metrology (OINL), and the General Agreement for Tariffs and Trade (GATT). Standards for noise and vibration measurement instrumentation are the responsibility of IEC Technical Committee 29 (Electro-Acoustics) and its Sub-Committee 29C on measuring devices. Those international standards that include all aspects of noise, except instrumentation, are the responsibility of ISO Technical Committee 43, Acoustics, and its Sub-Committee 1 (Noise).

Of particular interest are a group of standards for transit system noise and vibration measurements that are in the process of being prepared by the UITP. At the present time a draft standard for the measurement of groundborne noise and vibration (Ref. 4.28) has been completed.

ANSI has developed standards both for the measurement of noise and vibration and for the instrumentation required for such measurement. The four committees responsible for the development of ANSI standards are:

- S1 (Acoustics)
- S2 (Mechanical Shock and Vibration)
- S3 (Bio-Acoustics)
- S4 (Sound Recording)

Committees S1, S2, and S3 are concerned primarily with fundamental noise and vibration topics, including the measurement and assessment of noise and vibration. These committees do not specify test codes for specific kinds of machinery or equipment.

Tables 4-1 through 4-3 list some of the standards and codes that are particularly applicable to rail transit noise and vibration. Table 4-1 lists basic standards for instrumentation used for noise and vibration measurements. The standards developed by ISO and ANSI for noise and vibration measurement procedures are listed in Table 4-2. Test codes of particular interest to rapid transit engineers are listed in Table 4-3. Note that more test codes can be obtained from promulgating organizations.

A test code describes a recommended procedure for measurement and evaluation of noise and vibration from a specific type of equipment, such as a transit vehicle or motor alternator set. These test codes, which are essentially voluntary, have been developed by a number of organizations.

Labeling of equipment for noise and vibration is required, by federal regulation, to specify the test code used to determine noise or vibration levels produced by such equipment. Such test codes must be agreed upon by all concerned parties: federal agencies, the manufacturer, specifier, installer, and the ultimate user of the equipment. Usually, a voluntary standard, or test code, is developed for the necessary testing. However, the federal government may specify a test code in the absence of agreement or consensus.

Listed in Table 4-4 are the names and addresses of various organizations responsible for developing the various standards and test codes listed in Tables 4-1, 4-2, and 4-3. More complete lists are given in References 4.1, 4.2, and 4.4.

TABLE 4-1 INSTRUMENTATION STANDARDS

American National Standards Institute - ANSI

S1.4-1971	Specification for Sound Level Meters
S1.11-1976	Specification for Octave, 1/2 Octave and 1/3 Octave Filter Sets
S1.12-1972	Specification for Laboratory Standard Microphones
S2.3-1970	High-Impact Shock Machine for Electronic Devices
S3.6	Specifications for Audiometers

International Electro-Technical Commission - IEC Recommendations

Publication 123	Recommendation for Sound Level Meters (1961)
Publication 179	Precision Sound Level Meters (1973)
Publication 179A	Additional Characteristics for the Measurement of Impulsive Sounds (1973)
Publication 225	Octave, 1/2 Octave, and 1/3 Octave Band Filters Intended for the Analysis of Sounds and Vibration (1966)
Publication 184	Specifying the Characteristics of Electromechanical Transducers for Shock and Vibration Measurements (1965)

TABLE 4-2 BASIC MEASUREMENT STANDARDS

American National Standards Institute - ANSI

S1.1-1960	Acoustical Quantities
S1.2-1976	Method for the Physical Measurement of Sound
S1.5-1963	Loudspeaker Measurements
S1.6-1976	Preferred Frequencies and Band Numbers for Acoustical Quantities
S1.7-1970	Method of Test for Sound Absorption of Acoustical Materials in Reverberation Rooms
S1.8-1974	Preferred Reference Quantities for Acoustical Levels
S1.10-1976	Method for the Calibration of Microphones
S1.13-1976	Methods for the Measurement of Sound Pressure Levels
S2.2-1959	Calibration of Shock and Vibration Pickups
S2.10-1971	Analysis and Presentation of Shock and Vibration Data
S2.11-1969	Calibration and Tests for Electrical Transducers Used for Measuring Shock and Vibration
S3.4-1968	Computation of the Loudness of Noise
S3.5-1969	Calculation of the Articulation Index
Z24.21-1957	Specifying the Characteristics of Pickups for Shock and Vibration Measurements
Z24.24-1957	Calibration of Electroacoustical Transducers

International Standards Organization - ISO Recommendations

R131-1959	Expression of the Physical and Subjective Magnitudes of Sound or Noise
R140-1960	Field and Laboratory Measurements of Airborne and Impact Sound Transmission

TABLE 4-2 BASIC MEASUREMENT STANDARDS (continued)

ISO Recommendations (Continued)

R266-1962	Preferred Frequencies for Acoustical Measurements
R357-1963	Expression of the Power and Intensity Levels of Sound or Noise
R495-1966	Preparation of Test Codes for Measuring the Noise Emitted by Machines
R532-1966	Method for Calibrating Loudness Level
R717-1968	Rating of Sound Insulation for Dwellings
R1996-1971	Assessment of Noise with Respect to Community Response
R1999-1971	Assessment of Occupational Noise Exposure for Hearing Conservation Purposes
R2204-1973	Guide to the Measurement of Acoustical Noise and Evaluation of its Effect on Man
R3740-1978	Determination of Sound Power Levels of Noise Sources - Guidelines for Use of Basic Standards and Preparation of Noise Test Codes
R3744	Determination of Sound Power Levels of Noise Sources - Engineering Methods for Free-Field Conditions over a Reflecting Plane
R3746	Determination of Sound Power Levels of Noise Sources - Survey Method
R3747	Determination of Sound Power Levels of Noise Sources - Methods Using a Reference Sound Source

TABLE 4-3 TEST CODES

ISO/R362	Measurement of Noise Emitted by Vehicles
ISO/R495	General Requirements for the Preparation of Test Codes for Measuring the Noise Emitted by Machines
ISO 3085-1975	Acoustics - Measurements of Noise Emitted by Rail Bound Vehicles
ISO 3381-1976	Acoustics - Measurements of Noise Inside Rail Bound Vehicles
ISO R1680-1970	Test Code for the Measurement of the Airborne Noise Emitted by Rotating Electrical Machinery
ANSI S5.1-1971	Test Code for the Measurement of Sound from Pneumatic Equipment
ASHRAE Standard 36-72	Methods of Testing for Sound Rating Heating, Refrigerating, and Air-Conditioning Equipment
IEEE 85-1973	Test Procedure for Airborne Sound Measurements on Rotating Electric Machinery
ARI Standard 270-1967	Standard for Sound Rating of Outdoor Unitary Equipment
AGMA Standard 293.03-1968	Specification for Measurement of Sound on High Speed Helical and Herringboen Gear Units
AFBMA Standard No. 13-1968	Rolling Bearing Vibration and Noise
NEMA Standard MG1-12.49-1972	Motors and Generators. Methods of Measuring Machine Noise
NEMA Standard TR1-1972	Transformers, Regulators, and Reactors (Section 9-04, Audible Sound Level Tests)
ASTM E90-75	Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions
ASTM E336-71	Measurement of Airborne Sound Insulation in Buildings

TABLE 4-4 STANDARDS AND TEST CODE ORGANIZATIONS

Acoustical Society of America (ASA)

Standards Secretariate
335 East 45th Street
New York, New York 10017

Air-Conditioning and Refrigeration Institute (ARI)

1815 North Fort Meyer Drive
Arlington, Virginia 22209

American Gear Manufacturers Association (AGMA)

1 Thomas Circle
Washington, D.C. 20005

American National Standards Institute (ANSI)

1430 Broadway
New York, New York 10018

American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE)

345 East 47th Street
New York, New York 10017

American Society for Testing and Materials (ASTM)

1916 Race Street
Philadelphia, Pennsylvania 19103

Antifriction Bearing Manufacturers Association, Inc. (AFBMA)

60 East 42nd Street
New York, New York 10017

Institute of Electrical and Electronic Engineers (IEEE)

345 East 47th Street
New York, New York 10017

TABLE 4-4 STANDARDS AND TEST CODE ORGANIZATIONS (continued)

National Electrical Manufacturers Association (NEMA)

2101 L Street, N.W.
Washington, D.C. 20037

Society of Automotive Engineers (SAE)

400 Commonwealth Drive
Warrendale, Pennsylvania 15096

International Organization for Standardization (ISO)

1, Rue de Varembe
1211 Geneva 20
Switzerland

International Electrotechnical Commission (IEC)

1, Rue de Varembe
1211 Geneva 20
Switzerland

CHAPTER 4 REFERENCES

4.1 C. M. Harris, C. E. Crede (Editor), Shock and Vibration Handbook, Second Edition, (McGraw-Hill Book Publishing Company, NY, 1976).

A general handbook on shock and vibration. Included are detailed discussions of various types and proper use of instrumentation for shock and vibration measurements.Other topics include measurement techniques, testing and data reduction, shock and vibration data analysis, and environmental specifications and testing.

4.2 C. M. Harris (Editor), Handbook of Noise Control, Second Edition, (McGraw-Hill Book Publishing Company, NY, 1979).

This is a valuable reference concerning the measurement, analysis, and control of noise. A description of a sound level meter and its use is included, together with a discussion of various analysis techniques. Another chapter is devoted to standards for measurement of noise.

4.3 L. L. Beranek (Editor), Noise and Vibration Control, (McGraw-Hill Book Publishing Company, NY, 1971).

This text presents a complete discussion of general methods of noise and vibration control, including a good general introduction to the basic instrumentation and methods for noise and vibration measurements.

4.4 P. G. Peterson, E. E. Gross, Jr., Handbook of Noise Measurement, Seventh Edition. (GenRad, Inc., Concord, MA, 1974).

The methods and instrumentation for the measurement and analysis of all types of noise and vibration are described. This is a very good general reference on noise measurements; needless to say, it focuses on GenRad instrumentation.

4.5. "Methods for the Measurement of Sound Pressure Levels," American National Standard S1.13, (American National Standards Institute, NY, 1971).

A detailed method for the general measurement of sound levels is presented. Included are general definitions of various types of noise and recommended procedures for measurement by type.

4.6 "Specification for Octave, Half-Octave, and Third Octave Band Filter Sets," American National Standard S1.11, (American National Standards Institute, NY, 1971).

The electrical characteristics of octave, 1/2 octave, and 1/3 octave band filters are specified. Included are standard center frequencies and band numbers for each band.

4.7 "Environmental Criteria and Standards," Code of Federal Regulations, Title 24, Part 51, Department of Housing and Urban Development.

The acceptable noise environment for federally funded housing projects is specified. The noise criteria are defined in terms of the statistical noise level L_n.

4.8 "Laboratory of Measurement of Airborne Sound Transmission Loss of Building Partitions," ASTM E-90-75, 1976 Book of ASTM Standards - Part 14, (American Society for Testing and Materials, Philadelphia, PA, 1976).

A method is specified for evaluating the sound-insulating properties of building partitions in a controlled laboratory environment.

4.9 "Measurement of Airborne Sound Insulation in Buildings," ASTM E336-71, 1976 Book of ASTM Standards - Part 14, (American Society for Testing and Materials, Philadelphia, PA, 1976).

A method for evaluation of the sound-insulating properties of building partitions installed in buildings is specified. The procedure is specifically intended for field evaluation.

4.10 L. Cremer, M. Heckle, Structure-Borne Sound, Translated by E. E. Ungar,(Springer-Verlag, NY, 1973).

This text concerns vibration of mechanical structures and the theory of vibration propagation. Included is a discussion of measurement techniques for evaluating vibration. Specific methods of particular interest are those presented for the measurement of mechanical driving point impedance.

4.11 J. S. Bendat, A. G. Piersol, Random Data: Analysis and Measurement Procedures, (Wiley - Interscience, NY, 1971).

The authors present the theory and practice of random data analysis. Rigorous definitions of random data types are included. The concepts of auto- and cross-correlation and auto- and cross-spectral density are discussed with respect to their Fourier transforms.

4.12 H.P. Verhas, "Measurement and Analysis of Train Induced Ground Vibration," (Dissertation for University of Southampton, Faculty of Engineering and Applied Science, Southampton, England, 1977).

Verhas presents the results of groundborne vibration measurements at ballast-and-tie section of a mainline railroad. Measurements were performed with transducers mounted on aluminum discs set in plaster-of-Paris in pits. The response of the mounted transducer was evaluated by striking them through a force transducer and comparing the applied force with the response velocity. His conclusion was that the transducer mounted resonance was about 475 Hz. Cross-correlation techniques were used to measure the vibration propagation velocity in the ground; the measured speed was about 200 m/sec.

4.13 "Specifications for Sound Level Meters," American National Standard S1.4, (American National Standards Institute, NY, 1971).

Specifications are given for standard sound level meters. Included is the response characteristics for fast and slow meter responses, specifications for A-, B-, and C-weighting characteristics for Type 1, 2, and 3 sound level meters.

4.14 D. R. Hurst, "Sound and Vibration Frequency Analysis", (Toronto Transit Commission Technical Reports, Toronto Transit Commission Subway Construction Branch, Toronto, Canada 1976).

The instrumentation used by the Toronto Transit Commission for aquisition, analysis, and presentation of noise and vibration data is described. The data aquisition equipment consists of custom built, portable, battery-operated frequency modulators and tape recorders, and standard accelerometers and microphones. A digital 1/3 octave band analyzer, interfaced to a programmable calculator and plotter, is used for data analysis and presentation.

4.15 "Noise and Vibration," Guidelines for Design of Rapid Transit Facilities, Section 2.7, (American Public Transit Association, Washington, D.C., January 1979).

Measurement techniques included in this document are for wayside and interior transit vehicle noise.

4.16 B. A. Kugler, D. E. Commins, W. J. Galloway, "Highway Noise, a Design Guide For Prediction and Control," NCHRP Report 174, (Transportation Research Board, Washington, D.C., 1976).

One of a series of National Cooperative Highway Research Program reports on prediction and control of highway noise. Included in this report and several previous reports is a simple method for manual determination of statistical noise levels, L_n, using a hand-held sound level meter.

4.17 "Acoustics - Measurement of Noise Emitted Railbound Vehicles," ISO Standard 3095-1975, (International Standards Organization, Switzerland, 1975).

A standard procedure is recommended for the measurement, analysis, and presentation of noise data for rail vehicles. Included is a description of microphone locations.

4.18 S. L. Wolfe, H. J. Saurenman, P. Y. N. Lee, "Assessment of Urban Rail Noise Climates and Abatement Options," Interim Scientific Report, DOT-TSC-850-2 (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, March 1976).

Noise measurements at the Bay Area Rapid Transit system are described. Measurement locations and procedures are described for platform noise and interior car noise.

4.19 "Acoustics - Measurement of Noise Inside Railbound Vehicles," ISO Standard 3381-1976, (International Organization of Standardization, Switzerland).

A standard procedure is presented for the measurement, analysis, and presentation of rail vehicle interior noise data, including a description of microphone measurement locations.

4.20 G. P. Wilson, J. T. Nelson, "Metrorail Operational Sound Level Measurements: Fan and Vent Shaft Noise Levels," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for the Washington Metropolitan Area Transit Authority, June 1976).

Measurements were performed at the WMATA Metro system to assess noise produced at fan and vent shafts. Measurement locations were at the shaft grating and at ear level approximately 30 feet from the shaft.

4.21 H. Nolle, "High Frequency Ground Vibration Measurements," <u>The Shock and Vibration Bulletin</u>, No. 48, Part 4, (Shock and Vibration Information Center Naval Research Laboratory, Washington, D.C., September 1978).

A detailed description is presented of the experimental evaluation of various types of spikes for mounting accelerometers to groundborne vibration. Aluminum spikes which maximize the ratio of soil contact area to mass are shown to perform adequately at frequencies below 200 Hz.

4.22 J. E. Manning, R. G. Cann, J. J. Fredberg, "Prediction and Control of Rail Transit Noise and Vibration - A State-of-the-Art Assessment," Interim Report Report, UMTA-MA-06-0025-74-5, (U.S. Department of Transportation, Urban Mass Transportation Administration, Cambridge, MA).

The state-of-the-art in groundborne noise and vibration measurement, prediction, and control are described. Of particular interest for measurements is the recommendation of a 15 cm diameter aluminum disc for mounting accelerometers when measuring vibration at soil surfaces.

4.23 H. P. Verhas, "Measurement and Analysis of Train Induced Ground Vibration," Noise Control Engineering, 13(1), (Institute of Noise Control Engineering, Poughkeepsie, NY, July-August 1979), pp. 28-41.
Essentially a summary of Reference 4.12.

4.24 T. G. Gutowski, L. E. Wittig, and C. L. Dym, "Some Aspects of the Ground Vibration Problem," Noise Control Engineering, 10(3), (Institue of Noise Control Engineering, Poughkeepsie, NY, May-June 1978), pp. 94-100.

The article reviews the physics of groundborne vibration; applicable building damage and human response criteria; the dynamics of vibration transducer coupling with the ground surface; and results of measurements of ground vibration from a 30,400 Joule piledriver in sandy layered soil and of ground vibration near the support columns of an elevated highway. Three types of transducer mounts are shown together with corresponding estimates of frequency response. Also presented is a theoretical discussion of the dynamics of a transducer mounted on soil surfaces and a number of references.

4.25 J. T. Nelson, H. J. Saurenman, G. P. Wilson, "Metrorail Operational Sound Level Measurements: Ground-Borne Vibration and Noise Levels," Preliminary Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, December 1979).

This report presents the results of measurements performed at WMATA Metro shortly after the system was operational. Measurement locations on subway structures and at the ground surface near subway structures and a ballast-and-tie track are described.

4.26 "Yonge Subway Northern Extension Noise and Vibration Study," Toronto Transit Commission Subway Construction Branch, Reports RD115/2 and RD115/3, (Toronto Transit Commission, Toronto, Canada, October 1976).

A very extensive set of measurements of groundborne noise and vibration at the Toronto Transit Commission is described, including measurement locations at the ground surface at various distances from the subway structure and on the invert of the subway structure. Noise and vibration measurements inside residential structures are also described.

4.27 R. D. Bruce, E. K. Bender, E. E. Ungar, "Noise and Vibration Measurements in the Toronto Transit Commission Subway," Draft Report, Report No. 1899, (Bolt Beranek and Newman, Inc., Cambridge, MA, Prepared for Washington Metropolitan Area Transit Authority, November 26, 1969).

A comprehensive set of measurements at various subway station locations is described. Measurement locations in the subway were at a variety of points, including bench, wall, and ceiling. Additional measurements were performed inside a house and substation to assess vibration transmission characteristics.

4.28 "Draft Standard Method for the Measurement of Vibrations in Buildings Adjacent to Railway Tunnels and Tracks--Test Code for the Measurement of Vibrations Inside Underground Tunnels," UITP International Metropolitan Railways Committee Working Group on Measurement of Noise and Vibration (Unpublished Circular and Minutes of Committee Meeting).

A tentative method is suggested for the measurement of vibration on subway structures. Measurement locations, described in detail, include points on the invert between the rails and on the wall. Additional recommended locations at 20 meters to either side of the primary location. Transducer orientation is recommended to be perpendicular to the measurement surface. Presentation of data is to be in terms of vibration velocity levels in dB re 10^{-8} m/sec. One-third octave bands are recommended for presenting spectral data.

4.29 G. P. Wilson, S. L. Wolfe, "BART Car Floor Vibration Measurements," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinckerhoff-Tudor-Bechtel, March 1974).

Measurements of vehicle floor vibration are described. The accelerometer was mounted on a steel disc with support legs capable of protruding through the carpeting without damaging the material. An integrator was used to convert the vibration acceleration analog signal to vibration velocity and vibration displacement analogs.

4.30 R. R. Bouche, "Ensuring the Accuracy of Shock and Vibration Measurements," <u>1965 Annual Technical Meeting Proceedings</u>, (Institute of Environmental Sciences, 1965).

Techniques are described for proper selection and mounting of accelerometer transducers and signal conditioning equipment. Attention is focused on the mounted resonance frequency of various accelerometer mounting techniques.

4.31 D. Pennington, "Piezo-electric Transducers," Endevco Technical Paper TP-225, (Endevco Corporation, San Juan Capistrano, CA, 1970).

Various types, characteristics, and selection criteria are discussed for piezoelectric accelerometers. This is a valuable reference for someone in the process of selecting accelerometers.

4.32 W. J. Webster, J. W. Farinacci, "Use of Graphic Level Recorders as Indicating Instruments," (Bureau of Noise, NY State Department of Environmental Conservation, Albany, NY, 1974).

The dynamic characteristics of two level recorders are compared with the dynamic characteristics of sound level meters. Only under a specific set of operating conditions was one of the level recorders capable of simulating a sound level meter fast and slow response.

CHAPTER 5

5. VEHICLE NOISE AND VIBRATION CONTROL

Transit vehicles are the source of most of the noise associated with transit systems. Hence, designing vehicles so that noise and vibration are minimized is the first step in controlling patron noise exposure and noise impact on adjacent communities. This chapter focuses on specific vehicle modifications and design features applicable for control of noise and vibration. Control of structureborne sound from aerial structures and of groundborne noise (both caused by the transit vehicle) are discussed in detail in Chapters 7 and 8.

Transit vehicle noise is due to wheel/rail interaction, propulsion equipment, and auxiliary equipment (the remainder of the vehicle equipment). Some causes of interior and exterior noise produced by the transit vehicle cannot be eliminated in the design of a new vehicle or the retrofit of an existing vehicle; however, many design details can significantly reduce noise and vibration produced by the transit vehicle.

5.1 NEW VEHICLE DESIGN

Within the last 10 to 15 years, a number of new transit systems and extensions to existing systems have been built. Because of expansions and replacement of old vehicles, transit systems will always require large numbers of new transit vehicles. Patrons, wayside communities, and transit properties have demanded vehicles quieter than those associated with older systems. A number of design changes and innovations for new transit vehicles have reduced vehicle interior noise and vibration affecting patrons and employees, and vehicle exterior noise and vibration affecting patrons in stations and the wayside community.

5.1.1 Vehicle Noise Criteria

The primary reasons for reducing vehicle noise and vibration are to produce a reasonably pleasant environment for the patrons, to allow conversation between patrons with normal vocal effort, and to control or minimize noise experienced by wayside communities. The inclusion of vehicle noise criteria or standards as part of the vehicle procurement specifications is one of the best methods of ensuring that noise goals are achieved. The APTA Guidelines for vehicle noise (see Appendix B) are designed to ensure that state-of-the-art techniques are used to minimize wayside noise and to create a pleasant interior environment. The APTA Guidelines recommend maximum A-weighted noise levels for different modes of operation and for specific equipment and operations of auxiliary equipment. As discussed in Section 2.1.1.5, "Vehicle Noise Standards," meeting the APTA Guidelines means that at low speeds on ballast-and-tie trackbed, the subjective rating of the interior noise is "quiet," while at high speeds in subways, the level is "intrusive" but not "annoying." Note that UMTA is in the process of preparing the "Standard Core Technical Specifications for Procurement of Rapid Rail Cars" that is intended to be the standard for UMTA-funded transit car purchases. The present draft includes a chapter on noise and vibration that generally specifies the same conditions and noise levels as the APTA Guidelines.

It should be noted that the vehicle noise and vibration design goals given in the APTA Guidelines are based on a balance between what is desirable and what is economically feasible. In addition to determining "baseline" noise levels, measurements of noise levels on existing transit cars have been used in conjunction with special testing programs to determine what may be achieved with reasonable effort and current car design techniques.

Noise and vibration specifications which are included as part of the vehicle procurement specifications should include definitions of the desirable limits of noise and vibration and the procedures and techniques to be used in testing for conformance to the specifications. These conformance tests include determination of acoustical performance of vehicle components, auxiliary equipment, and the complete vehicle assembly.

5.1.2 Control of Wheel/Rail Noise

At medium to high speeds, either the propulsion equipment noise or the wheel/rail noise is the dominant source of wayside and car interior noise. At medium speeds, it is more common for the wheel/rail noise to dominate. This section discusses some features that can be included in the design of new transit cars or, in some cases, retrofitted to existing transit cars to reduce wheel/rail noise. Only a brief discussion of wheel types and other factors of vehicle design affecting wheel/rail noise is presented here. More information on wheel/rail noise is given in Chapter 6.

5.1.2.1 Resilient and Damped Wheels -- There are basically three different types of wheels tested or used on transit systems today. These are standard wheels (either solid steel or with an aluminum center), resilient wheels, and damped wheels. The latter two do not greatly reduce roar noise or impact noise either inside or outside the transit vehicle, although they can produce measurable reductions of overall noise levels when propulsion system noise is low enough to prevent masking of the reduction. Most resilient and damped wheels are very effective at controlling wheel squeal noise; in many cases, the wheel squeal is entirely eliminated.

Three types of resilient wheels which are used or which have been tested on U.S. rapid transit systems are the Penn Cushion or Bochum wheel, the Acousta Flex wheel, and the SAB wheel. Results of an extensive series of tests performed at SEPTA are presented in Reference 5.1. Resilient wheels have had extensive use on transit cars in Europe and on PCC streetcars in the United States. The Acousta Flex wheel is currently in use at Boston and San Francisco on the Standard Light Rail Vehicle. Experience with resilient wheels in the SEPTA tests indicates that they should not be used on cars with tread brakes because of the potential for overheating.

For transit systems with short-radius curves, either resilient wheels or damped wheels are recommended for control of wheel squeal. Both constrained layer-damped wheels and ring-damped wheels are very effective at reducing wheel squeal. Ring-damped wheels, which have been in use since 1938 at London Transport, are currently in use at the CTA on the 2400 series cars, and they will also be used on the new 2600 series cars. Constrained layer-damping treatment has not been extensively tested in North America, however, following a 4-year trial period, 2400 wheels of the Paris Metro have been equipped with constrained layer-damping treatment.

5.1.2.2 Truck Primary Suspension Stiffness -- The effects of primary suspension stiffness on groundborne vibration are discussed in Chapter 8. However, since primary spring type and stiffness can also affect car noise, this discussion is included to indicate the effects of rubber journal sleeve characteristics on trucks that do not incorporate primary springing.

Tests were made at CTA on a pair of 2000 series vehicles with the rubber journal sleeves on the trucks replaced with softer journal sleeves to create the effect of soft primary springing on a truck without primary springs. The stiffness of the journal sleeves was reduced by a factor of 30 to 1. Noise level reductions, both inside and outside the vehicle, ranged from a minimum of 0 to 4 dBA on steel elevated structures to a maximum of 7 to 10 dBA for at-grade and subway operations. The softer journals apparently increase the total damping in the system and decrease the vibration transmitted from the wheels and axles to the truck frames and vehicle body. The softer journals also apparently reduce the impact forces which occur at the wheel and rail interface, thus decreasing the vibration level and radiated noise of the wheels and rails.

It should be noted that if soft primary springs are used on systems with concrete aerial structures, the noise reduction on the aerial structures would be similar to that found at CTA for at-grade and subway operations.

5.1.2.3 <u>Sound Absorption Treatment on Underside of Car Floor and Inside Face of Car Skirts</u> -- With concrete trackbed, absorption material applied to the underside of the car floor, especially above each truck, can be effective for reducing the car interior noise level and the wayside noise levels. Tests on a BART car, with approximately 140 sq ft of absorption material located over each truck, showed a noise reduction in the car of 3 to 4 dBA at-grade and 2 to 3 dBA in subway (Ref. 5.2). A wayside noise reduction of only 1 dBA was measured for at-grade ballast-and-tie tracks, as would be expected because of the ballast absorption. However, in subways, the wayside noise reduction was 3 to 4 dBA, and similar results would be expected on concrete aerial structures. On aerial structures, use of undercar absorption treatment and a nonabsorptive sound barrier wall should reduce wayside noise 3 to 4 dBA compared to the sound barrier wall and no undercar absorption treatment.

A skirt extension used in conjunction with a sound barrier wall can be an extremely effective method of reducing wayside noise. The use of car skirts can also decrease the required height of sound barrier walls. The existing skirts on the BART cars, for example, provide significant noise reduction, although not specifically designed for noise control purposes. Extending the car skirts on the BART cars would provide even more noise reduction.

The addition of undercar absorption material is relatively inexpensive in terms of materials and labor costs and adds little weight to the car. When used with extended car skirts, the undercar treatment can give significant reduction of overall noise exposure for both patrons and the wayside community.

5.1.2.4 Nonskid Braking System -- A nonskid braking system is a braking system with a slip-spin or skid control system that maintains maximum friction or adhesion between the wheels and rails. If slower than normal or if differential wheel speed is detected during deceleration, braking effort is decreased at that wheel or axle until normal wheel rotation is restored. Such a system is used on BART and is currently being installed in the CTA 2400 series cars and on older NYCTA cars. A principal advantage of nonskid braking systems is the reduction in the number and severity of wheel flats and rail burns. Wheel flats and rail burns can increase overall noise levels by up to 10 dBA.

Since nonskid braking systems can be effective at reducing the occurrence of wheel flats, implementation of a nonskid system can reduce the need for wheel truing.

5.1.3 Control of Propulsion System Noise

The propulsion motors (primarily the fans for self-ventilating motors) are usually the main source of noise at high speeds, although in some instances, gearbox noise is also a large contributor.

5.1.3.1 Ducted Forced Ventilation of Propulsion Motors -- Ducted forced ventilation of the propulsion motors can reduce propulsion system noise considerably (at least 6 dBA and up to 10 dBA) because self-ventilated propulsion motors generally have open air inlets and exhausts and a cooling fan mounted on the motor shaft. The fans often produce strong pure tones at the blade passage frequency. The forced ventilation system uses a constant speed, lower noise fan with air ducted into the propulsion motors. With this system, noise controls are easier to incorporate into the car design so that, at most operating speeds, the propulsion noise will be lower than the wheel/rail noise. One problem which exists with some forced air propulsion systems is excessive noise emanating from the motor cooling blowers when the vehicle is stationary or moving at slow speed. Due to space limitations and cooling requirements, the cooling blowers are often high-speed radial blade centrifugal fans. The radial blade centrifugal fan is the fan type which produces the highest noise level for a given airflow. Consideration should be given to use of centrifugal fans with a relatively wide wheel, operating at a relatively low rpm, with either a backward or forward curved airfoil blade. This will produce considerably lower noise levels for the same airflow and static pressure than a high-speed radial blade centrifugal fan. Table 5-1 gives representative noise levels from traction motor blowers. Also, the fan enclosure and the air ducts which supply the air to traction motors should be acoustically treated to avoid excess noise being transmitted into the passenger area.

5.1.3.2 Propulsion System Chopper Noise -- A number of new transit systems and some existing systems are acquiring cars with propulsion chopper control systems which use high-power semiconductors rather than the traditional cam or resistance-controlled system. Although the chopper controls do result in significant power savings, at slow speeds both variable pulse width and variable frequency chopper control systems generate pure tone noise from the chopper reactor. This noise can be reduced by one or a combination of the following: vibration isolation, partial or full enclosure of the chopper reactor unit, and extra floor insulation in the area where the chopper reactor unit is mounted.

5.1.3.3 Propulsion System Gearbox Noise -- Although most of the propulsion system noise originates from the traction motors, the reduction gears do contribute to the noise produced by the propulsion system. Many modern transit vehicles have motors and drive systems with right angle hypoid gear sets rather than a traditional gear and pinion arrangement. Compared to other gear configurations, hypoid gears have a higher overall tooth contact ratio and thus generate less noise. Car procurement noise specifications usually have a section pertaining to noise limits for equipment prior to installation on the car. These limits are not only applicable to the traction motors and undercar auxiliary equipment, but also to the propulsion system gearing. With the use of hypoid gears and appropriate design, the gearing system will normally meet the car procurement specifications and produce noise less than or comparable to that of propulsion motors.

5.1.4 Control of Auxiliary Equipment Noise

Apart from wheel/rail noise and propulsion system noise, the transit vehicle also produces noise from its auxiliary equipment which can have a significant impact when the vehicle is stationary or traveling at a slow speed. Noise-producing auxiliary equipment includes such items as: the air-conditioning compressor, condensor, and evaporator, motor alternator, low voltage power supply, equipment cooling blower, brake compressor, brakes, and other miscellaneous motors and blowers necessary for the operation of the transit vehicle. Different transit vehicles have different types of auxiliary equipment depending on the requirements of the transit system. The following is a discussion of the auxiliary equipment which has the potential for producing excessive noise.

5.1.4.1 Air-Conditioning System Noise -- The typical air-conditioning system on a transit vehicle consists of three main units: compressor, condensor, and evaporator. Depending on the design, one or more of these units may be attached or combined

TABLE 5-1 TRACTION MOTOR BLOWERS, TYPICAL NOISE LEVELS

	Noise Level @ 4.7 m (15 ft)
1. Calculated Levels from Typical Fans @ 1000 CMF and 1" Static Head	
Backward Curved Centrifugal	44 dBA
Forward Curved Centrifugal	45 dBA
Radial Blade Centrifugal	54 dBA
2. Measured Levels from Typical Centrifugal Fans	
Direct Drive In-Line Centrifugal	
@ 860 RPM - 1090 CFM @ 1/8" Static Head	40 dBA
@ 1750 RPM - 1000 CFM @ 1.6" Static Head	63 dBA
3. Calculated Levels from Direct Drive High-Pressure Radial Blade @ 3450 RPM	
850 CFM @ 1/8" Static Head	53 dBA
1000 CFM @ 4" Static Head	66 dBA
1600 CFM @ 6" Static Head	71 dBA

with another unit. On the BART car, the air-conditioning compressor is attached to the motor alternator unit, while on other car designs, the compressor and condensor make up a single unit.

Compressor unit noise results from the drive motor and the reciprocating compressor compressing the refrigerant vapor from the evaporator. Condensor noise results from the fans circulating air past the condensor coils. Evaporator noise results from fans circulating air past the evaporator coils and into the transit vehicle. Low-frequency noise inside the car can result if the unit is improperly mounted or if rotating elements are poorly balanced; the resulting vibration of the car body can radiate high levels of low-frequency noise inside the cars.

General guidelines for minimizing the noise from the air-conditioning system are:

- Maximize the areas and minimize thicknesses (or number) of rows of condensor and evaporator coils. Increased coil area will allow for reduced head loss and reduced fan speed for an equivalent airflow and heat transfer. This can reduce noise since fan noise is approximately proportional to the 5th power of the fan rpm or fan blade tip speed.

- Use effective shear mounts in place of resilient plate-form mounts for the air-conditioning equipment under the car. Shear mounts providing vibration isolation appropriate for the unit rpm will reduce the noise radiated from the floor. The fail-safe mounting provision of a plate-form type of mounting has been successfully duplicated in a shear mount for the air compressor on the WMATA Metro car. Changing to tube-form shear mounts reduced both vibration and noise.

- Position the evaporators underneath the car rather than above the ceiling or in the wall. It is much easier to control interior noise produced by the evaporator fans if they are located outside of the passenger compartment or transit car shell.

- Use sound absorptive material to line the air ducts and outlets. Air passages should be streamlined to avoid unnecessary bends. Duct cross-sections should be large enough to avoid high air velocity and the associated noise.

- Make certain that the natural frequencies of any cover panels and refrigerant lines are different from basic compressor and fan rotational frequencies.

- Avoid the use of steep propeller blade fans and high-speed radial blade centrifugal fans, which produce high noise levels.

- Give particular attention to proper mechanical balancing of all motors and fans. Include specific mechanical imbalance limits and performance tests in the purchase documents.

5.1.4.2 Motor Alternator and Low Voltage Power Supply Noise -- Depending on the design, the transit car will have either a motor alternator or a low voltage power supply for providing power to the car's auxiliaries. Although noise problems, which arise from this equipment, usually must be handled on a specific case basis, some general guidelines include enclosing the units and placing sound absorption material on the enclosure walls, vibration isolation of individual components of the unit, or vibration isolation of the entire unit.

5.1.4.3 Equipment Cooling Blower Noise -- Equipment cooling blower noise is similar to that produced by the traction motor blower, which has been previously discussed. Again, if possible, use of high-speed radial blade centrifugal fans should be avoided. Table 5-2 gives representative sound levels of equipment cooling blowers. Also, the cooling ducts should be acoustically treated to avoid excess transmission of noise into the passenger compartment and into the wayside community.

5.1.4.4 Brake System -- Brake system noise sources include air exhaust, air compressor intake and exhaust, air dumping, and brake squeal. Mufflers should be installed to control noise from all air intake and exhaust ports. Relatively inexpensive mufflers that do not significantly restrict airflow can completely control noise from both intakes and exhausts. Note that this will reduce complaints from communities near stations, terminals, and yards.

Break squeal can be a major source of irritation. Any brake system that generates a significant amount of squeal should be redesigned; usually the stiffness of the mounting system must be modified to control the resonance that is excited by the brake shoes rubbing on the brake discs or drums. At other times, damping treatment must be used to reduce the vibration amplitude. In any case, there are numerous rail transit cars that do not have significant problems with brake squeal: a clear indication that with careful evaluation and redesign, brake squeal can be controlled.

5.1.4.5 Other Auxiliary Equipment -- Other miscellaneous items of auxiliary equipment include various fans and motors for other vehicle functions. The basic guidelines outlined above for other auxiliary equipment items are also applicable to these items. Well-balanced, properly mounted and designed auxiliary equipment will not produce excessive noise.

TABLE 5-2 TYPICAL NOISE LEVELS OF EQUIPMENT COOLING BLOWERS AT 4.7 m (15 ft)

1. Calculated From General Fan Type	
a. Backward Curved Centrifugal	
2000 CFM @ 1" Static Head	47 dBA
3000 CFM @ 2" Static Head	55 dBA
b. Forward Curved Centrifugal	
2000 CFM @ 1" Static Head	48 dBA
3000 CFM @ 2" Static Head	56 dBA
c. Radial Blade Centrifugal	
2000 CFM @ 1" Static Head	57 dBA
3000 CFM @ 2" Static Head	65 dBA
2. Measured Levels of Typical Centrifugal Fans	
a. Direct Drive Centrifugal @ 1160 RPM	
2000 CFM @ 3/4" Static Head	52 dBA
3000 CFM @ 1.6" Static Head	58 dBA
b. Direct Drive Centrifugal Model @ 1750 RPM	
4000 CFM @ 1/2" Static Head	64 dBA
3000 CFM @ 1.6" Static Head	64 dBA

5.1.5 Miscellaneous Equipment Noise

Miscellaneous equipment that can produce excessive noise includes door operation equipment, public address systems, and lighting systems.

5.1.5.1 Door Operation Noise -- Specifications for door operation noise should be included as part of the vehicle procurement noise specifications. There are three types of door configurations in general use for transit vehicles:

- Sliding Doors. The doors slide on an upper and lower track into and out of a pocket in the wall of the car.

- Bifold Doors. Each of the two door parts is hinged in the middle and at the connection to the car wall, so that as they open or close from the center, the door simply folds out of the way against the car wall.

- Plug Doors. Similar to the sliding door configuration, the doors slide on a track which takes them outside of the vehicle wall instead of sliding into a car wall pocket.

Sliding doors are the type in use in most of the newer transit vehicles. These doors are fast operating and are generally efficient and relatively quiet when moving. However, due to their size and weight, significant impact noise can be generated when the doors are unlocked and when the doors are closed. This can be alleviated by better insulation of the door lock actuator, by eliminating metal-to-metal impacts of the door lock actuator, and by using a more resilient rubber channel to cushion the impact of the two doors when closing.

Plug doors are used on the SLRV and provide superior sound insulation characteristics due to their sealing nature, whereas both sliding and bifold doors allow sound leaks along the vertical edges of the door as well as at the door tracks. As with the sliding doors, significant impact noise can be generated from the doors unlocking and from the impact when the doors are closed. To alleviate these problems, the same techniques applicable to the sliding door configuration are also applicable for the plug door configuration.

Bifold doors are not generally used in new cars, principally because of the access requirements of disabled patrons. Bifold doors are generally quieter during operation than the other two-door configurations because of their smaller size and lower mass which thus requires less force for door operation. The drawback of this type of door involves its sound insulation characteristics. Sound leaks are not only possible around the vertical edges and at the door tracks, but also at the hinged door sections. Also, the lightweight character of the doors usually does not provide sufficient sound insulation for a modern transit vehicle.

5.1.5.2 Public Address System and Lighting System Noise -- Although not generally in the same category of noise-producing items as those previously discussed, these items are generally provided with noise specifications so that they do not produce excessive noise.

The public address system should be free of hum and noise which could interfere with the intelligibility of announcements. Also, the public address system should use a noise-canceling type of microphone so that car interior noise is not amplified when the announce microphone is used. The lighting system of most new transit vehicles consists of florescent lamps with associated fixtures and ballasts. Improper installation or the use of noisy ballasts can give rise to a significant pure tone noise (hum) which, although low in level, can still be a significant source of annoyance to passengers in the car.

5.1.6 Car Body Sound Insulation

Sufficient sound insulation or transmission loss (TL) of the car body floor, wall, and ceiling assemblies is necessary in order to achieve the design goal interior noise levels of the APTA Guidelines. At speeds in excess of about 50 km/hr, noise from the air-conditioning system and other auxiliaries is dominated by propulsion motor noise and wheel/rail noise. Since both of these sources are outside of the car shell, the TL of the floor, walls, and doors determines the interior noise of the car operating at high speed. The TL of the floor is the most important characteristic for operation at-grade on ballast-and-ties or on concrete aerial structure, since the dominant noise sources are located beneath the floor. The TL of the walls, doors, and ceiling is more important for operation in subways, where the sound field surrounding the car is at a higher level and is more uniform over the entire exterior surface of the car.

The sound energy transmitted by mechanical vibration from the trucks to the body of a modern transit vehicle supported by air springs is far below the airborne sound energy transmitted through the car body elements, and efforts at detecting body leaks and effectively sealing them are usually successful. Therefore enhancement of the transmission loss of the car body elements is one of the best available means of reducing operating noise levels inside the car.

A field sound transmission loss test is required to determine whether the transmission loss of the car body elements is sufficient to achieve the goals for interior noise levels. The required sound transmission loss at each of four frequencies for the various car body elements is usually part of the modern new car procurement specifications. The procedures to be used are outlined in ASTM E 336-71, "Recommended Practice for Measurement of Airborne Sound Insulation in Buildings."

Table 5-3 is excerpted from the latest CTA transit car specifications (2600 Series) indicating the transmission loss requirements.

An important design consideration is that the vehicle shell be a nonhomogeneous, sandwich wall type barrier as opposed to a limp mass type barrier. The sandwich wall type provides superior transmission loss for the equivalent mass of wall. The superior performance capability of sandwich wall construction is now generally recognized and has been successfully incorporated into numerous designs; the basic design includes impervious barriers separated by a layer of sound absorption material. When the separation of the impervious layers is too small or the area density (weight per sq ft) of one or both of the barriers is too low, the sandwich effect is lost in the low- and mid-frequency range, and the transmission loss characteristics tend more towards that of a limp mass.

Figures 5.1, 5.2, and 5.3 indicate the floor, sidewall, and ceiling construction used on the SOAC vehicle tested at the Pueblo Test Track and at various transit systems (Ref. 5.3). A new transit vehicle utilizing these features for car body construction would meet the transmission loss requirements of the CTA specifications or other similar new car specifications. It is, however, extremely important that the transmission loss effectiveness of these elements is not reduced by flanking paths around windows, doors, through ventilation ducts, through floor penetrations for cables, pipes, or through wall penetrations for heating elements, etc. The flanking paths can be minimized by proper design and careful workmanship of window molds, door paths (effective brush seals) and proper sealing of any necessary penetrations through the inner liner or through the floor. Usually, sealing the floor penetrations is the most critical requirement.

TABLE 5-3 TRANSMISSION LOSS REQUIREMENTS OF CTA 2600 SERIES TRANSIT CAR SPECIFICATIONS

Car Body Transmission Loss

1. The sound transmission loss of the car body floor, wall, and ceiling assemblies in completed form shall be adequate to achieve the interior noise level limits specified in 15.02, E, but in no case shall the average sound transmission loss of each characteristic section of the car body be less than specified in the following table.

SOUND TRANSMISSION LOSSES BY OCTAVE BANDS

Octave Band Center Frequency	Entire Floor	Walls Including Windows but Excluding Doors	Ceiling or Roof	Doors
250 Hz	27 dB	23 dB	23 dB	14 dB
500 Hz	35	31	31	22
1000 Hz	38	34	34	25
2000 Hz	38	34	34	25

The sound transmission loss shall be averaged over each characteristic section of the car body defined in the above table and must include the influence of all sound energy which transmits through all weak areas such as apertures, door seals, air ducts, or openings for supply and return ducts.

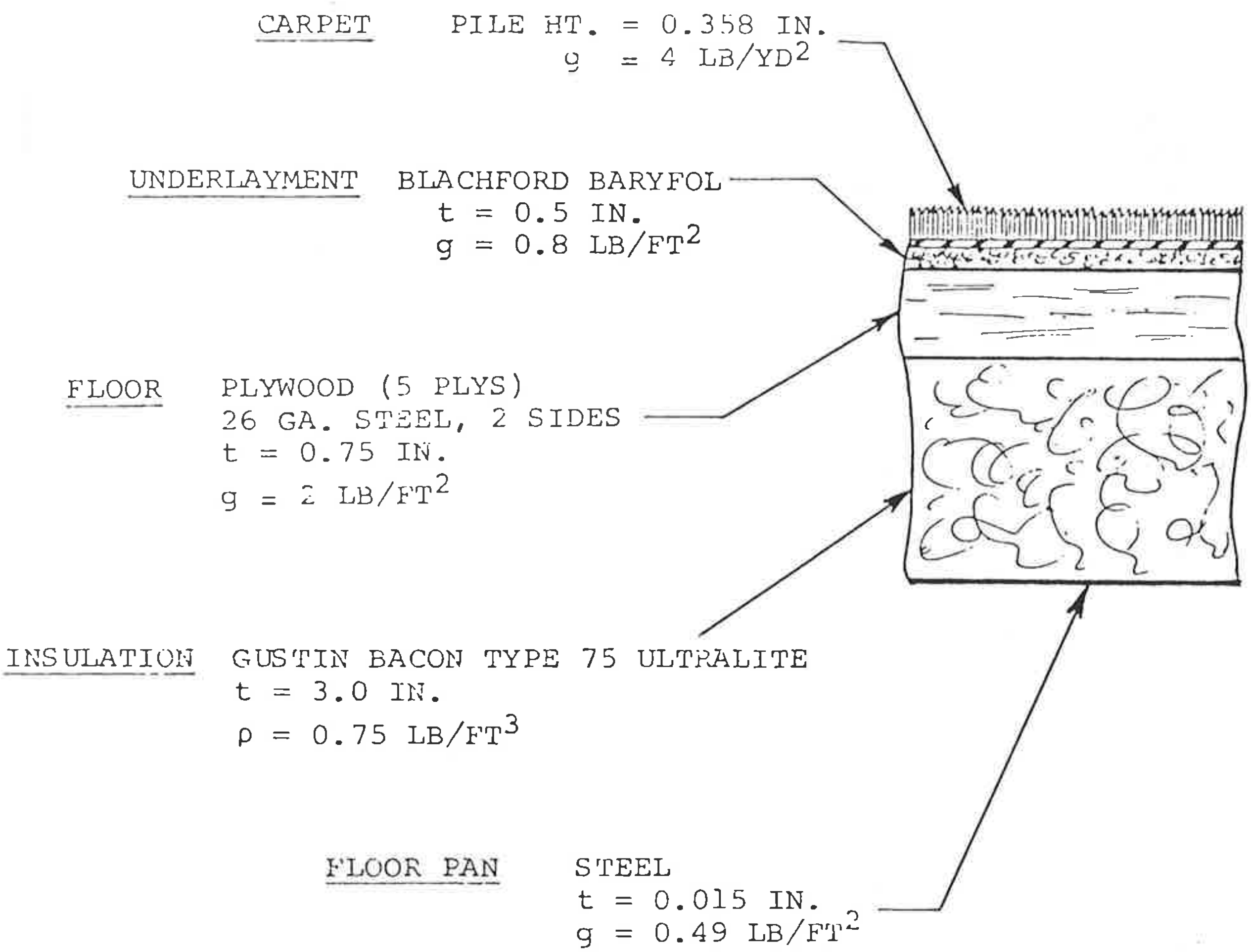
2. The Contractor shall, using the procedures outlined in ASTM E336-71, Recommended Practice for Measurement of Airborne Sound Insulation in Buildings, perform tests before delivering the cars to CTA to determine the sound transmission loss of the car body sections. Evaluation of the sound transmission loss of each characteristic section of the completed car body shall be done using one or both of the following procedures:

   With the car located outdoors on at-grade, ballast-and-tie track, or indoors in a space where reflected sound from nearby walls or floors will not influence the sound radiated from the car body by more than 2 dB, the Contractor shall, using portable loudspeakers in a manner reviewed by the Engineer, create random noise of constant level for the frequency range

TABLE 5-3 TRANSMISSION LOSS REQUIREMENTS OF CTA 2600 SERIES TRANSIT CAR SPECIFICATIONS (Continued)

encompassing the 250 Hz to 2000 Hz center frequency octave bands. With sufficient Sound Pressure Level inside the car, the noise transmitted through the car body is at least 10 dB higher than the outside ambient SPL in each octave band and with sufficient diffusion or distribution, the sound level in the car is uniform within 3 dB at 12 inches or more from any body surface (achieving a uniform sound field over the car floor will require removal of the seats). Using this procedure, the car body section sound insulation can be evaluated by using a sound level meter and octave band analyzer to measure the space average (SPL) inside the car in the 250, 500, 1000, and 2000 Hz center frequency octave bands. The car body section sound insulation also can be evaluated by measuring the exterior SPL for each of these octave bands at a distance of 12 inches from all car surfaces at a sufficient number of locations to determine the average noise reduction for each characteristic body section, such as the floor, walls, roof, and doors. The measurements must include the influence of any sound leakage such as at ducts, seals, or openings and the influence of flanking sound transmission paths at locations such as the floor/wall juncture. The difference between the interior space average Sound Pressure Level and the average exterior Sound Pressure Level at each section is the Noise Reduction provided by the car body sections. Noise Reduction measured in this manner is 6 dB greater than the transmission loss. The measurements must be corrected to transmission loss in accordance with procedures given in ASTM E336-71 in order to determine compliance with the specified minimum sound insulation of each car body section.

Alternatively, with the car located near highly reflective surfaces, such as over a maintenance and inspection pit, the transmission loss may be measured in accordance with the two room reverberant sound field methods indicated in ASTM E336-71. To create a satisfactory reverberant condition outside the car and to define the boundaries of the space, temporary baffles or barriers reviewed by the Engineer shall be placed between the car body exterior and the reflective surfaces, such as between the car body exterior walls at the floor level and the edges of a maintenance pit to define a closed space beneath the car for testing the car floor transmission loss. The temporary baffles both define the space exterior to the car and prevent flanking paths outside the car from influencing the measurements, for example, by preventing sound transmitted through the car walls or doors from bypassing the floor during a test of the floor.



t = THICKNESS

g = WEIGHT PER UNIT AREA

FIGURE 5.1 CAR BODY NOISE REDUCTION FEATURES, FLOOR CONSTRUCTION

WINDOW GLASS

t = 0.25 IN.
g = 3.25 LB/FT^2

SKIN STEEL-SHEET, t = 0.078 IN.
CORRUGATED, t = 0.042 IN.

DAMPING 1/32 IN. SPRAYED INSULMAT

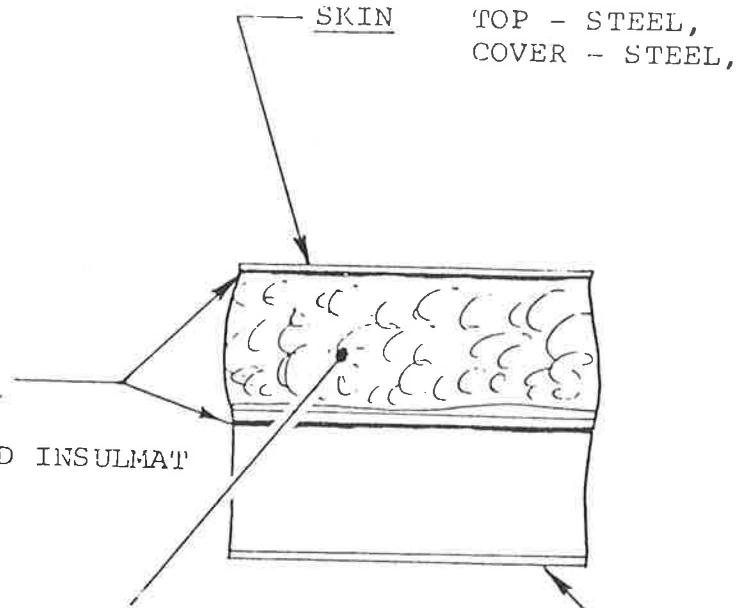
LINER

FIBERGLASS REINFORCED PLASTIC
t = 0.125 IN.
g = 0.016 FT^2
OR
MELAMINE ON ALUMINUM
t = 0.154 IN.
g = 1.75 16/FT^2

INSULATION - GUSTIN BACON TYPE 75 ULTRALITE
0.002 IN. ALUM. FACING (INNER FACE)
THICKNESS - 1.0 IN.
DENSITY - 0.75 LB/FT^3

t = THICKNESS
g = WEIGHT PER UNIT AREA

FIGURE 5.2 CAR BODY NOISE REDUCTION FEATURES, SIDEWALL CONSTRUCTION



t = THICKNESS

g = WEIGHT PER UNIT AREA

FIGURE 5.3 CAR BODY NOISE REDUCTION FEATURES, CEILING CONSTRUCTION

5.1.7 Vehicle Vibration

The characteristics of vehicle vibration are covered in Chapter 3 of this handbook. Vehicle vibration originates from wheels rolling on rails and from operation of auxiliary equipment. Vibration caused by train movement falls mainly into the category of ride quality (which is outside the scope of this handbook). Vibration from auxiliary equipment can have a strong influence on passenger comfort, when the vehicle is both moving and stationary. Equipment vibration can be an important factor influencing passenger comfort since this vibration usually affects the passenger whether or not the train is moving.

Standards of car vibration in both the vertical and horizontal plane are usually included in the car procurement specifications. A typical specification is given in Figure 3.6 and is applicable to vertical and horizontal vibrations anywhere on the floor, walls, ceiling, or seat frames of the stationary car. The vibration specifications are not difficult to meet and can be achieved by using the same methods for reducing the noise from auxiliary equipment, i.e., appropriate vibration isolation and mechanical balance.

New transit vehicles with undercar-mounted auxiliary equipment have some isolation between the units and vehicle. However, this isolation is not always optimized for the operating rpm or weight of the particular piece of equipment. Reduction of vibration can be achieved by one or a combination of the following: improving the balance of rotating parts, optimizing the stiffness of the vibration isolation mounts, improving bearing smoothness, changing the methods of mounting (i.e. from plate-form mounts to effective shear mounts as discussed previously) and possibly, stiffening of the support beams (Ref. 5.4).

Another effective method for reducing auxiliary equipment vibration (and consequent noise) is the use of an isolated floor. This form of isolation is employed in the SOAC vehicle and is accomplished by the use of leaded vinyl laminated on resilient foam laid under the carpet. This has the effect of decreasing the interior noise, most of which is probably caused by radiation from floor vibration, by 4 to 6 dBA.

5.2 MODIFICATION OF EXISTING VEHICLES

Many of the same techniques used for minimizing noise and vibration on new vehicles are applicable to older transit vehicles. Although only a major overhaul or rebuilding will create an interior noise environment which is equivalent to the new generation (post 1960's) of transit cars, older cars can be modified to significantly affect the passenger environment. Possible treatments are closing and sealing operable windows and vents with the addition of forced air ventilation or air-conditioning; improving seals on side doors; reducing noise of overhead fans; improving sound insulation characteristics of the car body; and redesigning traction motors for quieter operation (e.g., with forced air ventilation). Obviously, there are some procedures that are more cost effective than others and this will also depend on the transit system and the predominant noise problem associated with each group of older cars.

5.2.1 General Considerations

A number of nonacoustical questions should be addressed before any acoustical modification of the vehicle is planned. The projected life of the vehicle must be considered; obviously, if the vehicle is to be retired within 5 years or so, the cost effectiveness of acoustically improving the car is questionable. However, if there still is a long life expectancy or if the car is to be rebuilt (ostensibly for nonacoustical reasons), then a significant improvement in the vehicle acoustics can be achieved for a relatively low cost.

Major considerations should also include the scope of benefits obtained from the various retrofit options. The most apparent modification for cars with operable windows is to seal the windows and vents and add air-conditioning; however, the resulting noise reduction only benefits passengers inside a train operating in subway and on older steel elevated structures. It does not reduce the noise levels in the stations or the wayside community. In fact, due to the air-conditioning equipment noise, the noise in the stations could possibly increase when the train is stationary. Reducing the propulsion system noise and wheel/rail noise will have the benefit of not only reducing noise inside the transit car, but also of decreasing the station and wayside community noise.

An extensive study done at the NYCTA (Ref. 5.5) to determine ways of quieting older subway cars showed that significant noise reduction could be achieved in the car (10 to 15 dBA in most cases) by treating the door seals, traction motor fan, and floor for an estimated cost of less than $25,000 per car, not including the air-conditioning cost. Although this has not yet been verified by a production retrofit program, it does indicate that a significant degree of noise reduction can be achieved at reasonable cost.

5.2.2 Reducing Running Noise

As discussed previously, there are two principal sources of noise when the transit train is moving: the propulsion system noise and the wheel/rail noise. For many older transit cars, except at slow speeds, the predominant noise is from the propulsion system. Thus, even if the vehicle operates on smooth welded rail rather than rough jointed rail, significant noise reduction is usually not realized because of the dominance of the propulsion system noise.

5.2.2.1 Reducing Propulsion System Noise -- Most older transit cars have self-ventilated traction motors with armature-mounted radial blade cooling fans. The fan noise is the main component of the propulsion system noise and as discussed previously, forced ventilated traction motors are much quieter. The NYCTA car study (Ref. 5.5) recommends the redesign of this fan to one which is more like an axial vane fan. It is estimated that this will reduce the noise from the motor by 10 dB. Other studies (Ref. 5.6) have shown that a redesign of this fan results in reducing motor noise by 6 dB at most, and that the cooling is not always sufficient. An alternative approach is to retain the same traction motor, remove the integral fan, and provide forced air cooling via newly installed ducts and blowers which supply air to the motor air inlet openings. This provides more efficient cooling and reduces the noise level from the traction motors by 10 dBA or more. Another alternative is to replace the existing fan impeller with a randomly spaced blade impeller. This reduces blade frequency noise and generally gives about 6 dBA overall reduction.

Reducing the propulsion system noise will reduce overall noise levels inside the transit vehicle, in the stations, and in the wayside community.

5.2.2.2 Reducing Wheel/Rail Noise -- Assuming that the propulsion system noise has been significantly reduced, the next area of concern is wheel/rail noise. The mechanism of wheel/rail noise and methods for its reduction are discussed extensively in this handbook. Wheel/rail noise reduction techniques applicable to an existing transit vehicle are discussed in Chapter 6.

A primary method used to reduce wheel/rail noise on a transit vehicle is truing of the wheels. Trued wheels in conjunction with smooth and welded rail can significantly reduce noise (up to 10 dB) inside the vehicle, in stations, and in the wayside community. The amount that the noise level is reduced is strongly dependent on the condition of the wheels.

Experimental modification of a CTA 2000 series car indicated that using soft rubber journal sleeves between the axles and truck frames resulted in a substantial reduction of both in-car and wayside noise (Ref 5.7). This reduction in the stiffness of the bearing sleeves at the axles reduces the effective unsprung weight at the wheels while, at the same time, provides better vibration isolation between the wheels/axles and the truck frame. The overall result is a 3 to 5 dB reduction of wayside noise and up to 8 or 9 dB reduction of car interior noise for some operating conditions. This is a substantial reduction achieved by a relatively minor vehicle modification, although application in service has not yet been tried. This procedure should also be applicable to other transit systems using trucks with stiff journal sleeves rather than primary springs.

The use of resilient or damped wheels for controlling wheel squeal is another area where significant noise reduction is achievable on an existing vehicle. Either wheel type will substantially reduce wheel squeal noise for operation on all track structures and for both car interior and wayside noise. If the propulsion noise has been reduced significantly with the use of forced ventilated motors, roar noise can also be somewhat reduced with the use of damped or resilient wheels.

5.2.2.3 <u>Treatment of the Car Body</u> -- As discussed previously, the treatment of the car body (e.g., sealing leaks and improving sound insulation of the car body elements) can be an effective way of reducing the noise produced in an existing transit car, but this treatment decreases only the noise experienced by passengers in the cars.

The most obvious treatment for cars with operable windows is to seal the windows and vents and provide forced ventilation or air-conditioning, if the car can successfully withstand the added weight. This would be most beneficial for subway operations and also for trains operating on older steel elevated structures.

Improving the door seals on cars with sliding doors should be the next area for improvement. For cars with bifold doors, effective sealing of the doors is probably impossible because of the high number of hinged parts.

Improving the sound insulation of the floor is a procedure which would help reduce interior noise when a car is operating on any type of way structure. Two different types of floor treatment are among the most viable. The first is to increase the mass of the floor, easily accomplished by adding a leaded vinyl pad or even by adding a steel plate to the top of the existing floor. The second type of floor treatment is more expensive to install but will not add as much weight. This treatment involves putting 3 inches of glass fiber beneath the floor, held in place by an isolated steel or aluminum plate. This method would reduce the noise transmitted through the floor by up to 10 dBA.

If the car is being rebuilt and the inner liner to the passenger area is exposed, the transmission loss properties of the walls and roof can be improved by sealing all possible sound leak paths and using 1 to 3 inches of acoustical insulation in the wall cavity.

It is also important to be sure that the windows and vents are properly sealed so that no edge leaks or other paths exist to degrade the acoustical performance of the car walls.

5.2.3 Reducing Auxiliary Equipment Noise

On older transit system vehicles, the predominant noise occurs when the vehicle is moving. However, when the car is stopped or operating at a slow speed, auxiliary equipment noise can often dominate. In many instances, the high noise levels from auxiliary equipment is not due to the intrinsically high noise level associated with a particular piece of equipment, but rather to the lack of proper maintenance or repair. Thus, improving the balance of rotating parts, replacing rough bearings, and greasing or oiling noisy equipment can reduce the auxiliary equipment noise substantially.

As with new car designs, there are a number of areas and general considerations which can contribute to a reduction of noise from auxiliary equipment. Some of the techniques and areas where noise control is beneficial are:

- Isolation and Balancing of Auxiliary Equipment Components. Most transit vehicles with undercar-mounted auxiliary equipment have some isolation between the units and vehicle. Optimizing the stiffness of the isolators to minimize vibration, changing the methods of mounting to provide for increased isolation, and rebalancing of different rotational elements can reduce the noise transmitted through the floor.

- Air Brake Vent Mufflers. Many low cost mufflers are available which will have no detrimental effect (e.g., restriction of airflow) on the performance of the braking system. Installing mufflers on the brake vents of transit vehicles utilizing air brakes can provide a substantial decrease in the noise produced by escaping air as the transit train stops at a station. The mufflers would provide a large decrease in the overall noise, experienced by commuters on the station platform. In addition, they can help reduce complaints from people living near terminals, stations, or yards. The use of mufflers to reduce the noise from "dumped" air is particularly important in controlling community noise near yards. The mufflers do not provide significantly reduced noise inside the vehicle. However, some benefit can be realized inside the vehicle if windows or doors are typically open at the time of the air exhaust.

- <u>Redesign, Repair, and Maintenance of Doors</u>. An annoying and intrusive interior noise can be caused by the operation of the transit vehicle doors. The noise level produced from the operation of the doors can usually be decreased with repair and maintenance of older doors. Results obtained on older transit cars indicate that a 5 to 8 dBA decrease in door operation noise can be achieved.

- <u>Redesign, Repair, and Maintenance of Overhead Fans and Blowers</u>. One problem with older non-air-conditioned cars is the high noise levels produced by overhead fans. A noisy overhead fan can increase the noise by 10 dBA or more inside a transit car that is stopped or operating at slow speed. Quieter motors, proper lubrication, and replacement or worn bearings can substantially reduce this noise. For those situations where the fan is enclosed behind the ceiling or wall panel, reduction of fan noise can be accomplished by adding sound absorptive duct liner to the ducts and/or isolating or enclosing the fan motor.

- <u>Air Compressor Intake and Exhaust Mufflers</u>. The air compressor is frequently a noisy, if not the noisiest, item of the auxiliary equipment complement. Use of simple, lightweight air intake mufflers can considerably reduce compressor operation noise. Similar to reduction of air brake operation noise, the application of mufflers to the pressure relief or air exhaust valve can considerably quiet the operation, without inhibiting airflow or the safety aspects of the valve operation.

CHAPTER 5 REFERENCES

5.1 H.J. Saurenman, R.L. Shipley, G.P. Wilson, "In-Service Performance and Costs of Methods to Control Urban Rail System Noise Initial Report," Final Report, UMTA-MA-06-0099-80-1, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., December 1979).

This report compares the effectiveness of five methods of controlling wheel/rail noise and vibration on urban rail transit systems. Measurements performed on the SEPTA system provided noise and vibration data from rail grinding, wheel truing, resilient wheels, ring-damped wheels, and welded versus jointed rail track. These data were compared to similar studies on other North American transit systems and the relative effects of each method and its overall cost are discussed.

5.2 G.P. Wilson, "BARTD Prototype Car 107 Noise Tests with Standard, Damped and Resilient Wheels," Final Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinckerhoff-Tudor-Bechtel and Bay Area Rapid Transit District, June 1972.)

This report describes the effects of rail grinding operations evaluated with resilient and damped wheels on car interior and wayside noise levels for surface tracks and subway structures. Included is a comparison of in-car noise levels on concrete aerial structure and ballast-and-tie track with sound absorptive material installed between the trucks and the car underbody.

5.3 Boeing Vertol Company, "SOAC Engineering Tests at DOT High

Speed Ground Test Center," Vol. IV, Final Test Report, UMTA-MA-06-0025-75-4, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., January 1975).

Volume IV of this report presents results of extensive car interior and wayside noise and vibration measurements performed on SOAC Test Vehicles. The measurements were performed using solid steel and resilient (Acousta Flex) wheels at various speeds and track sections. All modes of vehicle operation were evaluated including selective operation of car auxiliary equipment. Data are presented in terms of A-weighted noise levels, 1/3 octave band plots of acceleration levels, and narrowband spectral analysis.

5.4 G.P. Wilson, S.L. Wolfe, "BART Car Floor Vibration Measurements," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinckerhoff-Tudor-Bechtel, March 1974).

Test data are presented on floor vibration levels caused by the undercar auxiliary equipment on ten BART cars. General procedures for reducing the vibration levels are discussed including vibration isolation mounting and balancing of rotating equipment.

5.5a P.J. Remington, L.E. Wittig, M.M. Myles, "Diagnosis and Control of Noise in Older Rapid Transit Cars," Paper presented at the 96th Meeting of the Acoustical Society of America, (Honolulu, Hawaii, November 1978).

5.5b P.J. Remington, L.E. Wittig, M.M. Myles, "Summary Report: Noise Control Study of Older Rapid Transit Cars for the New York City Transit Authority," BBN Report 4059, (Bolt Beranek and Newman, Inc., Cambridge, MA, Prepared for New York City Transit Authority, February 1979.

An experimental study, which after identifying primary noise sources, characterizes the noise reduction realized from various noise control techniques on NYCTA type R-33 and R-38 transit cars. It was determined that propulsion system noise rather than wheel/rail noise was the predominant noise source. It was also determined that airborne noise contributed to the interior noise to a much larger degree than structureborne noise. The basic conclusion for the most effective methods for reducing interior noise included (1) reduce the noise from the traction motor fan (most effective when the train is on welded rail), (2) sealing the doors, and (3) improve the isolation and insulation characteristics of the transit car floor (very effective if the vehicle has air-conditioning).

5.6 J. R. Tucker, "WMATA Motor Noise Test Results - Axial Discharge Design," (Private letter communication by Westinghouse Transportation Division, May 2, 1973).

Presented are noise levels from the WMATA vehicle traction motor at various positions with different noise control designs.

5.7 G. P. Wilson, "Noise Levels from Operations of CTA Rail Transit Trains," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Chicago Transit Authority, May 1977).

This is an extensive report on an evaluation of CTA car and track system noise using specially designated test trains on all types of track roadbed structures. The objective was to develop recommendations for improving and updating existing facilities to reduce noise levels. A detailed study was made on the effects of substituting softer rubber journal sleeves on the axles of 2000 series cars in terms of overall car interior and wayside noise levels.

5.8 T.D. Northwood, "Rail Vehicle Noise," Research Paper No. 155, (Division of Building Research, Ottawa, Canada, April 1963).

This paper discusses noise from rapid transit vehicles with an emphasis on interior noise levels and proposed and successful methods used to help reduce this noise.

5.9 E. Marlowe, "Comparison of Noise Levels in Rapid Transit Vehicles," Presented to the Panel on Noise Control, American Transit Association Rail Transit Group Conference, (Chicago, Illinois, May 5-8, 1964).

A study of the interior noise level of 13 different transit systems. Although the results are not exhaustive, they indicate possible areas which contribute to the different noise levels observed for the various systems.

5.10 L.G. Anderson, "Rapid Transit Rail Car Noise," Chicago Transit Authority, April 27, 1964, Paper which appears to have been presented to the Panel on Noise Control, American Transit Association Rail Transit Group Conference, (Chicago, Illinois, May 5-8, 1964).

A discussion of a number of noise reduction techniques and their effectiveness when tested on the CTA. Most of these fixes did not provide a substantial amount of noise reduction and were rejected.

5.11 J.J. Heffernan, "Noise Attenuation - The Expo Express," Presented at the Rail Transit Group Conference of the American Transit Association, (Montreal, Quebec, April 1967).

This paper deals with the noise reduction features considered and adopted for the Mass Transit System for the 1967 World Exhibition. Areas of noise reduction included an "acoustic fence," rail vibration isolation, rail lubrication, and wheel treatment.

5.12 Toronto Transit Commission, "Noise and Vibration Control - Rolling Stock," Report RD109C, (Toronto Transit Commission, Toronto, Canda, May 1967), pp. 10-13.

This somewhat older report covers the various noise sources which radiate from the car and paths the noise takes to radiate into the car. Possible areas of future tests are briefly discussed.

5.13 R. L. Colegate, "Standards of Rider Comfort: Noise Vibration and Age of Rider as Factors," Final Report, NTIS N75-18891, (June 1974).

This study of passenger comfort is somewhat preliminary and was to be continued under NASA Grant NSG-1074. Although the study was designed for all forms of mass transit, the study correlated people's responses to measured noise and vibration data obtained when riding on certain controlled bus routes. No quantitative data are given and few concrete conclusions are presented.

5.14 American Iron and Steel Institute, "Steel Structures for Mass Transit," Chapter 4, Comfort Criteria, (Washington D.C., 1977 est).

A relatively brief but thorough discussion of vehicle body acceleration and jerk and the effects on people. Various comfort limits from previous studies are given along with a general recommended range of criteria. Since the publication, for the most part, deals with structures, the design of structures is related to the vehicle body acceleration and hence, passenger comfort.

5.15 United States Environmental Protection Agency, "Passenger Noise Environments of Enclosed Transportation Systems," EPA-550/9-75-025, (Washington, D.C., June 1975).

This report gives noise levels and noise exposure levels for various enclosed transportation modes (cars, aircraft, rapid transit, etc.). It gives a number of references and presents a compilation of data from these references which allows for a comparison of interior noise data.

5.16 E. J. Rickley, R. Quinn, G. Byron, "Noise Level Measurements on the UMTA Mark I Diagnostic Car (R42 Model)," Technical Report, DOT-TSC-UMTA-72-3, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C.,October 1971).

This report contains noise levels measured inside an R42 model NYCTA car on the "N" Line (subway) and at the DOT Pueblo Test Track (at-grade). Contained are tabulations of the measured noise levels, time history charts, 1/3 octave band frequency analyses, and pertinent comments on the information obtained. No profound conclusions or suggestions for reducing the noise are given.

5.17 E. E. Ungar, "Noise in Rail Transit Cars: Incremental Costs of Quieter Cars," Final Report, EPA-550/9-74-012, (U.S. Environmental Protection Agency, Washington, D.C., June 1974).

This is a relatively comprehensive report for the time of the study (1974). It covers basic interior noise levels for several different systems and discusses methods of noise reduction applicable to new cars. Retrofit methods on old cars are not discussed due to high costs involved.

5.18 C. M. Harris, B. H. Aitken, "Noise in Subway Cars," Sound and Vibration, 2(5), (February 1971), pp. 12-14.

A brief report of noise levels measured inside subway vehicles. It gives some general qualitative noise reduction tips, and it indicates some factors which tend to change the interior noise level (i.e., loading, trackbed, age of car, etc.).

5.19 P. J. Remington, et. al., "Wheel/Rail Noise and Vibration Control," Interim Report, UMTA-MA-06-0025-73-15, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., May 1974).

This report covers the interim results of a program under the UMTA Urban Rail Supporting Technology Program to develop a basic understanding of urban transit wheel/rail noise generation for application to the evaluation and improvement of wheel/rail control devices. Useful background material applicable to the vehicle noise and vibration control include wheel truing, antilock devices (for brakes), resilient wheels, wheel damping, wheel skirts, noise deadening rings, and titanium wheel treads.

5.20 W.C. Caywood, H.L. Donnelly, N. Rubinstein, "Guideline for Ride-Quality Specifications Based on Transpo '72 Test Data," Final Report, UMTA-MA-06-022-77-3, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., October 1977).

These are guidelines for ride quality specifications suitable for use in Automated Guideway Transit (AGT) specifications. These were developed with the aid of new data obtained from four prototype AGT systems. Although less applicable to heavy rail transit, a set of acceleration and jerk values is presented.

5.21 P. J. Remington, M. J. Hudd, I. L. Ver, "Wheel/Rail Noise and Vibration, Volume II: Applications to Control of Wheel/Rail Noise," Final Report, UMTA-MA-06-0025-75-11, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., May 1975).

This report covers the final results of a program which began as reported in Reference 5.19. This volume covers among other items general techniques for the suppression of wheel/rail noise, applicable to vehicle noise and vibration control. Further work is also suggested.

5.22 L. G. Kurzweil, R. Lotz, E. G. Apgar, "Noise Assessment and Abatement in Rapid Transit Systems -- Report on the MBTA Pilot Study," Final Report, UMTA-MA-06-0025-74-8, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., September 1974).

This report is an assessment of the noise climate of the MBTA. It covers both interior and exterior noise. Useful for vehicle noise and vibration control is a table of rapid transit noise abatement techniques as they relate to car treatment (Table 3-3). Various scenarios for reducing noise from the MBTA are given, and relative costs are also given.

5.23 P. W. Eade, C. G. Stanworth., "Acoustic Aspects of Railway Vehicle Design," Proceedings of The Institution of Mechanical Engineers (Railway Divison), Vol. 190, No. 58/76, (October 18, 1976), pp. 515-525.

This paper discusses noise control techniques tried and proposed for reducing interior noise of railway coaches. Although the main discussion focuses on unpowered vehicles, there is some discussion relative to powered vehicles.

Included is a discussion of airborne and structureborne noises and the manner that these noises are transmitted into the vehicle. Also discussed is an experimental vehicle constructed with a floating floor, resiliently mounted internal panels, upholstered seats, and thick pile carpet. The A-weighted noise level inside the vehicle was lower than the standard vehicle, however, low-frequency noise was still a problem. The authors suggest some areas of further study.

CHAPTER 6

6. SURFACE TRACK NOISE CONTROL

The noise from trains operating on surface tracks is one of the most severe environmental problems faced by rapid transit systems. One of the principal concerns in the acoustical design of new transit facilities is the control of transit noise affecting adjacent communities.

The main sources of community noise from trains operating on surface track are:

- Wheel/Rail Interaction: The noise from the wheels rolling on steel rails.

- Structure Vibration: The vibration of the rails transmitted through the rail fastening system into the transit structure. The vibration of the transit structure can have a sounding board effect that significantly increases wayside and car interior noise levels. Noise radiated from aerial and elevated structures is considered separately in Chapter 7 because of its singular importance.

- Propulsion Equipment: The propulsion equipment, particularly the traction motors and the reduction gears, is an important source of wayside noise. Since the propulsion equipment is part of the vehicle design and is covered by the vehicle specifications, control of the noise radiated by the propulsion equipment is discussed in Chapter 5.

- Auxiliary Equipment: This category includes compressors, air handling equipment, braking systems, and motor generator sets. At normal operating speeds, the noise from this equipment is usually well below that of the propulsion equipment and the wheel/rail interaction. Auxiliary equipment does not usually create wayside noise problems; however, it does create problems in storage areas and at station platforms.

Other noise sources are third rail shoe sliding and aerodynamic noise. Third rail shoe noise is rarely a problem and can generally be ignored. Aerodynamic noise, caused by the turbulence of the air flowing past the moving transit car, does not become important until reaching speeds of 200 to 250 km/hr.

The purpose of this chapter is to discuss the methods that can be used to reduce the levels of wayside noise once the basic track and vehicle designs have been determined. Once this point has been reached, there are relatively few options available. In laying out the route alignments, efforts should be made to avoid noise-sensitive areas. This is particularly important when locating special trackwork and insulated joints. The impact noise from special trackwork joints increases noise to levels which can be distinctly annoying. Careful placement of crossovers and turnouts can save a considerable amount of trouble after the system is operational. The use of bonded insulated joints, as opposed to conventional bolted joints, will significantly reduce the impact noise at these joints.

The importance of considering community noise levels during the design phase of new transit systems cannot be overemphasized. The first step is to ensure that the transit vehicles are as quiet as possible. With new vehicles, the use of sound level limits in the contract documents effectively controls noise emissions. Other general goals are to lay out the system so that the impact on noise-sensitive areas is minimized; to use at-grade ballast-and-tie or open cut instead of aerial structures; and to use state-of-the-art technology, such as welded rail instead of jointed. Of course, it is also important to provide an ongoing program of wheel and rail maintenance because wheel and rail wear cause increases in wheel/rail noise.

The remainder of this chapter is divided into three sections. The first presents a summary of the parameters that affect the levels of wayside noise, including a brief discussion of the most practical methods for controlling the noise. The second section reviews the generation and control of wheel/rail noise. Although wheel/rail noise contributes to car interior as well as wayside noise, the problem of wayside noise is usually more critical; therefore, wheel/rail noise is covered in this chapter. The final section, 6.3, discusses the design of sound barrier walls. Included in Section 6.3 are design curves for estimating the attenuation that will be achieved with sound barriers, cuttings, and other obstructions blocking the line-of-sight between the trains and the receiver.

6.1 PARAMETERS THAT INFLUENCE WAYSIDE NOISE

In this section, the factors that influence the levels of wayside noise are enumerated and discussed. The factors range from selection of alignment and profile to train speed and selection of wheel type. In virtually all cases, strategies for control of wayside noise will involve one or more of the parameters listed below. Note that we do not cover the methods that can be used to improve the sound insulation of buildings. Although there are times when the most economical method of reducing noise impact is to modify, or even to purchase the affected building, this is rarely a viable alternative for U. S. transit systems.

6.1.1 Alignment and Layout

The reduction of community impact that can be achieved by modifications to the transit route alignment and layout should not be overlooked. Of course, the acoustical benefits must be balanced against other factors, many of which can be significantly more important than acoustics. Designing transit routes to take advantage of existing transportation corridors, such as highways and rail routes, can have a number of acoustical advantages. The noise from pre-existing transportation systems makes the area near the route less sensitive to noise. Also, the transit structure can often be arranged to take advantage of an existing highway or rail line as a buffer zone or to provide acoustical shielding between the transit route and noise-sensitive land use such as a residential area.

Another factor to consider when laying out the transit system is the location of special trackwork. The noise levels are significantly increased at special trackwork due to the wheel impacts at the rail discontinuities. The same principle is true for short-radius curves. Such curves should be avoided, if possible, because of the wheel squeal. If they cannot be avoided, an effort should be made to locate them as far as possible from noise-sensitive areas.

Track profile can also provide acoustical effects. Generally, the lower the track, the better. Ballast-and-tie track in open cut will have relatively little acoustical impact. At-grade ballast-and-tie track will have more impact because there is direct line-of-sight between the noise source and the receiver. Having the track on a berm can be expected to increase the noise impact slightly, since the attenuation caused by ground effects is reduced. An aerial structure will impact a significantly greater area than at-grade ballast-and-tie track because the noise source is elevated above grade. The relative impact of at-grade and aerial structures can be strongly dependent on the design of the aerial structure. In some locations, the structure can be arranged to shield the direct path between the receiver and the transit car noise source. If structure-radiated noise is not a problem, then it is possible for the wayside noise at the shielded area to be lower than it would be with at-grade track.

6.1.2 Vehicle Design

The vehicle parameters that affect the wayside noise levels are the wheel condition and type, the propulsion equipment noise, and the auxiliary equipment noise. The manner in which the car acts to shield and absorb the noise is also important. Factors that do not appear to be particularly important are the truck design and aerodynamic noise. Since one of the main sources of wayside noise is the propulsion equipment and a secondary source is the auxiliary equipment, it is clear that the design of the vehicles will have a strong influence on the levels of wayside noise. In addition to minimizing the strength of the noise sources, it is possible to design features that will reduce the levels of noise radiated to the wayside. Two methods that have been successful are the use of deep side skirts on the cars and the use of absorption material under the cars. The side skirts trap the sound energy under the car, and the absorption material absorbs sound energy before it radiates to the wayside. The side skirts and absorption material, when used separately, can each reduce wayside noise by 2 to 3 dB. Used together, the reduction can be as high as 4 to 6 dB.

Specific examples of the use of deep side skirts and undercar absorption are presented in Chapter 5. Note that although side skirts have been successfully used in fleets of transit cars (BART is one example), undercar absorption has been used on an experimental basis only.

When evaluating the relative benefits of noise control treatments, it is important to recognize that modifications to the transit vehicles will benefit the entire system as well as patrons and the wayside communities. Although treatments such as sound barriers are often required in locations where specific neighborhoods must be protected, their benefits are inherently limited in scope. Modifications to the transit vehicles, although probably more expensive than dealing with individual wayside problems, could provide a more cost-effective solution in the long run because the benefits are systemwide.

6.1.3 Trackbed and Rail Support

With respect to noise generation, the most important rail and trackbed parameters are the number and size of rail discontinuities that cause impacts and the acoustic absorption of the trackbed. The type of rail fastening system, the weight of the rail, the type of rail (i.e., heat-treated rail compared to mild steel rail) appear to have relatively little effect on the noise levels, although heavier rail sections do tend to be quieter. There are some qualifications, e.g., the development of rail corrugations can increase wayside noise levels up to 10 dBA. Why specific sections of rail develop corrugations are still unknown, although it could be related to any of the rail parameters, including rail weight and hardness.

The effective damping of the rail is another factor that may have an influence on the noise radiated from the rail. The principal source of rail damping is the rail fastening system. Hence, a fastening system with a higher damping effect could result in reduced levels of wayside noise. This does not appear to be a particularly important point since most rail fastening systems used for at-grade track have approximately equivalent damping. Rails that are rigidly attached to lightweight elevated structures are an important exception and are further discussed in Chapter 7.

One factor that has helped to significantly reduce the impact of rail noise is the general trend towards the use of welded rail, which requires considerably less maintenance than jointed rail. Removing the joint impacts also reduces both the overall noise level by 3 to 8 dBA and changes the character of the sound. Because the "clickety-clack" of the joint impacts is removed, the passby noise is less noticeable and, to many people, much less annoying.

To reduce noise annoyance, trackwork should be designed so that rail discontinuities are minimized. This includes the use of welded rail, the use of epoxy bonding to reduce the magnitude of the impacts at insulated joints, and the controlled use of special trackwork. Although it would be difficult to justify deletion of special trackwork from the system design on the basis of acoustical considerations, a concerted effort should be made to locate it in areas that are not noise-sensitive. If jointed rail is used, the maintenance of the joints and the ballast can significantly reduce the impact noise. The joints should be maintained in good alignment, the ballast should be maintained in good condition to prevent settling of the ballast under joints, and the tie down bolts or spikes should be kept tight so rattling at joints is prevented.

The acoustic absorption of the trackbed can have a somewhat surprising influence on the noise levels. Ballast is a relatively good absorber of acoustic energy. As a result, noise levels at concrete trackbeds are typically 3 to 5 dBA higher than at ballasted trackbeds. Little acoustical benefit would be realized from increasing the absorption at a ballasted section of track. However, placing acoustical absorption treatment on concrete trackbeds can reduce noise levels by 3 to 5 dBA, a significant reduction.

6.1.4 Wheels

Wheel contour types (conical or cylindrical) do not appear to have a strong influence on the levels of wayside noise. In fact, even though conical wheels would be expected to reduce crabbing on curves and, hence, reduce wheel squeal, the difference does not appear significant.

Resilient wheels and damped wheels have been found to be very effective at controlling wheel squeal noise. However, the reductions of wayside noise levels on tangent track are small at best.

6.1.5 Wheel and Rail Surface Condition

The condition of the wheel and rail surface has a very strong influence on the levels of wheel/rail noise that are generated. The most important factors are whether the wheels have flats or the rail has developed corrugations, both of which can increase noise levels by up to 10 dBA compared to the noise levels with smooth wheels and rails. Another condition that occurs fairly often on European systems is "corrugated wheels." This condition is apparently caused by cast iron tread brakes and does not occur with composition tread brakes or disc brakes. Measurements of wayside noise levels with corrugated wheels have shown up to 10 dBA increases in noise levels.

Besides the effects of wheel flats and wheel or rail corrugations, shelling, spalling, and pitting resulting from normal wear can also cause discontinuities on the running surfaces of the wheels and rails. Regular wheel truing and rail grinding are important features of any transit system noise control program. Although these techniques have been shown to provide acoustic benefits even on unflatted wheels and uncorrugated rails, the major acoustic benefits are derived from the removal of the large-scale discontinuities.

As is discussed in Section 6.2, wheel/rail noise on unjointed tangent rail is largely caused by the small-scale roughness of the wheels and rails. Perfectly smooth rails would still cause noise from variations in the stress field at the contact point, the constant moving of the contact point, and variations in the surface hardnesses. However, the theoretical mechanism indicates that wheel/rail noise can be lowered by reducing the surface roughness (Ref. 6.2). In this situation, roughness refers to wavelengths in the range of about 1 mm to 500 mm. The micro-roughness of the surface has only a small effect on the noise levels.

Wheel truing and rail grinding can result in significant reductions of wheel/rail noise. However, the wheels or rails must be in relatively poor condition initially. Hence, grinding new rail that still has the mill scale or grinding corrugated rail will result in significant reductions of wayside noise. Grinding rail that appears relatively smooth will result in, at best, small reductions of wheel/rail noise.

6.1.6 Train Speed and Length

The levels of wayside noise are a function of both the train speed and the train length. The maximum levels are proportional to 20 to 30 log V, where V is the train speed. The maximum levels also increase with train length (Fig. 3.18). When the distance from the track is short relative to the train length, increasing the train length will result in only slight increases in maximum noise levels. When the distance from the track is much longer than the train length, each doubling of train length will double the radiated sound power and increase the maximum level by approximately 3 dB.

In many cases, specifications and noise ordinances give limits in terms of the maximum levels only. However, for an environmental assessment of transit noise and a comparison of the acoustical impact of rapid transit with that of other community noise sources, the energy equivalent level, L_{eq}, is usually a better metric. The total acoustic energy of a train passby is dependent on the maximum level, the train length, the distance from the track, and the train speed.

L_{eq} is a measure of the total sound energy which depends on the maximum level and the time history of the sound. If the train speed is constant, the level of L_{eq} will increase 3 dB if the train length is doubled or if the number of trains are doubled.

Since each transit car radiates essentially the same sound energy, the change in the level of L_{eq} can be related to the total number of cars irrespective of how the cars are distributed.

6.1.7 Sound Barriers and Other Obstructions

Any obstruction in the direct line-of-sight between a sound source and a receiver will result in some attenuation of sound. The attenuation is not generally significant unless most of the sound source is blocked from the receiver's view. Sound barriers placed close to the track can be a very effective means of reducing wayside noise. Indeed, once the vehicle design, route alignment, and structure configuration have been determined, sound barriers are often the only viable alternative for further reductions of wayside noise. Because of the importance of sound barriers in the control of wayside noise, a more complete discussion along with design guidelines is presented in Section 6.3.

In the general definition, a sound barrier is any obstruction which shields, or partially shields, the receiver from the sound source. Based on this definition, walls, earth berms, the sides of depressed cuts, the edge of an aerial structure, buildings, and any other obstruction of sufficient size can act as sound barriers. Unless the direct path between the source and the receiver can be blocked, a barrier should not be considered to provide any attenuation. When the barrier blocks the direct path, the sound only reaches the receiver by:

- diffracting over or around the edges of the barrier.
- passing directly through the barrier.
- taking a reflected path that acts to short circuit the barrier.

Each of these paths is discussed in more detail in Section 6.3.

A general rule of thumb with respect to barriers is that 5 dBA attenuation is relatively easy to obtain; 10 dBA attenuation can be achieved with careful attention to the barrier design; and 15 dBA is usually the physical limit in field installations of barriers, although it is difficult to achieve with practical designs. Barrier theory indicates that attenuations up to 25 or 30 dBA can be achieved with barriers, and laboratory experiments have generally proven the validity of the theory. However, the theory does not include the effects of temperature variations, air turbulence, ground effects, and reflections, all of which act to reduce the effectiveness of barriers.

6.1.8 Trackbed Absorption

A final factor that can influence the wayside noise is the acoustical absorption of the trackbed, particularly the area between the rails. The effect of trackbed absorption is clearly illustrated by the fact that wayside noise levels adjacent to ballast-and-tie track will be 2 to 4 dBA lower than those adjacent to a concrete trackbed when all other factors are the same. This is because ballast has relatively high acoustical absorption and concrete has very low acoustical absorption that reflects sound from under the cars to the wayside.

Adding absorption treatment to a ballasted trackbed would provide very little acoustical benefit. However, on a concrete invert trackbed, the wayside noise levels and the car interior noise levels would be reduced by 2 to 4 dBA. The effectiveness of trackbed absorption has been proven with experimental installations, but there are no successful permanent installations of trackbed absorption on North American transit systems. Those installations used in revenue service became rapidly contaminated with dirt and brake dust and were therefore considered unsuccessful.

In most circumstances, the 3 to 4 dBA attenuation that can be achieved with absorption treatment of a concrete trackbed is not sufficient to justify the cost. An exception is when the material is used in conjunction with a sound barrier and placed on the barrier to avoid the contamination problem.

6.2 WHEEL/RAIL NOISE

Wheel/rail noise is generally divided into three categories:

- Roar noise created by the interaction of wheels rolling on continuous sections of rail.
- Impact noise caused by rail joints, wheel flats, and other discontinuities in the wheel or rail running surfaces.
- Squeal noise created when trains traverse short-radius curves.

This section focuses on the existing state-of-the-art with respect to control of wheel/rail noise on tangent track. Although this chapter addresses wayside noise, the discussion is equally applicable to the levels of wheel/rail noise inside transit cars. Any method that reduces the levels of wheel/rail noise will reduce both car interior and wayside noise. This section does not cover wheel squeal, which is discussed in Chapter 12.

Wheel/rail interaction and the propulsion equipment are usually the dominant sources of wayside noise; the wheel/rail noise is often comparable to or slightly higher than the propulsion equipment noise. It is clear that any program to reduce wayside noise must consider wheel/rail noise. However, when propulsion equipment noise is comparable to wheel/rail noise, significant reductions of wayside noise usually require that both wheel/rail and propulsion equipment noise be treated. One reason that barriers are a very attractive noise control method is because they shield the receiver regardless of which noise source is dominant.

A considerable amount of UMTA-sponsored research has been recently completed regarding wheel/rail noise. The first project (Ref. 6.2) was a theoretical study of the mechanisms of wheel/rail noise. This study is a valuable investigation of the causes of wheel/rail noise and includes mathematical models for the prediction; however, it does not cover the practical aspects of control. The second study (Ref. 6.1) involves the field testing of four methods of controlling wheel/rail noise: rail grinding, wheel truing, use of resilient wheels, and use of damped wheels. The most important conclusions regarding wheel/rail noise on tangent track were: grinding rail and truing wheels that are in good condition (based on visual inspection) will not result in significant reductions of noise; resilient wheels and ring-damped wheels do not result in significant reductions of wheel/rail noise on tangent track even though they dramatically reduce the squeal noise on curves; and welded rail results in significantly lower noise than jointed rail.

The studies of wheel/rail noise have not yet revealed a solution for reducing the noise beyond maintaining wheel and rail running surfaces, keeping joints as tight as possible, and whenever possible, changing from jointed rail to welded rail. A large number of techniques, however, have been considered. The following focuses on some of the more promising methods that may provide reductions of wheel/rail noise in the future. Methods such as sound barriers and long side skirts that reduce the wheel/rail noise by preventing some of the noise from reaching the receivers are not discussed. The following discussions are relatively brief; more detailed review of the methods for controlling wheel/rail noise is available in Reference 6.3.

6.2.1 Rail Grinding and Wheel Truing

Rail grinding and wheel truing consist of removing rail corrugations, spalled and shelled areas on wheels and rails, wheel flats, and corrugations (spotting) from wheel running surfaces. Wheel corrugations are apparently caused by using cast iron brake shoes on transit cars with tread brakes. Although it is known that truing wheels to specific profiles will have an affect on wheel wear, the effects of profile on noise levels have not been determined.

Rail grinding and wheel truing are expensive procedures that would be difficult to justify strictly for their acoustical benefits. However, they also promote regular wear patterns and lead to longer life for wheels and rails, improved ride quality, and increased safety. Wheel truing is needed to maintain wheel flange contour and width, and rail grinding recontours the railhead. Because of these benefits, most transit systems have some sort of wheel truing and rail grinding programs.

6.2.2 Resilient Wheels

Resilient wheels have never been shown to provide more than a 2 to 4 dBA reduction of wheel/rail noise on tangent track which is, in itself, insufficient to justify the extra cost of resilient wheels. However, there are a number of other benefits of resilient wheels worth considering. Resilient wheels are very effective at controlling wheel squeal and have been shown to reduce groundborne vibration and wheel/rail interaction forces. Resilient wheels could be more cost effective than solid steel wheels because they may last longer and because the reduced shock loading on truck components may reduce wear. Reduction of wheel/rail noise on tangent track would be an additional benefit.

6.2.3 Damped Wheels

Damped wheels have much the same acoustical performance as resilient wheels. They are very effective at reducing wheel squeal noise, but only small reductions of noise on tangent track have been observed. There is some indication that the Krupp-tuned dampers may reduce impact and roar noise, but a test installation of these dampers is needed before their effectiveness can be fully quantified. Damped wheels are not likely to affect wheel, rail, or truck component wear.

6.2.4 Joint Maintenance

On jointed rail, the impact noise can be significantly reduced by a regular program of joint maintenance. This includes tightening of joint bars, regapping, and ballast dressing to avoid excessive movement of the ties on the ballast. In some situations, grinding of jointed rail will reduce the vertical misalignment at joints. One specific example is a series of tests that were performed at SEPTA (Ref. 6.1). After replacing the joint bars in an attempt to improve the alignment of the joints, a 1 to 2 dBA reduction in wayside noise was observed. Subsequent grinding was observed to further reduce the wayside noise another 1 to 3 dBA for a total of 2 to 5 dBA. Note that maintaining joints in good condition significantly reduces the rail end batter and the problems with loose bolts and ballast settling or degradation.

6.2.5 Smooth Transition Rail Joints

Various smooth transition rail joints have been developed which transfer the wheel loads from one section of rail to the other, while still allowing room for rail expansion and contraction. Because of the cost of such joints, they are rarely considered for jointed rail. However, they could be an alternative to bonded insulated joints.

6.2.6 Welded Rail

Welding rail is clearly the most effective method of removing joint impact noise. When placing new rail, either shop or field welding should be used whenever possible. Although most welded rail is installed because of the reduced maintenance, there are clear acoustical benefits also. Both the overall noise level is reduced and the character of the sound is changed since the clickety-clack noise at the rail joints is removed. Most people find noise from trains on welded rail less annoying than trains on jointed rail.

Some problems do exist with welded rails, such as replacing worn sections of rail or welds occasionally breaking. These maintenance problems, however, are offset by the elimination of joint maintenance and the reduced incidence of track degradation. At this point, it is generally accepted that welded rail has significant advantages over jointed rail. However, welded rail cannot be used on the older lightweight steel elevated structures or on new elevated structures unless designed to take the loads caused by expansion, etc.

6.2.7 Wheel Slip-Slide Prevention

Some of the newer transit cars have a slip-slide protection system to prevent the wheels from sliding on the rails. The system acts to minimize the sliding during deceleration and the spinning during acceleration. The main effect of these devices, with respect to wheel/rail noise, is a reduction in the incidence of wheel flats. As a result, the wayside noise levels will be reduced.

6.3 DESIGN OF SOUND BARRIERS

Sound barriers can be a very effective means of controlling wayside noise levels. A considerable effort has been expended on the development of mathematical models to predict their effectiveness at reducing environmental noise. Because of the variability of environmental effects and the difficulty in accurately accounting for the effects of the ground, barrier predictions remain relatively inaccurate.

Despite the problems with applying the complex theoretical models of sound barrier insertion loss, adequate barrier design can be developed following relatively simple design principles:

- The barrier must break the line-of-sight path between the noise source and the receiver and block all possible paths that the sound can travel from the source to the receiver.

- Open areas in the barrier, drainage holes, for example, should be kept as small as possible. They provide paths for the sound to "short circuit" the barrier.

- The barrier should be constructed of a material that is sufficiently heavy to control the transmission of sound through the barrier. In most cases, virtually any material that is sufficiently strong to withstand the wind loads and other structural requirements will also be sufficiently massive to control sound transmission through the barrier.

- Barriers must block both the direct path and any reflected paths between the source and the receiver.

- The most effective location for barriers is either close to the noise source or close to the receiver.

In almost all cases, these basic guidelines will result in a barrier with 5 to 8 dBA attenuation. To obtain greater attenuation, more care must be taken in the design of the barrier. In any case, the barrier configuration should be checked with barrier design charts to ensure that the design is adequate.

When designing a barrier, the first consideration is that all of the possible paths the sound can travel, from the source to the receiver, will be controlled. As illustrated in Figure 6.1, there are four basic paths. These are discussed below.

The direct path refers to sound that travels directly to the receiver without being affected by the barrier. For this path, the barrier does not block the line-of-sight and, therefore, provides no attenuation.

Sound waves that pass over the top or around the ends of the barrier are diffracted into the area that is visually shielded by the barrier. The amount of sound diffraction into the shadow zone is dependent upon the diffracted angle, illustrated in Figure 6.1. The greater the diffraction angle, the less the amount of sound energy that is diffracted into the shadow zone. The diffraction of sound is also dependent on the shape of the barrier. Generally, a wider barrier, such as an earth berm, is more effective than a narrow barrier. However, the improvement is slight and most barrier prediction methods ignore the effect of barrier shape.

Another path is the transmission of sound directly through the barrier. Generally, if the barrier is constructed of a solid material such as concrete and does not have any significant open areas, the sound transmitted through a barrier will be negligible. An often used rule of thumb is if the surface or area density, i.e., the material density multiplied by the thickness, is greater than 4 lb/ft^2, the transmission of sound through the barrier will be negligible compared to the sound energy diffracted over or around the barrier (Ref. 6.5).

The final path shown in Figure 6.1 is the reflected path. In special circumstances, sound can be reflected into the shadow zone of a barrier, thus acting to negate the effectiveness of the barrier. In Figure 6.1, this effect is illustrated by a reflection of an overpass. Reflected paths can also be important when they reduce the diffraction angle. This effect can be important for transit barriers, as illustrated in Figure 6.2. The reflection between the barrier and the side of the train can act to reduce the effectiveness of the barrier.

Predictions of attenuation achieved with barriers are based on the path length difference between the direct path and the diffracted path. Given the source, barrier, and receiver geometry illustrated in Figure 6.3, the path length difference is:

$$D = A + B - C$$

The distance (A+B) is the shortest path over the barrier's edge from the source to the receiver and C is the direct distance through the barrier from the source to the receiver. For rail transit, the source is usually assumed to be at axle height at the near rail or the track centerline.

For a given path length difference, the barrier attenuation is a function of the frequency spectrum of the source. Barriers are more effective at controlling high-frequency, short wavelength sound than low-frequency, long wavelength sound. Based on a typical frequency spectrum for rail transit noise, it is possible to construct curves of attenuation as a function of path length difference. Figure 6.4 presents curves that can be used to design barriers. The top curve is a theoretical curve based on an average railcar spectrum. Note that the theory predicts 5 dB attenuation even for very small path length differences, something that cannot be depended upon for field installations. Also shown on Figure 6.4 are curves that should be used for design predictions. Two curves are shown: the first one is a linearized design curve from the "Design Guide for Reducing Noise In and

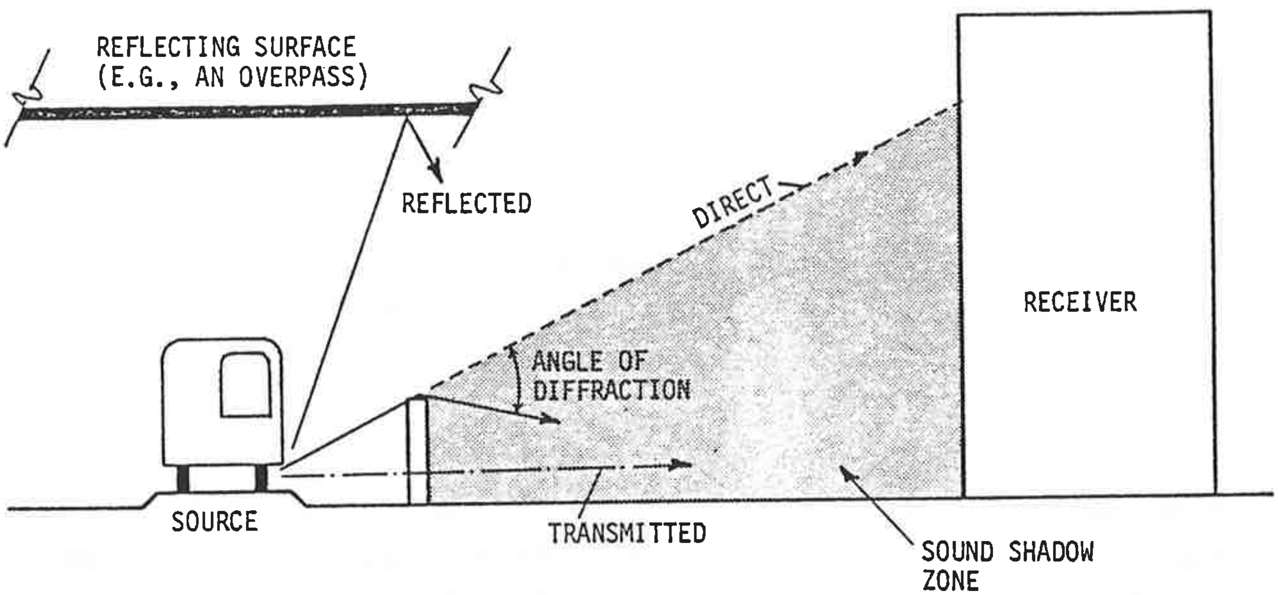


FIGURE 6.1 PATHS ALONG WHICH SOUND ENERGY CAN TRAVEL FROM THE SOURCE TO THE RECEIVER

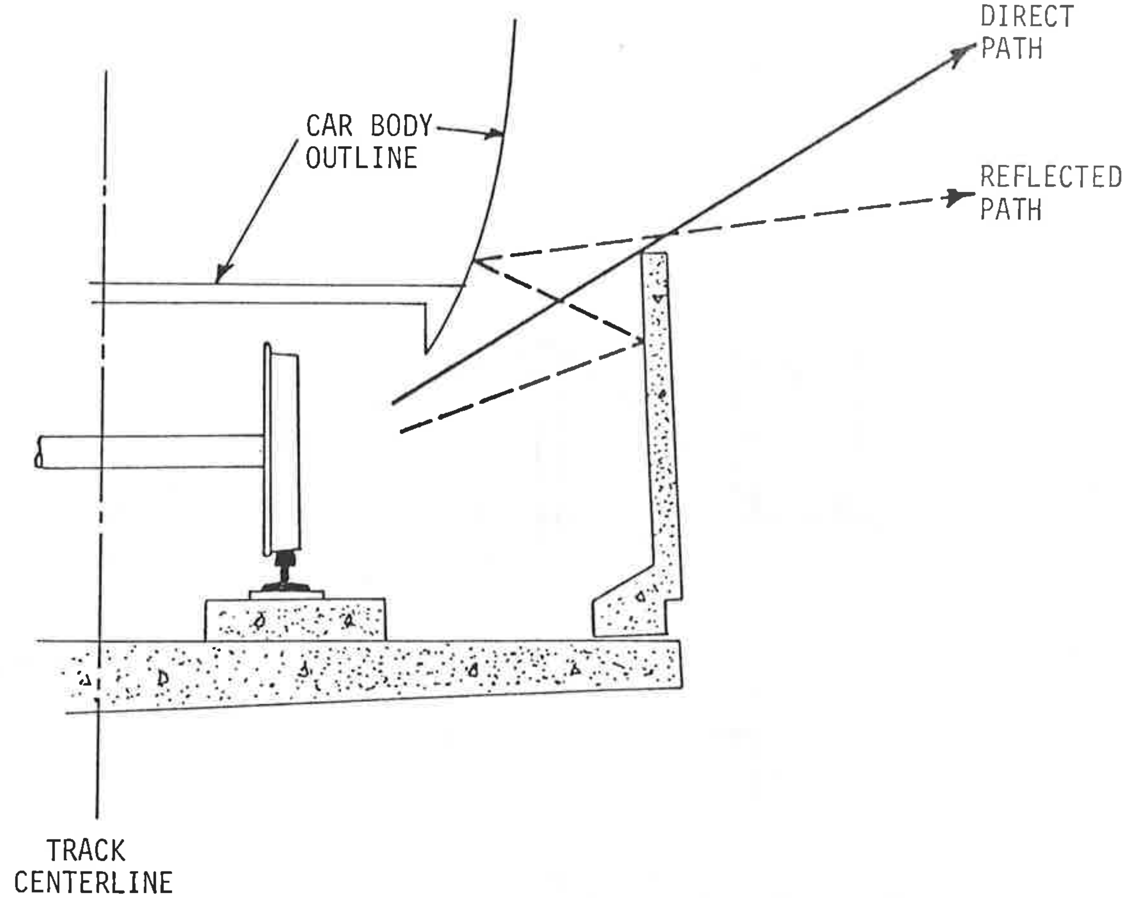
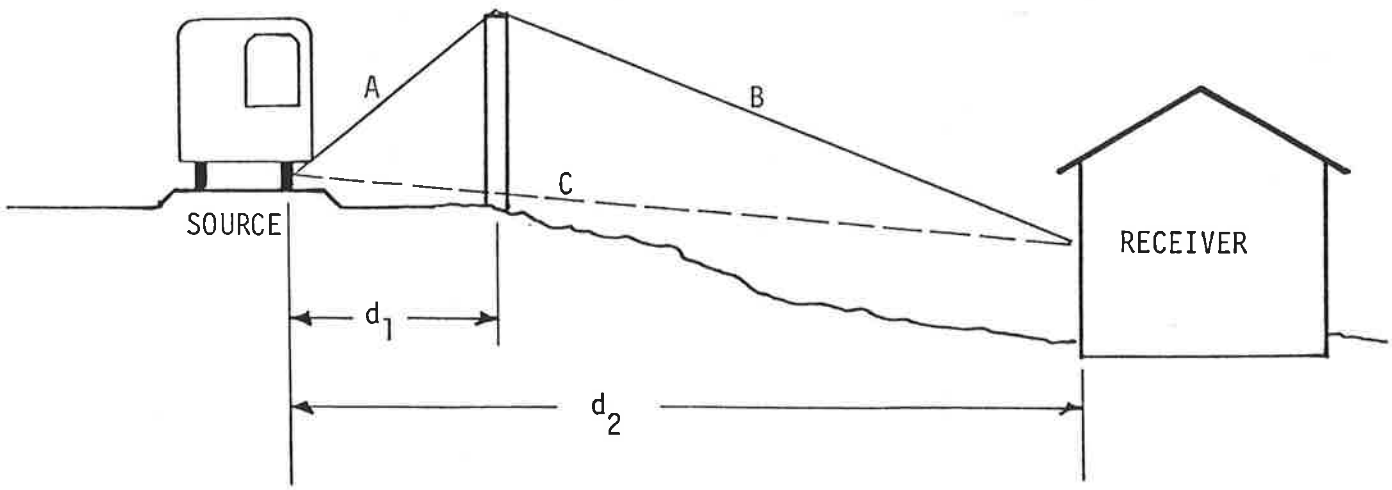


FIGURE 6.2 REFLECTIONS REDUCING BARRIER EFFECTIVENESS



$A = (H_B - H_S)^2 + d_1^2$

$B = (H_B - H_R)^2 + (d_2 - d_1)^2$

$C = (H_S - H_R)^2 + d_2^2$

$D = A + B - C$ = PATH LENGTH DIFFERENCE

H_S = SOURCE HEIGHT

H_B = BARRIER HEIGHT

H_R = RECEIVER HEIGHT

FIGURE 6.3 PATH LENGTH DIFFERENCE - D

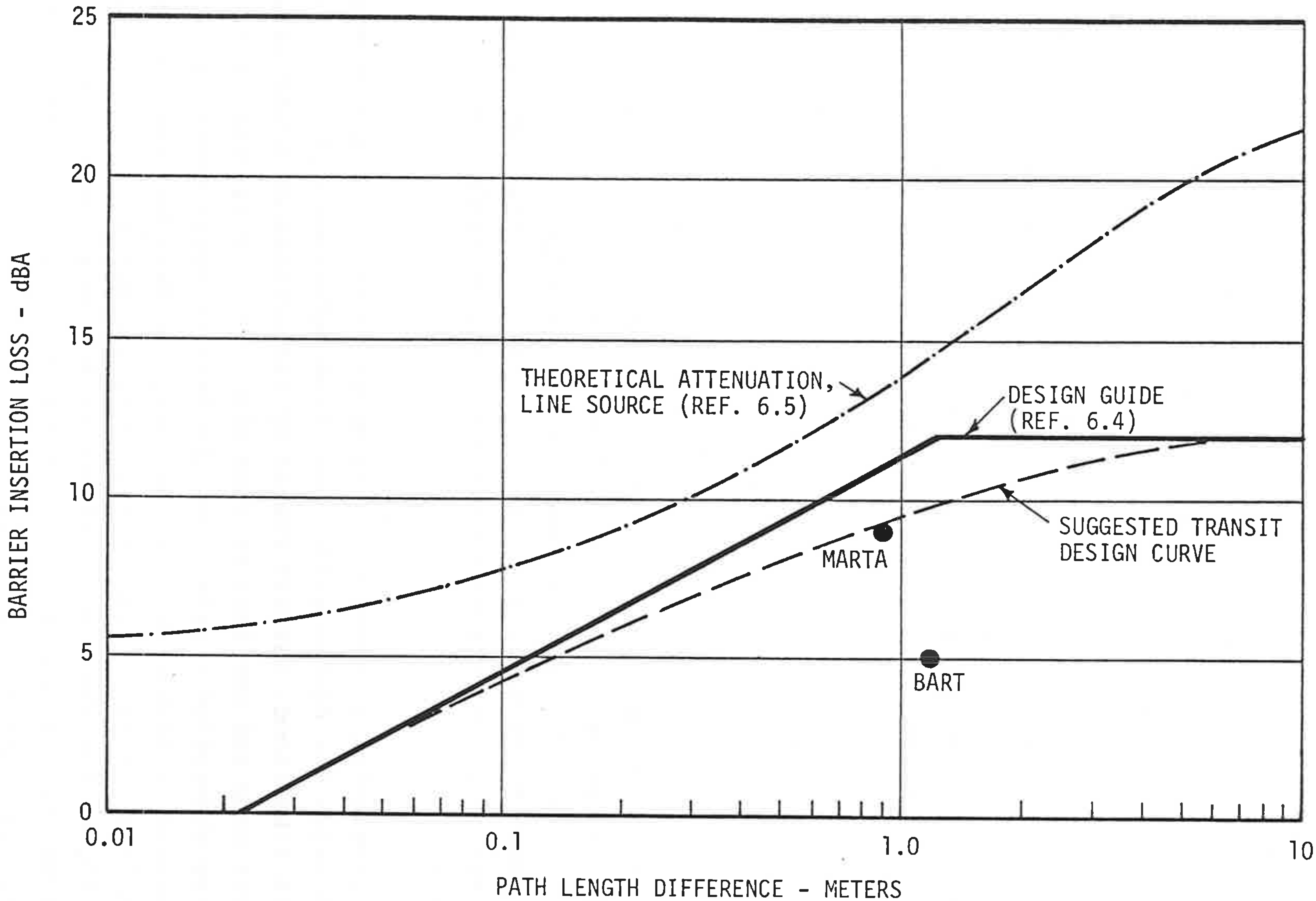


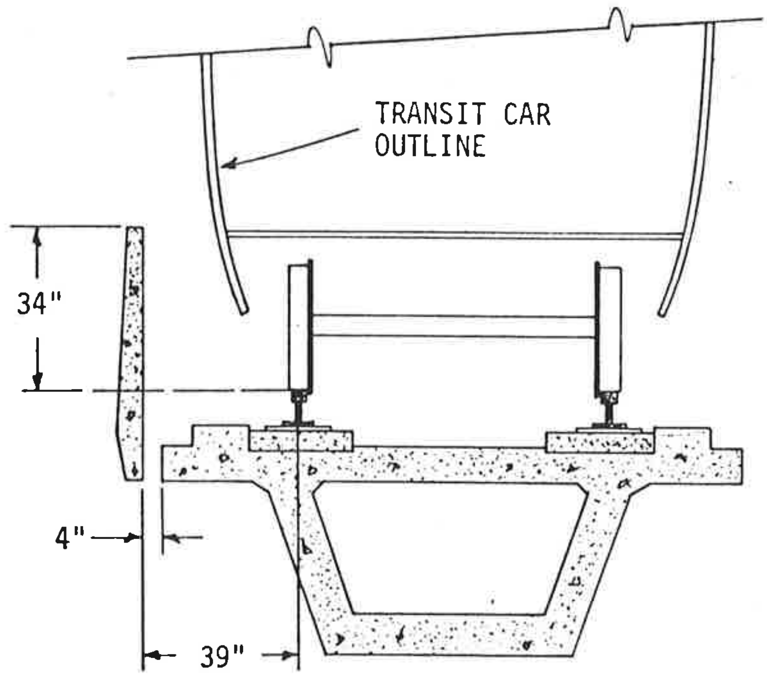
FIGURE 6.4 BARRIER ATTENUATION DESIGN CURVE

Around Buildings" (Ref. 6.4). This curve was developed to provide conservative estimates of barrier attenuation for a wide range of different transportation noise sources (e.g., highways and railroads). The second design curve is suggested for use with rapid transit systems. As indicated in Figure 6.4, the linearized design curve overestimates the insertion loss achieved with the MARTA aerial structure barriers. The suggested transit design curve has been drawn to reflect the attenuation measured at MARTA. Using this curve will result in a conservative barrier design which will be effective over a relatively wide range of meteorological conditions. Since wind, thermal gradients, and other conditions can significantly change the performance of a barrier, the theoretical curves should not be used directly for design purposes.

Shown on Figure 6.4 are the attenuations that were achieved with barriers on aerial structures at BART and MARTA. As discussed in Chapter 7, BART and MARTA have concrete deck aerial structures. With these structures, the noise radiated directly from the propulsion systems and from the wheels and rails dominates the wayside noise (structure-radiated noise is only significant at low frequencies). Hence, the sound barriers are just as effective as they would be on surface track (actually more effective because they can be placed closer to the cars than possible for surface track).

The attenuation achieved with the MARTA barriers is approximately 2 dBA below the design guide curve from Reference 6.4. This is, in all likelihood, largely caused by reflections of sound between the car body and the barrier (see Fig. 6.2). This shows that placing absorption treatment on the barrier would result in 2 to 3 dBA greater attenuation. In contrast, the BART barriers are 4 to 5 dBA less effective than predicted by the design curve. Figure 6.5 illustrates the cross sections of the aerial structures and the sound barrier walls. It is clear that the gap at the bottom of the BART barrier and possibly the gap between halves of the structure compromise the effectiveness of the BART barrier.

BART



MARTA

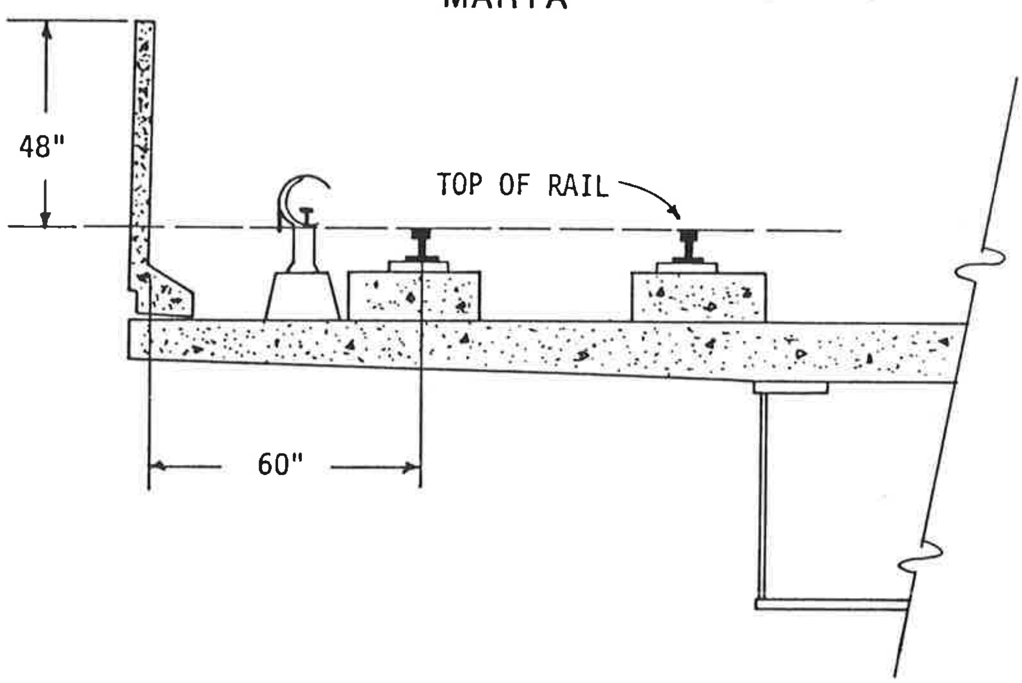


FIGURE 6.5 SOUND BARRIERS FOR AERIAL STRUCTURES, BART AND MARTA

The performance of the BART and MARTA barriers is shown in more detail in Figure 6.6. Measurements of the BART barriers, with the gap at the bottom plugged by rubber gasketing material, reduced the wayside noise levels further by 1 to 2 dBA. This illustrates the importance of designing barriers with minimized openings.

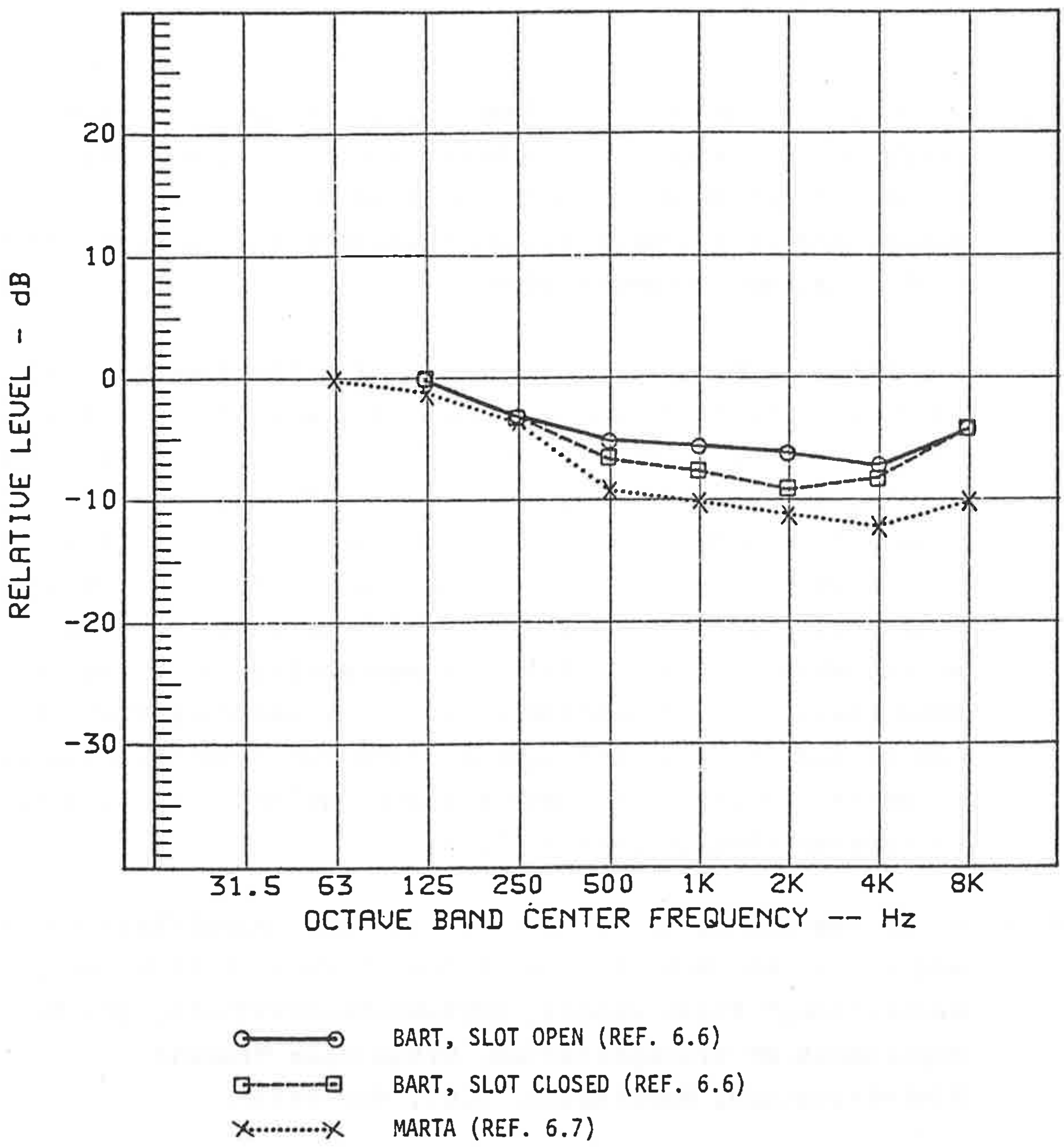


FIGURE 6.6 SOUND LEVEL WITH SOUND BARRIER WALL RELATIVE TO NO SOUND BARRIER - 15 m FROM TRACK CENTERLINE, 112 km/hr (70 MPH)

CHAPTER 6 REFERENCES

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The final report of a study evaluating the acoustic and economic effectiveness of controlling wheel/rail noise with rail grinding, wheel truing, resilient wheels, ring-damped wheels, and welded vs. jointed rail. The general conclusions regarding control of wayside noise are: grinding rail without large-scale roughness or visible corrugations will result in only small reductions of wheel/rail noise; truing wheels without visible imperfections will result in only small noise reductions; resilient and damped wheels can virtually eliminate squeal noise but result in little or no noise reduction on tangent track; welded rail is 2 to 5 dB quieter than jointed rail.

6.2a P. J. Remington, M. J. Rudd, I. L. Ver, "Wheel/Rail Noise and Vibration, Vol. I: Mechanics of Wheel/Rail Noise Generation," Final Report, UMTA-MA-06-002575-10, (U. S. Department of Transportation, Urban Mass Transit Administration, Washington, D.C., May 1975).

6.2b P. J. Remington, M. J. Rudd, I. L. Ver, "Wheel/Rail Noise and Vibration, Vol. II: Application to Control of Wheel/Rail Noise," Final Report, UMTA-MA-06-0025-75-11, (U. S. Department of Transportation, Urban Mass Transit Administration, Washington, D.C., May 1975).

These two volumes present the results of a research study into the mechanisms and methods for control of wheel/rail noise. Analytical formulas for the prediction of wheel/rail noise are presented; the formulas are at least partially verified by experiments. Some new devices for control of wheel/rail noise are suggested, and a number of old methods are evaluated using the information generated as part of the study.

6.3 L. G. Kurzweil, L. E. Wittig, "A Critical Evaluation of Wheel/Rail Noise," Paper presented at The American Public Transit Association 1980 Rapid Transit Conference, (San Francisco, CA, June 1980).

This paper briefly presents the results of an in-depth review of the available information on each of the known (or conceptualized) methods for controlling wheel/rail noise. Identified are requirements for further research, development, and testing in this area.

6.4 D. S. Pallett, R. Wehrli, R. D. Kilmer, T. L. Quindry, "Design Guide for Reducing Transportation Noise In and Around Buildings," Report No. NBS BSS 84, (National Bureau of Standards, Washington, D.C., April 1978).

This design guide presents a unified procedure for the selection of noise criteria in and around buildings and for the prediction of exterior and interior noise levels as a consequence of transportation systems including railroad and rail transit. This is a good, general guide to the evaluation of transportation-caused community noise problems. The guide is aimed at people with technical backgrounds who are not familiar with acoustical analysis.

6.5 U. J. Kurze, G. S. Anderson, "Sound Attenuation by Barriers," Applied Acoustics 4(1), (January 1971), pp. 35-63.

This is a widely referenced paper on the sound attenuation by barriers. Of particular interest is the evaluation of reduction of noise from a line source.

6.6 G. P. Wilson, "Aerial Structure Sound Barrier Walls," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Baltimore Metropolitan Transit Authority, January 1973).

This report presents measurement results of the effectiveness of the sound barriers on the BART aerial structures and presents recommendations for the sound barriers to be used at the Baltimore rapid transit system.

6.7 G. P. Wilson, "Vibration and Noise Control at MARTA," Presented at the 99th Meeting of Acoustical Society of America, (Atlanta, GA, April 1980).

This paper reviews the noise and vibration control features that were incorporated in the MARTA system. Of particular interest is the review of the effectiveness of floating slabs, sound barriers, and damping treatment to control low-frequency noise radiation from composite steel/concrete aerial structures.

CHAPTER 7

7. AERIAL STRUCTURE NOISE CONTROL

Train operation on lightweight elevated structures is one of the most severe environmental problems facing transit systems. The purpose of this chapter is to discuss the charactertistics of elevated structure noise, the methods of reducing noise levels from existing aerial structures, and the design features for prevention of excessive noise levels from new aerial structures.

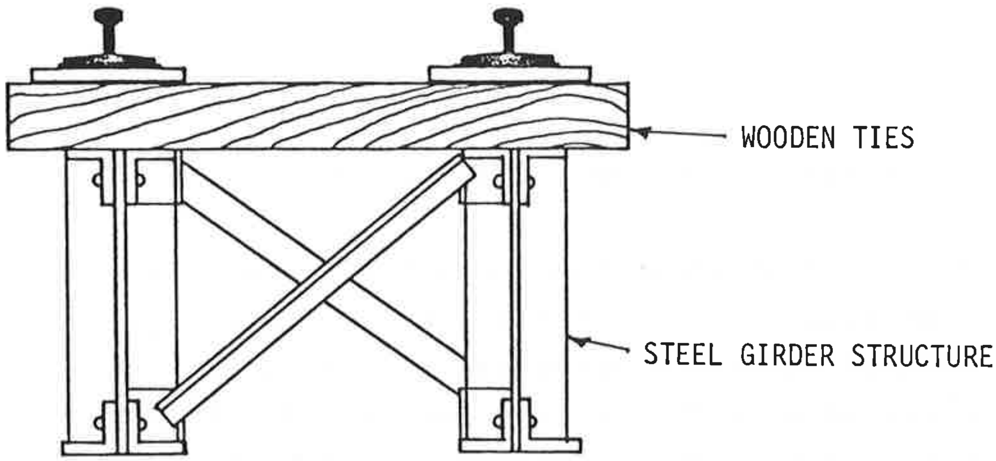
Aerial structure refers to any elevated guideway or bridge. The aerial structures used on North American transit systems can generally be divided into three broad classes, examples of which are shown in Figure 7.1.

The first is the relatively lightweight steel elevated structure with rail ties rigidly attached to the steel girder. There are many miles of these types of structures on older transit lines, particularly in New York and Chicago. This type of structure acts as a very large sounding board with very high noise levels radiated to the wayside community.

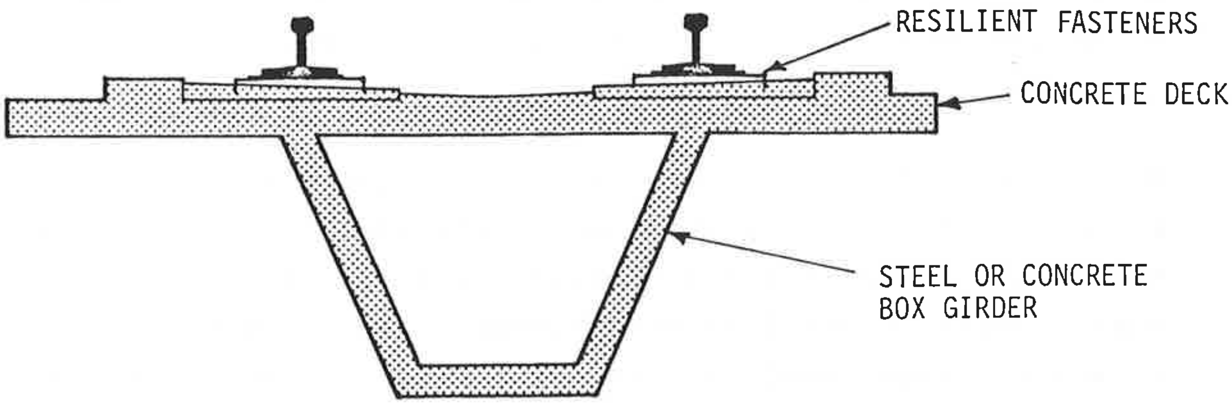
The second is a concrete deck supported by a steel or concrete girder and has entirely different characteristics. The rails are attached directly to the concrete deck with resilient direct fixation fasteners. As a result of the mass and inherent damping of concrete deck aerial structures, their acoustic characteristics are closer to at-grade track than to lightweight steel elevated structures.

The third category of aerial structure has ballast-and-tie track on top of the deck. This type of structure produces about the same noise characteristics with either concrete deck or steel deck and generally the noise is comparable to or less than that for the concrete deck aerial structure with direct fixation fasteners.

7.1a LIGHTWEIGHT STEEL ELEVATED STRUCTURE



7.1b CONCRETE DECK AERIAL STRUCTURE



7.1c BALLAST-AND-TIE TRACK ON CONCRETE OR STEEL DECK

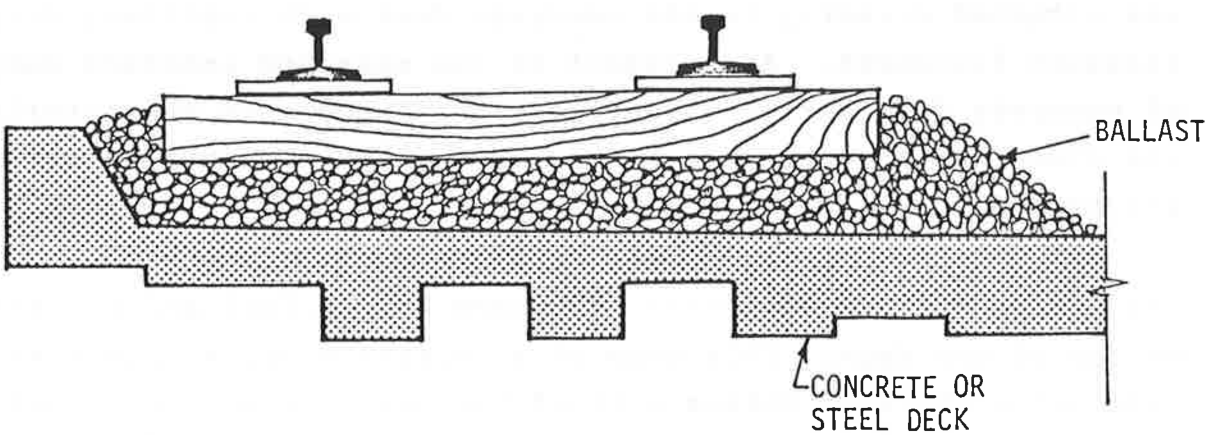


FIGURE 7.1 THREE BASIC CLASSES OF ELEVATED STRUCTURES

Thus, the noise characteristics of the ballasted track elevated structure are similar to the concrete deck aerial structure and produce much less noise impact than the lightweight steel elevated structure.

The large differences in the sound radiation characteristics of the lightweight steel and concrete deck types of aerial structures result in lightweight steel structures being referred to as "elevated" and concrete deck structures as "aerial". This is done to avoid public association of traditional elevated structure noise with proposed new sections of elevated trackway. The term "elevated structure" tends to be associated with the noisy, lightweight steel elevated structures.

As part of another TSC project, a survey was performed of elevated structure noise from U. S. rapid transit systems (Ref. 7.1). The results of this study, summarized in Table 7-1, provide some interesting insights about the problem of elevated structure noise. First, NYCTA and CTA account for 94% of the residential population exposed to L_{dn}'s greater than 75 dB. Virtually all of this noise exposure is caused by the steel elevated structures. In contrast, the newer transit systems account for relatively low levels of noise exposure, a result of the changes in the designs of elevated trackway structures. For Table 7-1, the statistics for BART, MARTA, WMATA, and PATCO have been combined. Of this group, BART accounts for approximately 67% of the route miles and 90% of the people in the table, indicating that the BART aerial structures create more noise impact than the other aerial structures. This is because BART has more route miles of aerial structure in residential areas than the other systems and not because of differences in aerial structure design.

Table 7-1 clearly shows that when considering aerial structure noise, there are two separate topics:

TABLE 7-1 ESTIMATES OF NOISE EXPOSURE OF WAYSIDE RESIDENTIAL POPULATION NEAR U.S. ELEVATED TRANSIT STRUCTURES (AFTER REF. 7.1)

Transit System	Number of People Exposed Within Various L_{dn} Ranges				
	55 - 65	65 - 75	75 - 85	85 - 95	Total
NYCTA	-	23,596	114,330	114,696	252,622
CTA	-	2,082	61,564	14,242	77,888
MBTA	539	1,246	285	-	2,070
SEPTA	-	4,253	23,497	-	27,750
MARTA, BART, WMATA & PATCO	16,509	1,643	-	-	18,152
TOTAL	17,048	32,820	199,676	128,938	378,482

- First is the control of noise from the existing lightweight steel elevated structures. Most of these structures are at NYCTA and CTA, although significant amounts also exist at MBTA and SEPTA.

- Second is the control of noise from aerial structures on new systems. Although the problem may be of a different order of magnitude in terms of the number of people exposed, it is very important to ensure that noise from new aerial structures will be compatible with the existing land uses in the transit corridor.

The control of noise from concrete deck aerial structures (and from ballasted deck structures) is almost identical with the control of noise from at-grade track. Over most of the frequency range, the radiation directly from the car, the wheels, and the rails is at least 10 dB higher than the noise radiated from the structure. Also, the structure does not significantly increase the noise radiated by the wheels and rails. The only time that structure-radiated noise becomes important is at low frequencies or when a sound barrier effectively controls the wheel/rail noise and the car auxiliary equipment noise leaving only the structure-radiated noise.

In contrast, the control of noise from lightweight steel elevated structures is a very complex topic. To control the noise from steel elevated structures, the first step should be to reduce the joint noise by changing to welded rail whenever possible, and to use rail grinding and wheel truing to maintain the wheels and rails in good condition. The next steps are less obvious. Although it is commonly believed that the principal noise source is the vibration of the steel girders, recent investigation indicates that the vibration of the rails, ties, and girder are all approximately equal sources of noise (Ref. 7.2). These conclusions, if verified, indicate that applying damping treatment to the girders is not an appropriate treatment for reduction of elevated structure noise.

Although there are examples of measures being taken to alleviate the noise problem, the research for this handbook has uncovered few examples of the successful control of noise from lightweight steel elevated structures. Design of successful noise control treatments is clearly an undertaking that requires the services of engineers experienced in the evaluation of noise and vibration problems and the design of treatments. It is important to recognize that for the older steel elevated structures, there may be no practical, economical solution to the wayside noise problem. Even partial solutions may be very expensive, and the only thorough solution may require that the structure be completely rebuilt or replaced.

The remainder of this chapter is divided into four sections. The first section (7.1) discusses factors that influence aerial structure noise, including a general discussion of noise control methods. Sections 7.2, 7.3, and 7.4 discuss the specific methods that can be used to reduce noise from lightweight steel elevated structures, concrete deck aerial structures, and ballasted deck structures.

7.1 PARAMETERS AFFECTING AERIAL STRUCTURE NOISE

As illustrated in Figure 7.2, there are four basic sources of community noise due to train operation on elevated structures:

- Noise radiated from on-car equipment such as traction motors, compressors, and motor generators.

- Wheel/rail noise radiated directly from the vibration of the wheels and rails. Wheel/rail noise includes impacts at rail joints, squeal noise on curves, and normal rolling noise.

- Noise radiated from the vibrating components of the structures.

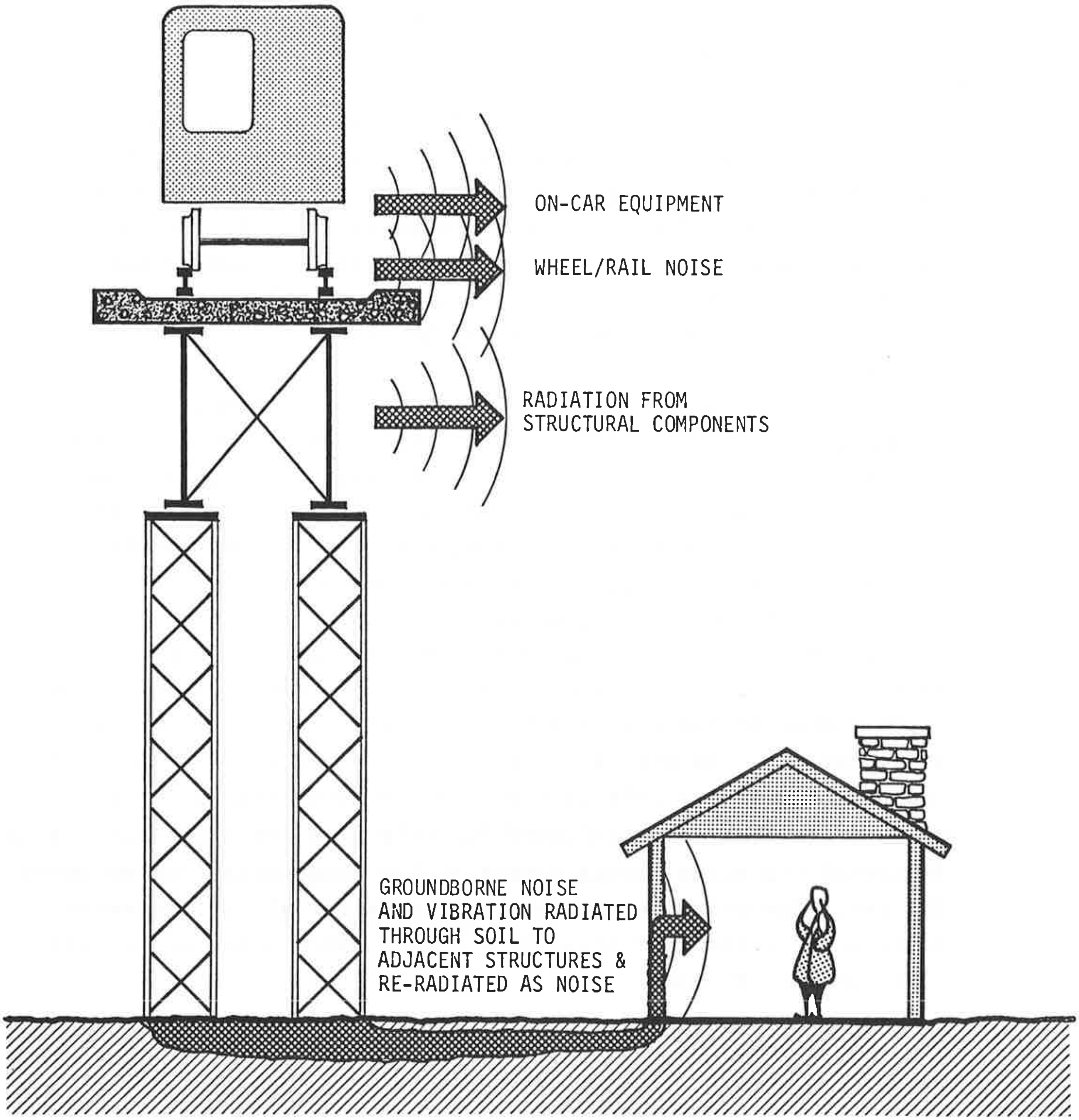


FIGURE 7.2 NOISE SOURCES FROM ELEVATED STRUCTURES

- Groundborne noise transmitted through the support columns to nearby buildings.

The noise from on-car equipment and the wheel/rail noise are largely the same for an aerial structure as for at-grade track. This chapter includes only cursory evaluation of these noise sources; for more detail refer to Chapters 5 and 6. Groundborne noise is a very detailed topic in itself and is covered separately in Chapter 8. However, it is relatively unusual to have groundborne noise problems from aerial structures.

This chapter focuses on the radiation of noise from structural components. However, recognition of the relative importance of the various noise sources before attempting to design abatement procedures is necessary. The importance of this recognition is exemplified by studies done on prototypes of the BART aerial structure design. Pertaining to the problem of structure-radiated noise, the noise-radiating characteristics of the structures were thoroughly studied and damping treatments were developed to control noise radiated from structures with steel girders. The study looked at the mid- and high-frequency range and the damping treatment was very effective in this range. However, from full-scale tests with transit cars at the BART Test Track, it was evident that the noise radiated directly from the trains and rails dominated the noise levels except at low frequencies (below about 125 Hz). The point is that one must consider all of the noise sources for control of noise from aerial structures, as for all noise control problems.

The main concern of this chapter is the reduction of noise radiated from the structural components of aerial structures. When a train operates on an aerial structure, the action of the wheels rolling on the rails results in vibration energy flowing through the rail fasteners into all components of the structure.

The level of the sound radiated from a specific component depends on the magnitude of vibration, the radiating area, and the efficiency with which the structure radiates sound. In the context of this chapter, structureborne noise refers to the process by which the vibratory energy caused by the wheels rolling on the rails ends up being radiated as sound by components of the aerial structure.

In contrast to airborne sound, structureborne sound can be transmitted by several different wave types, the most common being bending waves and compression waves. Shear waves and torsional waves can also be important in some cases. Airborne sound is only transmitted as compression waves in the atmosphere. The manner in which the vibration energy is transmitted will have a strong influence on the resulting noise radiation. Typically, bending waves radiate airborne noise much more efficiently than compression waves. The manner in which vibrational energy is distributed throughout the structure as structureborne noise significantly influences the design of effective noise control methods.

There are a number of different factors that influence the levels of structureborne sound. Following Kurzweil (Ref. 7.3), the factors that have applicability to rail transit systems, particularly to the reduction of aerial structure noise, can be divided into seven basic methods:

- Reduce the energy input to the system by reducing wheel/rail forces and vibration.

- Use vibration isolation to reduce energy flow into the structure.

- Apply vibration damping to major structural components to reduce the magnitude of vibration.

- Increase the mass against which the forces are acting.
- Reduce the noise-radiating area.
- Arrange shielding to block the noise path from the structure to receivers.
- Apply acoustic absorption.

These methods and their general applicability to aerial structures are discussed in the following paragraphs.

Reducing Wheel/Rail Forces: Since the interaction of the wheels and rails create vibration energy radiated as noise, any method which reduces the forces of interaction should result in reduced noise levels. Methods include rail grinding, wheel truing, use of resilient wheels, and modifications to the trucks. Also, any methods that reduce impacts such as slip-slide detectors to prevent wheel flats and using welded rail instead of jointed rail will significantly reduce the noise levels. The methods of reducing the wheel/rail forces are discussed in more detail in Chapter 6.

Vibration Isolation: The concept of vibration isolation is discussed in Appendix A. In the simple model of a single-degree-of-freedom spring mass system, the magnitude of the force transmitted to the base is dependent on the frequency of the applied force. Below the resonance frequency, the force is transmitted essentially unchanged to the base. At frequencies near resonance, the force is amplified with the amount of amplification depending on the amount of damping. Above resonance, the force transmitted to the base will be less than that applied to the mass. This basic concept can be applied to concrete structures with heavy decks; however, the model is considerably more complex for lightweight structures because they cannot be considered or modeled as rigid foundations. With improper design, the desired noise reduction may not be achieved, possibly increasing the noise level due to coupling between structural resonances and the vibration isolation system. Hence, design of an appropriate, effective vibration isolation system requires use of a multi-degree-of-freedom model and measurement of the necessary structural parameters that are inputs to the model.

Methods that have been used for vibration isolation of the rail from aerial structures include use of resilient fasteners, resiliently supported ties, ballast, and floating slabs.

Vibration Damping: Damping reduces the amplitudes of vibration by transforming vibrational energy into heat. Damping the steel components of elevated structures and railway bridges has been investigated by a number of researchers. Damping treatment essentially consists of applying viscoelastic material, usually with a restraining plate, to the components of a structure radiating the most noise. The frequency range affected by the damping treatment is predominantly determined by the characteristics of the damping materials. There is a wide enough assortment of commercial damping materials available to meet the requirements of almost any specialized requirement.

Another area where damping may play an important role in aerial structure noise is the damping of the rail vibration by the fasteners. It has been proposed that the noise reduction achieved with the use of resilient direct fixation fasteners on lightweight elevated structures is a result of the internal damping of the fasteners and not the vibration isolation (Ref. 7.4).

Mass Addition: One of the principal reasons that the concrete aerial structures are much quieter than all steel structures is the mass of the concrete which the force must act against. For a simple single-degree-of-freedom, spring-mass system driven by a force at a frequency well above the resonance of the system, the force transmitted through the system is inversely proportional to frequency times mass. Hence, increasing the mass in the high-frequency range will reduce the vibration amplitudes and the resulting radiated noise. The simplest model gives a reduction of 6 dB for each doubling of mass of the affected structural component. Clearly, it is not economically or structurally feasible to greatly increase the mass of the elevated structures for the purpose of noise control. However, mass addition can be feasible for specific components, especially since the methods used to increase the mass will generally result in increased damping (e.g., use of ballasted track or all concrete structure vs composite steel/concrete). Change in effective mass is a realistic consideration in the design phase of new structures but may not be appropriate as a retrofit procedure.

Reduction of Radiating Area: Reducing the area that radiates noise will reduce the resulting noise levels. This is primarily a consideration for new construction, e.g., minimizing large plate girders using narrow deck or building two physically isolated elevated structures instead of one double-track structure. Although reducing the radiating area could be a feasible noise control method in specific cases, we are not aware of any case history of this technique being used to reduce elevated structure noise.

Acoustic Shielding: The primary examples of this technique are sound barrier walls. On concrete elevated structures where the noise radiated directly from the train and from the wheels and rails dominates, using a barrier to shield the receiver from the radiated noise can be very effective. However, when the structure itself radiates the noise, reducing the noise by acoustic shielding requires shielding the structure components that radiate noise, i.e., usually the entire structure (which is rarely feasible).

Acoustic Absorption: Absorption occurs whenever a sound wave is reflected from a surface. At an acoustically hard surface, such as concrete or steel, almost 100% of the incident acoustic energy is reflected. At a surface with high acoustic absorption such as acoustical tile or glass-fiber insulation, only a small portion of the incident sound energy is reflected.

As an example of the effects of acoustic absorption, noise levels on ballast under similar operating conditions are 3 to 4 dBA quieter than on concrete trackbed because ballast has a much higher absorption coefficient than concrete. Similarly, adding absorption to the inner face of a sound barrier on concrete deck aerial structure can further reduce noise by 2 to 3 dBA. Chapter 6 has more information about the effectiveness of acoustic absorption.

7.2 CONTROL OF NOISE FROM LIGHTWEIGHT STEEL ELEVATED STRUCTURES

Noise from lightweight steel elevated structures exposes large numbers of people to very high noise levels. The noise affects both the wayside communities and the passengers. Maximum noise levels at the wayside of over 100 dBA are not uncommon. Noise levels inside the transit cars can increase 10 to 15 dBA as the train moves from at-grade, ballast-and-tie track to an elevated structure. Given the high noise levels created by these structures as well as their worldwide prevalence as transit system elevated structures and railway bridges, it is not surprising that control of their noise has received a considerable amount of attention.

There are relatively few examples of lightweight steel elevated structures, such as those in Chicago and New York, being modified to the point where the noise impact was removed. Some effective modifications, however, can be undertaken. The first step is to put the wheels and rails in as good condition as possible. The next step is to change from jointed rail to welded rail which substantially alleviates community annoyance. It is not always possible to install welded rail on existing steel elevated structures because of problems with anchoring rail and the stresses that are caused by expansion of the rail. However, whenever possible, welded rail should be used. Even field welding of 60% to 70% of the joints will result in a noticeable improvement. It is important to recognize that even when the impact noise only slightly increases the overall noise level, the impacts make the noise much more noticeable. This effect should not be underestimated. It is much the same as having a rattle in an automobile caused by something such as a loose muffler. The rattle can be very annoying but may not have a significant effect on the overall noise level. The next step is to use resilient direct fixation fasteners instead of attaching the rail directly to the structure. This will reduce the vibration energy transmitted from the rail into the structure and, because of its damping characteristics, will absorb some of the vibration energy. A noise reduction of 3 to 4 dBA was observed at an experimental installation of resilient rail fasteners at NYCTA.

These three basic steps will serve to markedly reduce the noise impact of the elevated structures. However, they cannot be expected to cure the problem. Further noise reductions require obtaining information about the relative strengths of the various noise radiators. For example, applying damping treatment to steel girders will not help if the dominant source of noise is the vibration of the rails. Before an effective treatment can be designed, it is necessary to identify the noise source. Often, the most effective methods of isolating the noise sources are to selectively apply various noise control methods to test sections of elevated structure.

Information on techniques that have been used to reduce elevated structure noise and the measured results are listed in Reference 7.3. Some of the specific methods that may have applications to U.S. transit systems are:

- Damping Treatment: Damping treatment is generally applied to steel plates of the structure. The general use of damping treatment is discussed in Reference 7.9. The effectiveness of specific treatments to rail structures is discussed in References 7.5 to 7.8. Optimization of the noise-reducing capabilities of a restrained layer-damping system involves: selection or development of a synthetic damping material with high internal damping characteristics as well as good corrosion resistance and good long-term mechanical and thermal performance characteristics; determining the relative thicknesses of the restraining layer, the damping material, and the base panel to give damping which is effective over the appropriate frequency and temperature ranges; and developing a system to attach the damping material and restraining panel to the elevated structure components. In order to achieve maximum noise reduction, all of the above parameters must be considered as they apply to a particular elevated structure configuration. Experience with restrained layer-damping has proven that middle and higher frequency noise can be reduced by up to 15 dB (Ref. 7.5).

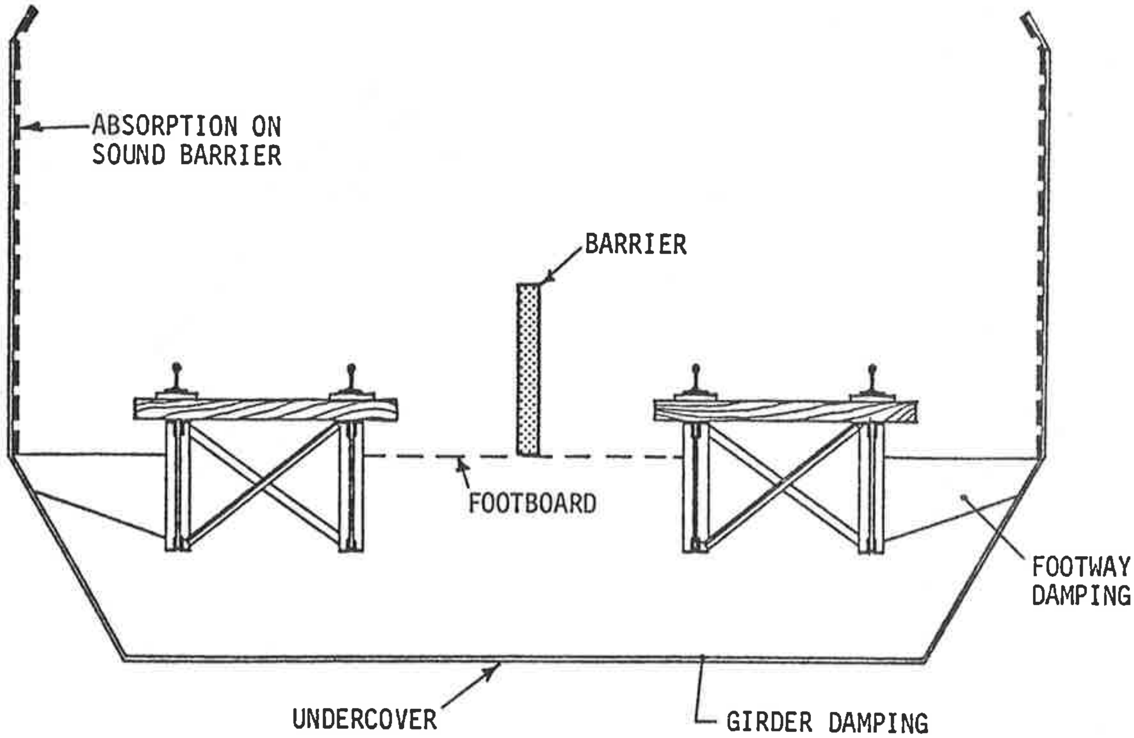
  Other factors requiring careful design consideration include the corrosion behavior of the steel girders in the vicinity of the damping material, especially the fastening system, and the long-term performance of the restraining layer system under mechanical vibrating loads when affected by temperature aging. A solution to the problem of corrosion at the point of attachment has been found through the use of a fastening element manufactured by Hilti (Ref. 7.10).

- Addition of Concrete Deck or Ballast: If the structures can take the extra weight, significant noise reductions can be achieved by adding a concrete deck or ballasted trackbed on top of the structure. The extra mass and damping will completely change the character of the sound. However, this is rarely a viable option for existing structures.

- <u>Ballast Mats</u>: If the structure already has a ballasted trackbed, JNR testing has shown that ballast mats can significantly reduce the structure-radiated noise while also improving the durability of the ballast (Ref. 7.6).

- <u>Sound Barriers</u>: Sound barriers can be a very effective method of reducing the noise. To be effective, all of the important noise-radiating elements must be blocked. Figure 7.3, summarizing some JNR tests, is an example of how a sound barrier might be used on a steel elevated structure. First, a sidewall was placed to block the sound directly radiated from the wheels, rails, and car. The noise reduction of 7 dBA indicates that these were the dominant noise sources. The damping of the footway and the main girder provided only 2 dBA reduction. The most dramatic reduction, 19 dBA, was achieved with the final treatment, the damping of the sound barriers, and placing a high transmission loss cover under the structure. Effective barriers on most lightweight elevated structures require both vertical sidewalls and a high sound insulation factor undercover.

Identification of the relative source strengths is an important step in the design of noise control treatments for elevated structures. Unfortunately, there are no simple rules on how the evaluation of source strength should proceed. One example is given in Reference 7.4: using a relatively complex analytical model of elevated structure noise radiation and some measurements of the vibration magnitudes of various components of the structure, the relative source strengths for the NYCTA steel girder elevated structures were estimated. Figure 7.4 is one result of this evaluation. It is of considerable interest that



TREATMENT	REDUCTION OF WAYSIDE NOISE	COMMENTS FROM REFERENCE 7.3
1. Sidewall (no absorption) plus footboard below ties	7 dBA	Since sidewalls reduce sideline noise by 7 dBA, the direct radiation from the wheels, rails, and car must have dominated the structureborne noise for the untreated structure. Also, since the main girder and footway damping (after reduction of the wheel/rail radiation) only reduce the level by another 2 dBA, the direct and secondary radiation after Treatment 1 were probably about equal. Comparison of acoustic and vibration spectra indicate that after Treatment 3, the radiation is probably due to vibration of the sidewalls. This points up the need for isolating the walls from the structure.
2. Treatment 1 plus damping of footway and girders	9 dBA	
3. Treatment 2 plus barrier between tracks plus absorption on barrier and sidewalls plus resiliently hung high TL undercover	19 dBA	

FIGURE 7.3 JNR TREATMENT OF LIGHTWEIGHT STEEL ELEVATED STRUCTURE (ADAPTED FROM REF. 7.11)

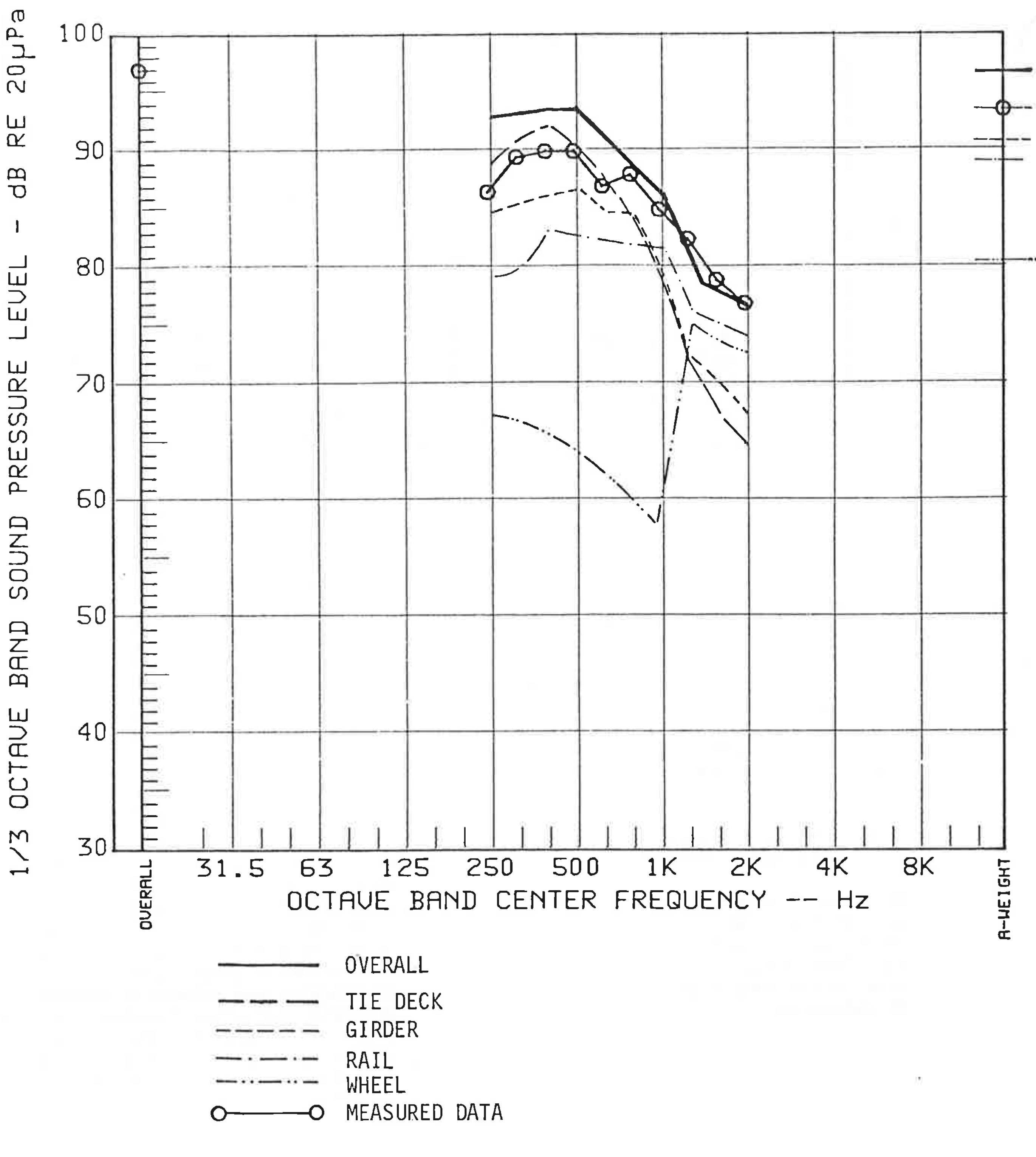


FIGURE 7.4 ESTIMATES OF RELATIVE SOURCE STRENGTHS, 10-CAR TRAIN AT 35 km/hr ON NYCTA 10TH AVENUE ELEVATED STRUCTURE (ADAPTED FROM FIG. 40, REF. 7.4)

the rail, the tie deck, and the girder were all identified as major sources of noise. The implication is that to achieve significant noise reductions, all of these noise sources must be treated.

As is always the case with analytical models, the results of Figure 7.4 require experimental evaluation before they can be used for major design decisions.

Another approach to the evaluation of source strength is to use some of the sophisticated experimental techniques, such as modal analysis, to develop very detailed information about the vibration modes and transfer functions of the entire structure and individual components of the structure. The basic technique consists of exciting the structure with a large shaker or impact and measuring the input force and resulting vibration. The analog signals are digitized and the transfer functions between the two measurement locations derived with a digital computer. Depending on the measurement locations selected, the types of transducers, and the computer software, the same basic technique can be used to:

- Evaluate mechanical impedances.
- Perform modal analysis.
- Evaluate energy flow through the structure.
- Determine a transfer function between an input force and resulting sound pressure level.

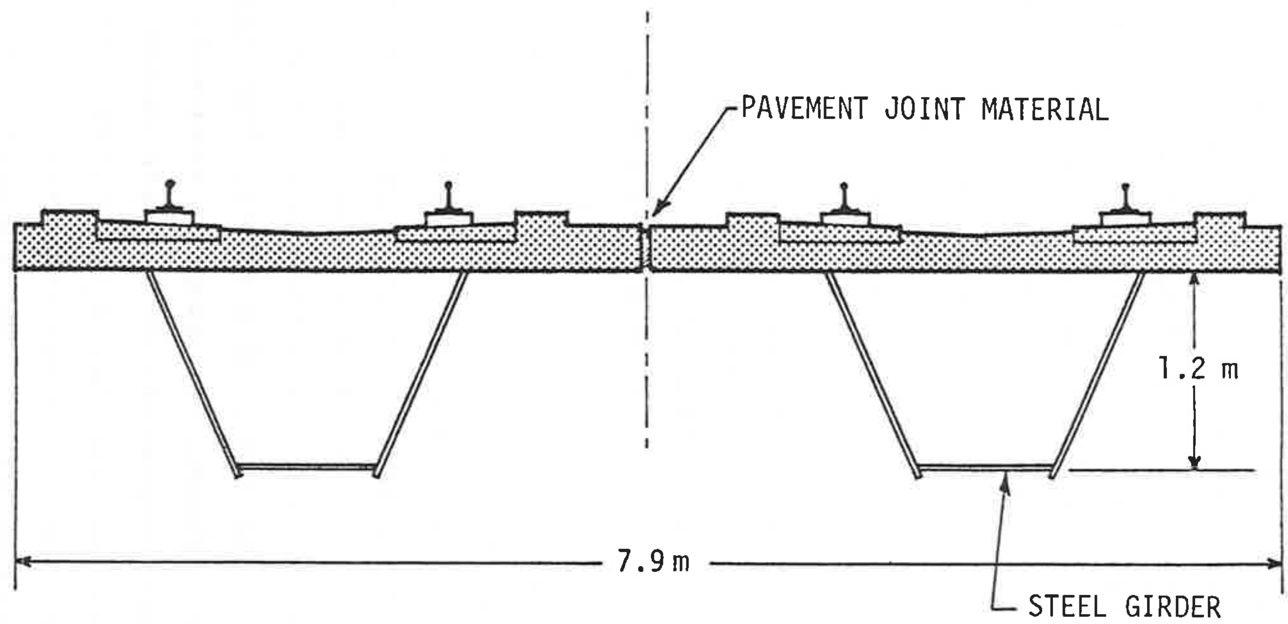
Since the same technique can be used to evaluate an entire elevated structure or individual components of the structure, the analysis can be extended to relatively high frequencies. This is a method that has proven to be an important diagnostic tool for many industries. However, it should be noted that it does not result in direct information about source strengths. Once the analysis is complete, the results will have to be carefully evaluated to infer the relative source strengths.

7.3 CONTROL OF NOISE FROM CONCRETE DECK AERIAL STRUCTURES

With most concrete deck aerial structures, the noise problems and the noise reduction methods are essentially the same as for at-grade track. With welded rail, resilient direct fixation fasteners, concrete girders, and sound barrier walls, concrete deck aerial structures can be placed even in quiet residential areas without extensive noise impact. However, the problem of noise impact can occur with all concrete structures. This has occurred on the high-speed Shinkansen lines in Japan (Ref. 7.6) and has the potential of occurring on aerial structure designs such as the BART, WMATA, and MARTA designs using concrete deck in various configurations (Ref. 7.13).

Most of the noise problems with the modern concrete deck aerial structures relate to low-frequency noise radiation from composite steel/concrete structures as shown in Figure 7.5. Although some discussion remains about the source of the noise from such structures, the bulk of the evidence indicates that the noise is radiated by the vibration of the panels of the steel girders (Ref. 7.13, 7.14). Hence, damping treatments designed to control the low-frequency noise radiation have been recommended when composite steel/concrete aerial structures are to be located near noise-sensitive land uses. It is important to recognize that because the damping treatment is designed to control low-frequency noise, and the A-weighted sound levels are dominated by the mid-frequency range, the damping treatment will not have a significant effect on the <u>outdoor</u> A-weighted sound level. The effect will be much more pronounced inside buildings adjacent to the structure. The building walls and ceiling do not reduce low-frequency noise as efficiently as high-frequency sound. Hence, the low-frequency sound has greater relative importance once the sound is transmitted indoors.

Another factor to be aware of is that the A-weighted level tends to underestimate the annoyance quality of low-frequency sound. Even though the presence of the low-frequency sound may result in



APPROXIMATE TOTAL WEIGHT = 2680 kg/m (1800 LB/FT)

FIGURE 7.5 BART COMPOSITE STEEL/CONCRETE AERIAL STRUCTURE -- WALNUT CREEK BRIDGE only a 2 to 3 dBA increase in indoor sound level, it may be the cause of a significant increase in annoyance to building occupants.

The newly constructed MARTA aerial structures are the most recent example of vibration damping being used on a U.S. transit system (Fig. 7.6). The original recommendation was for approximately 80% coverage of the steel girder plates with constrained layer-damping. Based on a research project sponsored by the American Iron and Steel Institute, an optimized damping configuration was developed that reduced the required damping area to approximately 40 percent (Ref. 7.15). Details of the design are shown in Figure 7.7.

Shown in Figure 7.8 are summary results of measurements performed on the damped and undamped MARTA aerial structure. Figure 7.8a illustrates the vibration velocity level of the side plate of the steel girder and Figure 7.8b presents the sound levels at 50 ft from the structure. Figure 7.8a clearly illustrates that above 16 Hz the damping treatment is very effective at reducing the vibration amplitude of the steel girders. However, as can be seen in Figure 7.8b, there is only a small reduction in the wayside sound level. In this particular example, the only significant reductions are in the 20, 25, and 50 Hz 1/3 octave bands. There are several factors which may contribute to the reduction of wayside noise level being small even though the damping achieves a large reduction of girder vibration. At high frequencies (above 100 to 200 Hz), the noise radiated from the aerial structure is significantly lower than the noise radiated directly from the wheels, rails, and transit train. Hence, the damping treatment does not affect the high-frequency components of wayside noise. At low frequencies, the structure-radiated noise dominates. On a steel girder, aerial structure with a single-track width concrete deck, the steel girder usually dominates the low-frequency radiated noise. However, the MARTA structure has a double-track width concrete deck. This increases the flexibility and the sound radiating area of the deck and hence increases the level of noise radiated from the concrete deck relative to the level

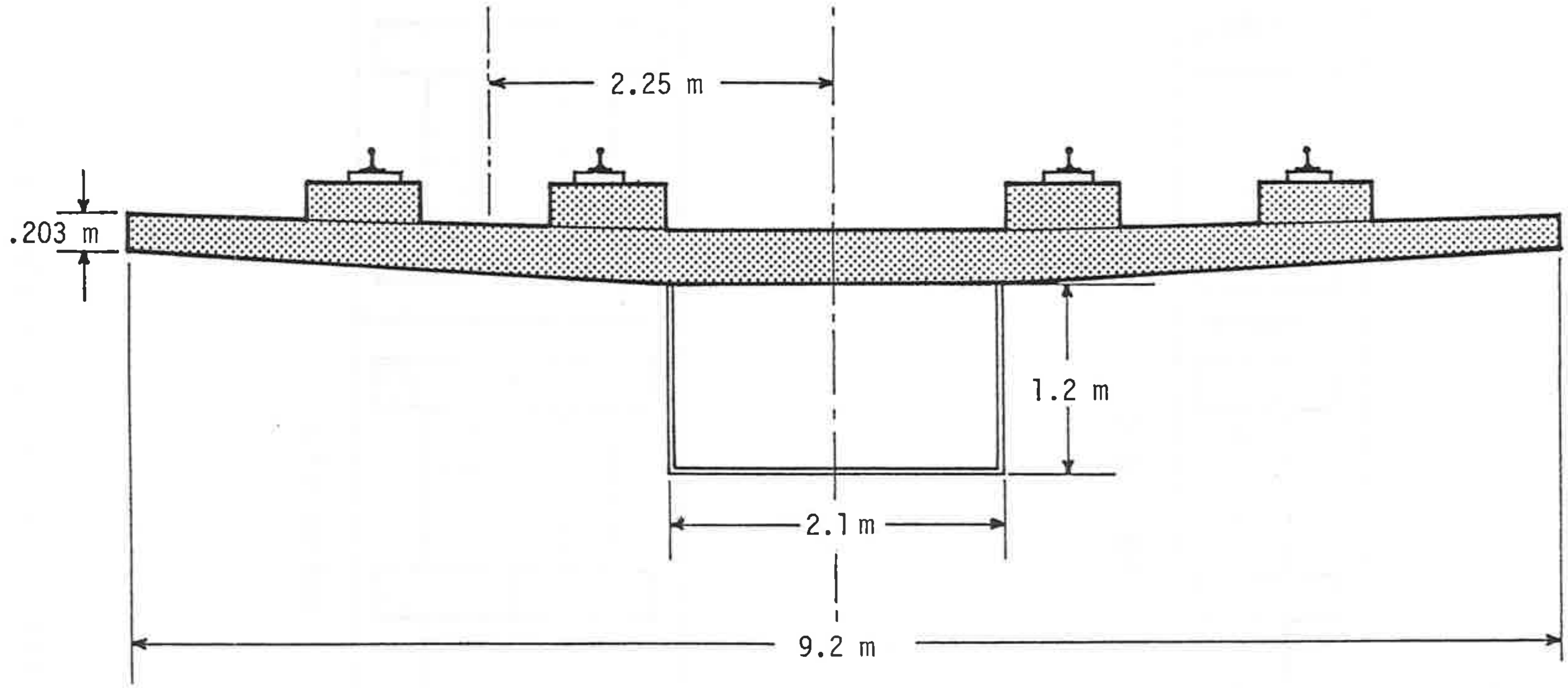
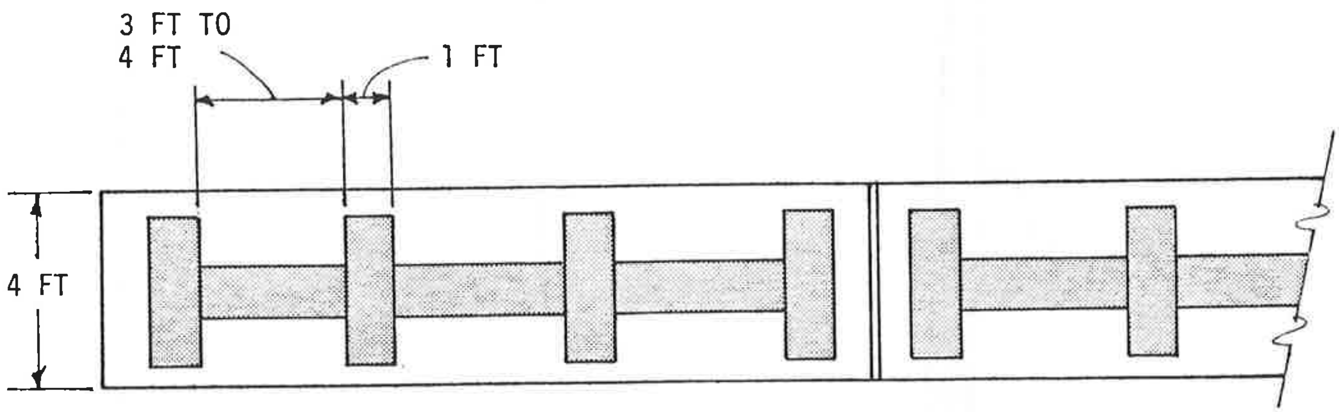
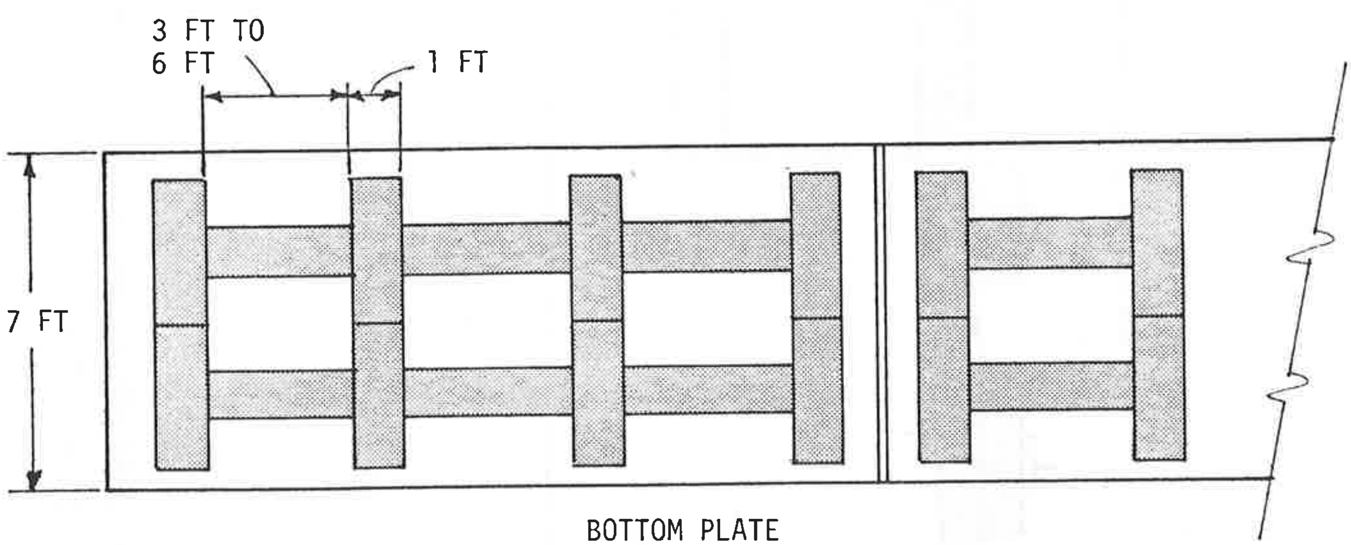


FIGURE 7.6 CONFIGURATION OF MARTA COMPOSITE STEEL/CONCRETE AERIAL STRUCTURE, 21 m (70 ft) SPAN



SIDE PLATES



BOTTOM PLATE

FIGURE 7.7 DAMPING TREATMENT PATTERN USED ON STEEL GIRDERS OF MARTA COMPOSITE STEEL/CONCRETE AERIAL STRUCTURES

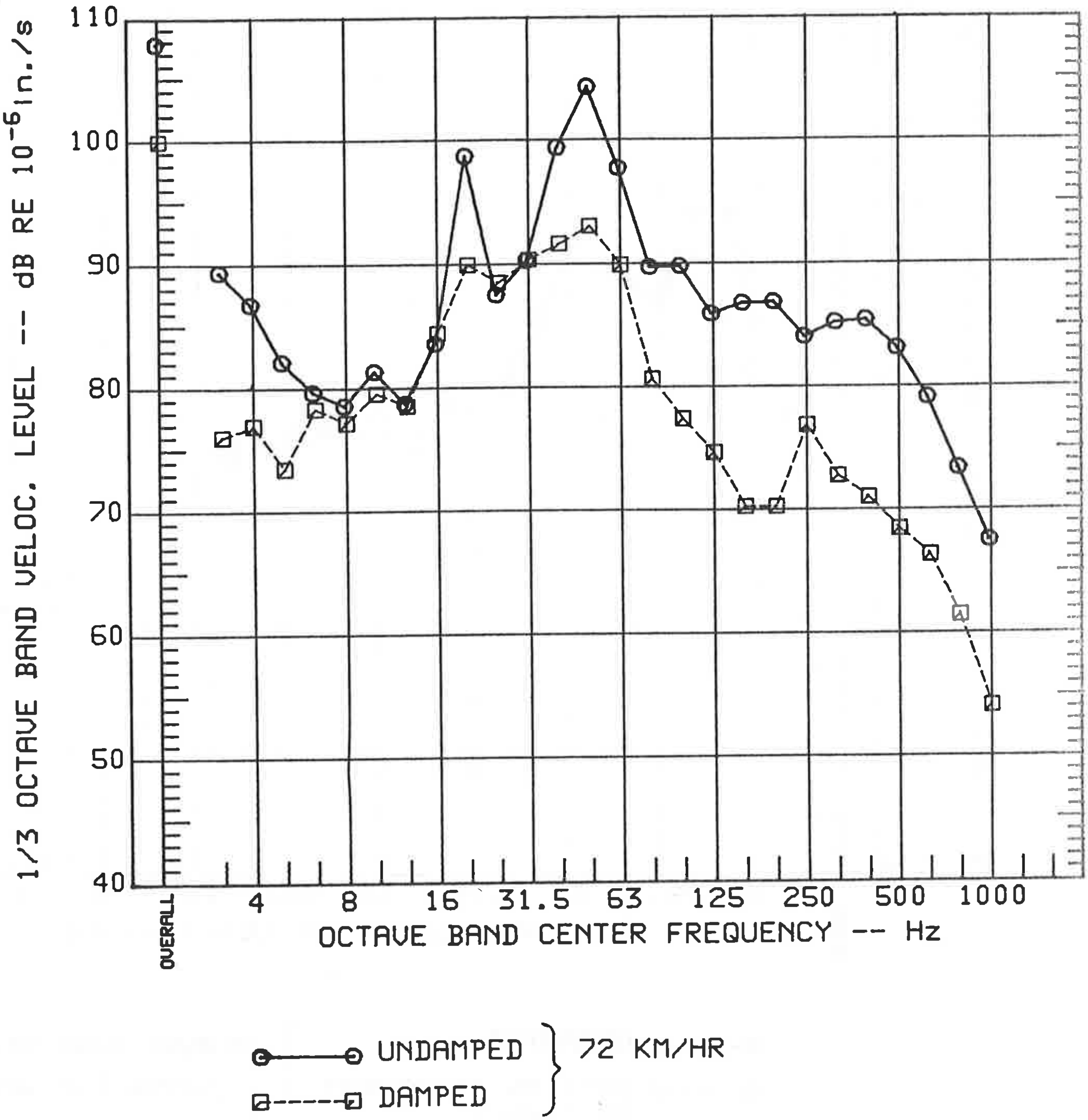


FIGURE 7.8a VIBRATION VELOCITY LEVEL OF SIDE PLATE OF MARTA AERIAL STRUCTURE WITH AND WITHOUT DAMPING TREATMENT

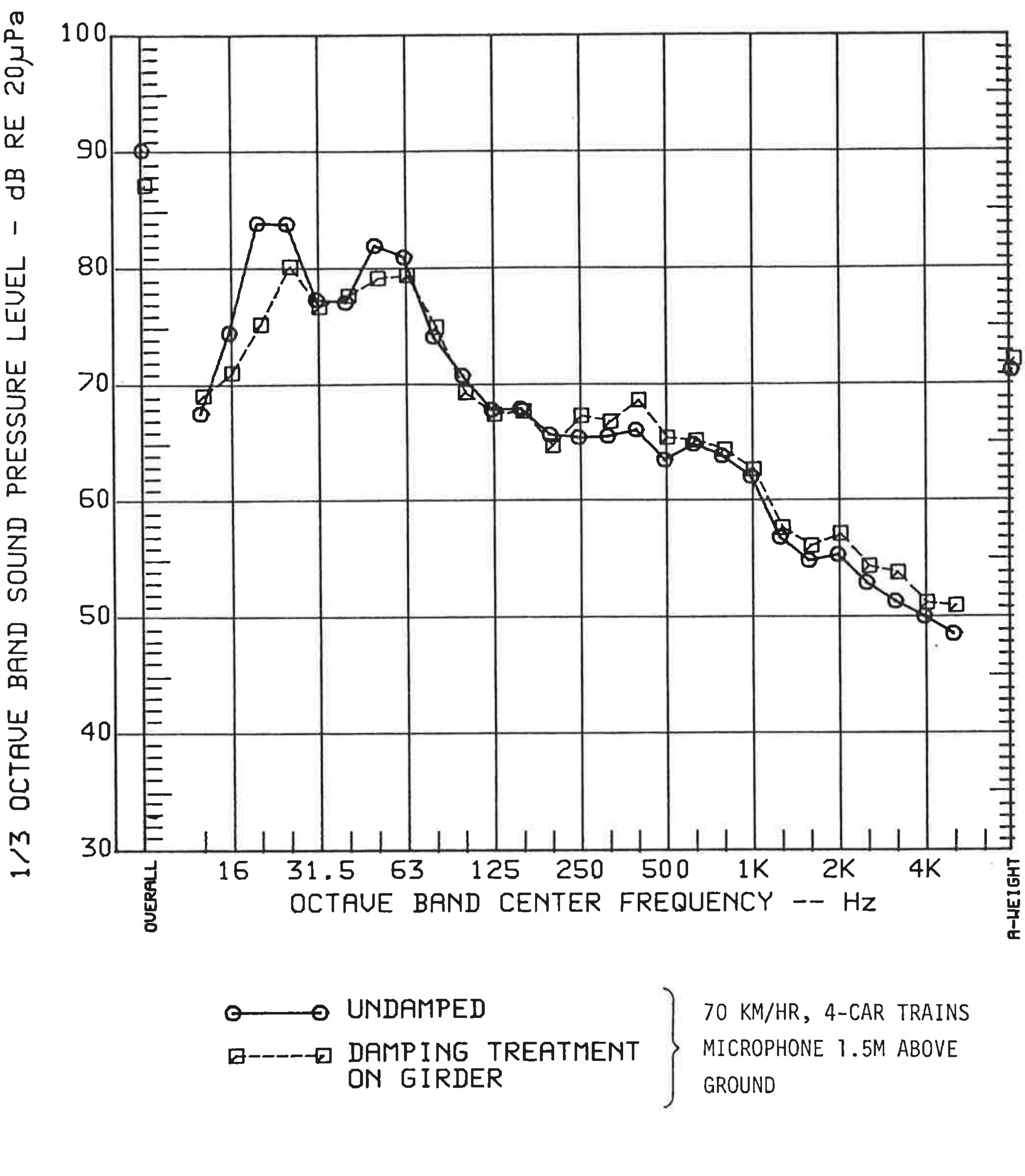


FIGURE 7.8b WAYSIDE NOISE LEVEL 15M (50 FT) FROM DAMPED AND UNDAMPED MARTA AERIAL STRUCTURES WITH SOUND BARRIER WALLS

of noise from the girder. The conclusion is that the damping treatment results in only a small reduction of the overall low-frequency sound level because the sound radiated from the wide concrete slab is nearly as high as the sound radiated from the steel girder.

The general conclusion is that damping treatment applied to the steel girder of a composite aerial structure acts to reduce low-frequency noise radiation to the point that it is nearly identical to that of concrete structures. However, it is also possible for the vibration of the concrete sections of the structure to radiate noise. This can be a significant problem for structures with unusually large decks such as the double-track aerial structure used at MARTA.

7.4 CONTROL OF NOISE FROM BALLASTED DECK STRUCTURES

Because ballasted deck elevated structures or bridges are the quietest form of aerial structure, the possibilities for additional treatments for further reduction of the noise are limited. The damping provided by the ballast precludes the use of further damping for reduction of structure-radiated noise. The mass of the ballast and its associated support elements is sufficient, and the addition of further mass to the structure will not significantly reduce structure-radiated noise. The use of ballast mats and sound barriers provide the possibility for reducing noise from ballasted deck structures.

Treatments appropriate for ballasted deck elevated structures are similar to those treatments discussed in relation to toher types of aerial elevated structures. The reader is referred to Section 7.2 for details on these items which can be applied to ballasted deck structures. The only exception is that addition of sound absorption to sound barrier walls will not have any beneficial effect on a ballasted deck structure; the sound absorption of the ballast provides all of the noise reduction that can be achieved with absorption treatment.

CHAPTER 7 REFERENCES

7.1 D. A. Towers, "Noise Impact Inventory of Elevated Structures in U.S. Urban Rail Rapid Transit Systems," Draft Report, Report No. 4239, (Bolt, Beranek & Newman, Inc., Cambridge, MA, Prepared for U.S. Department of Transportation, Transportation Systems Center, January 1980).

This report is part of a TSC-sponsored project to evaluate the impact of elevated structures and design noise control features. This report presents the results of the inventory and assessment of community impact of U.S. elevated structures. Included in the assessment are MARTA, BART, CTA, the Dade County system that is under construction, MBTA, NYCTA, PATCO, SEPTA, and WMATA. The basic conclusion is that the open deck, steel elevated structures account for 91% of the total nationwide noise impact.

7.2 S. K. Oleson, "Aerial Structure Noise Comparisons," Technical Report, SRI Project PH-4579, (Stanford Research Institute, Menlo Park, CA, October 1965).

An experimental study of the structure-radiated noise of the proposed BART aerial structures. Tests were performed on a concrete aerial structure and a steel/concrete aerial structure. The damping treatment (without constraining layer) on the steel girder was found to significantly reduce structure-radiated noise, however, the noise reduction is in the frequency range where noise radiated directly from the wheels, rails, and car dominates.

7.3 L. G. Kurzweil, "Prediction and Control of Noise from Railway Bridges and Trackbed Transit Elevated Structures," Journal of Sound and Vibration, 51(3), (April 1977), pp. 419-439.

A review of the current approaches to the prediction and control of noise radiated from elevated structures. Included is a summary of noise control case histories for a number of different types of elevated structures. The article also describes an analytical model that was developed to attempt to estimate the effects of structural parameters on vibration transmission and noise radiation.

7.4 P. J. Remmington, L. E. Wittig, R. L. Bronsdon, "Prediction of Noise Reduction in Urban Rail Elevated Structures," Draft Report, Report No. 4347, (Bolt, Beranek & Newman, Inc., Cambridge, MA, Prepared for the U.S. Department of Transportation, Transportation Systems Center, March 1980).

This report covers the development of an analytical model for the prediction of noise radiated by urban rail elevated structures. The work focuses almost exclusively on analysis of lightweight steel elevated structures and does not consider the concrete or composite steel/concrete aerial structures being installed on new transit lines. Of particular interest is the conclusion that the tie deck, girder, and rail are approximately equal strength noise radiaters. This indicates that the noise from all three sources must be controlled if significant noise reductions are to be achieved. Also of interest is the noise reduction achieved with the installation of a resilient fastener on a test section of NYCTA elevated structure. The overall noise level was reduced by 5 dBA.

7.5 U. J. Kurze, "Schalledampfte Stahlbrucken fur Schienenverkehr" (Sound-Damped Steel Bridges for Rail Traffic), Report prepared by a commission of the Study Committee for Applied Technology of Iron and Steel, (Dusseldorf, Germany, January 1973), (In German).

This report on the control of noise from steel elevated

structures presents background information on noise radiation by steel structures, a discussion of noise abatement procedures, some evaluation of the economics of various control procedures, and presents a summary of some tests that were performed.

7.6 Japanese National Railways, "Shinkansen Noise," Report and Supplemental Report (II), (Japan, August 1975).

This report summarizes the efforts of JNR to evaluate the noise and vibration problems of the Shinkansen and design control methods.

7.7 J. J. Hanel, T. Seeger, "Reduction in Noise Level on Steel Railway Bridges by the Application of Sandwich Type Sound-Absorbing Systems to the Girders," VDI-Berichte, No. 278, (1977).

A report on the design considerations for applying constrained layer-damping to steel railway bridges and on the results of a full-scale test on two box-girder bridges. The damping treatments were very effective; noise reductions of 13 and 17.5 dB were achieved. The conclusion is that fully treated steel box-girder bridges have almost the same acoustic characteristics as open track.

7.8a International Union of Railways, "Noise Abatement on Bridges: Noise Development in Steel Railway Bridges," Interim Report No. 1, D105/RP1/E, (Office of Research and Experiments, Utrecht, Holland, October 1966).

7.8b International Union of Railways, "Noise Abatement on Bridges," Final Report, D105/RP3/E, (Office of Research and Experiments, Utrecht, Holland, April 1971).

These reports present extensive measurement data on a large number of rail elevated structures. Much of the data is presented in summary form in Reference 7.3.

7.9 L. Cremer, M. Heckl, Structure Borne Sound, Translated and revised by E. E. Ungar, (Springer-Verlag, NY, 1973).

This is a very good textbook on structureborne sound which contains a collection of information that, to our knowledge, is not available in any other English text.

7.10 Hilti, "Intermediate Report on Endurance Tests for ST52 Rods with Countersunk Screws, Settling Bolts, and Weld-On Screw Bolts," (Technische Hochschule Darmstadt, Institute for Statics and Steel Construction).

A solution is discussed to the problem of corrosion at the point that the restraining layer is attached for constrained layer damping.

7.12 G. P. Wilson, "Reduction of Noise Radiated by the Walnut Creek Bridge Structure," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Letter report to BART, November 1973).

Report on an acoustical analysis of The Walnut Creek Bridge structure. Because of a high level of low-frequency rumble that contributes to community annoyance, a damping treatment to reduce the vibration amplitude of the steel plates of the bridge is recommended. The report includes the results of detailed noise measurements, including narrowband analysis, as trains traverse the bridge.

7.13 H. J. Saurenman, "Vibration and Noise Control Recommendations for Aerial Structures," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinckerhoff-Tudor-Bechtel and Metropolitan Atlanta Rapid Transit Authority, September 1975).

This report evaluates the expected noise radiation characteristics of the aerial structures proposed for and subsequently used at the MARTA system. The analysis concluded that the concrete aerial structure would not create any problems, but that low-frequency noise radiation from the girder of the composite steel/concrete structure could be a problem. A constrained layer-damping treatment was recommended for the plates of the steel girder to control the low-frequency noise.

7.14 J. R. Curreri, J. E. Koch, "Noise Control of Steel Aerial Structures: Computer Program for Optimizing Partial Coverage Constrained Layer Vibration Damping Treatments," (Soundcoat Company, Inc., NY, Prepared for Committee of Steel Plate Procedures of American Iron and Steel Institute, October 25, 1976).

This study reports on a computer program that was developed to optimize damping treatment of composite steel/concrete aerial structures. The analysis of the study focused on a 120 ft span MARTA design aerial structure.

7.15 G. P. Wilson, "Vibration and Noise Control at MARTA," Presented at the 99th Meeting of Acoustical Society of America, (Atlanta, GA, April 1980).

See Reference 6.7.

CHAPTER 8

8. GROUNDBORNE NOISE AND VIBRATION CONTROL

Groundborne noise and vibration are caused by vibration originating at the wheel/rail interface and propagating from the transit structure through the intervening soil and rock to nearby buildings. The resulting vibration may be perceptible as mechanical motion, and the acoustic radiation by the building components may cause an audible low-frequency rumble. Figure 8.1 illustrates the relationships and vibration propagation paths between the transit vehicle, the subway, and the structures affected along the right-of-way. The transit structure shown is a subway, but many of the considerations discussed apply to aerial and at-grade structures as well.

While the three basic transit configurations -- aerial, at-grade and subway have similar vibration sources, subways have received the greatest attention regarding groundborne noise and vibration even though the other configurations can produce similar levels. Airborne noise from trains on at-grade or aerial structures generally overpowers the groundborne noise and vibration, whereas the groundborne noise and vibration from subway structures are not similarly masked, which probably accounts for differences in attenuation. Communities near the WMATA and TTC systems have complained of groundborne noise and vibration from at-grade ballast-and-tie configurations. Near the CTA, similar complaints have occurred from elevated structure operations. Such complaints indicate that the problem of groundborne noise and vibration cannot be ignored for any transit structure configuration.

Groundborne noise and vibration from rail transit trains, transmitted into buildings, generally fall in the frequency range of 10 to 200 Hz and are usually concentrated in only one or two octaves. The typical octave band rms acceleration levels at the ground surface, at distances of 15 to 30 m from a subway, are 50 to 70 dB (re 1 micro g) with the peak frequency between 16 to 63

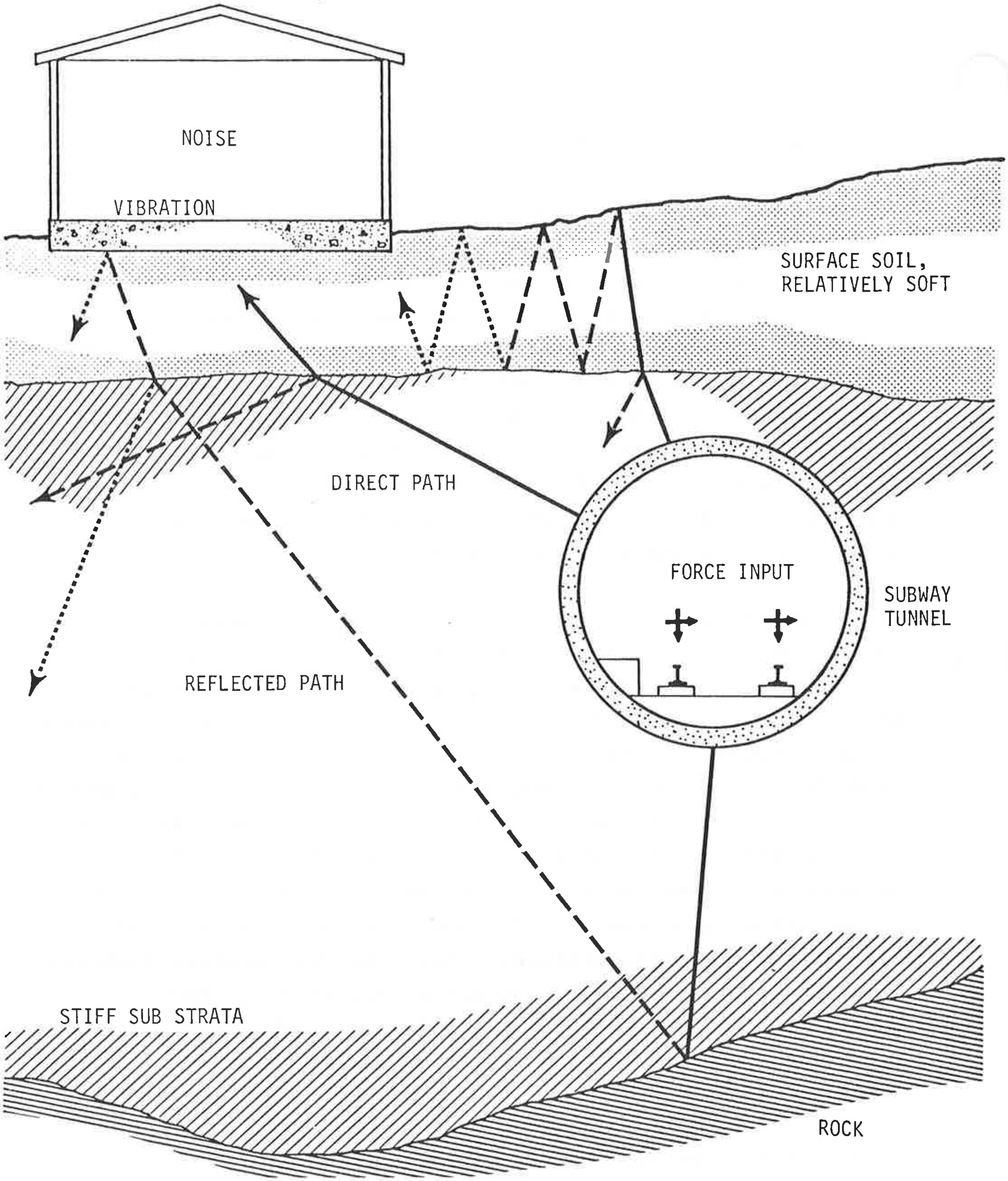


FIGURE 8.1 GROUNDBORNE VIBRATION TRANSMISSION PATHS

Hz. Figure 8.2 indicates some examples of ground surface vibration levels for three different transit systems -- TTC, WMATA, and MARTA. Groundborne noise levels in buildings near subways can be in the range of 30 to 45 dBA. Several spectra of groundborne noise, measured within residential structures, are presented in Figure 8.3 for the TTC system.

Groundborne noise and vibration inside buildings are often near the threshold of human sensitivity. In this range, a small increase in vibration or noise levels can cause quite significant increases in human response, so precise estimates of such vibration would be valuable during the design phase. Unfortunately, variability in soil and rock conditions and building designs make prediction more difficult than for airborne noise levels.

Both floating-slab and rigid invert trackbeds are represented in Figure 8.2. The differences between the TTC, WMATA, and MARTA data should not be interpreted as measures of the vibration isolation effectiveness of different designs. The data were collected with different truck designs, at different distances from the subway structure, and under different soil conditions. This information does, however, illustrate the fact that vibration control measures such as floating slabs may effectively reduce audible groundborne noise at frequencies above 20 to 30 Hz, but low-frequency groundborne vibration, in the range of 10 to 30 Hz, may still be relatively significant. Clearly, attention must be paid to reducing low-frequency vibration, where necessary, as well as noise. Subaudible, low-frequency vibration can make windows and dishes rattle, and residents may interpret such vibration as damaging to their homes and offices, although damage-risk levels are rarely, if ever, approached. Criteria for acceptable levels of both groundborne noise and vibration are discussed in Chapter 2.

The remainder of this chapter focuses on vibration control

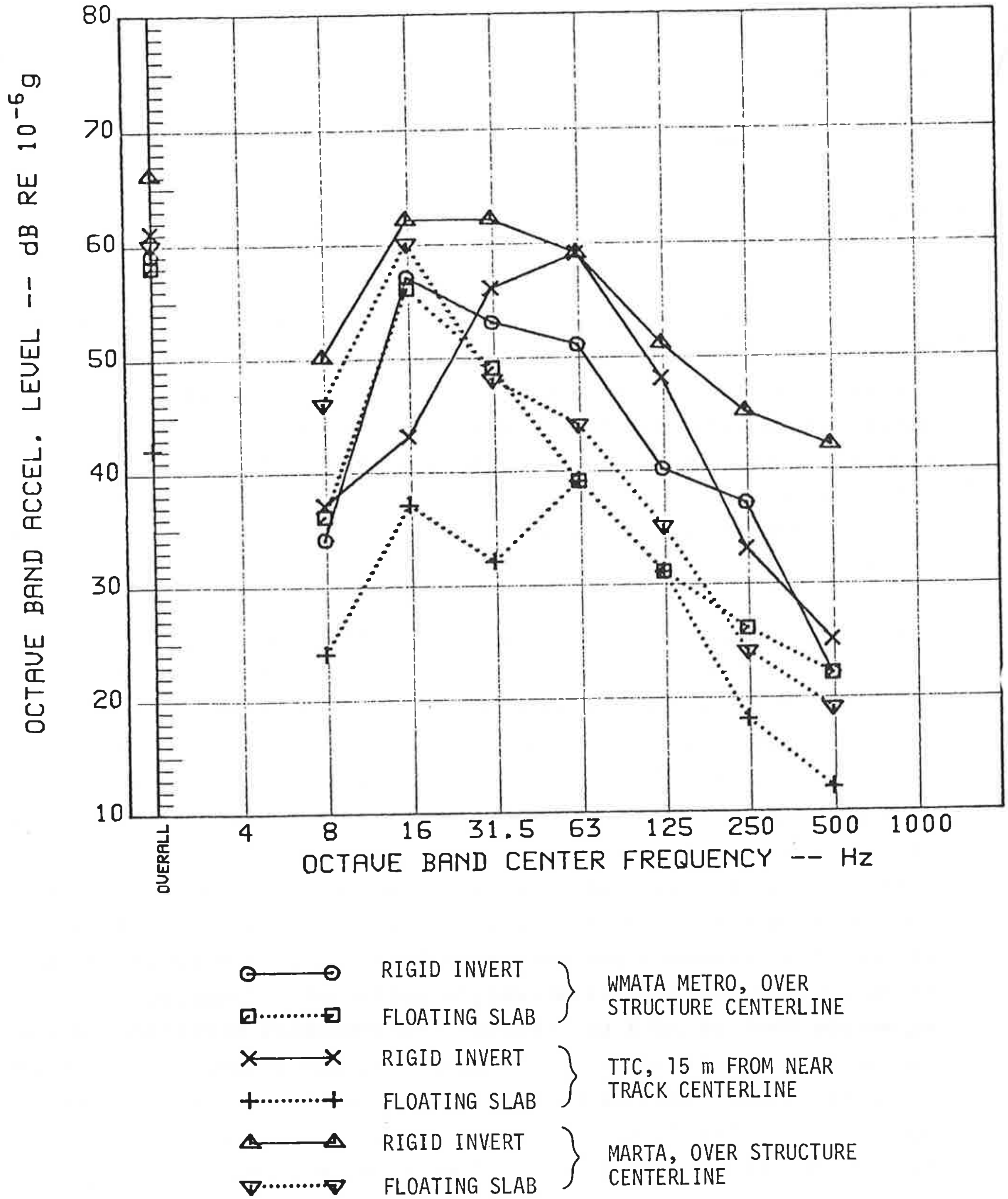


FIGURE 8.2 GROUND SURFACE VIBRATION, DOUBLE-BOX SUBWAYS FOUNDED IN SOIL

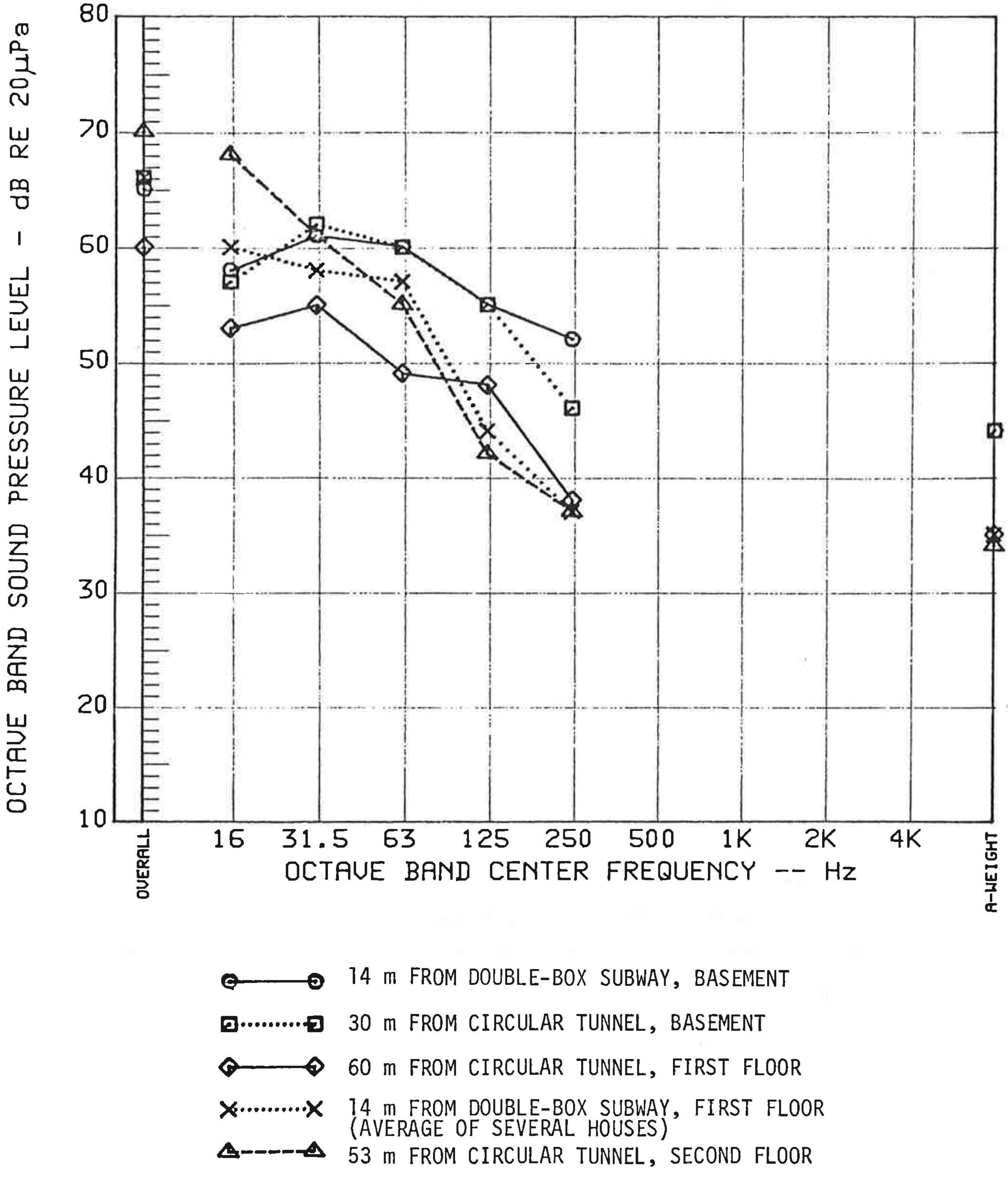


FIGURE 8.3 GROUNDBORNE NOISE AT VARIOUS DISTANCES FROM TTC SUBWAY

provisions, the design of track fixation and floating-slab trackbeds with respect to vibration control, and the prediction of groundborne noise and vibration levels within buildings.

8.1 PARAMETRIC DEPENDENCE OF GROUNDBORNE NOISE AND VIBRATION

The parametric dependence of groundborne noise and vibration is presented in terms of octave bands of 1/3 octave bands. This approach is more general than using "single-number" descriptors, such as overall acceleration level or velocity level. It allows the two-fold problem of low-frequency (10 to 30 Hz) vibration and higher frequency (30 to 500 Hz) groundborne noise to be addressed simultaneously. Single-number descriptors for vibration occur throughout the literature and are useful in many situations. However, caution must be exercised because, unless otherwise indicated, such single-number descriptors often refer to A-weighted groundborne noise levels and do not adequately describe low-frequency noise.

The influence of various transit system components and conditions are summarized in Table 8-1.

8.1.1 Wheel/Rail Roughness

Groundborne noise and vibration are generally assumed to result from surface irregularities on the wheels and rails (e.g., roughness of wheels and rails, rail joints, etc.). The information presently available on this topic is very limited at wavelengths relevant to groundborne vibration. It is clear, however, that even if all surfaces were perfectly smooth, some vibration would be generated by variations in the hardness of the surfaces in contact.

TABLE 8-1 FACTORS INFLUENCING GROUNDBORNE NOISE AND VIBRATION

Factor	Influence
Truck primary suspension:	Very significant causing up to 10 or 15 dB differences in groundborne vibration levels.
Wheel/rail roughness and wheel flats:	Increases vibration by 6 to 15 dB.
Resilient wheels:	Decreases vibration above 40 to 50 Hz by 5 to 10 dB.
Rail fasteners:	Vibration level proportional to 20 log K above 50 Hz, K = rail support modulus.
Floating slab (14 Hz resonance):	Decreases vibration above 20 to 30 Hz by up to 20 or 30 dB.
Structure type:	Circular tunnels produce higher vibration levels than double-box subways at frequencies above 50 Hz.
Train speed:	Vibration increases by 4 to 6 dB per doubling of train speed.
Car body suspension:	No apparent effect.

Wheel/rail roughness can be controlled by grinding rails and truing wheels. Inadequate maintenance of wheels and rails may cause significantly higher noise levels (Ref. 8.1). Roughness (without large corrugations) does not increase vibration levels significantly on jointed track because the impacts at the rail joints or from wheel flats will dominate the groundborne vibration levels.

Table 8-2 summarizes the increase of A-weighted groundborne noise levels relative to smooth, continuous welded rail with balanced and trued wheels, as functions of rail and wheel condition and track configuration.

This table indicates that flatted wheels, corrugated rail, and loose rail joints will greatly increase groundborne noise, compared to smooth, welded rail and trued wheels. Special trackwork also produces high groundborne noise levels, without significantly altering the spectrum. Noise levels are particularly high at double crossovers with center diamonds and switch frogs. Special, jointless switch frogs are available, which reduce the magnitude of the impacts and, hence, the levels of groundborne noise and vibration.

8.1.2 Transit Vehicles

The transit vehicle consists of two major components, the vehicle body and the trucks. The vehicle body is generally supported on the trucks by a secondary suspension with a very low resonance frequency, on the order of 1 to 2 Hz. Groundborne vibration at frequencies below about 10 Hz rarely causes any difficulty, thus the vehicle body support system is not a significant parameter for groundborne noise and vibration.

The primary suspension stiffness of the vehicle truck is the most important property of the transit vehicle with respect to groundborne vibration. The primary suspension supports the truck frame on the axles, and reduction of the primary stiffness generally

TABLE 8-2 INCREASE IN A-WEIGHTED GROUNDBORNE NOISE DUE TO VARIOUS WHEEL/RAIL CONDITIONS RELATIVE TO SMOOTH WELDED RAIL AND TRUED WHEELS

	Condition	Increase
I	Wheel flats, loose rail joints, rail corrugation	10 - 20 dBA
II	Rough wheels, and rails (continuous welded rail without corrugation and no wheel flats)	3 - 6 dBA
III	Jointed track, wheel flats	5 - 10 dBA
IV	Special trackwork	10 - 15 dBA

TABLE 8-3 RAIL SUPPORT SYSTEMS

	Configuration	Typical Rail Support Modulus (lb/in. per in.)
I	Rigid fixation (non-resilient fixation to concrete invert)	>20,000
II	Ballast-and-tie	2,000 to 4,000
III	Resilient direct fixation (elastomer pad in compression)	2,500 to 10,000
IV	High resilience direct fixation (elastomer pad in shear, e.g., Cologne "Egg")	1,000 to 1,500
V	Resilient supported ties (RS STEDEF or Semperit-Voest)	≈2,500

leads to a reduction of the dynamic load of the transit vehicle at the rail (i.e., a reduction of wheel/rail forces at frequencies below about 100 Hz). An additional advantage is a reduction of the shock loading on the truck frame and components. At CTA, the use of softer primary suspensions on the 2400 series cars resulted in substantial reduction of groundborne vibration compared to the vibration produced by the 2000 and 2200 series CTA cars (Ref. 8.2). The 2400 series truck design was proposed by the German truck manufacturer Wegmann and incorporated into the 2400 series cars by Boeing Vertol.

The design features for the CTA 2400 series trucks include:

- Flexible rubber springs to support the frame at the journals.
- A combination steel-rubber bolster spring to support the body.
- No metal-to-metal contact between moving elements.
- Rubber bushings or pads at all contact points.
- A special rubber-and-steel "sandwich" center bearing to absorb torsion motions when the truck pivots under the kingpin.
- Flexible frame.

Figure 8.4 summarizes the average reduction of 1/3 octave band vibration levels achieved over a wide range of train speeds, using the low stiffness primary suspension of the 2400 series CTA car. This is compared to the standard stiff primary suspension used in the earlier 2000 and 2200 series CTA cars (Ref. 8.2). The tests were performed on the same day with identical instrumentation and measurement locations.

The vibration reduction achieved with the CTA 2400 series truck was substantial on ballast-and-tie track with welded continuous rail, but less reduction was observed on the jointed-rail elevated structure. The lower reduction on the elevated structure is probably due to a lower impedance of the elevated structure roadbed and the presence of rail joints. Nevertheless, the

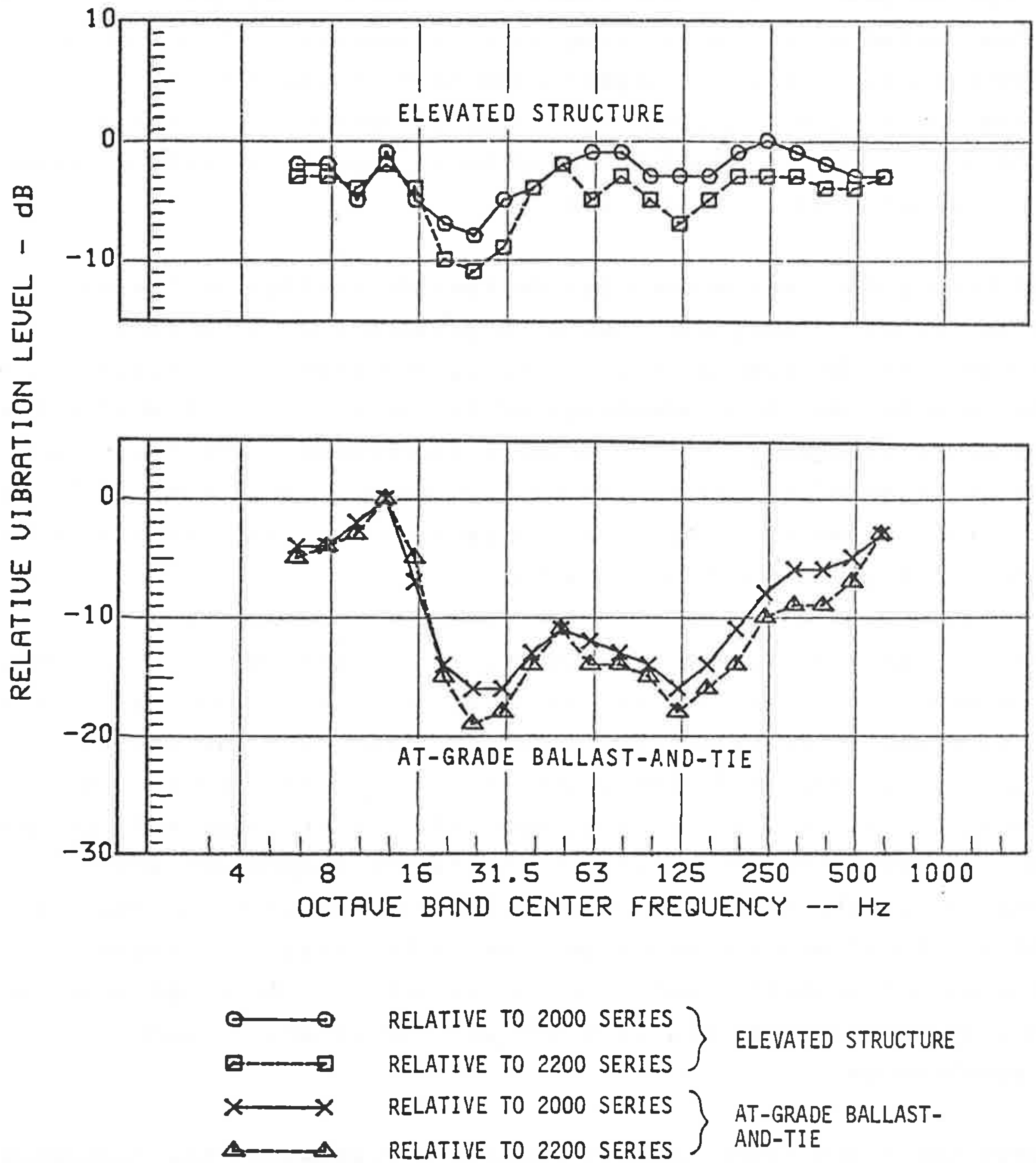


FIGURE 8.4 VIBRATION LEVEL WITH CTA 2400 SERIES CARS (WEGMANN TRUCKS) RELATIVE TO CARS WITH STIFF PRIMARY SUSPENSION

vibration reduction on the elevated structure was measurable. In subway structures, the driving point impedance of the subway invert is similar to or higher than that of ballast-and-tie surface track, so that the vibration reduction performance of the 2400 series car in subways should be similar to or greater than that illustrated in Figure 8.4.

Additional data are needed for further definition of the effects of the truck primary suspension on groundborne vibration. However, at the time of this writing, the limited available data indicate the definite advantage of low stiffness or low effective resonance frequency for the primary suspension. The preferred design is an effective resonance frequency of less than 12 Hz for the truck frame on the primary suspension, and some data indicate 7 to 10 Hz gives the best results.

Besides the lower primary suspension stiffness, other resilient elements in the CTA 2400 series cars may reduce wheel rail forces by increasing the structural damping of the truck assembly. Vibration energy is directly fed into the truck, and in the absence of structural damping, some of it must eventually return to the track. Adding resilient, inelastic components and bushings to the truck may allow vibration energy to be absorbed, which effectively removes a portion of the available energy otherwise transmitted into the way structure. Although this is only a secondary effect, it could be significant in some circumstances.

Above the truck frame support resonance frequency, the impedance of the truck as seen by the rail is determined primarily by the wheel mass. Thus, further reduction of wheel rail forces by modification of the truck must involve the wheel sets. The use of resilient wheels, such as Penn Bochum, SAB, or Acousta Flex, will partially decouple the wheel, axle, and gearbox masses from the rail and consequently reduce wheel/rail forces.

Figure 8.5 illustrates vibration reductions achieved through the use of resilient wheels and pneumatic rubber tires. Resilient wheels on trucks with tread brakes failed on the SEPTA system (Ref. 8.3), but they have been successfully used on many vehicles with disc brakes (Ref. 8.4). Resilient wheels will reduce shock loading of truck-mounted and axle-mounted equipment, with possible improvement in reliability and service life. The limited data available indicate that the stiff or low static deflection resilient wheels, such as the Penn Bochum or Acousta Flex wheel, give a small reduction of ground vibration above a frequency of 40 or 50 Hz (probably the effective resonance of the axle/hub mass on the resilient pad at the tire). However, the data also show that very soft or high static deflection wheels, such as the SAB or the PCC Super Resilient wheel, give substantial reduction of groundborne vibration in the low-frequency range - giving about 10 dB reduction in the 31.5 and 63 Hz octave range (see Fig. 8.5).

Currently, available technology ahd hardware can be incorporated directly into the design of the transit vehicle truck. This will significantly reduce groundborne noise and vibration by reducing the dynamic load of the vehicle truck at the rail. In fact, some of the wide variation between groundborne noise and vibration levels observed at various systems is attributable to differences in vehicle truck design. These considerations are particularly important when selecting and developing specifications for new transit vehicles.

8.1.3 Rail Support

The design of the rail support or track fixation system is important in controlling groundborne vibration. Decreasihg the fastener stiffness and increasing the rail mass usually reduces trackbed forces. Use of heavier rails can, therefore, result in reduced groundborne vibration, resulting in reduced airborne noise from the rail due to a lower vibration amplitude.

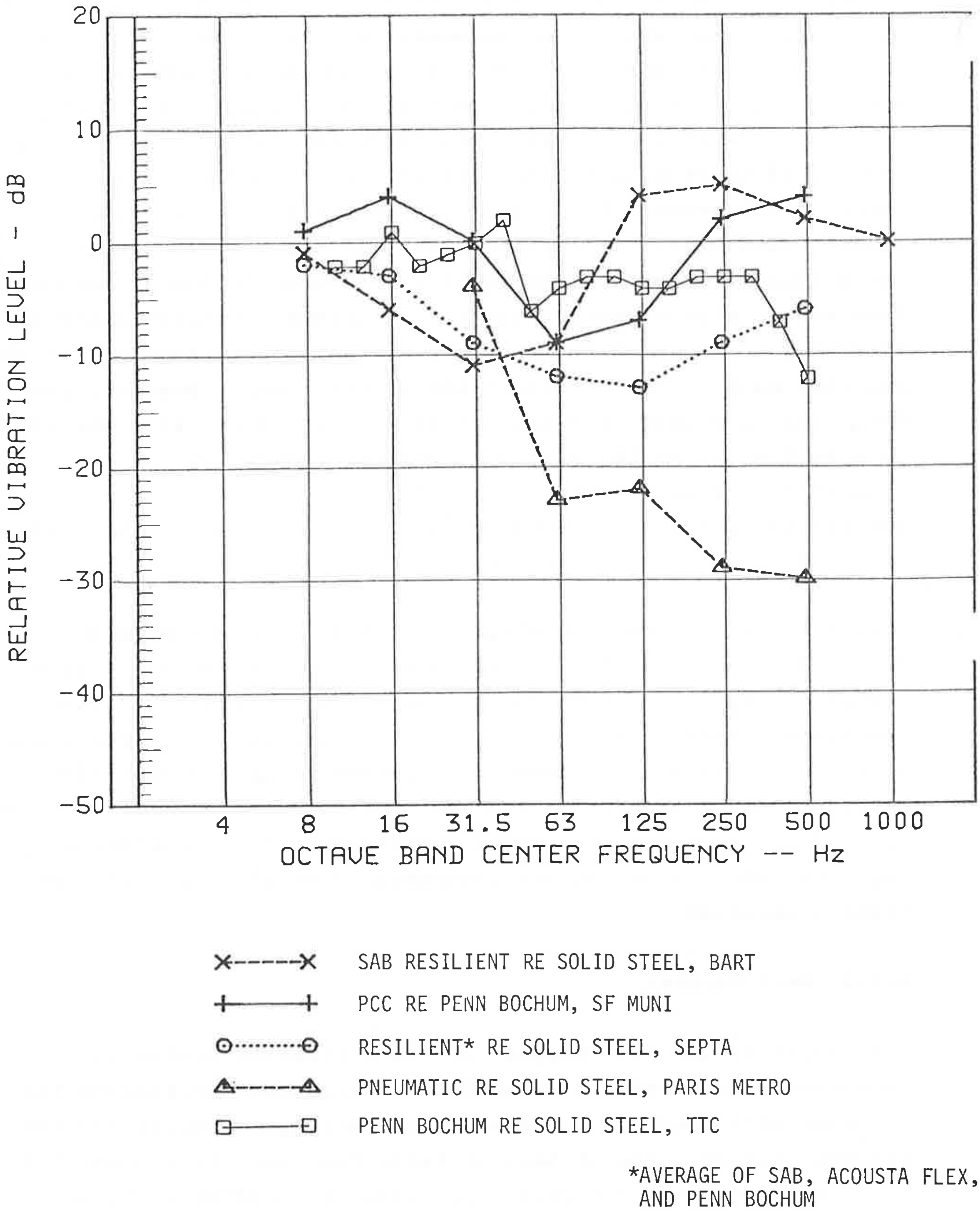


FIGURE 8.5 RELATIVE VIBRATION LEVELS WITH PNEUMATIC TIRES AND VARIOUS RESILIENT WHEELS

Rail fastener designs may be grouped under a number of categories, based on the general design. Each category has a characteristic range of rail support stiffnesses. Table 8-3 lists various major rail support categories. The floating-slab vibration isolation system is discussed separately in Section 8.3, because of its inherently unique performance, size, and weight and because it is a secondary support system independent of the rail fastening system.

The term "rigid fixation" applies to rail fastening systems giving direct rigid attachment of the rail to a continuous concrete invert or to an elevated structure. It usually consists of wooden blocks or ties between the rail and structure for fastening and load or stress distribution. Direct fixation refers to fasteners bolted directly to a continuous concrete trackbed without intervening ballast or wood blocks. Direct fixation with concrete trackbed is generally of longer life and is cheaper to maintain than ballast-and-tie, but the initial cost is higher. All of the categories in Table 8-3, except the ballast-and-tie, represent variations of the direct fixation concept.

Generally, the lower stiffness, conventional, resilient direct fixation fasteners are a form of vibration isolation which will adequately control the groundborne vibration for many situations. The "Cologne Egg" is a recently developed resilient direct fixation fastener of nonconventional design, incorporating elastomer in shear. With this fastener, low vertical stiffness has been obtained without sacrificing lateral stability. Developed by the Cologne Transport Authority, it is reputed to have excellent vibration isolating characteristics (Ref. 8.8).

The RS-STEDEF ballastless track system is a two-block concrete tie system with the ties supported directly on the invert by elastomer boots. This system reduces groundborne noise and vibration levels through the isolating effect of the concrete tie mass supported on soft pads. Thus, it provides some inertial mass in addition to reducing the effective track support stiffness. A thin resilient pad is also placed between the rail and the concrete ties; however, this pad is too stiff to provide any vibration isolation benefits and provides only for proper anchoring of the rail to the tie without stress concentration. At frequencies above the resonance of the concrete tie/resilient pad system, the vibration reduction is quite large -- Colombaud reports 9 dB in the 63 Hz octave band, relative to tie-on-ballast fixation (Ref. 8.9). Resilient tie systems may reduce construction costs by circumventing the need for floating-slab track in vibration trouble spots. Wilson stresses that the resilient tie systems originally designed for railroads could be adapted to rapid transit operation with improved vibration reduction performance by enlarging the tie spacing and using softer pads (Ref. 8.10).

Generally, the lower the support modulus of a rail fixation system, the lower the levels of groundborne vibration. The rail support modulus, K, is equal to the fastener stiffness divided by the fastener spacing. Groundborne vibration levels are approximately proportional to 20 log K in the range of 50 to 4000 Hz and 5 log K in the range of 20 to 30 Hz (Refs. 8.5, 8.6) when the rail support modulus, K, is less than 20,000 lb/in. per lineal inch of rail.

A very complete, theoretical analysis of the effect of rail fastener stiffness on vibration levels has been performed by Bender, et al. (Ref. 8.7). However, these authors recommend a fastener stiffness which results in a rail support modulus of about 800 to 1000 $lb/in.^2$, inadequate for rail stability with most fastener designs. Most practical design rail fastening systems give a rail support modulus in the range of 2500 to 20,000 lb/in^2. The practical lower limit for conventional flat pad direct fixation fasteners is a vertical stiffness of 90,000 to 100,000 lb/in., giving a rail support modulus of 2500 to 3300 $lb/in.^2$, depending on fastener spacing.

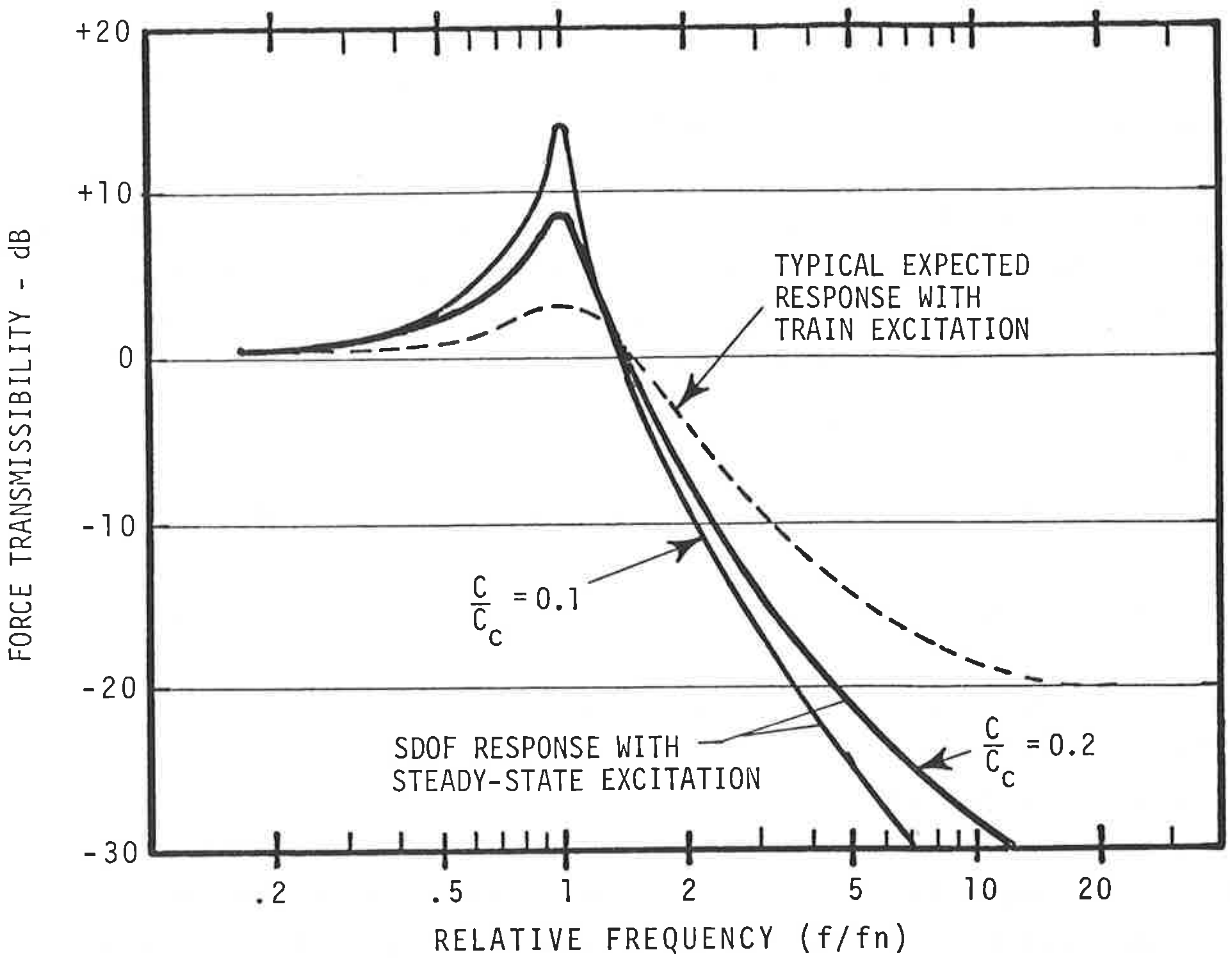
A much, more comprehensive discussion and description of rail fastener performance and design is included in Section 8.4.

8.1.4 Floating-Slab Vibration Isolation

Floating slabs, though relatively expensive, are extremely effective in controlling groundborne noise and vibration. Because of their importance, their design and performance are discussed in detail in Section 8.3.

A number of different floating-slab configurations will provide substantial vibration isolation. Two designs being used on many of the new transit lines are: the lightweight continuous floating slab and the discontinuous floating slab. The continuous floating slab consists of a resiliently supported slab, 0.2 to 0.3 m thick, cast-in-place in a recess in the subway invert. The discontinuous floating slab consists of precast blocks, 0.2 to 0.3 m thick and 1.5 m long, supported on elastomer pads with longitudinal and lateral stability, which is provided by preloaded elastomer blocks located at the sides and between blocks. The discontinuous floating slab is often referred to as a double-tie floating slab since each precast block supports two sets of rail fasteners.

Most floating slabs that are in use on North American transit systems have been designed for resonance frequencies of 14 to 16 Hz, with a train on the slab. A detailed, theoretical analysis of floating-slab performance, including the effect of rail fastener stiffness and damping (Ref. 8.16), concludes that with reasonable damping, a single-degree-of-freedom model is sufficient for floating-slab design. The vibration reduction for an ideal single-degree-of-freedom isolation system and the reductions that would be predicted for a floating slab are presented in Figure 8.6. The broken line indicates that above 125 Hz, 20 dB is the maximum reduction expected for design purposes. Wave motion in slab supports, slab bending modes, and other secondary effects



C = DAMPING COEFFICIENT

C_c = CRITICAL DAMPING COEFFICIENT

f = FREQUENCY

f_n = UNDAMPED NATURAL FREQUENCY

FIGURE 8.6 TRANSMISSIBILITY OF FLOATING-SLAB VIBRATION ISOLATION SYSTEMS prevent greater vibration reduction at higher frequencies although in some cases, greater reductions (25 to 30 dB) have been measured for installations now in service.

For design of floating slabs, the resonance frequency indicated in Figure 8.6 is 12 to 16 Hz. This figure is determined by the sum of the masses of the slab, the rail, the unsprung mass of the vehicle truck, and by the sum of the support stiffnesses of the slab supports, the perimeter isolation pads, and any entrapped air. There is no entrapped air to consider with discontinuous floating slabs. Tests conducted at TTC, WMATA, and MARTA confirm the theoretical prediction shown by Figure 8.6 (Refs. 8.17-21); however, significant deviations from the ideal isolation curve sometimes occur (Ref. 8.22), apparently because of interaction between the trucks and the floating slab or, possibly, between the slab and tunnel coupling to the soil.

8.1.5 Ballast Mats

Ballast mats are usually thick, resilient layers of elastomer, cork, fiberglass, or rock wool, placed under ballast. Although they are normally used to improve electrical isolation, water drainage, or to reduce pulverization of the ballast, ballast mats also reduce rail forces from the roadbed. Although widely tested and installed in Europe and Japan (Ref. 8.23-27), they have received only limited attention in North America. Some recent tests were performed at TTC and a ballast mat has been installed by MBTA on a railroad bridge (low-speed freight only) over a transit station lobby. Unfortunately, there are no measurement data available for the MBTA installation; however, it is considered a success by MBTA. Although several researchers report satisfactory results, there is a considerable amount of disagreement about the effectiveness of ballast mats (Ref. 8.8). Tajima (Ref. 8.23) indicates that ballast mats made from used automobile tires reduce ballast pulverization, in addition to aerial structure noise. Installation of the ballast mat was timed to coincide with normal ballast removal and maintenance operations. The nominal increase in cost was easily outweighed by the reduced maintenance frequency achieved.

Oelkers and Kock (Ref. 8.26) present measurement data for the reduction of subway structure vibration obtained with the use of Triplex "Isolif" ballast mats. The 1/3 octave band vibration reductions achieved by introducing a ballast mat are shown in Figure 8.7 and range from about 4 to 12 dB, at 31.5 Hz, to about 10 to 25 dB at 315 Hz. No data are given for lower frequencies.

These reductions suggest that a ballast mat could be useful in a ballasted subway or aerial structure. The relatively low installation cost and time reported by Tajima and Kiura indicate that a ballast mat could be used for general retrofit. Before ballast mats are installed to control groundborne noise and vibration, more research is needed to ensure that the installation will provide significant benefit at the frequency range of interest. Although some of the research looks promising, questions remain as to actual effectiveness and life expectancy.

Ballast mats have not been incorporated in any new subway or aerial structure constructions, primarily because direct fixation fasteners and low maintenance concrete roadbed are usually preferred. In the case of ballast-and-tie track on earthen embankments, the foundation stiffness of the subgrade, on which the mat would lie, may be insufficient for effective vibration reduction. A thicker, more resilient mat than that used with concrete bases might prove effective.

8.1.6 Type of Transit Structure

Groundborne vibration levels and spectra are strongly influenced by the type of transit structure -- subway, at-grade ballast-and-tie, or aerial structure. Since the relationship between the soil and the structure is different for each structure type, the vibration coupling for each structure type is fundamentally different.

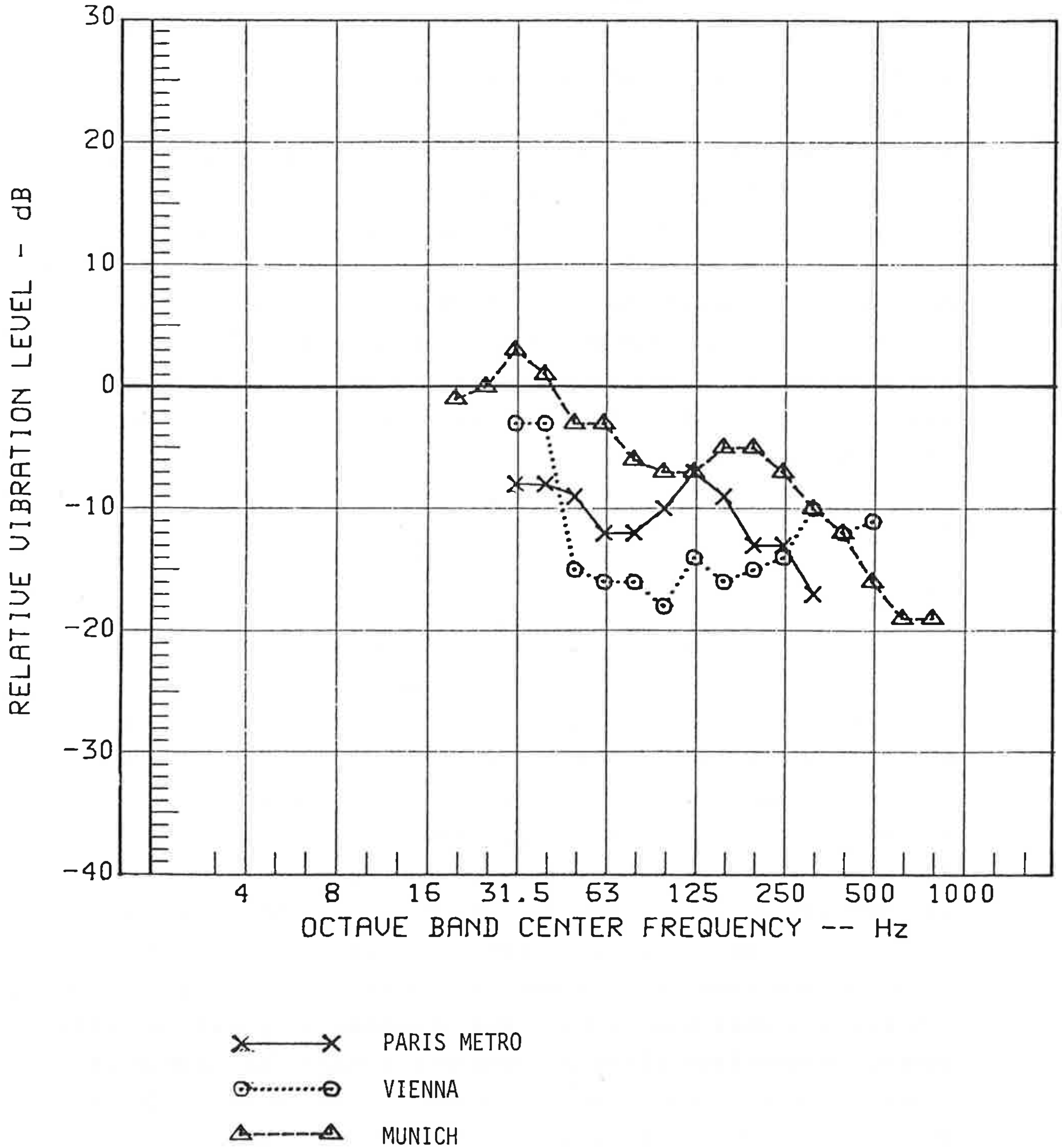
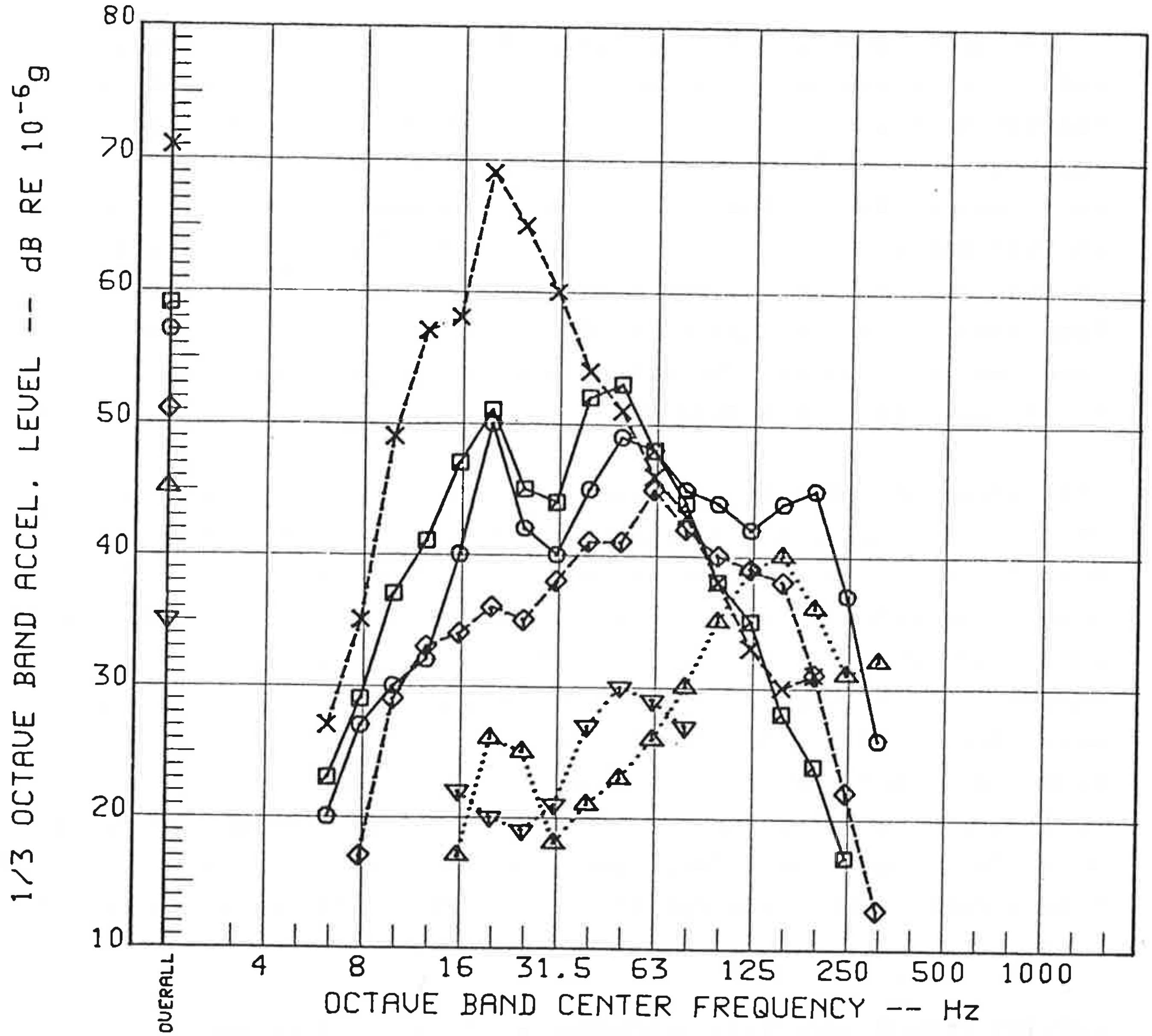


FIGURE 8.7 MEASURED VIBRATION REDUCTION WITH "ISOLIF" BALLAST MATS

In each basic structure, vibration levels and spectra are affected by dimensional and mass parameters. In subways, the wall flexural stiffness, mass per unit wall area and subway length, and structural bending stiffnesses may all be important. In an aerial structure, the bending stiffness and mass per unit length of each span, the footing bearing surface area and depth, and footing and support column mass are significant. Important characteristics of ballast-and-tie track, at-grade and on earthen embankments, include ballast thickness, the material properties of the embankment, and perhaps the height and cross section of the embankment. Some recent data indicate reduced wayside vibration levels for elevated berms compared to low berm or at-grade tracks.

8.1.6.1 Differences Between Subway Types -- All types of transit structures are subject to complaints due to groundborne noise and vibration. Although problems are more common near subway structures, many valid complaints come from people living near at-grade ballast-and-tie track, as well as near aerial structures. Most complaints associated with aerial structures are for lightweight steel elevated structures. Groundborne vibration and noise problems are very rare near concrete deck aerial structures. Figure 8.8 presents a comparison of groundborne vibration levels measured at the ground surface for various way structure configurations at the WMATA system (Ref. 8.21). Because these data were all obtained on the same system, direct comparisons can be made among levels with soil-founded, concrete, double-box subway, soil-founded circular concrete tunnel, mixed-face circular concrete tunnel, horseshoe or circular rock tunnels, and at-grade ballast-and-tie track structures. No information is available on the ground vibration levels near WMATA aerial structures.

The data presented in Figure 8.8 span a range of almost 50 dB at frequencies in the neighborhood of 20 Hz to about 10 dB at 100 Hz. The at-grade ballast-and-tie track produced the highest vibratic



□———□	DOUBLE-BOX, SOIL	70 km/hr	15 m
○———○	CIRCULAR CONCRETE TUNNEL, SOIL	70 km/hr	15 m
◇- - - -◇	CIRCULAR CONCRETE TUNNEL, MIXED-FACE	88 km/hr	15 m
△·········△	HORSESHOE ROCK TUNNEL	70 km/hr	30 m
▽·········▽	CIRCULAR ROCK TUNNEL	120 km/hr	30 m
×- - - -×	AT-GRADE BALLAST-AND-TIE	95 TO 110 km/hr	20 m

FIGURE 8.8 AVERAGE GROUND SURFACE VIBRATION LEVELS FOR DIFFERENT TYPES OF WMATA STRUCTURES

acceleration level, with a major peak at about 22 Hz. While the rock tunnels produced the lowest vibration levels, they still exhibit at least a minor peak at about 20 Hz. However, the maximum peak in the 1/3 octave band acceleration spectra for the rock tunnels occurred at about 125 Hz, compared with 20 Hz for the ballast-and-tie track. The soil-founded subways produced the highest levels of vibration from subway structures, exhibiting high levels over the range of 20 to 200 Hz. In the frequency range of 10 to 63 Hz, the mixed-face tunnel produced about 5 to 10 dB lower vibration than did the soil-founded subways.

The differences between groundborne noise and vibration from at-grade ballast-and-tie track or subway structure, and groundborne vibration from aerial structures, are difficult to assess empirically, since measurement data are limited. However, some measurements were performed at the BART test track for both concrete aerial structure and ballast-and-tie tracks on a low berm (Ref. 8.12). The results are presented in Figure 8.9 in terms of octave band acceleration levels. Results for ballast-and-tie track on the TTC system and the WMATA systems are shown for comparison. These data refer to smooth, continuous ground rail, with train speeds of 70 to 80 km hr and distances of 9 to 15 m.

Judging from Figure 8.9, groundborne vibration from aerial structures in less than or similar to at-grade ballast-and-tie track, depending on the frequencies considered and upon the soil properties of the locality. The difference between the WMATA and TTC groundborne vibration data on at-grade ballast-and-tie track is apparently due to differences in properties of the soil. These are similar to the differences observed for groundborne vibration from subways at these two systems.

8.1.6.2 <u>Effects of Subway Structure Type</u> -- A number of measurements of groundborne vibration have been made for various subway designs. Much of this data is complicated by differences in rolling stock and local or regional variations in soil

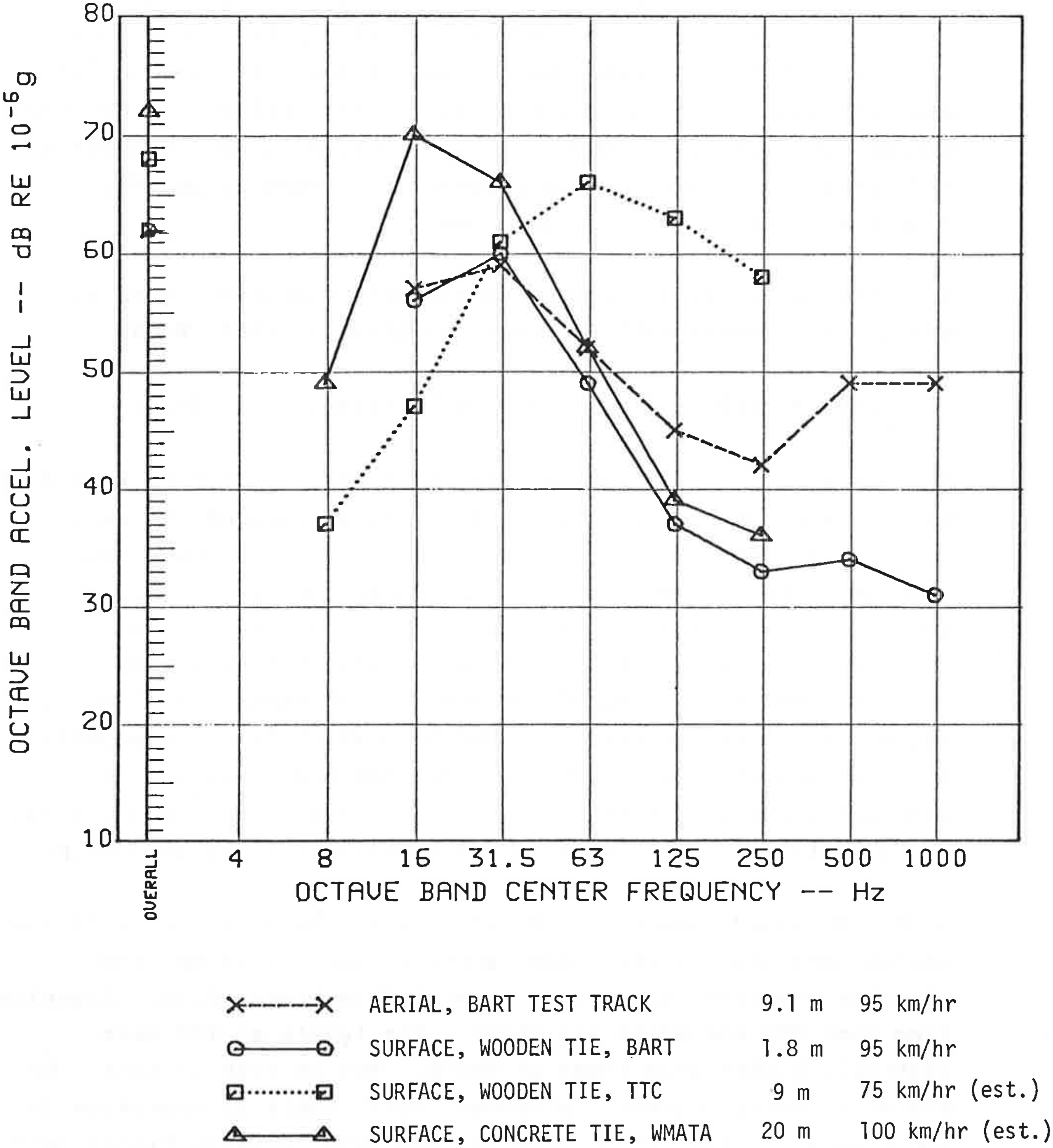


FIGURE 8.9 GROUND SURFACE VIBRATION, CONCRETE AERIAL STRUCTURE AND BALLAST-AND-TIE SURFACE TRACK

parameters, but the data do provide a basis for estimates of the vibration level changes caused by differences in subway mass and geometry. All such measurements suggest that groundborne noise and vibration levels decrease significantly with increased overall subway mass per unit length. This reduction is not adequately understood, although various theories have been proposed to describe subway soil/structure interaction.

An empirical relation between subway wall vibration velocity level and average subway wall thickness is given as (Ref. 8.29):

$$L_v (\text{dB re } 5\text{x}10^{-8}\ \text{m/sec}) = 69 - 56 \log(d/40) \qquad (8.1)$$

for values of average subway wall thickness, d, of 0.40 to 1.25 m. The average subway wall thickness is the average of the wall, ceiling, and invert thicknesses. This relation is based on results of measurements made by the Curt-Risch Institute, published in a series of internal reports, and evidently summarizes the institute's investigations. The reduction predicted for a doubling of average wall thickness is 12 dB to 18 dB, significantly greater than the 6 dB which is often expected for doubling of mass. Although Reference 8.29 indicates a remarkably low deviation of, $\pm$ 2.5 dB, there are no measurements at North American transit systems which verify this relationship.

A more detailed comparison of 1/3 octave groundborne acceleration levels, measured at the ground surface for double-box and circular concrete tunnels, is presented in Figure 8.10. Examples from both TTC and WMATA are shown. The levels at TTC were generally higher than those at WMATA. But in both systems, the circular concrete tunnels produced lower levels of vibration in the frequency range of 6 to 30 Hz and substantially higher levels of vibration above 125 Hz than the concrete double-box structures.

The circular concrete TTC tunnel is constructed of bolted precast concrete elements, with a nominal average thickness of 0.13 m (5 in.). The WMATA tunnel walls are cast-in-place concrete, 0.2 to

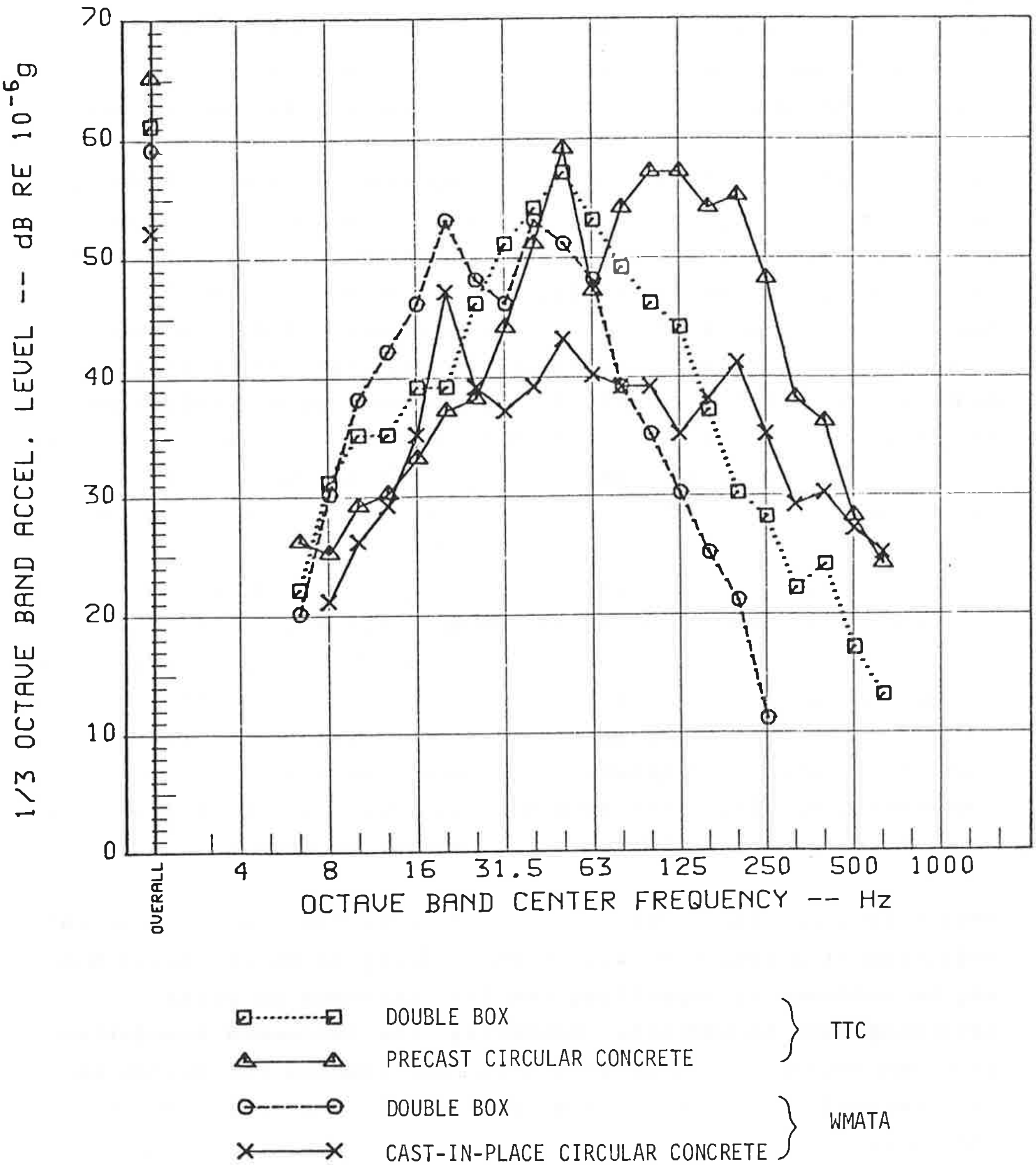


FIGURE 8.10 GROUND SURFACE VIBRATION FROM SOIL-BASED SUBWAYS, 15 m FROM NEAR TRACK CENTERLINE

0.3 m (8 to 12 in.) thick, depending on tunnel excavation. Unfortunatley, a direct comparison of vibration produced by these circular tunnels (of differing mass, all other parameters remaining unchanged) is not possible using the available data.

8.1.6.3 Effect of Aerial Structure Foundation Mass -- Although no data concerning the effect of aerial structure foundation design are available, a basic understanding can be gained from the literature on machinery foundation design (Ref. 8.30, 8.31). The dynamics of an aerial structure foundation and a machine foundation are similar. Therefore, at audible frequencies, a doubling of aerial structure foundation and support column mass should result in a 6 dB reduction of vibration. However, simply increasing the mass may result in significant amplifications of ground amplitudes at the fundamental resonance of the foundation.

8.1.6.4 Groundborne Vibration From Ballast-and-Tie Track -- Very limited information is available on how design variations of ballast-and-tie track influence the levels of wayside groundborne vibration. Limited measurements at Canadian railway facilities indicate that embankment height or cross-section area may have a significant effect on groundborne vibration; the higher the embankment, the lower the levels of vibration at the wayside. At best, this can be considered a preliminary conclusion.

Recent research in Sweden (Ref. 8.32) indicates that substantial reduction of groundborne vibration velocity in nearby buildings may be achieved by supporting the rail trackbed on piles extending down to bedrock. Evidently, the increased foundation reaction obtained through this technique impedes the motion of the trackbed, thus restricting the radiation of vibration into the soil.

8.1.7 Propagation of Vibration in Soil and Rock

Several types of waves can be excited by the passage of a transit train. These wave types are discussed in detail in Reference 8.31 and may be listed in two groups:

Body Waves

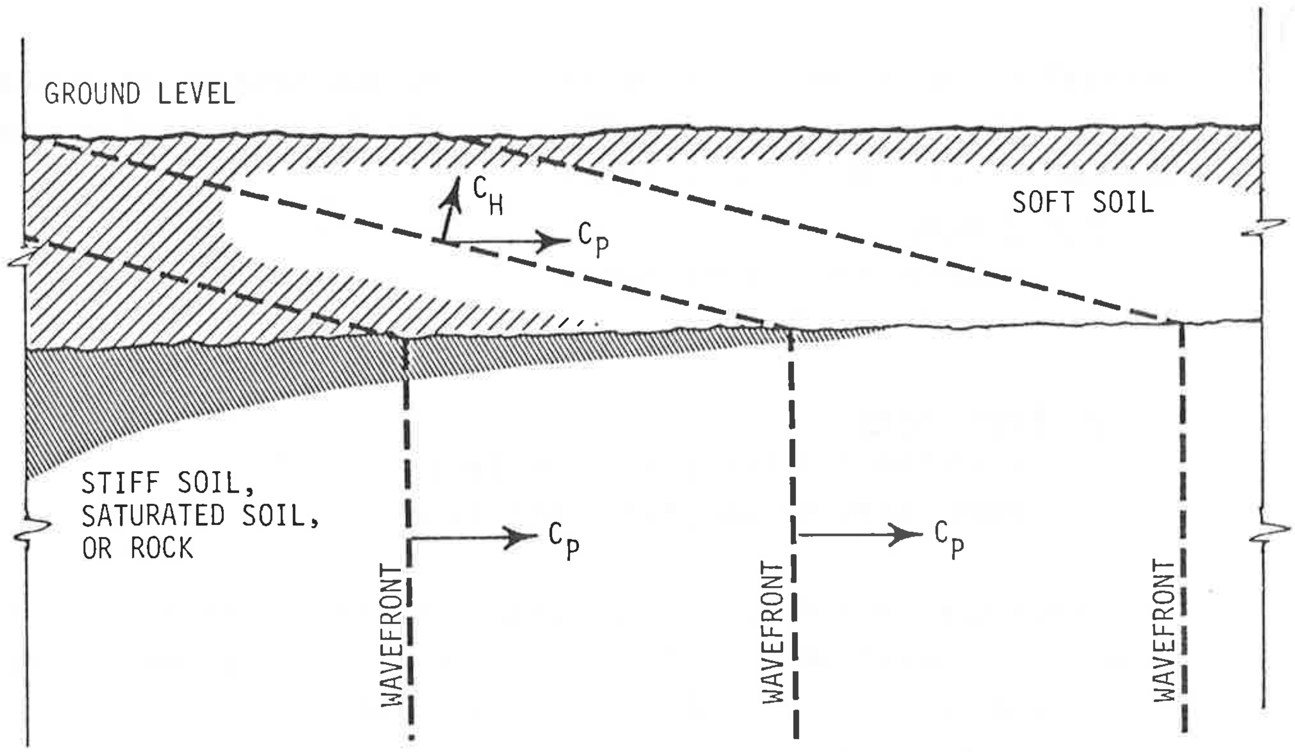
1. Dilatational (Compression)
2. Rotational (Shear)

Surface Waves

1. Rayleigh (at the ground surface)
2. Love (between subgrade soil layers)

Each wave type has different propagation speeds and different rates of attenuation with distance. Thus, the development of any comprehensive model of groundborne noise and vibration requires an understanding of the relative importance of the various wave types. However, little information is available concerning the partition of vibration energy between these wave types, at least for typical transit system sources. Ungar and Bender (Ref. 8.33) assume that the compression wave is the most significant for groundborne noise in buildings near subways. However, measurements have shown that either shear or Rayleigh surface waves are predominant for groundborne vibration from at-grade, ballast-and-tie track (Refs. 8.32, 8.35). Shear waves may also be a very important component of the vibrational energy propagated from subway structures, but no definite data have been presented.

An additional wave type which has not received attention is the so-called "headwave," illustrated in Figure 8.11. A headwave is produced by a compression wave propagation at high speed, with little attenuation, in a subsurface soil layer or water table. A combined shear and compression wave can result in the upper soil layer that propagates at a velocity equal to that of the compression wave in the lower layer. Propagation within a water table which creates a headwave could result in very efficient propagation of groundborne noise and vibration.



C_H = PHASE VELOCITY OF HEADWAVE NORMAL TO WAVE-FRONT IN SOFT UPPER SOIL LAYER

C_P = PHASE VELOCITY OF HEADWAVE IN SOFT SOIL AND IN STIFF SUBSTRATA IN HORIZONTAL DIRECTION

FIGURE 8.11 ILLUSTRATION OF HEADWAVE IN UPPER SOIL LAYER CAUSED BY FASTER WAVE IN LOWER STRATA

TABLE 8-4 SPREADING LOSS VALUES FOR COEFFICIENT "K"

Wave Type	Source Distribution Point	Source Distribution Line
Body	-20	-10
Surface	-10	0
Headwave	-10	0

TABLE 8-5 FREQUENCY-DEPENDENT SOIL DAMPING (REF. 8.35)

$$\Delta L_D(\text{dB}) = -\alpha(f/c)x$$

Δ is the attenuation in dB

α is a loss-factor in dB per wavelength

f is the frequency in Hz

c is the propagation velocity in m/sec

x is the distance in meters over which the attenuation is predicted

Representative values:

Soil Type	α(dB/wavelength)	c(m/sec) (compression)	c(m/sec) (shear)
Clay	4.4	1,500	150
Loess	2.6	800	260
Sand	0.86	500	150

The attenuation of vibration level in dB as a function of distance from the source is represented by the sum of the attenuation due to geometric spreading loss and to material damping in the soil.

8.1.7.1 Geometric Spreading Loss -- Geometric spreading is the attenuation of ground vibration caused by the spreading of the vibration energy as it propagates away from the source. Spreading loss may be represented as:

$$\text{Attenuation (dB)} = K \log(r/r_o) \quad (8.2)$$

where r_o is the distance between the source and a reference location (a point where the vibration level is known), r is the distance between the source and the point where the level is to be predicted, and K is a dimensionless constant, the value of which depends on the character of the source and the type of wave (e.g., surface or body wave). Theoretical values for K are given in Table 8-4. In very stiff soils or rock with low material damping, geometric spreading will be the most significant cause of vibration attenuation at locations relatively close (less than 15 m) to the transit structure. Spreading loss is independent of wavelength and frequency, therefore representing the simplest and most conservative method of estimating attenuation with distance.

8.1.7.2 Material Damping in Soil -- In the average soil, material damping is of much greater significance than geometric spreading loss, especially in the audible frequency range. Detailed discussions of material damping for various soils can be found in the literature (Refs. 8.30, 8.31, 8.35). Material damping is strongly dependent on frequency. One method of accounting for this frequency dependence is presented in Table 8-5. In Loess, the damping of compression waves, due to material damping alone, will be about 0.09 dB/m at 25 Hz. For shear waves, it is about 0.24 dB/m. At higher frequencies, the damping will be proportionately higher. Implicit in Table 8-5 is the assumption that the loss factor in dB per wavelength is independent of the type of wave, whether compression, shear, or Rayleigh surface.

An alternative formulation of material damping relies on a model independent of frequency or wavelength. At higher frequencies, this model is more conservative than the frequency-dependent model. The frequency-independent model is summarized in Table 8-6. It predicts higher damping at low frequencies than the frequency-dependent model does; the opposite is true at high frequencies.

Both models represent damping in terms of dB per unit distance, while attenuation caused by geometric spreading is represented in terms of the ratios of distances, e.g., dB per doubling of distance. Geometric spreading loss will thus dominate at locations near to the source whereas material damping will dominate at locations far from the source.

Neither of the above models for prediction of attenuation with distance has been universally accepted. In fact, Gutowski and Dym (Ref. 8.35) report that dissipative attenuation in dB is proportional to the logarithm of the number of wavelengths traveled, in opposition to both of the above models. Obviously, additional work in this area is needed.

8.1.8 Coupling of Buildings to Ground Motion

A given building's response to ground vibration is very complex and perhaps impractical to predict analytically. The building foundation is necessarily distributed over a relatively large area in comparison with the wavelength of vibration in soil for audible frequencies. It may therefore not be valid to model the building as a rigid body on an elastic half-space. Further complicating the problem, the vibration of the ground cannot be viewed as independent of the building foundation vibration.

TABLE 8-6 FREQUENCY-INDEPENDENT DAMPING (REF. 8.35)

$$\Delta(dB) = -a\,x$$

Δ is the attenuation in dB

a is a damping coefficient in dB per meter

x is the distance in meters over which the vibration propagates

Representative values for "a" in dB/m	
Clay	0.35 to 1.0
Loess	0.86
Sand & Silt	0.35

Although a building and foundation are necessarily a multi-degree-of-freedom system (at audible frequencies, the modal vibration force relationships are quite complex), significant cancellation of building vibratory force loads will occur at the foundation. The result is that at audible frequencies, the foundation response to ground vibration may be primarily determined by its mass and geometry.

Building foundations can be divided into four basic types:

- Slab-on-grade.
- Foundation on spread-footings.
- Foundation on piles founded in earth.
- Foundation on piles supported by rock.

Generally, basement floors and certain residential buildings can be considered the same as concrete, slab-on-grade floor. The surface area of the slab is large, and the slab itself is in intimate contact with the underlying soil. Therefore, the vibration of such floor slabs is similar to that which would exist in the soil without the slab, giving a coupling loss of 0 dB -- at least for low frequencies -- and up to the resonance frequency of the slab on the soil (usually in the range of 200 to 400 Hz). However, the vibration of the ground surface may be changed by the presence of the concrete slab if the slab is sufficiently heavy. In foundations built on piles founded on rock, the effective mass of the foundation may be much greater than the actual mass of the foundation: therefore, the vertical response to ground vibration in the earth will be reduced. However, for subway or aerial structures founded on the same bedrock as the building foundation, the vibration of the rock is used to estimate foundation vibration, with the coupling loss generally assumed to be negligible.

For the remaining foundation types, Wilson (Ref. 8.36) supplies empirical curves for estimating foundation vibration levels from ground surface vibration levels. These curves are presented in Figure 8.12.

Floors, walls, and ceilings are subject to significant amplifications of vibration, relative to the foundation vibration. This effect is especially noticeable in residential wood frame houses. In many cases, the resonance of the floor assembly may cause amplification of the vibration in the 10 to 30 Hz range. Problems arise when these floor resonances coincide with peaks in the groundborne vibration spectrum. The amplification of floor, ceilings, and walls is difficult to predict, but it is typically in the range of 5 to 15 dB in the frequency range of 16 to 80 Hz.

8.1.9 Propagation of Vibration Through Buildings

The propagation of vibration through a building structure and the response of walls, floors, and ceilings to vibration are very complex. In multistory buildings, a common value for the attenuation of vibration from floor-to-floor is approximately 3 dB; however, Ishii and Tachibana (Ref. 8.37) show approximately 1 db floor-to-floor attenuation in the upper floor regions at low frequencies and greater than 3 dB attenuation at lower floors. These data result from experiments in floor-to-floor vibration attenuation in a 10-story, steel framed, reinforced concrete building (summarized in Table 8-7) for a point vibration source below the building. The figures given are reasonable for uses in predicted propagation of structureborne vibration from trains even though a train is not a point source.

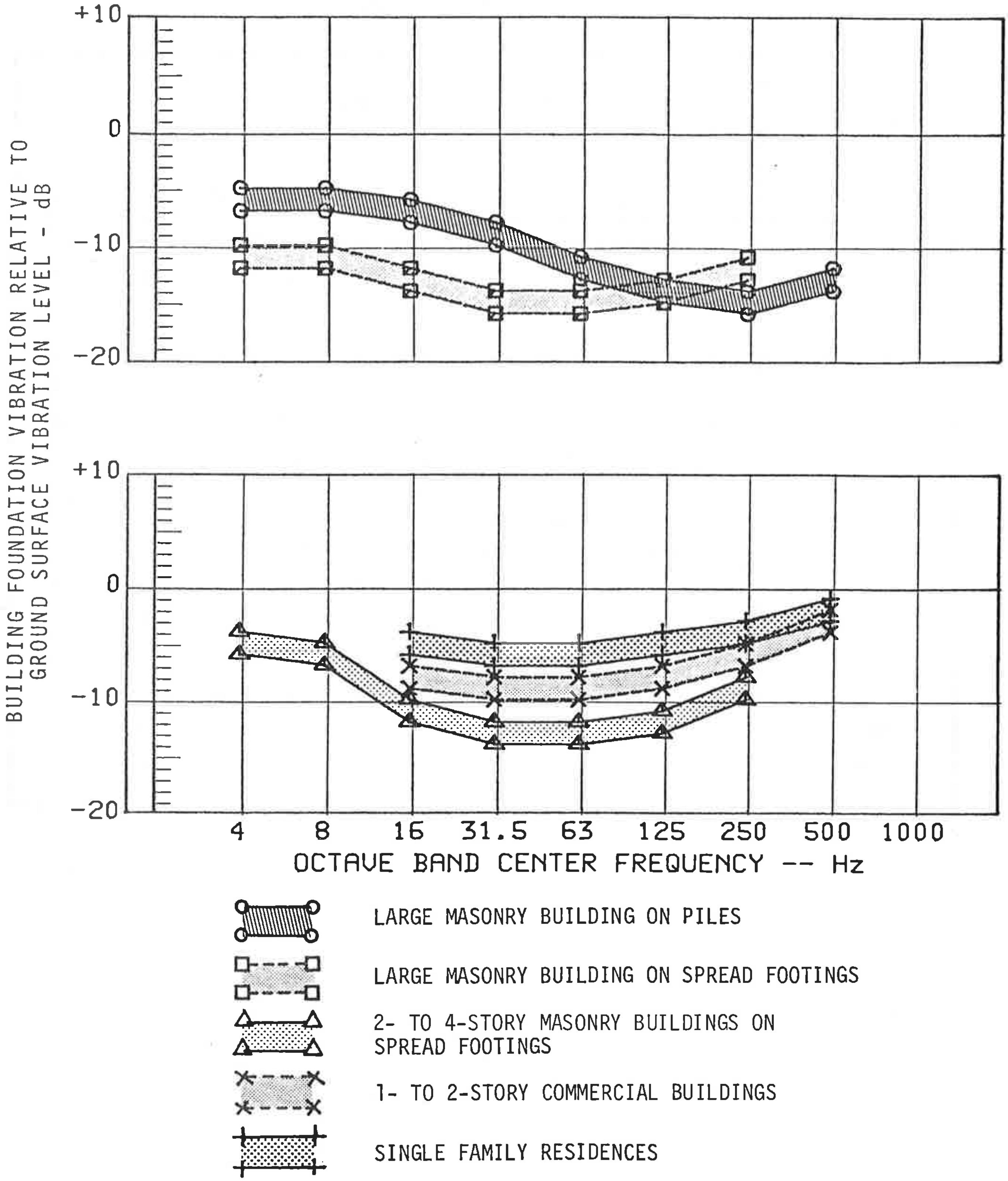


FIGURE 8.12 COUPLING LOSSES FOR BUILDING FOUNDATIONS

TABLE 8-7 POINT SOURCE BELOW BUILDING - ATTENUATION PER FLOOR WITH ACCELERATION IN dB

Frequency Hz	Floor Level Above Grade									
	1	2	3	4	5	6	7	8	9	10
	Floor-to-Floor Distance: 10 ft									
31	2	2	2	1	1	1	1	1	1	1
63	3	2	2	2	2	1	1	1	1	1
125	3	3	2	2	2	2	2	1	1	1
250	3	3	3	3	3	3	3	2	2	2
500	4	4	3	3	3	3	3	3	3	3
1K	5	5	4	4	4	4	4	3	3	3
	Floor-to-Floor Distance: 12 ft									
31	2	2	2	2	1	1	1	1	1	1
63	3	3	2	2	2	1	1	1	1	1
125	3	3	3	2	2	2	2	1	1	1
250	4	4	3	3	3	2	2	2	2	2
500	4	4	4	4	4	3	3	3	3	3
1K	5	5	5	4	4	4	4	4	4	4

8.1.10 Interior Noise Levels

Groundborne noise is the result of acoustic waves radiating from the vibrating walls, floors, and ceilings of a room surface. The relationship between the vibration of the room surfaces and the noise level is dependent on the amount of absorption in the room, the room size and shape, and the distribution of the vibration velocity level on the room surfaces. Figure 8.13 presents the results of simultaneous measurements of floor vibration and noise level during train passbys (Ref. 8.36). The average relationship is:

$$L_p = L_a - 20 \log(f) + 36 \qquad (8.3)$$

where:

L_p = sound pressure level, dB
L_a = rms vibration acceleration level for the floor, (dB re 10^{-6} g), and
f = frequency, either octave band or 1/3 octave band center frequency, Hz

Although the data are rather widely spread about the mean line, the relationship of Equation 8.3 is identical to the approximate relationship derived from theoretical considerations.

8.1.11 Train Speed

Train speed has a significant effect upon groundborne noise and vibration, though less than is often expected. Vibration levels as a function of train speed vary from 10 log V to 20 log V; that is, a doubling of train speed results in an increase of about 3 to 6 dB in the groundborne vibration and noise level.

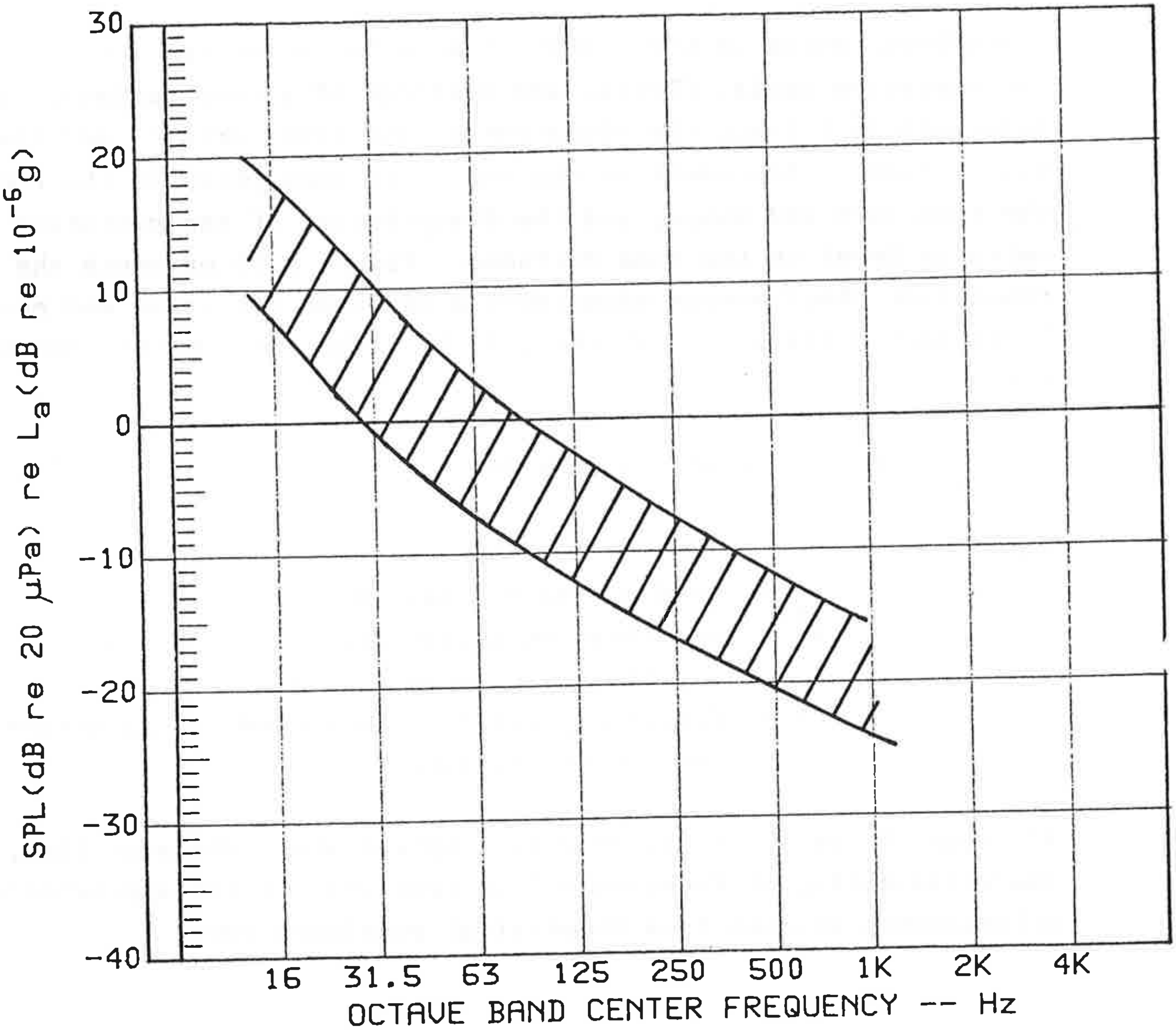


FIGURE 8.13 ROOM INTERIOR SOUND PRESSURE LEVEL RELATIVE TO AVERAGE FLOOR ACCELERATION LEVEL

8.2 PREDICTION OF GROUNDBORNE NOISE AND VIBRATION

The purpose of this section is to present information that can be used to obtain reasonable estimates of expected groundborne noise and to supply some background on prediction techniques. It is important to recognize that because the prediction of groundborne noise and vibration is empirically based, extreme caution must be used when evaluating the need for vibration control. The predictions should be performed by someone who is familiar with the most recent advances in the field and with the ramifications of various design parameters.

8.2.1 Background

Most groundborne noise and vibration prediction procedures are relatively simple. A series of empirical corrections are applied to a reference vibration spectrum or reference overall level to account for tunnel mass, distance to affected buildings, ground type, vibration control measures, fastener type, building type, train speed, and train length. The starting point for most of the groundborne vibration prediction methods is an assumed octave band vibration spectrum, based on measured levels, either at the subway wall or invert or at some standardized distance from the subway structure.

None of the prediction procedures account analytically for the radiation characteristics of subway structures. Wilson (Refs. 8.36, 8.38) provides octave band and overall level corrections for heavy box subways, circular concrete tunnels, and circular steel tunnels. He accounts for the radiation characteristics of the tunnels by starting with the vibration spectrum in soil or rock at 7.5 m (25 ft) from the structure walls. Ungar and Bender (Refs. 8.33, 8.39) assume that the vibration in soil outside of the subway wall is equal to that of the subway wall. The vibration attenuates with increasing distance from the subway by geometric spreading and dissipation. This approach ignores possible decoupling at low frequencies of the subway structure from the surrounding soil, brought on either by wavelengths in the soil in excess of the subway dimensions or by wave propagation velocities in the subway structure lower than in the surrounding soil.

Wilson accounts for attenuation with distance within soils by application of attenuation curves for each octave band of vibration in average soils. These curves are largely based on data from the BART test track and on data obtained at the Toronto Transit System, where relatively high levels of groundborne noise have been observed and documented.

The method used by Ungar and Bender incorporates the propagation velocity and loss factor for a given soil type, but recommends that only the compression wave parameters be used. Ungar and Bender's discounting of the shear wave relies on two assumptions:

- The subway structure does not radiate shear waves effectively.

- Shear waves, compared to compression waves, are heavily damped over comparable distances.

However, these assumptions ignore wave conversion at various interfaces between soil strata and at the surface. Furthermore, analytical work by Miller and Pursey (Ref. 8.40) indicates that if a circular footing (resting on the surface of an elastic half-space with Poisson ratio 0.25) is vibrating vertically, 65% of the total energy is radiated away in the form of Rayleigh waves, 26% as shear waves, and only 7% is radiated as compression waves. Although a buried subway is not directly comparable to a circular footing at the ground surface, compression wave energy appears to be significantly less than shear wave energy.

Tokita, et al. (Ref. 8.41) uses a method similar to Ungar and Bender, with apparently favorable comparisons between measured and predicted overall vibration levels as a function of frequency. These authors assume geometric spreading from a line source and a correction formula for impedance mismatch between soil strata. The correction formula accounts for the angle of incidence and for transmission through the interface.

Building coupling loss is defined as the difference between building foundation vibration and the incident ground vibration in the absence of the buildings. This coupling loss has been studied analytically by Bender, et al. (Ref. 8.39) using simple models of building piles and structure loading and measurements performed by Blazier (Ref. 8.42). Wilson presents a set of octave band correction factors, based upon empirical data collected in Toronto and in Washington, D.C. Coupling losses vary according to whether the building is a relatively small, two- to four-story building, a large masonry on spread footings, or a large masonry on pilings. Floor vibration at ground level for slab-on-grade construction is usually taken as equal to the incident groundborne vibration, which is similar to the approach taken by Ungar and Bender in estimating basement wall vibration (Ref. 8.33).

Ungar and Bender assume the propagation of vibration through the building structure to decrease by about 3 dB for each floor (Ref. 8.39). Wilson, however, stresses that in lightweight frame buildings, little attenuation from floor to floor may be expected, as well as little coupling loss (Ref. 8.36). Recently, Wilson's prediction method has been modified to incorporate the research results of Ishii and Tachibana's (Ref. 8.37) estimations of floor-to-floor attenuation of structureborne vibration in large multistory buildings.

The conversion of groundborne vibration levels to interior noise levels has been approached in a number of ways. Kurzweil (Ref. 8.1) presents a very simple, empirically based formula for estimation of A-weighted noise levels, as a function of distance from the subway structure, based on work by Lange (Ref. 8.27).

This formula does not account for spectral characteristics and assumes a data spread of ± 10 dB. This approach is useful as a first estimate, but knowledge of the groundborne vibration or noise spectrum is required to evaluate the vibration control effectiveness of noise control provisions during design.

At TTC, the analysis of many simultaneous measurements of building floor vibration and sound levels in residential structures showed that the average sound pressure level can be directly related to the vibration velocity level (Ref. 8.15). The data were widely spread around the best fit line, but accounting for such factors as acoustic absorption, surface area, and room volume would reduce the scatter. A similar result was reported by Kalic (Ref. 8.43), who derived essentially the same relationship empirically in Reference 8.15. A formula which also includes the effect of the average room absorption coefficient is presented in Reference 8.39.

A significant limitation of existing prediction methods is that they are mainly designed to estimate A-weighted sound pressure levels or to derive sound pressure levels to compare with NC curves. Vibration levels below 30 Hz have not been emphasized. Recent experience at the WMATA and MARTA systems has pointed out the need for evaluating vibration in the range of 10 Hz to 30 Hz in order to predict mechanical motion and secondary noise radiation (rattling dishes, etc.). Most floating slabs in particular have been designed to reduce audible noise, since the vibration isolation system has a resonance frequency of 12 to 16 Hz. As a result, low-frequency vibration is not attenuated. In fact, coincidence with other resonances, such as building floor resonances, can amplify low-frequency vibration at the floating-slab resonance frequency. This would adversely affect the vibration environment at some residential locations (Ref. 8.44). Therefore, the prediction of groundborne noise and vibration must include the subaudible range.

8.2.2 Groundborne Noise from Tunnels, a Simple Estimate

It is always difficult to estimate groundborne noise and vibration accurately. This problem is often further complicated by the lack of detailed information on tunnel construction, vehicle parameters, dynamic characteristics of the trackbed, and dynamic properties of the soil. Lange (Ref. 8.27) has developed an approximate relationship for the A-weighted sound level, $L_A(R)$, in basement rooms between 1 and 20 m (3 to 65 ft) from the subway:

$$L_A(R) = 59 - 20 \log(R R_o) + 10 \text{ dBA} \qquad (8.4)$$

where R is the distance from the tunnel to the building and R_o a reference distance of 1 m (3.3 ft). There is some question as to whether R should be the distance from the building wall to the tunnel wall or the distance from the building to the track centerline. However, since this is a very approximate estimate, either distance will provide the same approximate accuracy. The data used to derive Equation 8.4 are based on a wide range of vehicle types, condition, and speeds; track types and conditions; tunnel and building constructions; and soil types. In general, systems with poor wheel and rail conditions, stiff rail fastening designs, or jointed rail will produce higher vibration levels than systems with smooth wheels and rails, resilient fasteners or ballast-and-tie trackbed, and welded rails. Furthermore, under similar conditions, the levels in lightweight wood frame buildings will be higher than those in heavy masonry buildings on piles or spread footings.

Equation 8.4 can be quite useful in the preliminary planning stages of a new transit line. However, it cannot be used in any serious evaluation of environmental impact or for final design considerations. Once beyond the preliminary planning stages, more detailed techniques are necessary to account for the specific parameters of the transit line under consideration. One such method is presented in the following section.

8.2.3 Prediction of Groundborne Noise from Tunnels, a Detailed Estimate

Figure 8.14 illustrates the steps for estimating the maximum groundborne noise and vibration levels and the need for control procedures. The starting point is the octave band spectrum of the ground vibration at a distance of 7.5 m (25 ft) from the transit structure. Since the vibration is strongly dependent upon the transit structure, the vehicle type, and the type of soil, it is necessary to develop a standard spectrum for each major structure type. Figure 8.15 shows the beginning spectra used for the original predictions at WMATA, along with those developed from measurements after the system was operational. The original curves were based on measurements at TTC and the BART Test Track facilities with similar vehicles and track, but with soil conditions considerably different than in the Washington, D. C. area.

Notice that the curves in Figure 8.15 are based on the maximum expected levels of groundborne noise and vibration during a train passby. Unfortunately, insufficient data are available to allow a firm statistical basis for these levels. Experience, however, has shown that the levels will be exceeded only in relatively unique situations.

Once the maximum expected level at 7.5 m has been determined, the vibration levels are shifted up or down to account for speed variations, type of trackwork, and structure type. The correction for speed can be determined using the relationship:

$$L = 20 \log (V/V_o) \qquad (8.5\ )$$

where V is the train speed; V_o is the train speed for the standard spectrum; and L is the number of decibels by which the spectrum is adjusted.

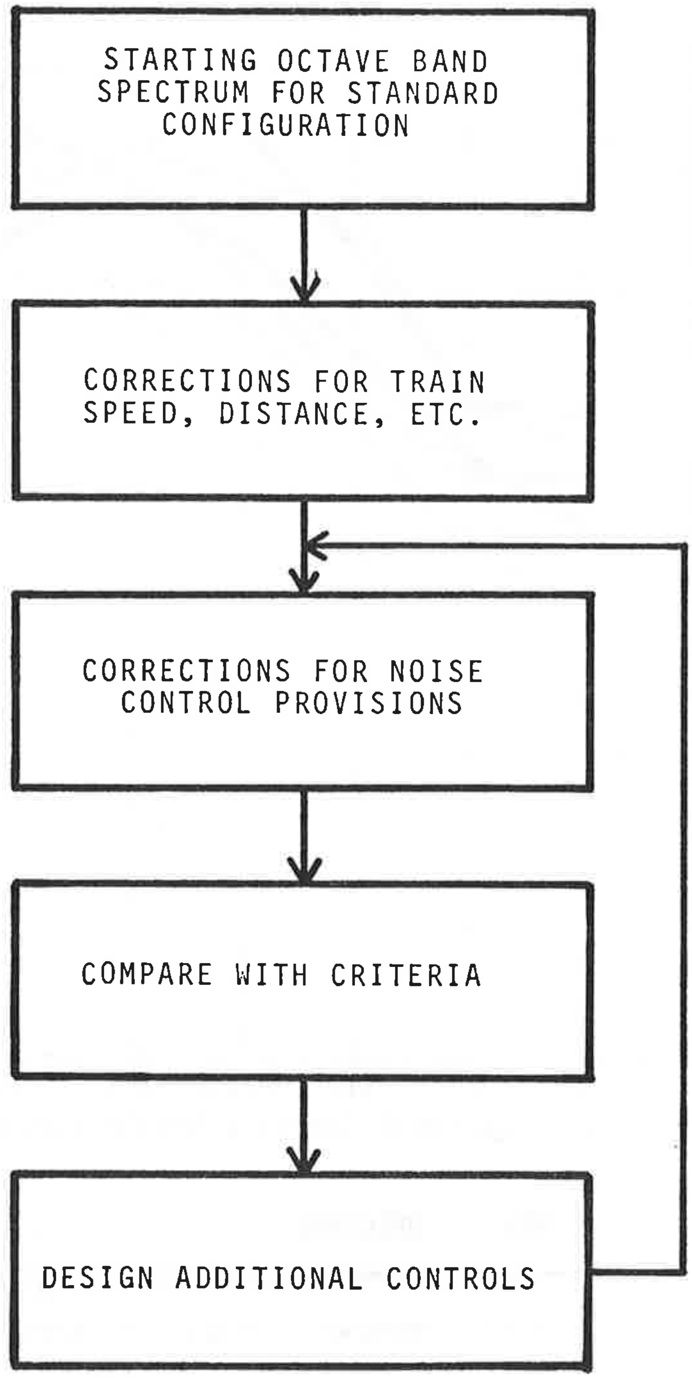


FIGURE 8.14 ESTIMATION PROCEDURE FOR GROUNDBORNE NOISE AND VIBRATION

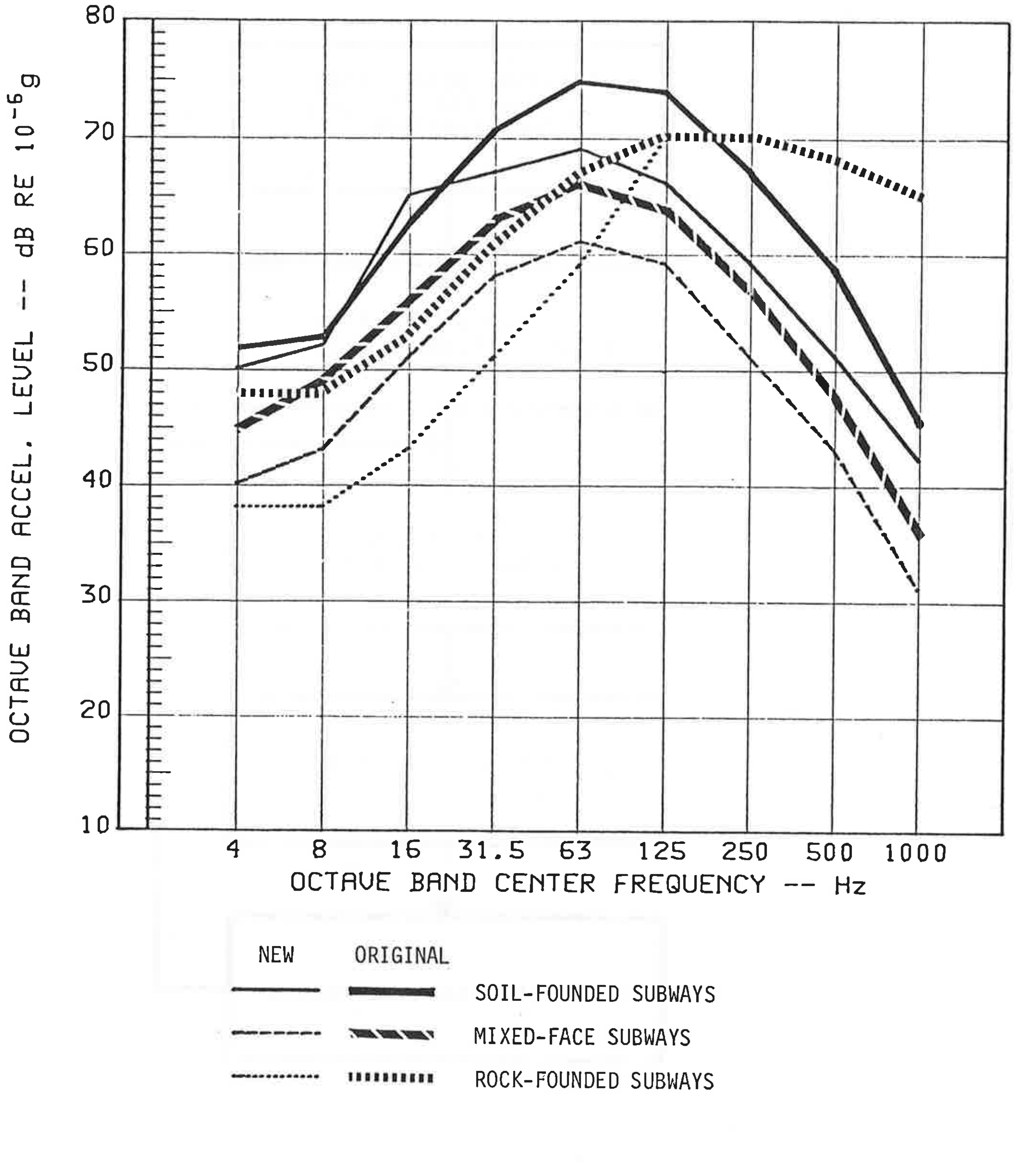


FIGURE 8.15 MAXIMUM EXPECTED VIBRATION LEVELS AT 7.5 to 9 m (25 TO 30 FT) FROM TRACK CENTERLINE AT THE TOP-OF-RAIL

The adjustments to be made for special trackwork and for the type of structure are given in Table 8-8. The corrections for tunnel structure type only apply to tunnels founded on soil. If the structure is founded in or on rock, the impedance of the concrete invert and tunnel liners is assumed to be approximately equal to that of the rock, so the specific type of tunnel construction will not have a strong influence on the vibration levels.

The corrections for tunnel structure types are based on the assumption that the mass of the tunnel is the most important factor. As discussed in Section 8.1.6, Reference 8.29 indicates that doubling subway wall thickness, hence doubling the mass per unit length, will reduce the groundborne vibration by 12 to 18 dB. This has not been verified by other researchers, and data support the generalized corrections given in Table 8-8, hence use of the listed factors is recommended.

Once the spectrum for the specific tunnel type, train speed, etc. has been established, the next step is to estimate the attenuation caused by geometric spreading and internal damping as vibration propagates through the ground. Figure 8.16 presents a chart that can be used to estimate the amount of vibration attenuation that occurs through geometric spreading. Since the internal damping of rock is relatively low, this curve can be used directly to estimate the attenuation with distance from a rock subway tunnel. Because geometric spreading dominates, the attenuation with distance is not frequency dependent.

The curves of Figure 8.17 can be used to estimate the attenuation in soil caused by dissipation and geometric spreading. The figure, extracted from Reference 8.36, is based on empirically determined values of soil dissipation listed in the literature and on measurements of attenuation with distance from soil-based subways. This curve was developed for "typical" soil; the attenuation is less rapid in very stiff soil. Figure 8.17 emphasizes the fact that the attenuation of vibration in soil is strongly frequency dependent. The high frequencies attenuate much

TABLE 8-8 GROUNDBORNE VIBRATION LEVEL ADJUSTMENTS FOR SPECIAL TRACKWORK AND TYPE OF STRUCTURE

Factor Affecting Vibration Level	Relative Vibration Level - decibels Octave Band Center Frequency - Hz 16	31.5	63	125	250
Special Trackwork	+10	+10	+10	+10	+10
Subway Structure - soil-founded structures only					
Double box	0	0	0	0	0
Single box or concrete tunnel	-3	-3	0	+5	+5
Cast iron or steel tunnel	-3	-3	+1	+6	+6
Triple box or crossover structure	0	0	-2	-2	-2
Station	0	-1	-3	-4	-4

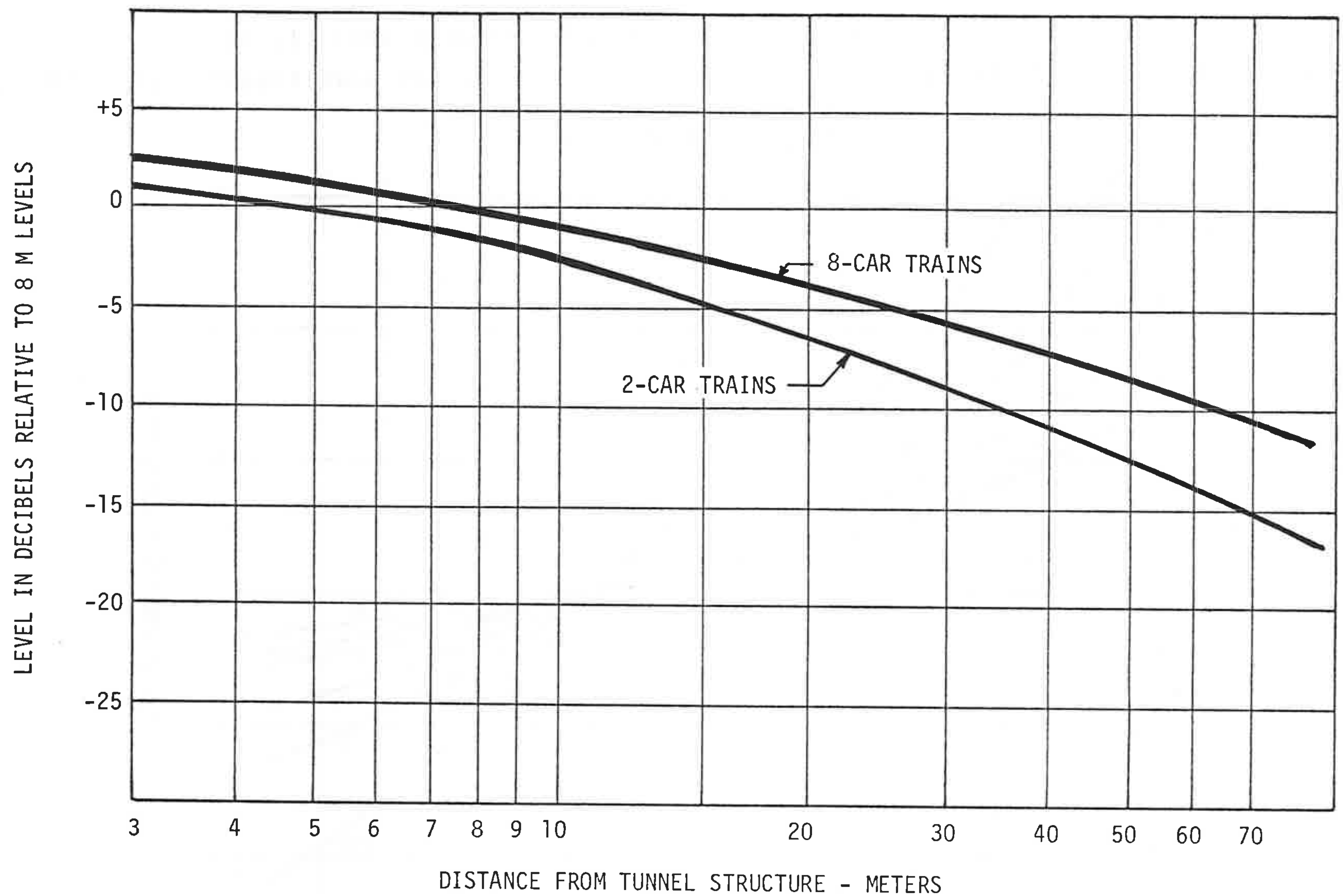


FIGURE 8.16 VIBRATION LEVELS IN ROCK AS A FUNCTION OF DISTANCE FROM A ROCK TUNNEL STRUCTURE

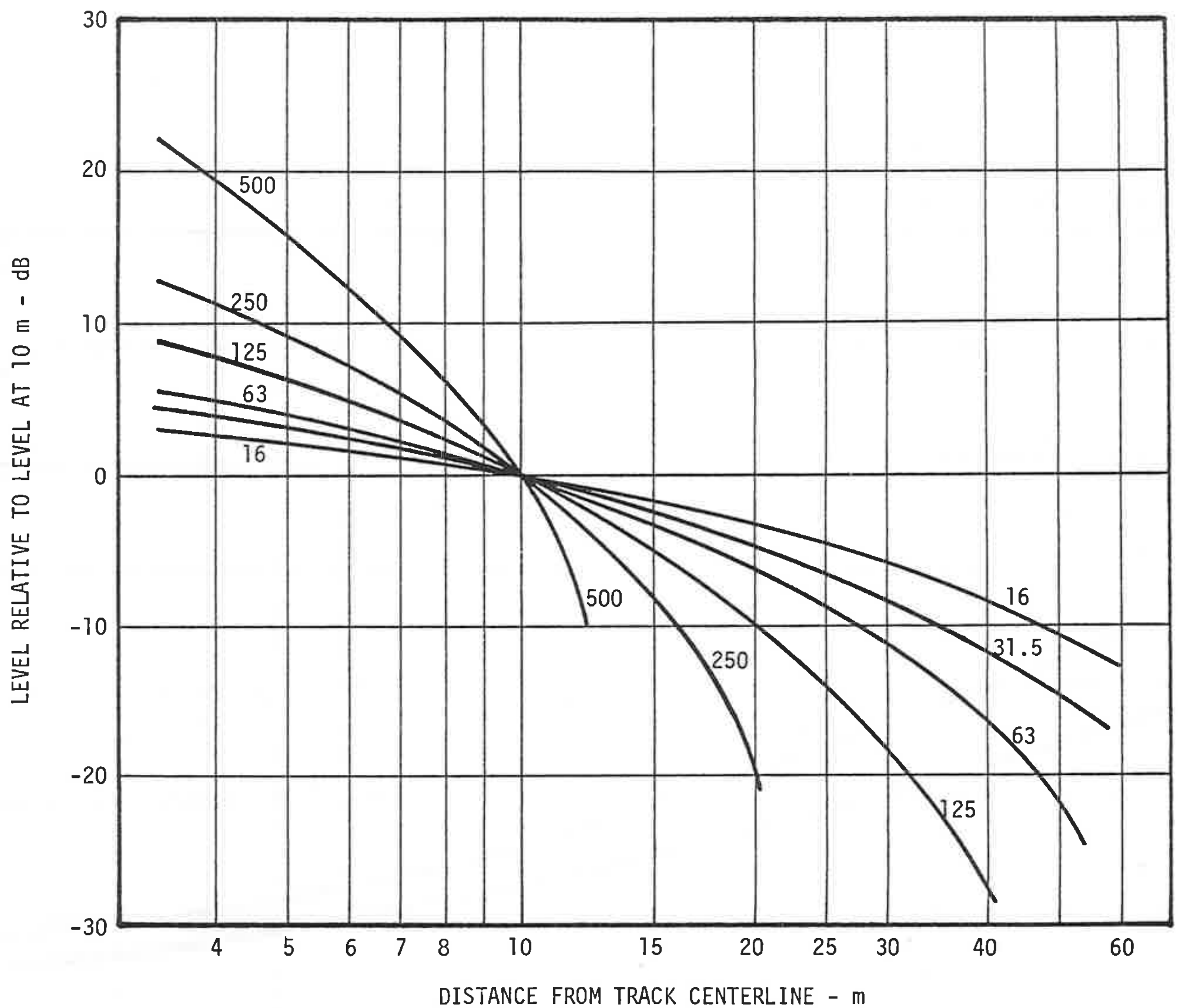


FIGURE 8.17 APPROXIMATE GROUND VIBRATION LEVELS RELATIVE TO THE LEVELS IN SOIL AT 10 m (33 FT) FROM A SOIL-BASED OR MIXED-FACE SUBWAY

more rapidly than the low frequencies.

At this point, the information presented allows estimation of vibration at the ground surface some distance away from the subway structure. It can be assumed that the vibration at the foundation of a lightweight structure is the same as the ground surface vibration without the structure. Heavier buildings, especially buildings on spread footings or piles, will cause coupling loss; that is, the vibration of the ground floor of the building will be less than that of the ground. Figure 8.12 shows curves that can be used to estimate the coupling loss. These curves were empirically derived from simultaneous building floor and ground surface vibration measurements.

Table 8-7 may then be used to estimate floor-to-floor attenuation. Once the vibration of the floor of a building has been estimated, the levels can be compared with the appropriate criterion, and the need for measures to control mechanical vibration can be evaluated.

In many cases, even when the vibration criteria are not exceeded, excessive noise may be caused by acoustic radiation from the vibrating room surfaces. It is therefore necessary to estimate levels of groundborne noise that will be caused by vibration, which is the final step in the estimation procedure.

Figure 8.13 indicates the approximate relationship between sound pressure level and acceleration or velocity level at the floor of a room. The chart is based on measurement data with typical sound absorption; a range of results can be expected, depending on whether the room has typical, unusually large, or small amounts of sound absorption. The chart is intended to cover a range from typical, highly reverberant spaces to spaces that are acoustically dead. Given the octave band sound pressure level, one can evaluate the acceptability by either plotting the levels and comparing NC curves or by calculating the A-weighted levels.

Note that the above outlined procedure covers only the cases for soil-founded subway and soil-founded building or for rock-founded subway and rock-founded building. For cases involving mixed-faced structures or soil and rock layers between the subway and the building, the propagation estimates are more complex and require consideration of the effects of interfaces and differing propagation in different strata. Such procedures are beyond the scope of the handbook.

8.3 FLOATING-SLAB DESIGN

In critical locations, control of rail transit groundborne noise and vibration requires special vibration reduction features beyond such standard design features as smoothly ground continuous welded rail or resilient rail support on rigid invert. One of the most practical means yet developed to provide this reduction is the floating-slab trackbed. Operations of transit trains on floating-slab track have shown that floating slabs effectively reduce the groundborne noise. Their performance resembles the mathematically predicted performance of the single-degree-of-freedom isolator system (Refs. 8.16 - 8.21).

The floating slab consists of a concrete slab supported on resilient pads. The design is similar to the inertia bases on springs that are used to support stationary machines. The floating-slab trackbed reduces noise intrusion in nearby buildings and allows new rail transit subways to be placed closer to buildings than might otherwise be practical. Because they are expensive, floating slabs should be used only where extra reduction of groundborne noise or vibration is required.

A number of designs for vibration isolated trackbed have been developed, ranging from heavy, bridgelike structures with thick rubber support pads and damping applied to the bridge deck (Ref. 8.45) to relatively lightweight concrete slabs without damping, supported on resilient pads. The lightweight concrete slabs have evolved into two basic forms: continuous cast-in-place slabs and discontinuous precast slabs (Figs. 8.18, 8.19). The first extensive use of floating slabs in North America was on the WMATA Metro System, using the cast-in-place continuous slab design. Trains have been operating on these slabs since 1975 with excellent results. The discontinuous floating slab has been used at TTC (Toronto), MARTA (Atlanta), MURLA (Melborne), and in Hong Kong where both continuous and discontinuous designs were adopted. The basic differences between the two systems are that the continuous floating slab is cast-in-place, using a sheet metal form which remains in place, whereas the discontinuous floating slab is constructed using precast concrete block elements. The discontinuous floating slab more closely approximates a single-degree-of-freedom vibration isolator system, whereas the continuous floating slab, because of its distributed nature, may support bending waves reducing vibration isolation effectiveness. Tests on the WMATA system indicate that in single track slabs, the continuous floating slab performs to expectations based on the single-degree-of-freedom isolator model (Ref. 8.20). However, wider two track or crossover slabs showed less favorable results until joints were introduced to limit the size of continuous slabs.

8.3.1 Design Parameters

Floating slabs have three frequency regimens: the low-frequency, stiffness-controlled range; the resonant, damping-controlled range; and the high-frequency, mass-controlled range. There is essentially no reduction of the transmitted vibration in the low-frequency range. There is a small amplification of the transmission in the damping-controlled range near the resonance frequency (this amplification depends upon the effective damping factor of the system). At frequencies above the resonance frequency, the reduction of vibration transmission increases with frequency and depends on mass ratios and damping.

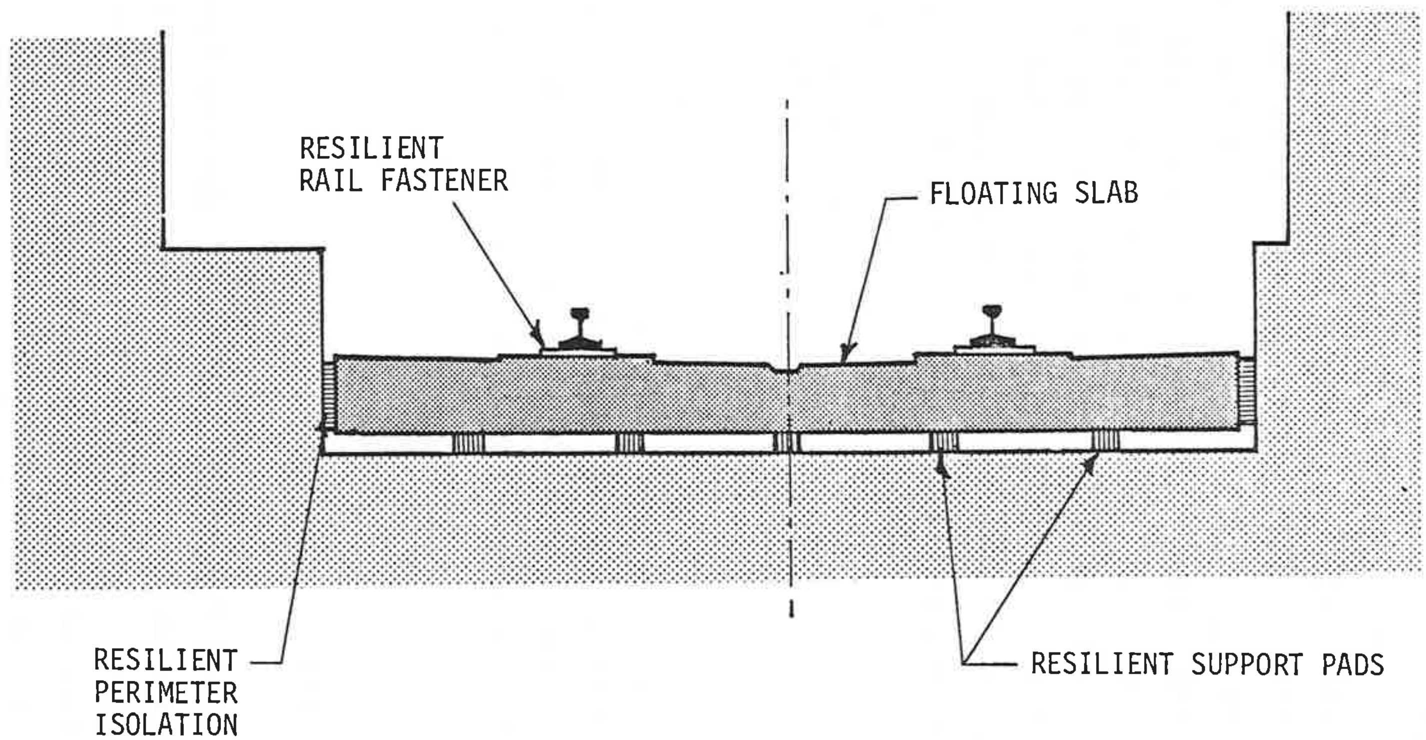


FIGURE 8.18 SECTION OF CONTINUOUS CAST-IN-PLACE FLOATING SLAB FOR CONCRETE BOX STRUCTURE

To control structureborne noise from a machine, the natural frequency of an inertial mass on the supporting resilient mounts must be below 70% of the operating frequency of the machinery. The primary design consideration for floating-slab trackbed systems is that the fundamental vertical natural frequency of the slab-support system be low enough to provide adequate vibration isolation of groundbourne vibration and audible noise.

Because audible noise inside buildings adjacent to the subway is the primary form of intrusion, floating-slab designs focus on controlling groundborne noise in the audible frequency range. Recent experience on the WMATA, MARTA, and NYCTA systems indicates that in some locations, significant intrusion (perceived as vibratory motion) from groundborne vibration can occur, especially if the vehicle's primary suspension resonance is approximately equal to the floating-slab resonance. This is complicated by the fact that many wood frame structures have fundamental resonances in the neighborhood of 10 to 20 Hz. In these cases, the slab resonance frequency should be lowered to avoid coincident amplification of groundborne vibration.

To provide noise and vibration reduction over the audible-- frequency range, the vertical motion, natural frequency of the slab-support system should be less than 15 Hz when loaded with the weight of the car bodies and truck. The lower the vertical natural frequency, the more effective the vibration isolation. Increasing the mass of the floating slab lowers its frequency, however, cost and available space often limit the amount of mass that can be used on the floating slab. Further, allowable deflection of the rail prevents the use of very soft supports, which would give very low natural frequencies.

Therefore, designing the floating-slab system involves finding a favorable compromise between the rail deflection, produced by the static load of the trains and the floating-slab mass achievable in the space available in the subway structure. Round tunnels present the greatest problem because they provide relatively little space. Cut-and-cover structures usually provide enough space for good designs that incorporate appropriate mass ratios. To be effective as a vibration reduction element, the floating-slab mass must be sufficient to match the car truck and body mass. The floating-slab's natural frequency for lateral motion should be approximately the same as, or below, the natural frequency for vertical motion, but high enough to prevent interaction with the hunting motion of the transit car trucks. Another factor in designing a floating-slab trackbed system is that the supports must be stiff enough to support the rails without excessive vertical or lateral motion. A typical requirement is that the static load total rail deflections be less than 3 mm (1/8 in.) when transit cars pass over. This figure includes motion in the vertical and lateral directions. The limitation is based on considerations of rail stress and stability.

In evaluating the expected performance of a floating-slab system, the motion of the slab support system can be assumed to be uncoupled from the motion of the transit car body and from the motion of the subway structure. The low stiffness and vertical natural frequency of the car body's secondary suspension (1.0 to 2.0 Hz natural frequency) assures that the car body mass will be uncoupled from the floating slab. The large mass of the subway structure and the resilient supports for the floating slab assure that the motion of the slab is uncoupled from the subway structure. However, the floating-slab mass should not be more than 1/10 the effective mass of the structure.

In deriving appropriate mass ratios for design, the simplest and most straightforward assumption is that the entire truck mass is unsprung mass, and that it, therefore, is the driving mass of the floating-slab system. If this is assumed, deriving the necessary mass ratios for the floating slab results in a conservative design. In fact, the primary suspension system causes only a portion of the truck mass to function as an unsprung mass applying forces to the inertia base or floating-slab system.

If ratios of slab mass to car and truck mass are not high enough, multiple resonances or interaction effects may develop, either of which may compromise the effectiveness of the slab. If the ratio of the slab mass to the car mass is greater than 1 to 1 and the ratio of the slab mass to the total mass of the trucks is greater than 3 to 1 (assuming the truck mass is distributed over the car length), interaction effects should be avoided. In a conservative design, the ratio of slab mass to car mass should preferably be in the range 1.5 to 1, and the ratio of the slab mass to truck mass should be 5 to 1 or more. In order that interaction effects and multiple resonances between the subway structure and the floating slab be avoided, the mass of the subway structure should be about 10 or more times the mass of the slab; although the minimum ratio could be about 3 to 1 in very light tunnels (considering only the tunnel structure and not accounting for mass loading due to the surrounding soil). It is sometimes possible to design vibration isolation systems that do not meet all of these guidelines, but still provide adequate vibration isolation. However, the design should be very carefully evaluated, using a multi-degree-of-freedom model because of the good chance of unforeseen problems developing.

It is obviously desirable to use direct fixation type rail fasteners on floating slabs to minimize height. But it should also be noted that the rail fixation, though resilient, should not be resilient enough to allow interactive effects between the rail support and slab support systems. Direct fixation fasteners providing a rail support modulus of 2.5 to 10 Kn/cm per cm of rail (3,000 to 12,000 lb/in. per inch of rail) are appropriate.

One further design parameter of the floating-slab system is the damping coefficient. If the damping is too low, the amplification at resonance may be excessive. However, too high a damping ratio will reduce the attenuation of groundborne noise and vibration at frequencies above the resonance frequency. The desired damping ratio for floating-slab trackbeds is about 5% to 10% of critical, the minimum recommended being about 3% and the maximum around 20%

of critical. For the design discussed here, which uses only rubber pads with no auxiliary damping features, the expected damping factor for steady-state excitation is 3% to 5% of critical. This implies that a large amplification could occur at resonance. However, experimental data from continuous floating slabs at WMATA show only 2 to 3 dB of amplification for moving trains, indicating that the effective damping of the system is much higher than the 5% predicted by the mechanical characteristics of the support springs alone. Possibly, this is due to the moving loads and impact nature of the applied forces.

With a loaded resonance frequency of 14 Hz for uniform vertical harmonic motion and with maximum load static deflection of 3 mm, the weight of modern rail transit vehicles dictates a design consisting of concrete slabs 300 to 375 mm thick and 3.0 to 3.5 meters wide supported on 75 mm thick elastomer pads. The slabs must be completely isolated from the subway structure. This demand requires elastomer pads for the lateral and longitudinal supports in addition to the main load supports. The dynamic vertical stiffnesses of vertical and lateral support pads and any entrained air must all be included in calculating the resonance frequencies and in determining the elastomer pad characteristics.

Figure 8.18 indicates the cross section of the design for the continuous floating slabs designed in 1970 for the Washington, D.C. Metro. This design uses slabs 305 mm (12 in.) thick and 3.4 m (11 ft, 2 in.) wide, supported on resilient pads spaced 600 mm (24 in.) apart. The support pads are either 150 mm-square loadbearing fiberglass or 140 mm-diameter round polyurethane, all 75 mm thick. The side isolation is either a continuous pad of expanded neoprene or continuous strips of neoprene cemented in place. A detailed performance specification was prepared for purchasing the resilient pads to ensure long life and an appropriate stiffness range. The concrete is poured at the site using a waterproofed sheet metal form placed on top of the resilient support pads.

Figure 8.19 illustrates the plan and cross-section view of the discontinuous floating-slab design developed in 1974, for an extension of the Toronto rail transit system. The cross section is similar to the continuous floating slab illustrated in Figure 8.18. For box section structures, this design uses individual precast concrete blocks, about 300 mm thick and 3.2 m wide, each weighing 2500 kg and supported on four resilient pads. Side and end pads are provided for lateral and longitudinal restraint. In the Toronto system, the support pads are of natural rubber, 330 mm in diameter and 75 mm thick, spaced at 760 mm intervals longitudinally. The side pads are 300 mm x 150 mm x 50 mm, and the end pads are 300 mm x 150 mm x 75 mm, also of natural rubber. The side pads are preloaded by about 15 Kn via the slotted mounting angles and anchor bolts. Similar designs have been adapted for the Atlanta, Melbourne and Hong Kong transit facilities, with some variations in slab mass and shape and in support pad size to accommodate different loadings and other requirements. Support pads from 330 mm to 375 mm diameter have been used.

If either design is to be used in single-track round tunnels, a narrower slab is required. Because little space is available, the mass is usually less than it is in the box section, but can still be sufficient to give adequate results. The discontinuous slab does not entrain the air, an advantage that provides lower natural frequency than does the continuous slab design for equivalent static deflection under load.

8.3.2 Materials for Resilient Elements

The materials for floating-slab support pads, side pads, and end pads can be either high grade natural rubber, blended natural and synthetic rubber, polyurethane, or loadbearing fiberglass. For continuous floating slabs, the perimeter isolation is usually of closed-cell expanded neoprene material.

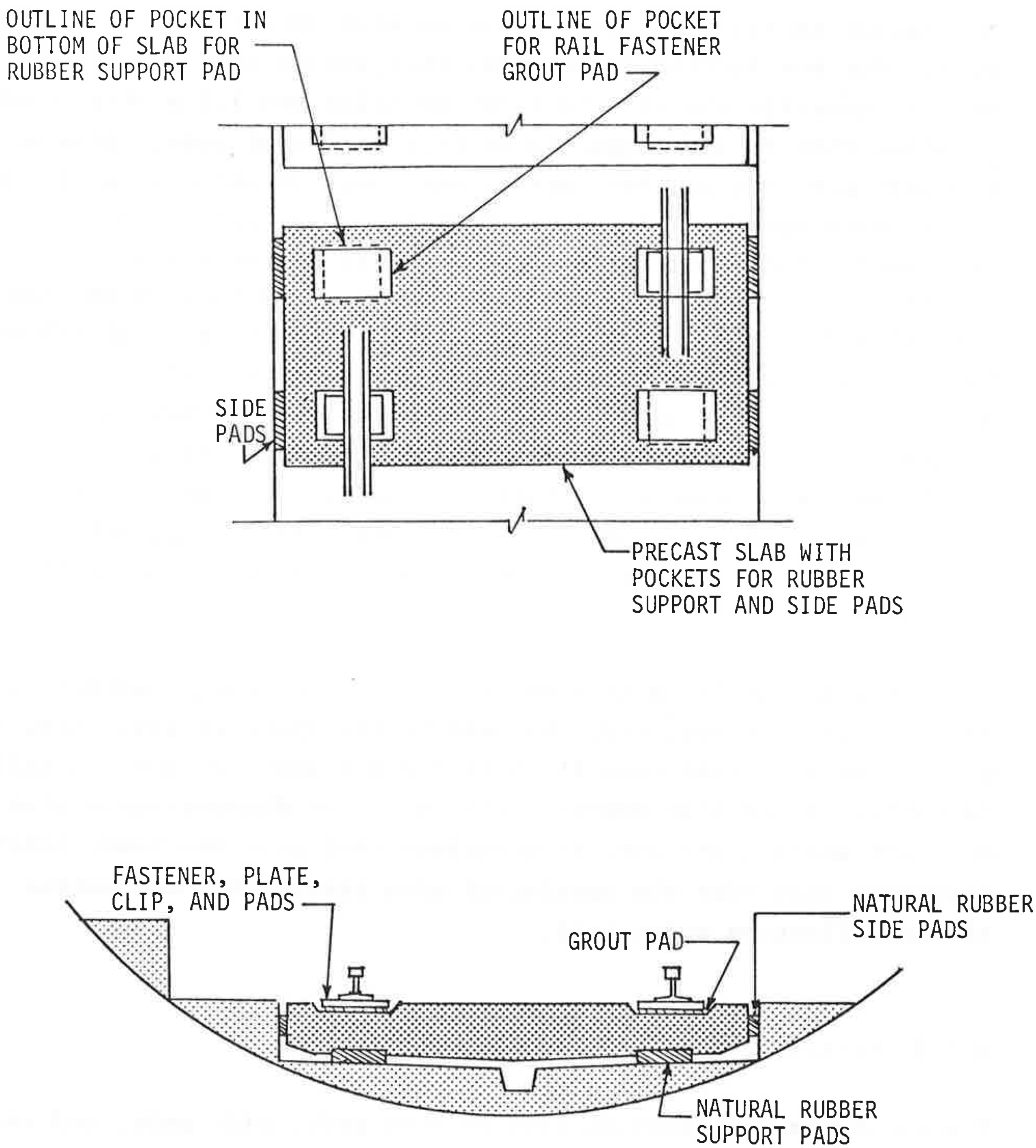


FIGURE 8.19 PLAN AND SECTION OF THE DISCONTINUOUS FLOATING SLAB

The elastomer material for the support pads should be compounded with suitable reinforcing agents, antioxidants, antiozonants, etc. This gives the pads adequate resistance to frequently repeated loadings and adequate impact, abrasion, weathering, and aging properties. The material should be vulcanized in a manner appropriate to provide stable dynamic properties, giving static and dynamic spring rates that are in accordance with the design values.

No ingredient should be used in the manufacture of the elastomers or as a mold release agent that will interfere with the cementing of the isolation materials to metals or concrete. Adhesives used for cementing elastomeric isolation materials to the concrete invert, or to the metal form, should be those recommended by the manufacturers of the elastomers. The adhesive should not affect the elastomers physically or chemically.

The dimensions, hardness, and numbers of supporting pads must be the first determination in the design of the floating-slab resilient supports. These parameters determine the static stiffness of the pads in compression. Typically, support pads are 300 mm (12 in.) to 375 mm (15 in.) in diameter and 75 mm (3 in.) thick. The hardness of the elastomer material is measured on the Shore A durometer by procedures given in ASTM D2240-1968. In general, elastomer hardness of 40 to 50 durometer is appropriate. The ratio of dynamic stiffness to the static stiffness should be in the range of 1.2 to 1.4, if satisfactory results are to be obtained. With proper formulation of the support pads, a life expectancy of 50 years can be achieved for the support system.

8.3.3 Floating-Slab Performance

The vibration isolation effectiveness of several floating slabs has been evaluated at several rapid transit systems. These include the continuous floating slabs used at the Washington Metro subway (WMATA) structures and the discontinuous or double-tie floating slab used at TTC and MARTA. Figure 8.20 shows the overall configuration of the arrangement for testing of continuous floating slabs at the WMATA system (Ref. 8.20). The rail was 11 to 12 m below the surface at a section of subway with uniform conditions, a section of standard rigid lnvert trackbed, and a section of floating slab. The tests were performed in October 1975, and the results are given by Figures 8.21 and 8.22. A 2-car train was operated at constant speeds of 32, 48, and 64 km/hr through the test area. The vibration was measured on the subway structure at the side curbs and center bench and on the ground surface using a sidewalk or parking lot asphalt surface for mounting the accelerometer. Several accelerometer locations were used, and the results averaged.

Figures 8.21 and 8.22 indicate typical results. There is a small amplification of vibration levels near the floating-slab fundamental resonance and substantial reduction in frequencies above 31.5 Hz. For the surface measurements, background noise and vibration from surface traffic prevented obtaining accurate data for the floating slab at frequencies above 100 Hz.

In October 1978, vibrations from subway train passbys were measured at the MARTA system, which has discontinuous floating slabs. The rail was 7 to 8 m below the surface, in a section of double-box subway structure in which both standard rigid trackbed and a long floating-slab trackbed are installed. A 2-car train was operated at a constant speed of 56 km/hr through the test, while vibration was measured on the subway structure and on the ground surface. Figures 8.23 and 8.24 indicate the measured vibration levels and show the difference between levels with and without the floating slab. As is evident, there is slight amplification at the resonance of the floating slab, and substantial vibration reduction, at frequencies above 31.5 Hz.

Figure 8.25 compares the insertion loss performance of WMATA continuous and MARTA discontinuous floating slabs plotted as response functions. The amplification at resonance is only 3 dB

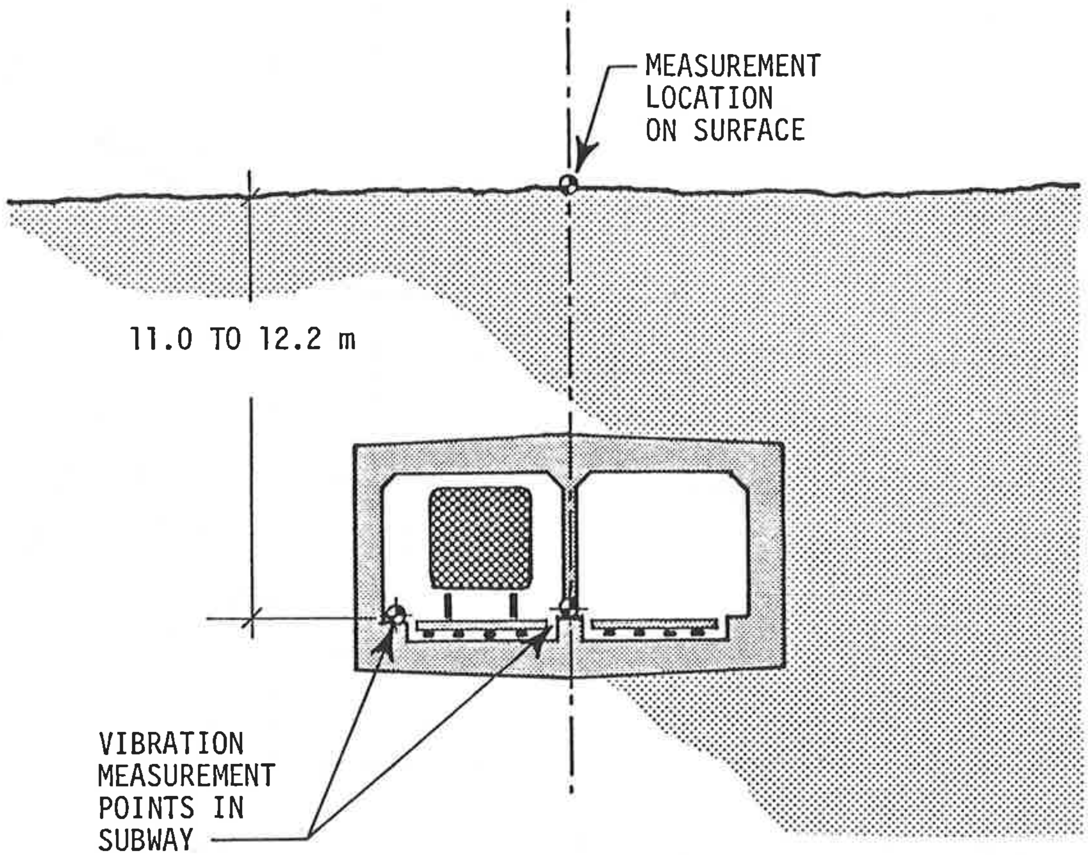


FIGURE 8.20 OVERALL CONFIGURATION FOR THE WMATA FLOATING-SLAB TRACKBED TESTS

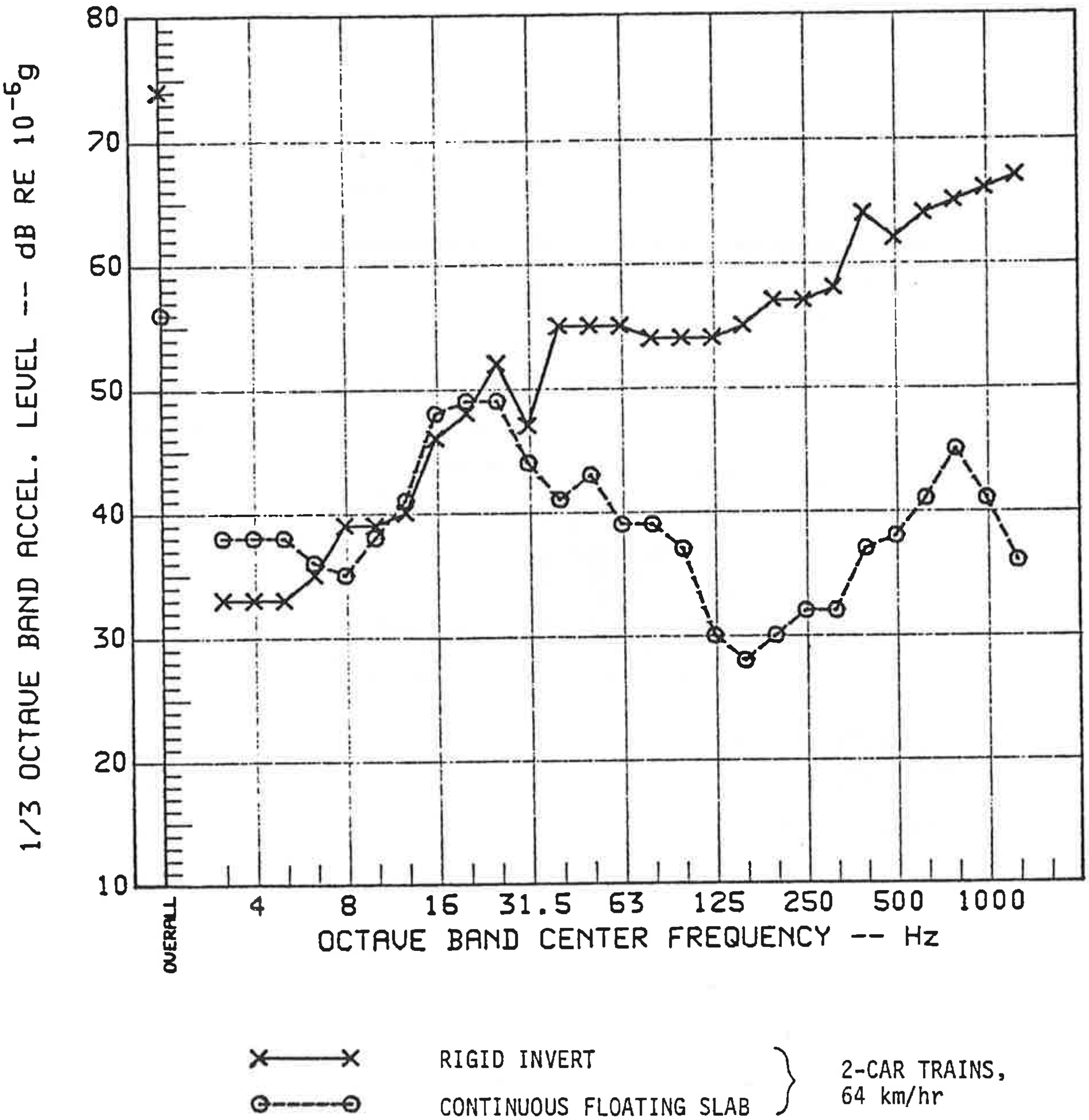


FIGURE 8.21 SUBWAY STRUCTURE (SAFETY WALK) VIBRATION, WMATA METRO DOUBLE-BOX SUBWAY

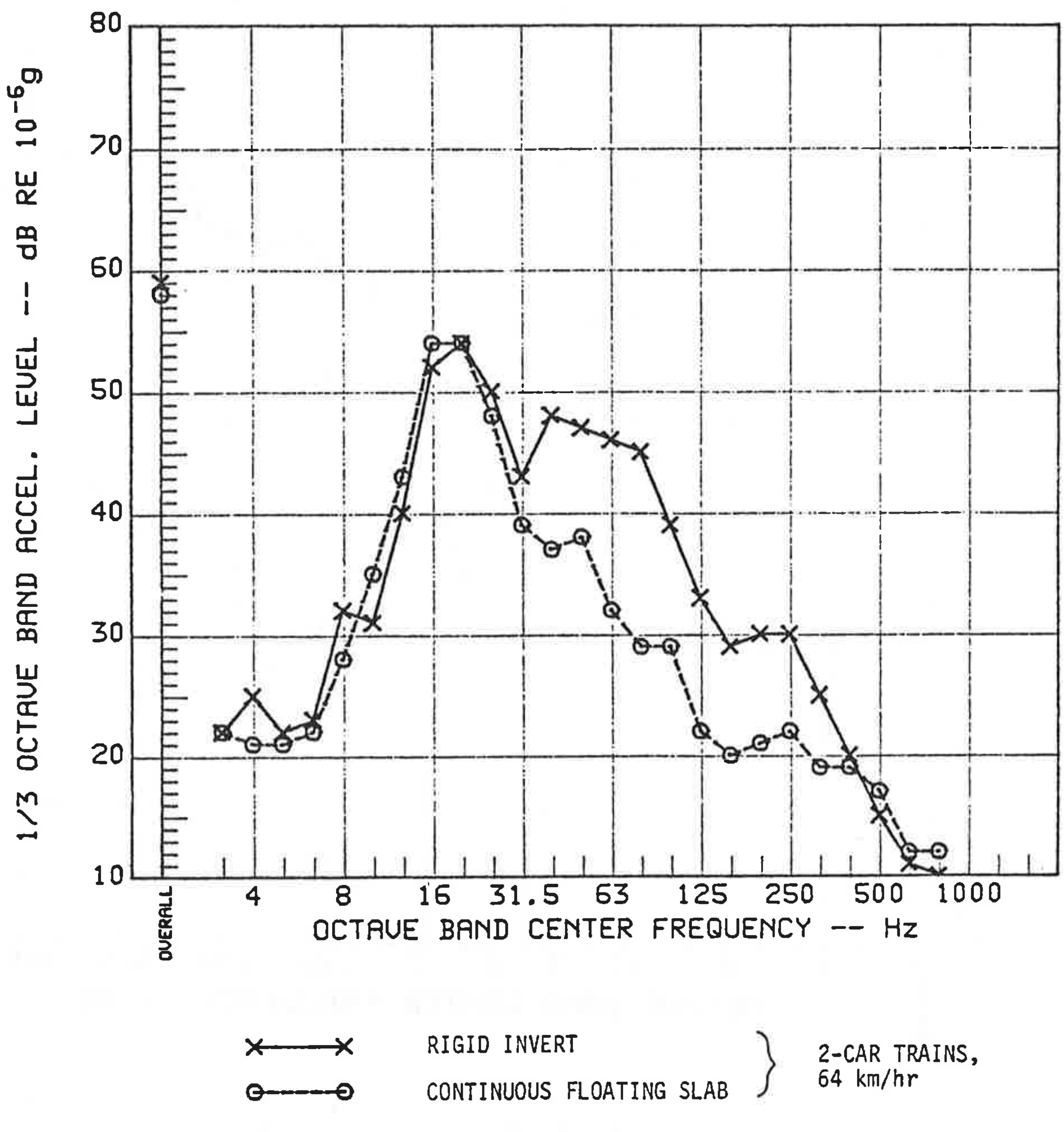


FIGURE 8.22 GROUND SURFACE VIBRATION ABOVE WMATA METRO DOUBLE-BOX SUBWAY (TOP-OF-RAIL 11 to 12 m BELOW SURFACE)

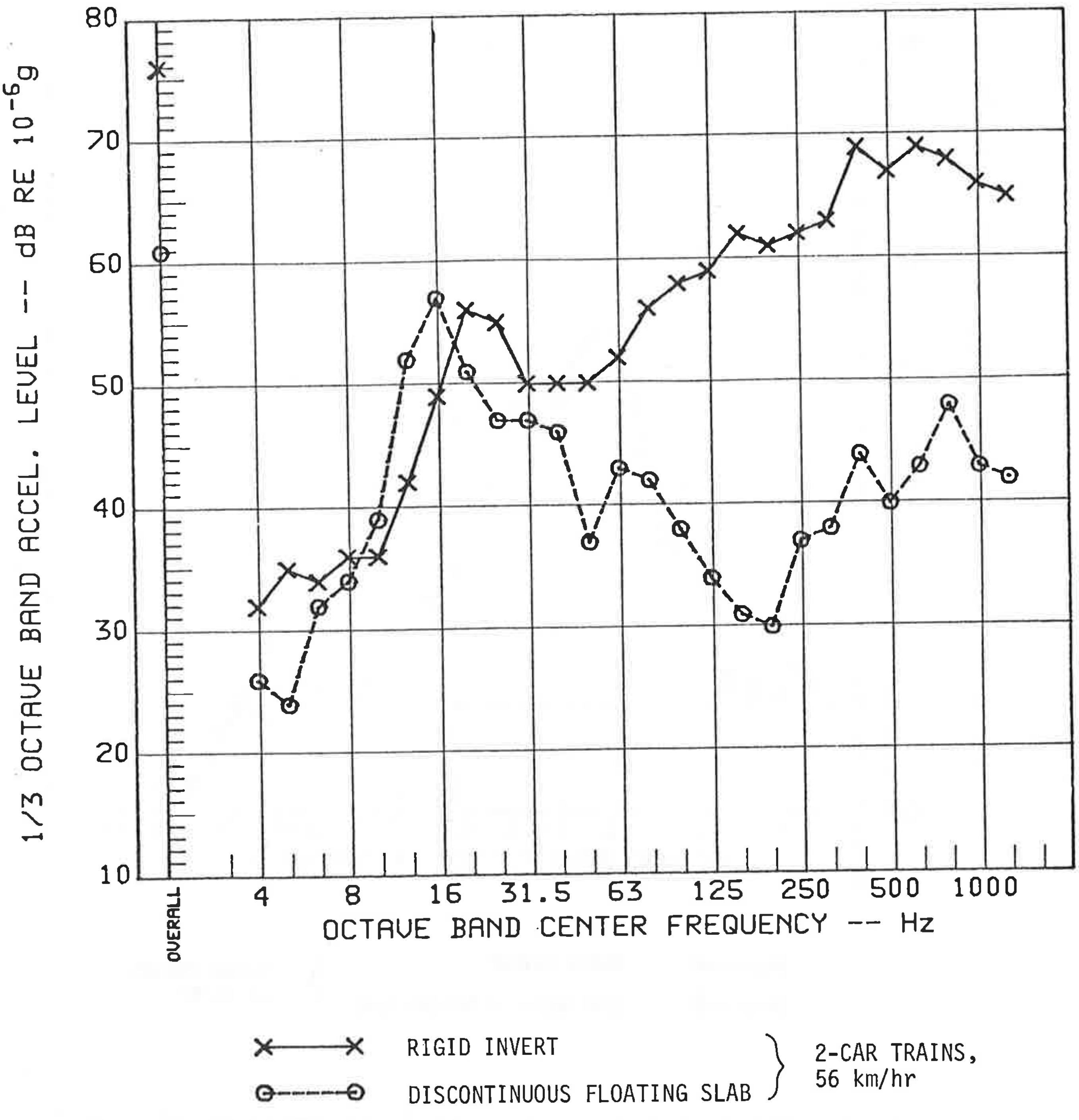


FIGURE 8.23 SUBWAY STRUCTURE (SAFETY WALK) VIBRATION, MARTA DOUBLE-BOX SUBWAY

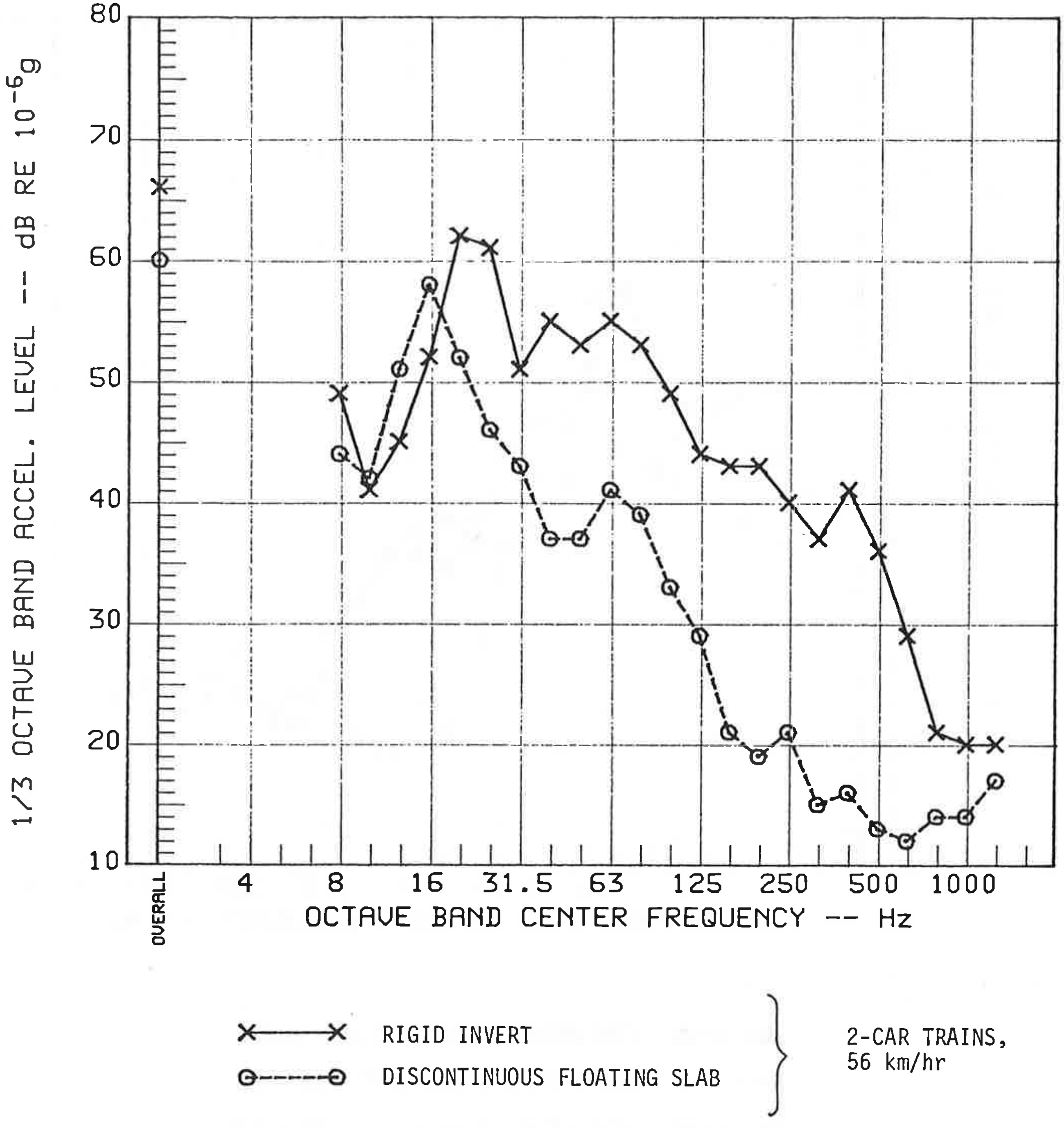


FIGURE 8.24 GROUND SURFACE VIBRATION LEVELS ABOVE MARTA DOUBLE-BOX SUBWAYS WITH AND WITHOUT FLOATING SLABS

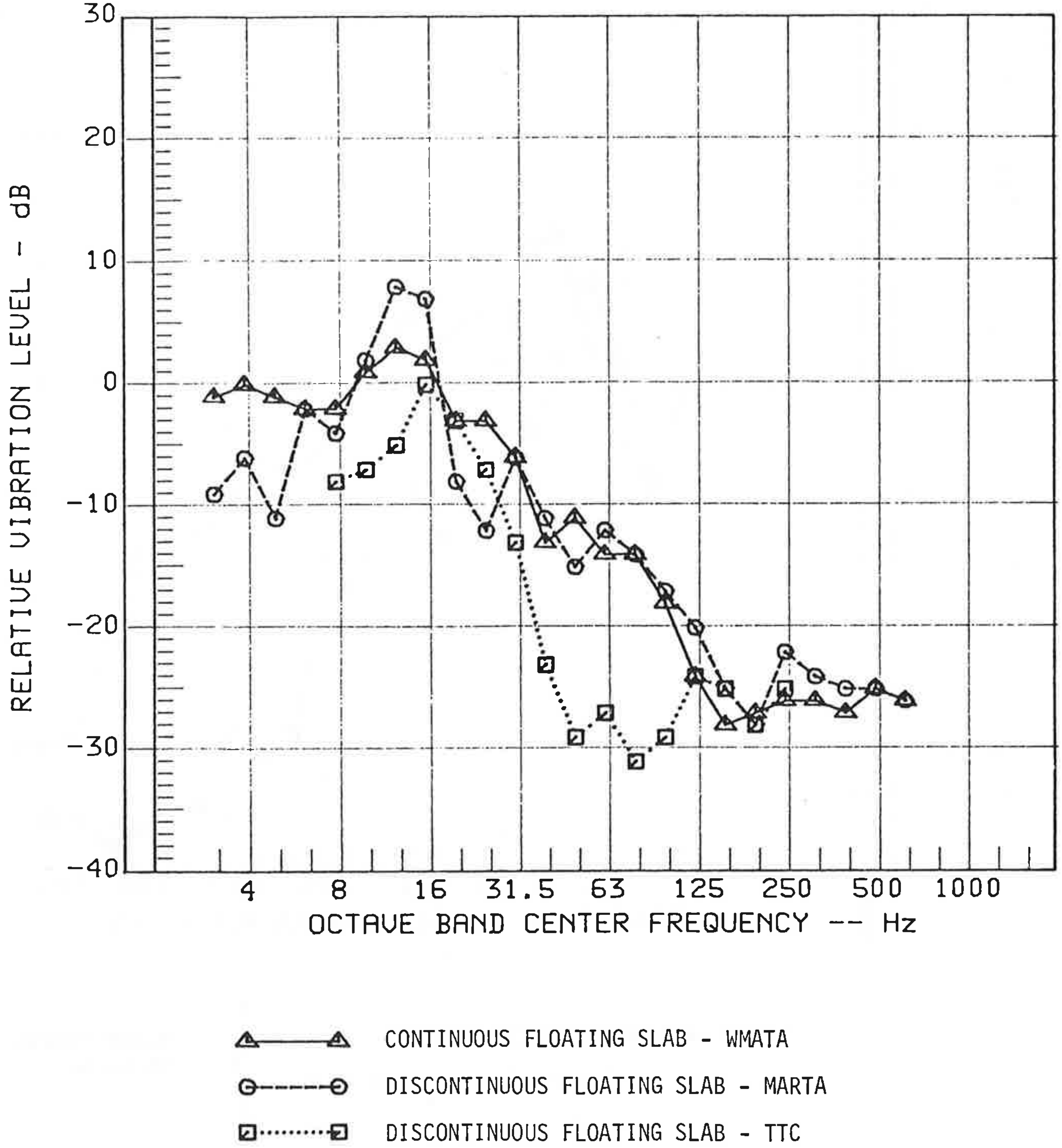


FIGURE 8.25 VIBRATION LEVELS WITH TWO TYPES OF FLOATING SLAB RELATIVE TO LEVELS WITH RIGID INVERT

with the continuous floating slab, but 8 dB with the MARTA discontinuous floating slab. Noted that the resulted above 31.5 Hz are essentially the same. Although there is a significant difference between the two curves in Figure 8.25 at the resonance frequency, measurements at the TTC system in Toronto indicate little or no amplification at the design resonance for the discontinuous double-tie system. In addition, the vibration isolation of the TTC system at frequencies above 31.5 Hz is better than that of either the WMATA and MARTA floating slabs. These variations may be caused by any number of factors, but differences between rolling stock on the MARTA and TTC systems are the most likely causes. These results suggest that both types of floating slab give similar insertion loss performance.

8.3.4 Noise Radiation of Floating Slabs

Because the lightweight continuous floating slab is an undamped concrete plate, bending waves can radiate noise into the subway tunnels. This noise can affect patrons riding in cars and consequently, has been the subject of much discussion. During the tests on the WMATA Metro system, noise outside the train was measured. In Figure 8.26, reverberant noise levels around a train running on continuous floating-slab track are compared with those of trains running on standard rigid invert. The continuous floating slab does cause an increase in noise level, primarily at low frequencies, outside the car, but the change ln car interior noise level is barely measureable and not noticeable to patrons.

8.4 RAIL FASTENER DESIGN

One of the most effective and practical developments for reducing the vibration from subway installations is the use of resilient rail fasteners. The most widely used resilient fastener in the U.S. is a relatively simple and economical design, consisting mainly of some form of tie plate plus an elastomer pad between the

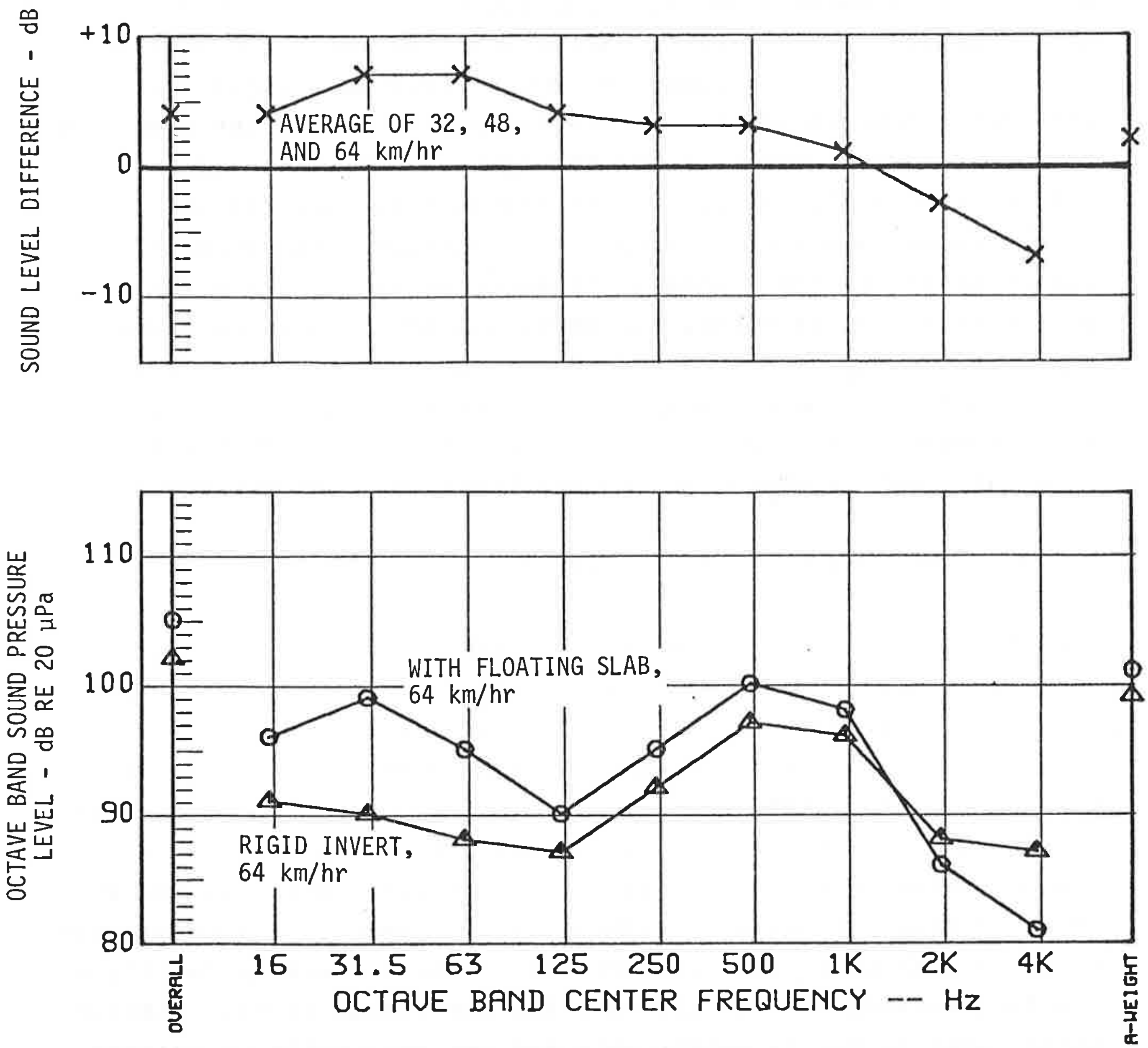


FIGURE 8.26 AVERAGE WAYSIDE NOISE LEVELS IN SUBWAY, 2-CAR WMATA METRO TRAINS

rail and invert. However, this type of fastener cannot provide the necessary low-frequency vibration reduction at critical locations.

The use of resilient rail fasteners, along with other design features such as continuous welded, smooth ground rail and smooth ground wheel surfaces, has produced considerable reduction of vibration levels. This is compared to those in standard systems using wood ties cast in the concrete invert or ties and ballast in tunnels. The use of direct fixation resilient rail fasteners with concrete invert also reduces height requirements in subways, weight requirements of aerial structures, and maintenance costs, and, in addition, provides electrical isolation of the rail.

Use of the same rail fastener for all applications (i.e., for aerial structures, at-grade concrete trackbed, and subways) is desirable from the point of view of production and installation. Under these circumstances, the fastener design must be a compromise and will not give the optimum vibration isolation in all types of structures. With direct fixation fasteners, the requirements for minimum airborne noise from above-grade operations conflict somewhat with the requirements for minimum ground vibration. Results of the test programs at the BART Test Track and experience in Toronto and Paris indicate that an excessively soft direct fixation fastener adds to the undercar and wayside airborne noise. A fastener stiff enough to avoid the added noise does not reduce the groundborne vibration as much as the softest fasteners permissible within rail support stability requirements. Theoretical models of an elastically supported rail do not support measurement results (cited above) of increased noise radiation with the use of softer fasteners. Additional work in this area is necessary.

8.4.1 Direct Fixation Fasteners

Figure 8.27 indicates some of the direct fixation fasteners that have been tested. The figure is adapted from the final report on the acoustic studies at the BART Test Track (Ref. 8.12) and indicates the fasteners tested during the BART Test Track program.

Figure 8.28 and 8.29 illustrate the resilient direct fixation fasteners used at TTC and BART. Of all the various types of fasteners which have been developed in the U. S. and other countries, these two represent the most commonly used configurations. The Toronto fastener consists of a standard rail tie plate with compression rail clips; a 12.5 mm thick, cored rubber pad is placed between the tie plate and the invert grout pad. This is an unbonded fastener with anchor bolts passing through the elastomer pad, so that the elastomer pad is somewhat precompressed by the anchor bolt tension. Springs are used between the anchor bolt nuts and the rail tie plate hold-down washers to prevent anchor bolt fatigue from occurring as the rail fastener deflects in normal use. This type of fastener is also in use at the PATCO system.

The BART fastener, manufactured by the Landis Sales Company, is a bonded elastomer pad fastener, with flat steel plates on both sides of the elastomer pad. It consists of a 12.5 mm thick, steel top plate with rigid rail clips, a 19 mm thick, bonded elastomer pad, and a 6 mm steel bottom plate. The assembly is anchored to the invert using sleeves to prevent contact between the anchor bolt and the upper steel plate. In the bonded BART fasteners (i.e., with top and bottom steel plates bonded to the rubber pad), there is no precompression of the elastomer element.

Two direct fixation fastener designs developed by SNCF (French National Railway) and RATP (Regime Autonome de Transports Parisiens) in Paris are illustrated in Figure 8.30. The RATP fastener consists of a single elastomer pad about 20 mm thick, grooved on both sides. It is located between the rail base and a steel plate, which is fixed to the concrete invert with anchor

TEST SECTION	TYPE ROADBED	TYPE FASTENER	TYPE RAILPAD	REFERENCE DRAWING[c]
OGAS.[b]	CONCRETE	IA (DOUBLE BOLT AND CLIP)	$65^{\pm 5}$ DUROMETER, NEOPRENE 8" X 14" X 5/8" THICK	CT-46
1	CONCRETE	I (DOUBLE BOLT AND CLIP)	$45^{\pm 5}$ DUROMETER (OLIVER SINGLE RATE, TYPE 1)	CT-45
2	CONCRETE	I (DOUBLE BOLT AND CLIP)	$45^{\pm 5}$ DUROMETER (OLIVER DOUBLE RATE, TYPE 2)	CT-45
3	CONCRETE	XI A (DOUBLE BOLT AND VICTORY CLIP)	$70^{\pm 5}$ DUROMETER, BUTYL RUBBER 6" X 8" X 3/4" THICK RAIL SEAT PLUS 1/4" THICK PAD UNDER PLATE – LIBERTY 1	CT-54
3a	CONCRETE	XI A (DOUBLE BOLT AND VICTORY CLIP)	3/4" GOODRICH CONTINUOUS ELASTOMER STRIP ON UNDER-SIDE OF RAIL, "DEADBEAT" RIBS ON BOTTOM	CT-54
4	CONCRETE	III (MOLDED RAIL SEAT- 4 BOLT CONFIGURATION)	$70^{\pm 5}$ DUROMETER, BUTYL RUBBER 8 5/8" X 12" X 7/8" THICK RAIL SEAT	CT-50
5	WOOD TIE & BALLAST	IIA (SPIKED RAIL AND PLATE)	$65^{\pm 5}$ DUROMETER, NEOPRENE 8" X 11" X 1/4" THICK	CT-47

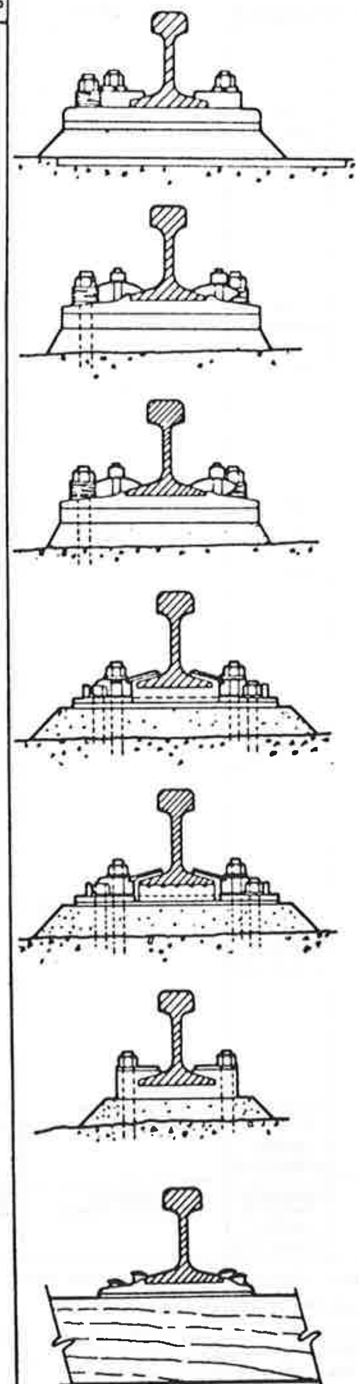
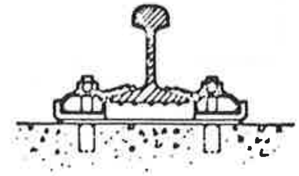
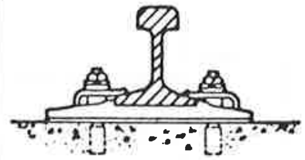
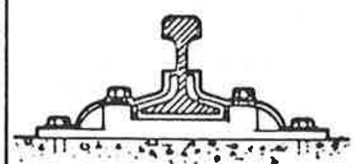
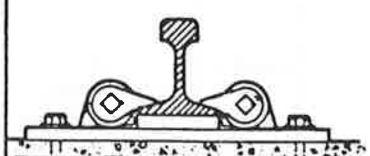


FIGURE 8.27 DIRECT FIXATION RAIL FASTENERS TESTED AT THE BART TEST TRACK

TEST SECTION	TYPE ROADBED	TYPE FASTENER	TYPE RAILPAD	REFERENCE DRAWING[a]	
6	GERWICK CONCRETE TIE AND BALLAST	VI (SINGLE BOLTED CLIP)	$70^{\pm5}$ DUROMETER, BUTYL RUBBER 6"X8"X5/8" THICK RAIL SEAT PLUS 1/4" THICK PAD UNDER RAIL PLATE	CT-53	
7	ARR CONCRETE TIE AND BALLAST	VII (SINGLE ARR BOLTED CLIP)	$65^{\pm5}$ DUROMETER, SOLID NEOPRENE 5"X14"X1/4" THICK	CT-56	
8	SWEDISH CONCRETE TIE[c] AND BALLAST	V (SWEDISH FIST)	$70^{\pm5}$ SHORE DIN 53503, RUBBER TYPE UNKNOWN 6"X6"X1/4" THICK RAIL SEAT, WITH GROOVES	CT-52	
10	CONCRETE PHASE II AERIAL STRUCTURE	PANDROL LOCKSPIKE LTD	$71^{\pm5}$ DEGREES B.S. & I.R.H TEST B.S.903 PART A7:1957 3/16" THICK ELASTOMER BONDED CORK PAD		
11	CONCRETE PHASE II AERIAL STRUCTURE	GENERAL TIRE & RUBBER (PROTOTYPE)	$60^{\pm5}$ DUROMETER REINFORCED ELASTOMER 3/8" THICK		
12	CONCRETE PHASE II AERIAL STRUCTURE	B.F. GOODRICH (PROTOTYPE)	$70^{\pm5}$ DUROMETER, EPT ELASTOMER 5"X5-1/2"X1/2" THICK		
—	CONCRETE PHASE II AERIAL STRUCTURE	STANDARD (TORONTO TYPE)	$65^{\pm5}$ DUROMETER, SOLID NEOPRENE 8"X14"X5/8" THICK	CT-45	SAME AS TYPE I INSTALLED IN TEST SECTION I, EXCEPT FOR RAIL PAD

a. BART CONTRACT DRAWINGS IC4161, C416

b. OAK GROVE AERIAL STRUCTURE

c. SHORT CONCRETE TIE SUPPORTS EACH RAIL. OPPOSITE TIES ARE JOINED WITH STEEL TRANSVERSE SPACING BAR.

FIGURE 8.27 (CONT.) DIRECT FIXATION RAIL FASTENERS TESTED AT THE BART TEST TRACK

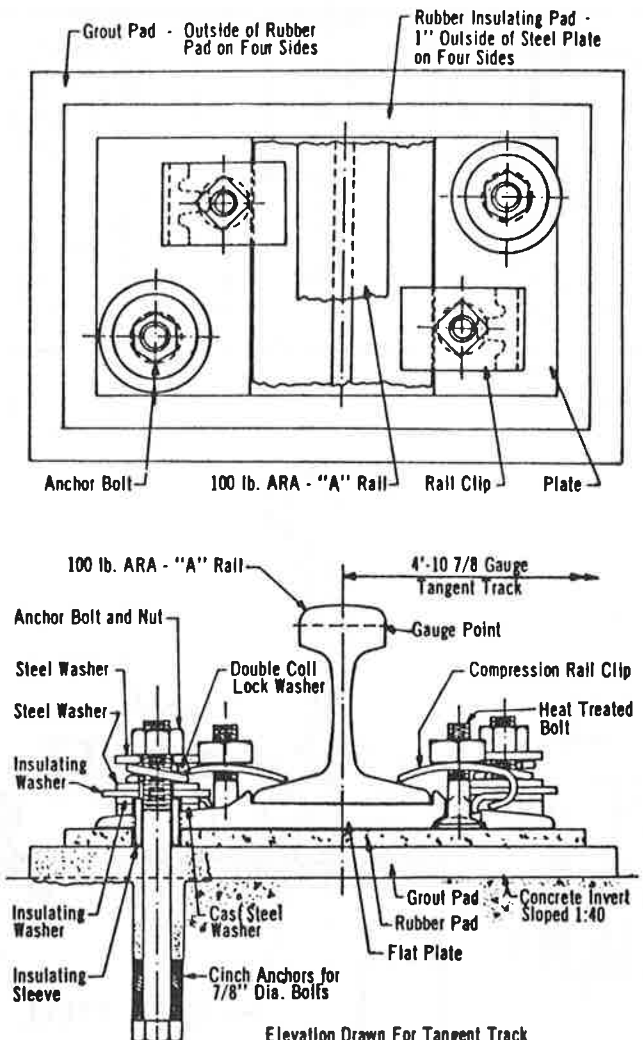


FIGURE 8.28 TORONTO TRANSIT COMMISSION DIRECT FIXATION RESILIENT RAIL FASTENER WITH UNBONDED NEOPRENE PAD

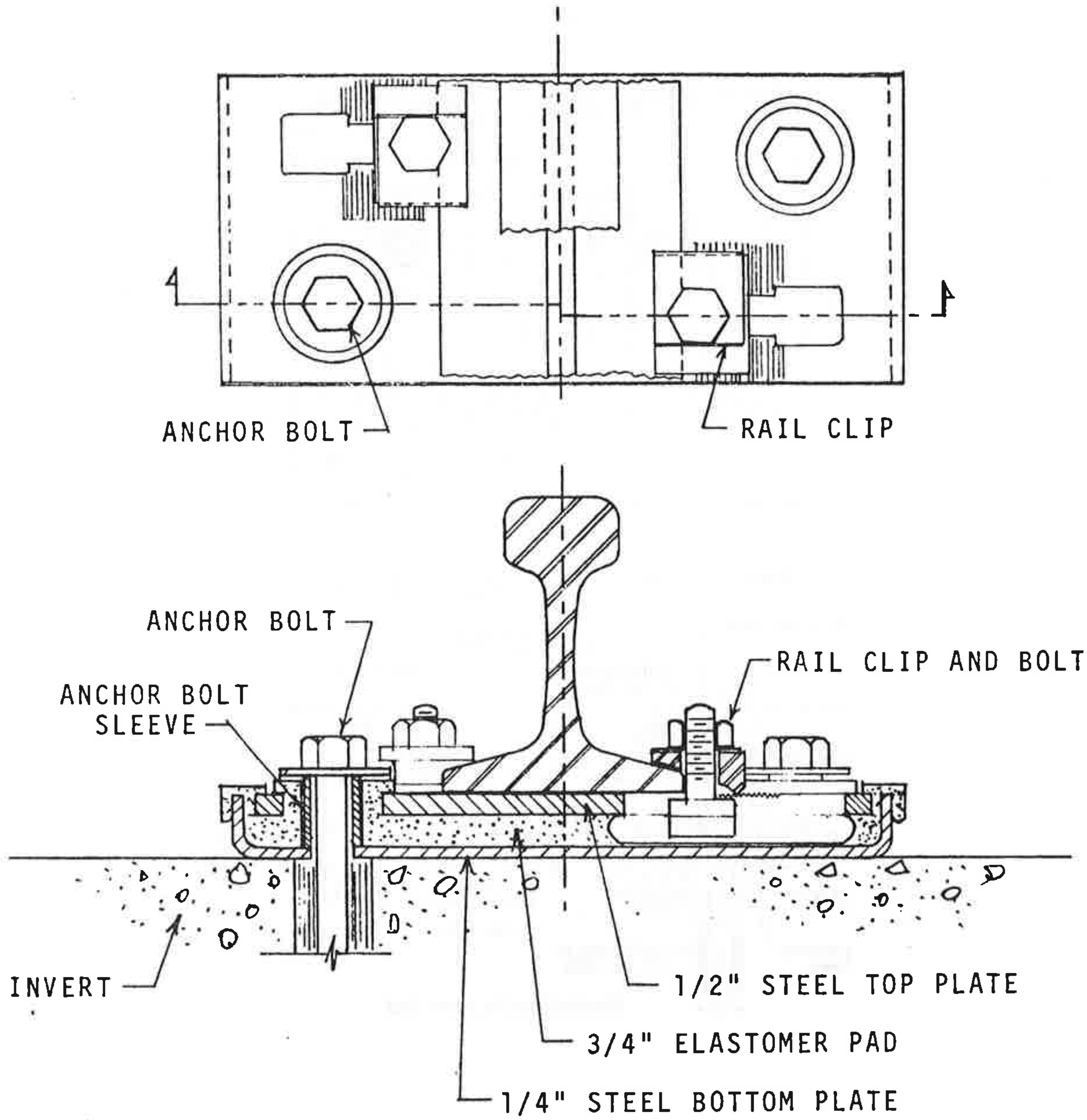


FIGURE 8.29 BART DIRECT FIXATION RESILIENT RAIL FASTENER WITH BONDED ELASTOMER PAD

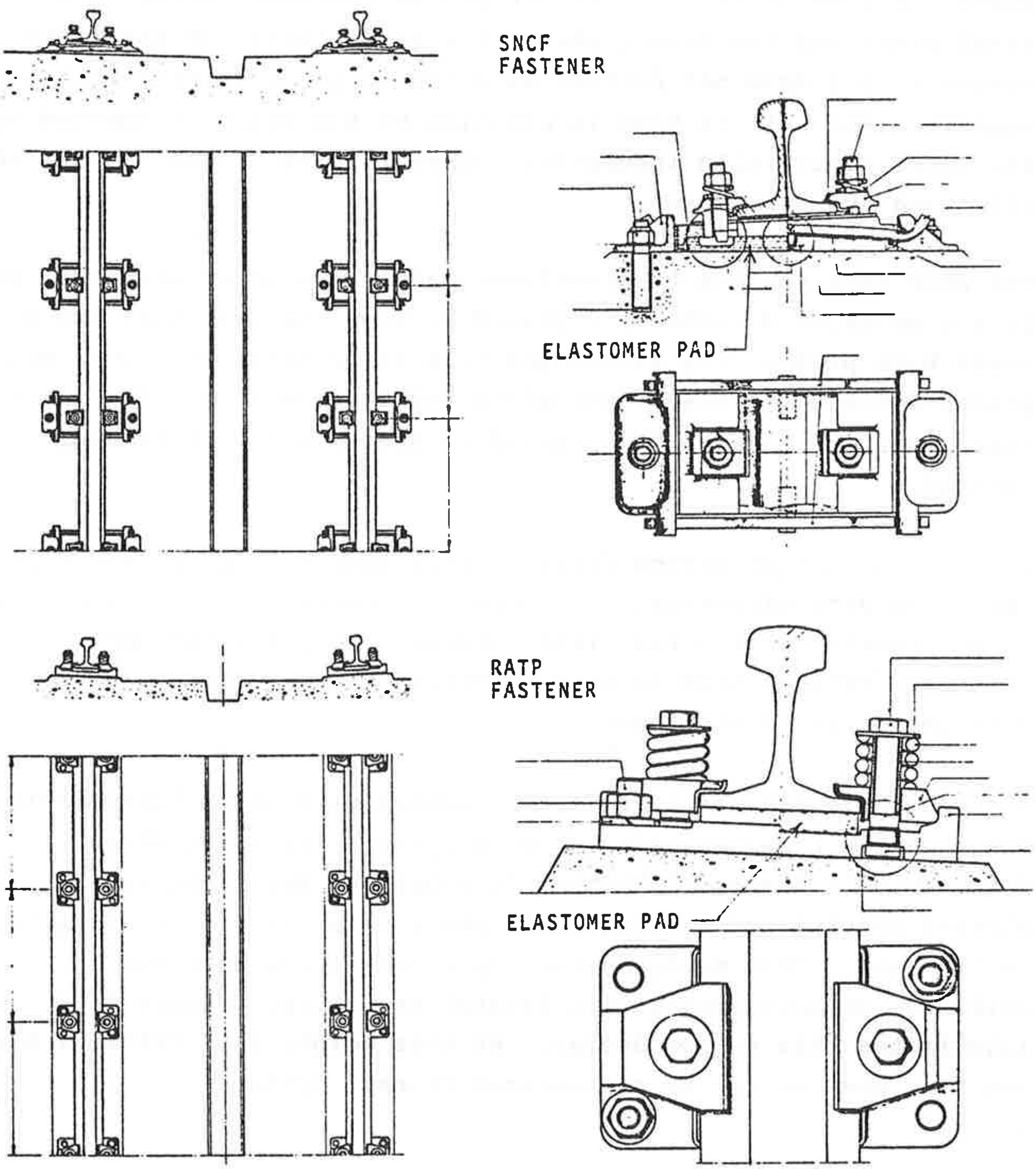


FIGURE 8.30 DRAWINGS OF THE SNCF AND RATP RESILIENT DIRECT FIXATION FASTENER DESIGNS

bolts. A 3 mm thick solid rubber pad is included between the steel plate and the invert plate for good bearing of the plate on concrete, but does not perform as a second stage of resilient support. The rail is kept in position by two clips, supported on the steel plate, with insulating rubber inserts placed between the clips and the rail bases.

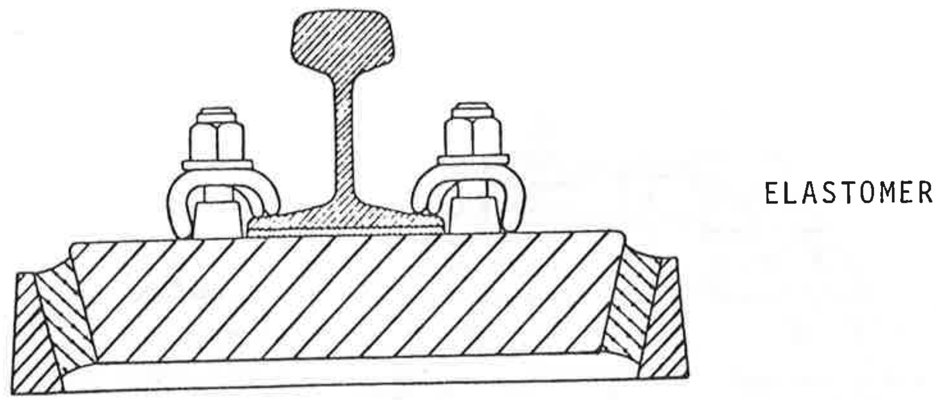
The SNCF fastener has two resilient pads. The upper elastomer pad is a 5 mm grooved rubber pad placed between the rail base and a steel base plate. The second pad is a 22 mm thick elastomer pad, placed between the steel base plate and the concrete. The steel base plate is elastically anchored to the invert with lateral springs.

Other varieties of direct fixation rail fasteners have been tried, including very elaborate, multilayer fasteners using an elastomer in compression as the resilient element. However, the most elaborate designs seem no more effective than a simple fastener with one layer of elastomer.

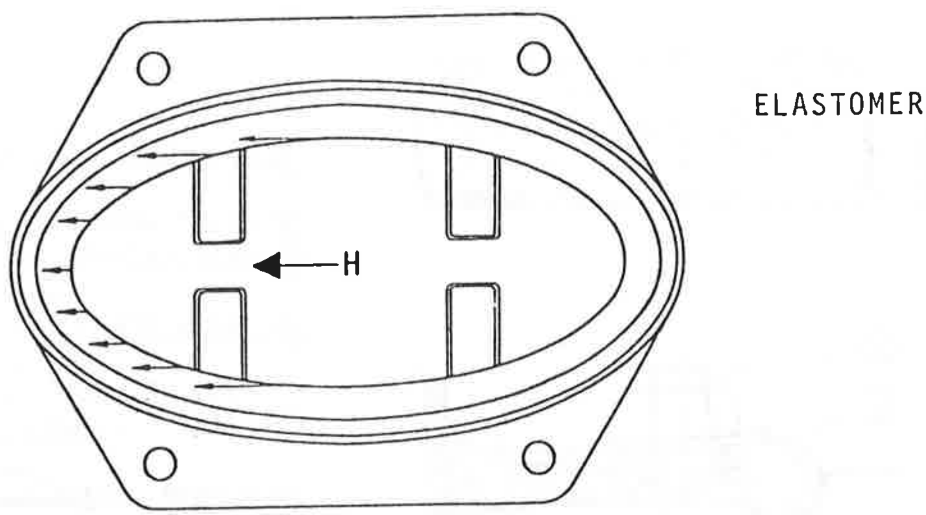
A recent European fastener design successfully uses elastomer-in-shear as the resilient element of a bonded, resilient direct fixation rail fastener. Dubbed "The Cologne Egg," the resilient element consists of an oval ring, whose major axis is transverse to the rail. This design evidently achieves low vertical stiffness without sacrificing lateral stability. Figure 8.31 illustrates this unique design. At this point, this fastener has not been used on any North American transit system.

8.4.2 Resiliently Supported Ties

The resiliently supported tie concept is illustrated in Figure 8.32. The tie is supported by elastomer boots, soft enough to provide vibration isolation. Both concrete and plastic ties have been used with this system. This resiliently supported tie system adds another significant element to the vibration isolation



CROSS-SECTION



PLAN VIEW

FIGURE 8.31 CLOUTH 1403/c ("COLOGNE EGG") RAIL FASTENER

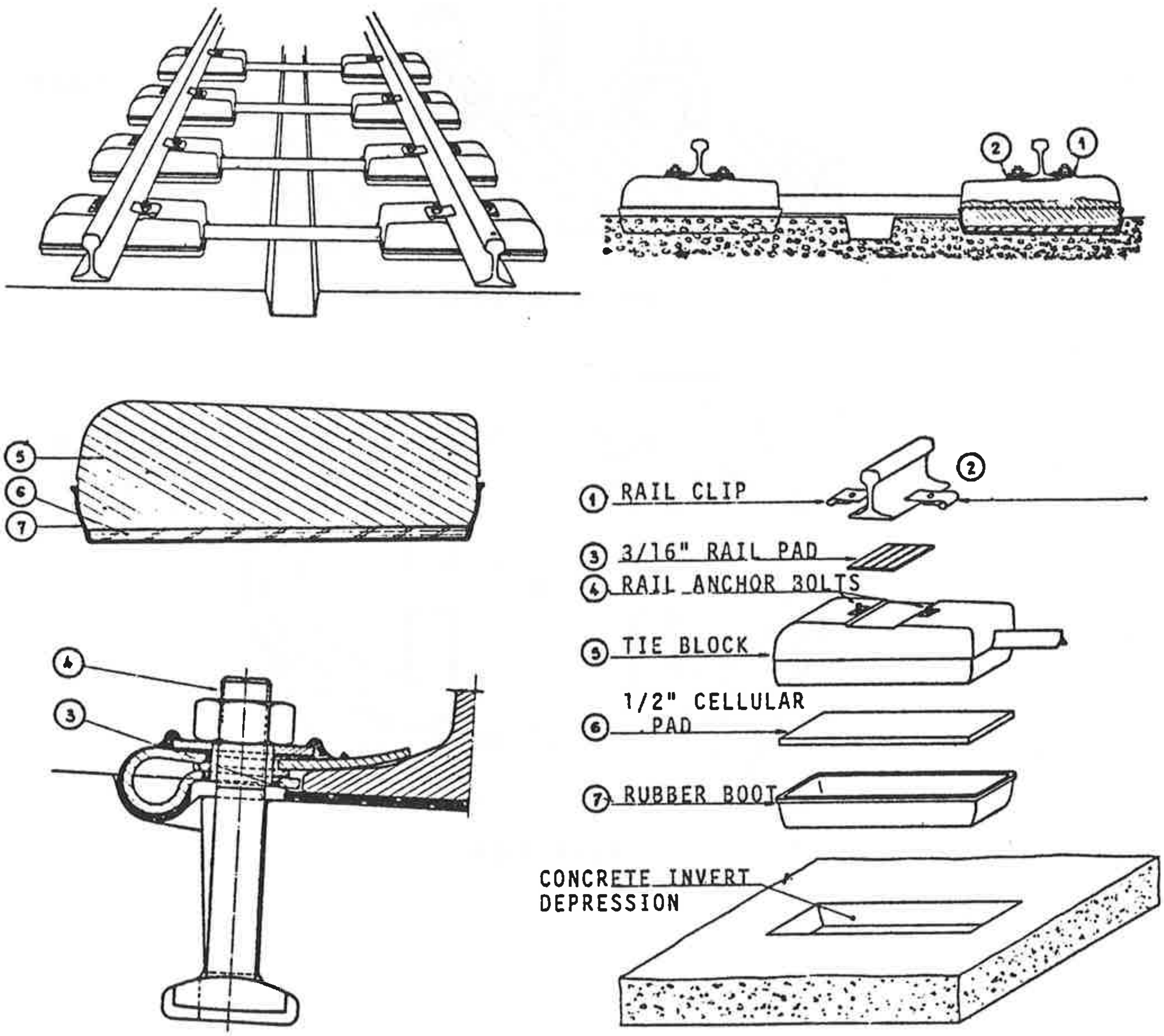


FIGURE 8.32 RS-STEDEF RESILIENTLY SUPPORTED RAIL TIE SYSTEM COMPONENTS

system: the mass of the concrete tie between the rail and the elastomer pad. This design may reduce ground vibration levels more effectively than a direct fixation fastener because of the mass reactance of the concrete tie block and the fact that a softer elastomer pad can be used than is possible with a direct fixation fastener.

One variety that has been shown to be effective, and on which technical information is available, is the STEDEF ballastless track design, which uses resiliently supported -- RS type -- double-block concrete ties. The RS-STEDEF design has been shown to significantly improve groundborne vibration over direct fixation rail fasteners and standard tie and ballast installations in subway tunnels. The SEMPERIT Rubber Company in Vienna, Austria has developed a similar system, using one piece PVC ties, that should give similar performance.

8.4.3 Fastener Performance

The fasteners shown in Figure 8.27, including five types with 4 to 6 mm thick elastomer pads and eight types with 13 to 25 mm thick pads, were tested at the BART Test Track (Ref 8.12). Little difference in noise and vibration performance was found for the widely different configurations. This was primarily because the fasteners were improperly tested. All of the tests were done with a car equipped with SAB high deflection resilient wheels. The resilience of the wheels was so great that differences in fastener stiffness or performance were obscured and, in effect, the test results gave no valid evaluation of the comparative performance of the different fasteners.

The RATP in Paris has used the construction of a new section of the RER subway as an opportunity to install and test several varieties of track fastening systems (Ref. 8.9). The test installations included two direct fixation type fasteners on concrete invert and the RS-STEDEF resiliently supported tie system. The three types of fastening systems were installed at adjacent sections of concrete invert subway. The standard track construction in this two-track subway tunnel was timber tie on ballast, with about 50 cm total depth of ballast. Tests were conducted to evaluate the differences in groundborne vibration, noise in the cars, and noise in the tunnels for the three experimental and the standard track fastening systems under controlled conditions. The rail was continuous welded, with glued insulating joints, and was ground smooth prior to the tests.

Figure 8.33 indicates the vibration levels observed in the drainage tunnel for 80 kph (50 mph) test runs. The vibration levels with the RS-STEDEF system were lower than the other fastening systems, particularly in the 63 Hz octave band.

The fasteners tested by the RATP are similar to those used by the TTC, BART, and WMATA Metro systems. These results indicate the probable performance characteristics of the TTC, BART, and WMATA fasteners, relative to ballast-and-tie track or to the RS-STEDEF resiliently supported tie system.

As a part of the RER test program, a series of noise measurements were also performed, including measurements inside the car and in the tunnel near the car. Figure 8.34 presents these results. It is apparent from the chart that lack of sound absorption by ballast, and to a lesser extent the softer direct fixation fastener, results in considerably greater noise levels than in the standard ballast-and-tie track -- particularly in the middle frequency range (between 500 and 1000 Hz octaves) where the wheels and rails are efficient noise radiators. The added mass and rail anchor stiffness of the RS-STEDEF fastening system reduces the rail vibration, so that even for the concrete trackbed without ballast for sound absorption, the noise level is only about 5 dBA greater than for the standard ballast-and-tie.

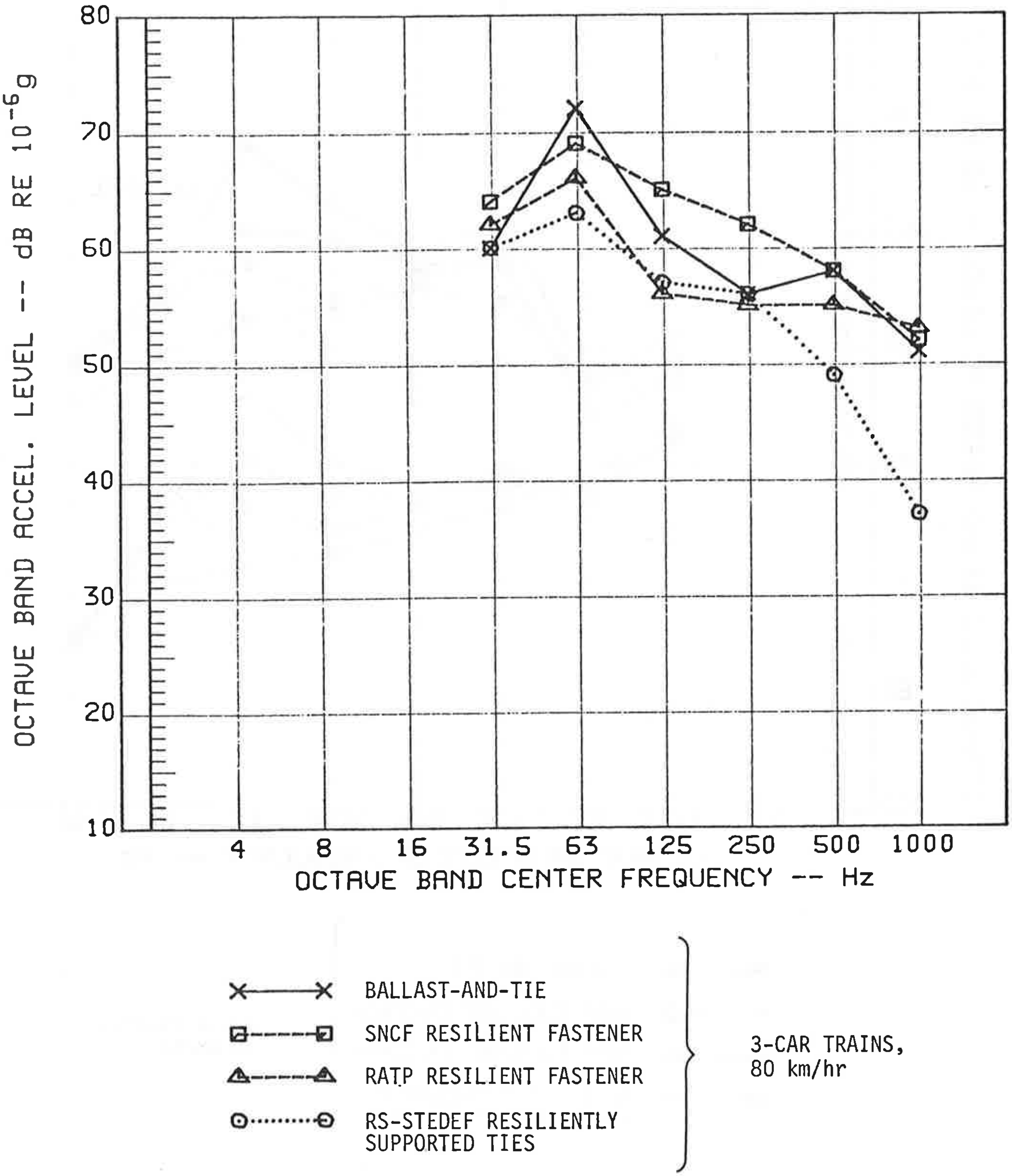


FIGURE 8.33 VIBRATION LEVEL IN DRAINAGE UNDER VARIOUS TYPES OF RER SUBWAYS. MEASUREMENTS BY RATP (REF. 8.9)

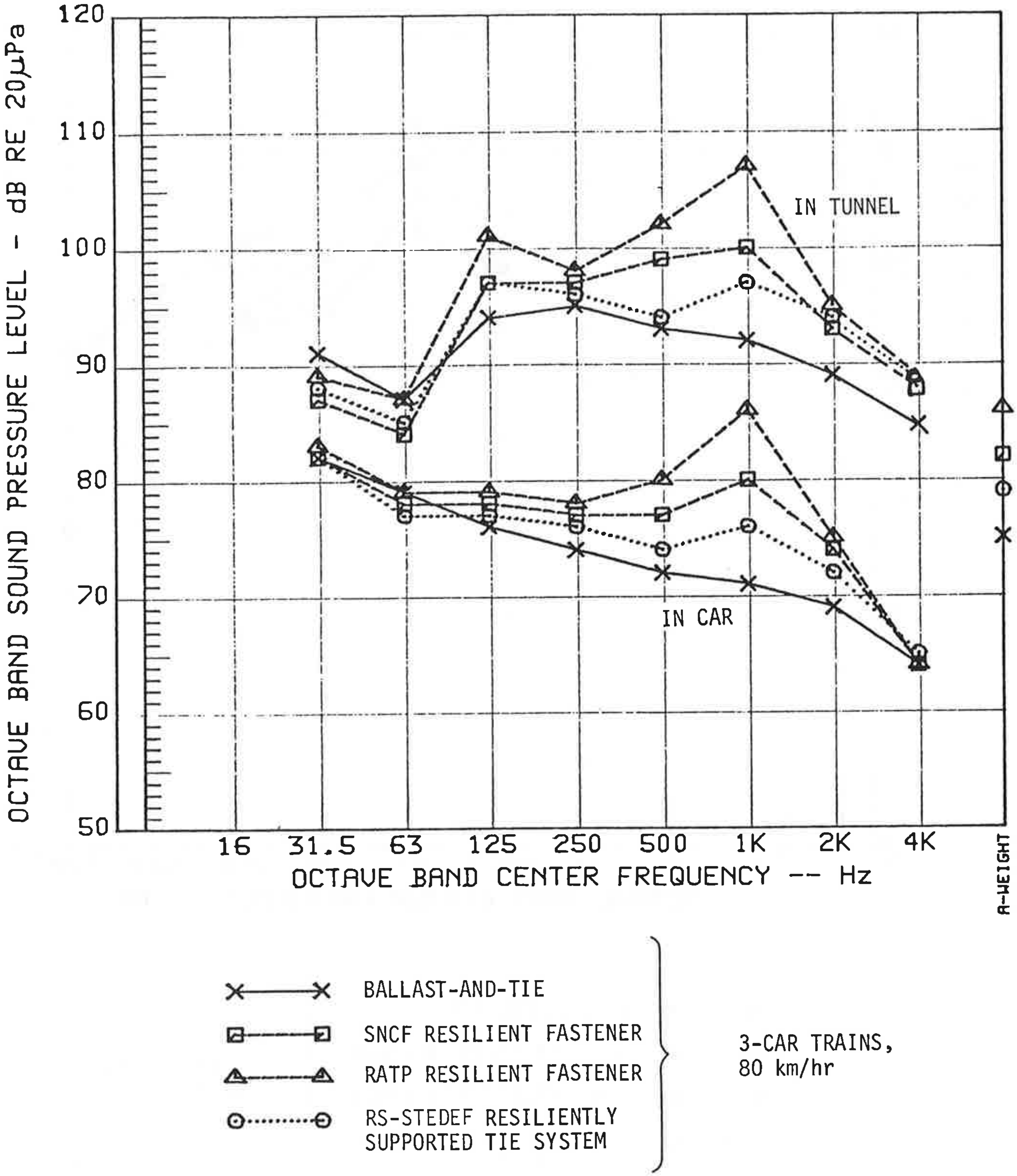


FIGURE 8.34 SUBWAY NOISE LEVELS OBSERVED AT RER SUBWAY TEST INSTALLATIONS

8.4.4 Fastener Design Parameters

The single most important design parameter affecting the transmission of vibration to the ground is the dynamic vertical stiffness, or spring rate, of the assembly. Since an elastomer or rubber spring is a nonlinear spring, this stiffness must be determined from the slope of the load-deflection curve when loaded with a transit vehicle. Furthermore, the dynamic stiffness of the elastomer pad, which affects the transmissibility of vibration to the invert, is greater than the stiffness determined from static load-deflection tests; the ratio of static-to-dynamic stiffness depends on the type and quality of the elastomer. Many factors affect the stiffness of an elastomer support, and these must all be considered in designing and evaluating a direct fixation resilient rail fastener.

The dynamics of the cars and trucks should, of course, be included in the analysis. The design and performance information on direct fixation rail fasteners presented here is based on results obtained from cars whose support systems represent current typical practice on major transit systems.

Fastener design or selection generally involves trading off many technical and economical factors. Direct fixation fasteners with great resilience minimize the vibration transmitted to the subway structure. In contrast, more rigid fasteners provide lower rail stresses and greater rail stability, require less space, are generally less expensive, and provide slightly lower airborne noise levels. The best designs in the past have given fastener vertical stiffness in the range of 90,000 to 150,000 lb/in. per fastener, as determined from static load-deflection curves.

Direct fixation fastener performance has been analyzed theoretically, using models not yet proven to be valid and from these analyses, the optimum vertical stiffness was determined. One analysis, performed by Stanford Research Institute (Ref. 8.47), indicated that a vertical stiffness of about 33,000 lb/in. per fastener would be the optimum design for the BART system. Another analysis (Ref. 8.7) recommended that the vertical stiffness should be 24,000 lb/in., for fastener spacing of 30 in. for the WMATA Metro system. Stiffnesses in the range of 25,000 to 35,000 lb/in. certainly would appear to offer the possibility of improved reduction of the groundborne vibration from the transit trains. However, the practical requirements for rail stability, the increased noise radiation from the rail with softer fasteners, and size and economic limitations, all indicate that the minimum practical vertical stiffness is in the range of 90,000 to 150,000 lb/in. per fastener.

The diminishing return, in terms of reduced groundborne vibration levels, of increasingly softer fasteners is particularly noticeable for stiffnesses less than about 80,000 lb/in. The small reduction of vibration produced by direct fixation fasteners with stiffness less than 90,000 lb/in. (for 30 in. spacing) is more than offset by other disadvantages. Thus, using a direct fixation fastener with a vertical stiffness of 25,000 to 35,000 lb/in. is not necessarily practical or desirable. Achieving improved reduction of the groundborne vibration usually requires a different design principle, i.e., the floating-slab trackbed or the resiliently supported tie rather than softening the direct fixation fastener, which may lead to lateral instability.

A possible exception is the use of the elliptic ring elastomer-in-shear, "Cologne Egg," described earlier. The disadvantage of reduced lateral stiffness, attendant upon most fastener designs of very low vertical stiffness, seems not to apply to the Cologne Egg fastener. The information available indicates that the maximum deflection under full load is about 2.5 to 4 mm, depending on design. Additional research regarding the Cologne Egg design is required, including full-scale experimental evaluation on a U.S. system.

Table 8-9 shows the basic design characteristics of various fasteners for which considerable technical information is available. For most of the fasteners listed in Table 8-9, there are experimental data available on groundborne vibration from transit train operations, either from standard revenue track installations or from test installations where measurements were made under controlled conditions.

In evaluating a fastener design, the static deflection of the fastener cannot be used to determine dynamic characteristics, since an elastomer pad is a nonlinear spring. The maximum dynamic forces are transmitted through the spring when the fastener is loaded by the weight of the passing transit vehicle. Under the lighter loadings, as the vehicles approach and depart, the effective spring rate is less than that at heavier loadings, and the forces transmitted are correspondingly less.

Fastener pads of natural rubber, neoprene, or SBR elastomer compound can all give satisfactory results. There are, however, other properties of the various elastomers which seem to indicate that neoprene is a better elastomer for use in direct fixation rail fasteners. Natural rubber has the best dynamic performance, but in the past it has been more expensive than neoprene when compounded for resistance to the deterioration caused by aging or exposure to air, ozone, or oil. The electrical properties of SBR compounds are only fair. Neoprene has good resistance to weather, ozone, oil, and other factors which deteriorate elastomers over a period of time, and it has the distinct advantage of flame resistance, which is especially important in tunnels. In general, neoprene has been selected as the best compromise.

Although the popular view has been that it is difficult to compound natural rubber to obtain a life expectancy comparable to neoprene or SBR compounds, this is not true; natural rubber should be considered because of its favorable dynamic properties. The manufacturers of elastomer pads use many additives in the elastomer materials, including various types of carbon blacks and waxes, to adjust the mechanical and chemical properties of the

TABLE 8-9 RAIL FASTENER CHARACTERISTICS

Type of Fastener	Vertical Stiffness From Static Tests - lb/in.	Fastener Spacing - in.	Rail Support Modulus - lb/in.^{2} of rail
DIRECT FIXATION FASTENERS			
TTC - Standard	110,000	24	4600
TTC - Double Thickness Pad	60,000	24	2500
BART - Surface	400,000	36	11,000
BART - Subway	130,000	36	3600
WMATA Metro	90,000/100,000	30	3000/3300
MARTA	100,000	30	3300
SNCF	200,000	29.5	6800
RATP	90,000	29.5	3000
RESILIENTLY SUPPORTED RAIL TIE			
RS-STEDEF	70,000	29.5	2400
Cologne Egg	≈30,000	29.5	1000

elastomers. These additives can provide the SBR compounds with many of the age-resistant characteristics of neoprene. The choice of natural rubber, SBR, or neoprene must be based on the basic elastomer properties and on the property adjustments obtainable through compounding.

The actual life spans of elastomer pads depend greatly on the fillers and compounding used by the elastomer manufacturer. Unfortunately, the fillers used in adjusting the mechanical, chemical, and electrical properties of an elastomer all interact, and an element added to improve electrical insulation may, for example, result in increased permanent set or hardening with age. It is, therefore, necessary to specify many tests and a variety of properties to insure that the elastomer will remain effective over the desired life of the rail fastener unit.

The main factor to be considered during compounding is that the elastomer compound should minimize the hardening and creep that can occur as the fastener ages. If a fastener elastomer hardens enough to increase the durometer as much as 15 points, the spring rate could increase from a nominal value of 90,000 lbs/in. to about 160,000 lbs/in. This would increase the vibration levels by about 4 dB, which may not be acceptable. An increase of 10 points in the elastomer durometer, due to aging and permanent set, would increase the spring rate to about 130,000 lbs/in., which should be acceptable. An increase of hardness by 10 durometer points for the fastener life is the maximum desired but is, of course, difficult to predict, and the tests which can be performed may not be realistic.

CHAPTER 8 REFERENCES

8.1 L. G. Kurzweil, R. Lotz, "Background Document for Development of an Urban Rail Noise Control Handbook," Draft Report, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, April 1978).

A summary is included of the effects of various factors on groundborne vibration from rapid transit systems, including fastener stiffness, train speed, and rail and wheel roughness. Also presented is a prediction procedure based on Lange's data and another more detailed procedure based on the concept of corrections representing various factors.

8.2 S. Wolfe, "Ground-borne Vibration from 2000, 2200, and 2400 Series CTA Transit Trains," Preliminary Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Chicago Transit Authority, October 1978).

The effect of vehicle truck design was evaluated on the Chicago Transit Authority system. Wegman trucks on the 2400 Series vehicles were shown to produce substantially lower levels of groundborne vibration that the trucks on the 2000 and 2200 Series vehicles. The reductions observed for operation on welded rail on ballast-and-tie track were greater than those observed on elevated structures with jointed rail. The reductions were consistent for all train speeds up to the maximum speed of 65 mph.

8.3 H. J. Saurenman, G. P. Wilson, R. L. Shipley, "In-Service Performance and Cost of Methods to Control Urban Rail System Noise - Final Report," UMTA-MA-06-0099-80-1, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, December 1979).

Included in this study was one series of ground vibration measurements. Groundborne vibration reductions produced by resilient SAB and Penn Bochum resilient wheels were found to be significant at frequencies above 50 Hz. Reductions of both subway structure vibration and ground surface vibration were observed.

8.4 Communication by Wilson, Ihrig & Associates, Inc. staff with BART Engineering Staff (Oakland, CA, November 14, 1979).

The BART engineering staff indicated that Acousta Flex wheels have performed without difficulty on the BART system over their expected life span.

8.5 J. E. Manning, R. G. Cann, and J. J. Fredberg, "Prediction and Control of Rail Transit Noise and Vibration - A State-of-the-Art Assessment," UMTA-MA-06-0025-74-5, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, April 1974).

The state-of-the-art in groundborne noise and vibration control is summarized. Measurement data are presented for vibration produced by subways, ballast-and-tie, and elevated structure transit train operation. Vibration propagation and attenuation in soil and the effects of various factors such as fastener stiffness and train speed are discussed. Attention is called to the variability of presentation and questions of reliability of vibration data.

8.6 E. E. Ungar and E. K. Bender, "Vibration Produced in Buildings by Passage of Subway Trains; Parameter Estimation for Preliminary Design," Proceedings of Inter-Noise 75, (Sendai, Japan, August 27-29, 1975), pp. 491-498.

A prediction procedure for groundborne vibration in buildings near rapid transit systems is presented. The procedure accounts for layered soil and variability in soil stiffness. The subway wall is considered to be the source of vibration radiated horizontally from the subway structure. Maximum expected subway wall vibration levels are presented, based on measurements in Toronto and New York.

8.7 E. K. Bender, U. J. Kurze, P. R. Nayak, E. E. Ungar, "Effects of Rail Fastener Stiffness on Vibration Transmitted to Buildings Adjacent to Subways," 1832, (Bolt Beranek and Newman, Inc., Cambridge, MA, August 28, 1969).

The effects of rail fastener stiffness on groundborne vibration are described. It is concluded that at audio frequencies, groundborne vibration will increase by 6 dB per doubling of fastener stiffness. Curves are presented which describe groundborne vibration levels for various rail support moduli relative to the standard TTC rail support modulus of 4500 lbs/in.^{2} of rail.

8.8 H. Braitsch, "Das 'Kohner Ei' (Oberbau 1403/c)," Verkehr und Technik, 7, pp. 285-289; 8, pp. 323-328; 10, pp. 460-465; 12, pp. 523-530; (Germany, 1979) (In German).

A four-part article describing a new fastener concept, dubbed the "Cologne Egg", is described by Braitsch to equal the vibration isolation performance of floating-slab isolation systems. The design uses elastomer in shear to achieve low vertical stiffness consistent with requirements for lateral stability.

8.9 J. L. Colombaud, "Noise and Vibration Levels Suit Ballastless Track for Underground Railways," Rail Engineering International, 3(5), (June 1973), pp. 235-240.

Colombaud describes measurement results performed in Paris for evaluation of the vibration isolation performance of the RS-STEDEF resiliently supported tie track system. The results are compared with those for direct fixation fasteners and are shown to be favorable.

8.10 G. P. Wilson, "Noise and Vibration Design Criteria and Recommendations for the Baltimore Region Rapid Transit System," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for the Baltimore Region Rapid Transit System, February 1974).

Recommendations for direct fixation fastener design criteria are presented. The report presents general recommendations for control of groundborne vibration for the Baltimore Regional Rapid Transit System.

8.11 A. W. Paolillo, "Control of Noise and Vibration in Buildings Adjacent to Subways - A Case History," Noise-Con 73 Proceedings, (Washington, D.C., October 15-17, 1973), pp. 152-157.

Measurements were performed at the NYCTA system to evaluate candidate fastener designs for control of groundborne vibration at a specific section of track. Ribbed neoprene was found to provide good vibration reduction relative to standard fastener designs. One design, decided on as appropriate for retrofit, involved encasement of the entire rail foot with elastomer and lateral restraint being provided by the outer fastener body.

8.12 G. P. Wilson, "Diablo Test Track Noise and Vibration Measurements," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinckerhoff-Tudor-Bechtel, June 1967).

Measurements of vibration isolation performance of a large variety of direct fixation fasteners were performed at the BARTD Diablo Test Track. Also tested were resilient and damped wheels, both of which gave reduction of groundborne vibration. Measurements were performed at both at-grade ballast-and-tie and elevated track sections.

8.13 R.D. Bruce, E.K. Bender, E.E. Ungar, "Noise and Vibration Measurements in the Toronto Transit Commission Subway," Report No. 1899, (Bolt Beranek and Newman, Inc., Cambridge, MA, November 26, 1969).

Measurements were performed at the Toronto Transit Commission subways to define the vibration characteristics of subway structures and transmission of vibration to nearby residential structures. Vibration levels for bench, wall, and ceiling subway vibration are presented. These data form in part the starting point for vibration prediction methods used by the authors.

8.14 R. J. Murray, "Noise and Vibration Studies: Track Fastenings," Report No. RD106, (Toronto Transit Commission, Subway Construction Branch, Ontario, Canada, May 1967).

A variety of direct fixation track fasteners are described in terms of geometry and elastomer characateristics. No vibration data are presented.

8.15 "Yonge Subway Northern Extension Noise and Vibration Study Measurement Program Results," Report No. RD115/3, (Toronto Transit Commission, Subway Construction Branch, Ontario, Canada, October 1976).

A study of groundborne vibration was undertaken by the Toronto Transit Commission at the YSNE tunnels. Measurements were performed at numerous locations at the ground surface, inside residences, and on the subway structure. Computer models of vehicle truck and track support dynamics are described. Digital analysis of impulse signals were used to assess the characteristics of the vibration transmission path from the subway to the soil surface. Various vibration control techniques were evaluated, including halving of the fastener and primary suspension stiffnesses.

8.16 J. E. Manning, D. C. Hyland, G. Tocci, "Vibration Prediction Model for Floating Slab Rail Transit Track," Final Report, UMTA-MA-06-0025-75-13, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, August 1975).

A vibration force reduction model is developed for the coupled interaction of rail and floating-slab bending modes. In the latter, transverse bending of the floating slab is included. Discontinuous floating slabs are treated by setting the floating-slab bending stiffness to zero. For slab support springs with sufficient damping, the model is shown to give results similar to those predicted by the single-degree-of-freedom spring mass analogy.

8.17 "TTC Track Vibration Isolation System," (Toronto Transit Commission, Engineering Department, Ontario, Canada, 1978).

The TTC discontinuous floating-slab vibration isolation system adopted for control of groundborne vibration is described. Vibration reduction data are presented which indicate that the system performs very well at frequencies above 25 Hz.

8.18 G. P. Wilson, K. G. Knight, "Design and Performance of a Floating Slab Trackbed," <u>Noise-Con 73 Proceedings</u>, (Washington, D.C., October 15-17, 1973).

The measured vibration isolation performance of the WMATA continuous floating slabs is described. The data indicate performance exceeding expectations. The vibration reductions were evaluated with a mechanical shaker attached to the slab.

8.19 G. P. Wilson, "Ground-Borne Vibration Reduction of the B-1 Contract Demonstration Floating Slabs for the WMATA Metro System," Final Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for De Leuw, Cather & Company and the Washington Metropolitan Area Transit Authority, November 1973).

The vibration reduction of the WMATA continuous floating-slab design was measured with a mechanical shaker under preload. The data indicate performance consistent with expectation for a stationary point excitation. Following these results, the continuous floating slab concept was adopted by the WMATA system as a standard vibration control provision for subway structures near sensitive buildings.

8.20 G. P. Wilson, Letter report to DeLeuw Cather & Co. regarding floating slab evaluation at the WMATA system, (Wilson, Ihrig & Associates, Inc., Oakland, CA, October 31, 1975).

Vibration reductions were measured for continuous single-track and double-crossover floating slabs during controlled tests with 2-car transit trains at the WMATA system. The single-track floating slabs performed as expected, while a double-crossover slab did not provide the expected reduction of vibration transmitted to the structure. This is apparently due to transverse bending modes excited in the double-crossover slab.

8.21 J. T. Nelson, H. J. Saurenman, G. P. Wilson, "MetroRail

Operational Sound Level Measurements: Ground-Borne Vibration and Noise Levels," Preliminary Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, December 1979).

Measurements were performed at the WMATA Metro system to characterize groundborne vibration from the subway and ballast-and-tie track structures. Measurement locations at the ground surface were over the subway and 50 feet from the track centerline for subways, and approximately 70 feet from the ballast-and-tie track. Measurements were also performed within the subway structure to assess the vibration transmission properties between the structure and the ground surface. Substantially lower levels of vibration at the ground surface were observed for rock tunnels than for earth-based subway structures. Estimates of floating-slab vibration reduction and building coupling losses are included.

8.22 J. T. Nelson, "Ground-Borne Vibration from Rail Rapid Transit Train Operations," Presented at the 96th Meeting of the Acoustical Society of America, (Honolulu, Hawaii, November 27-December 1, 1978).

Measurements of groundborne vibration at the WMATA system for a variety of subway configurations and founding conditions are described. Also presented are measurement results for groundborne vibration levels for the CTA 2400 Series vehicle, using Wegman trucks, relative to levels for the 2000 and 2200 Series vehicles. Estimates of vibration reductions for floating slabs used at WMATA, TTC, and MURLA are presented.

8.23 H. Tajima, K. Kiura, "Laying of Track Ballast Mats in Shinkansen," <u>Permanent Way</u>, 15(3), (1974), pp. 11-19.

Use of ballast mats on a Shinkansen elevated structure was found to reduce ballast pulverization and structure vibration. The mats were constructed of discarded automobile tires.

8.24 L. Steinbeisser, "Korperschallmessungen in Zurich and Munchen," VDI-Berichte, No. 217 (1974) (In German), pp. 37-42.

Measurements of vibration attenuation across a floating-slab vibration isolation system indicate successful isolation of a commercial space directly below the subway. One-half octave band data are presented.

8.25 J. Hofer, J. Sailler, "Versuche fur einen Schotterlosen Oberbau mit Elastische gelagerten Querschwellen," Verkehr und Technik, 9, (1972), pp. 381-386.

The vibration reduction characteristics of ballastless track constructions are described, including the use of polyurethane plastic ties, ribbed elastomer sheathing, and fiberglass pads. These components were eventually decided upon for future installation on the Vienna U-Bahn.

8.26 H. D. Oelkers, H. W. Koch, "Results of Noise Measurements for Different Types of Superstructure Existing in Urban Traffic," VDI-Bertchte, No. 170, (1971) (In German).

Measurements are summarized for groundborne noise and vibration at five subway systems in Europe. Data presented includes: (1) standard ballast-and-tie track, (2) elastomer tie support pads, (3) ballasted floating slab, (4) ballast with ISOLIF TRIPLEX ballast mat, and (5) ballast with ballast mat constructed of pieces of rubber. The ballasted track with ISOLIF TRIPLEX ballast mat gave about 15 dB reduction of impact vibration velocity at the invert relative to the ballasted track configuration.

8.27 J. Lang, "Messergebnisse zum Korperschallschutz fur U-Bahnen," ("Measurement Results Seen in Relation to the Protection from Structure-Borne Noise in Underground Systems"), <u>Proceedings Seventh International Congress on Acoustics</u>, (Budapest, Hungary, 1971), pp. 421-424.

Measurement results are presented for A-weighted noise and vibration velocity reductions using impulse sources. Several trackbed configurations were tested, including: (1) wood ties on concrete base, (2) wood ties on 20cm and 40cm of ballast, (3) wooden ties on rubber pads on concrete invert, (4) ballast-and-wooden tie track on floating slab, (5) ballast and wooden tie track with Isolif-Triplex ballast mat, and (6) ballast-and-wooden tie track a "bed of rubber pieces." The Isolif-Triplex ballast mat gave about 15 dB reduction of impact vibration at the invert. Also presented are measurements of A-weighted noise levels in cellars and groundfloor rooms of residential buildings near several subway systems as a function of distance from the subway.

8.28 A. Devaux, "Reduction of Vibrations and Noise Transmitted by Trains and Construction Sites," <u>Monatschrift der International Eisenbahn</u> - Kongress - Vereinigung, XLVI(11) (November 1969) (In German).

Elastomer ballast mats of 27 mm thickness placed beneath the ballast have resulted in vibration reductions at all systems where it was tested. Reductions of 1 to 10 dB are reported for invert vibration.

8.29 H. W. Koch, "Comparative Values of Structure-Borne Sound Levels in Track Tunnels," <u>Journal of Sound and Vibration</u>, 66(3), (October 1979), pp. 377-380.

The effects of tunnel wall thickness and trackbed configuration are summarized with respect to groundborne vibration from both transit and railroad tunnels. The tunnel wall vibration is found to decrease by about 17 dB per doubling of average tunnel wall thickness. The average tunnel wall thickness is indicated to be related to the subway structure mass per unit length. Increasing the ballast thickness from 30 or 40 cm to 70 cm produced a 5 dB reduction of vibration.

8.30 D. D. Barkan, Dynamics of Bases and Foundations, Translated by L. Drasheuska, (McGraw-Hill Book Company, NY, 1962).

This monograph thoroughly treats the subject of the dynamics of bases and foundations. Both theoretical models of wave propagation in soils, dynamic responses of foundations on idealized half-spaces, practical design formulae, data for wave propagation velocities, loss factors in soils, measurement results and a critical evaluation of various screening methods are all presented.

8.31 F. E. Richart, R. D. Woods, J. R. Hall, Vibrations of Soils and Foundations, (Prentice-Hall, Inc., NJ, 1970).

The authors summarize the theoretical and practical aspects of wave propagation in soils, dynamic soil structure interaction, screening of soil vibration, geophysical measurement techniques for evaluation of soil properties, and a variety of other topics. A good discussion of lumped parameter models of foundation interaction with the soil surface is presented.

8.32 L. Hannelius, "Vibration from Heavy Rail Traffic," (Geotechnical Department, Swedish State Railways, Stockholm, 1978).

Results are presented for a variety of measurements of groundborne vibration from a Swedish mainline railroad. Seismograph recordings indicate that ground vibration from at-grade track is primarily Rayleigh surface waves. Trackbeds, which are supported on piles and on bedrock, produce significantly lower levels of vibration than unsupported trackbeds. A variety of measurement locations was used for evaluation of building vibration.

8.33 E. E. Ungar, E. K. Bender, "Vibrations Produced in Buildings by Passage of Subway Trains; Parameter Estimation for Preliminary Design," Proceedings of InterNoise 75, (Sendai, Japan, 1975).

A method is outlined for prediction of groundborne noise and vibration from rapid transit subways. The method accounts for soil stiffness and damping, effects of layering, and sound absorption within building spaces. The method assumes compression waves in soil and the wall to be most significant to transmission. Loss factor and propagation velocity data are presented for a variety of soil types.

8.34 H. P. Verhas, "Measurement and Analysis of Train Induced Ground Vibration," (Dissertation for University of Southampton, Faculty of Engineering and Applied Science, England, 1977).

Verhas has performed a series of measurements to determine the transmission characteristics of vibration from a British mainline railroad to wayside ground surface locations. The velocity of propagation was determined to be similar to that of shear waves in soil at both low and high frequencies. The attenuation with distance is discussed in terms of body and surface waves from line and point sources.

8.35 T. G. Gutowski, C. L. Dym, "Propagation of Ground Vibration: A Review," Journal of Sound and Vibration, 49(2), (November 1976), pp. 179-193.

The authors summarize the practical theory of wave propagation and attenuation in soils. Models are presented for frequency-independent and frequency-dependent loss factors, neither of which are accurate for all measurement data. The frequency-independent loss factor is the most popular representation of damping in soils.

8.36 G. P. Wilson, "Ground-Borne Vibration Levels from Rock and Earth Base Subways for the WMATA Metro System," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for DeLeuw, Cather & Company and the Washington Metropolitan Area Transit Authority, September 1971).

Wilson presents methods for the prediction of groundborne noise and vibration from rapid transit systems for rock and earth-based subways on the WMATA Metro system. The method begins with expected ranges of vibration levels for rock, mixed-face, and earth-based subways, at 30 feet from the subway structure. Corrections are then applied to account for fastener stiffness, tunnel mass, train speed, special trackwork, building coupling loss, attenuation in soil, etc. Soil stiffness and loss factor data are not included. The general approach was first described in Ref. 8.38.

8.37 K. Ishii, H. Tachibana, "Field Measurement of Structure-Borne Sound in Buildings," Acoustical Society of America Reprint No. L10, Presented at the joint meeting of the Acoustical Society of America and the Acoustical Society of Japan, (Honolulu, Hawaii, April 1977).

Experimental results are presented for floor-to-floor attenuation of vibration in a large multistory building. Point excitation with a mechanical shaker was used. The

data indicate about 3 dB reduction per floor at the lower floor levels, but with less or no reduction at higher floor levels.

8.38 G. P. Wilson, "Noise and Vibration Characteristics of High-Speed Transit Vehicles," Technical Report, OST-ONA-71-7, (U.S. Department of Transportation, Office of Noise Abatement, Washington, D.C., June 1971).

Wilson discusses the general characteristics of noise and vibration from modern transit systems. The prediction of groundborne vibration and noise is discussed in detail. A simple spring mass analogy is used to estimate groundborne vibration produced by rock-founded subways relative to earth-founded subways.

8.39 E. K. Bender, U. J. Kurze, K. S. Lee, E. E. Ungar, "Predictions of Subway-Induced Noise and Vibration in Buildings Near WMATA, Phase 1," Report No. 1823, (Bolt Beranek and Newman, Inc., Cambridge, MA, Prepared for Washington Metropolitan Area Transit Authority, May 20, 1969).

Groundborne vibration levels are predicted for buildings near the WMATA Metro Transit system. The method of prediction and underlying assumptions are briefly described.

8.40 G. F. Miller, H. Pursey, "On the Partition of Energy between Elastic Waves in a Semi-Infinite Solid," Proceedings Royal Society, London, A:233, (1955), pp. 55-69.

The authors determine the partition of far-field vibration energy between shear, compression, and Rayleigh waves for excitation of the soil surface of an elastic half-space by a circular disc. The results indicated that Rayleigh waves, and to a lesser extent, shear waves are the dominant carriers of vibration energy. Compression wave energy is

least significant.

8.41 Y. Tokita, A. Oda, K. Shimizu, K. Kimura, "On the Ground-Borne Noise Propagation from a Subway," Presented at the Joint Meeting in Hawaii of the 96th Session of the Acoustical Society of America and the Acoustical Society of Japan (Honolulu, Hawaii, April 1978).

A method is presented for the prediction of ground surface vibration as a function of distance from subway structures. An effective, or equivalent, line source is assumed for purposes of calculation to be just beneath the subway structure. The structure is then assumed to act as a barrier. Soil stiffnesses, loss factors, and layering are included in the model in much the same manner as Reference 8.33. Curve fitting techniques are used to determine the source depth and strength for a given set of measurement data. The model is capable of predicting an observed maximum in the vibration level as a function of distance for a nonzero horizontal distance from the subway structure.

8.42 W. Blazier, "Study of Vibration Transmissability Properties of Soil at Boeing Development Center," Report No. 1392, (Bolt Beranek and Newman, Inc., San Francisco, CA, December 1966).

Results are presented for coupling loss of building piles in response to incident groundborne vibration.

8.43 D. D. Kalic, A Simple Formula for Estimating Vibrational Noise Level in Buildings," Proceedings of the 9th International Congress on Acoustics, (Madrid, Spain, July 1977).

Radiation of noise by vibrating building wall panels is discussed in detail. The effects of room absorption are included to derive interior reverberant sound levels in rooms.

8.44 A. W. Paolillo, "Ground-Borne Vibration Generated by Rapid Transit Rail Cars," Technical Report, (New York City Transit Authority, June 7, 1978).

Increased groundborne vibration associated with the design of rail vehicle suspension systems are reviewed. A primary suspension resonance at about 20 Hz is identified as contributing to increased groundborne vibration and overstressing of the truck frame. Coincidence of the truck suspension resonance and the resonances of nearby buildings is identified as a contributing factor.

8.45 P. Grootenhuis, "Floating Track Slab Isolation for Railways," Journal of Sound and Vibration, 51(3), (April 1977), pp. 443-448.

Floating-slab vibration isolation is discussed. A continuous, constrained layer-damped floating slab is recommended for small cross-section circular tunnels. Continuous floating slabs are deemed more desirable than discontinuous double ties such as those employed by the TTC system due to concentration of local vibration energy. No supportive data are presented.

8.46 G. P. Wilson, Private Communication, (Wilson, Ihrig & Associates, Inc., Oakland, CA, 1979).

Measurements were performed which indicate that coincidence between the resonance frequencies of the floating slab and residential structure floors produce amplification of low-frequency vibration.

8.47 V. Salmon, S. K. Oleson, "Noise Control In The Bay Area Rapid Transit System," Interim Report, (Stanford Research Institute, Menlo Park, CA, Prepared for Parsons

Brinckerhoff-Tudor-Bechtel, February 1965)

General recommendations are given for control of noise and vibration at the then proposed BART system. In particular, a fastener vertical stiffness of 33,000 lbs/in., with a lateral compliance not in excess of 20% of the vertical compliance, is recommended.

CHAPTER 9

9. STATION NOISE CONTROL

Transit stations have four main sources of noise: trains entering, leaving, and passing through the stations; ancillary equipment, such as HVAC equipment and escalators; crowds; and, in above-ground stations, street or highway traffic and railroads. A typical example of the latter is a station platform located in a freeway median. This section discusses methods of controlling each of these sources of station noise.

The APTA (American Public Transit Association) Guidelines (see Appendix B) present design goal maximum levels for various sources of station noise. For reference in the following discussions, these criteria are summarized in Table 9-1.

The principal means of controlling noise in subway stations are: the use of sound absorption treatment to control reverberant buildup of acoustic energy, careful design of ancillary equipment to ensure that it meets appropriate sound emission criteria, and the use of barriers to block the sound from the source (e.g., trains passing through stations) to the receiver (usually the patron).

Figure 9.1 illustrates an example of an acoustically treated subway station. The figure illustrates the use of a platform height sound barrier to control train noise and appropriate locations for acoustical absorption treatment. In most cases, it is possible to maintain satisfactory levels of train noise on the platforms through the use of sound absorption materials. The noise sources on a transit car are primarily located in the confined space beneath the transit cars. Hence, sound-absorbing materials on the trackbed or on the walls near undercar spaces effectively absorb the sound energy and reduce the level of train noise on the station platform. In effect, such treatment reduces

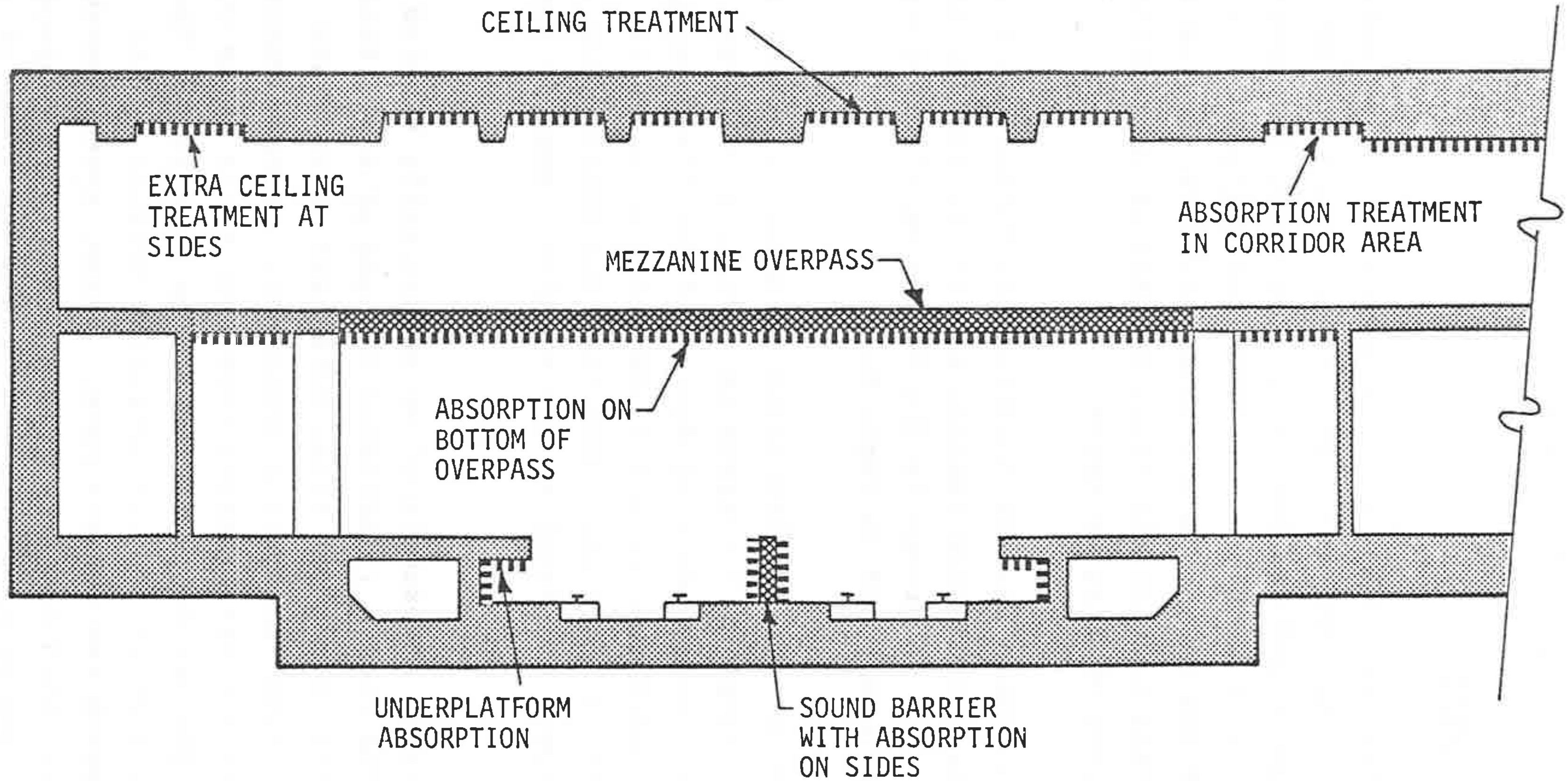


FIGURE 9.1 ACOUSTIC TREATMENT IN A SUBWAY STATION

the sound power radiated into the platform area in addition to limiting the reverberant buildup of sound energy.

On side platform stations, further reductions can be achieved with sound barriers to block noise from far track trains. Because most of the sources of train noise are concentrated beneath the car, the barriers only need to be as high as the platform level to achieve significant reductions of train noise.

A platform height barrier between the near and far tracks of a side platform station can reduce sound levels on the platform by as much as 10 dBA. The amount of reduction is dependent on the design of the barrier and the measurement location. The greatest reduction occurs on the far platform.

9.1 ACOUSTICAL TREATMENT OF STATIONS

9.1.1 Background and Criteria

Rapid transit system stations are often reverberant, noisy spaces. Designers have traditionally used acoustically reflective materials, such as concrete or ceramic tile, on all surfaces of train platform areas for durability, abuse resistance, and ease of cleaning. With these materials, sound energy from train operations is not dissipated, but is reflected back into the room on impinging a wall.

Applying absorption treatment to the walls and ceilings prevents excessive buildup of reverberant sound energy and substantially reduces noise. This treatment reduces crowd noise, exterior noise transmitted through entrances and shafts, mechanical equipment noise, and other noise. It also greatly improves the intelligibility of public address systems.

All new rail transit subway stations in the United States have, or will have, sound-absorbing material on the station interior surfaces. This includes stations at BART (Bay Area Rapid Transit District), WMATA Metro (Washington Metropolitan Area Transit Authority), MARTA (Metropolitan Atlanta Rapid Transit Authority), BRRT (Baltimore Region Rapid Transit System), NFTA (Niagara Frontier Transit Authority, Buffalo), and new stations added to the TTC (Toronto Transit Commission), NYCTA (New York City Transit Authority), and CTA (Chicago Transit Authority) facilities.

For subway stations, acoustical design criteria require that sound-absorbing material be installed in underplatform areas, platform walls and ceilings, and in enclosed concourse spaces such as fare collection areas, stairs, escalators, and corridors. Similarly, enclosed areas of above-grade stations should have ceiling- and wall-mounted absorption treatment. These considerations create an attractive acoustic environment for the transit patrons. Fortunately, the platform areas of most surface stations are only partially enclosed. Since the open areas do not reflect sound into the reverberant sound field, they have the same effect as acoustical treatment. Additional acoustical treatment to control reverberation is generally not needed in surface station platform areas or other spaces with large openings to the outdoors. However, absorption should be considered to control reverberation under canopy areas. The absorption can be particularly important when the station platform is exposed to traffic noise.

9.1.2 Placement of Materials

Obtaining maximum benefit from acoustic treatment requires that suitable amounts of the material be installed in the proper locations. In general, for control of any noise source, it is best if absorption treatment can be located close to the noise source so that some of the sound energy can be absorbed before it reaches the reverberant sound field. Inappropriate placement of treatment can control reverberation without obtaining satisfactory or efficient reduction of train noise.

Figure 9.2 indicates the reverberation times measured in WMATA Metro subway stations and BART subway stations, respectively, before and after acoustical treatment on ceiling and underplatform overhang surfaces were installed. Noise levels from the transit train operations are much more acceptable than those found in older systems with completely untreated, highly reverberant stations. The reverberation times measured in treated BART and WMATA stations are typically 1.3 to 1.5 seconds at 500 Hz, as compared with 7 to 9 seconds for untreated stations.

Figure 9.3 shows noise levels on the platforms for trains passing by at 64 km/hr at several subway stations. The noise levels at BART and WMATA platforms are in the range of 87 to 89 dBA. Similar measurements at untreated CTA stations, under similar operating conditions and using similar trains, indicate train noise levels as high as 108 dBA on the platform of stations with concrete trackbed and 93 dBA on the platform of stations with ballast-and-tie tracks. The 15 dBA difference due to the absorption properties of ballast confirms that the ballast provides a significant amount of absorption, reducing the reverberant sound energy buildup of train passbys.

Noise measurements inside WMATA Metro cars indicate that acoustical treatment of subway stations can substantially reduce car interior noise levels. Figure 9.4 shows the results of these measurements. In a box structure with no sound absorption treatment, the interior noise level for a 2-car train operating at 64 km/hr was 79 dBA, whereas in passing through an acoustically treated station, the interior level was 68 dBA. The same type of measurement indicated 64 dBA for at-grade, ballast-and-tie stations where no reflective sound impinges on the transit car.

Figures 9.5 and 9.6 show the noise levels achievable through the use and placement of sound absorption materials in subway stations. Figure 9.5 presents typical noise levels, measured in TTC tunnel stations having sound absorption treatment on the underplatform overhang surfaces only (an insufficient amount to

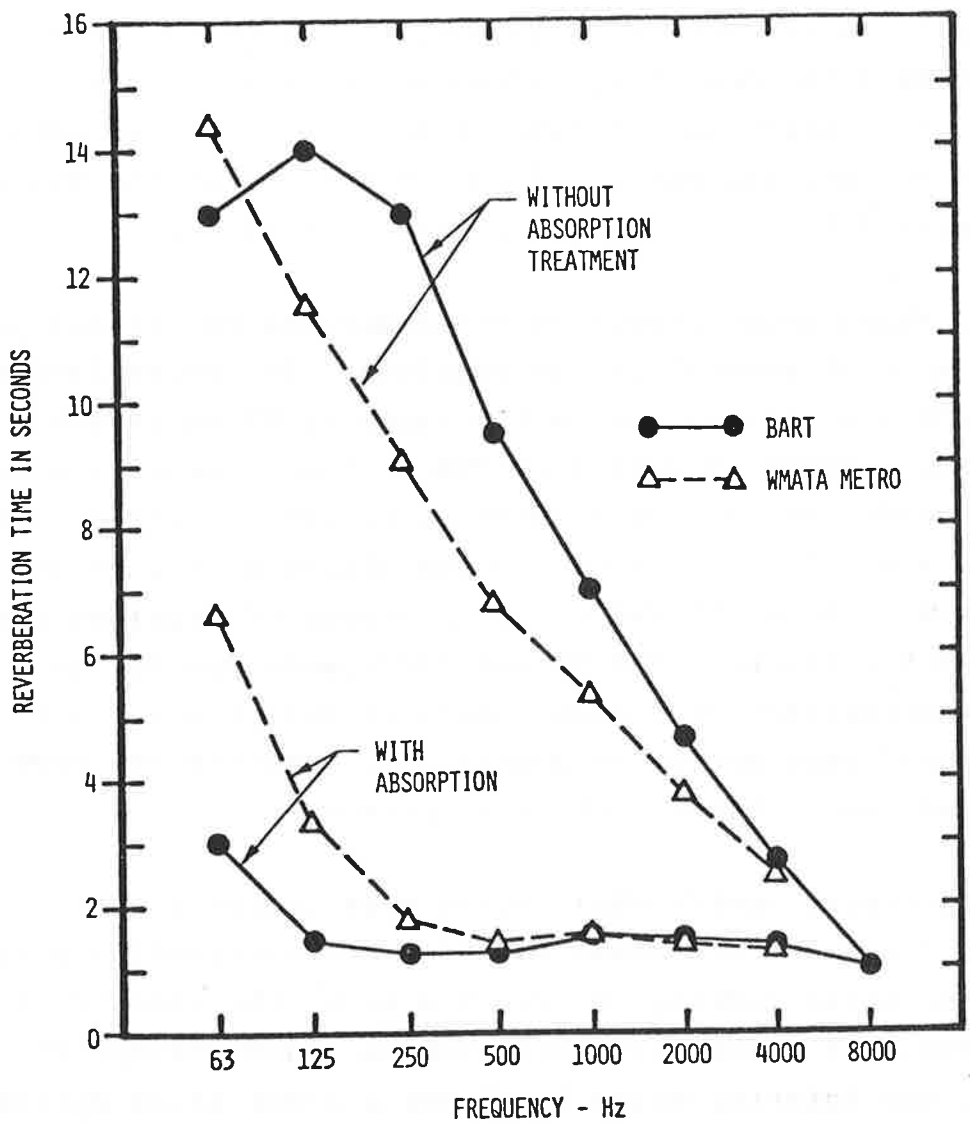


FIGURE 9.2 SUBWAY STATION REVERBERATION TIMES BEFORE AND AFTER INSTALLATION OF SOUND ABSORPTION MATERIALS

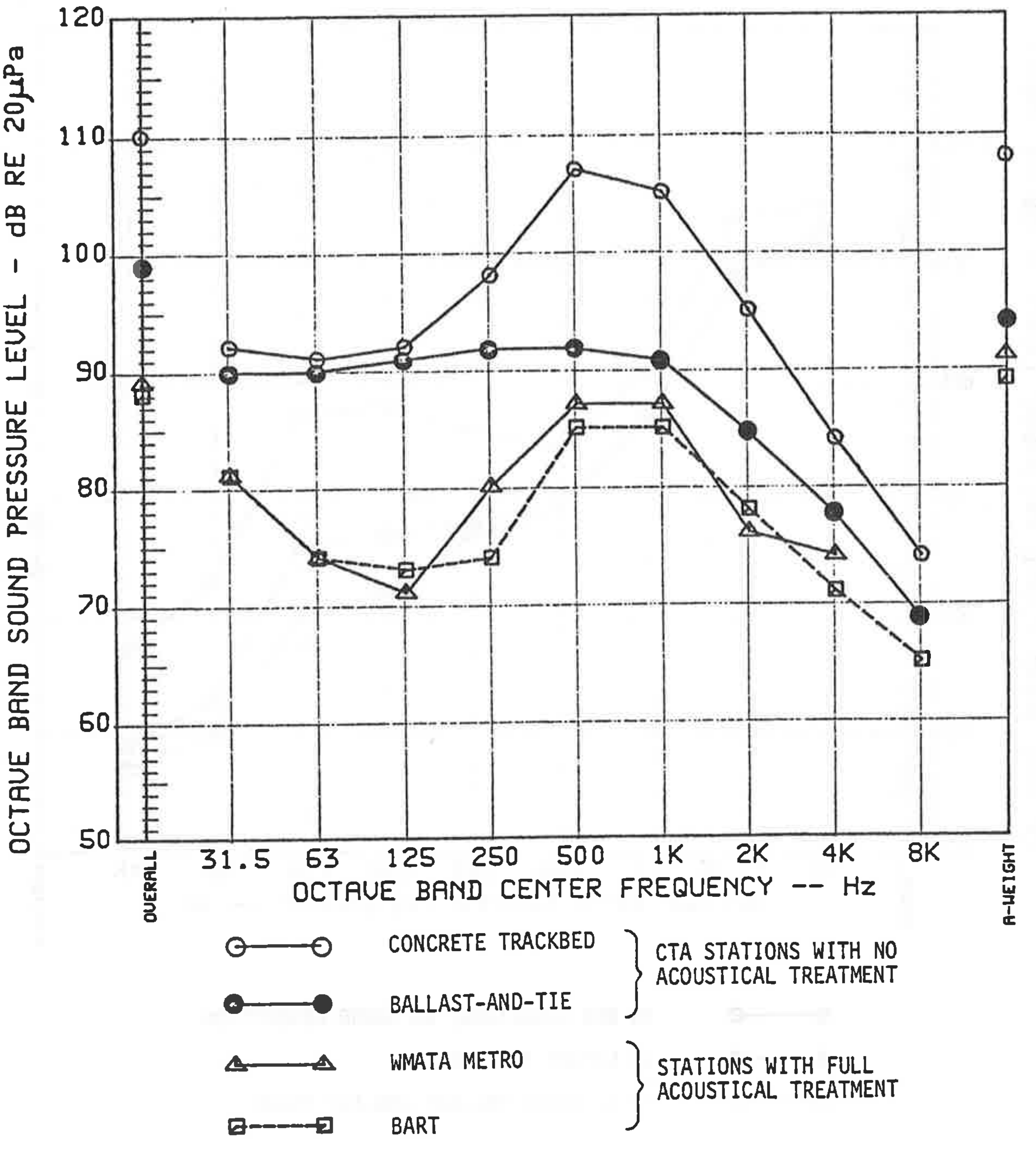


FIGURE 9.3 SUBWAY STATION PLATFORM NOISE LEVELS WITH TRAINS PASSING THROUGH AT 64 km/hr (40 MPH)

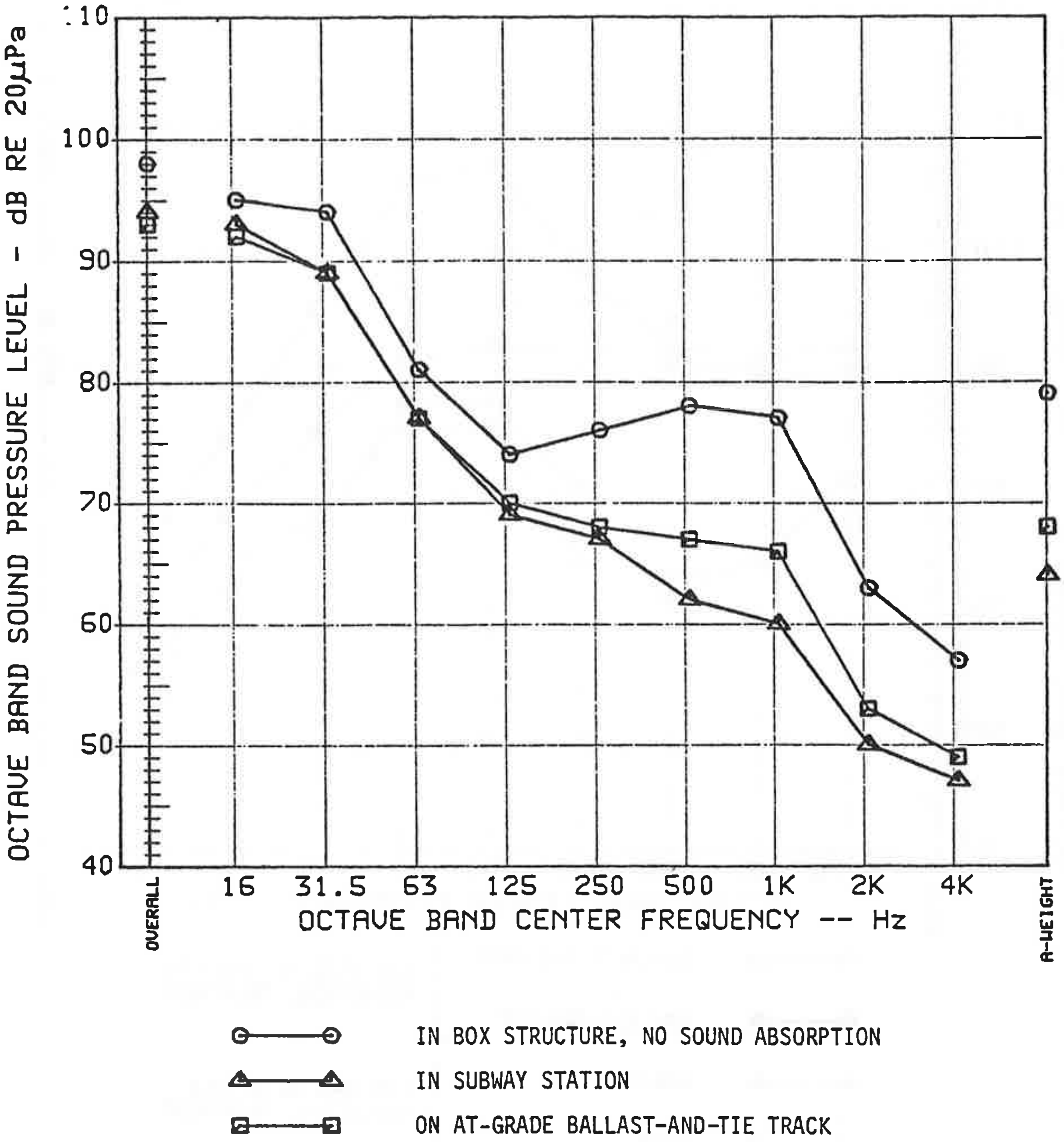


FIGURE 9.4 WMATA METRO CAR INTERIOR NOISE LEVELS, 2-CAR TRAIN AT 64 km/hr

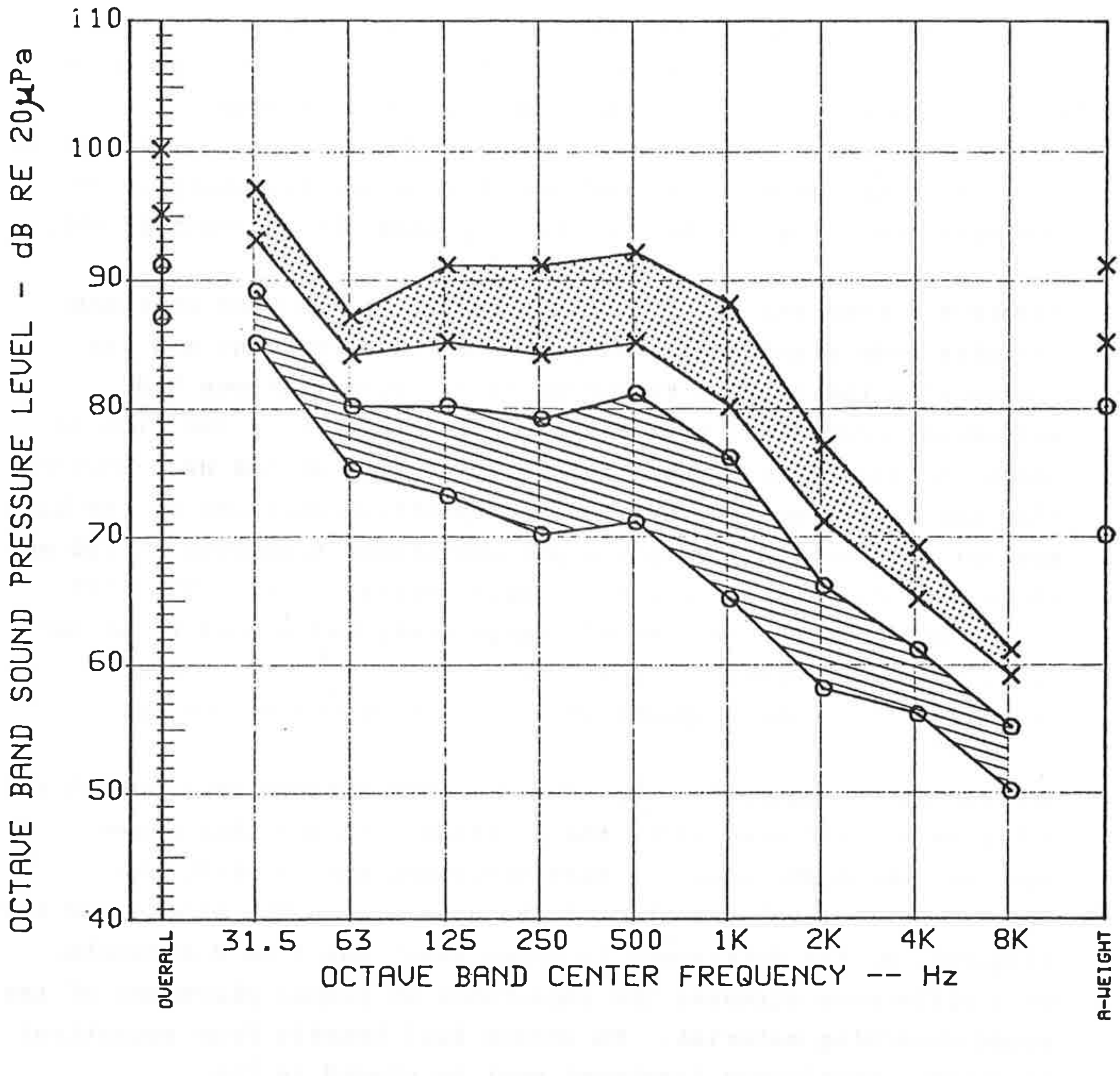


FIGURE 9.5 MAXIMUM WAYSIDE NOISE LEVELS ON PLATFORMS OF TTC TUNNEL STATIONS WITH TRAINS ENTERING AND LEAVING

control reverberation), and in a station in which the entire ceiling, as well as the underplatform, has been treated. The range shows the typical maximum levels that occur on the station platforms as trains arrive and depart. The sound absorption on the ceiling in this case is provided mainly by a suspended acoustical tile ceiling, an arrangement which gives nearly uniform absorption and noise reduction over the entire frequency range. The effective noise difference is very dramatic -- about 13 dBA.

Figure 9.6 compares noise levels observed in two BART stations: one with underplatform overhang and ceiling treatment and the second with the ceiling treatment only. Both stations had sufficient acoustical treatment to reduce reverberation time to about the same range, e.g., about 1.3 seconds at 500 Hz. However, they were different in that the underplatform surfaces at the Lake Merritt Station had a complete and continuous treatment of 100 mm (4 in.) thick glass wool with a sheet plastic cover. The 19th Street Station, at the time of measurement, had almost no acoustical treatment under the platform edge and only one row of acoustical tile units spaced at about 0.6 meters on center.

The charts in Figure 9.6 show the dramatic effects of treating the relatively small area under the platform. In the 19th Street Station, where the underplatform treatment was omitted, the average noise level was about 5 dBA greater; in the middle and low frequencies, the difference in noise level was 5 to 8 decibels. This difference stresses the importance of proper placement of the sound-absorbing material. To obtain full benefit from acoustical treatment, continuous treatment <u>must</u> be placed in the underplatform overhang area.

The use of sound-absorbing materials is, however, to some extent governed by the law of diminishing returns. It seems logical that if some acoustical absorption material provides good results, more will provide even better. But the amount of noise control achievable in this way is limited; beyond a certain point, it becomes very uneconomical and inefficient to use more absorption material,

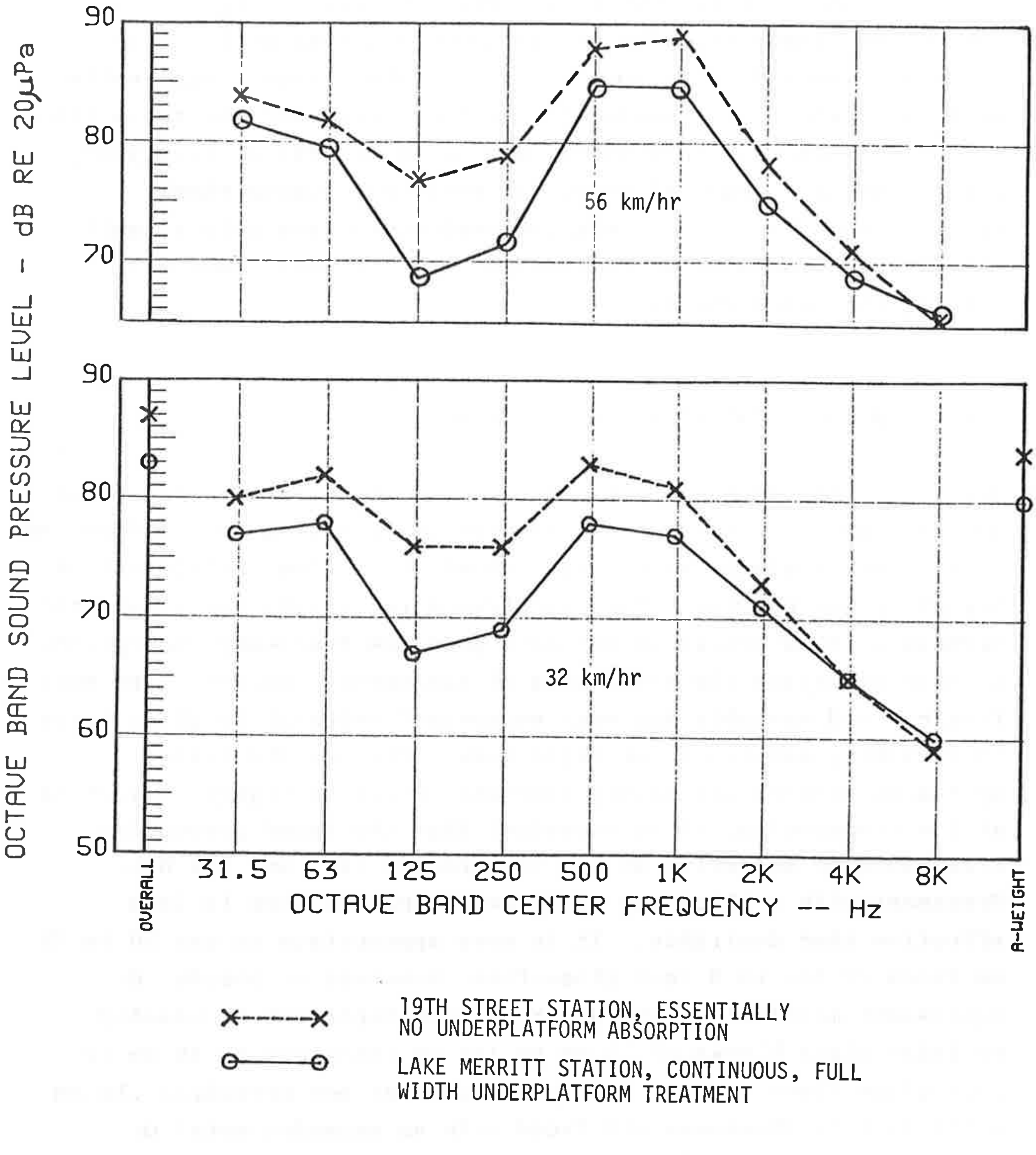


FIGURE 9.6 NOISE LEVELS ON BART UNDERGROUND STATION PLATFORMS WITH TRAINS PASSING THROUGH AT CONSTANT SPEED

and other noise control procedures must be used.

The guidelines in Table 9-2 are designed for the efficient use of materials and indicate the diminishing utility of excess sound absorption treatment. The recommended treatment will control reverberation and train noise to the greatest extent achievable with sound absorption material. Further noise and reverberation control is possible through using greater amounts of treatment, but the amounts required would be more than double those recommended. The improvement achieved would have only a small effect on the acoustical environment and certainly would not justify the added costs.

9.1.3 Types of Materials and Coverings

Because of the frequency characteristics of transit system noise and the lack of low-frequency absorption in untreated stations, an acoustical treatment with a high sound absorption coefficient at 500 Hz should be used. This requires a relatively thick material because thin materials do not have good low-frequency absorption. It also minimizes the total area of treatment required. The most flexible and probably the most economical material is glass fiber in flexible, semirigid, or rigid form. Because the transit system structures are mainly concrete, which is highly reflective at low frequencies, it is essential that the sound absorption treatments be effective at low frequencies (125 and 250 Hz). Treatment with a thickness of 25 mm (1 in.) or less is less effective than desirable. It is more appropriate to use 50 to 75 mm thick (2 in. to 3 in.) glass-fiber blankets or boards or equivalent materials. The underplatform treatment is usually cellular glass blocks of 50 mm to 100 mm thickness or 75 mm (3 in.) glass fiber wrapped in a plastic sheet not exceeding .10 mm (.004 in.) in thickness and faced with an expanded metal or perforated sheet metal facing. For protective and architectural purposes, facings or perforated metal or metal slit-and-slat are commonly used on the side walls and ceilings. In order to avoid

TABLE 9-1 DESIGN GOAL MAXIMUM NOISE LEVELS - TRANSIT STATIONS

Location	Level dBA
On platform, train entering and leaving	80 to 85
On platform, trains passing through	85
On platform, trains stationary	68
On platform or in mezzanine areas with only station ventilation system and other auxiliaries operating	55
In station attendants' booths or offices	50

TABLE 9-2 CRITERIA FOR ACOUSTICAL TREATMENT OF TRANSIT STATIONS

	Areas Exposed to Street Traffic	Enclosed Concourse Areas	Platform Areas
Maximum Reverberation Time (500 Hz)	1.2 to 1.4 sec.	1.2 sec.	1.5 sec.
Treatment:			
Minimum wall/ceiling area	20-25%	35%[1]	35%[2]
Minimum ceiling only	70-100%	--	--
Treatment Properties:			
Minimum 500 Hz absorption coefficient	0.6	0.6	0.6[2]
Minimum NRC	0.6	0.6	0.6[2]

[1]Including at least 50% of ceiling area.

[2]Underplatform treatment also required -- minimum absorption coefficient at 250 Hz = 0.4, at 500 Hz = 0.65 (75 mm to 100 mm thick material).

degradation of the sound absorption coefficients at high frequencies, the open area of the facing material should be at least 20 percent. For enclosed areas such as mezzanine, concourse, and entrances, noise and reverberation control requires acoustical treatment on the ceiling or both the ceilings and walls.

Note that as long as the noise created by the trains is consistent with the APTA Guidelines, then following the criteria of Table 9-2 will ensure that the design goals for station noise control are achieved.

9.1.4 Selection of Acoustical Material

Acoustical treatment for transit system stations consists of three basic elements:

- The sound absorption media or material.
- A protective covering.
- An architectural or trim facing.

In some treatments, each of these elements is an individual material; in others one material combines two or even all three of the functions. For example, glass-wool blankets, encased in plastic bags with a perforated or expanded metal covering, provide a treatment in which each material functions individually. Acoustical tile with painted or vinyl facing is an example of treatment in which functions are combined. Another factor that must be considered in the overall design is the fastening, since each type of treatment requires a different fastening system.

One very important consideration is the fire rating of the acoustic treatment. Clearly, the material must meet all of the safety standards, including fire and smoke ratings. Certain flammable materials are effective for sound absorption, however, nonflammable materials are also available; every effort should be made to use nonflammable materials for station acoustic treatment. In order to provide designers with sufficient information, the following discussion includes comments on both flammable and nonflammable materials and their effectiveness.

9.1.4.1 Sound-Absorbing Materials -- One of the most economical sound-absorbing materials is glass wool (also called glass fiber) boards or blankets. Unfortunately, the binders used in most of these materials are flammable. However, nonflammable configurations of glass wool are available which use no binder material at all. Although glass-fiber products without binders have a present oil residue that could produce smoke, the oil residue can be removed by a baking process. Glass fiber can be found in a number of different forms, including flexible, semi-rigid and rigid boards. Table 9-3 indicates the sound absorption coefficients that can be expected for various thicknesses of glass fiber. For acoustical treatment, the recommended density for glass fiber is 10 to 30 kg/cu m (2 to 6 lb/cu ft). This density range is assumed in Table 9-3. It is usually most economical to use multiple layers of 25 mm (1 in.) thick material for the thicker treatments, since the 25 mm thickness is more readily available than single layers of greater thickness. It is recommended that the treatment in subway station platform areas be made up of 50 mm (2 in.) to 100 mm (4 in.) thick absorption material. For platform ceilings and mezzanine areas, 50 mm thickness is adequate. Treatment 25 mm thick is sufficient in other areas of the stations such as entrances, corridors, etc.

The Owens-Corning Fiberglass Company provides a fireproof glass-fiber material denoted TIW, Thermal Insulating Wool (the same material is denoted M-1000 Insulation for marine applications). This is a multipurpose material for industrial applications, useful at temperatures up to 550 degrees C (1000 degrees F). Since this material does not have a binder, its use requires mechanical retention (e.g., a fiberglass cloth bag and metal screen).

A disadvantage of glass-fiber materials, particularly the

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

	Sound Absorption Coefficients				
Frequencies in Hz	125	250	500	1000	2000
25 mm (1 in.) thick Glass Fiber	.08	.30	.65	.80	.85
50 mm (2 in.) thick Glass Fiber	.20	.55	.80	.95	.90
75 mm (3 in.) thick Glass Fiber	.45	.80	.90	.95	.90

nonflammable products, is that a protective or retaining covering and facing is generally required. Some other nonflammable materials -- such as cellular glass blocks -- can be used with no protective covering or facing.

Geocoustic Blocks, made by Pittsburgh Corning Company, are another possibility for general application and particularly for under-platform overhang treatment. These blocks are noncombustible, low density, cellular glass that do not require protective covering or facing. The material is rigid and self-supporting and is typically held in place with an adhesive such as epoxy. The faces of the blocks are slotted to increase sound absorption; the 200 mm (4 in.) thick blocks have excellent sound absorption characteristics. This material generally should not be used in thicknesses less than 50 mm (2 in.). Although transit system experience indicates that they require little maintenance when used in areas not accessible to the public, one significant disadvantage of materials such as Geocoustic Blocks is that they are relatively fragile. As a result, they should not be used in any location subject to mechanical abuse.

It is important to recognize that there is a wide assortment of materials that can be used to increase the sound absorption in transit stations. The choice of the appropriate material is based on the amount of absorption that is required, architectural considerations, ability to withstand the pressure transient loading in stations, resistance to mechanical abuse, safety considerations, such as flame resistance, cost, and other considerations. In most cases, glass-fiber products are the most economical absorption treatment, however, there are many other products that should be considered. More discussion of materials that can be used for acoustical absorption, including spray-on materials, is presented in Section 10.1.2. Another material that can be used is Soundblox, concrete blocks with slots designed to make the hollow core of the block act as a sound absorber. Since they are concrete blocks, they can be used as structural elements in place of standard concrete blocks. By combining structural and acoustical absorption functions, they can be an economical material. Hollow ceramic tile with similar sound absorption mechanism is also available.

9.1.4.2 Facings -- A facing is needed for protection of glass fiber in subway station applications. To prevent accumulation of dust and to permit washing of the facing, glass fiber should be enclosed with a wrapping. Covering slightly decreases the high-frequency absorption of glass fiber and slightly increases the mid- and low-frequency absorption. The net effect is a slight improvement, compared to the bare material, in reducing the overall levels of train noise.

Plastic film as a protective covering cannot always meet fire resistance requirements. A substitute for plastic film covering, which also performs well against water and dust, is close-weave, glass-fiber cloth. Surface tension usually prevents water spray from penetrating the glass-fiber cloth.

9.1.4.3 Recommended Assemblies -- For underplatform overhang treatment, a recommended material assembly is a 75 to 100 mm (3 in. to 4 in.) thickness of nonflammable glass wool with an appropriate nonflammable plastic film or glass-fiber cover of not more than .1 mm (.004 in.) thickness and a facing of expanded metal or hardware cloth. For platform areas and mezzanine ceilings, the design recommended is 50 mm glass wool with appropriate covering, and either perforated sheet metal or slit-and-slat configuration facings. This treatment can be arranged in panels of appropriate size and shape to fit architectural requirements.

Geocoustic Blocks are most usefully applied to underplatform overhangs, in fan and vent shafts, and behind architectural facings. For areas other than platforms and mezzanines, ordinary acoustical tile or panels of 19 mm to 25 mm (3/4 in. to 1 in.) thickness are appropriate. These materials -- which may be of compressed glass wool or other appropriate fire resistant cellular material -- can be of the type with painted or vinyl facing. Also, as for mezzanine areas, panels of glass-wool blankets with perforated metal facing can be used. A consideration when designing treatment for mezzanine areas is that, depending on station configuration and ventilation, they may be subject to dust (including brake dust) from the trains.

9.1.5 Installation Procedures

The expected sound absorption coefficients for glass-fiber treatments are in Table 9-3. The figures in the table are the maximum that should be expected of these materials in a normal, practical installation. They are, in many instances, somewhat less than will be found in the literature. Laboratory tests are performed under ideal conditions that provide the maximum absorption coefficient; it is not realistic to assume that the same absorption will be achieved in a field installation.

For the underplatform edge treatment, an appropriate arrangement is the use of 75 mm to 100 mm mechanically retained glass-fiber material, of 10 to 30 kg/m^3 (2 to 6 lb/cu ft) density, wrapped with close-weave glass cloth. Alternatively, 200 mm thick slotted Geocoustic Blocks are equally effective. The material should be mounted to cover as much of the underplatform area as possible. At stations with significant platform overhangs, absorption material should be placed on the underside of the overhang surface as well as the vertical wall. The minimum treatment for the underplatform area is a .75 m (2.5 ft) width of continuous treatment on the vertical wall.

If glass fiber wrapped in glass cloth is used for the underplatform treatment, the panels should be held in place with either an expanded metal facing, hardware cloth facing, or perforated metal facing. For center platform stations, expanded metal or hardware cloth is the most economical material; it is satisfactory as long as the material is not visible to patrons. For a side platform station, where the material is visible to patrons on the opposite platform, a better appearance can be obtained through the use of perforated metal facing.

Wherever perforated metal or slit-and-slat facings are used, the open area should be at least 20% of the total area. Either expanded or perforated metal facings can be attached to the underplatform surfaces with simple metal brackets. Airspace should be plentiful around the edges to allow free circulation and to prevent air pressure transients, created by the train movements, from loading the acoustical panels. Panels with perforated metal or slit-and-slat facings -- in underplatform, ceiling, and wall installations -- should have a dimpled screen placed between the metal facing and the face of the acoustic blanket. This establishes an airspace of about 12 mm thickness between the perforated facing and the blanket or glass-cloth bag that serves two purposes: it allows the sound waves to diffuse over the entire face of the acoustic material, thereby assuring full efficiency as a sound absorber. It also allows free airflow for pressure equalization, thus helping to prevent loading of the facing by air pressure transients -- especially necessary if high-flow resistance material is used as a cover for the glass wool.

For the ceilings and walls of the train rooms, a number of treatment configurations are available. Table 9-4 indicates some of the basic materials. Materials equivalent to the glass-fiber products in Table 9-4, marketed by companies, such as Johns-Manville and Pittsburgh Plate Glass, should be given equal consideration. The list is intended to be indicative, not exhaustive. The last two materials listed in Table 9-4 are appropriate only where flammable materials are acceptable.

Sectioned or continuous panels (consisting of either a metal or plastic slit-and-slat system or a perforated metal facing) with fiberglass or cellular glass blocks between the facing and the concrete surface are appropriate for treating flat, continuous

TABLE 9-4 SOUND ABSORPTION MATERIALS FOR CONSIDERATION AS ACOUSTICAL ABSORPTION TREATMENT IN TRANSIT STATIONS

Material	Approximate Sound Absorption Coefficients With Rigid Backing	
	250 Hz	500 Hz
100 mm Thick Geocoustic Blocks, .3 m x .45 m, *Slotted:*		
Unspaced	1.0	1.06
Spaced 50 mm, both directions	0.90	1.06
Spaced 150 mm, both directions	0.60	0.66
100 mm Thick Geocoustic Blocks, .3 m x .45 m, *Perforated:*		
Unspaced	0.79	0.84
Spaced 50 mm, both directions	0.82	0.94
Spaced 150 mm, both directions	0.53	0.59
50 mm Thick Geocoustic Blocks, .3 m x .45 m, *Perforated:*		
Unspaced	0.79	0.73
Spaced 50 mm, both directions	0.74	0.71
Spaced 200 mm, both directions	0.42	0.60
50 mm Thick Plain Glass wool of 2 to 6 lb/cu ft density wrapped with glass cloth	0.60	0.80
50 mm Thick Owens-Corning Aeroflex Duct Liner (3 lb/cu ft density) or Type 702 Blanket faced with a vinyl or neoprene coating	0.55	0.80
50 mm Thick Owens-Corning Glass Cloth Faced Boards backed with Type 703, 704, or 705 Board	0.55	0.85

surfaces and platform or mezzanine ceiling areas. However, if a continuous panel system or a suspended acoustical tile ceiling is used, it must have gaps or openings to permit free airflow between the acoustical treatment panels and the concrete surface behind. If pressure equalization provisions are not provided, the loading due to air pressure transients can eventually cause fatigue failure of the fastenings, allowing the panels to come loose from the mounting surface and fall.

Note that several combinations of spaced and unspaced Geocoustic Blocks are listed. The absorption coefficients for the spaced configurations are based on the gross area of the treatment, i.e., the block area plus the area of the spaces between blocks. Use of spaced configurations can result in material economy, but to avoid loss of low-frequency absorption, the 100 mm (4 in.) thick units should be spaced not more than 150 mm (6 in.) and the 50 mm (2 in.) thick units not more than 100 mm (4 in.) apart. For low cost and nonflammability, Geocoustic Blocks should be specified unpainted and without surface coating or wrapping.

A suitable covering for any side wall treatment is perforated sheet metal with at least 30% open area. Perforation patterns, such as 1/16 in. diameter holes staggered at 7/64 in. centers, 1/8 in. diameter holes at 3/16 in. centers, or 3/16 in. diameter holes at 5/16 in. centers, provide adequate open area. There are, of course, other combinations of equivalent performance.

A basic panel system for ceilings and walls in the mezzanine and corridor areas can provide acoustical absorption very simply. The panel may be of perforated metal, a slit-and-slat configuration of boards or metal, or some form of architectural trim. If the latter, it should have at least 20% open area and no bars or sections greater than 50 mm (2 in.) width between the openings. Such an arrangement will provide a completely transparent acoustical face. Acoustical material can then be located 12 mm to 150 mm (1/2 in. to 6 in.) behind the face; cellular glass blocks or nonflammable glass wool of 50 mm (2 in.) thickness would be suitable materials.

In corridors and entrances, the sound absorption treatment can be the same as that described above for platforms and mezzanines, or it can consist of an application of 19 mm to 25 mm (3/4 to 1 in.) thick acoustical tile, acoustical ceiling board, or cellular glass blocks. Another choice might be a sound absorption assembly, such as perforated sheet metal with fiberglass blankets behind the sheet metal facing. The absorption coefficient should be at least the value listed in Table 9-2 for each type of space, taking the type of mounting into account.

In ancillary spaces, two basic types of material are appropriate. For spaces with equipment which radiates relatively low noise levels or in which the noise is intermittent (such as in switchgear rooms or shops), the recommended acoustical treatment is a 25 mm (1 in.) thick glass-fiber application. An alternative could be the use of 19 mm to 25 mm (3/4 to 1 in.) thick acoustical tile, acoustical ceiling board, or painted duct liner board for the absorption material. In spaces with noisy equipment, such as fans, pumps, and chillers, the acoustical treatment should have a minimum thickness of 50 mm (2 in.). In such spaces, the material need not have an architectural trim facing. Application of 50 mm thick (two layers of 25 mm thickness) duct liner blanket to the walls and ceiling, perhaps with hardware cloth facing for mechanical protection, gives an economical and effective sound absorption treatment with appropriate absorption characteristics.

In ancillary spaces housing noisier equipment, the treatment requires 30% of the total wall area and 50% of the ceiling area. The sound absorption material must be distributed reasonably uniformly over the ceiling, in panels or patches, and the wall material must be distributed over at least two adjacent walls. That is, the material should not be concentrated on one part of the ceiling or concentrated on two opposite walls, but distributed between the ceiling and walls, with the wall treatment located to give an approximately equal division of area between walls located at right angles to each other.

9.2 CONTROL OF STATION NOISE DUE TO ADJACENT HIGHWAYS OR RAILROADS

It is relatively common for surface stations to be located so that the platform area is exposed to noise from traffic or railroads. The noise created by highways and railroads is very different -- railroads create intermittent high noise levels, while traffic noise is more continuous -- however, the methods of controlling the noise are basically the same. When the basic station design is complete, the principal means by which noise from highways or railroads can be reduced is to place a barrier that obstructs the direct path from the source to the receiver. In partially enclosed areas, absorption treatment should also be considered to control the buildup of reverberant sound.

The remainder of this section concentrates on the control of highway noise, a more common problem than railroad noise. However, most of the discussion of designing barriers to control highway noise is equally applicable to railroad noise.

9.2.1 Criteria for Traffic and Railroad Noise on Platforms

Transit system facilities and motor vehicle highways or railway systems are often located along the same right-of-way. On the platform of above-grade stations adjacent to freeways, the traffic noise, unless controlled, can produce an uncomfortable environment, difficulty in conversation between patrons, and a loss in the intelligibility of announcements from the public address system. While trains on adjacent tracks are passing, train noise can also generate similar detrimental effects.

Because railroad noise has much the same character as transit train noise, the same criterion can be applied. Following the APTA Guideline, the criterion for maximum level for railroad noise should be 85 dBA. In contrast, noise due to freely flowing traffic on a highway is a variable quantity, depending on the traffic flow rate, the speed and type of vehicles,and the number of traffic lanes. Because of the variability of traffic noise, statistical analysis techniques are needed for analyzing and describing traffic noise.

The most widely accepted analysis uses the statistical distribution of A-weighted sound level in decibels. Traffic noise is generally analyzed in terms of the 10, 50, and 90 percentile levels, abbreviated L_{10}, L_{50}, and L_{90}, or in terms of the energy equivalent sound level, L_{eq}. At this time, L_{10} and L_{eq} are generally accepted as the best indicies for rating traffic noise. The noise level which is exceeded 10% of the time, L_{10}, turns out to be approximately equal to the average values of the peak noise levels occurring as vehicles pass by an observation point.

Appropriate maximum levels of traffic noise on platforms can be determined from the sound levels already deemed appropriate for planning and design in residential areas near highways, the need for speech communication between patrons, and the need for intelligibility of public address system announcements. The most commonly used criterion for traffic noise in residential areas is that the maximum value of L_{10} over any hour of the day not exceed 70 dBA or that the maximum hourly L_{eq} not exceed 68 dBA. With these noise levels, speech communication with normal speaking voices can be achieved with a separation distance of 3 to 4.5 m (9 to 14 ft). The conclusion is that maintaining the peak hour L_{10} due to traffic noise below 70 to 75 dBA will maintain a comfortable acoustic environment for the patrons. In terms of L_{eq}, the peak hour level should be a maximum of 68 to 73 dBA.

9.2.2 General Discussion of Traffic Noise Levels at Station Platforms

This section presents the noise levels for freeway median station platforms, based on measurements and observations of actual freeway traffic noise performed at existing freeways and at BART system stations located in the median strip of highway systems. The noise on station platforms was recorded for later statistical analysis, and the passing traffic was counted to determine the flow rate. The data obtained were checked with the predictions made according to methods given by the National Physical Laboratory reports and the Highway Research Board Report #174 and found to be consistent with the levels expected. Using the highway noise prediction rules and charts, with small adjustments necessary for correspondence to freeway median situations, the data obtained have been generalized to provide a means for estimating noise levels at any station platform near highway traffic lanes.

Figure 9.7 presents L_{10} traffic noise levels measured at the BART Rockridge Station at mid-day, when the traffic flow was about 2900 vehicles per hour (vph), and at an evening rush hour time, when the flow was about 5100 vph. For these traffic flow rates, the L_{10} levels were 78 to 81 dBA and 81 to 84 dBA, for the open platform areas and the covered platform areas respectively. These levels are 3 to 9 dBA greater than the criterion for stations, or in other words, roughly twice as loud as they should be.

The traffic census indicated that about 4.5% of the flow on the route adjacent to the Rockridge Station consisted of trucks and buses. Figure 9.7 also shows a cross-section drawing of the Rockridge Station, showing the configuration and its relation to the highway traffic lanes. There are no noise reduction elements included along the platform, and there is no sound-absorbing treatment on the underside of the canopy roof. The curbs on the highway structure are solid, but are not high enough to block the line-of-sight between persons on the platform and the vehicles, even for the lanes nearest the railing. These walls, therefore, do not function as sound barriers and do not reduce the noise at the station platform.

One of the factors in station design demonstrated by noise data at the Rockridge Station is the role that the canopy, or roof with open walls, plays in causing increased noise levels, especially when there is no sound absorption material on the underside of the canopy. A complete tabulation of the traffic noise levels at the BART Rockridge Station is given in Table 9-5. These data show that the nonabsorptive roof covering has little effect on background noise; L_{90} is only 1 dBA higher than under the roof. However the reflections from the roof increase the average noise level, L_{50} by 2 dBA and L_{10} by 3 dBA. Apparently, any canopy or roof over a platform area exposed to traffic noise via a direct line-of-sight must be considered in determining noise levels that will be experienced by patrons on the platform.

One of the major parameters affecting the level of traffic noise at the station platform areas is the number of vehicles using the highway. To predict the anticipated noise levels at stations accurately, it is necessary to determine the traffic noise levels as a function of traffic flow rate, vehicle speed, vehicle mix, and distances from the traffic lanes. Complex formulae and charts have been developed to predict levels for a wide range of situations, but under general conditions at freeway median station platforms, it is possible to summarize the information on one chart for typical traffic conditions and then apply correction factors for other conditions.

Figure 9.8 presents the expected traffic noise level as a function of distance from the edge of the nearest traffic lane for freely flowing traffic with a flow rate of 5000 vph, average speed of 96 kph (60 mph), and including about 5% trucks. The chart shows L_{90}, L_{50}, L_{10} and L_1 to permit general use, but the curve for

TABLE 9-5 TRAFFIC NOISE LEVELS AT THE BART ROCKRIDGE STATION

Traffic Flow Rate	Noise Levels in dBA			
	Open Platform Area		Center of Covered Platform Area	
	2900 vph	5100 vph	2900 vph	5100 vph
L_1	82	85	85	87
L_{10}	78	81	81	84
L_{50}	74	77	76	79
L_{90}	70	74	71	75

TABLE 9-6 CORRECTIONS TO L_{10} FOR DIFFERENT PERCENTAGES OF HEAVY VEHICLES

% Trucks	ΔL_{10}
0	-1 dBA
5	0
10	+1
20	+2

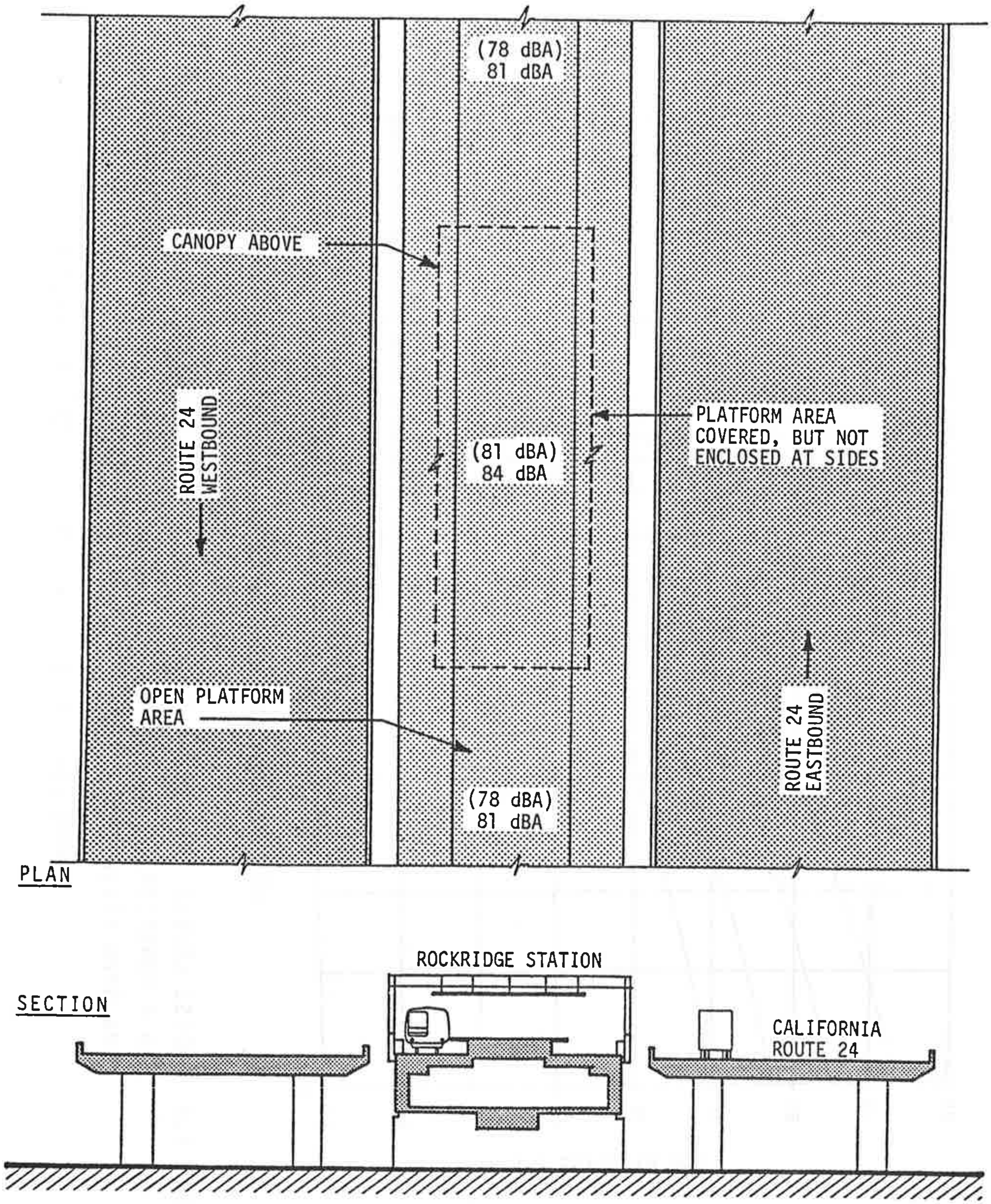


FIGURE 9.7 MEASURED FREEWAY TRAFFIC L_{10} NOISE LEVELS AT BART ROCKRIDGE STATION. TRAFFIC FLOW RATES OF (2900 VPH) AND 5100 VPH.

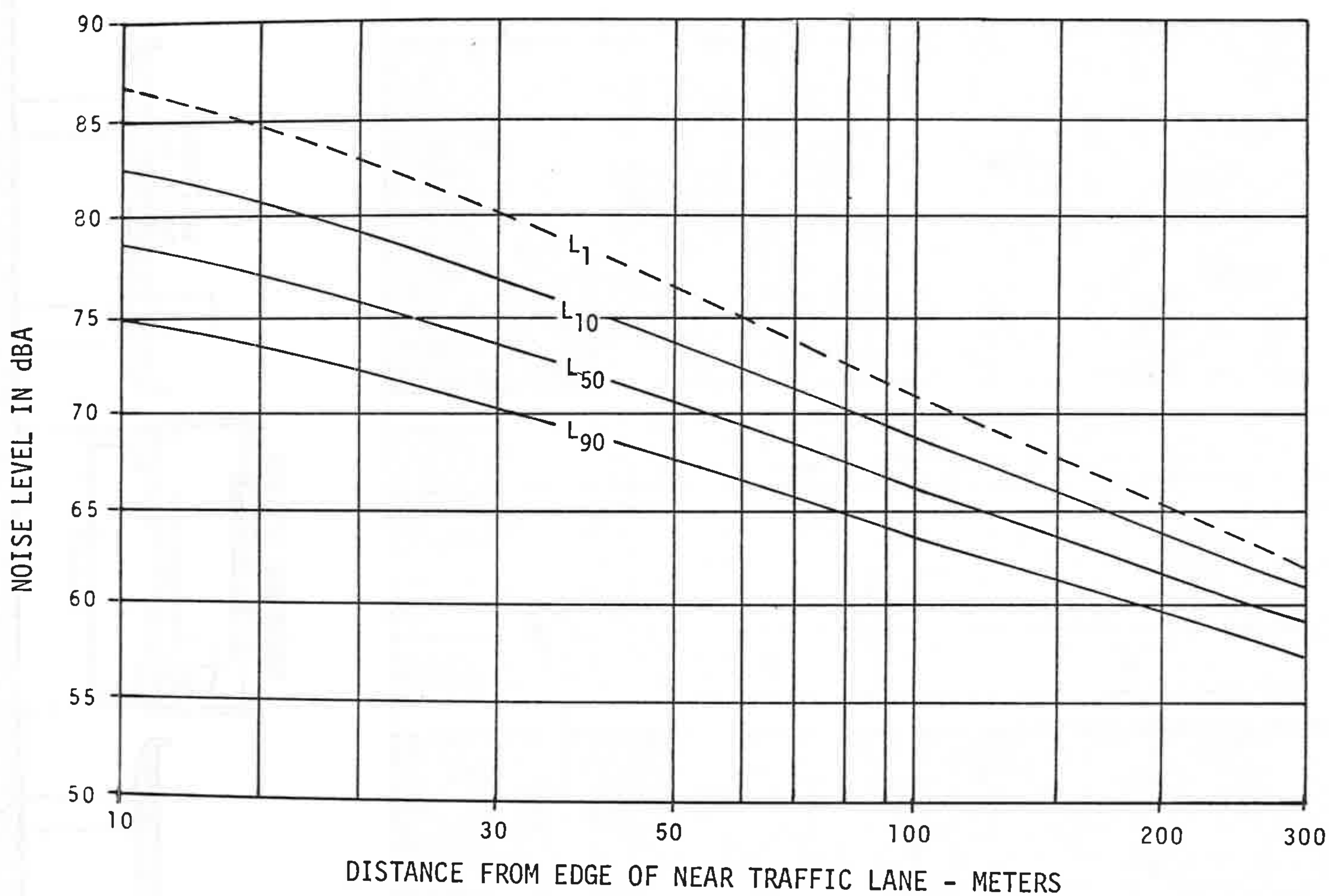


FIGURE 9.8 NOISE LEVELS FOR FREELY FLOWING TRAFFIC ON A MULTILANE DIVIDED HIGHWAY AS A FUNCTION OF DISTANCE FROM THE EDGE OF THE NEAREST TRAFFIC LANE FOR OPEN LEVEL TERRAIN

5000 VPH - 95 km/hr AVERAGE SPEED - 5% TRUCKS

L_{10} is most relevant to freeway median stations. Greater traffic flow rate or speed will give higher levels, and lesser flow rate or lower speed will give lower levels. For distances to about 60 m (200 ft) from the near traffic lane, some simple general rules can be applied to determine adjustments of the values shown in Figure 9.8 for different flow rates, speeds, and percentage of trucks. Examination of the chart shows that at distances greater than 45 to 60 m (150 to 200 ft), the L_{10} level is less than 75 dBA; the noise is less than the criterion demanded of station platforms, making noise-reduction treatment unnecessary under these conditions.

For distances in the range of 9 to 60 m from the near traffic lane, with flow rates of 2000 to 15,000 vph and average speeds of 60 to 110 kph, the curve for L_{10} as a function of distance in Figure 9.8 can be adjusted up or down. For flow rates other than 5000 vph and speeds other than 88 kph (55 mph):

$$L_{10} = 16 \log (V/95) \quad \text{for speed correction;}$$

$$L_{10} = 8 \log (Q/5000) \quad \text{for flow rate correction;}$$

where V is the average vehicle speed in kph and Q is the traffic flow rate in vph. These relations are equivalent to a change of 4.8 dBA per doubling of speed and 2.4 dBA per doubling of flow rate. The changes for L_{50} are somewhat different: about 4.0 dBA per doubling of speed and about 3.0 dBA for doubling of flow rate.

For different percentages of heavy vehicles, the approximate corrections to L_{10} are given in Table 9.6.

With these relationships and the curve for L_{10} in Figure 9.8, it is possible to estimate the expected noise levels on open platform areas of any station located near highway or high-speed boulevard traffic lanes.

Figure 9.9 indicates the typical frequency spectrum of freely flowing traffic noise. This range of octave band levels can be used to derive the predicted effectiveness of various heights and configurations of sound barrier walls in reducing the platform noise levels. Since barrier walls are not equally effective for all frequencies and for all traffic noise sources, it is necessary to consider the frequency spectrum of the noise in determining the effectiveness of a sound barrier wall.

9.2.3 Noise Reduction with Sound Barrier Walls

An effective and economical method for reducing the wayside noise from street and highway traffic is to interpose a sound barrier wall between the traffic lanes and the observation position. In the absence of an enclosure, the sound barrier wall is, in fact, the only means available for reducing the traffic noise, unless the noise at the source can be controlled. Sound barrier walls between traffic and the station platforms constructed using a relatively inexpensive wall construction can provide noise reductions of 5 to 15 dBA. In many cases, the barrier wall can simply be an extension of a curb, retaining wall, or median barrier already required in the overall assembly. For freeway median stations, the sound barrier wall can be an extension of the grade beam, retaining wall, or traffic barrier which is usually necessary at the outer edge of a station structure.

To be most effective, a sound barrier wall should be placed close to either the sound source or the sound receiver. Because freeway traffic lanes are wide and the noise sources are spread out over a large area, the best results are obtained by locating the sound barrier wall as close as possible to the station platform. The optimum location is on the outer edges of the station structure corresponding with the retaining wall, grade beam, or traffic barrier on the outside edge of the track structure. Another possible location for the sound barrier wall would be at a median barrier installed as part of the highway structure. However, such a barrier may be 2 to 3 m farther away from the station platform

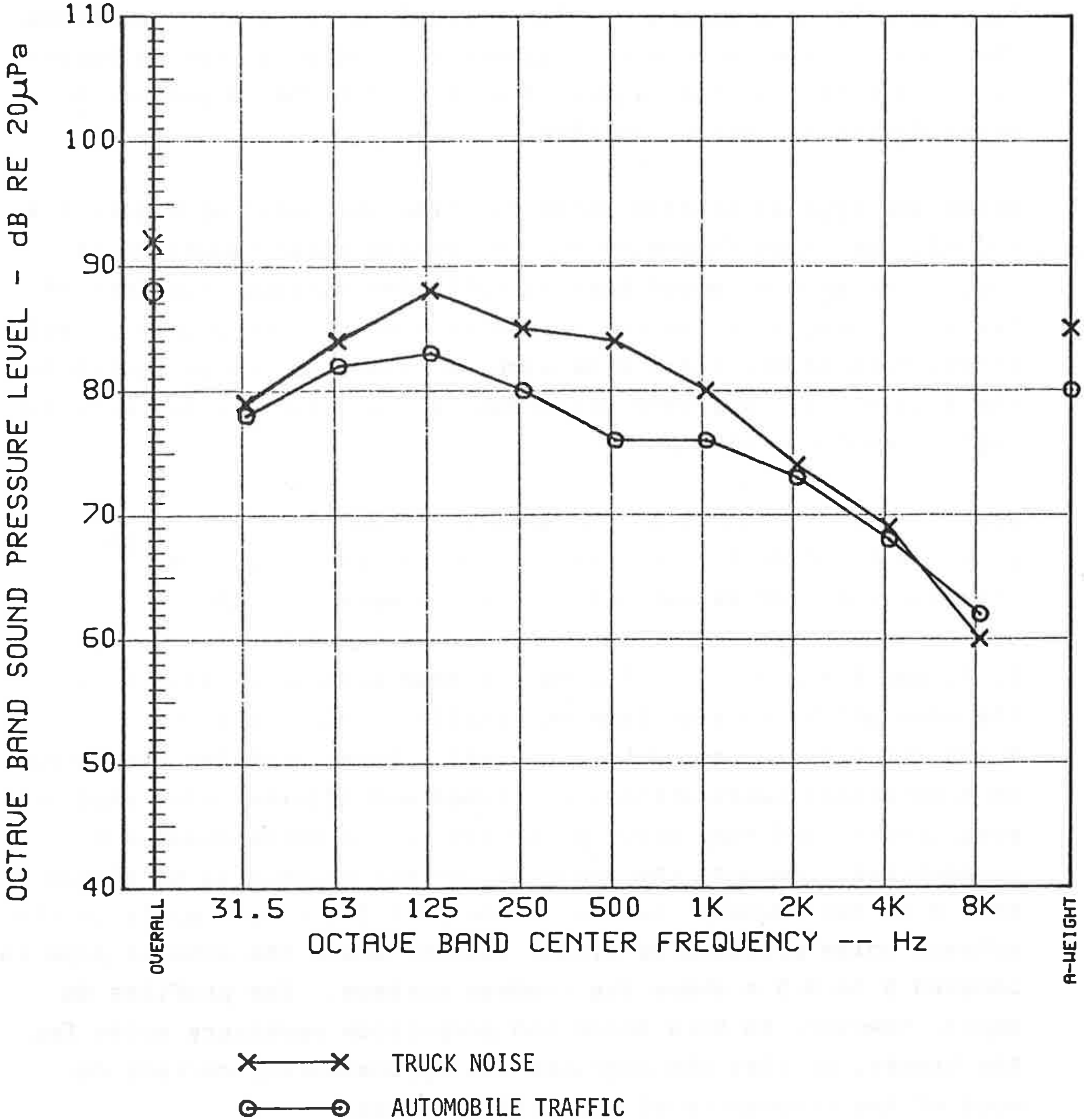


FIGURE 9.9 TYPICAL OCTAVE BAND LEVELS FOR FREEWAY TRAFFIC, 10 TO 25 m FROM EDGE OF THE ROADWAY

except where freeway traffic lanes are placed directly adjacent to the transit system structure. Therefore, its position is less favorable for minimizing the noise level on the station platform. The general rule is that the farther the sound barrier is located from the platform, the higher it will need to be to produce a given degree of noise reduction.

Using the typical traffic noise spectrum indicated by Figure 9.9, calculations have determined the effective noise reduction of various heights of sound barrier walls for various locations of traffic lanes, relative to the position of the sound barrier wall. Discussions of the results of such calculations are presented in the literature. For this presentation, calculations are made for typical median stations.

Figures 9.10 through 9.13 indicate the reduction in the L_{10} provided by sound barrier wall positions and locations characteristic of situations at freeway median stations. The charts show noise-reduction profiles for sound barrier walls of 1, 2, 3, and 4 m high, relative to the road surface elevation, where the edge of the nearest lane of traffic is 6.1 m (20 ft) and 10.7 m (35 ft) from the sound barrier wall. These profiles are based on theoretical calculations, confirmed and adjusted with empirical meaurements, and they apply primarily to the noise created by automobiles. Because the noise source was assumed to be within .5 to 1 m of the roadway surface, these profiles do not apply to the exhaust noise produced by diesel trucks, where the exhaust pipe is located 3 to 3.5 m above the roadway surface. The profiles do apply, however, to tire noise and propulsion machinery noise for the trucks, so that the profiles are approximately correct for most of the components of the traffic noise.

In Figures 9.10 through 9.13, the noise-reduction profiles indicate the approximate noise reduction to be expected from a sound barrier wall for any elevation of the platform relative to the elevation of the roadway. It is immediately apparent that higher elevations of the platform result in greater noise levels

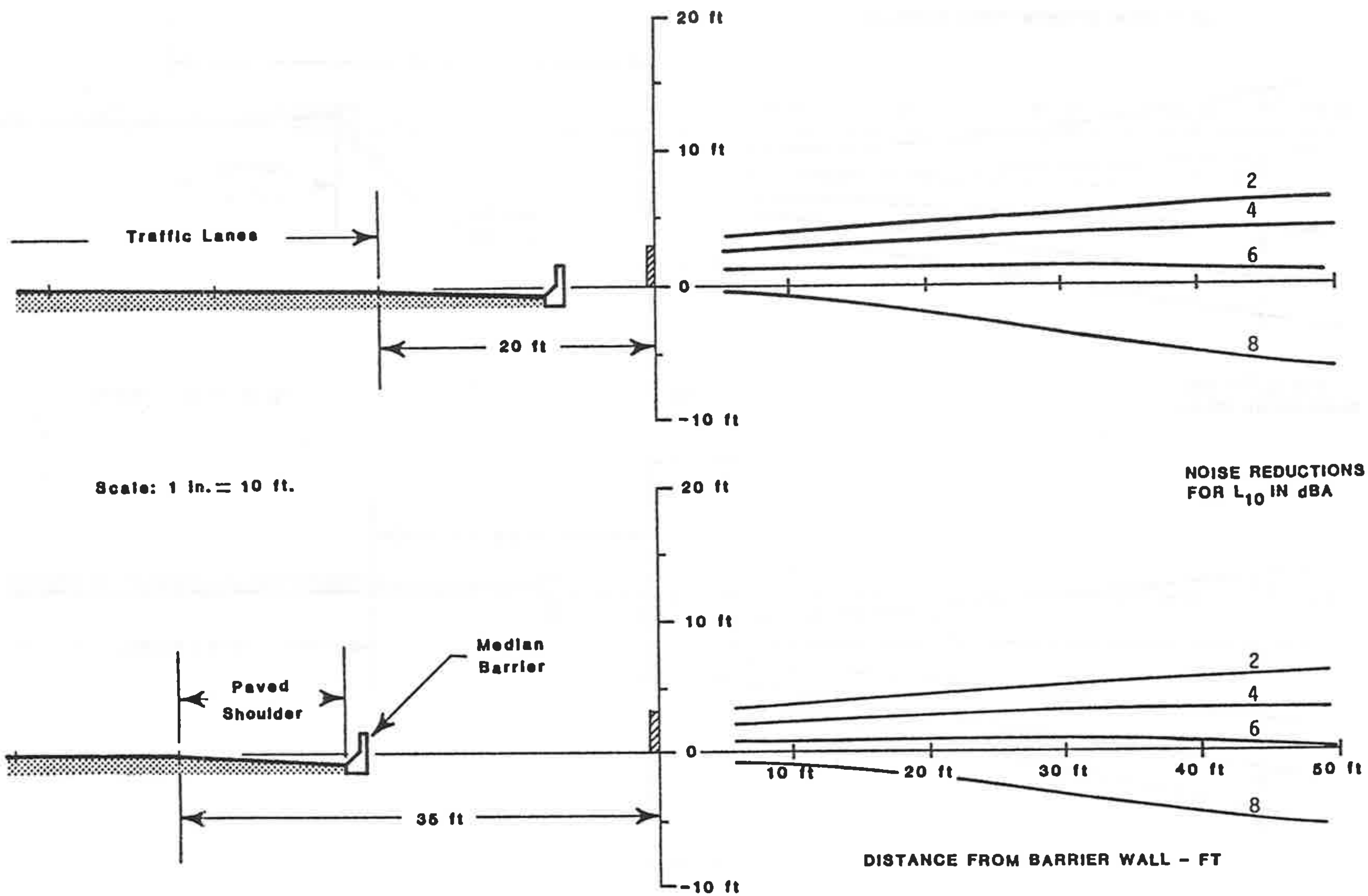


FIGURE 9.10 NOISE REDUCTION CONTOURS FOR A 1 m (3.3 FT) HIGH SOUND BARRIER WALL

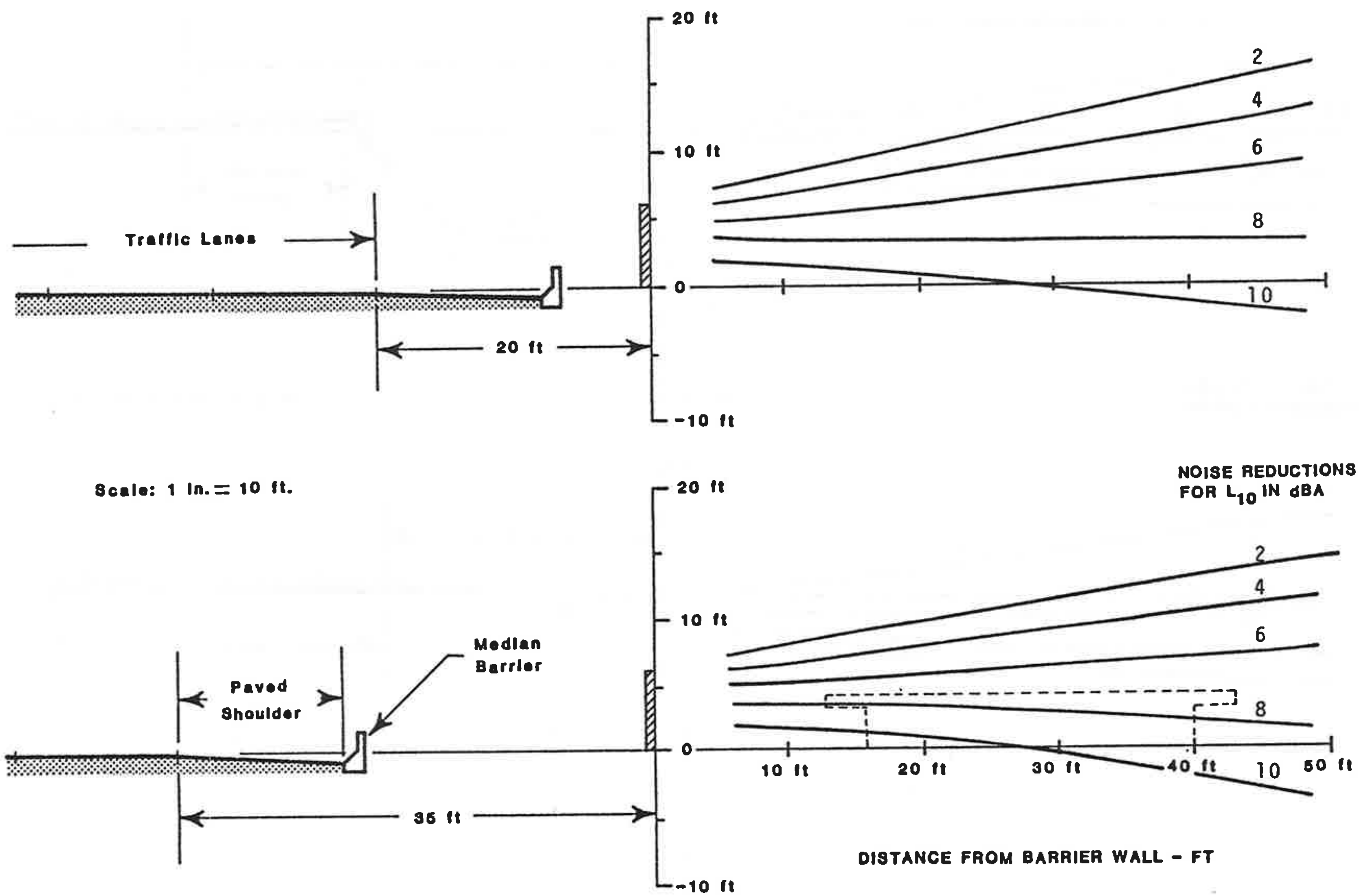


FIGURE 9.11 NOISE REDUCTION CONTOURS FOR A 2 m (6.6 FT) HIGH SOUND BARRIER WALL

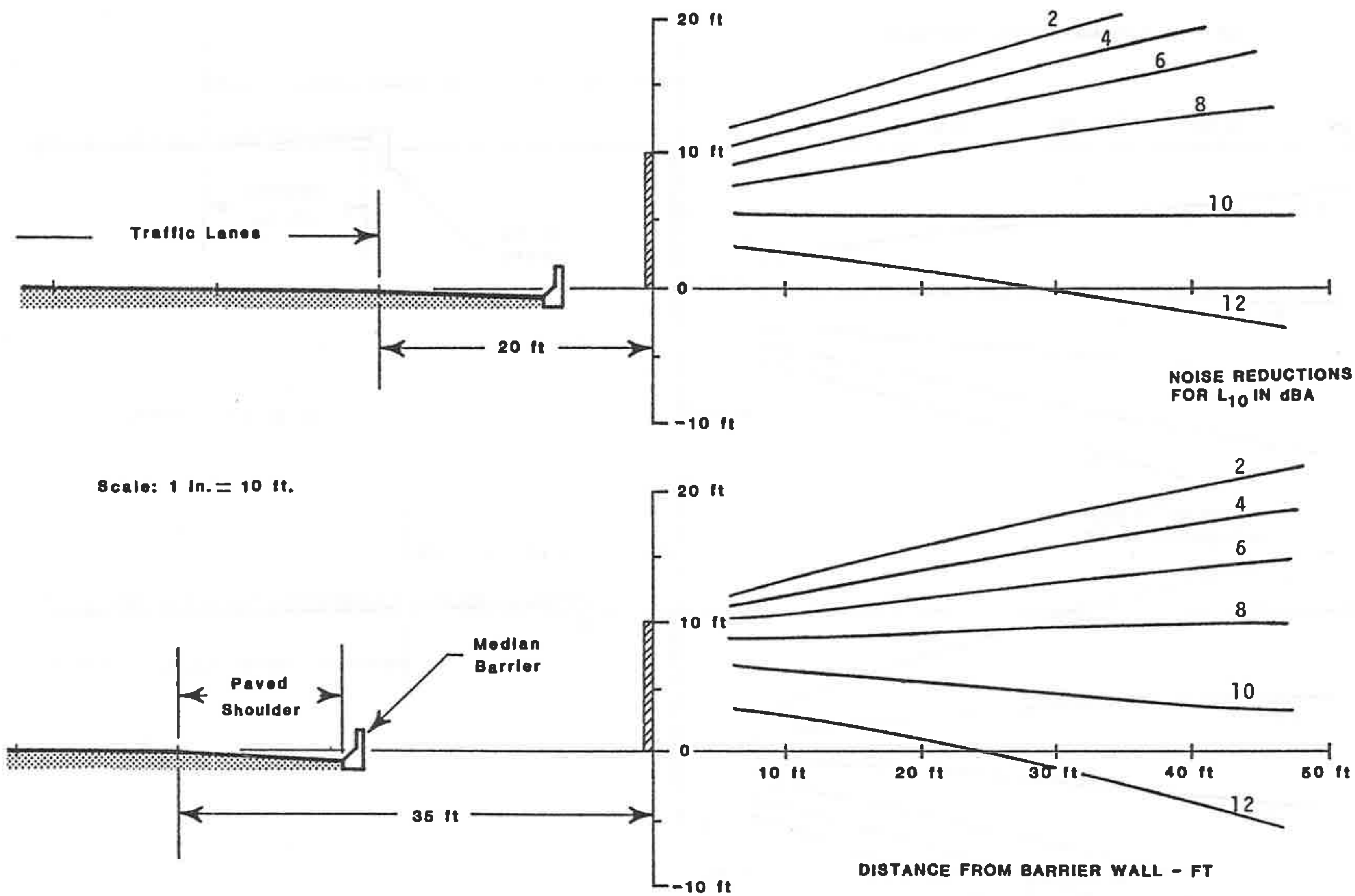


FIGURE 9.12 NOISE REDUCTION CONTOURS FOR A 3 m (10 FT) HIGH SOUND BARRIER WALL

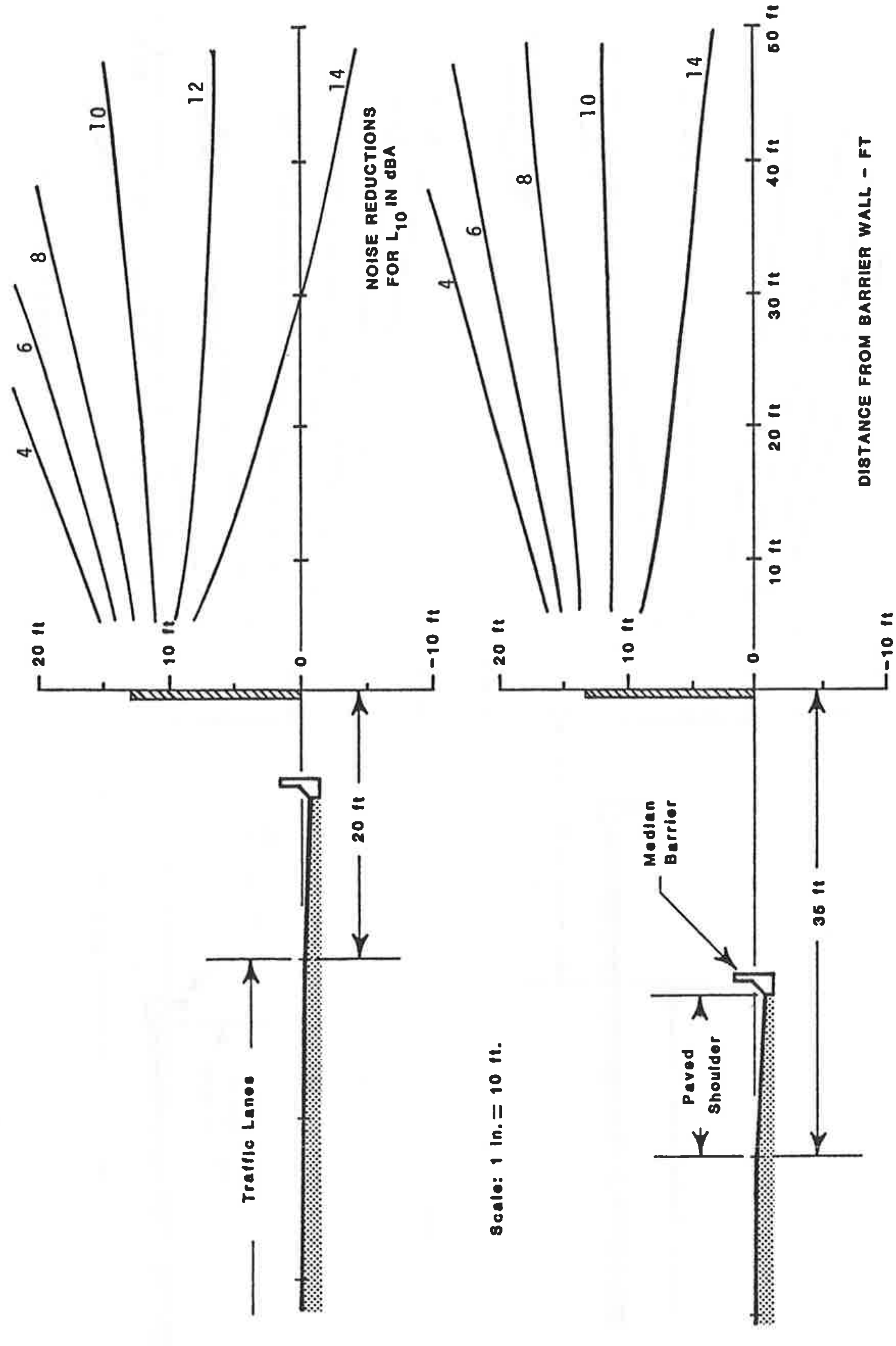


FIGURE 9.13 NOISE REDUCTION CONTOURS FOR A 4 m (13 FT) HIGH SOUND BARRIER WALL

than platform elevations equal to or lower than the elevation of the roadway. In other words, the higher the elevation of the platform, with respect to the roadway, the higher must be the elevation of the top of the sound barrier wall in order to provide a given degree of noise reduction (or a given maximum noise level). Where the platform is at a high elevation and directly adjacent to the freeway, as is possible in an outside platform design, the platform itself forms part of the shielding barrier.

In Figure 9.11, the expected reduction from a sound barrier wall of a given height can be determined by placing a cross-sectional drawing of the station platform (as shown by the dotted line) on the noise-reduction profile drawing at the appropriate elevation and location. The expected noise reduction from the combination should then be determined by finding the profile line about 5 ft above the surface of the platform. This will give the expected noise reduction, compared to the situation without a sound barrier wall, for an open platform with no covering roof.

One important point in the use of sound barrier walls and noise-reduction profiles is that the calculations and profiles apply for a continuous sound barrier wall, that is, for a sound barrier wall extending a significant distance beyond each side of the observation location. Because of architectural and economic reasons, sound barrier walls for freeway median stations are desirably terminated either at the ends of the platforms or at some other location past the end of the platform, within the station or beyond it. Regardless of economic or architectural considerations, the acoustic barrier wall must extend beyond the platform end in order to reduce noise on all locations of the platform. An observer located near the end of a sound barrier wall will be exposed to traffic noise which propagates around the end of the wall. Figure 9.14 presents a plan view drawing, indicating the typical situation for freeway median stations. It also shows a chart indicating the decrease in performance of the sound barrier walls as a function of the position of the observer

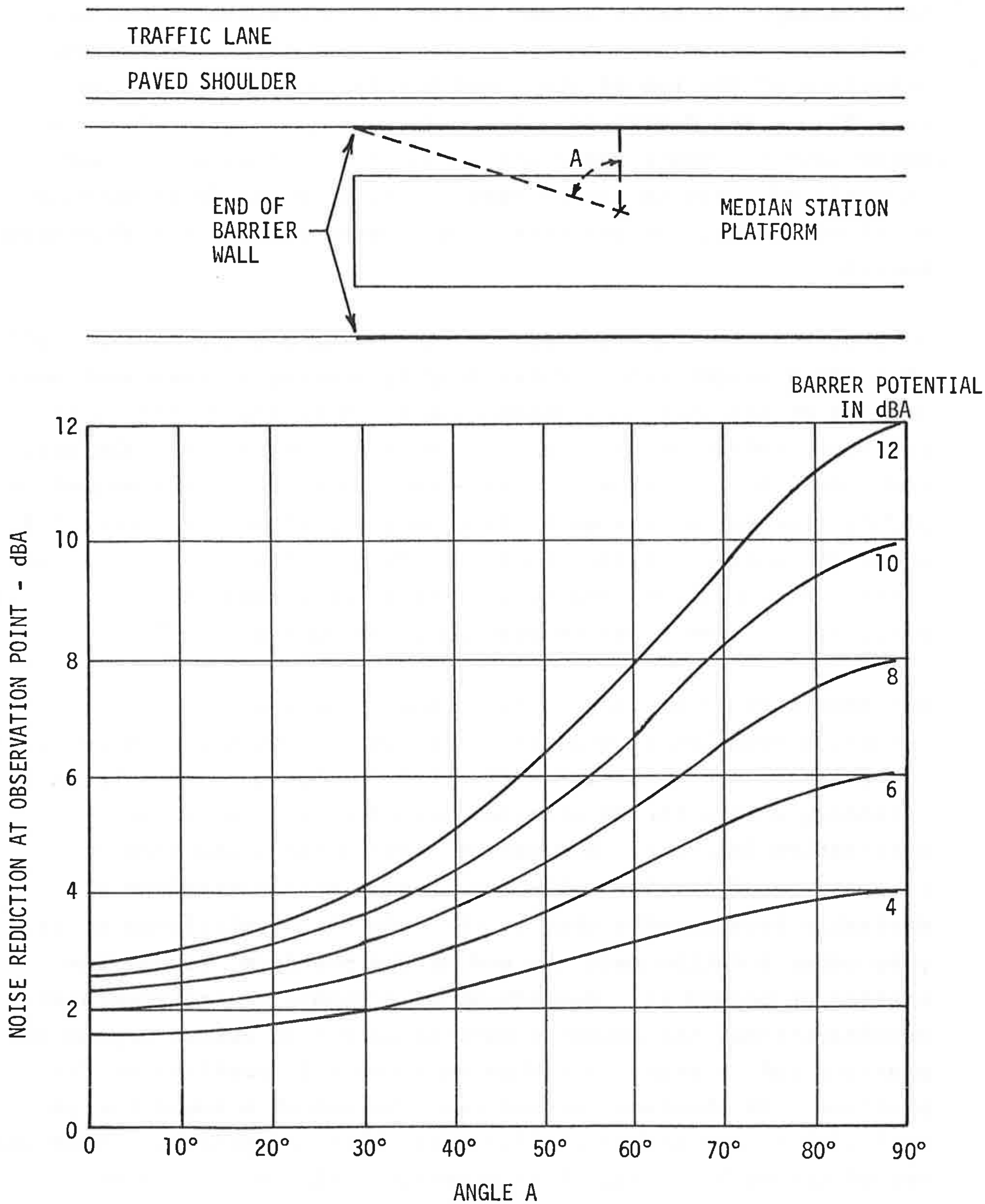


FIGURE 9.14 CHART SHOWING THE REDUCTION IN EFFECTIVENESS OF A SOUND BARRIER WALL DUE TO SOUND TRANSMITTED AROUND ONE END OF THE WALL

relative to the end of the wall. The factor used to determine the decrease in effectiveness is the angle A, measured between the line from the observer to the end of the barrier and the perpendicular line from the observer location to the barrier.

The typical geometry of a transit system station requires that the sound barrier wall should extend about 30 m beyond the end of the platform. An angle of 70 degrees to 80 degrees between the observer (at the end of the platform) and the end of the barrier wall, which gives nearly the full potential of the barrier, is thus obtained; this dimension can therefore stand as the needed extra length of barrier to control noise at the end of the platform.

As an example, consider the situation of a sound barrier wall with a potential 10 dBA noise reduction if the wall is very long. Such a sound barrier wall, if terminated at the ends of the platform, will provide about 3 dBA noise reduction at the end of the platform and about 8 dBA noise reduction 30 m from the end of the platform; it will only reach its full potential in the middle of the platform area, 90 m (or more) from the end.

9.2.4 Traffic Noise Reduction Without Sound Barrier Walls

If for nonacoustical reasons the sound barrier walls as discussed in Section 9.2.3 cannot be implemented, then acoustically treating the platform roofs of above-grade stations will help reduce the noise from traffic on adjacent highways and roads.

Platform noise levels may be increased by up to 3 dBA due to reflection of the highway noise down to the platform by the roof of the station. The platform roofs of above-grade stations should be acoustically treated to minimize these reflections. Additionally, it is advisable to shape the ceiling carefully so that noise interception and reflection downward are minimized. This absorption treatment is not necessary for reduction of the transit train noise, and in the absence of the noise from traffic on the roadway, trains entering and leaving will meet the noise design goals for above-grade stations without acoustical absorption material. The treatment area required is 50% to 60% of the ceiling or the extent allowed by architectural limitations. To minimize the reflections of the traffic noise to the platform area, the sound absorption material should mostly be located directly over the transit train tracks and the outer edges of the platform.

9.3 Ancillary Equipment Noise Control in Stations

The ancillary equipment that may generate noise in rapid transit stations includes air-conditioning system equipment, ventilation fans, elevators, escalators, and transformers. The acoustical criterion for noise from ancillary equipment is 55 dBA for platform or mezzanine areas, as indicated in Table 9-1.

Chapter 35 of the ASHRAE Handbook (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1976 Systems), entitled "Sound and Vibration Control," presents a detailed discussion and design procedure for noise and vibration control on air-conditioning noise and the design of acoustical control features. It is recommended that anyone designing noise control for station ancillary equipment use this reference as a general guide.

Ductborne transmission is the main source of noise intrusion in public areas from ventilation equipment. However, structureborne and airborne sound also emanate from this equipment. It is therefore necessary to provide adequate noise and vibration control by physically isolating this equipment from public spaces. Proper selection of equipment, possibly including the use of vibration isolation for fans, pumps, compressors, and connecting elements, and the provision of sound attenuators and lined ducts, is also necessary.

Controlling noise from ventilation fans in subway stations usually involves the use of sound absorbent materials in the fan room, attenuators of the fan intakes and/or discharges, and sound-insulating doors at the machinery rooms. Chapter 10.1 of this handbook presents a complete discussion of fan noise transmission in stations and the results of typical acoustical measurements.

Noise from escalators and elevators does not usually constitute a serious acoustical problem in stations. However, equipment purchase documents should include appropriate specifications to ensure that the noise is comparable to typical background noise in the stations.

Transformers are generally installed in separate rooms away from public areas; the humming noise they produce is, most of the time, at a reasonably low level. By providing proper sound isolation at the transformer room doors (e.g., by using metal doors with weatherstripping), transformer noise is usually adequately controlled.

CHAPTER 9 REFERENCES

9.1 M. Rettinger, "Acoustic Design," Vol. I; "Noise Control," Vol. II, - Acoustic Design and Noise Control, (Chemical Publishing Company, NY, 1977).

One of many good texts on architectural acoustics and noise control. In these volumes, the author covers the physics of sound, room acoustics and design, and noise and its reduction.

9.2 G. P. Wilson, "Sound Absorption System for Metro Tunnels and Station Platforms," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, June 1971).

This report discusses the design considerations for acoustical treatment to control noise in the WMATA Metro subway stations and tunnels. Specific design guidelines for the type and placement of acoustical material are given.

9.3 G. P. Wilson, "At-Grade and Aerial Stations in Highway Medians," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, February 1973).

The purpose of this report is to present specific design recommendations for control of traffic noise on transit station platforms. The recommendations are largely based on measurements performed at the BART Rockridge Station.

9.4 G. P. Wilson, "Sound Control in Stations, Running Tunnels, and Shafts for the Melbourne Underground Railway Loop," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Melbourne Underground Railway Loop Authority, November 1974).

Presented are general discussions of sound control treatments for MURLA stations, tunnels, and vent shafts, along with specific recommendations for the treatments to be used on MURLA facilities.

9.5 G. P. Wilson, H. J. Saurenman, "Acoustical Treatment for Sound Control in Stations," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Niagara Frontier Transportation Authority Metro Construction Division, November 1978).

The use of acoustical absorption treatment to control noise in the NFTA stations is discussed. Specific recommendations for the treatment of underground stations are included. A significant conclusion is that treatment is not required for the platform areas in surface stations.

9.6 G. P. Wilson, "Rail Transit System Noise Control," Noise-Con 77 Proceedings, (1977 National Conference on Noise Control Engineering, NASA Langley Research Center, Hampton, VA, October 1977).

This paper summarizes the progress that has been made in the control of rail transit noise. Included is a discussion of the effectiveness of acoustical treatment in various types of subway stations.

9.7 M. E. Delany, "A Practical Scheme for Predicting Noise Levels (L_{10}) Arising From Road Traffic," Acoustics Report, AC 57, (National Physical Laboratory, Department of Trade and Industry, England, July 1972).

A simple, relatively accurate scheme is presented for predicting values of L_{10} out to 120 m from freely flowing traffic on straight, level roads without the need for calculation.

9.8 "Highway Noise: A Design Guide for Prediction and Control," NCHRP 174, (Transportation Research Board, National Research Council, Washington, D.C., 1976).

One of a series of National Cooperative Highway Research Program reports on prediction and control of highway noise.

9.9 "Sound and Vibration Control," ASHRAE Handbook & Products Directory; 1980 Systems Handbook, Chap. 35, (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., NY, 1980 - Updated Periodically).

See Reference 10.1 for annotation. Note that the updated versions sometimes leave out or condense important information. Hence, the older versions remain valuable references.

CHAPTER 10

10. ANCILLARY EQUIPMENT NOISE

Noise-producing ancillary equipment is often located in areas sensitive to noise, which may cause complaints from the community. Some ancillary equipment can also be irritating to patrons in subway stations. This chapter focuses on the major sources of transit system ancillary equipment noise and suggests some methods for control. Ancillary equipment can be located in very sensitive areas without significant community annoyance, if the sites selected are a reasonable distance from nearby buildings and suitable control measures are applied. Vibration from ancillary equipment is occasionally felt inside stations, but only very rarely is it a problem.

Designing noise control features for ancillary equipment requires that the maximum sound level ratings be clearly specified. Specifications applicable to various types of equipment are discussed in the following sections. The advantages of specifying quiet equipment should be emphasized; it may be more expensive to purchase quiet equipment in the first place, but the reduced amount of acoustical treatment necessary can more than make up the difference.

The APTA Guidelines (American Public Transit Associaton) (Appendix B) recommend criteria for ancillary equipment noise. These criteria, summarized in Table 10-1, specify the maximum allowable levels 15 m (50 ft) from the facility, at the setback line of the nearest building, or at the nearest occupied area, whichever is nearest to the source. "Transient noise" refers to intermittent noise, such as that caused by trains. The criteria in Table 10-1 cover a 25 dBA range with the appropriate level depending upon the type of neighborhood in which the facility is located; the criteria are least restrictive for industrial areas and the most restrictive for low-density residential areas. Ancillary facilities located in industrial areas generally do not require any noise control features. However, when an ancillary facility is located in a low-density residential area, particular care must be taken to ensure that the noise levels are not excessive.

When the noise from the ancillary equipment contains identifiable pure tones (e.g., transformer hum), the maximum noise levels of Table 10-1 should be reduced by 5 dBA. This penalty allows for subjective annoyance, which is significantly increased when audible pure tones are present.

In most situations, maintaining the community noise levels below the limits in Table 10-1 will prevent community annoyance. However, when developing criteria for a specific installation, it is important to determine if any local noise ordinance is applicable. If there is a local noise ordinance, the noise limits normally take precedence over the transit system criteria.

10.1 FAN AND VENT SHAFT NOISE

This section discusses the available methods for reducing ventilation fan and train noise radiated from shaft openings at the surface and for preventing fan noise from penetrating into stations. The results of fan noise measurements on station platforms and near fan shafts are also presented. Most of the material in this section is largely adapted from References 10.1, 10.5, 10.6, and 10.7.

Figure 10.1 illustrates the cross section of a typical fan/vent shaft configuration and the paths by which noise travels to patrons and adjacent communities. When the ventilation fans are operating, the noise travels from the fans through the tunnel to the station platform and through the shaft to the outdoors. Without acoustical treatment, fan noise can be intrusive to nearby residential areas and to patrons on the station platform. Fan noise is also transmitted through the transit car body, but

TABLE 10-1 GUIDELINES FOR NOISE FROM TRANSIT SYSTEM ANCILLARY FACILITIES[1]

Community Area Category[2]		Maximum Noise Level Design Goal[3]	
		Transient Noises	Continuous Noises
I	Low-Density Residential	50 dBA	40 dBA
II	Average Residential	55	45
III	High-Density Residential	60	50
IV	Commercial	65	55
V	Industrial/Highway	75	65

[1]Adapted from APTA Guidelines, see Appendix B

[2]Community cateogry definitions are in Table 2-8.B of the APTA Guidelines

[3]Noise level goals to be applied at 15 m (50 ft) from ancillary equipment, at the setback line of the nearest building, or at the nearest occupied area.

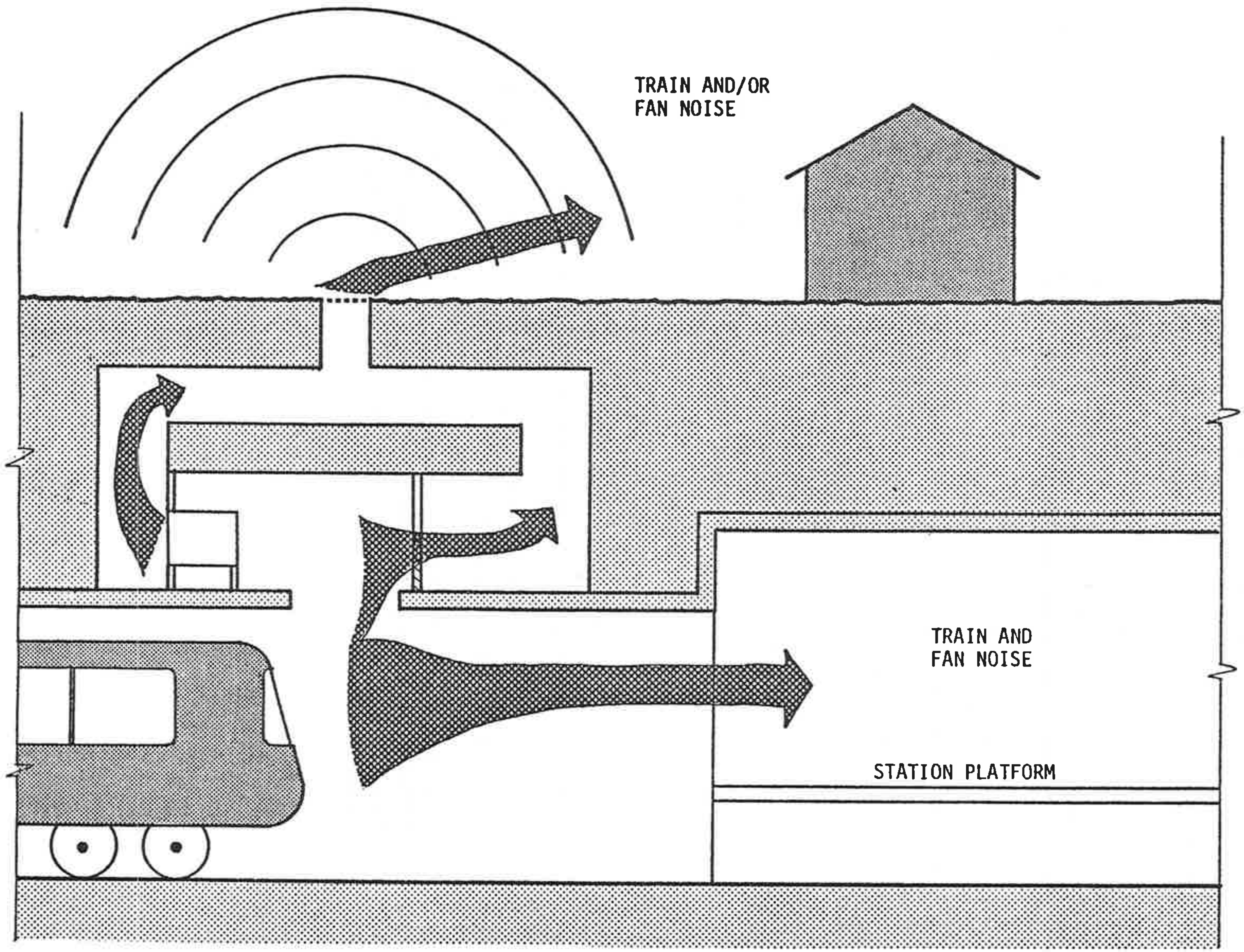


FIGURE 10.1 SKETCH OF SOURCES AND PATHS OF FAN AND VENT SHAFT NOISE

because of the sound-insulating properties of the car bodies and of the train noise, it is rarely noticeable inside the transit cars.

10.1.1 Procedures for Attenuating Fan and Vent Shaft Noise

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

- Use acoustical absorption material to line surfaces of fan rooms and shafts, tunnel walls and ceilings near vent shafts in subways, and areas between the subway and the shaft.
- Attach sound attenuators (also called silencers, mufflers, and sound traps) to the fans.
- Use specially constructed splitters and acoustical louvers in the fan and vent shafts and in the fan room or the tunnel opening.

Typical fan shafts with appropriate noise control treatments are shown in Figure 10.2.

The sound reduction achieved by lining a fan or vent shaft depends on the placement, area to be covered, and type of material used. The characteristics of absorption material are discussed in detail in Section 10.1.2. Section 10.1.3 covers the importance of placement and area of coverage.

If sufficient space is available, prefabricated silencers can control fan noise. In some cases, attaching an attenuator to the fan may be preferable, and more economical, than lining the fan shaft. Attenuator units are usually selected on the basis of required sound attenuation, permissible head loss, and available space. They may be either round or rectangular, depending on the type of fan outlet cone or transition section used. Information concerning the head loss, as well as the sound attenuation with flow at appropriate face velocity, is given in the catalogs published by manufacturers. When attenuators are to be used for

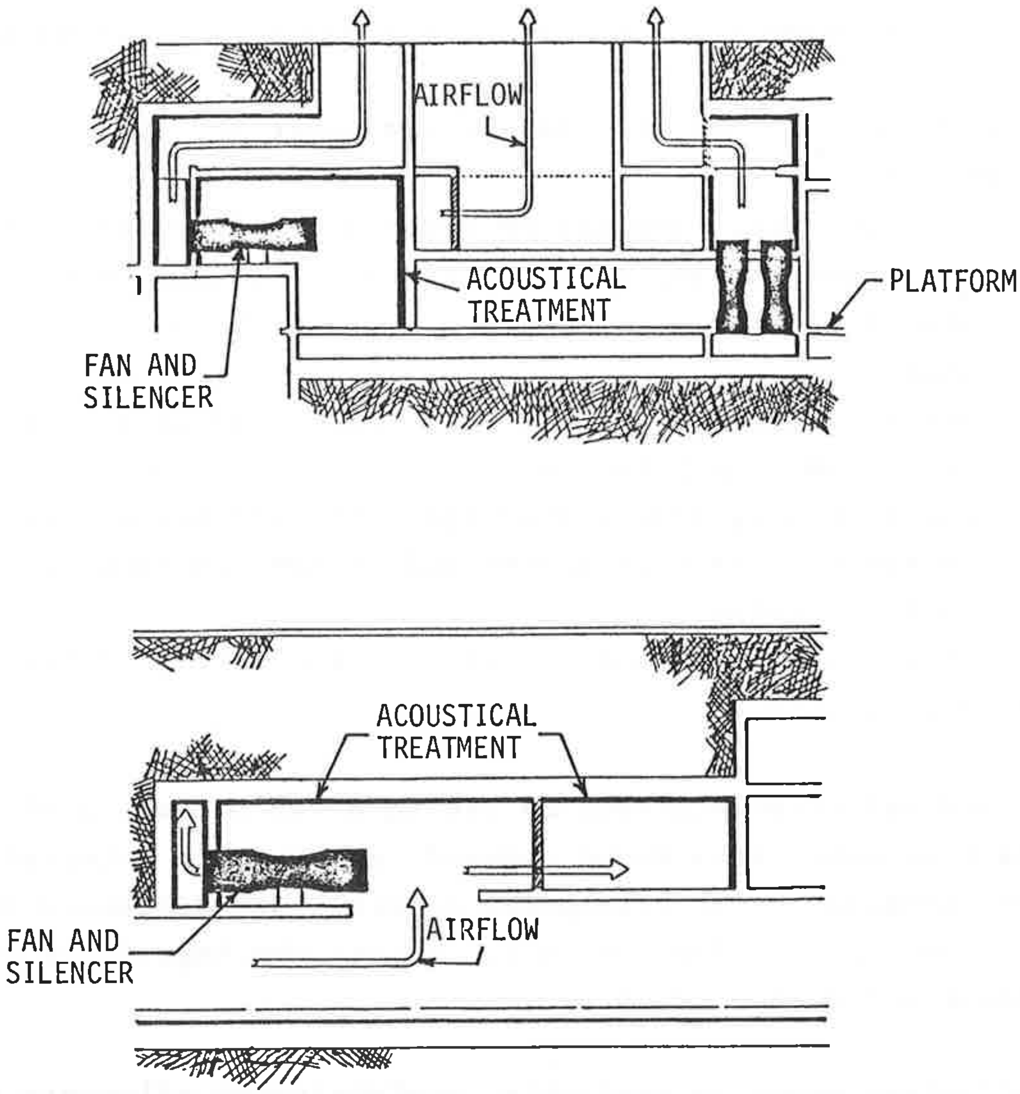


FIGURE 10.2 TYPICAL FAN SHAFT NOISE CONTROL TECHNIQUES

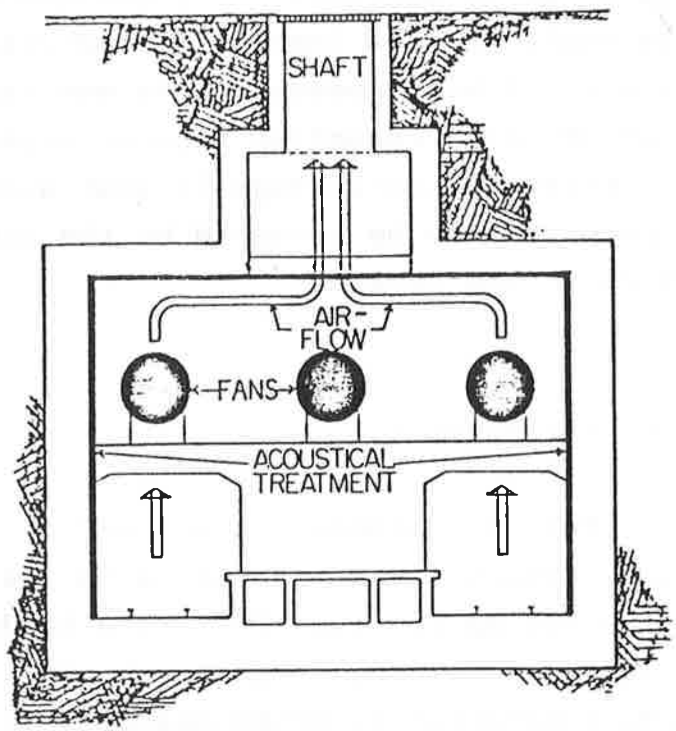


FIGURE 10.2 (CONT'D)   TYPICAL FAN SHAFT NOISE CONTROL TECHNIQUES

reducing fan shaft noise, simply select a unit that can accommodate the necessary airflow and will give the necessary attenuation at the expected air velocity.

It is also possible to build an attenuator into the shaft, either by installing several absorbent splitters (acoustically absorbent panels oriented parallel to airflow) or by putting acoustical louvers inside the shaft or at a damper. Commercial splitters and louvers are readily available, however, they generally cost more for the same amount of noise reduction than absorption material or duct attenuators. Another disadvantage is that access for maintenance or other purposes may be hindered by the presence of louvers or splitters in the shafts.

10.1.2 Sound Absorbent Materials

Absorbent materials must be durable and economical; they must provide extremely efficient sound absorption in the frequency range covered by the 500 Hz and 1000 Hz octave bands and adequate sound absorption in the remaining frequency bands. The efficiency of a sound absorption material is often measured in terms of the Noise Reduction Coefficient, NRC, a single number rating of the absorptive coefficients at 250, 500, 1000, and 2000 Hz. However, since reduction of transit-related noise generally requires maximum efficiency in the 500 Hz and 1000 Hz octave bands, it is recommended that the absorption coefficients in these octave bands be used rather than the NRC. Materials of similar NRC's may perform altogether differently at 500 Hz and 1000 Hz. For example, thin layers of material provide the best absorption at 2000 Hz, less absorption at 1000 Hz, and considerably less at 500 Hz and 250 Hz. The averaging by which the NRC is obtained could thus obscure deficient performance in the 500 Hz frequency range, resulting in an overestimation of sound attenuation properties.

Three basic kinds of material can be used for sound absorption in the fan and vent shafts:

- Spray-on materials, such as those appropriate for use on the subway walls.
- Conventional glass-fiber boards or blankets, mechanically attached to the fan and vent shaft interior surfaces.
- Cellular glass blocks, mechanically attached or adhered to walls.

Spray-on materials provide the easiest installation procedure and may be cheaper than glass-fiber material. Glass-fiber boards or blankets provide the highest sound absorption coefficient and, therefore, the largest amount of sound attenuation for a given area of coverage. Cellular glass blocks (Geocoustic Blocks), a proprietary product of Pittsburgh Corning, absorb sound very effectively. A particular advantage for subway applications is the incombustibility of the blocks.

10.1.2.1 Spray-on Materials -- Although spray-on materials have been successfully applied to the walls and ceilings of several transit systems, the number of satisfactory spray-on products is much more limited than glass-fiber blanket or board materials. Many spray-on materials either provide very little sound absorption or are not durable enough for use in subway fan and vent shaft installations. Those recommended include:

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

All these materials have similar absorption characteristics when applied in thicknesses of 16 mm to 20 mm (5/8 in. to 3/4 in.).

When properly applied, Sound Shield 85, Kilnoise, and Pyrok are all durable enough to withstand repeated cleaning or washing with water spray. Pyro-Spray may also be sufficiently durable, but at this time, its resistance to mechanical abrasion and water spray has not been determined. Pyrok use in WMATA tunnels has not been satisfactory, apparently due to incorrect installation procedures. Although many successful architectural applications of Pyrok exist, if it is used in subway applications, the installation procedure must be fully defined and carefully monitored. As for other spray-on materials, incorrect installation can be responsible for reduced acoustical effectiveness and inadequate adhesion to the concrete surface.

Spray-on materials such as Cafco Sound Shield 85, applied directly to fan and vent shaft surfaces in thicknesses of 16 to 20 mm (3/8 in. to 3/4 in.) will provide absorption coefficients of .45 to .55 at 500 Hz and .70 to .80 at 1000 Hz. Pyrok has reported absorption coefficients for 25 mm (1 in.) thick application, applied directly to a concrete surface, of .24 and .42, at 500 and 1000 Hz respectively. Evaluation of the initial Pyrok installations at WMATA confirmed that these figures are reasonably accurate; the measured values for 25 mm (1 in.) thick WMATA subway installations of Pyrok were .17 and .39 at 500 and 1000 Hz respectively (Ref. 10.11). Although this is less absorption than is indicated by laboratory tests, it will still provide significant noise reduction. With correct installation, Pyrok would give very good results.

10.1.2.2. Glass-fiber Boards and Blankets -- A wide range of glass-fiber blanket and board materials will satisfactorily control fan and vent shaft noise. The material should be 25 to 100 kg/cu m (1.5 to 6.0 lb/cu ft), with or without sprayed-on vinyl or neoprene coating for protection. The most economical and appropriate material is glass-fiber duct liner, as used in ventilation system ducts. This material is generally available in 25 mm (1 in.) thickness; two layers can be used to obtain 50 mm thickness. A recommended material for sound absorption treatment is Owens-Corning's Fiberglass rigid or semirigid board; Johns-Manville and Certain-Teed/St. Gobain (Gustin-Bacon) supply similar materials, with equivalent mechanical and acoustical characteristics. Where mechanical protection of the material is necessary (e.g., in areas where treatment is subject to contact by personnel or equipment), the installation may include an outer covering of acoustically transparent materials, such as hardware cloth or expanded metal. Dust or dirt collecting on the surface of the glass fiber will not significantly affect its sound absorption characteristics although dust can be a fire or smoke hazard. Water has no permanent degrading effect on the sound absorption ability of glass fiber, but absorption is reduced while the material is wet. Over the course of time, the detergents used in tunnel washing may leave an accumulation of residue, the effects of which are not yet known.

Glass-fiber material can be kept from collecting dirt and absorbing water by enclosing it in an envelope of plastic film. Up to 0.1 mm (0.004 in.) thick plastic film can be used without significantly decreasing noise reduction. Any plastic film with a thickness of 0.1 mm or less is acoustically satisfactory; it is only important that its weight be less than about 0.09 kg/m^2 (0.3 oz/sq ft). The selection of a plastic film must be based on the life expectancy of the tunnel and the fire resistance of the material. In many outdoor environments, mylar or polyethylene film is used. However, when the fire resistance capacity of these materials is unacceptable, a polyamide film such as DuPont Kapton or any other fire-resistant material should be considered.

A number of procedures can be used to attach glass-fiber boards and blankets to concrete surfaces. In machinery rooms and concrete ducts, "Stic Klips" (similar to large-headed nails) can be used, either cast in the concrete or fastened to the concrete surface with cement or epoxy. The glass-fiber material is impaled on the rod, a washer is placed over the rod, and the rod is bent over the washer to retain the material and any added protective covering. Wood or metal furring strips, attached to the concrete surface, can be used with mechanical fastening to support and retain the glass fiber. The Owens-Corning Fiberglass "Acoustical Stud" or similar shaped metal extrusions can be used to attach 50 mm (2 in.) thick materials to concrete walls, without using fasteners that penetrate the glass fiber or protective coating. Such mountings are especially convenient when a waterproof covering is to be used.

Fire safety is a major concern when specifying acoustical absorption material for subway applications. A glass fiber with no binder is necessary to achieve an incombustible product, however, very few products are manufactured without binders because glass fiber tends to lose fibers when there is no binder material. Temp-Mat, from Pittsburgh Corning, is a glass-fiber product held together by a mechanical felting process that contains no binder. It does contain a small amount of residual oil, used in the manufacturing process, which can be baked out by the manufacturer. It has a density of 180 kg/m^3 (11.25 lb/cu ft) in the 25 mm (1 in.) thickness, which is two or three times the density normally recommended for glass-fiber acoustical absorption. The density of Temp-Mat makes it cost more than lower density materials, but its acoustical performance will be as satisfactory as lower density 100 kg/m^3 materials of the same thickness.

10.1.2.3 <u>Cellular Glass Blocks</u> -- Geocoustic Blocks have been used successfully in a number of subway applications. They are made of rigid glass foam 50 mm (2 in.) and 100 mm (4 in.) thick and are slotted to increase effective surface area and absorption. The manufacturer states, and tests confirmed, that these blocks have absorption coefficients above 0.90 at 500 and 1000 Hz. Geocoustic Blocks have the significant advantage of being completely inorganic and incombustible; however, they have the disadvantage of shedding small glass granules. The manufacturer is experimenting with a spray-on coating to control the shedding problem, but this adds an organic component to the material that might increase fire-related problems.

10.1.3 Fan and Vent Shaft Lining

Using acoustical absorption material in air handling systems can be a very effective method of controlling the noise propagated through the duct system. As a general rule of thumb, attenuation is maximized when absorbent material is applied to the surfaces on which the sound energy impinges most directly, especially at bends in the shafts. The following discussion is largely based on the ASHRAE Guide (Ref. 10.1) which contains more detailed information on the control of noise in ventilating systems.

There are three basic methods for applying absorption treatment to ventilating systems: lining straight sections of duct, lining bends in the ducts, and lining plenum areas. The least effective is to line straight sections of ducts. Because the ducts in vent and fan shafts are usually large, typically 10 m^2 (100 sq ft), large amounts of treatment are needed to achieve significant attenuation. Lining bends and plenums can be very effective and economical methods of reducing noise. Note that because fan and vent shafts are large, it is not always appropriate to think of a shaft as a duct. Often, it is more appropriate to analyze the shaft as though it were an acoustically lined plenum.

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

$$\text{Attenuation (dB)} = 1.05\, d(P/A)\bar{\alpha}^{1.4} \qquad (10.1)$$

where d = length of lining

P = duct perimeter

A = duct area

$\bar{\alpha}$ = average absorption coefficient
(a function of frequency)

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

This formula does not account for line-of-site propagation, which limits attenuation at high frequencies. For a duct with a minimum cross-section dimension of about 1 m (3 ft), the maximum attenuation in a straight-lined duct is 10 dB in the 2000 Hz octave band. The attenuation in the 1000 Hz octave band will be approximately midway between 10 dB and the value calculated from the equation. The frequency above which the 10 dB limit applies is inversely proportional to the shortest dimension of the duct. For a large vent shaft with a minimum dimension of about 3 m (10 ft), the 10 dB limit applies to frequencies above approximately 600 Hz. Note that for a 3 m square shaft (9 m^2 or about 30 ft^2) lined with 50 mm (2 in.) thick fiberglass, with an absorption coefficient of 0.8 in the 500 Hz octave band, the formula requires lining nearly 10 m (30 ft) of shaft to obtain 10 dB attenuation. This length, usually not available, indicates why lining lengths of straight shafts is usually impractical.

This discussion of sound attenuation by lining straight ducts assumes that axial sound waves move down the duct. Since jogs and bends in the shafts will excite nonaxial propagation modes, lining sections of straight shafts will sometimes provide slightly more attenuation than is indicated by the formula given above. In cases where the sound is transmitted down the duct in primarily nonaxial modes, significant attenuation can be achieved by lining straight sections of duct (e.g., ducts connected to fan intake or exhaust plenums). The noise field entering the duct from the plenum is of random incidence. Lining a length of duct equal to 3 to 4 times the duct width will give sound attenuation approximately equal to that of a bend lined after the bend only. After a length of 3 to 4 times the duct width, nonaxial modes are essentially eliminated; lining beyond this length will give only the attenuation equal to that of a normal, lined straight duct.

10.1.3.2 Right Angle Bends -- Acoustical treatment before and after bends can be very effective. The sketch in Figure 10.3 indicates the most effective locations for placing absorbing material to achieve noise reduction at shaft bends. The definitions of shaft depth, d and D, and shaft width, W, as they apply to sound attenuation treatment in a fan or vent shaft, are also indicated in the figure. Note that the sketch shows acoustical material on only the sides normal to the plane of the bend. Additional material on the sides parallel to the plane of the bend would contribute to the total sound attenuation; but such placement is inefficient because the added material acts only as lining in a straight duct.

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

- two shaft depths before or after the bend, if a high absorption lining is used ("thick" treatment),

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

Extending the lining for additional lengths does not appreciably increase sound attentuation. For each additional length of lined duct equal to the duct width (d or D), the added attenuation will be about 1.5 dB for thick treatment and 0.8 dB for thin treatment. The sides of the shaft parallel to the bend should be lined only if shaft width and depth are comparable.

In practice, lining the wall and ceiling areas of a shaft is practical, but lining the floor is not. This restriction means that in some cases, only one side of the duct shaft can be lined,

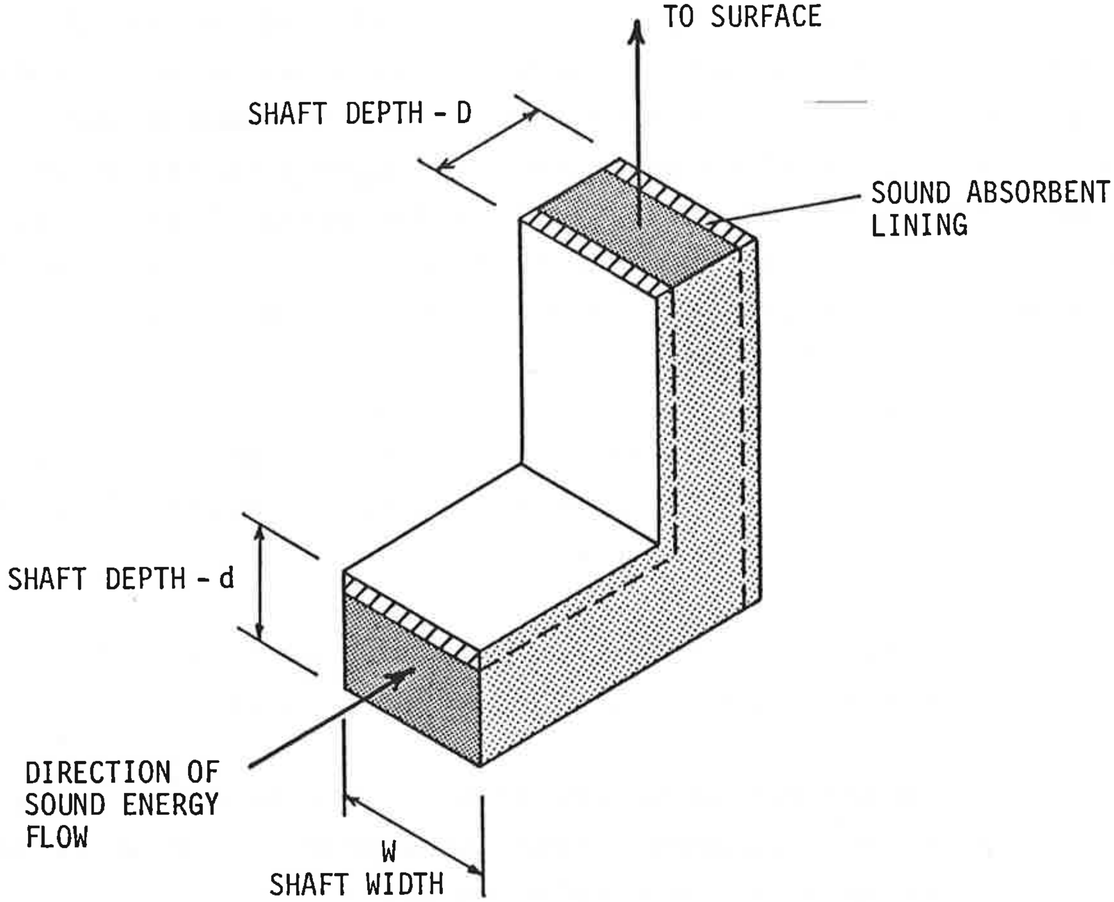


FIGURE 10.3 RIGHT ANGLE BEND IN DUCT

either just before or just after a bend.

Table 10-2 indicates the approximate amount of attenuation that can be achieved by applying either "thick" or "thin" sound-absorbing treatment at shaft bends. The "thick" treatment would typically consist of 50 mm (2 in.) thick glass fiber or of a spray-on material on metal lath backed by a 25 mm (1 in.) airspace. A "thin" treatment would consist of a directly applied spray-on material 16 mm to 20 mm (5/8 in. to 3/4 in.) thick. Note that 25 mm glass wool, applied directly to the concrete surface, would provide absorption and attenuation about halfway between the "thick" and "thin" treatments as described earlier.

The figures in Table 10-2 show the effects of lining straight portions of the shafts, as well as the effects of interaction at a bend and are intended for use with rectangular shafts of a width greater than 2D. For shafts of less than 2D, the sides parallel to the plane of the bend should also be lined. The results apply to round shafts also if the lining on one-half the circumference is considered equal to lining on one side of a rectangular shaft, and the lining around the entire circumference equals the lining shown on two sides. In shafts where the available length of straight duct to be lined before or after a bend is less than 1.5D, attenuation will be less than that given in Table 10-2; in these shafts, other methods of achieving noise reduction may be appropriate.

The attenuations listed in Table 10-2 are equally valid for small, medium, or large shafts. This generalization is possible because shafts are usually designed so that the lining area, as defined by the table, is approximately proportional to the shaft size. For full 180 degree bends in the shaft, the attenuation obtained by lining the shaft either before or after the bend will be approximately 1.5 times that listed in Table 10-2.

The following example illustrates the use of Table 10-2 to

TABLE 10-2 APPROXIMATE ATTENUATION OF SOUND ACHIEVED BY LINING RIGHT ANGLE BENDS IN FAN AND VENT SHAFTS

Lining Location	Lining Area: Thick Treatment[1]	Lining Area: Thin Treatment[2]	Sound Attenuation: 500 Hz	Sound Attenuation: 1000 Hz
Ahead of Bend				
1 side	2dW	3dW	2 dBA	3 dBA
2 sides	(4d + D)W	(6d + D)W	6	8
After Bend				
1 side	(2D + d)W	(3D + d)W	5	6
2 sides	(4D + d)W	(6D + d)W	8	10
Ahead of and after Bend				
1 side + 1 side	(2D + 3d)W	(3D + 4d)W	7	10
1 side + 2 sides	(4D + 3d)W	(6D + 4d)W	10	14
2 sides + 1 side	(3D + 5d)W	(4D + 7d)W	11	16

[1] "Thick treatment" consists of 50 mm (2 in.) thick glass fiber or spray-on material on metal laths. Absorption coefficients at 500 Hz and 1000 Hz of 0.8 to 0.9.

[2] "Thin treatment" consists of 15 to 20 mm (5/8 to 3/4 in.) thick spray-on material with absorption coefficients of at least 0.5 at 500 Hz and 0.7 at 1000 Hz.

estimate the reduction caused by lining a vent shaft. Figure 10.4a indicates a shaft bend, similar to many found in both station and line vent shafts. In this situation, several circumstances prevent the maximum utilization of a lined bend. It is generally impractical to line the floor of a shaft or any area open to the weather; the only remaining area available for treatment is the ceiling before the bend. The 5.5 m of thick treatment on the ceiling, indicated in Figure 10.4a, will give a reduction of only about 3 dB. However, if the shaft is rearranged, as shown in Figure 10.4b, the bend can be lined much more effectively.

If the design is modified according to Figure 10.4b, the total attenuation will be about 15 dB. The attenuation achieved by lining one side before and one side after the bend is about 10 dB, and the extra attenuation of a 180 degree bend, compared to a 90 degree bend, is 1.5 times 10 dB. Clearly, such alterations will complicate the airflow; increased flow resistance must be considered.

Vent and fan shaft designs often fail to allow sufficient area for acoustical treatment, and some of the potential attenuation provided by a lined bend is lost. Although lining such bends will not provide the greatest possible attenuation, it will be greater than that achieved by lining a straight shaft with identical treatment. Using half the recommended length of lining on a bend will result in approximately half the attenuation.

10.1.3.3 Plenums -- Analyzing the section of shaft illustrated in Figure 10.5 as a large bend, with thick absorption material on the two sides before the bend, gives 8 dB attenuation at 500 and 1000 Hz. Following the estimates of treatment area given in Table 10-2, 85 sq m of treatment would be needed. However, the shaft section is more accurately modeled as a plenum than a lined bend.

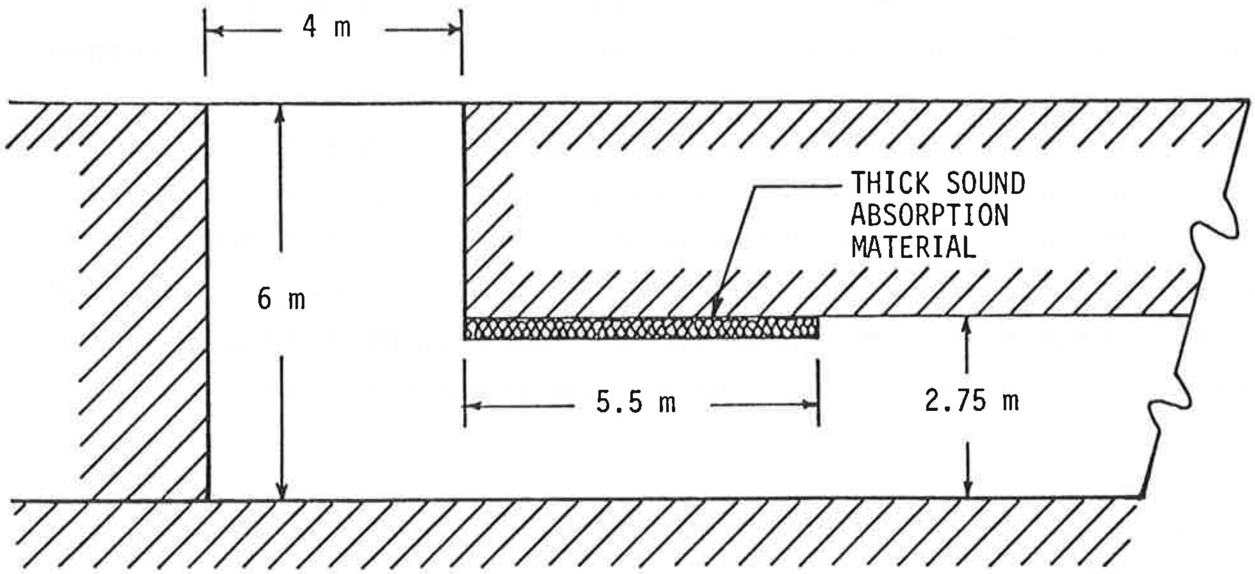


FIGURE 10.4.a

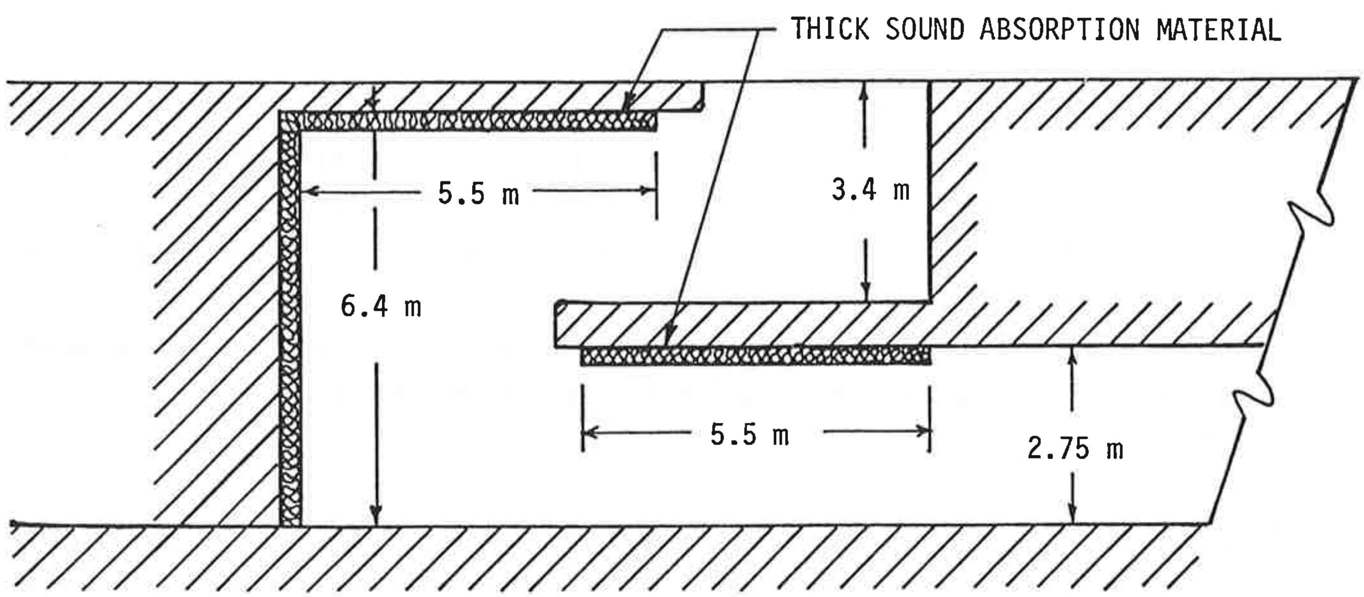


FIGURE 10.4.b

FIGURE 10.4 TYPICAL BEND IN SHAFT

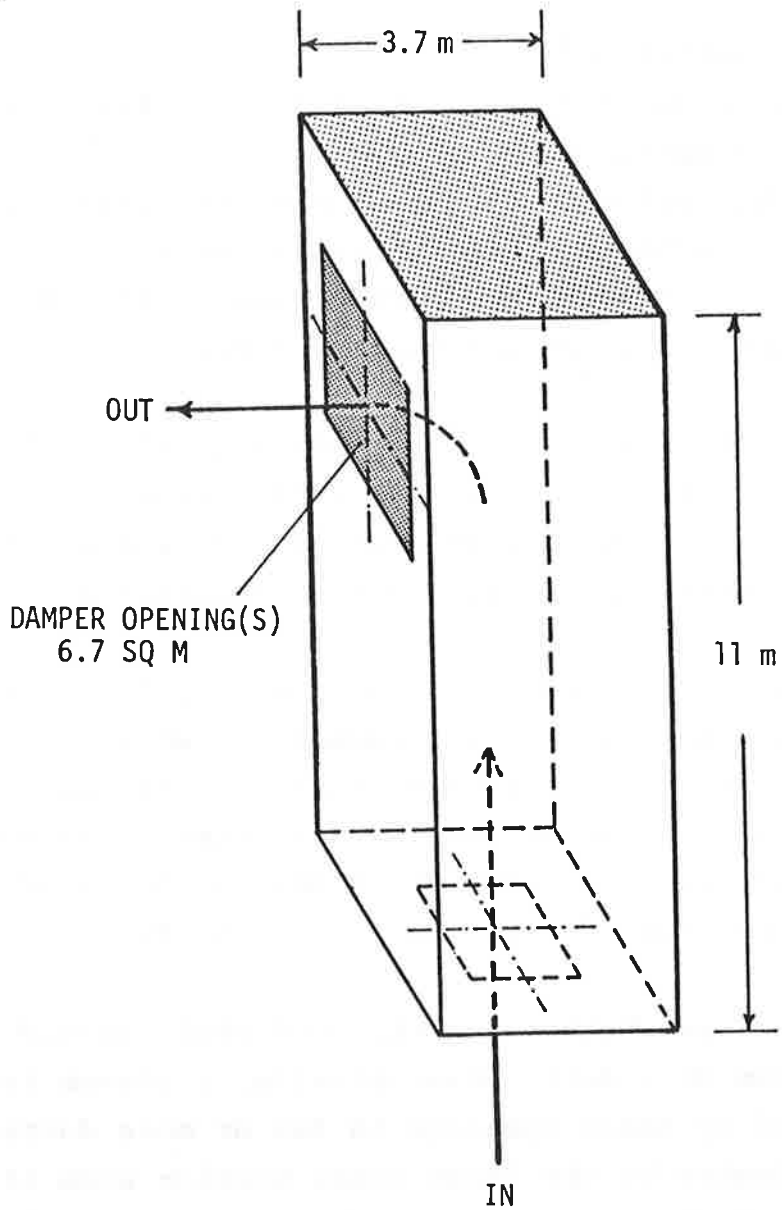


FIGURE 10.5 TYPICAL PLENUM CONFIGURATION

The formula for approximating reduction obtained by lining a plenum is:

$$\text{Attenuation (dB)} = 10 \log \left[S \left(\frac{\text{Cos}\theta}{2\pi q^2} + \frac{1-\bar{\alpha}}{a} \right) \right] \qquad (10.1)$$

where S = area of outlet, m^2

$\bar{\alpha}$ = average random-incidence absorption coefficient of plenum, dimensionless

a = total absorption in plenum chamber in sabins, equal to area of treatment times absorption coefficient, m^2

q = diagonal distance between inlet and outlet, m

θ = angle between inlet and outlet, degrees.

In the design sketched in Figure 10.5, assuming 85 m^2 of absorptive lining with an absorption coefficient of 0.9, the reduction is 12 dB -- 4 dB more than the lined bend estimate. Doubling the treatment area increases the reduction to 19 dB.

This formula, for attenuation by a lined plenum, is accurate within a few decibels even at high frequencies where the acoustic wavelength is less than the plenum dimensions. At lower frequencies, the equation is conservative; actual attenuation sometimes exceeds the calculated value by 5 to 10 dB, due to sound reflections caused by the expansion at the plenum inlet.

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

10.1.4 Vent Shaft Entrances

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

In vent shafts and stations, the added attenuation achieved with a "thick" absorption treatment on the tunnel walls and ceilings is 5 to 7 dBA, compared to no lining at all. If a "thin" treatment is used, the expected noise reduction amounts to 3 to 5 dBA, at 500 and 1000 Hz. These figures assume that the walls and ceiling of the subway are lined for 15 m (50 ft) on each side of the vent shaft entrance. Train noise radiated from the vent shaft can be further reduced by lining the shaft with acoustic material, as is discussed in the preceding section.

10.1.5 Fan Attenuators

Many types of attenuator units are available for installation on ventilation fans to reduce noise either radiated from the surface outlets or audible in the station. Prefabricated attenuators are available in both rectangular and round shapes, so that either a conical outlet or a sheet metal transition section, going from round to rectangular, can be used on the fans.

In general, the selection of a fan attenuator depends on the amount of attenuation required. If all required noise reduction is to be provided by the attenuator, a unit is needed that provides sufficient attenuation in the 500 Hz and 1000 Hz octaves to meet whatever criteria are demanded. A number of factors should be considered when selecting an appropriate attenuator. The first is whether a rectangular or a round unit is needed, a choice that will probably be decided on the basis of length and convenience of installation. In general, rectangular units are shorter and often less expensive than round ones. Rectangular attenuators are typically supplied as modules .9 m, 1.5 m, 2.1 m, or 3 m (3 ft, 5 ft, 7 ft, or 10 ft) in length. The cross-sectional area depends on the limitations on head loss and face velocity of the airflow through the unit. For subway ventilation fans, available round units of a size giving sufficiently low head loss units vary from 3 to 5 m (10 to 16 ft) long. The round units are therefore less desirable, in view of probable space limitations.

After the configuration to be used is selected, the maximum permissible head loss determines the size and type of unit necessary. Rectangular units can provide sufficient noise reduction with satisfactorily low head loss, if the appropriate cross-sectional area is selected. For example, rectangular units, ranging from 2.3 m^2 (25 sq ft) to 9.3 m^2 (100 sq ft) in total cross-sectional area can be obtained, with appropriate attenuation ratings and head loss in the range of 25 to 75 Pa (0.1 in. to 0.3 in. of water) at 1700 m^3/min (60,000 cfm) airflow. Rectangular units are probably a better choice, since they are available in relatively short lengths and offer good noise attenuation. A .91 m (3 ft) length can provide attenuation at 500 Hz in the range of 12 to 22 dB, depending on the head loss rating. At 500 Hz, a 1.5 m (5 ft) unit can provide attenuation in the range of 17 to 37 dB, and a 2.1 m (7 ft) long unit can provide attenuation in the range of 23 to 46 dB. Head loss of 1.5 m (60 in.) diameter round units, which could be directly attached to the fans, vary from 25 to 100 Pa (0.1 in. to 0.4 in. of water), at 1700 m^3/min. (60,000 cfm) airflow. Attenuations achievable by these units vary all the way from 10 dB to 34 dB. However, the minimum available length is 3 m (10 ft), and some units are as long as 4.5 m (15 ft).

It is essential to ensure that the rating for a selected attenuator is applicable to whatever airflow velocity will exist when the unit is operating. Although not a general practice, some attenuator catalogs still present sound attenuation data under static conditions. When air is flowing in the same direction as the sound propagation, attenuation will be less than it is under static conditions.

10.1.6 Fan Rooms

When a fan room acts as an intake or discharge plenum, significant noise reduction is possible through lining the fan room with acoustical treatment. However, when axial fans are installed as an integral part of the ducts, sound-absorbing material in the fan rooms will not reduce noise radiated from the shaft or into the station. Treatment in the fan room can protect maintenance personnel who must be in the fan room when the fans are operating.

Sound radiated out of a fan room into a duct can be estimated using the following formula:

$$PWL_{OUT} = PWL_{FAN} + 10 \log \left[S_E(1-\bar{\alpha})/S\bar{\alpha}\right] \qquad (10.3)$$

where PWL_{OUT} = power level radiated into the shaft
PWL_{FAN} = sound power level created by the fan
S_E = exit area of the shaft
= average absorption coefficient of fan room
S = total surface area of fan room
$S\bar{\alpha}$ = total absorption of fan room

The sound power level radiated into the shaft with and without treatment of the fan room must be calculated in order to judge the reduction of sound power. In the following example (sketched in Fig. 10.6), the fan room is a cube, 9 m (30 ft) on each side, with a 10 m^2 (110 ft^2) opening into the shaft. If the walls are concrete, with an absorption coefficient of .015 at 500 Hz, calculations of the power level in the 500 Hz octave band transmitted into the shaft, with and without room treatment, will be:

<u>WITHOUT TREATMENT</u>

Surface Area: 476 m^2
Shaft Openings: 15 m^2 = 15 units absorption
Absorption of Surface: .015 x 476 = 7.1
Total Absorption: $(S\bar{\alpha}) = 15 + 7.1 = 22.1$
$= 22.1/476 = .046$

$10 \log [S_E(1-\bar{\alpha})/S\bar{\alpha}] = 10 \log(10 \times .946/22.1) = -3.6$

$PWL_{OUT} = PWL_{FAN} - 3.6$

<u>WITH TREATMENT</u> (200 m^2 of 50 mm (2 in.) thick fiberglass)

Absorption of Untreated Surface Area: 276 x .015 = 4
Absorption of Treated Area (=0.8): 200 x .8 = 160
Shaft Opening: 15 m^2
Total Absorption: $(S\bar{\alpha}) = 4 + 160 + 15 = 179$
$= 174/476 = .38$

$10 \log [S_E(1-\bar{\alpha})/S\bar{\alpha})] = 10 \log(10 \times .63/179) = 14.6$

$PWL_{OUT} = PWL_{FAN} - 14.6$

<u>ATTENUATION DUE TO TREATMENT: 11 dB</u>

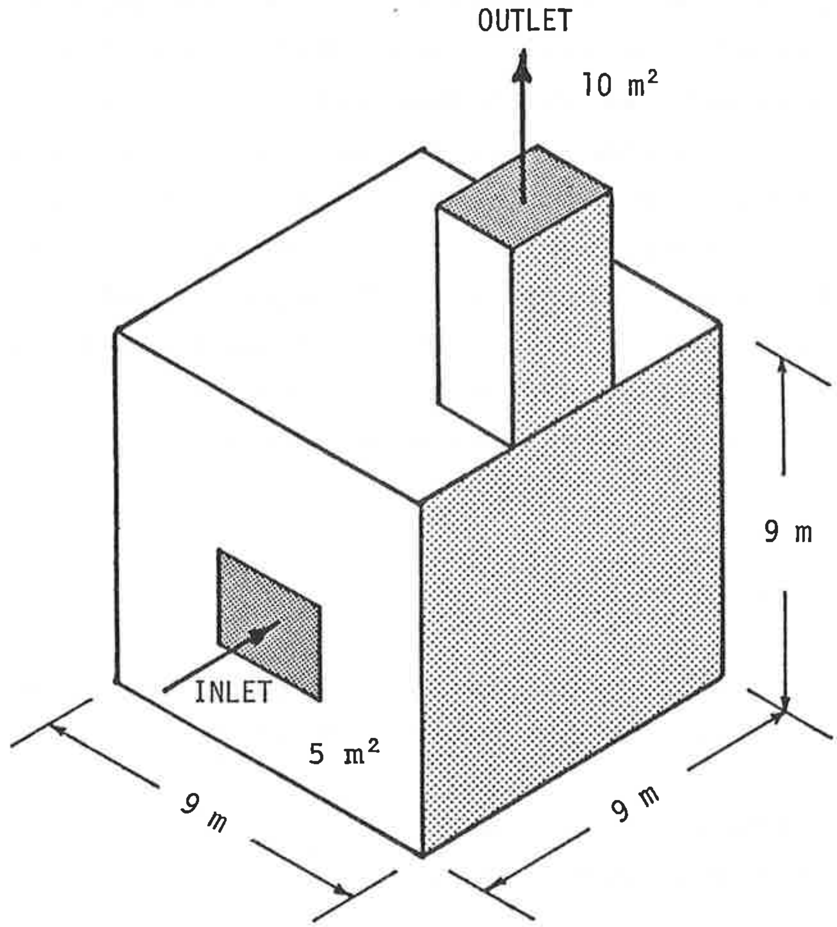


FIGURE 10.6 FAN ROOM EXAMPLE

10.1.7 Wayside Measurements of Fan Shaft Noise

Noise radiated from station ventilation fans into the station and out from the fan shafts to the community has been measured on several occasions at BART stations. To determine the directivity of noise radiated from fan and vent shafts, the fans were set to operate in both exhaust and supply modes, and measurements were taken on the ground floor, out the windows of the second floor, the fourth floor, and the seventh floor of a building across the street. These locations corresponded approximately to angles of less than 10 degrees, 30 degrees, 45 degrees, and 60 degrees relative to horizontal. Sound pressure measurements were also taken at the grating to estimate the sound power level. The directivity index was then estimated according to the following relation:

$$L_{p\theta} = L_W + DI_\theta - 20 \log(r) - 8 \qquad (10.4)$$

where $L_{p\theta}$ = sound pressure level measured at distance and angle from the source, dB re 2×10^{-5} Pa

L_W = sound power level, dB re 10^{-12} watt

r = radius of hemisphere, in m

DI_θ = directivity index.

Assuming that the average sound level at the shaft exit, L_{pe}, is known, the sound power level emitted from the shaft can be estimated as:

$$L_N = L_{pe} + 10 \log(A/2) \qquad (10.5)$$

where L_{pe} = average sound pressure level at the shaft exit, dB re 2×10^{-5} Pa

A = shaft exit area, in m^2

The sound level at distance "r" from the shaft can then be expressed in terms of the sound level at the shaft exit:

$$L_p = L_{pe} + 10 \log(A/2r) + DI_\theta - 8 \qquad (10.6)$$

These three expressions are valid regardless of the noise source. The factor "A/2" in Equation 10.5 is used instead of "A" to account for the random incidence characteristics of the sound field within the shaft (see p. 4-49).

The result of the directivity measurement (Fig. 10.7) shows that the directivity index increases as the angle above the horizon decreases. From these results, the influence of the height above the ground can be plotted. The levels shown in Figure 10.8 are relative to the level at 1.5 m above the ground. Note that the directivity and the actual diagonal distance from the shaft opening to the building tend to offset each other. Therefore, on high floors (where the sound level contribution from directivity is greater), the increase in distance will reduce the noise level to essentially the noise level on the ground. Of course, as the angle increases beyond 60 degrees, the directivity index becomes small enough so that distance determines the anticipated noise level.

Actual noise levels in the community depend upon the sound power created by the fan, the attenuation in the shaft, and the distance and angle from the grating to the receiver. In an untreated concrete shaft, a maximum of 2 to 3 dB attenuation is possible; essentially, all available sound power is radiated to the community.

Table 10-3 summarizes the results of two sets of measurements of wayside fan noise at the BART Berkeley Station. The first set of measurements was performed in November 1969 before the station was completed and the second set in October 1975 after the station had been in use for several years.

Since the fans are intended for emergency use only, the fan shafts were not treated. Figure 10.9 is an isometric sketch of one of the fan rooms and shafts, and Figure 10.10 is a photograph of one

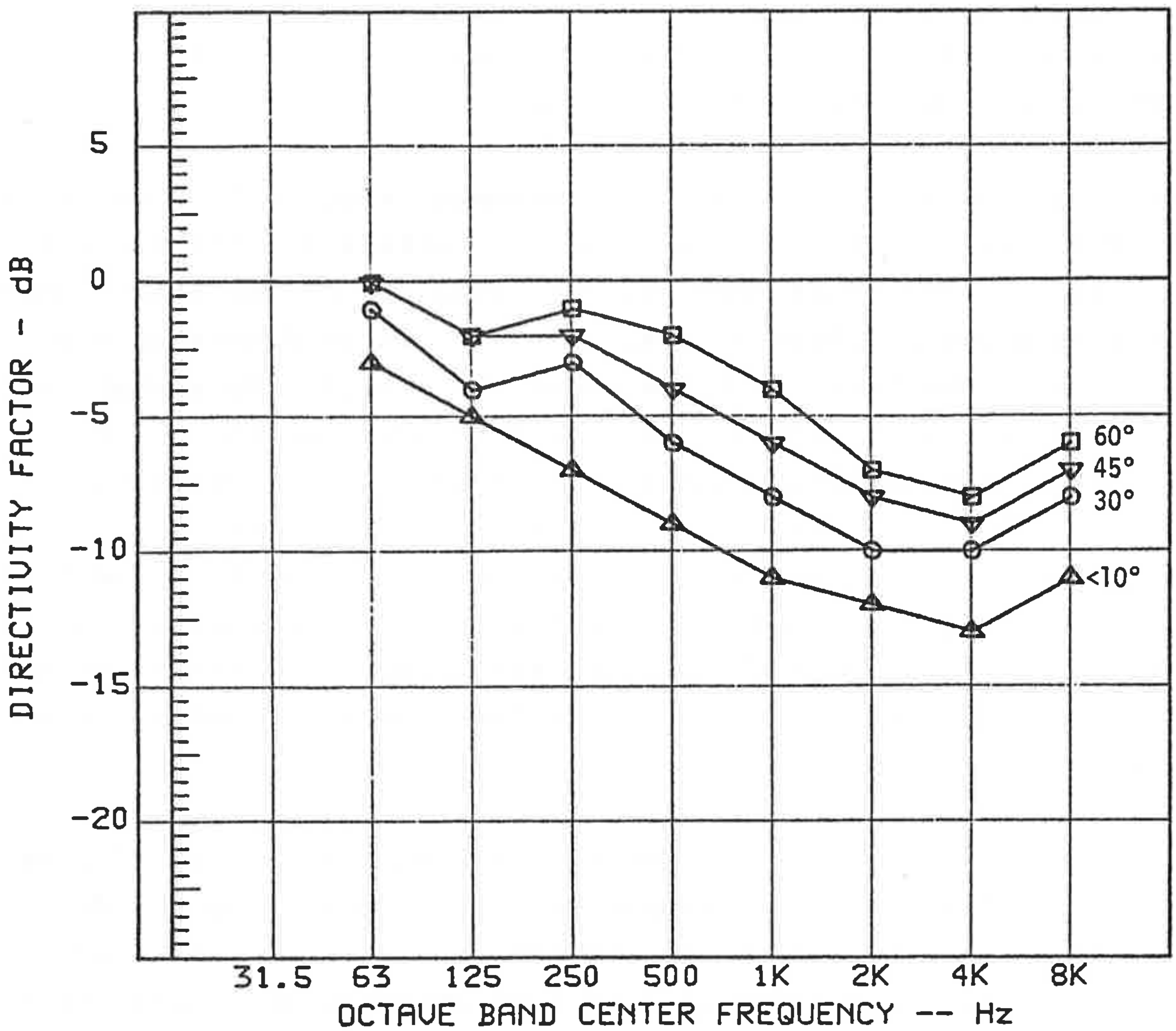


FIGURE 10.7 MEASURED DIRECTIVITY FACTORS FOR VARIOUS ANGLES ABOVE THE HORIZONTAL SURFACE

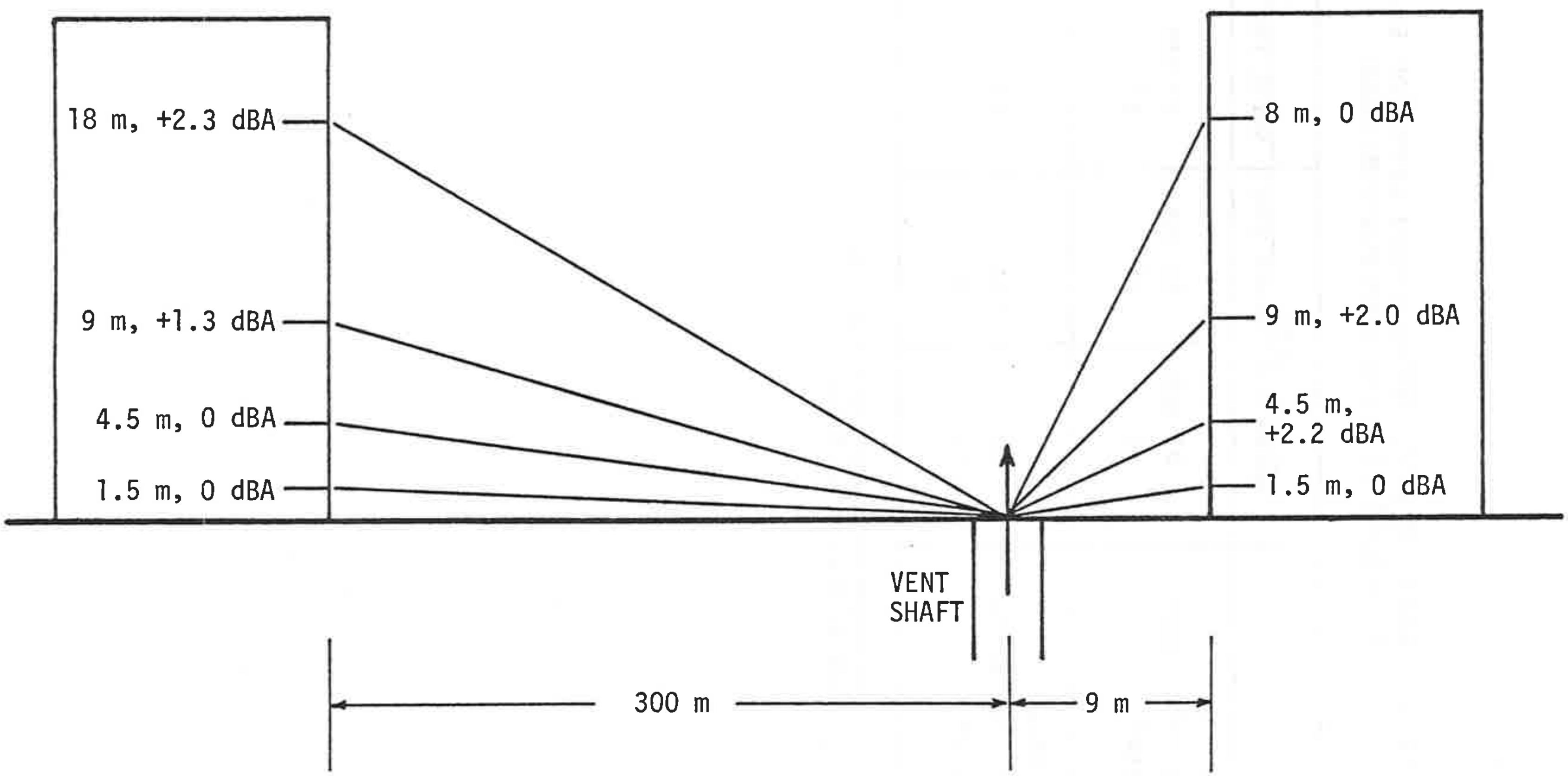
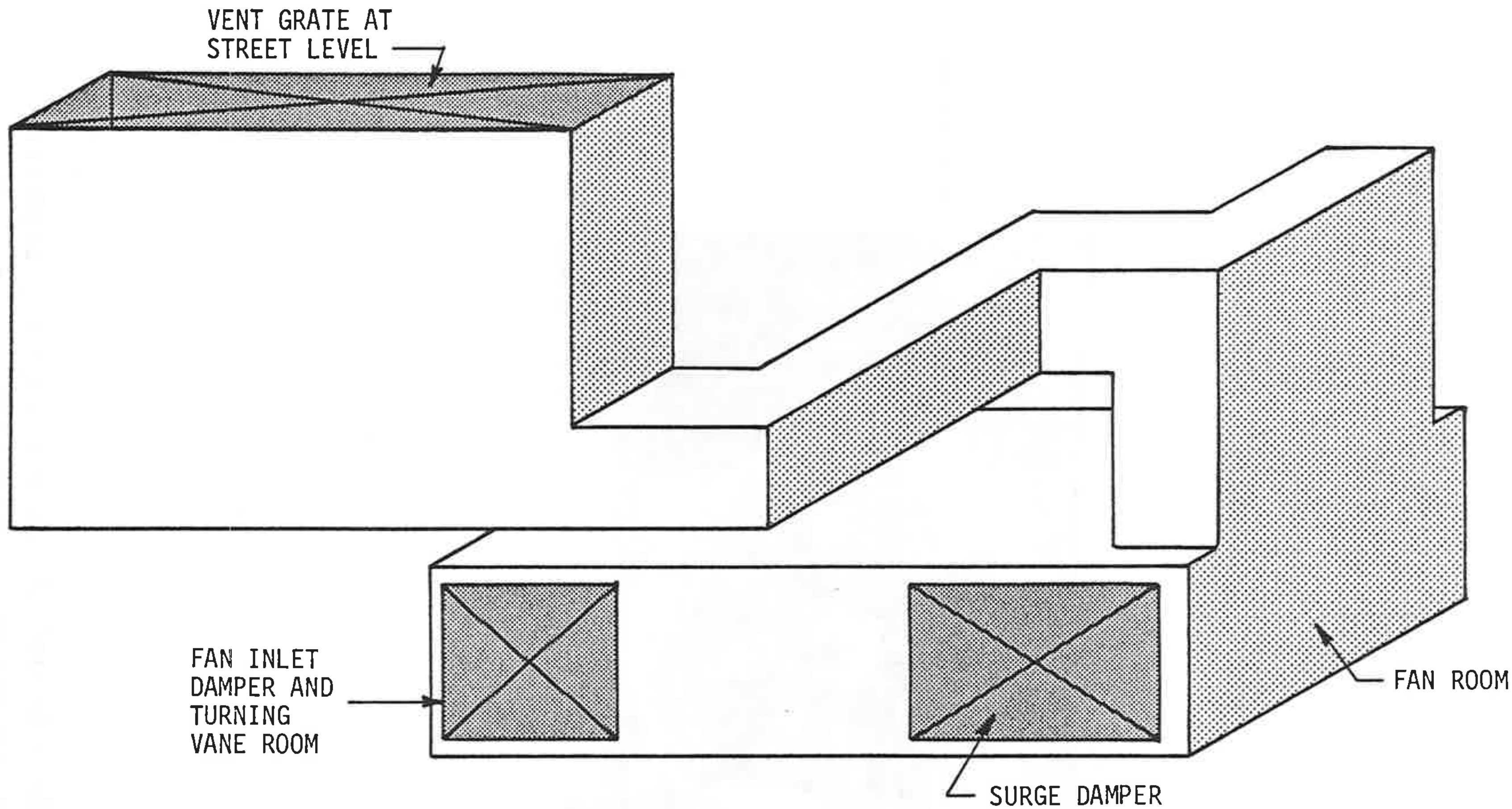


FIGURE 10.8 TYPICAL INFLUENCE OF HEIGHT ABOVE GROUND ON NOISE LEVEL RADIATED OUT VENT SHAFT - THE LEVELS SHOWN ARE RELATIVE TO THE LEVEL 1.5 m ABOVE THE GROUND

TABLE 10-3 WAYSIDE A-WEIGHTED SOUND LEVEL OF BART BERKELEY STATION VENTILATION FANS

Location	1969		1975	
	North Fan	South Fan	North Fan	South Fan
Fan in Exhaust Mode				
SPL at shaft exit grate	92 dBA	95 dBA	93 dBA	95 dBA
9 m from grate*	69	69	67	70
Fan in Supply Mode				
SPL at shaft exit grate	97	102	98	102
9 m from grate*	74	76	75	79

*Measurement location is 1.5 m above ground surface.



NOT TO SCALE

FIGURE 10.9 SKETCH OF FAN ROOM AND VENT SHAFT - BART BERKELEY STATION

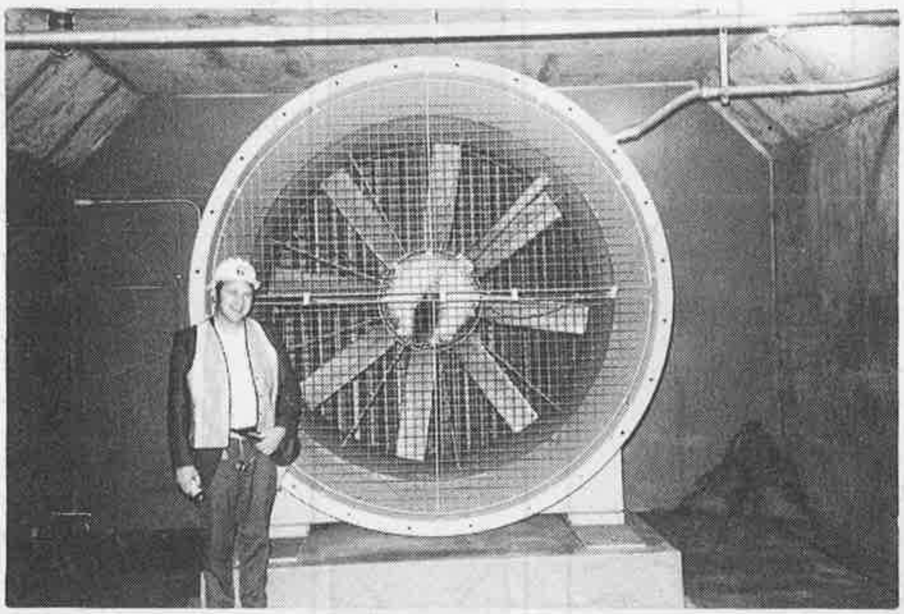


FIGURE 10.10 PHOTOGRAPH OF ONE OF THE 2.1 m (7 FT) DIAMETER STATION AND TUNNEL EXHAUST FANS IN THE BART BERKELEY STATION

of the fans inside the fan room. In the exhaust mode, the fans pull air out of the tunnels, through the turning vane room, through the fan into the fan room, and up the shaft. The fans are Joy Axivane Fans, 2.1 m (84 in.) in diameter, run by 84 HP motors.

The BART fan data in Table 10-3 are typical for station and line fan noise without treatment. The sound levels, 9 m (30 ft) from the vent shaft exit, averaged 69 dBA in the exhaust mode. If the fans were used in the exhaust mode on a regular basis, attenuation from 11 dBA to 36 dBA (depending on the type of land use near the shaft) would be required to meet criteria suggested by the APTA Guidelines. With appropriate selection of silencers attached to the fans and acoustical treatment of the fan rooms and shafts, the criteria could be met.

Train noise radiated out from fan and vent shafts can cause annoyance in the community. The noise reaching a receiver position at the surface is dependent upon the sound power emission of the train, which is speed dependent; the sound power transmitted from the subway into the shaft; the modification of sound power as it moves up the shaft to the surface; and finally, upon the distance and angle of the receiver relative to the shaft opening. Figure 10.11 illustrates levels of vent shaft noise calculated at the WMATA Metro System.

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

$$L_W = L_p + 10 \log(A_s) - 6 \text{ dB} \qquad (10.7)$$

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

A_s = area of shaft opening, in m^2

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

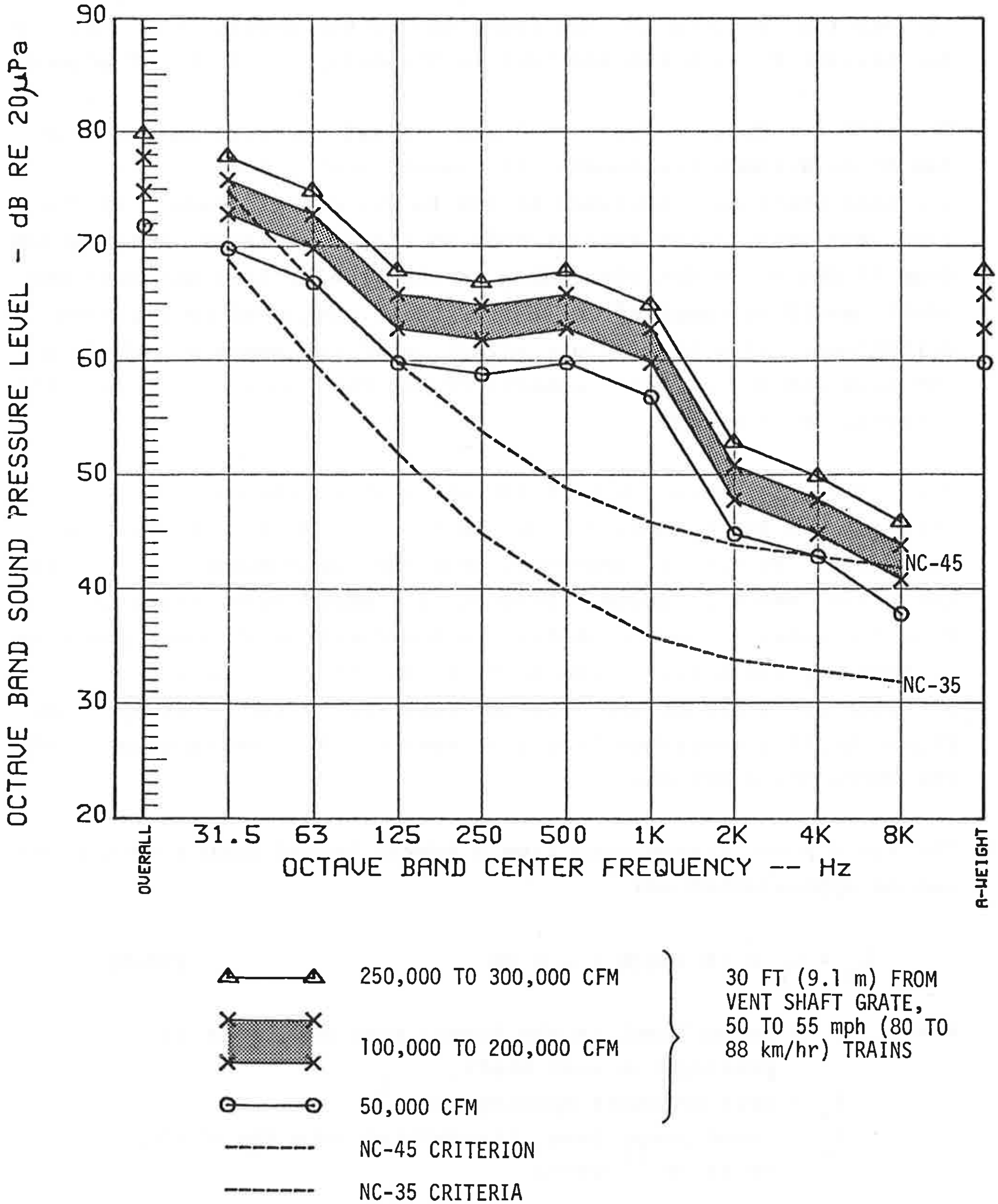


FIGURE 10.11 WMATA DESIGN CURVES FOR MAXIMUM EXPECTED TRAIN NOISE FROM VENT SHAFTS

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

10.1.8 Measurements of Fan Noise at Station Platform

To evaluate patron exposure and to develop methods for predicting fan noise levels in stations, noise measurements were taken inside the BART Berkeley Station, with the fans operating in both supply and exhaust modes. These are the same fans that were discussed above. Measurements were taken in 1969, when there was no acoustical treatment in the station, and in 1975, after acoustical treatment had been installed on the walls and ceiling. The treatment effectively reduced reverberation time and noise levels inside the station.

Table 10-4 indicates that only small changes in the fan noise occurred at the exit grate. This implies that operating conditions during the two periods measured were relatively similar. Table 10-5 shows the measurement results inside the station. At the south end of the platform, closest to the fan rooms, the noise levels differ by about 4 dBA, probably because the tunnel leading away from the station was blocked off in 1969, but open in 1975.

Table 10-6 shows the reverberation times measured in the central area of the station platform during the two measurement periods. The reverberation times were substantially reduced after the treatment. The effectiveness of the absorption treatment is evident when comparing the measured fan noise levels for the two sets of measurements at 30 m and 60 m from the platform end and at the platform center. Figure 10.12 shows the added attenuation due to station treatment. At 30 m from the south end, an average reduction of about 6 dBA was obtained. At the platform center, a reduction of 13 dBA was measured.

TABLE 10-4 AVERAGE CHANGE IN A-WEIGHTED LEVELS BETWEEN 1969 AND 1975 MEASUREMENTS

Location	Average Change (1975 minus 1969)
Fan shaft exit grate	+0.5 dBA
9 m west of grate	+3.25
9 m east of grate	-2.25
9 m south of grate	+1.75
Fan room	+2.00
At damper into tunnel	+0.75
South end of tunnel	-3.25
30 m from south end	-9.25
Platform center	-16.25

TABLE 10-5 NOISE MEASUREMENTS INSIDE STATION

Location	North Fan Before Treatment	North Fan After Treatment	South Fan Before Treatment	South Fan After Treatment
Fan in Exhaust Mode				
Fan room	104 dBA	105 dBA	105 dBA	106 dBA
At damper in tunnel	102	101	103	102
South end of platform	94	90	95	91
30 m from south end	89	80	91	78
60 m from south end		72		72
Platform center	81	66	83	64
30 m from north end		58		59

TABLE 10-6 SUMMARY OF REVERBERATION TIME MEASUREMENTS AT STATION PLATFORM AREA

Octave Band (Hz)	Reverberation Time (sec) Before Treatment	Reverberation Time (sec) After Treatment
125	14.0	1.5
250	13.0	1.3
500	9.5	1.3
1000	7.0	1.6
2000	4.7	1.5
4000	2.8	1.4
8000	1.0	1.0

TABLE 10-7 RANDOM INCIDENCE END CORRECTIONS FOR A TYPICAL SUBWAY STATION*

Octave Band (Hz)	End Correction (dB)
63	0
125	2
250	6
500	9
1000	10
2000	10
4000	10
8000	10

*Assuming the station can be modeled as a large duct

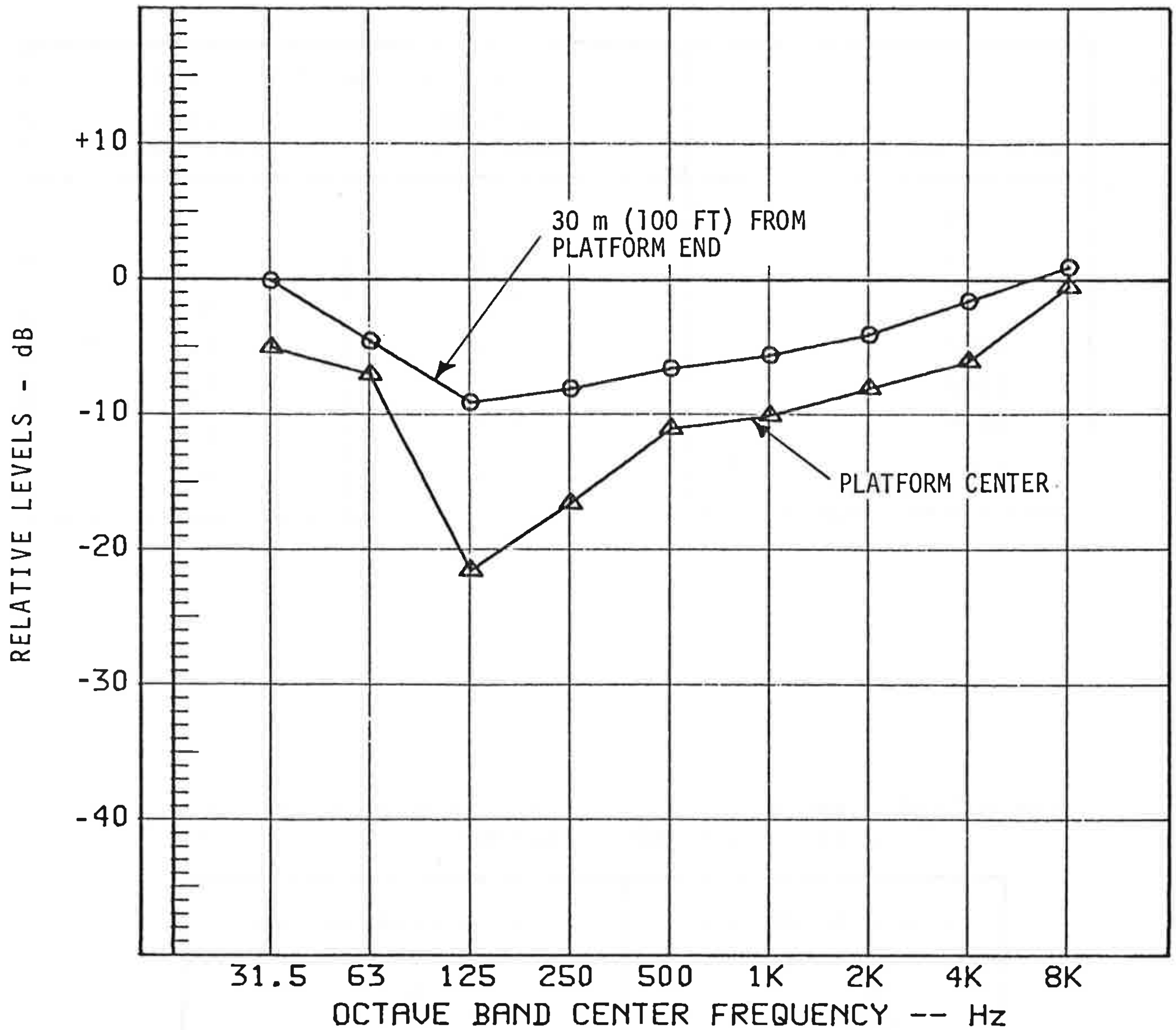


FIGURE 10.12 ADDED ATTENUATION DUE TO STATION ACOUSTICAL TREATMENT

The data from these measurements can be used to evaluate various means of estimating station noise levels caused by fan noise originating at one end of the station. To some degree, the typical transit system station is like a long tube and also like a large reverberant room. According to the statistical theory of sound reverberation, it does not really fall within either category; therefore, some of the common methods for making simple sound level estimates in a room are not appropriate. Hence, it is necessary to confirm the various room analysis methods.

First, consider the station as a reverberant room. The sound power that goes into the reverberant field will be the power into the station minus an end correction due to the random incidence of the noise sources. The random incidence end correction is different from the end reflection loss caused by termination impedance between the subway tunnel and station opening. For the size of the subway tunnel and station opening, the end reflection loss is negligible in the frequency range of interest. The random incidence end corrections for the typical subway station (assuming that the station can be modeled as a large duct) are given in Table 10-7. The reverberant level is given by:

$$L_p = L_w + 10 \log(4/S\bar{\alpha}) - \text{end correction} \qquad (10.8)$$

where S = total area of all room boundary surfaces, in m^2,
$\bar{\alpha}$ = average absorption coefficient of the room surfaces,
L_p = reverberant sound level, dB re 2×10^{-5} Pa

The denominator of the argument of the logarithmic term is related to the reverberation time of the room by:

$$S\bar{\alpha} = 0.161\ V/T \qquad (10.9)$$

where V = volume of room, in m^3
T = reverberation time, in sec

Combining the two relationships gives:

$$L_p = L_W + 10 \log(24.8\ T/V) - \text{end correction} \qquad (10.10)$$

The levels measured at the platform center and the calculated reverberant levels are illustrated in Figure 10.13. The reverberant model gives reasonable results for the level at the center of the untreated station and relatively poor results for the level in the station after treatment.

Next, consider the station as a long tunnel. The attenuation, L, over a length of tunnel d can be approximated as:

$$L = 60d/cT = 0.18d/T \quad \text{(dB/m)} \qquad (10.11)$$

where: T = measured reverberation time, sec
c = speed of sound in air (340 m/s)

This analysis is a one-dimensional theory which takes into consideration the propagation of sound waves along the tunnel and the decay of sound from absorption by the tunnel surface.

Since the level at the platform end was measured very close to the tunnel opening, an adjustment must be made to approximate the complete spreading of the sound energy over the station cross section. The appropriate correction is approximately 5 dB. Figure 10.14 shows the attenuation of the A-weighted level down the platform relative to the adjusted measured level at the platform end. Also indicated is the attenuation predicted using the given formula and the measured reverberation times.

The results in Figure 10.14 indicate that the noise level continuously decreases as a function of distance from the noise source. This explains why the reverberation room theory is not appropriate, especially for the treated stations. In a reverberant room, the noise level remains constant beyond a certain

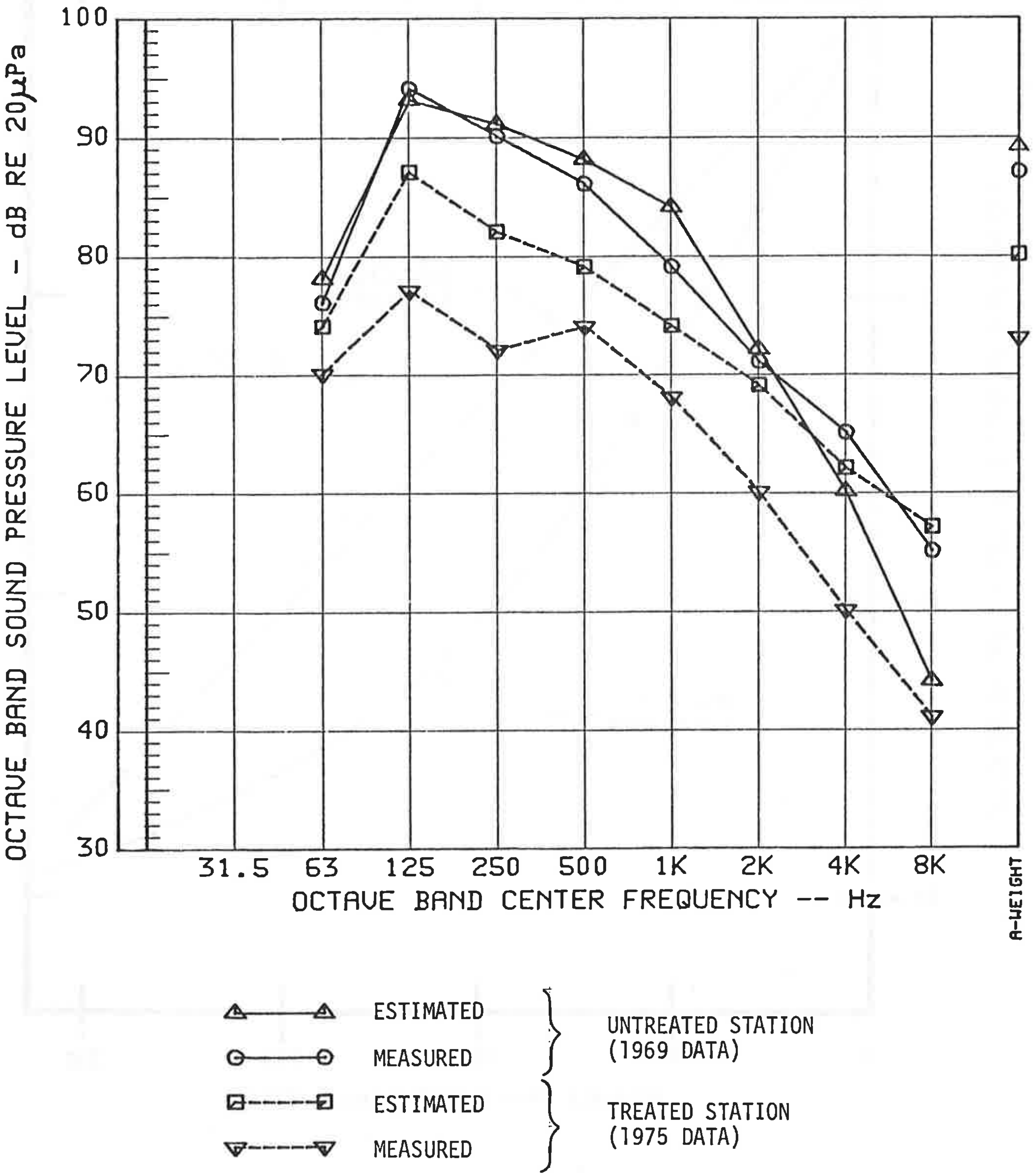


FIGURE 10.13 COMPARISON OF ESTIMATED REVERBERANT LEVELS AND THE MEASURED LEVELS IN THE STATION CENTER

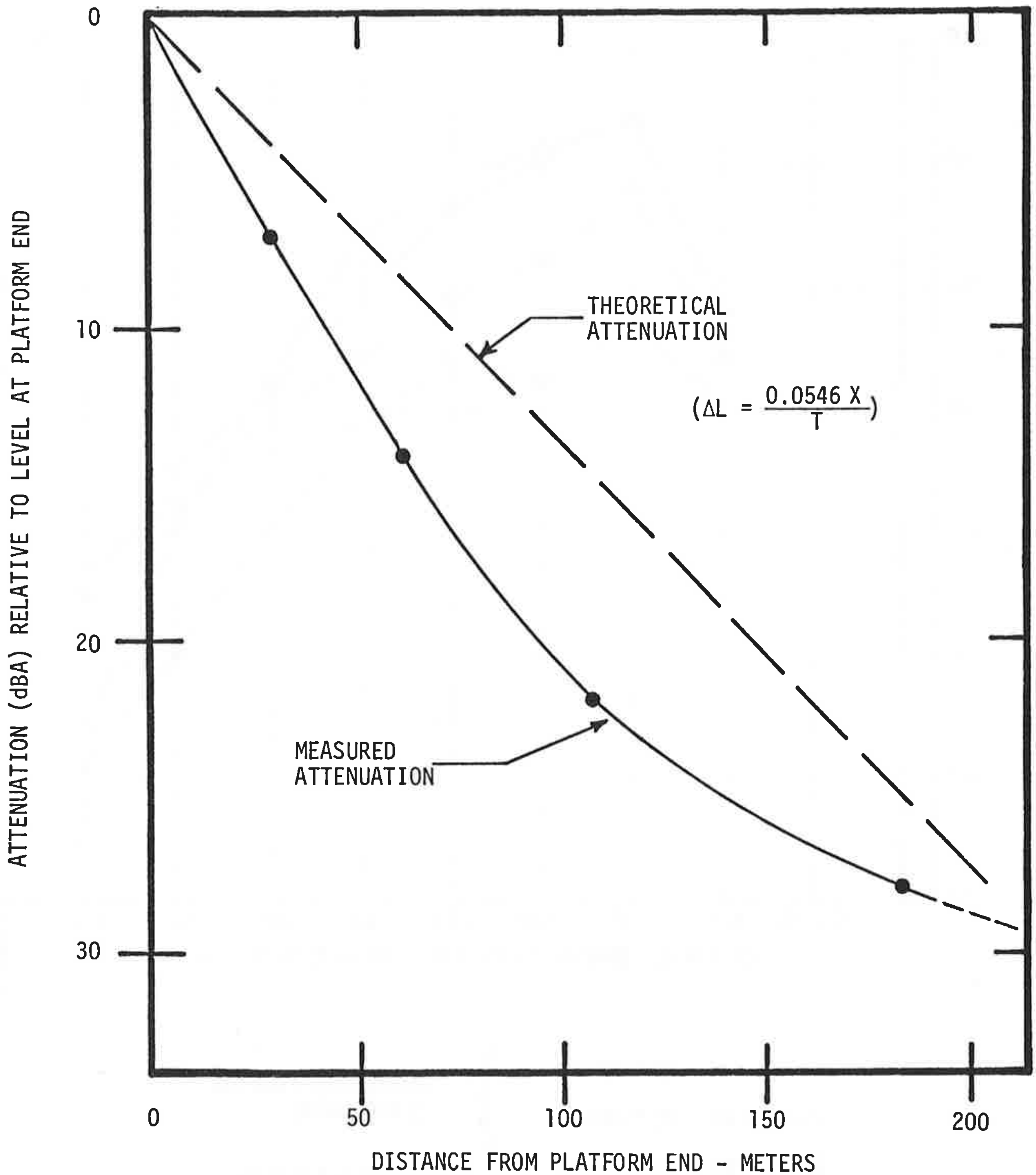


FIGURE 10.14 A-WEIGHTED ATTENUATION OF FAN NOISE ALONG PLATFORM AT BART BERKELEY STATION

distance from the noise source.

It may also be noted that the long tube model of the station does not show the same results as calculated. Except at the platform end opposite the fan rooms, the measured attenuation was higher than the predicted attenuation. The data show that the attenuation down the first half of the station had the general form that would be expected from the long tube theory, that is, $L = Kd/T$. However, the data indicate that the value of K is approximately 0.25 instead of the theoretical 0.18. To some degree, the discrepancy can be accounted for by obstructions (such as stairways) and by the reflection of sound energy from the opposite end back toward the fan room end of the station. As a result of the influence of end reflections, the attenuation becomes nonlinear beyond the center of the platform, thus differing from the tube theory. The reverberant sound theory would most likely give better approximation for distance from the noise sources beyond the middle of the platform.

10.2 SUBSTATIONS

Transformer hum and cooling fan noise are the two main sources of noise in substations. Because the transformer hum is a pure tone or tonal noise, it is more noticeable and more annoying than broadband noise at the same level. Transformer hum generally consists of component frequencies equal to multiples of the fundamental, which equals twice the supply frequency. On 60 Hz lines, the fundamental is 120 Hz. In smaller transformers, harmonics up to 1200 Hz can be detected. The dominant tones of the large transformers used in transit system substations are generally tones at 120 Hz and 240 Hz. However, measurable tones at 60 Hz sometimes occur.

Location determines a substation's potential for community noise impact. Underground substations rarely cause noise problems, however above-ground substations in quiet residential neighborhoods often do. Substations protected only by acoustically transparent fencing require a 150 to 200 m (500 to 700 ft) buffer zone between the substation and noise-sensitive activities or buildings. A buffer zone is not usually an economical method of noise control, unless the substation is located in a rural area where land is relatively inexpensive.

Substations in commercial or residential areas are often surrounded by a high wall or enclosure that blocks the view of the equipment and provides protection. These walls can be designed to provide sufficient noise control in all but the most sensitive areas. However, when the substation is protected only by an acoustically transparent fence, other noise control measures should usually be considered.

10.2.1 Substation Noise Estimates

The primary factors in substation noise are:

- The transformer sound power level and directivity
- The sound power level and directivity of the cooling fans
- The distance between the substation and the receiver
- The presence of barriers, or other obstructions, between the substation and the receivers

Sound power data are rarely published for individual transformers. Instead, NEMA publishes maximum values of A-weighted sound levels for various classes of transformers. These sound levels are specified for a distance of 30 cm (1 ft) from the source. The attenuation of sound level with distance from the transformer must be based on empirical data, up to a distance of approximately 10 m, beyond which a 6 dB attenuation with each doubling of distance can be assumed (Ref. 10.3). That is, beyond 10 m, even a large transformer can be modeled as a point source. Figure 10.11 (adapted from Ref. 10.3) presents empirical data on the attenuation of transformer noise with distance. With the sound level at a given distance from a transformer, the curves on Figure 10.15 can be used to estimate the sound level at any receiver location.

It is possible to obtain quieter transformers, which produce noise levels 15 dBA or more below normal (Ref. 10.3). These "low noise" transformers generally have more iron in their cores which reduces the flux density. The result is a larger, more expensive unit. Barriers or enclosures are usually cheaper.

The cooling fans at transit system substations often make more noise than the transformers. Sound power level specifications can usually be obtained from fan manufacturers, which help in estimating the noise from proposed substations. The method suitable for estimating the noise from cooling fans at a given receiver location depends on the specific configuration of the substation. Here are some basic configurations and simple methods that can be used:

- Fans in the open: assume uniform nondirective sound radiation.
- Fans in the open with a wall or other obstruction between the fans and the receiver: assume nondirective sound radiation, with an extra attenuation factor based on standard estimates of barrier attenuation (see Appendix A).
- Fans in an enclosed courtyard with walls on all sides: same as second configuration.
- Fans in an enclosure: the sound power radiated via fan dampers or other openings should be based on reverberant room theory. Normally, much of the sound power will be absorbed by the surfaces of the enclosure and only a fraction radiated from the enclosure openings. For the purposes of estimating the sound level at a receiver position, the sound radiated from enclosure openings can be assumed to spread uniformly over a hemisphere (see Appendix A).

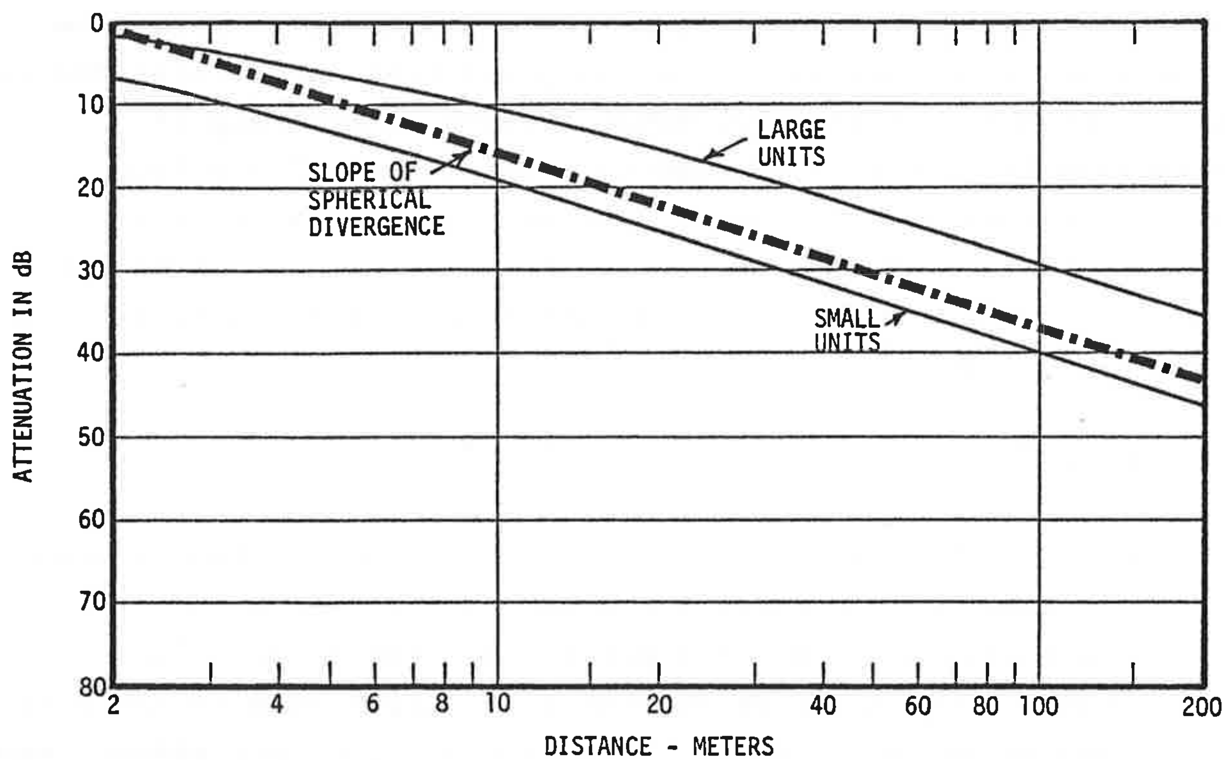


FIGURE 10.15 TRANSFORMER NOISE ATTENUATION WITH DISTANCE (ADAPTED FROM REF. 10.3)

10.2.2 Measurements of Substation Noise

The results of noise measurements at several WMATA above-ground power substations are reported in Reference 10.12. The three substations measured all have similar designs consisting of a transformer courtyard, surrounded on three sides by a masonry wall approximately 5.5 m high and bordered on the fourth side by a switchgear building. No roofs cover the courtyard areas in which the transformers are installed. The courtyard entrances are secured with acoustically transparent gates constructed of steel bars.

Although noise measurements were made at three substations, substation noise could only be clearly distinguished from background noise at two of the substations. The measured A-weighted levels at these two substations, as well as projected levels, are presented in Table 10-8. Note that the projected levels are significantly higher than the measured levels. The forecasts were based on the maximum allowable sound levels for Type OA transformers, as specified in NEMA Publication TR1, "Transformers, Regulators and Reactors". Since the measured levels are lower than the estimates of maximum levels, the transformers in the two WMATA substations emitted less noise than the NEMA specifications allow.

Figure 10.16 presents 1/3 octave band spectra for the noise measured at the Fort Totten/Tacoma substation. Significant pure tones (characteristic of transformer hum) at 60, 120, and 240 Hz were present in the spectra at the courtyard entrance. At 9 m (30 ft) from the structure, the pure tones are audible components of the substation noise but cannot be identified in the 1/3 octave band spectra. This indicates that the courtyard walls and switch building are effective at controlling these relatively low-frequency components. At 9 m from the substation structure, transformer hum is comparable with or lower than the cooling fan noise and general background noise.

TABLE 10-8 COMPARISON OF PREDICTED AND MEASURED A-WEIGHTED SOUND LEVELS OF WMATA TRACTION POWER SUBSTATIONS

Substation	*Forecasted Level at 9.1 m	Measured Levels - dBA		
		9.1 m From Solid Wall	9.1 m From Courtyard Gate	9.1 m From Front of Switchgear Building
Brookland/Fort Totten (3 transformers)	54 to 56	45	47	52
Fort Totten/Tacoma (2 transformers)	52 to 54	43	49	51

*For 9 m (30 ft) from solid courtyard walls, corrected for number of actual transformers installed. Forecast was for maximum level of transformer noise that could be expected.

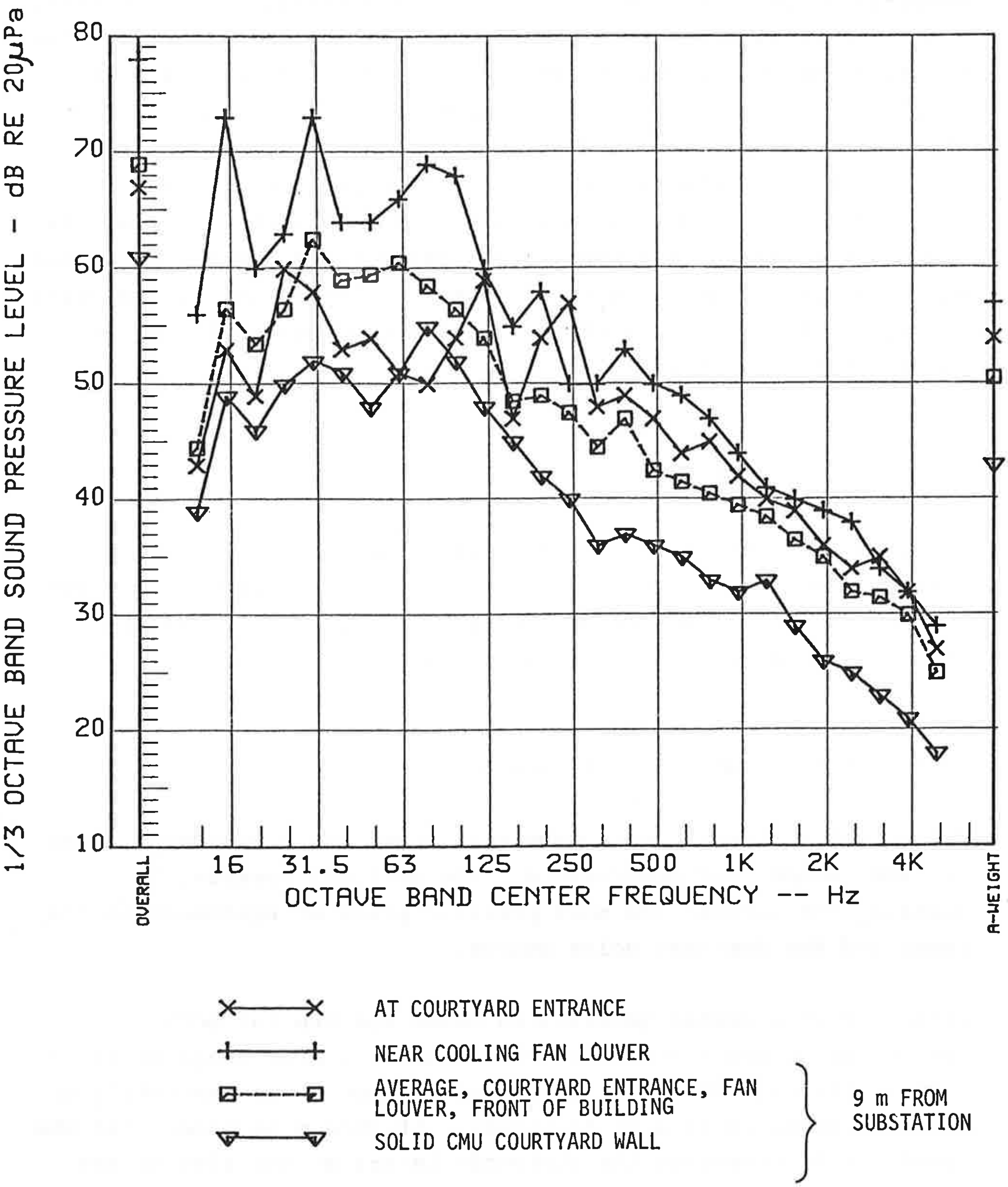


FIGURE 10.16 NOISE LEVELS AT WMATA METRO FORT TOTTEN/TACOMA SUBSTATION WITH TWO TRANSFORMERS IN PLACE

In general, the measurements of the WMATA substations showed compliance with both local noise ordinance limits and WMATA Metro design criteria. The only exceptions were the noise radiated from the front of the structures through the courtyard gate and the cooling fan louvers. Assuming appropriate orientation of the structures and reasonable distance from nearby property lines, noise from the front of the structures complies with local ordinances even though it exceeds the WMATA design criteria. At the two substations measured, the cooling fan louvers are located on the exterior wall of the switchgear building. This noise would be significantly reduced if the cooling fan louvers faced the inside of the courtyard.

10.3 CHILLER PLANTS

If careful attention is paid to chiller plant location, orientation, and design, noise impact problems can be prevented. The two sources of noise from chiller plants are the mechanical equipment rooms and the cooling towers.

10.3.1 Mechanical Equipment Rooms

Mechanical equipment rooms generally house noisy equipment, such as fans, pumps, and compressors. The chiller, however, is normally the largest and most powerful piece of equipment in the rooms and the dominant noise source.

Figure 10.17 presents generalized noise spectra for both reciprocating and centrifugal chillers, at a wide range of capacities. The data shown are based on the upper 50th percentile of machines measured in a field survey. It should be noted that the sound levels represent the state-of-the-art at the time of the survey (1972); quieter machines are now available.

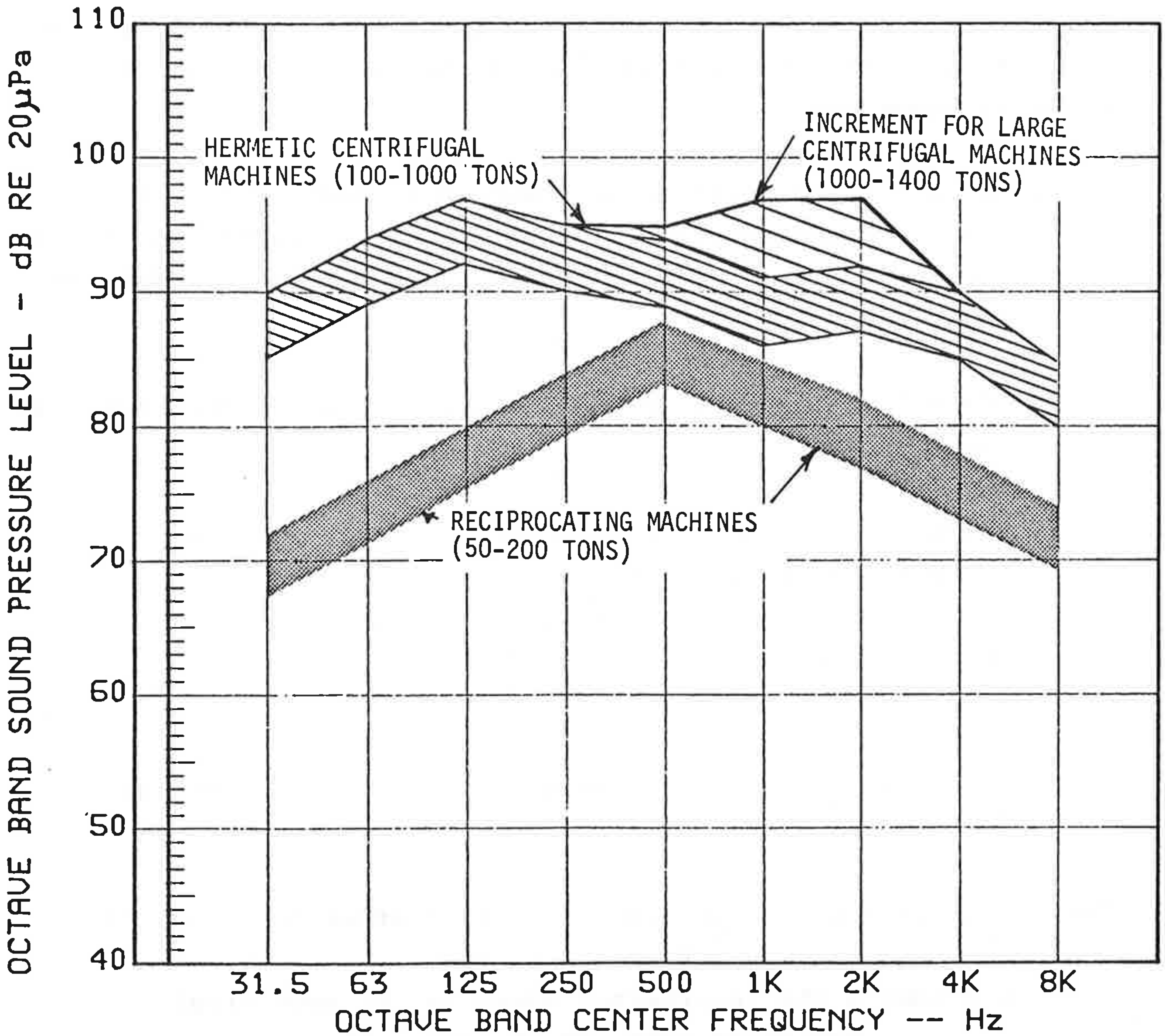


FIGURE 10.17 GENERALIZED SPECTRA FOR CHILLERS

The following general procedure should be used to determine the impact of noise radiated from mechanical equipment rooms. This impact includes the noise heard by patrons or transit personnel in station areas near the mechanical equipment room, as well as the noise radiated from ventilation louvers or other openings to community locations.

1. Obtain sound pressure or sound power level data for whatever mechanical equipment will be a significant source of noise. Major manufacturers now supply direct data for most equipment.

2. Calculate the reverberant sound level inside the equipment room.

3. Estimate the sound level at the receiver position. For an opening in the wall of area S, the sound level at a distance r from the opening (assuming there are no barriers or other obstructions between the observer and the opening) can be estimated as:

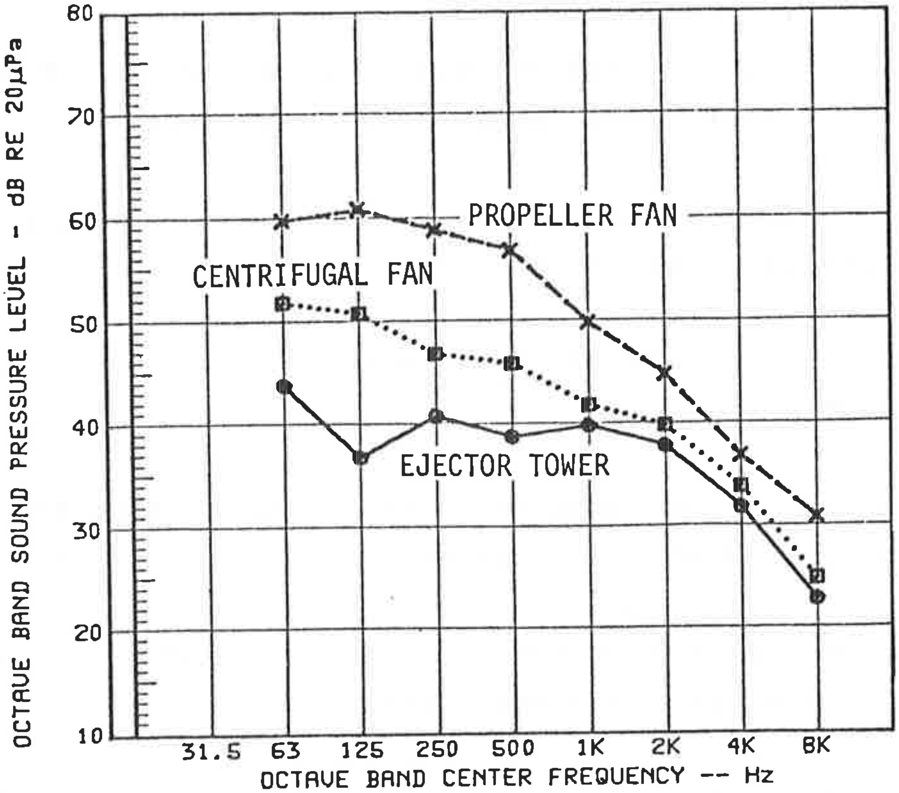
$$L_p(\text{obs}) = L_p(\text{room}) + 10\log(S/8\pi r^2) \qquad (10.12)$$

where L_p(obs) = the sound level at the observer location, dB re 2×10^{-5} Pa

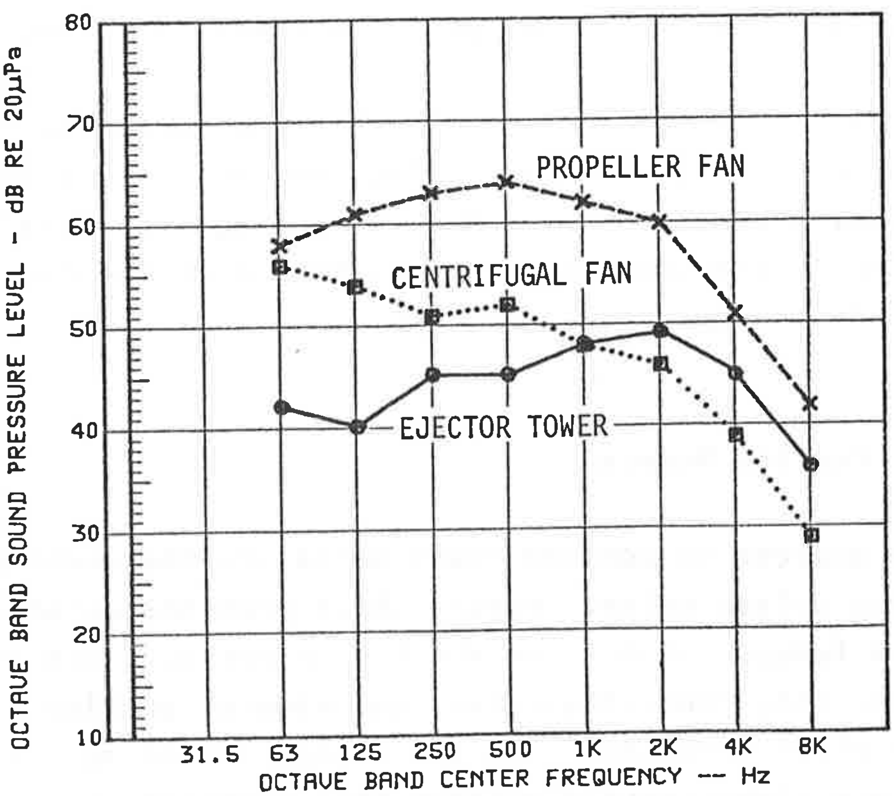
L_p(room) = the reverberant sound in the mechanical equipment room, dB re 2×10^{-5} Pa.

Equation 10.13 assumes that the sound radiates nondirectively from the opening into a hemispherical space.

If either estimates or measurements of mechanical equipment room noise indicate that sound control is required, the following basic methods can be used:



10.18a 30 m FROM FAN OR NOZZLE SIDE



10.18b 30 m FROM ENDS

FIGURE 10.18 SOUND PRESSURE LEVELS 30 m (100 FT) FROM TYPICAL PROPELLER FAN, CENTRIFUGAL FAN, AND EJECTOR COOLING TOWERS

- Install acoustical absorption material to reduce the reverberant sound level inside the room.

- Seal cracks around doors or at other locations with gasket material or resilient, nonhardening caulking material.

- Change the site of the room to a less noise-sensitive location.

- Change the orientation of the room, so that the noise will radiate towards the street.

- Place barriers or baffles between any opening in the equipment room and the receiver.

- Install sound traps or acoustical louvers to reduce noise transmitted through openings in the enclosure.

- Select equipment that produces lower noise levels.

Generally, noise radiated from mechanical equipment rooms to adjacent communities can be controlled through careful site selection, orientation of the room openings to minimize noise radiation to the community, and gasketing of the doors of sides with no louver openings.

10.3.2 Cooling Towers

The two sources of cooling tower noise are the cooling fans and the water splash noise. Figure 10.18 presents octave band sound pressure levels (30 m) from the fan or nozzle sides of comparable propeller fan, centrifugal fan, and ejector cooling towers with no noise control treatment. As the figure indicates, centrifugal fan towers are significantly quieter than propeller fan units. In centrifugal and propeller fan units, the fan noise typically dominates the noise levels except at high frequencies.

The ejector unit is a relatively new configuration that does not use a fan. Ejector units make less low-frequency noise than centrifugal fan units, but their high-frequency noise level is comparable to that of centrifugal fan units. Sound barrier walls, which effectively reduce higher frequency noise, can be used to effectively overcome this problem.

The noise levels expected from cooling towers should be based on sound level data supplied by the manufacturer. Typically, information is supplied at distances of 1.5 m (5 ft) and 15 m (50 ft) from the sides and top of the cooling tower. Data on the directivity of cooling tower noise is also important. Most towers are somewhat directive, with the highest noise levels radiated from the fan or nozzle sides. By knowing the directivity of the radiated sound, the tower can be oriented so that the highest levels of sound are directed toward the least noise-sensitive areas.

Cooling tower noise can be controlled by the following techniques:

- Select sites, such as ones near major streets or highways, that are not noise-sensitive.

- Install parapets or barrier walls high enough to block the direct line-of-sight of the cooling tower from receiver locations.

- Use packaged attenuators in the air inlet and exhaust openings. Packaged attenuators are available from cooling tower manufacturers and some muffler manufacturers. It is not usually possible to install attenuators on ejector units.

- Reduce motor speed when the cooling load is reduced. Operating fan towers at one-half speed will produce a 6 to 9 dB reduction near the blade passage frequency of the fans. Although this may not be a suitable solution for transit applications, under certain conditions (e.g., at night when reduced cooling requirements will match increased noise sensitivity), it can be effective.

CHAPTER 10 REFERENCES

10.1 "Sound and Vibration Control," ASHRAE Handbook & Products Directory, 1980 Systems Handbook, Chap. 35, (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., NY, 1980 - Updated Periodically).

A very good reference on the control of noise from air-conditioning systems and mechanical equipment installations. It contains a number of summary tables and examples that can be applied to outdoor and indoor noise control and includes vibration isolation of mechanical equipment, and a particularly valuable reference for the acoustical design of duct systems.

10.2 C. M. Harris (Editor), Handbook of Noise Control, 2nd Edition, (McGraw-Hill Book Publishing Company, NY, 1979).

This book is a collection of chapters on a wide variety of topics in noise and vibration control, oriented towards practical aspects. Each chapter was prepared by a different author. The chapters relevant to Section 10 of this Handbook follow.

J. B. Moreland, "Electrical Equipment," Chapter 25.

This chapter includes a brief discussion of transformer noise relevant to substations, based on Chapter 29 of the Handbook of Noise Control, 1st Edition, Reference 10.3

J. B. Graham, "Fans and Blowers," Chapter 27.

This chapter describes different types of fans and suggests procedures for estimating the sound power level that various fans radiate to an attached system.

H. F. Kingsbury, "Heating, Ventilation, and Air Conditioning Systems," Chapter 28.

This chapter discusses the propagation of noise in the ducts of air handling systems and the control of noise in duct systems.

10.3 C. M. Harris (editor), Handbook of Noise Control, 1st Edition, (McGraw-Hill Book Publishing Company, NY, 1957).

The first edition of the Handbook of Noise Control has been a standard reference for many years. The recently published second edition (Reference 10.2) has updated much of the original material, but some of this material was not repeated and is still relevant. Of particular interest is Chapter 29, "Reduction of the Noise of Iron-Core Transformers and Chokes," by A. S. King. The discussion of transformer noise in this chapter is significantly more detailed than in the second edition.

10.4 L. L. Beranek (editor), Noise and Vibration Control, (McGraw-Hill Book Publishing Company, NY, 1971).

This book is a valuable reference in acoustics, but the presentations are generally oriented toward technical and mathematical aspects. The book is not well-suited to the novice in acoustics. Of interest for the control of noise in ducts is Chapter 12, "Mufflers," by T. F. W. Embleton, which contains a detailed, mathematically oriented discussion of muffler design, duct and plenum linings, and other methods of controlling noise in ducts.

10.5 G. P. Wilson, "Acoustical Design Criteria for Subway Fan and Vent Shafts," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority Metro System, March 12, 1971).

Report prepared for the WMATA system by WIA on the prediction and control of fan and vent shaft noise. The report oovers criteria, expected noise levels, and methods for reducing the noise to acceptable levels.

10.6 Wilson, Ihrig & Associates, Inc., "Acoustical Considerations -- Fan and Vent Shafts," (Prepared for Baltimore Region Rapid Transit System, January 1976, revised June 1976).

WIA report on fan and vent shaft noise control, prepared for the BRRT (Baltimore) System. It is similar to Reference 10.5 except that the discussion is oriented to the shaft configurations planned for BRRT. This report includes data on fan noise in the community and in stations at BART (San Francisco).

10.7 P. Y. N. Lee, "Noise Control for Fan and Vent Shafts in Subways, Noise Control Engineering, 10(3), (Institute of Noise Control Engineering, Poughkeepsie, NY, May-June 1978), pp. 102-107.

A summary of WIA experience at several transit systems, including available methods for reducing fan and train noise propagated out of vent shafts and fan noise propagated into stations. Fan noise measurements on station platforms and outside the fan shafts at existing rapid transit facilities are also included.

10.8 H. Seelbach, F. M. Oran, "What To Do About Cooling Tower Noise," Sound, 2(5), (September-October 1963), pp. 32-41.

Although written 16 years ago, the discussions of cooling tower noise-generating mechanisms and the control of cooling tower noise are still relevant. The article is a good introduction to the subject.

10.9 "The Noise of Cooling Towers," Engineering Manual 251, (Baltimore Aircoil Company, Baltimore, MD, 1971).

This manual presents a sixteen-step approach for the evaluation of cooling tower noise and methods for controlling it. Although the manual is distributed by a manufacturer of cooling towers, it is general enough to apply to all types of cooling towers.

10.10 T. Kihlman, G. P. Wilson, "Tests of Sound Absorption Treatment in WMATA Metro Subway Structures," (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, May 1975) .

A report of testing performed to evaluate the effectiveness of absorption material placed in WMATA Metro running tunnels, vent shafts, and stations. The testing was designed to measure the absorption coefficients of the treatment material and the noise reduction produced by the treatments. One of the main purposes of the testing was to evaluate the in-place absorption of the 25 mm (1") thick Pyrok spray-on sound absorption treatment.

10.11 J. T. Nelson, G. P. Wilson, "Metrorail Operational Sound Level Measurements: Substation Noise Levels," Preliminary Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Washington Metropolitan Area Transit Authority, June 1979).

The results are presented of noise measurements produced by at-grade WMATA Metro traction power substation facilities. Measurements were performed at these substations, all with courtyards surrounded by high (4.5 to 5 m) solid walls. The substations were found to be in compliance with both the local noise ordinances and the Metro design criteria.

CHAPTER 11

11. YARDS AND SHOPS

The noise caused by activities in maintenance or storage yards is somewhat different than the wayside noise of normal mainline transit train operations and the noise of stationary ancillary facilities. Also, ordinance requirements and design criteria are generally different. Thus, the control of yard and maintenance operation noise requires different considerations.

The following are the major sources of yard and shop noise:

- Wheel squeal on curves
- Clicks and pings as wheels pass over joints and through switches
- Train rolling noise
- Transit car auxiliary equipment operations
- Coupling and uncoupling of cars
- Train horns
- Impact tools and machinery
- Shouting workmen
- Car washes
- Other noise created by maintenance work
- Telephone or warning buzzers, horns, announcement or call loudspeakers and cross gate warning bells.

Two additional sources of noise that are not included in the above list are brakes squealing and air release (frequently encountered with air brakes or dumping cycles of air compressors and air brake systems). These sources do not usually generate significant noise on modern transit vehicles because of the use of quiet operating brakes or systems which do not require dumping of air in the operating cycle; thus, the characteristic air release sound is eliminated. On those systems which do use air release, the air valves or exhaust pipes should be equipped with mufflers to limit the noise to levels comparable to or lower than the noise from the other auxiliary equipment, such as fans and blowers. The levels will then be in accordance with the noise limit specifications applied to auxiliary equipment on the trains.

Community noise impact from yard and shop operations can be limited through control of yard operating procedures, design of yard equipment, use of buffer zones and sound barriers, and through controls on the noise created by the transit vehicles and equipment. With the appropriate noise abatement procedures and techniques and with modern transit vehicles, a transit system storage or maintenance yard can be acoustically acceptable to nearby community areas.

An important consideration in the placement and design of yards is the zoning of nearby properties. For example, placement of yards in industrial areas has the obvious advantage of minimizing noise intrusion. However, this natural protection is assured only if the surrounding property is developed or maintained compatibly with the yard.

Figures 11.1 and 11.2 are photographs of the TTC Davisville Yard site when it was first built and after twelve years of development. Initially, this yard was in an open/light industrial area and produced no noise impact. However, high rise residential development directly adjacent to the yard resulted in many complaints about the noise impact. A plan to solve the problem that has received much attention is the building of a development over the yard, which would completely cover the storage tracks and contain the noise. However, no action has been taken on this plan. The point is that yard planning and design should include consideration of adjacent land zoning; also, the continued operation of a yard should include participation in the local area planning process in order to ensure compatible development of adjacent properties.

FIGURE 11.1 TORONTO TRANSIT COMMISSION DAVISVILLE YARD SITE - JANUARY, 1953

FIGURE 11.2 TORONTO TRANSIT COMMISSION DAVISVILLE YARD SITE - MAY, 1965

11.1 YARD ACTIVITY

The principle noises which create impact in residential or other noise-sensitive areas near transit system yards are the wheel/rail squeal on curves and the warning signals when trains move in and out of maintenance buildings. The next level of noise impact is caused by shouting workmen, telephone or warning buzzers or horns, and announcement or call loudspeakers. These sources produce randomly occurring noises which are considerably different than typical community background noise and which can be very intrusive, even if they do not significantly exceed the ambient community noise levels. Most of the noises produced by the newer transit vehicles are controlled (due to the specification requirements for wayside noise, in-car noise, and subway station platform noise) to a level that will avoid impact on adjacent areas, unless the separation distance from the yard and the noise-sensitive area is very small. For older cars, the noise from the auxiliary equipment may be significant and require consideration in any plans for noise control.

All auxiliary equipment on new transit cars should be required to meet a specification of 68 dBA maximum at 4.6 m (15 ft) from each individual item. With all equipment operating, the typical maximum allowable noise level should be 60 dBA at 15 m (50 ft) from the center of the vehicle. With typical older vehicles, air compressors and other items which operate either constantly or cyclically can produce noise levels as high as 75 to 80 dBA at 4.6 m (15 ft) from the car. With some vehicles, the level of air release noise is even higher. The noise limit specifications on auxiliary equipment for new transit vehicles usually eliminates these noises as sources of impact in the community near transit system yards. Thus, dealing with car auxiliary equipment noises in the design of a new yard requires consideration of car age and equipment characteristics. For existing yards, modification of car equipment may be a viable procedure for reducing noise transmitted to the community. Alternately, sound barriers or buffers may be the appropriate modification, depending on car equipment characteristics and yard location.

Controlling the noise from train horns, telephone or warning buzzers and announcement or call loudspeakers can be accomplished through design of the yard facilities and operating procedures. The use of light signals in lieu of audible signals and the confinement of announcements or calls to the vehicle radio and public address systems can eliminate much of this noise.

Often, because of safety requirements, it is not possible to eliminate the use of train horns as warning signals when trains move in and out of yard and maintenance buildings. Through the use of visual panels in the doors and walls and the use of specific operating procedures, it may be possible to minimize the use of train horns as warning signals. It is also possible to minimize some of the annoying quality of the train horns by changing the type of noise produced. The use of airhorns or raucous electronic horns or buzzers should be avoided. These are not necessary to provide adequate audibility of a warning signal. A melodious horn sound, designed to be relatively nonintrusive in neighboring areas, can be used to provide adequate audible warning signals. A good example is the warning signal used by BART trains.

In a railyard, squeal noise at short-radius curves or turnarounds and the impact noise when trains pass over rail joints or through switches can create annoyance. The squeals, pings, and clicks created by the wheel/rail interaction on the yard tracks can generate high levels of sound. Squeal levels as high as 80 dBA at 60 m (200 ft) have been reported for operation on 60 m (200 ft) radius curves. Methods available for reducing or eliminating these noises include: the use of sound barrier walls, retaining walls, berms, or natural topography around the yard tracks to provide shadowing barriers between the rail operations area and the adjacent community; the use of resilient wheels or damped wheels on the trains to reduce or eliminate the squeal noise at the source; and the use of rail lubrication on curved sections of track. The methods that can be used to control squeal noise;

are discussed in more detail in Chapter 12.

Buffer space, retaining walls, sound barrier walls, berms or natural topography barriers and building barriers provide general means for reducing noise transmitted to the adjacent community. Table 11-1 indicates typical train noise for various operating conditions and distances. The levels indicated can be used for comparison with noise ordinance requirements or design criteria to ascertain the need for noise control measures.

In the design of yard noise control, the location and layout are usually the most important items in establishing the community noise standards. Thus, in most cases, yard layout and sound barriers can be used to adequately control noise. In some cases, buffer space may also be needed. As mentioned above, the land use zoning and potential developments on adjacent properties must be considered in determining the layout and design of a yard. Note that if sound barrier berms or topography are used to reduce yard noise, they will have the additional benefit of reducing the visual impact of the yard.

Note that in the design of noise control barriers, the typical car noise characteristics are favorable for use of barrier walls or berms. Usually, all of the significant sources are located beneath the car so that the source height can be considered to be 0.5 to 0.75 m (1.5 to 2.5 ft). The maximum levels, in terms of loudness, are generally in the 500 or 1000 Hz octaves - or even higher frequencies - so that low, lightweight barriers are effective. Because yard tracks are generally ballasted, there is no need for use of absorption materials on sound barriers.

11.2 SHOP ACTIVITY

The noise from maintenance activities and work on the cars can, of course, be controlled by confining the activities to the interior of the inspection and maintenance shops. Therefore, noises from

TABLE 11-1 NOISE LEVELS FROM 2-CAR TRAINS OPERATING ON YARD TRACKS

Noise Source	Distance from Track Centerline 15 m (50 ft)	30 m (100 ft)	90 m (300 ft)	180 m (600 ft)
Train Stationary				
Auxiliaries operating	61 dBA	57 dBA	47 dBA	41 dBA
Train Moving at 30 km/hr*				
No shielding	70	66	57	51
1.2 - 1.4 m (4 - 4.5 ft) barrier	62	58	49	43
Deep cut shielding	55	51	42	36
Maximum Wheel Squeal 60 to 90 m (200 ft to 300 ft) radius curve				
No shielding	92	88	79	73
1.4 - 1.5 m (4.5 - 5 ft) barrier	77	73	64	58
Deep cut shielding	72	68	59	53

*When traversing trackwork with frogs and switches, the wheel impact noise results in noise levels about 6 dBA higher than shown in the table.

impact tools, machinery, and the PA system can be confined inside the buildings. In general, because the maintenance activities are carried out inside the shops, the actual work on the cars, including inspection and cleaning of the cars, does not contribute to the community noise impact of the yard operations. Note that the control of noise within shop buildings should include use of absorption materials on the building interior surfaces for reduction of reflected and reverberant noise.

11.3 CAR WASHES

It is normal for car washing equipment to be completely enclosed inside a building. Car washing equipment can create high noise levels and is likely to cause community impact if it is not enclosed. Therefore, when designing car washes, a complete enclosure should be provided. In addition, in order to lower the equipment noise levels inside car wash buildings and to provide a more acceptable environment for workers, sound absorption materials should be used on the building interior. Sound absorption treatment will reduce the noise inside the building as well as noise transmitted through openings to the neighboring community.

It is, of course, necessary to use sound absorption materials that are water resistant, such as sound-absorbing structural masonry units or sound-absorbing structural tiles.

Air blowers used to strip water from a car can be particularly noisy and the design or purchase documents for such equipment should include specification of maximum permissible noise levels. Such criteria should be based on industrial noise exposure criteria for completely enclosed units and on community noise limits for exposed units.

CHAPTER 12

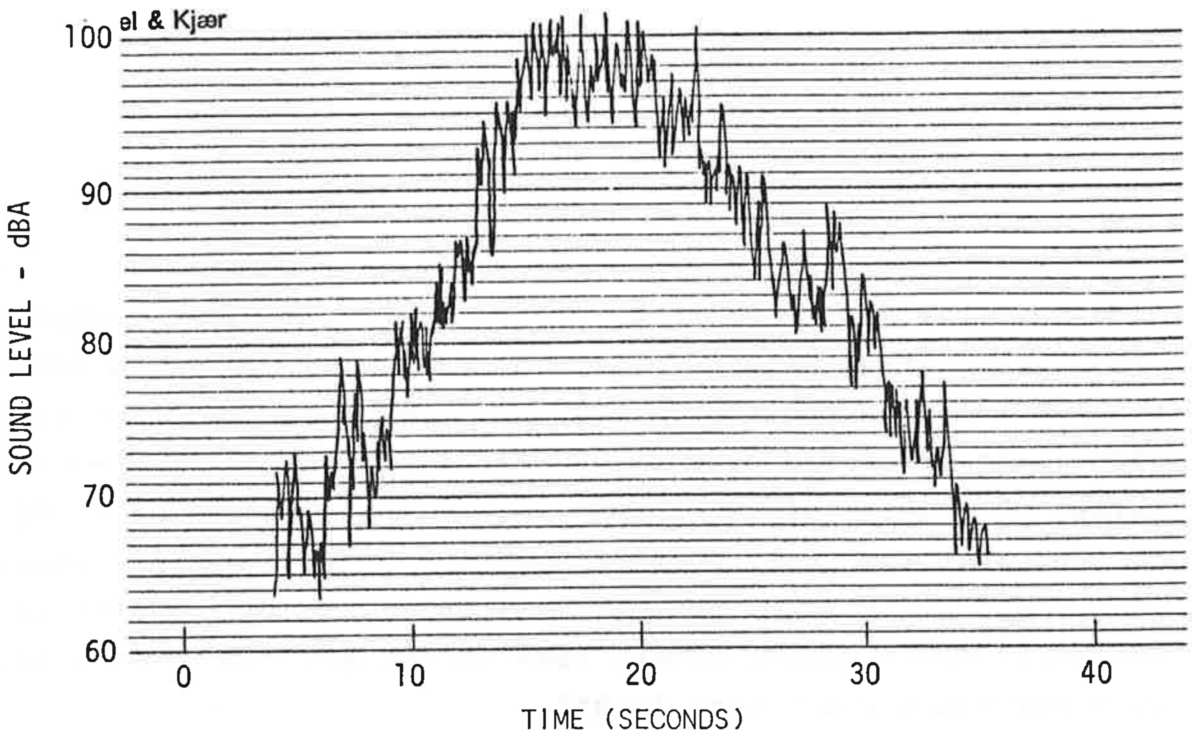
12. WHEEL SQUEAL

Wheel squeal is a phenomenon of rail transit that can be particularly irritating to patrons, communities, and transit employees. Usually occurring only at short-radius curves, it is an intense tonal noise, ranging from a high pitched squeal to a low pitched growl. It is caused by wheel surfaces rubbing or sliding on the rails. A common rule of thumb is that wheel squeal will not occur if the curve radius is greater than 100 times the truck wheel base. This rule seems to work well in most cases, but there are instances where squeal occurs even though the radius is greater than 100 times wheel base length.

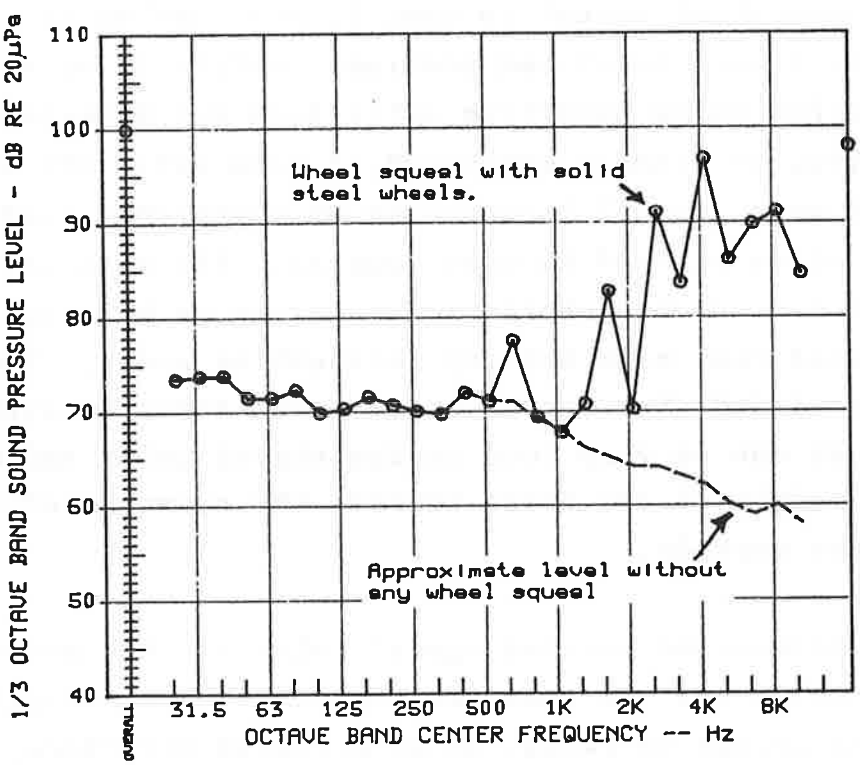
Because wheel squeal is usually confined to short-radius curves, it is a very localized problem. Difficulties arise if residential or other noise-sensitive activities are situated near short-radius curves, if transit employees in yard areas are exposed to significant wheel squeal from trains on turnaround curves or if patrons are often exposed to wheel squeal. The most severe problems are encountered when short-radius curves on heavily traveled lines are located near high density residential areas. The situation is intensified when the curve is on an elevated structure where the squeal can be very loud on the street below and where, because of the height of the noise source, the noise is efficiently radiated to the wayside.

The mechanisms causing squeal noise and the methods of controlling the noise are the same, whether the squeal originates at turnaround tracks in yards, on an elevated structure, an at-grade track, or in a subway.

Figure 12.1 illustrates a typical example of wheel squeal. This example, from the SEPTA system in Philadelphia, illustrates a train with standard, solid, steel wheels traversing a 43 m (140



12.1a SOUND LEVEL AS TRAIN PASSES BY



12.1b 1/3 OCTAVE BAND SPECTRA

FIGURE 12.1 WHEEL SQUEAL EXAMPLE, 2-CAR TRAIN, SEPTA 69th STREET TURNAROUND, 7.5 m (25 FT) FROM TRACK CENTERLINE ft) radius turnaround at the 69th Street Station. Figure 12.1a presents the A-weighted level at 7.5 m (25 feet) as the train passes by. In this example, the squeal noise is relatively steady, reaching a maximum level of nearly 100 dBA. Squeal noise is often highly variable, fluctuating over a range as great as 20 dBA. The frequency of wheel squeal noise often fluctuates as different resonance modes of the wheels are excited. The 1/3 octave band spectra for the wheel squeal is presented in Figure 12.1b. The figures represent the actual levels measured and the approximate levels that would occur if there were no squeal noise. The curves show that in this case, the squeal noise, which is concentrated in the 4 to 6 kHz range, dominates both the overall and the A-weighted noise levels.

Field studies of methods to control squeal noise have shown that the use of damped or resilient wheels can completely eliminate wheel squeal. Other methods, such as rail grinding and wheel truing, can reduce the levels of wheel/rail noise on the tangent track, but have little or no effect on wheel squeal (Ref. 12.1). In some cases, the levels of wheel squeal noise have increased after rail grinding.

12.1 MECHANISMS OF WHEEL SQUEAL

Wheel squeal is generally thought to be created by the slip-stick motion of the wheels on the rails. Most transit cars are supported on two 2-axle trucks. The wheels are rigidly attached to the axles, and the axles rigidly attached to the truck. When the truck goes around a curve, several factors can cause wheel squeal. The first is the difference in the distance traveled by the inner and outer wheels. On a sharp curve, the inner and outer wheels will attempt to roll at different velocities. The resulting differential movement is compensated by one or both of the wheels slipping on the rails, causing wheel squeal, or by elastic deformation at the wheel/rail contact point (Ref. 12.2). Another major cause of wheel squeal is the crabbing of the trucks as they traverse the curve. This is illustrated in Figure 12.2. Since the axles are forced to remain parallel, they can never both lie on the radii of the curve. At least one of the wheel pairs must roll at an angle to the rails, which creates a slippage of the wheel running surface across the rail. Crabbing can also force the wheel flanges into the side of the rail, and this rubbing also creates wheel squeal. Where guard rails are used, squeal can be caused by flanges rubbing against the guard rail, as the wheel is forced around the curve.

A relatively detailed theoretical study of wheel/rail noise by Remmington, et al. (Ref. 12.2) concluded that wheel squeal was primarily caused by the lateral motion of the wheels relative to the rails. In any case, it is clear that the interaction between the wheels and the rails excites resonant modes of the wheels. Present technology cannot yet predict accurately the level of wheel squeal that will occur for a specific set of conditions.

12.2 METHODS OF CONTROLLING WHEEL SQUEAL

Four general approaches can control wheel squeal:

- Reduce the energy, created by the wheel sliding or rubbing on the rail, by lubricating the rail surface.

- Absorb the vibrational energy before it is radiated by applying damping treatment or dampers to the wheels or using resilient wheels.

- Block the sound energy before it reaches the receiver. Community areas can usually be protected by sound barriers. The squeal noise inside the cars can be reduced by increasing the sound insulation of the car body or by placing absorption material in the undercar area near the wheels.

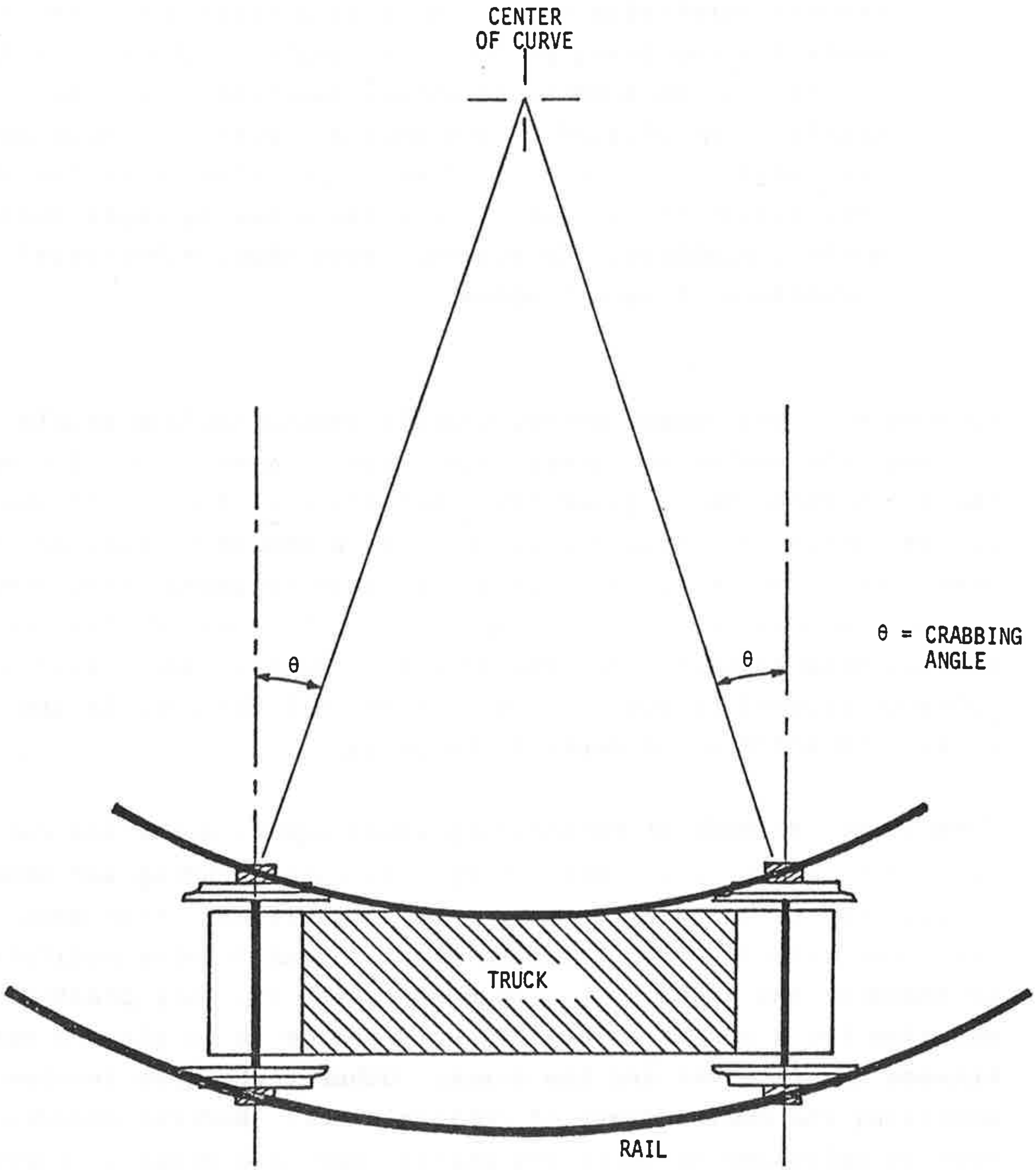


FIGURE 12.2 CRABBING OF TRUCK AROUND CURVE (AFTER REF. 12.2)

- Prevent stick-slip at the wheel/rail interface. Experiments are now being performed to evaluate the feasibility of articulated trucks, sometimes referred to as steerable trucks. Articulated trucks have a flexible arrangement that allows the rotation of one axle relative to the other. This allows the trucks to traverse a fairly tight curve without crabbing. Experiments have shown substantial reductions of squeal noise.

To control wheel squeal noise, transit system designs should minimize the number of curves with a radius less than 100 times the truck wheel base, generally about 230 m (750 ft). If short-radius curves are unavoidable, the curve should be located in an area that is not noise-sensitive. If this is impossible, other control techniques must be investigated. If short-radius curves on main line sections are unavoidable, some passengers will inevitably be exposed to squeal noise, unless all the cars in the system are modified to control the squeal.

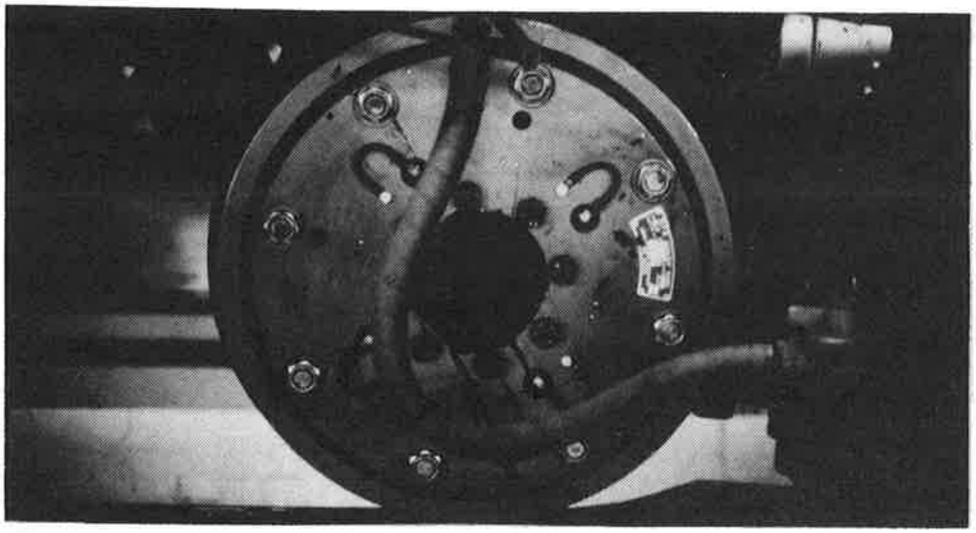
Some common methods of controlling wheel squeal noise are summarized in Table 12-1. All, except for rail grinding and wheel truing, can be quite effective. It is significant that most practical methods of controlling wheel squeal involve modifying or changing the wheel sets. In most cases, the only practical solution for a specific wheel squeal problem is to place a barrier between the receiver and the track. Other techniques involve modifying the entire fleet of rolling stock. Radical changes, such as switching to resilient wheels, may have other systemwide benefits such as reduced shock loads on truck-mounted equipment, but an individual case of wheel squeal will rarely provide sufficient economic motivation for such modifications. If barriers can be placed close to the track, wayside noise levels can be reduced by up to 10 dBA, even with relatively short 1 to 1.5 m (3 to 5 ft) barriers.

TABLE 12-1 SUMMARY OF WHEEL SQUEAL CONTROL METHODS

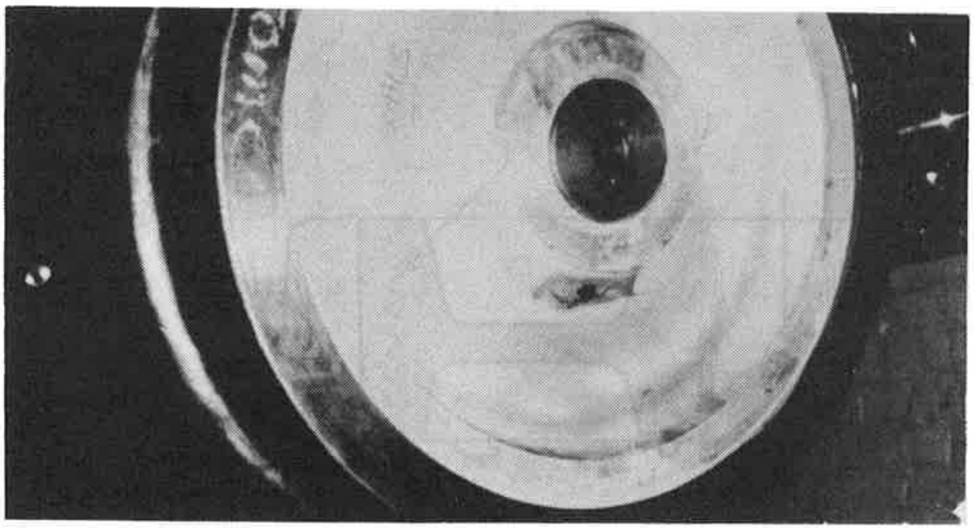
Method	Reduction at Squeal Frequencies (dB)	Comments
Resilient Wheels	15 to 30	Most resilient wheels essentially eliminate wheel squeal noise.
Ring Dampers	10 to 25	Very effective at controlling squeal above 1500 Hz. Appear to be one of the most cost-effective methods of controlling noise.
Viscoelastic Damping	10 to 25	Although not widely used or tested, constrained layer viscoelastic damping can reduce squeal noise.
Tuned Dampers	20 to 30	Made by Krupp of Germany. Although moderately expensive, they appear very promising.
Articulated Trucks	10 to 30	Still in experimental stage, little measurement results available.
Sound Barriers	5 to 15	Attenuation depends on barrier configuration.
Rail Grinding	+5 to -5	Inconsistent results have been achieved with rail grinding.
Wheel Truing	0 to -5	Marginal reductions of wheel squeal have been observed after wheel truing.
Rail Lubrication	0 to 25	Grease, water spray and water-and-oil mixtures have been used with varying degrees of effectiveness.

Resilient wheels, ring dampers, and tuned dampers have recently received considerable attention with respect to their effectiveness in controlling wheel squeal noise. Resilient wheels and ring dampers were field tested on the SEPTA system (Ref. 12.1) which uses vehicles with tread brakes. The three types of wheels tested are shown in Figure 12.3a and b. Only limited testing of the resilient wheels was possible because of difficulties while the wheels were in revenue service: the SAB and Penn Bochum wheels were damaged by overheating caused by problems with the brake systems; the Acousta Flex wheels were removed from service, after exhibiting bond failure between the elastomer and the wheel rim. Although all of the wheels effectively controlled wheel squeal (the Penn Bochum and Acousta Flex wheels being significantly more effective than the SAB wheels), overheating prevented all of the wheels from being used in revenue service for a significant period of time. Resilient wheels should not, therefore, be used on any system with tread brakes before more detailed evaluation of potential safety problems has been performed. Resilient wheels have been successfully used in Europe for many years on transit systems that do not have tread brakes. Also, similar test installations of the Bochum and Acousta Flex wheels have been in service on BART cars, using disk brakes for several years with no failures or other problems.

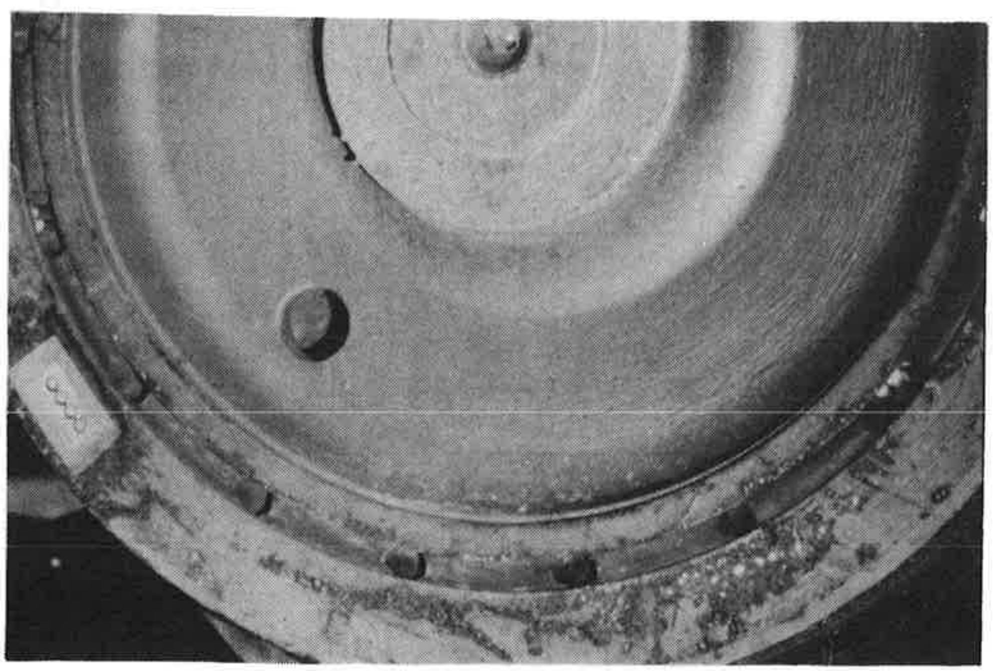
Ring-damped wheels promise to be one of the most cost-effective methods of controlling wheel squeal. The basic configuration for ring dampers is shown in Figure 12.4. A groove is machined into the tread of the wheel, either on the field or the flange side. The ring damper is then snapped into the groove. Damping, apparently, is provided by friction between the damping ring and the groove. In the test program described in Reference 12.1, the snap rings were made of untreated steel. Testing showed that the rings virtually eliminated squeal at frequencies above 1500 Hz. However, one very important problem was observed: after the ring-damped wheels had been in service for about 10 months, the rings had frozen rigidly in the grooves. Subsequent tests showed that the rings lost virtually all of their damping characteristics when



SAB RESILIENT
WHEEL



ACOUSTA FLEX
RESILIENT
WHEEL



PENN CUSHION
(BOCHUM)
RESILIENT
WHEEL

FIGURE 12.3a PHOTOGRAPHS OF RESILIENT WHEELS

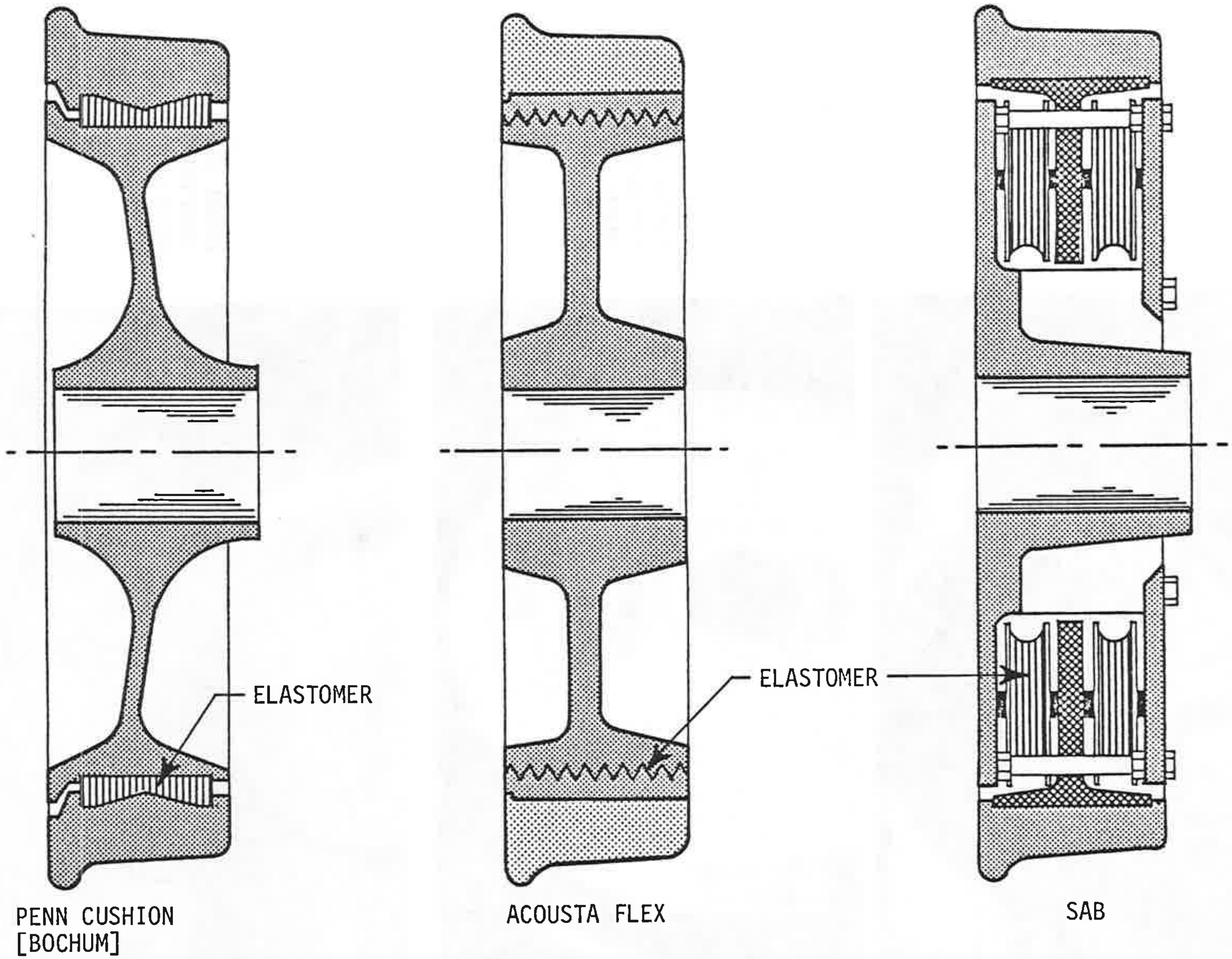
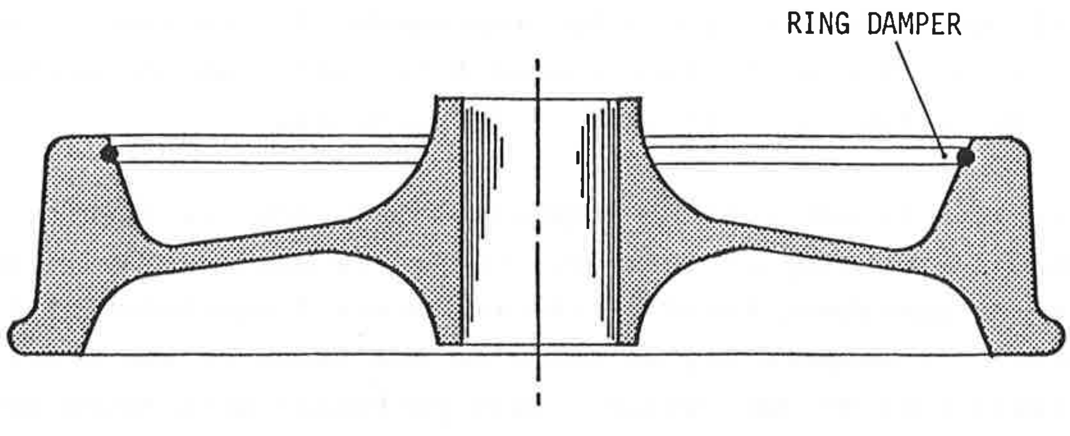
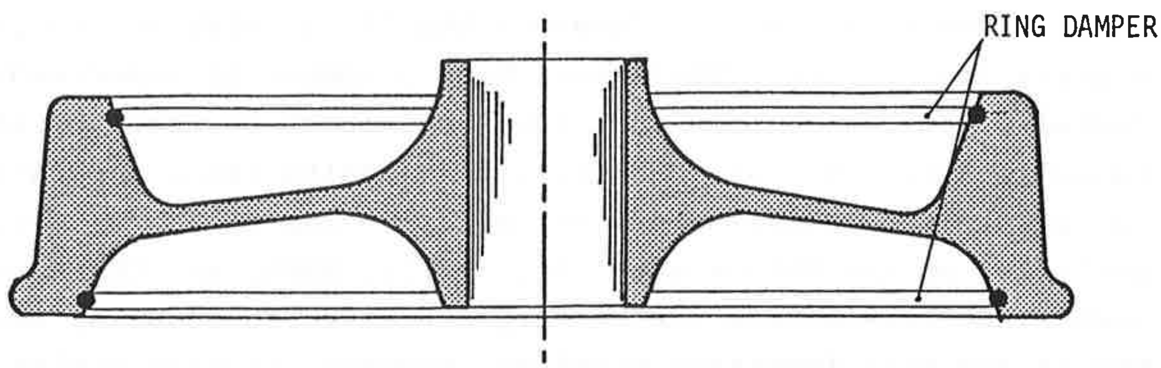


FIGURE 12.3b CROSS SECTIONS OF RESILIENT WHEELS



DAMPER ON FIELD SIDE



DAMPERS ON FIELD AND FLANGE SIDES

FIGURE 12.4 RING-DAMPED WHEELS

frozen in the grooves. This test program implies that ring-dampers are a cost-effective solution to the problem of wheel squeal, but before they can be recommended for widespread use on transit car fleets, further research is needed on the mechanism that causes the rings to bind in the grooves.

Figure 12.5 illustrates new squeal control devices developed by Krupp Manufacturing of Germany. These are mass/spring dampers or vibration absorbers tuned to the resonance frequencies of the wheels. The dampers are attached to the sides of the wheels, typically four to each wheel; tests performed with these dampers have shown them to be very effective (Ref. 12.4). To date, they have not been tested on any of the North American transit systems. They do not appear to be as cost effective as ring dampers, although they are very effective and can also measurably reduce wheel/rail noise on tangent track.

One method of squeal noise reduction not included in Table 12-1 is lubrication of the wheel flange or the flange side of the rail on short-radius curves. There have been a number of experiments and installations which show that lubrication can be used to reduce squeal and reduce wear. However, the results can be sporadic, and the maintenance requirements are severe. Tests such as those performed on the PATH system (Ref. 12.5), MBTA, and CTA have all shown that lubrication can be very effective in reducing squeal. One of the most important problems, however, is maintaining the lubricating equipment so that the lubricant remains confined to the side of the rail and the wheel flange area and does not migrate to the top of the rail.

In those areas where rail lubricating devices are used, the life of wheel flanges and rail is extended, but the degree of wheel squeal reduction is dependent upon the status of the equipment. Further, there are times when lubricant gets on top of the rail and causes loss of traction and/or wheel flats due to sliding of wheels. Some mainline railroads use on-vehicle equipment which applies lubricant to the wheel flanges rather than to the rail,

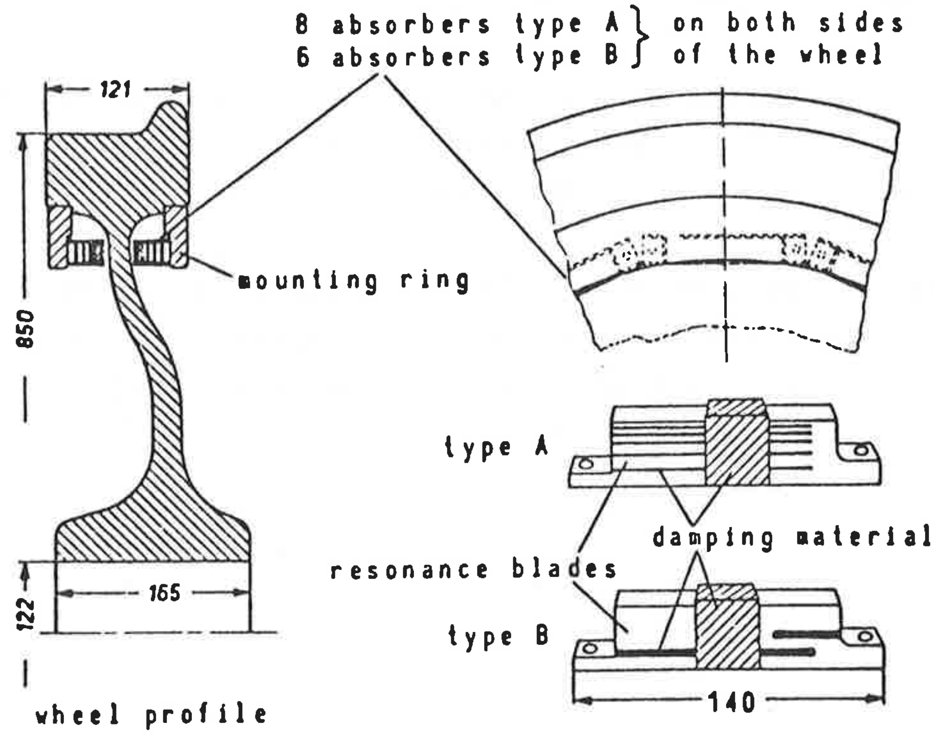


FIGURE 12.5 KRUPP TUNED VIBRATION ABSORBERS

but this is probably not practical for transit systems. Water spray lubrication has been tested by PATH, MBTA, and CTA and found to be effective, however, during the winter, freezing is a problem in exposed areas. Antifreeze may be added to the water to eliminate this problem, but negative side effects, such as slower evaporation and increased sliding, require further investigation. Water lubrication is also not appropriate for open decks or in areas where drainage is poor.

In summary, rail or wheel-flange lubrication can be an effective means for control of wheel squeal. However, the maintenance and operational difficulties make it less attractive than other direct means for control of wheel squeal.

CHAPTER 12 REFERENCES

12.1 H. J. Saurenman, R. L. Shipley, G. P. Wilson, "In-Service Performance and Costs of Methods to Control Urban Rail System Noise - Final Report," UMTA-MA-06-0099-80-1, (U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, December 1979).

This is the final report of a study evaluating the acoustic and economical effectiveness of five methods of controlling wheel/rail noise and vibration on urban rail transit systems. Evaluations of rail grinding, wheel truing, resilient wheels, ring-damped wheels, and welded vs. jointed rail were performed under revenue service conditions on the Market-Frankford Line of the Southeastern Pennsylvania Transportation Authority (SEPTA) rail transit system. Resilient wheels and ring-damped wheels were found to be very effective at controlling wheel squeal. Only marginal changes in squeal noise were observed after truing wheels and grinding rails. Ring-damped wheels were found to be ineffective at controlling wheel squeal if they were locked rigidly in the grooves.

12.2 P. S. Remmington, M. S. Rudd, I. L. Ver, "Wheel/Rail Noise and Vibration," Final Report, (2 volumes), UMTA-MA-06-0025-75-10 and UMTA-MA-06-0025-75-11, (U.S. Department of Transportation, Urban Mass Transportation Administration, Washington, D.C., May 1975).

The final report of a very detailed, relatively theoretical study of the mechanisms of wheel/rail noise generation. Analytical formulas are developed and at least partially verified with laboratory and field measurements. The purpose of this study was to develop a basic understanding of wheel/rail noise and some methods that could be used to reduce it.

12.3 G. P. Wilson, "BARTD Prototype Car 107 Noise Tests with Standard, Damped, and Resilient Wheels," Final Report, (Wilson, Ihrig & Associates, Inc., Oakland, CA, Prepared for Parsons Brinkerhoff-Tudor-Bechtel, June 1972).

A report on tests performed on BARTD prototype car 107, using various types of wheels. The testing included a section of subway short-radius curves where in-car and in-tunnel noise levels were measured with unground and ground rail and with various types of wheels. Acousta Flex, Bochum resilient wheels, and wheels with constrained layer damping epoxied to the wheels were tested. The resilient and damped wheels were found to reduce wheel squeal most effectively.

12.4 E. Raquet, G. Tacke, "Sound Emission of Railway Wheels and Tests on Noise-Damped Wheels for Long-Distance and Local Rail Traffic," (6th International Wheel Set Congress, Chicago, IL, October 1978).

A report on a new concept in damping the resonant motion of rail wheels. The Krupp Company has developed a damping device that very effectively controls wheel squeal. Several absorbers are bolted to each wheel. Tests have shown these dampers to be very effective at controlling wheel squeal and also indicate that they may provide reductions of wheel/rail noise on tangent track.

12.5 A. Senko, "PONYA Terminal Noise," Letter report to The Port of New York Authority, (Goodfriend-Ostergaard Associates, New York, NY, 8 March 1971).

A report on measurements taken to evaluate a water spray lubrication system at the PATH Church Station Terminal. The system was found to be very effective in reducing wheel squeal above 500 Hz.

CHAPTER 13

13. PRESSURE TRANSIENTS IN SUBWAYS

Pressure transients are large fluctuations in the ambient pressure, which occur whenever a train passes a discontinuity in a tunnel. If the pressure transient is large enough and rapid enough, it can create a pressure differential between the middle and outer ear that can cause patrons significant discomfort. Pressure transients of significant magnitude are a common problem, particularly in the newer transit systems with high train speeds and large blockage ratios (the ratio of the cross-section area of the train to the area of the tunnel).

Control of pressure transients is not usually a part of noise and vibration control. However, this short discussion is included because pressure transients are often neglected in transit system design. Note that pressure transients can cause problems with sound recording instruments; this problem is discussed in Chapter 4.

13.1 NATURE OF PRESSURE TRANSIENTS

A pressure transient can be expected to originate at any point where there is a discontinuity in the subway system. Transients primarily occur at the following locations:

- Portal entrances and exits.
- Fan and vent shafts.
- Tunnel transition sections (e.g., the transition from double-track tunnel to single-track tunnel or the transit from station to running tunnel).
- Areas where trains pass by in opposite directions in double-track tunnels.
- Connecting passages between two tunnels.

Portal entries are usually the greatest source of problems. However, large vent shafts will sometimes create worse transients because passing the shaft is equivalent to leaving one portal and rapidly entering another.

Some factors that significantly influence the form and magnitude of pressure transients are:

- Train Speed: The magnitude of the pressure transients is approximately proportional to the square of the train speed, V^2, and the rates of pressure change have a dependence in the range of V^2 to V^3.
- Blockage Ratio: The larger the blockage ratio, the larger the transients. The magnitude is approximately proportional to the cube of the blockage ratio.
- Train Length: The effect of length on the transients is very complex. In most cases, the transient is more severe for longer trains.
- Geometry of Portals and Vent Shafts: Flared portals and parallel top-slot portals can reduce the rate of pressure change and hence the discomfort due to portal entries and exits.
- Friction Factors or Mean Roughness Heights of the Tunnel and Train Surfaces: As the roughness increases, the magnitude of transients at vent shafts and portals will increase.

Figure 13.1 illustrates tunnel wall pressure profiles as a function of train position for three conditions: train entry, steady-state conditions prior to exit, and train exit. During entry, a positive pressure wave is created which propagates at the speed of sound (340 m/sec) toward the tunnel exit, where it is linearly reflected back to the train and tunnel entrance as a negative pressure wave. It then interacts with the train and is nonlinearly reflected back towards the exit. After several reflections, steady-state conditions are achieved because the train has progressed to a point well within the tunnel, as shown

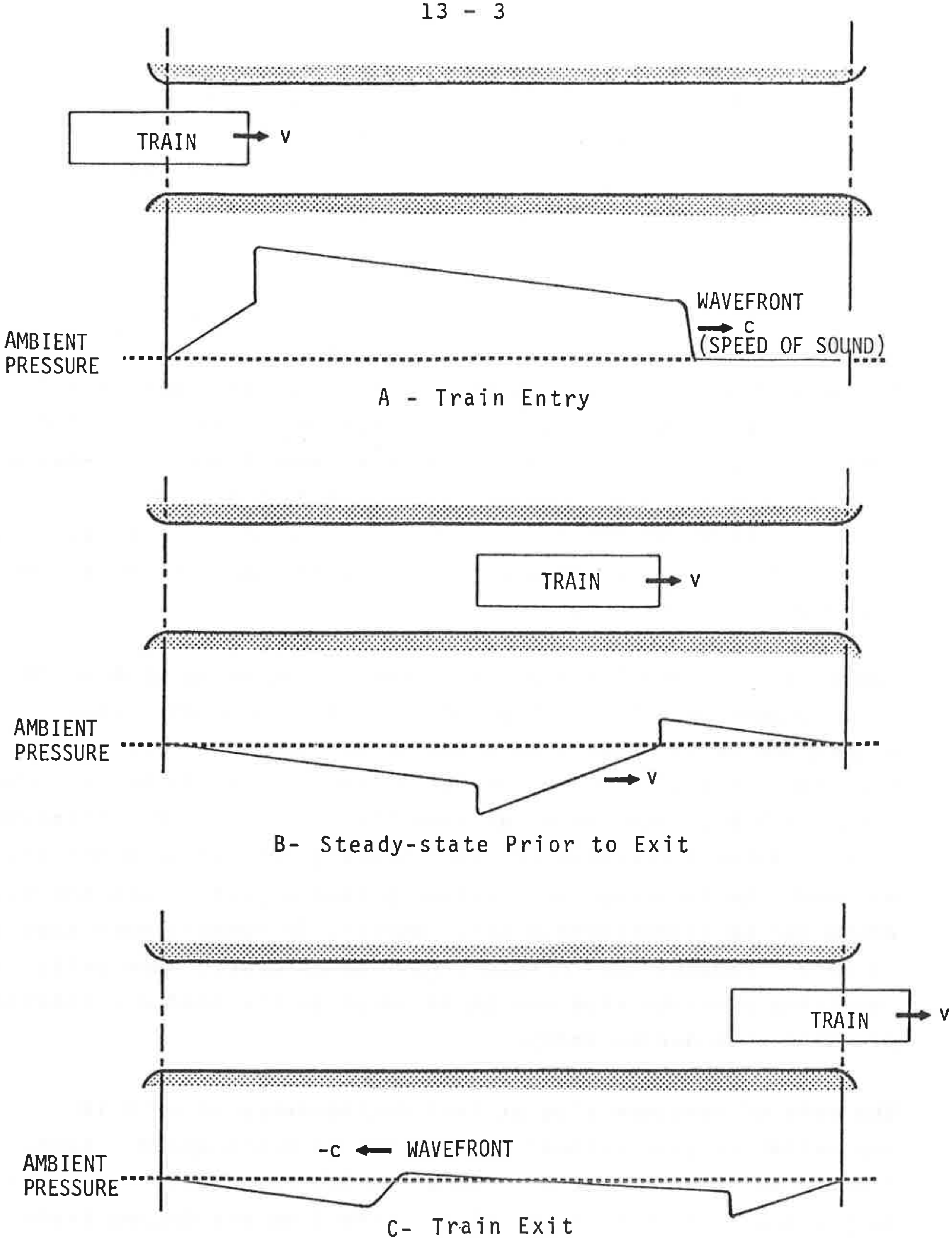


FIGURE 13.1 TUNNEL WALL PRESSURE AT VARIOUS PHASES OF TRAIN PASSAGE THROUGH A TUNNEL

in Figure 13.1.b. Upon the train's exit, another pressure transient is generated, also propagating at sonic speed, which is reflected back and forth within the tunnel for some time after the train has left the tunnel. Almost all portal configurations allow total wave reflection at each open end.

Examples of lead car and trailing car interior pressure transients, measured while a train passed the blast shaft of the BART Transbay Tube, are given in Figure 13.2. These examples illustrate the magnitude of car interior pressures that can occur in very long tunnels. Interestingly, the dampers and cross-passage doors of the Transbay Tube ventilation and pedestrian galleries were very leaky at the time of the measurements. If these leaks did not exist, the entry pressure transient magnitudes in Figure 13.2 might be even higher.

Magnitudes of interior lead car pressure can be as high as 10 in. water column (w.c.) or about 50 psf. In the worst cases, magnitudes at the subway wall can be as high as 15 in. w.c. following very high speed entries (greater than 120 km/hr) into long, high blockage ratio (greater than .55) tunnels. Negative (below ambient) pressure transients are generally less severe, although the trailing car interior pressure just before the train exits can be significantly below ambient in tunnels more than 1.5 km long. Because this pressure must be recovered upon exit, the resulting pressure rise can be as large as the lead car interior pressure rise during entry.

The rate of pressure rise or fall during entry or exit is approximately proportional to the cube of train speed. Thus, increasing the train speed from 80 to 125 km/hr will more than triple the rate of pressure rise in the lead car during train entry and in the trailing car during exit. Indeed, speed reductions were required to control the rate of pressure rise at entry and exit from the BART Berkeley Hills Tunnel and Transbay Tube.

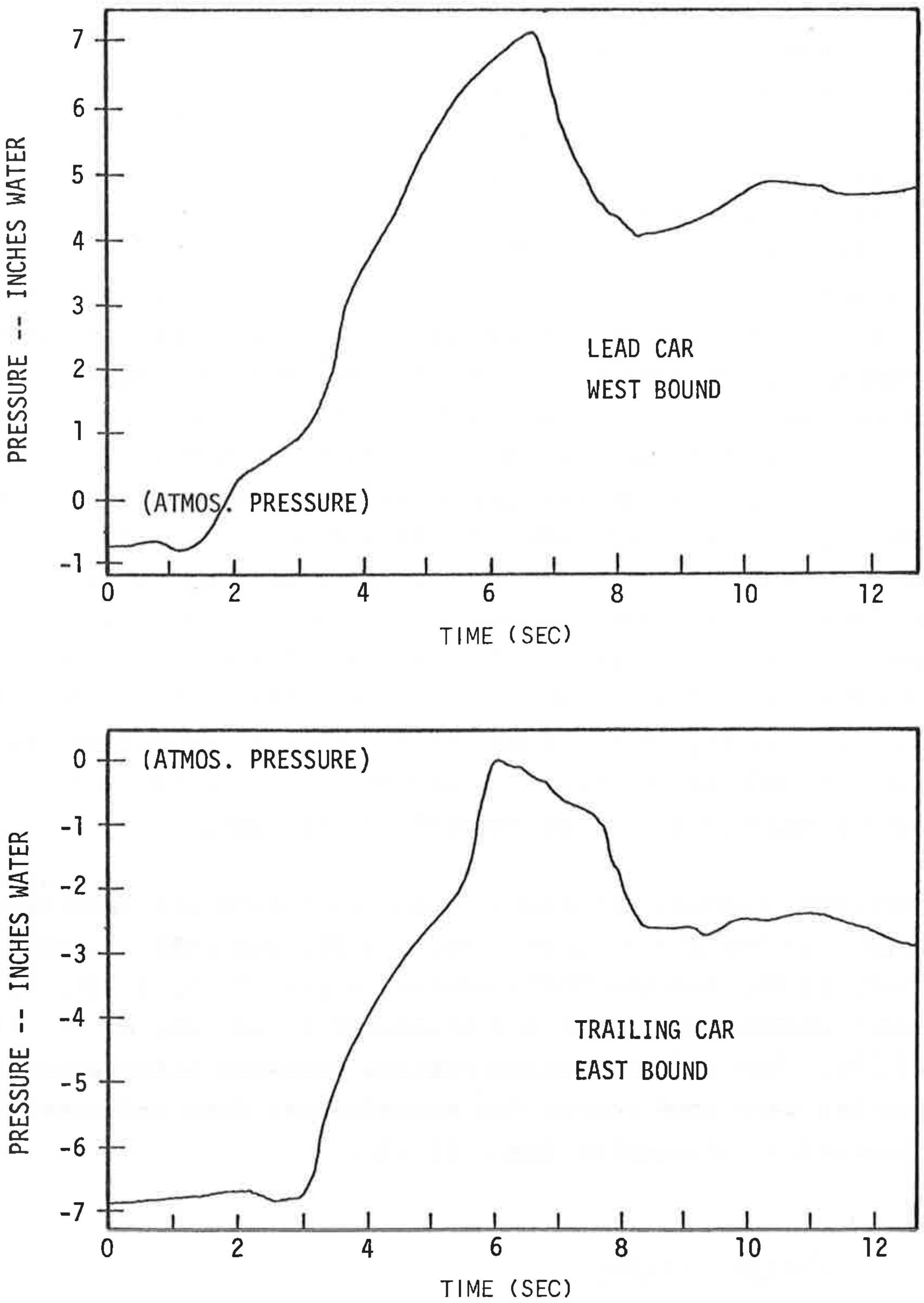


FIGURE 13.2 EXAMPLE PRESSURE TRANSIENTS INSIDE LEADING AND TRAILING CARS PASSING BY BLAST SHAFT OF BART TRANSBAY TUBE, 120 KM/HR, 5-CAR TRAINS

No simple, satisfactory prediction procedure has been published for estimating pressure transient magnitudes and rates of change. The problem is generally nonlinear although certain simplifying assumptions can be made under restricted circumstances (Ref. 13.1). A prediction procedure is included as part of the Subway Environmental Design Handbook (SEDH) (Ref. 13.2). This procedure treats unsteady air within the subway as an incompressible fluid. The procedure is useful for qualitative predictions of entry, exit, and vent shaft passage pressure transients and should be reasonably accurate for predicting steady-state conditions between these events. Fox and Henson (Ref. 13.3) used the method of characteristics for accurate prediction of pressure variation during train passage through a subway. The same method was later used by Fox and Vardy (Ref. 13.4) and by Hara (Ref. 13.5). The method of characteristics and its general application to compressible flows is discussed by Zucrow and Hoffman (Ref. 13.6). A good prediction procedure for pressure transients should consider the far field air pressure as compressible, but the near-field air in the vicinity of the train (i.e., in the annulus and immediately fore and aft of the train) can probably be considered as an incompressible fluid, as assumed in the SEDH.

Numerous pressure transient measurement data are reported for the BART Berkeley Hills Tunnel (Ref. 13.7), the BART Transbay Tube (Ref. 13.8), and the WMATA system (Refs. 13.9, 13.10, 13.11). Model experimental data are presented by Dayman, et al. (Ref. 13.12). The British Hydromechanics Research Association is another very good source for experimental data and general theoretical discussion (Ref. 13.13).

13.2 EFFECT ON PATRONS

Pressure transients can be very uncomfortable for patrons, especially those with sinus disorders or colds. One criterion used for most U.S. transit systems and included in the SEDH (Ref. 13.2) states that the rate of pressure change should be less than

0.06 psi per second (1.7 in. w.c./sec) when the pressure change is greater than 0.10 psi (2.8 in. w.c.). This criterion assumes that the earliest discomfort to the ears begins at 0.06 psi and that a healthy body can equilibrate to this pressure differential in less than a second. It should be pointed out that this criterion will not adequately evaluate certain complex pressure transients. Pressure transients can also cause a noise problem by the sudden banging of doors which can annoy or even frighten the unwary rider.

This criterion is exceeded on almost every modern transit system with high-speed trains and high blockage ratios.

13.3 EFFECT ON HARDWARE

13.3.1 Fan and Vent Shaft Dampers

Operable dampers used at fan and vent shafts for ventilation and fire control are one of the most common types of equipment to be damaged by tunnel pressure transients. Original WMATA specifications for vent shaft dampers at ancillary areas near stations required that the dampers withstand a maximum load of 20 psf. This specification was inadequate for line tunnel dampers located within the tunnel where pressure loads can be much higher, possibly in excess of 50 to 80 psf under unique conditions. Such magnitudes have never been measured, but have been projected on the basis of possible worst-case conditions (Ref. 13.1, 13.9, 13.10, 13.11). The more usual loads are generally about 50% of the worst-case loads, or about 40 to 50 psf, in long tunnels with high-speed trains.

13.3.2 Cross-Passage Doors

Cross-passage doors within line tunnels may experience worst-case pressure loads from 10% to 15% higher than those for vent or fan shaft dampers. This occurs if the worst-case positive pressure in one tunnel coincides with a negative pressure in the adjacent tunnel. Cross-passage doors within the BART Berkeley Hills Tunnel and Transbay Tube have required reinforcement of latches and hinges and, in some cases, removal. Using sliding doors will prevent doors swinging open and slamming shut in the event of latch failure; a swinging door can be dangerous if it is opened when there is a significant pressure differential across the door.

13.3.3 Concrete Masonry Partitions

Intertunnel partitions, constructed of concrete masonry blocks without reinforcement, have been literally blown down by a single train passby pressure transient. Such partitions should be substantially reinforced and perhaps even removed from long tunnels. Intertunnel partitions must withstand the same pressure differentials as cross-passage doors. Differential pressures can be as much as 100 psf for high-speed (125 km/hr) 8- or 10-car train entries into very long unvented tunnels with blockage ratios of 0.55. Short tunnels, lower train speeds, or locations close to tunnel portals or station ancillary areas require less sturdy concrete masonry partitions, although reinforcement is always advisable (Ref. 13.11).

13.3.4 Suspended Ceilings

Problems have been caused when transit trains pass under suspended ceilings in station areas (Ref. 13.14). The gradual rise and sudden drop of pressure that occurs when the train passes underneath the panels can dislodge them. Current specifications at WMATA require that ceiling panels withstand pressure differentials of at least 15 psf. Particular attention should be paid to the method of retention. If sound absorption material is not retained within the panel by plastic sheet or hardware cloth, the train passage can push or pull the material out of the perforated panel.

13.3.5 Vehicle Hardware

Pressure transients rarely cause problems with vehicle panels and windows, although the Japanese have had window failures during train passages through tunnels. Window pressure differentials are discussed by Voss and Wiebels (Ref. 13.15). Generally, the lead car interior pressure is the same as that in the annular space between the car and tunnel wall. The greatest pressure differentials exist across the lead car front windshield and the inter-car access door. Pressure differentials across panels of other cars are not significant. Measurements have been performed on the BART system (Ref. 13.16) to determine pressure differential across doors.

13.4 DESIGN GUIDELINES

13.4.1 Vehicle Design

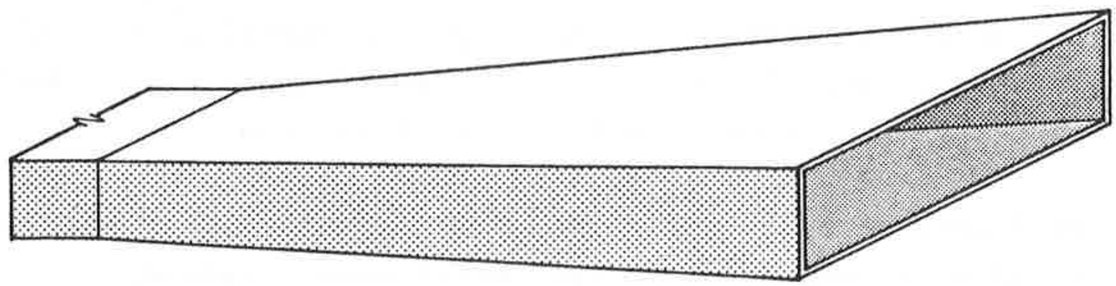
The shape of the transit vehicle nose has little effect either on the magnitude of pressure transients or their rate of change. During entry, the effective vehicle skin friction is the primary cause of the pressure transient. This is because the air velocity in the annulus during train entry is very high relative to the train, much higher than relative to the tunnel wall. Reducing the effective skin friction of the train will thus directly reduce pressure transient magnitudes. The effective skin friction is primarily determined by flow losses around the trucks, wheels, axles, and undercar equipment (Ref. 13.13). Efforts to reduce the effective skin friction of the vehicle should, therefore, be directed at the undercar area.

13.4.2 Tunnel Portal Design

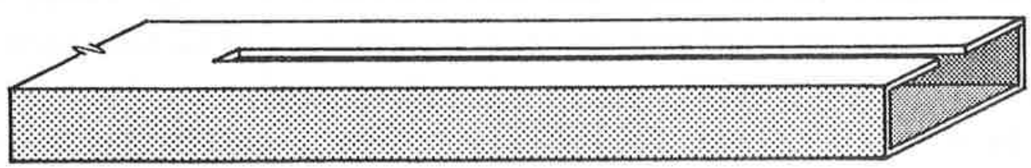
Tunnel portal transition sections have been incorporated at all modern transit systems with varying degrees of success. Two basic designs are currently being recommended: the flared portal transition and the parallel top-slot portal. These are shown schematically in Figure 13.3. The tapered top-slot portal is an older design (Refs. 13.17, 13.18), which has since been determined to be effective only at the narrower sections of the slot (Ref. 13.19). Neither the parallel slot or tapered flare design is necessarily more or less expensive than the other. The parallel slot requires a cantilevered ceiling which cannot be built into hillsides or covered. This construction sometimes requires an extension of the tunnel length. The flared portal design can be constructed as part of the tunnel, although additional excavation is required. In some cases, flaring can be achieved by slanting the ceiling and keeping the walls parallel. Either design can provide the necessary portal action. However, as of this writing, no measurements documenting the performance of the parallel top-slot portal have been performed, although its action has been theoretically described by Barrows (Ref. 13.20) and Wilson (Ref. 13.21).

During a train entry, the initial positive pressure wave generated by the nose entry is reflected back from the tunnel exit as a negative pressure wave which accelerates the air column in the direction of the train motion. It also tends to superpose with the train near field pressure. If possible, the portal transition should be long enough so that the train nose arrives at the end of the portal transition at about the same time as the arrival of the reflected initial wavefront. At a given train speed (v) and overall subway length (L) including portal length, the transition length (d) to satisfy this condition is:

$$d = 2\,Lv/(c + v) \qquad 2\,L\,v/c$$



FLARED PORTAL



PARALLEL TOP-SLOT PORTAL

FIGURE 13.3 SKETCHES OF TUNNEL PORTAL TRANSITIONS where c is the velocity of sound. This formula for transition length should be viewed as an optimum length (subject to cost and real estate limitations). In a typical tunnel with an overall unvented length of 500 m and train entry speed of 100 km/hr (28 m/sec), the minimal transition length is about 82 m.

The location of a cross passage may effectively shorten the tunnel length with respect to portal performance. Indeed, the cross-passage locations and sizes might be adjusted, in combination with effective portal design, to achieve an "optimum" pressure control strategy and to reduce portal transition lengths required. The location of a vent shaft will also essentially short-circuit tunnel air to the ambient pressure, reducing the effective tunnel length. However, an additional pressure transient will be created as the train passes the vent shaft, and this transient may be similar to that created at a blunt entry. This is especially true of large vent shafts, which require additional pressure transient control provisions.

If the tunnel is so long that the entire train can enter the tunnel before the initial pressure wave is reflected back to the train nose, the tunnel may be viewed as semi-infinite, and little can be done to reduce the entry pressure transient without very long portal transitions of length similar to that of the train.

13.4.3 Porous Center Walls

Where possible, as in double-box structures, the wall between subway tunnels should be made porous. The porous wall is the most effective means of controlling pressure transients. Center wall porosity is achieved with openings of approximately 3 m^2 (30 sq ft) on 7.5 to 9 m centers. The exact dimensions and spacings are not important. The porous wall has the effect of doubling the subway cross-sectional area. Because magnitudes and rates of rise are roughly proportional to the cube of the blockage ratio, porous walls will substantially reduce the pressure transients. Indeed, car interior pressure transients will often be within criteria, even during high-speed entries and exits of very long trains in tunnels without any effective portal transition. However, a high-speed blunt entry into a double-box subway, even with a porous wall, may cause very short duration pressure pulses, which can be uncomfortable at the moment of lead car entry. Some short transition is therefore probably desirable. At this time, there are no guidelines concerning minimum length requirements.

Portal flare rates are designed to provide a 4 to 6 degree equivalent conical flare angle. In very long transitions, however, the entry blockage ratio of the transition should not be less than 0.25 or 0.30. Equivalent conical flare angles of less than 4 degrees may be necessary. Barrows' recommendations (Ref. 13.20) should be followed for slotted portals. The slot width can be overestimated during design and then experimentally adjusted with steel plates to determine the optimum slot width (Ref. 13.19). Slot widths of 15 to 30 cm for a 30 m transition are reasonable.

13.4.4 Vent and Fan Shaft Design

The location of line fan and vent shafts can effectively reduce tunnel entry pressure transients by the reflection of pressure waves. However, no general control provisions have been developed for pressure transients produced by the trains passing shafts.

If possible, tapered or flared transition sections should be used at shafts, just as at an entry or exit portal. Such transitions are normally incorporated to avoid restricting the ventilating airflow, but often precede the shaft rather than follow it, as would be best for pressure transient control. A cross passage can be placed at a downstream position to reflect pressure waves back to the train, as in the discussion of entry portals above. If a cross passage already exists, it should be left partially or entirely open. The degree of opening might be experimentally determined to avoid generating another significant transient as the train passes the cross passage.

One method of reducing vent shaft passby transients is to restrict the airflow by placing the dampers in a partially closed position. This will produce a significant pressure drop across the damper prior to train passage, which induces airflow in the line tunnel beyond the shaft, consequently reducing the overall passby pressure transient magnitude and rate of change. The best damper position must be determined experimentally. The effect of partial closing on tunnel ventilation must be carefully evaluated. Such pressure transient control techniques will require an overall review of the subway environmental design.

13.4.5 Speed Reduction

Speed reduction at critical points where pressure transient criteria are exceeded is a very effective control technique because the magnitude and rate of change are roughly proportional to the square and the cube of train speed. When train control is computerized, such speed reductions can be implemented without seriously degrading system performance, unless a large number of speed reductions are required.

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APPENDIX A

FUNDAMENTALS OF SOUND AND VIBRATION

Because many readers of this handbook may be unfamiliar with some of the basic concepts and terminology of acoustics, this appendix on fundamentals has been included as an aid to the reader. A short introduction to important acoustical terms is included as Section 1.3.1 of the introduction. The purpose of this appendix is to cover the fundamentals of sound and vibration in more technical terms. The following topics are covered:

- Physics of sound and vibration.
- Methods that are used to characterize noise and vibration.
- The concepts of sound absorption and sound isolation emphasizing the difference between the two.
- The types of sound fields, especially reverberant and free sound fields.
- Propagation and radiation of sound.

This appendix should be used as a reference for the reader unfamiliar with some of the basic concepts of acoustics. For more detailed information, there are a number of good reference books on acoustics which provide both an introduction to the field for the nontechnically oriented reader, along with the more detailed information required by engineers designing noise control treatments. Many such books have been used as references for this handbook and are included in the reference lists at the end of each chapter. Several recommended references are:

- C.M. Harris, ed., Handbook of Noise Control, 2nd edition, (McGraw-Hill Book Company, NY, 1979).

  This is a very good, general reference on noise control. There are a total of 45 chapters on different aspects of noise control, each written by a different author. Most of the material is technical but does not require an acoustics background to be understood.

- L.L. Beranek, ed., Noise and Vibration Control, (McGraw-Hill Book Company, NY, 1971).

  This widely used text tends to be fairly detailed and mathematically oriented. Although a valuable source for acoustical engineers, it is probably not appropriate for someone who does not have a background in acoustics.

- M. Rettinger, Acoustic Design and Noise Control, (Chemical Publishing Co., NY, 1977).

  Written in two volumes (Volume 1, Acoustic Design, and Volume 2, Noise Control), the texts present a wealth of practical information on the design of noise control features.

- L. Kinsler, A. Frey, Fundamentals of Acoustics, (John Wiley and Sons, NR, 1962).

  Although this book is widely used as a textbook for undergraduate acoustics courses and hence is readily available, the material tends to be too theoretical for direct application to noise control problems.

- "Transportation Noise and Its Control," available from the U. S. Government Printing Office, Washington, D. C. 20402, Stock Number 5000-0057, (U.S. Department of Transportation, Office of the Secretary, Washington, D. C. 1972).

A pamphlet prepared by the U. S. DOT to describe transportation noise to the general public. The pamphlet describes aircraft, highway, and rapid transit noise and includes a good, general description of the physics of sound.

- R. Lutz, "Theory and Practical Application of Noise and Vibration Abatement for Railway Vehicles," Report DT 25/E, (Office for Research and Experiments of the International Union of Railways, ORE, April 1976.)

  Although this report focuses on noise abatement procedures that can be applied to diesel locomotives, most of the information is sufficiently general to also be applied to transit noise control.

This list of references is meant to be representative and is by no means exhaustive.

A.1 PHYSICS OF SOUND AND VIBRATION

Noise, usually defined as unwanted or excessive sound, is now broadly recognized as a form of environmental degradation. Noise can be annoying, can interfere with sleep, work, or recreation, and in extremes, may cause physical and psychological damage. Sound travels through air in the form of small waves of pressure fluctuations. These waves are similar to the circular waves seen on the surface of water after a stone is thrown in a pond.

Sound is characterized by the amplitude of the pressure fluctuation and the frequency (or pitch) of the fluctuations. Typically, sounds will contain many different frequency components, however, there are some sounds that consist of only one frequency. A tuning fork is an example of a single frequency sound. The sound of a train passby consists of a wide spectrum of frequencies, but usually the components at any particular frequency are not individually distinguishable.

Sounds are divided into three basic categories based on the frequency of the sound. Sound in the frequency range audible to humans is referred to as audio sound, sound at frequencies below this range is referred to as infrasound, and sound above the audible frequency range is referred to as ultrasound.

Sound is created by fluctuating disturbances that create pressure waves in a medium; in practical situations, the medium is almost always air. The most common source of disturbance is vibrating objects such as the drum illustrated in Figure A.1. The vibrating object or surface disturbs the air molecules, alternately causing compression (squeezing together) and rarefaction (pulling apart) of the air molecules. The compression and rarefaction results in a pressure wave that travels (propagates) away from the vibrating object at a constant speed. Note that as for waves on the surface of a quiet pond, there is no net transfer of matter away from the vibrating source.

The speed of sound in air is independent of frequency and varies only slightly with humidity and atmospheric pressure. At a temperature of 20° C (68° F), the speed of sound is approximately 344 m/sec (1127 ft/sec). The air temperature can have a significant effect on the speed of sound; the speed of sound increases about 0.61 m/sec for each 1 degree C increase in temperature.

It should be noted that there is no clear distinction between sound and vibration. Sound is a vibratory phenomenon, although vibration generally refers to motion of solid objects. However, transmission of vibration through solid objects is often referred to as structureborne sound and transmission of vibration through the ground is often referred to as groundborne sound or groundborne noise. Sound waves travel much faster in solid objects than in air. For example, the speed of sound transmission in concrete

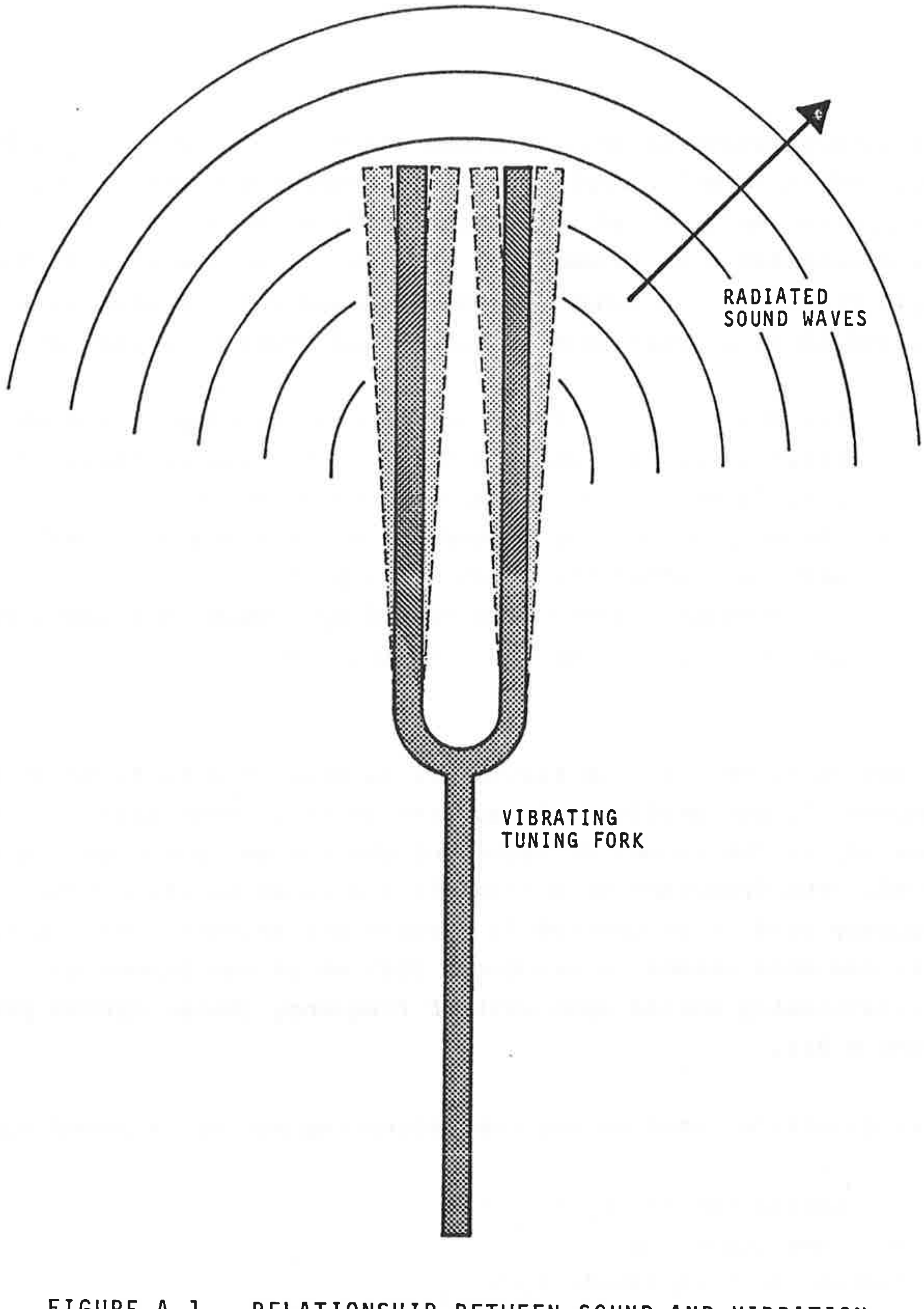


FIGURE A.1 RELATIONSHIP BETWEEN SOUND AND VIBRATION is approximately 10 times faster than in air.

A.1.1. Waveforms

Oscillatory phenomena such as sound and vibration are generally described in terms of amplitude, frequency, and phase. Figure A.2 illustrates amplitude as a function of time for the simple example of a sinusoidal wave (sine wave) - a wave of motion consisting of a single frequency. A single frequency sound wave is generally referred to as a "pure tone." Figure A.2 could be a plot of:

- The displacement of a freely vibrating simple harmonic oscillator. The classic example of a simple harmonic oscillator is a mass supported on a spring.
- The displacement of a pendulum oscillating at a small amplitude about the equilibrium point.
- The pressure fluctuation caused by a pure tone sound wave such as that created by a tuning fork.

The motion of the wave in Figure A.2 is described in terms of the frequency f, the amplitude A, and the initial phase angle. The frequency is the number of cycles of the motion that occur in one second. The frequency of a sound is analogous to its pitch. Frequency used to be denoted in "cycles per second" (cps) however, it is now more common to use Hertz (Hz) which has become an internationally agreed upon unit of frequency (Note: cycles per second = Hz).

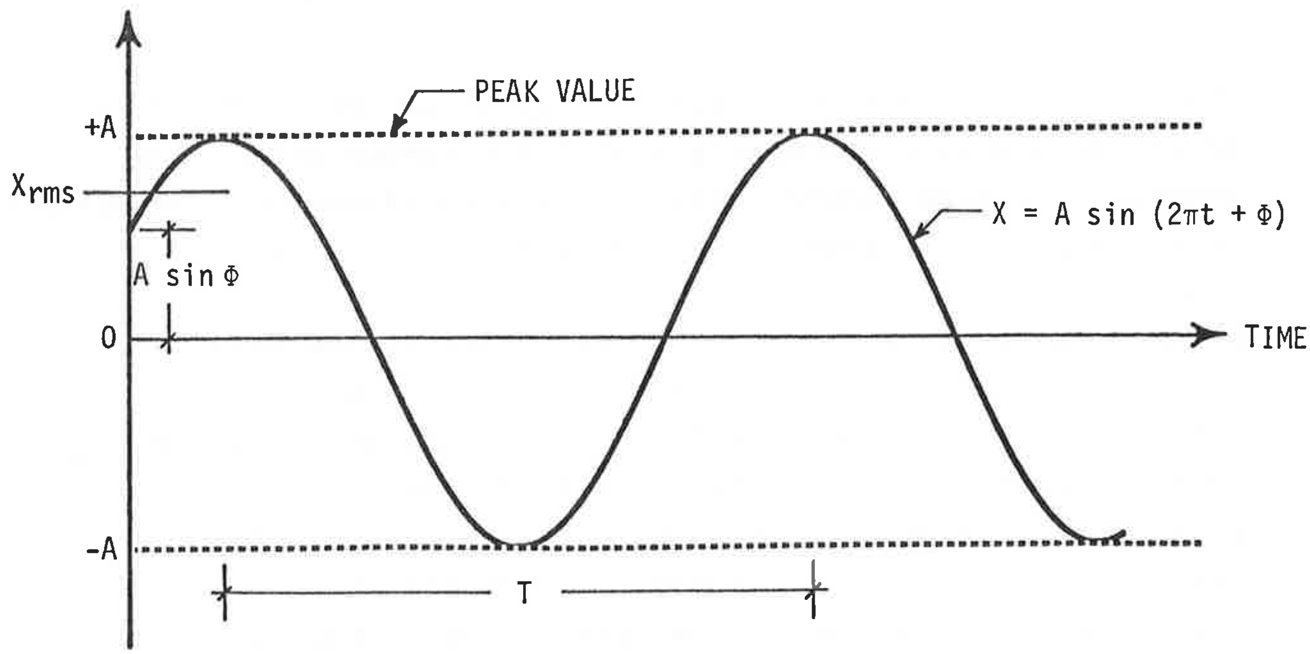
Other quantities used to describe vibrating motion or sound waves are:

T = period for one cycle = $1/f$

Peak amplitude = A

Peak-to-peak amplitude = $2A$

$$\text{Root mean square (rms)} = \left(1/T \int_0^T x^2(t)\,dt\right)^{1/2} \qquad \text{(A.1)}$$



A	=	AMPLITUDE
f	=	FREQUENCY IN HERTZ (CYCLES PER SECOND)
T	=	PERIOD FOR 1 CYCLE = 1/f
X_{rms}	=	ROOT MEAN SQUARE VALUE = $A/\sqrt{2}$
2A	=	PEAK-TO-PEAK AMPLITUDE

FIGURE A.2 PURE TONE OR SINE WAVE (EXAMPLE IS THE MOTION OF A SIMPLE HARMONIC OSCILLATOR) for a sine wave or pure tone, $x_{rms} = A/\sqrt{2} = .707A$

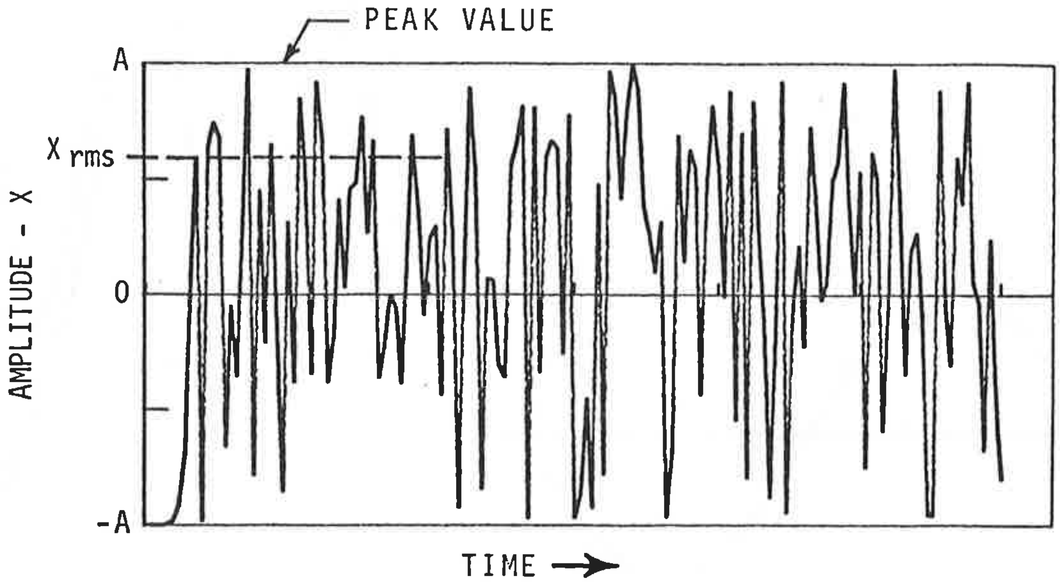
$$\text{Rectified average} = 1/T \int_0^T |x(t)| \, dt \qquad (A.2)$$

for a pure tone $= \frac{2}{\pi} A = .637A$

The rms value is a very common descriptor of amplitude of complex waveforms and is particularly useful for random noise. In general, the term "sound pressure" is used throughout the handbook to signify the rms sound pressure, unless otherwise indicated.

Figure A.3 is an illustration of broadband random noise such as that of a waterfall or such as might result from steel wheels rolling on rails. The term "broadband" indicates that the sound does not consist of a sum of discrete frequency components; the sound can be thought of as having all frequencies. The term "random" indicates that the magnitude of the noise cannot be precisely predicted for any instant of time. The rms value is the most common descriptor used for the amplitude of random signals, however, the rectified average and peak or peak-to-peak values are also encountered. Although a few applications do require the use of peak amplitude, in most applications, the rms value is more useful. There are two reasons for this. First, as discussed below, the rms value of a sound is a measure of the energy level involved and the rms values of components can be used to obtain the overall rms value. Second, the rms value of a sound or vibration is well correlated with human response. Measures such as the peak level are not as well correlated. The peak level is often an instantaneous event that occurs so rapidly that the human perception does not respond. Rectified average, although widely used for alternating current electricity, is not used in acoustics because the measure is highly dependent on waveform and cannot be directly related to other measures unless the waveform is known.

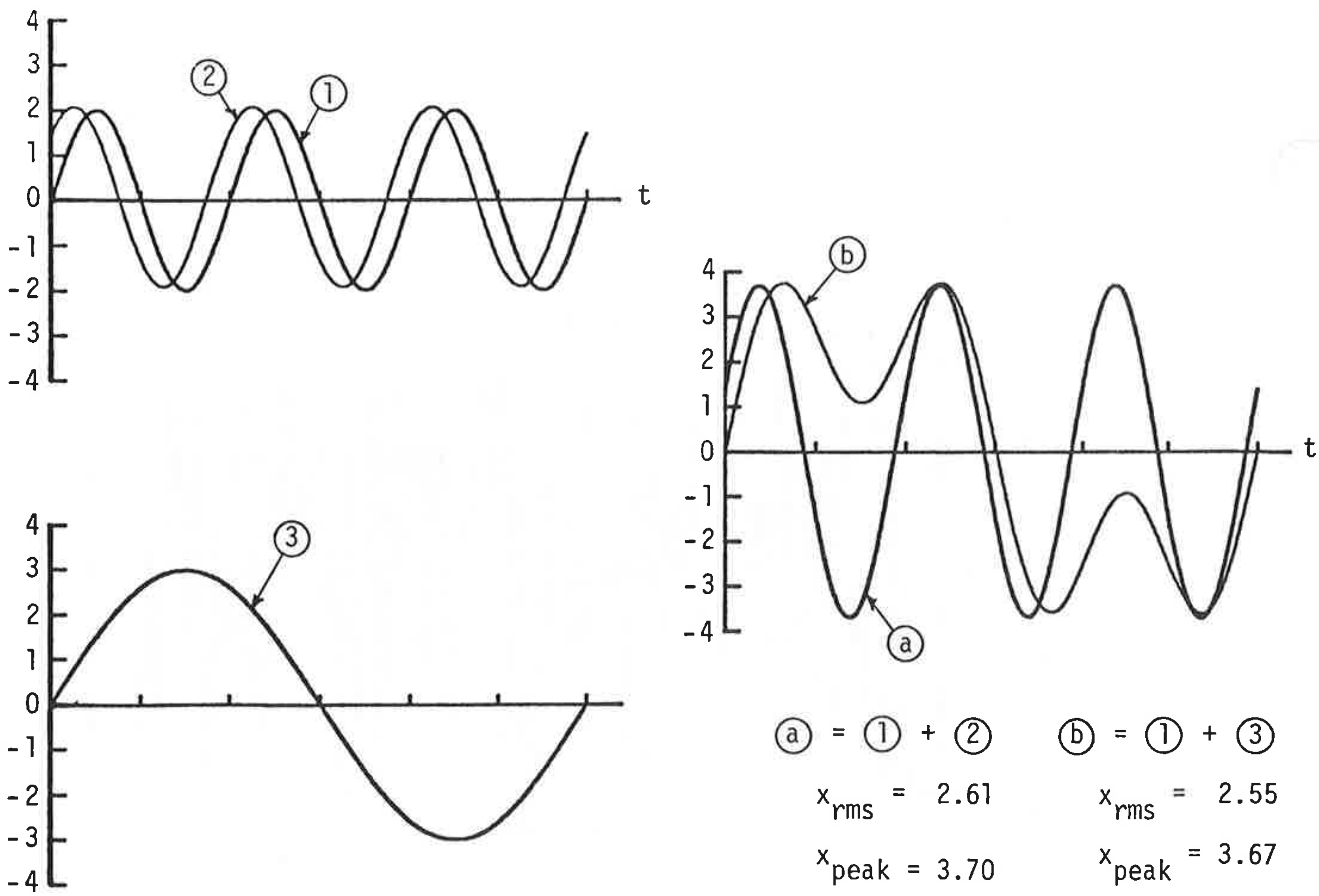
Figure A.4 illustrates the combining of two sound signals. Figure



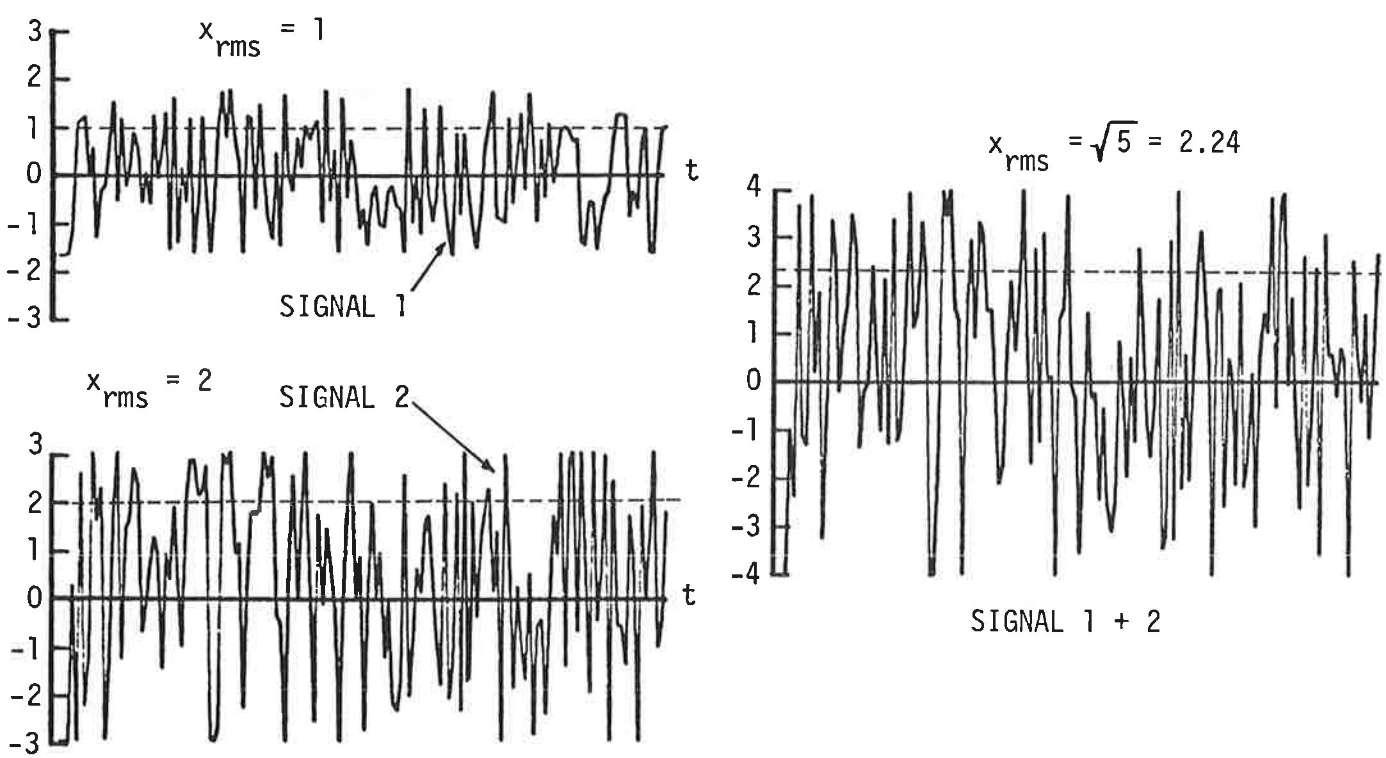
PEAK VALUE = A

ROOT MEAN SQUARE VALUE = X_{rms}

FIGURE A.3 RANDOM (BROADBAND) WAVE



a. PURE TONES



b. BROADBAND

FIGURE A.4 COMBINING SIGNALS

A.4a shows how two sine waves can be added to obtain a complex wave. The rms value of the combined signal is given by the formula:

$$(X_{3_{rms}})^2 = (X_{1_{rms}})^2 + (X_{2_{rms}})^2 \quad \text{(A.3)}$$

This equation is valid except when combining waves with pure tone components at the same frequency. When adding pure tone components, the relative phase angle will determine if there is constructive or destructive interference between the two waves. Note that in Figure A.4a, the rms value of the combined wave is given by Equation A.3, however, the peak value of the combined signal depends on the relative phase angle. It is even possible for the combined signal to have a lower peak value than one of the component signals. An important characteristic of the rms value is that it remains the same regardless of the relative phases of the constituent waves of a complex motion. As shown in Figure A.4b, the rms value of random signals add in the same manner.

A.1.2 Decibels

The human ear is capable of responding to a very wide range of pressures; at the threshold of pain, the sound pressure is one million times as large as the sound pressure at the threshold of hearing. Because there is such a large range of acoustic pressures that are of interest in noise measurements, the decibel (abbreviated dB) scale is used to compress the range of numeric values. To many laymen, the term "decibel" is uniquely associated with acoustic measurements, however, the term is also widely used in electrical engineering and represents a relative quantity. The general definition of the decibel is:

$$L_W = 10 \log (W/W_{ref}) \quad \text{(A.4)}$$

where W is a quantity proportional to power, W_{ref} is a reference power level, and L_W is the level in dB. Note that the decibel is a ratio or relative measure; there must always be a reference quantity (W_{ref}) and the reference quantity must always be explicitly defined or implied via conventional usage.

The decibel is used for quantities such as sound pressure level, sound power level, vibration acceleration level, and vibration velocity level. Whenever level is included in the name of a quantity, it indicates that the value is in decibels and that a reference power, pressure, or other quantity is stated or implied.

Table A-1 presents standard reference quantities used for noise and vibration measurements. Note that the definition of Sound Pressure Level (SPL) is:

$$L_p = SPL = 10 \log (p^2/p_o^2) \tag{A.5}$$

where p is the rms pressure for the sound in question. The quantity p^2 is used since pressure squared is proportional to power and decibels are generally used to indicate the ratio of two values of "power-like" quantities.

Since decibels are logarithmic, they are not added arithmetically but logarithmically. When adding two signals, the energy of the two signals is added and then the decibels calculated. For example, if the rms power of signal 1 is $6x10^{-7}$ watts and the rms power at signal 2 is $7x10^{-7}$ watts, combining signals 1 and 2 creates a new signal with an rms power of $13x10^{-7}$ watts. The power <u>level</u> of the combined signals (with a reference level of 10^{-12} watts) is:

$$L_W = PWL = 10 \log (13x10^{-7}/10^{-12}) = 61.1 \text{ dB}$$

TABLE A-1 STANDARD ACOUSTIC REFERENCE QUANTITIES (AS DEFINED IN ANSI S1.8, 1969)

Name	Definition	Preferred Reference Quantity - SI Units	Other Common Reference Quantity
SOUND PRESSURE LEVEL (Gases)	$L_p = 20 \log_{10} (p/P_0)$	$P_0 = 20\ \mu N/m^2$ $= 2 \times 10^{-5}\ N/m^2$	--
POWER LEVEL	$L_W = 10 \log_{10} (W/W_0)$	$W_0 = 1\ pw = 1 \times 10^{-12}$ watt	--
VIBRATORY LEVELS			
Acceleration	$L_a = 20 \log_{10} (a/a_0)$	$a_0 = 10^{-5}\ m/sec^2$	$a_0 = 1\ g = 39.4 \times 10^{-5}$ in./sec^2
Velocity	$L_v = 20 \log_{10} (v/v_o)$	$v_0 = 10^{-8}\ m/sec^2$	$v_0 = 5 \times 10^{-8}\ m/sec^2$ and $v_0 = 10^{-6}$ in./sec^2
Displacement	$L_d = 20 \log_{10} (d/d_o)$	$d_0 = 10^{-11}\ m$	$d_0 = 10^{-10}$ in.
Force	$L_F = 20 \log_{10} (F/F_0)$	$F_0 = 1\ \mu N = 10^{-6}\ N$	--

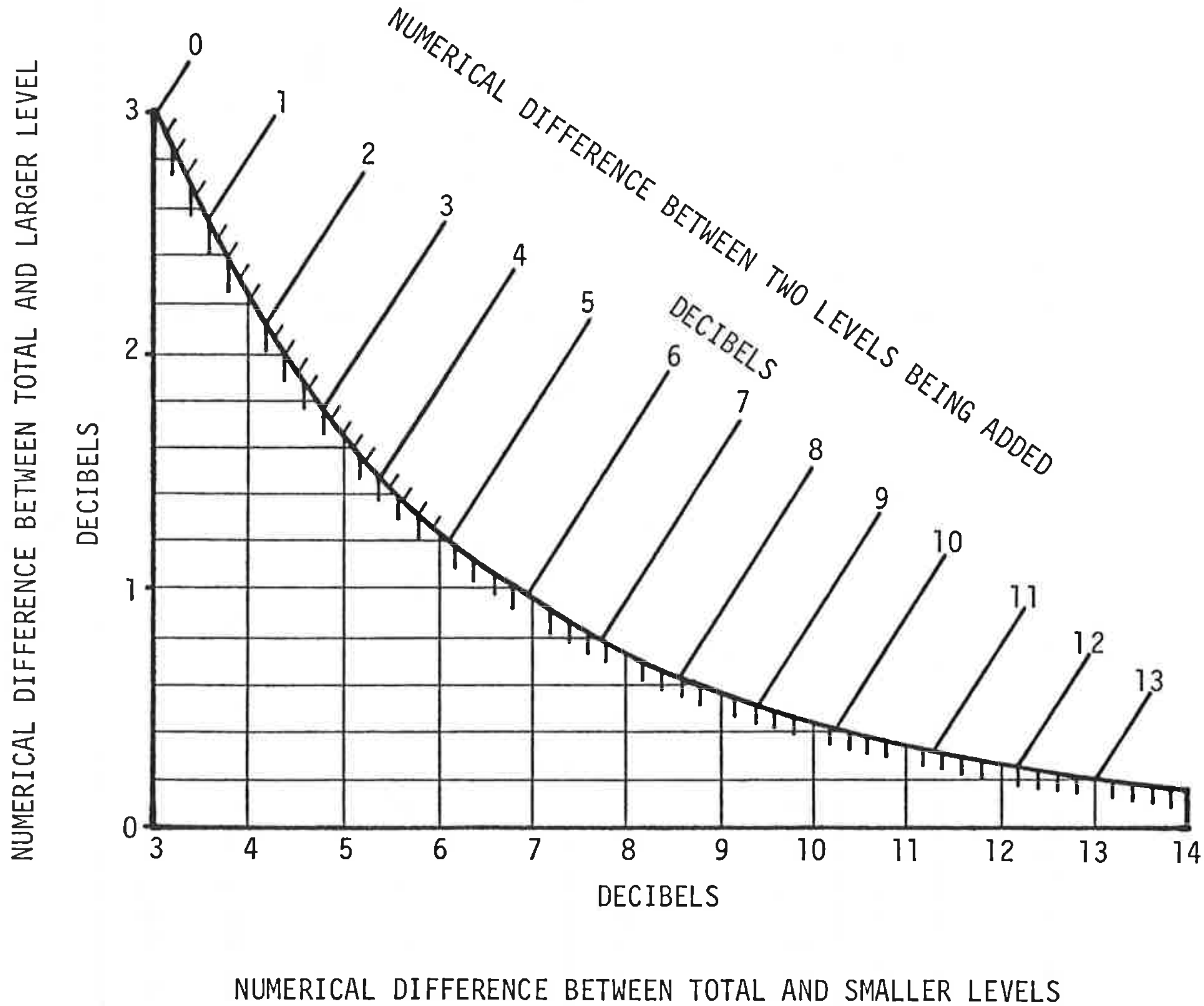


FIGURE A.5 CHART FOR COMBINING TWO SOUND LEVELS (ADAPTED FROM A.P.G. PETERSON AND E.E. GROSS, JR., HANDBOOK OF NOISE MEASUREMENT, GenRad, INC. MASSACHUSETTS, 1974)

Figure A.5 presents a simple chart that can be used to perform decibel addition when the values are given in decibels. As an example, consider adding the sound from two sources. Source A creates a level of 68 dB when source B is turned off, and source B creates a level of 65 dB with source A turned off. The question is what will the combined level be when both sources are active? Referring to Figure A.7, L_1 minus L_2 is 3 dB, hence the combined level is 68 dB plus approximately 1.8 dB giving 69.8 dB.

When the values are given in decibels, the combined source strength can also be calculated using the following relationship:

$$L_T = 10 \log (10^{.1xL_1} + 10^{.1xL_2}) \qquad (A.6)$$

where L_T is the decibel sum of L_1 and L_2. Using this relationship, the example given above of adding 68 dB and 65 dB gives a combined total of 69.76 dB.

A.1.3 Weighted Sound Levels

The human ear does not respond in a uniform manner to different frequency sounds. For example, a sound pressure level of 70 dB will sound much louder at 1000 Hz than at 100 Hz. To account for this, various weighting methods have been developed to reflect human sensitivity to noise. The purpose of the weighting methods is to de-emphasize the frequency ranges in which the human ear is less sensitive. Figure A.6 shows the weighting curves that are commonly found on sound level meters. Sound pressure levels measured with a weighting network are generally referred to as weighted sound levels. Of these weighting curves, the A-weighting curve is the only one widely used for transit-related noise measurements and specifications for community noise ordinances and standards.

The A-weighted curve shown in Figure A.6 is an almost universally accepted standard for measuring sound level in objective terms that can be related to subjective effects of the sound. Although

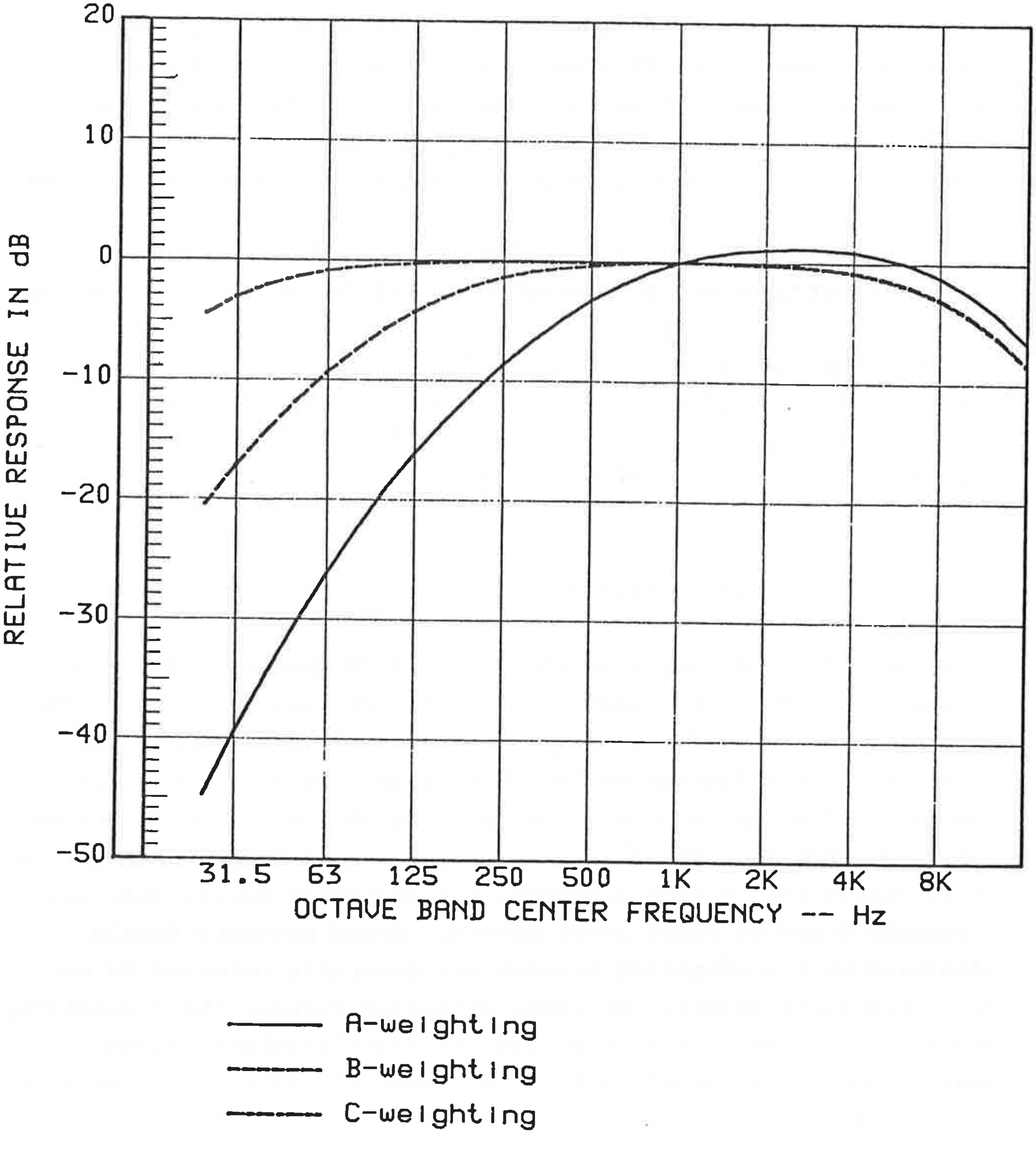


FIGURE A.6 FREQUENCY RESPONSE CHARACTERISTICS OF WEIGHTING NETWORKS a number of relatively complex techniques have been developed to reflect human perception of the loudness of sounds more accurately, psychoacoustic studies have shown these methods to be only marginally better measures of human response than the use of A-weighted decibels. Acoustics textbooks often present and discuss the "sone" and "phon" which are measures of the loudness level. However, because of the complex steps required to calculate sones and phons, they are rarely used to evaluate noise problems or the need for noise control.

As discussed above, the internationally recognized measure of sound is the A-weighted sound level, abbreviated dBA. The dBA has been shown to be highly correlated to human response to noise; that is the A-weighted level of a group of noises can be used to accurately rank order the subjective loudness of the noises. As a general rule of thumb, it has been found that in direct comparisons, a 2 dBA change in noise level can be detected, however, over a long period of time a 3 to 5 dBA change is required before the change is noticeable. A change in sound level of 10 dBA will typically be judged as a subjective doubling or halving of the loudness.

One notable disadvantage of the A-weighted sound level is that it does not accurately reflect the annoyance quality of audible pure tones, clicks, buzzes, rattles, etc. that may be part of a sound. Although the A-weighted levels of two sounds may be the same, the sound with identifiable components will be considerably more distracting, annoying, and intrusive to most people. A good example of this effect is the noise level of trains on jointed rail compared to welded rail. Although the A-weighted sound levels may be the same, the train on jointed rail will sound louder because of the joint impact noise. To account for this phenomenon, it is typical for community noise ordinances to add a 5 dBA penalty to noise that has identifiable pure tones or other annoying components.

A.1.4 Vibration Levels

In contrast to sound, vibration is often specified in terms of absolute magnitudes of displacement, velocity, or acceleration. Although this practice is traditional in a number of fields, the use of vibration levels is recommended because it is useful for sound measures. The standard reference quantities for vibration levels are presented in Table A-1. The use of these standard quantities is not uniform; some care must be used to ensure that reference quantities are clearly stated when vibration levels are presented.

Since acceleration level (L_a), velocity level (L_v), and displacement level (L_d) can all be used to describe the same motion, measurements of one quantity can be transformed to one of the other quantities. The basic relationship is:

$$a = dv/dt = d^2d/dt^2$$

where "a" is acceleration, "v" is velocity, and "d" is displacement. In terms of vibration levels, the relationships can be shown to be:

$$\begin{aligned} L_a &= L_v + 20 \log f - 44 \\ &= L_d + 40 \log f - 88 \end{aligned} \tag{A.7}$$

where f = frequency.

Although these relationships are strictly true only at specific frequencies, they can generally be used to transform octave or 1/3 octave values with only small errors if "f" is the center frequency for the band. Use of reference quantities other than those given in Table A-1 will change the constants of Equation A.7.

Note that in contrast to sound levels, there are no generally accepted weighting curves that can be used to evaluate human perception of vibration levels. Some recent work has been done in this area by the Committee on Hearing, Bioacoustics and Biomechanics Assembly of Behavioral and Social Sciences "CHABA" (Ref. 2.5). Based on the proposed ISO Standard, "Guide for the evaluation of human exposure to whole-body vibration" (Ref. 2.4), the CHABA Committee developed a tentative weighting curve for vibration that can be used to estimate human perception of the vibration environments. The use of the CHABA weighting curve and appropriate vibration criteria using the weighted levels are discussed in Chapter 2.

A.1.5 Frequency Analysis

With the use of weighting networks such as those shown in Figure A.6, one obtains a single number measure of the energy of a signal. As discussed above, the most common weighting for measurement of sound levels is the A-weighted network. In many instances, the A-weighted response will give the required information. However, it is often necessary to obtain more detailed information regarding the frequency content or distribution of a source signal. This type of analysis, called spectral analysis, yields a plot of amplitude as a function of frequency and is particularly useful when analyzing the characteristics of a noise or vibration source and when designing control measures.

Figure A.7 presents several methods of analyzing the frequency distribution of a complex signal including octave band, 1/3 octave band, and constant-bandwidth analysis. All of these charts represent analysis of the same signal. The signal consists of broadband noise with several pure tone components, the strongest pure tone component being at 1000 Hz.

Figure A.7a presents the octave band and 1/3 octave band analysis.

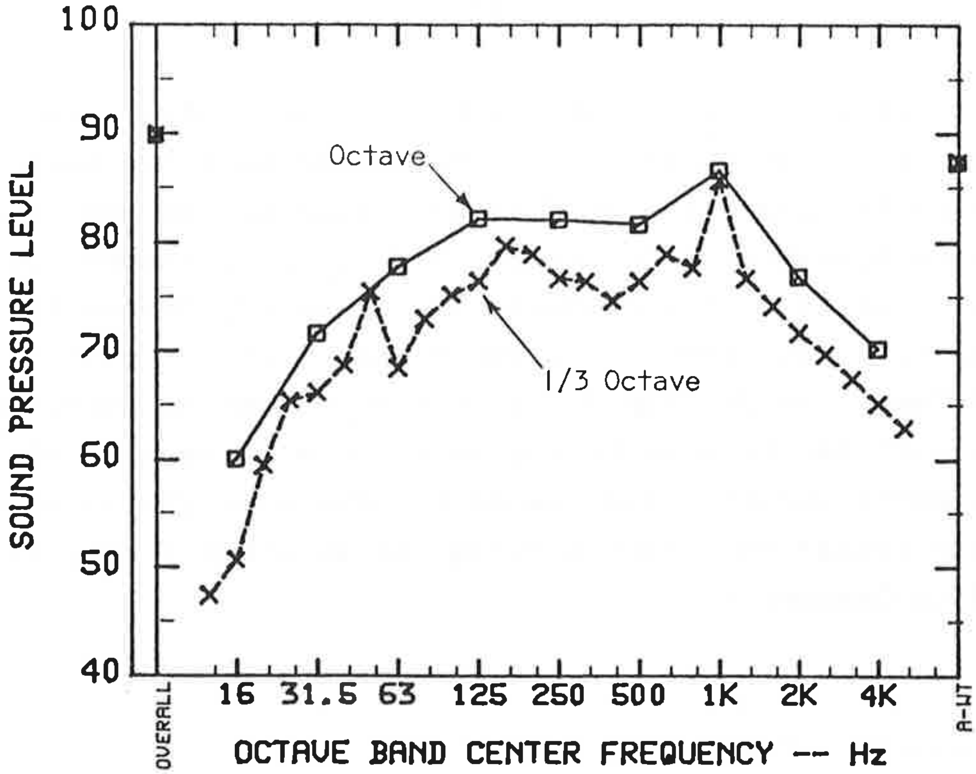


FIGURE A.7a OCTAVE AND 1/3 OCTAVE ANALYSIS

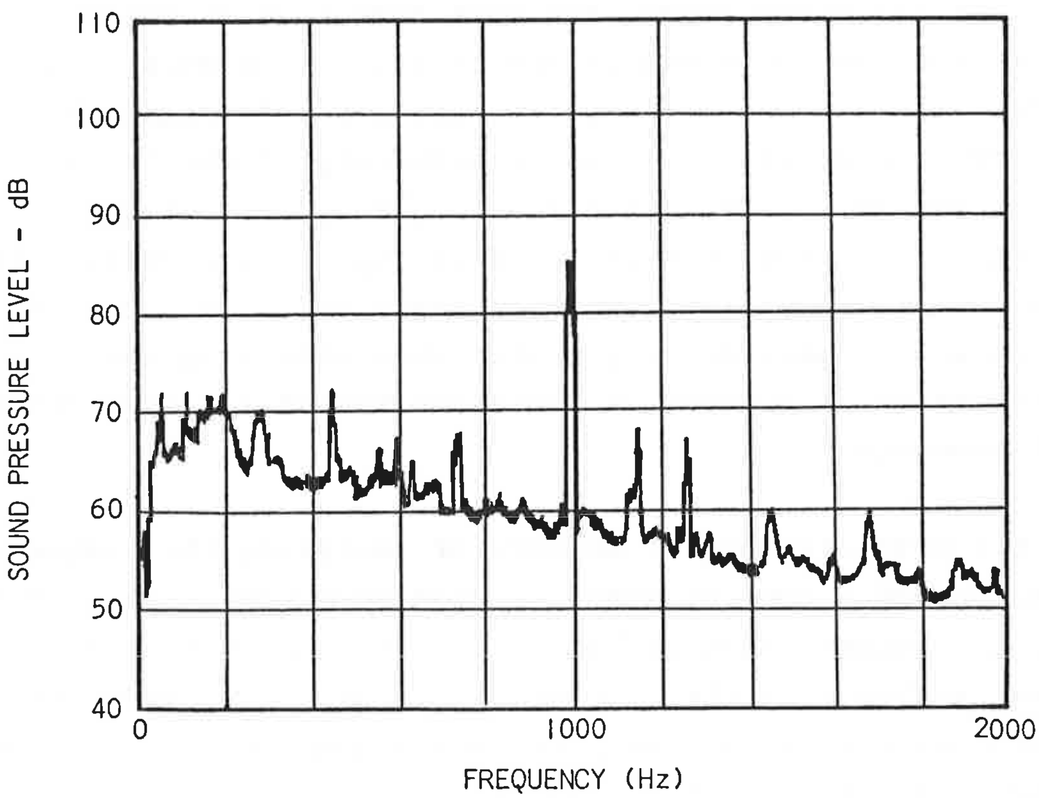


FIGURE A.7b CONSTANT-BANDWIDTH FREQUENCY ANALYSIS USING 400 LINE FFT ANALYZER

FIGURE A.7 FREQUENCY ANALYSIS

An octave covers a 2 to 1 range of frequencies, and each octave can be further subdivided into three 1/3 octaves to obtain more detail.

Figure A.8 shows the response of typical octave and 1/3 octave band filters. The ideal octave band filter would pass the portion of the signal within the frequency band of interest and remove all other frequency components. Since perfect filters are not possible (except with digital equipment which synthesizes the filter) as shown in Figure A.8, band pass filters do not have vertical skirts.

The preferred frequency limits used for octave band and 1/3 octave bands are defined in ANSI S1.6, 1967; these are presented in Table A-2. Note that the center frequencies are generally used to reference specific octave and 1/3 octave bands. Although the bands specified in Table A-2 are now an accepted standard, another series of octave bands was widely used in the past and are still occasionally encountered in publications, test codes, and noise ordinances that have not been updated.

Since octave and 1/3 octave band analysis consist of using a set of filters with contiguous frequency bands, the result is one value for each frequency band. For N contiguous frequency bands, decibel sum of the N levels is the same as would be measured with one filter that covered the entire frequency range of the N filters. In practice, this means that three 1/3 octave band levels can be combined to obtain the equivalent octave band level and that all octave or all 1/3 octave band levels can be combined to obtain the equivalent overall level.

Octave band levels are relatively easy to obtain and are often used when simple frequency analysis is required. The use of 1/3 octave bands requires more sophisticated equipment but provides more detail about the distribution of energy as a function of frequency. In Figure A.7a, the 1000 Hz pure tone could easily be overlooked if only octave band results were available, but it is

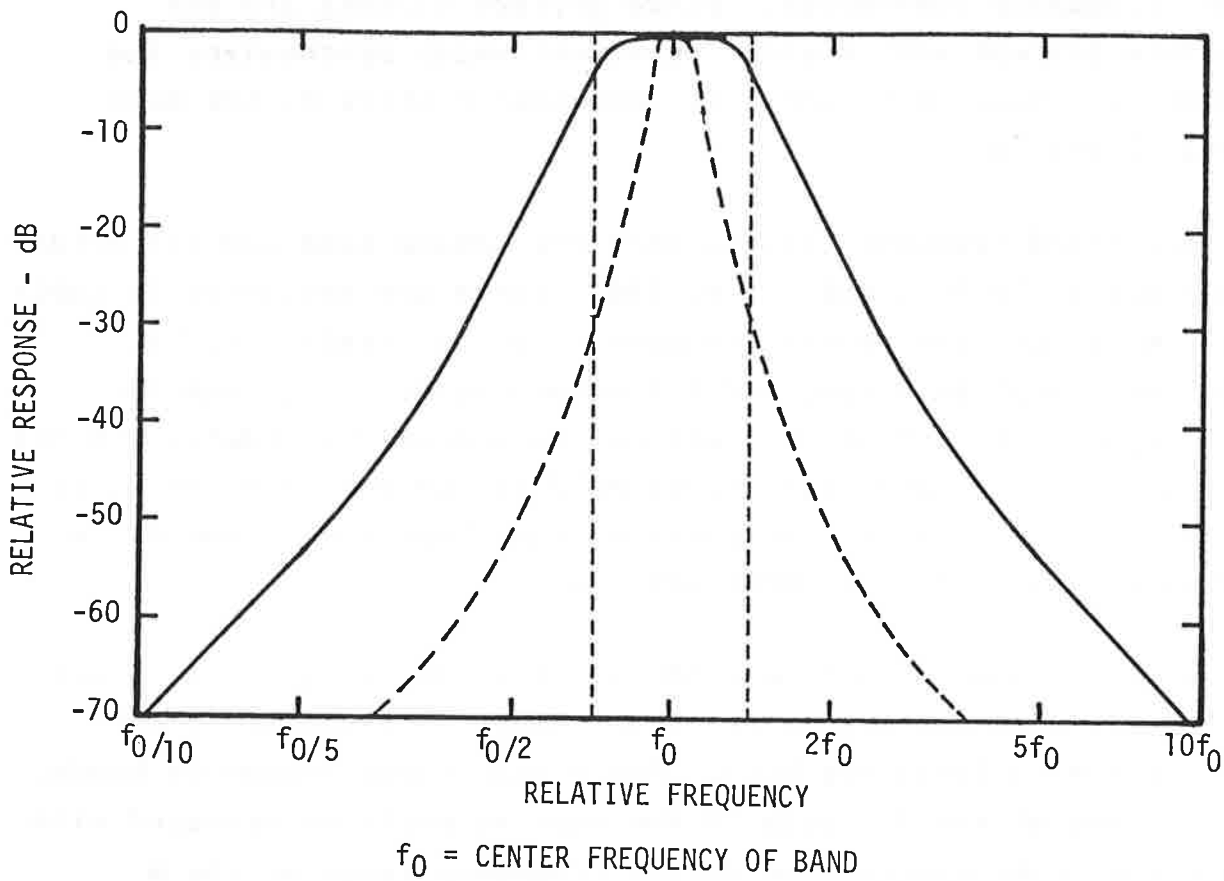


FIGURE A.8 NORMALIZED RESPONSE OF ACTUAL OCTAVE AND 1/3 OCTAVE BAND FILTERS (ADAPTED FROM L.L. BERANEK, ED., NOISE AND VIBRATION CONTROL. McGRAW-HILL BOOK COMPANY, NEW YORK, 1971).

TABLE A-2 FREQUENCY LIMITS FOR OCTAVE AND 1/3 OCTAVE BANDS (AS DEFINED IN ANSI S1.6, 1967[R1971])

Band	Frequency - Hz					
	Octave			One-Third Octave		
	Lower Band Limit	Center	Upper Band Limit	Lower Band Limit	Center	Upper Band Limit
12	11	16	22	14.1	16	17.8
13				17.8	20	22.4
14				22.4	25	28.2
15	22	31.5	44	28.2	31.5	35.5
16				35.5	40	44.7
17				44.7	50	56.2
18	44	63	88	56.2	62	70.8
19				70.8	80	89.1
20				89.1	100	112
21	88	125	177	112	125	141
22				141	160	178
23				178	200	224
24	177	250	355	224	250	282
25				282	315	355
26				355	400	447
27	355	500	710	447	500	562
28				562	630	708
29				708	800	891
30	710	1000	1420	891	1000	1122
31				1122	1250	1413
32				1413	1600	1778
33	1420	2000	2840	1778	2000	2239
34				2239	2500	2818
35				2818	3150	3548
36	2840	4000	5680	3548	4000	4467
37				4467	5000	5623
38				5623	6300	7079
39	5680	8000	11,360	7079	8000	8913
40				8913	10,000	11,200
41				11,220	12,500	14,130
42	11,360	16,000	22,720	14,130	16,000	17,780
43				17,780	20,000	22,390

clearly evident when the 1/3 octave band results are available. Referring to Figure A.7b, it can be seen that the 1000 Hz tone is the dominant component, and any noise control effort should first concentrate on the source of this peak.

When comparing 1/3 octave band levels with octave band levels, it is best to combine the 1/3 octave levels in groups of three to obtain equivalent octaves. However, another approach that is often used is to shift the chart down by 5 dB to approximately convert from octave results to 1/3 octave or upward 5 dB for the reverse. The 5 dB factor is based on the assumption that the levels in the three 1/3 octaves in each octave are approximately equal. When the three 1/3 octave band levels are combined, the total is 5 dB higher. As illustrated in Figure A.7a, this assumption is reasonable except in cases such as the 1000 Hz octave in Figure A.7a. Since the level of the 1000 Hz 1/3 octave dominates the 1000 Hz octave band, the level of the 1000 Hz octave band is only 1 dB above the level of the 1000 Hz 1/3 octave band.

In most cases, 1/3 octave band analysis will provide sufficient detail about the frequency content of noise and vibration. However, analysis with narrower frequency bands is essential for some purposes. Figure A.7b is an example of constant-bandwidth narrowband analysis using a Fast Fourier Transform (FFT) type analyzer. This technique is discussed in more detail in Chapter 5; for the purpose of this section, it is sufficient to say that Figure A.7b is a constant-bandwidth display. The advantage of this type of display is that each of the tonal components of the reference signal is clearly displayed. Constant-bandwidth analyzers can consist of either an analog filter that has the same bandwidth in Hertz over the full range of the analyzer or a digital device that implements algorithms such as the FFT.

It is interesting to compare the bandwidth of a constant-bandwidth analyzer with that of a 1/3 octave band analyzer. The 1/3 octave bands at 40, 400, and 4000 Hz have bandwidths of 9.26, 92.6, 926 Hz, respectively. Thus, a constant-bandwidth analyzer with a 20

Hz bandwidth is much narrower at 4000 Hz, but at low frequencies the constant-bandwidth analyzer is much wider than the 1/3 octave analyzer.

It is not uncommon for the results of narrowband frequency analysis to be presented in terms of spectrum level. The spectrum level of a noise is the level that would be measured if an analyzer had an ideal response characteristic with a bandwidth of 1 Hz at all frequencies. In other words, the level is normalized to a bandwidth of 1 Hz. The main uses of this concept are comparing data taken with analyzers with different bandwidths and checking compliance with specifications given in terms of spectrum level.

A.2 CHARACTERIZING NOISE ENVIRONMENTS

A.2.1 Measures of Sound Level

There are a number of methods that can be used to measure the instantaneous magnitude of sounds. However, there are only a few that one is likely to encounter being applied to rapid transit systems. The term "instantaneous" is used to differentiate from long-term averages that are discussed in the next section. The following are a list of measures that may be applied to various types of transit work.

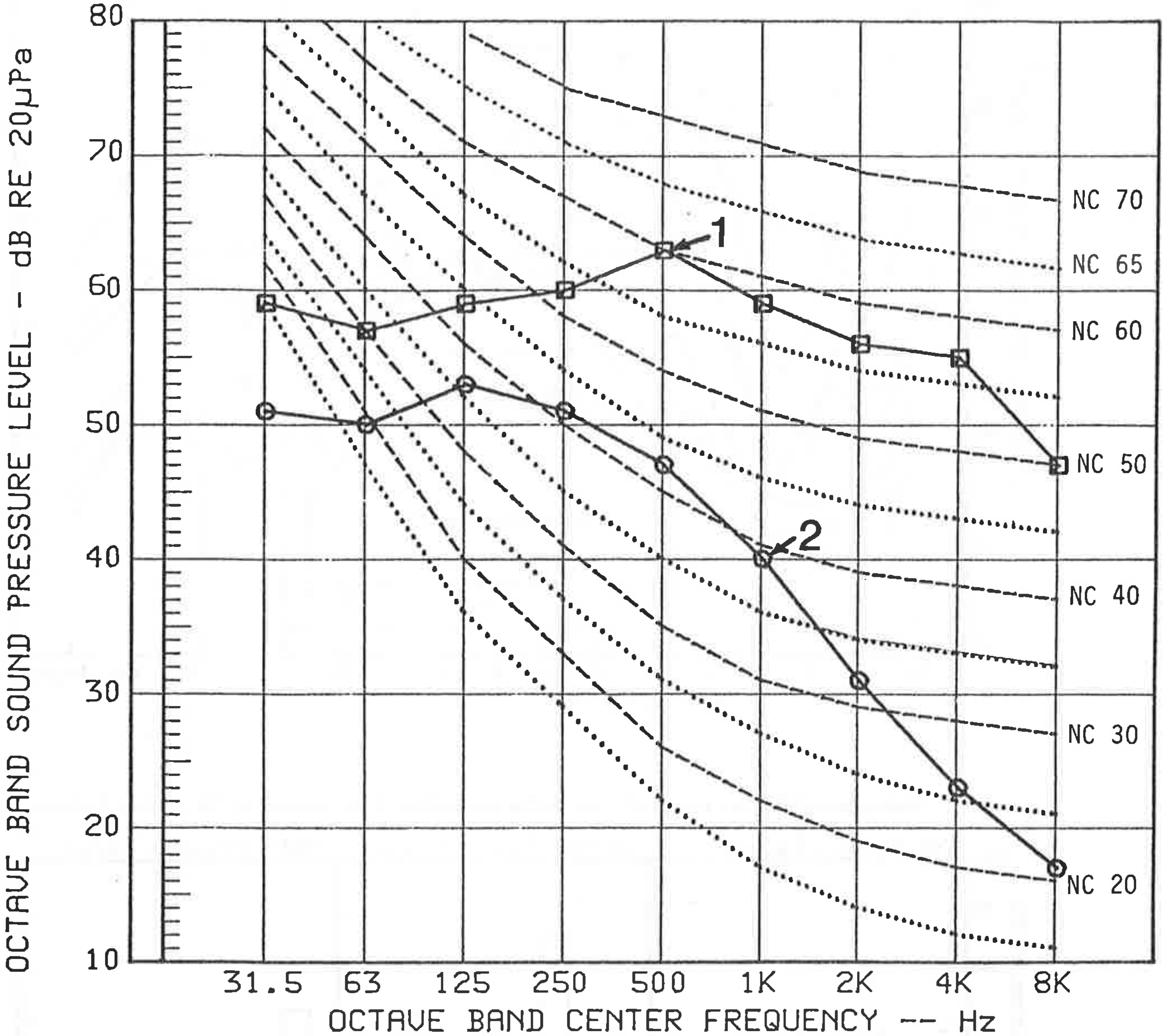
A.2.1.1 A-weighted Level -- As discussed in the previous section, the A-weighted level (dBA) is the most common metric used to measure overall sound level. Although other weighting curves, including the B, C, D, and E weighting curves, are available, the A-weighted level is universally accepted as being a reasonably accurate reflection of human response to noise.

A.2.1.2 Noise Criterion (NC) Curves -- NC curves are commonly used for noise criteria indoors and are widely used in architectural specifications; NC curves and their use are illustrated in Figure A.9. The NC curves are used by plotting the octave band levels graph paper with the NC curves superimposed. The NC level for a particular octave band spectrum is the maximum NC curve that the spectrum touches. Hence, using the NC curves gives a rating for the acceptability of an acoustic environment and indicates the frequency range that dominates the overall rating. Another form of the NC curves called the preferred noise criteria (PNC) curves, which were designed to overcome some objections to the NC curves, are also occasionally encountered. The NC curves were designed to be applied only for steady, continuous spectrum, broadband noise containing no significant pure tones. However, experience has shown that the NC curves can be used to specify noise environments or evaluate noise even when significant pure tones occur. Since the NC value for a spectrum is the highest NC curve reached by any component of the spectrum, NC curves are responsive to the effects of pure tones - considerably more responsive than overall measures such as the dBA.

A.2.1.3 Sones and Phons -- Sones and phons were developed to give a measure of the loudness of noises. The procedure is designed to approximate the mean of the responses of a large number of individuals who were asked to make subjective judgments of loudness level or loudness for that noise. However, because the procedure to calculate sones or phons from octave or 1/3 octave sound levels is relatively complex and the method has not been adapted to simple handheld meters, sones and phons are very rarely encountered as a measure of transit noise.

A.2.2 Measures of Sound Environment

Since noise, particularly outdoor noise, continuously and randomly varies with time, there is a need for measures of noise climate that account for these variations. Figure A.10 illustrates an example of measures that are commonly used to evaluate community noise. The quantities shown on the graph are defined below:



CURVE 1 IS NC 60

CURVE 2 IS NC 42

FIGURE A.9 USE OF NC CURVES

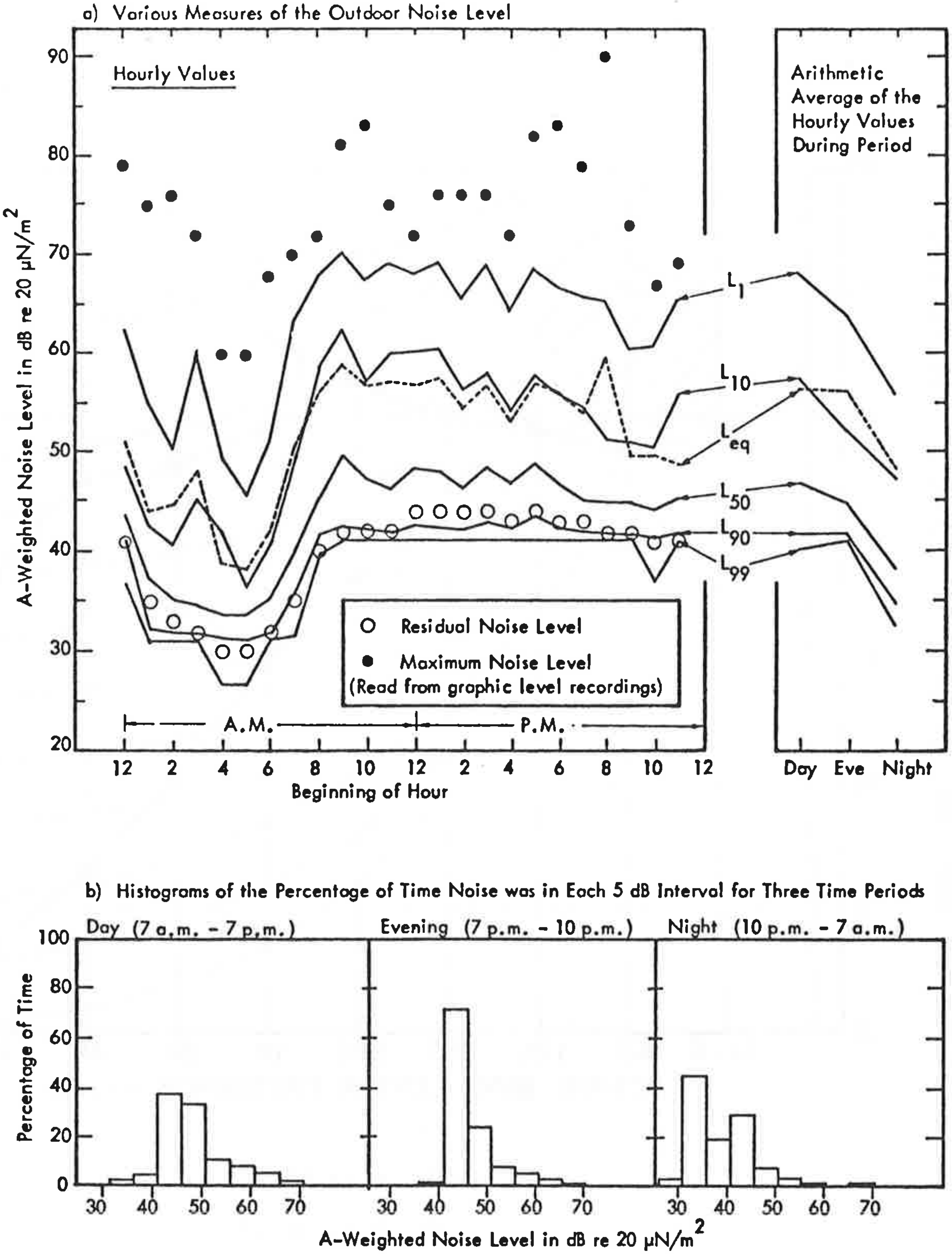


FIGURE A.10 SUMMARY OF THE 24-HOUR OUTDOOR NOISE LEVELS AT LOCATION M - SUBURBAN RESIDENTIAL AT CITY OUTSKIRTS (TAKEN FROM "COMMUNITY NOISE," NTID300.3, U.S. ENVIRONMENTAL PROTECTION AGENCY, DECEMBER 31, 1971, p. A-56)

A.2.2.1 <u>Equivalent Sound Level - L_{eq}</u> -- Sometimes referred to as the energy average sound level over the period of interest, L_{eq} has been widely accepted as a valid measure of community noise. The equivalent sound level is equal to the equivalent steady noise level which in a stated time period would contain the same energy as the time-varying noise during the same time period. Mathematically, it is defined as:

$$L_{eq} = 10 \log \left[\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} \frac{p^2(t)}{p_o^2} dt\right] \qquad (A.8)$$

where p(t) is the time varying pressure and p_o is the standard reference pressure of 20 Pa (0.0002 dynes/cm^2). Note that this is equivalent to the definition of rms level in Equation A.1, however, rms is generally used for much shorter periods of time than L_{eq}.

The average sound level over a period of time T is often symbolized as $L_{eq(T)}$ or simply L_T. Commonly used are:

L_h = hourly average sound level

L_{8h} = eight-hour average sound level

L_d = average daytime sound level

L_n = average nighttime sound level

The term "average" indicates energy average.

A.2.2.2 <u>Day-Night Average Sound Level - L_{dn}</u> -- L_{dn} is a 24-hour average sound level in which the nighttime noise levels occurring between 2200 and 0700 are increased by 10 dBA before calculation of the 24-hour average. The 10 dBA penalty is meant to account for people's increased sensitivity to noise during the nighttime hours when many people are asleep and the background noise is low. L_{dn} is the primary measure used for describing noise in Environmental Impact Statements.

A.2.2.3 Community Noise Equivalent Level (CNEL) -- CNEL is essentially the same as L_{dn} except that the 24-day period is broken into three periods -- day (0700 to 1900), evening (1900 to 2200), and night (2200 to 0700). Weightings of 5 dBA are applied to the evening period and 10 dBA to the nighttime period. In most cases, CNEL will be 0 to 1/2 dB higher than L_{dn}, an insignificant difference.

A.2.2.4 Level Exceeded "n" Percent of the Time - L_n: The level exceeded a certain percentage of time is a widely used measure of environmental noise. Typical measures are L_1, L_{10}, L_{50}, L_{90}, and L_{99}. The time period referred to can range from a full 24-hour period to a several minute spot check. To avoid confusion, the time period should be clearly specified.

Figure A.11 illustrates another form that is used to display the fluctuation of community noise. The vertical axis is the percent of time, and the horizontal axis is the noise level in dBA. As shown in the figure, typical community noise with the ambient noise resulting from a large number of relatively distant noise sources will be a nearly straight line on the chart. Adding a nonrandom noise source such as rapid transit trains that create relatively high noise levels for relatively short periods of time will modify the percent exceeded chart as indicated in Figure A.11. Note that in this example, the train noise increased L_1 by 15 dBA, L_{10} by 1 dBA, and L_{50} was unchanged. This illustrates the problem with using statistical measures such as L_{10} or L_{50} to evaluate community reaction to transit noise.

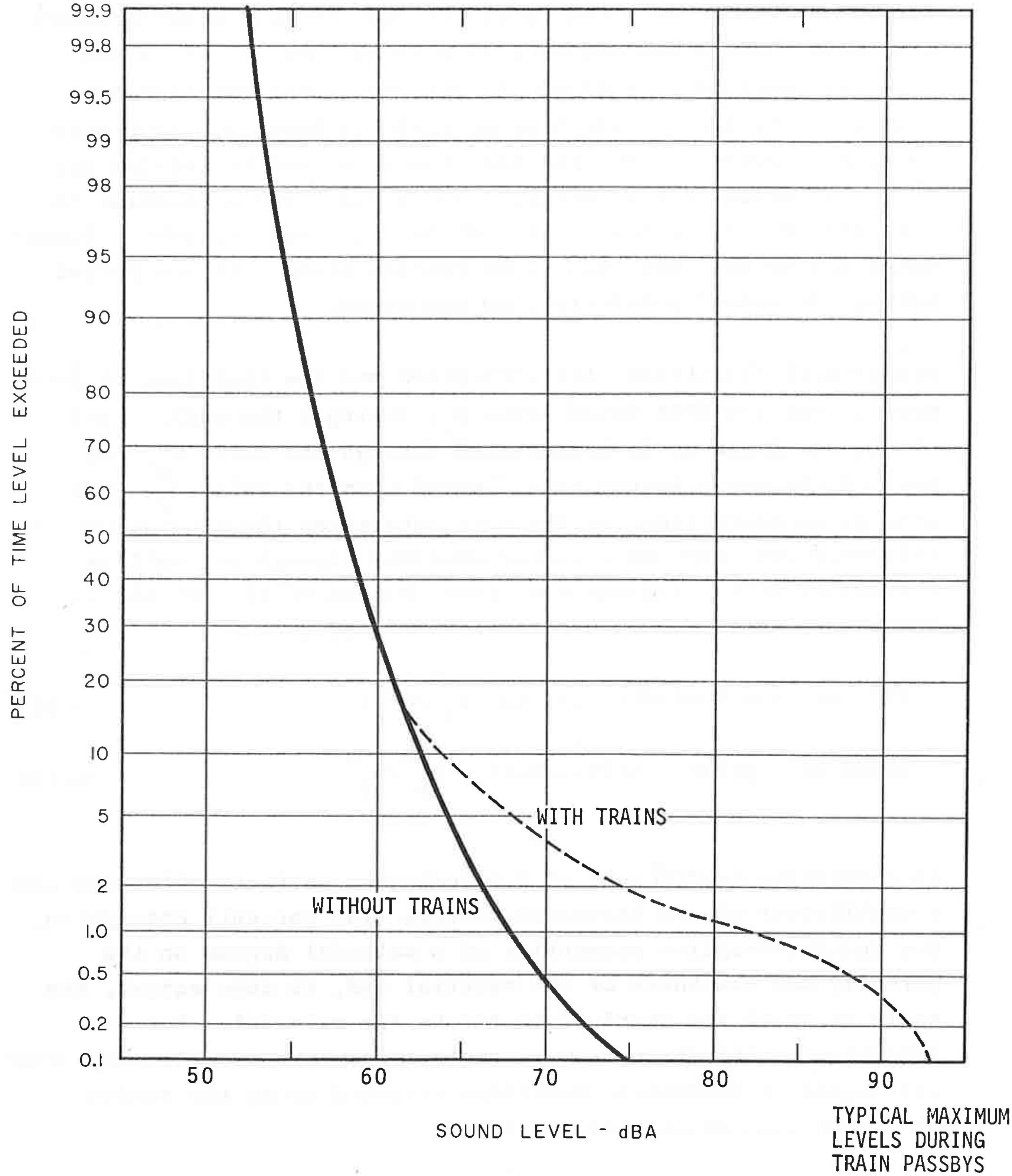


FIGURE A.11 STATISTICAL DISTRIBUTIONS FOR TYPICAL COMMUNITY NOISE, 7.5 TO 15 m FROM TRACK CENTERLINE

A.3 SOUND ABSORPTION AND ISOLATION

Sound absorption and sound isolation are commonly misunderstood phenomena. Sound absorption refers to the ability of porous materials such as fiberglass blankets or acoustical tile to convert sound energy into heat or fluid turbulence. Absorbing materials reduce sound reflected from a surface by letting air into and through the material. In contrast, for a material to effectively isolate sound, it must be solid and airtight. Porous materials do not stop, block, or contain sound. If air passes through or around a material, so can sound.

Figure A.12 illustrates the absorption and the isolation of sound waves. The incident sound wave, p_i, impinges the wall. Part of the sound energy is transmitted through the wall, p_t, and part of the sound energy is reflected from the wall, p_r. The absorption coefficient of the wall determines the strength of the reflected wave and the sound transmitted through the wall is determined by the transmission loss characteristics of the wall. These quantities are mathematically defined as:

$$\text{Transmission Loss(TL)} = 10 \log (p_i^2/p_t^2) \qquad \text{(A.9)}$$

$$\text{Sound Absorption Coefficient} = 1 - p_r^2/p_i^2 \qquad \text{(A.10)}$$

An absorption coefficient of 0.0 indicates perfect reflection and a coefficient of 1.0 indicates perfect (100 percent) absorption. The sound absorptive properties of a material depend on the porosity and thickness of the material and, to some extent, the angle at which the sound waves strike the material. For convenience, the absorption reported is generally the average over all angles of incidence, sometimes referred to as the random incidence absorption coefficient.

The transmission loss indicates the amount of incident energy that is transmitted through a wall or other barrier. Note that transmission loss is given in dB. A large TL indicates the wall is a very effective sound isolator, and a TL of zero indicates an

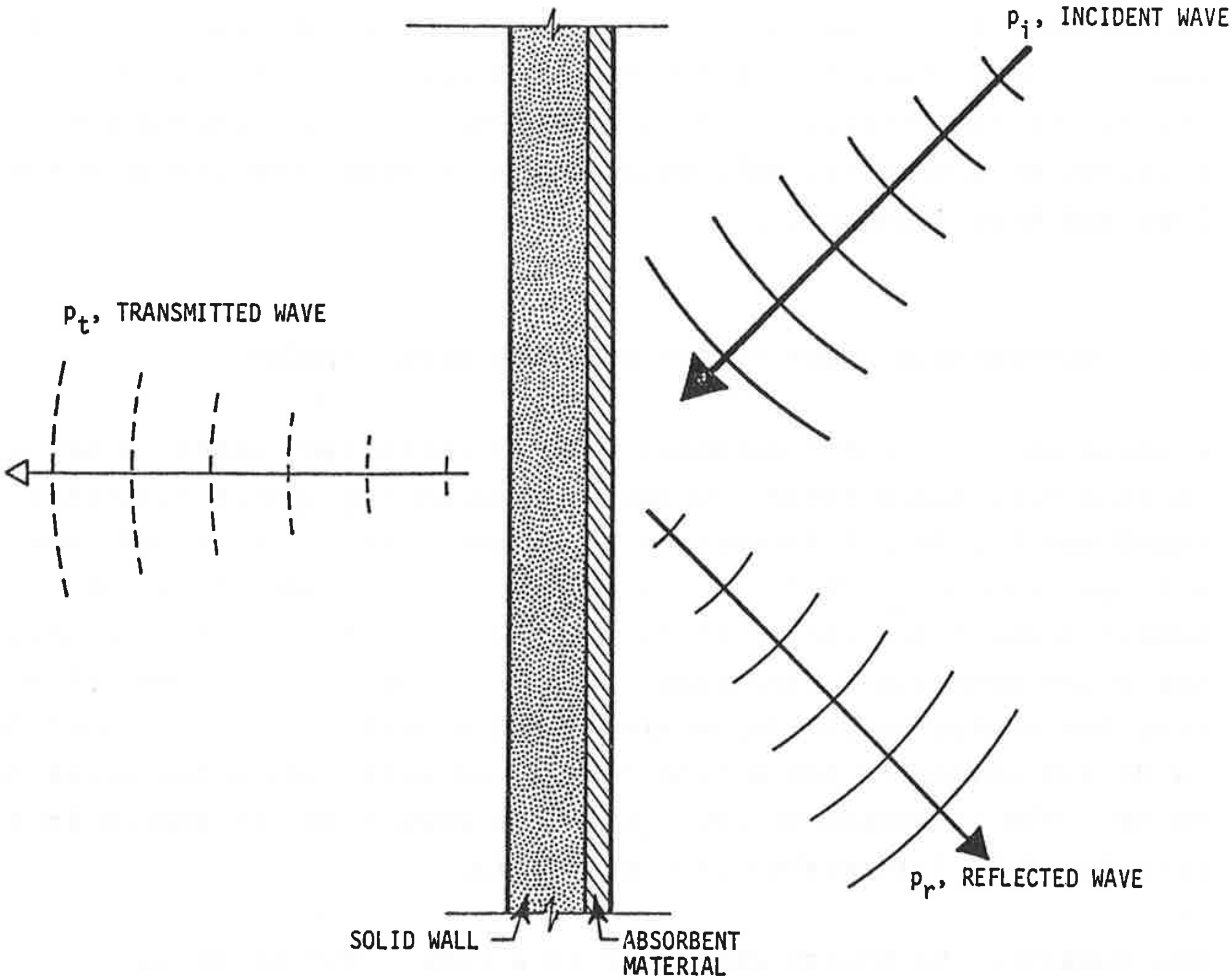


FIGURE A.12 INCIDENT SOUND WAVE ON AN ABSORBENT SURFACE acoustically transparent wall. Absorption coefficients and transmission losses of some common materials as a function of frequency are listed in Table A-3. Note that materials such as glass-fiber insulation have high absorption coefficients but very low transmission loss while in contrast, solid materials such as concrete walls have relatively high transmission loss but very low absorption coefficients. It is clear that mounting fiberglass blankets on a concrete wall will result in both high transmission loss and high absorption.

A.4 REVERBERANT SOUND FIELDS AND FREE SOUND FIELDS

A noise source located outdoors with no reflecting surfaces nearby is in a free sound field. Consider a pulsating sphere radiating sound equally in all directions in a free field. The sound level will decrease by 6 dB for every doubling of distance from the source because the energy is distributed over four times the area for every doubling of distance. For instance, if the level 10 m from the source is 100 dB, moving to 20 m will reduce the level to 94 dB and moving to 100 m from the source will reduce the level to 80 dB. The reduction of level with distance from the source in a free field is illustrated in Figure A.13.

Now consider the source contained in a room. The sound is reflected by the room surfaces with absorption of energy occurring at each reflection. The amount of energy absorbed is dependent upon the absorption coefficients of the surface materials. In this case, the sound level is no longer a simple function of distance from the source. The reduction of level with distance from the source in the room is also shown in Figure A.13. Close to the source, the level is essentially the same as before. Moving further away from the source, the level drops at a rate slower than 6 dB per distance doubling and finally reaches a plateau where the level no longer decreases with distance from the source.

TABLE A-3 ACOUSTICAL PERFORMANCE OF COMMON MATERIALS

MATERIAL	ABSORPTION COEFFICIENTS 125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Brick, unglazed	.03	.03	.03	.04	.04	.07
Brick, glazed, painted	.01	.01	.02	.02	.02	.03
Carpet, heavy, on concrete	.02	.06	.14	.37	.60	.03
Concrete	.01	.01	.015	.02	.02	.02
Concrete Block, coarse	.36	.44	.31	.29	.39	.25
Concrete Block, painted	.10	.05	.06	.07	.09	.08
Glass Fiber on Concrete						
1" thick	.08	.30	.65	.80	.85	.85
2" thick	.20	.55	.80	.95	.90	.90
Plaster on tile or brick	.14	.10	.06	.05	.04	.03
Gypsum Board, 1/2" nailed to 2 x 4's, 16" o.c.	.29	.10	.05	.04	.07	.09
Typical Acoustical Tile 3/4" thick -						
Cemented to plaster	.09	.31	.92	.94	.75	.71
Hung Ceiling	.56	.43	.71	.94	.79	.68

TABLE A-3 ACOUSTICAL PERFORMANCE OF COMMON MATERIALS (CONT.)

MATERIAL	TRANSMISSION LOSS 125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	STC Rating
6" Concrete with 1/2" plaster both sides (80 psf)	39	42	50	58	64	67	53
9" brick with 1/2" plaster each side (60 psf)	41	43	49	55	57	60	52
2" solid plaster on metal lath (18 psf)	20	22	22	27	36	42	24
2 x 4 studs 16" o.c., 1/2" gypsum board both sides (6 psf)	10	28	33	42	47	41	32
6" concrete block wall, painted (34 psf)	37	36	42	49	55	58	44
Sheet metal - 22 gage	16	20	24	29	35	43	29
2" thick glass fiber, (.25 psf) paper facing	0	0	6	12	16	23	10

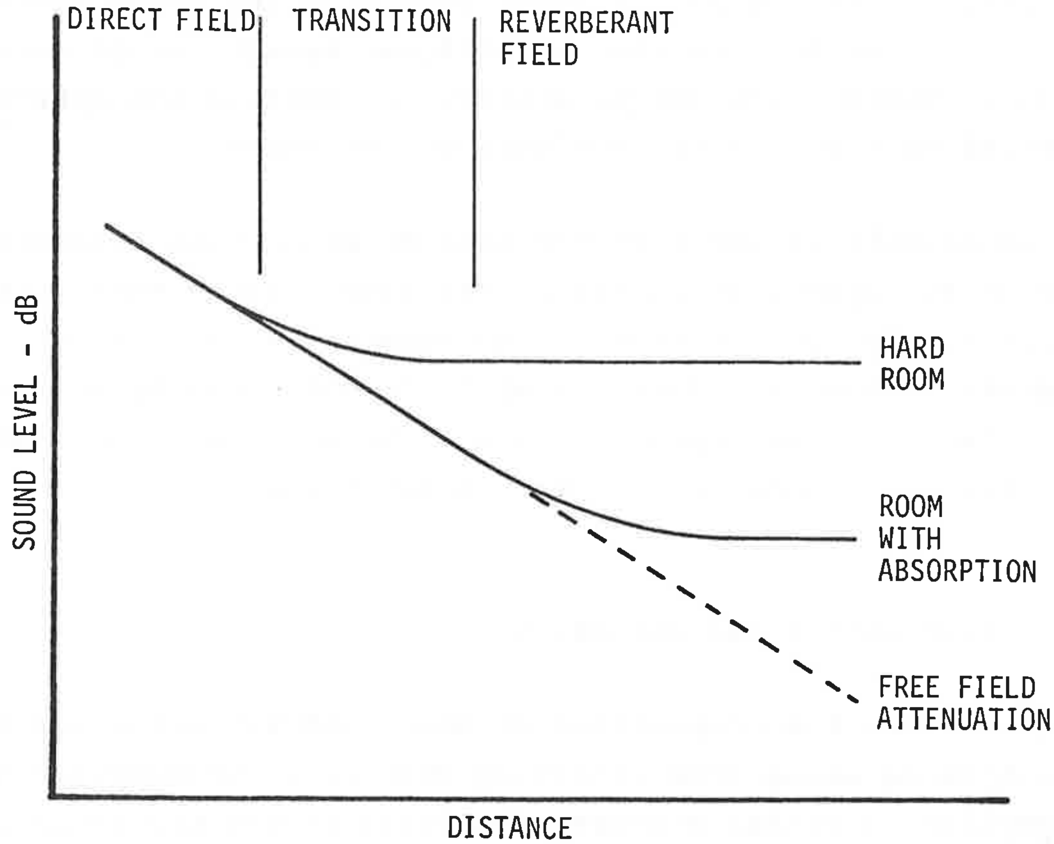


FIGURE A.13 SOUND LEVEL AS A FUNCTION OF DISTANCE FROM A SOUND SOURCE IN AN ENCLOSURE

The region where the sound level is constant is referred to as the reverberant field. In the reverberant field, the sound from the reflections dominate the direct sound. By increasing the absorption of the room surfaces, the reverberant sound level is decreased, and the transition from the direct field to the reverberant field occurs further away from the sound source. The implication of this is that significant reductions of sound level in an enclosed space can be achieved by putting absorptive material on the interior surfaces of the space.

A good example of the effectiveness of absorption treatment in reverberant spaces is in subway stations. Since subway stations traditionally consist of hard, nonabsorptive surfaces such as concrete, dramatic reductions of train noise can be achieved by installation of absorption treatment in previously untreated stations as is done in new station facilities.

A.5 PROPAGATION AND RADIATION

Understanding the propagation of sound and vibration and the radiation of sound from vibrating objects is fundamental before attempting to design measures that will reduce the resulting noise or vibration at the receiver. Radiation generally refers to the radiation of sound waves from vibrating objects, and propagation refers to the path by which the sound or vibration travels from one location to another. Propagation and radiation of sound and vibration are very complex subjects, and only the basic concepts will be discussed here.

A.5.1 Sound Radiation

When a structure is set into vibratory motion, the air surrounding the structure is set into motion resulting in sound radiation. The amount of sound radiated is described by the sound power level (abbreviated L_W or PWL). As distance from the sound source is increased, the sound level decreases. However, L_W is a measure of the acoustic energy radiated by the source, and hence L_W is not dependent on the distance.

There are several formulas that are useful for relating sound level to source L_W. Under the assumption that the sound intensity is uniform over an area S, the sound power and the sound level are related as follows:

$$L_p = L_w - 10 \log(S) \tag{A.11}$$

where S is the surface area in square meters and L_p is the sound level (SPL) in dB. For a sound source radiating uniformly in all directions (a monopole source) into a spherical space, the relation is:

$$L_p = L_w - 10 \log (4\pi r^2) \tag{A.12}$$

where r is the distance from the source in meters. If the monopole source is radiating into a hemispherical space (for example, a monopole source on a reflective ground surface), the relationship is:

$$L_p = L_w - 10 \log (2\pi r^2) \tag{A.13}$$

When the sound source does not radiate uniformly in all directions, a directivity factor must be included. The directivity index, assumed to be independent of distance from the source, is defined as:

$$DI(\Phi) = L_p(\Phi) - L_{ps} \tag{A.14}$$

where L_p is the level measured at a distance r from the source and L_{ps} is the level that would be measured at that location if the source radiated uniformly in all directions. Note that the directivity index is often written as $DI(\Phi)$ to indicate that it is a function of angle from a reference axis.

The directivity index, in the relationship for sound radiated into a hemispherical space, gives:

$$L_p = L_w - 10 \log (2\pi r^2) + DI \tag{A.15}$$

Now instead of a pulsating sphere consider a vibrating surface. If a large plane surface is vibrating in phase (e.g., a large piston) with a velocity amplitude of v, it will create a pressure wave with a magnitude of:

$$P = \rho c v \tag{A.16}$$

where P is the pressure, ρ is the air density, and c is the speed of sound in air. The quantity c is referred to as the acoustic impedance. The acoustic power radiated by a large piston vibrating in phase is:

$$W = \rho c v^2 S \tag{A.17}$$

where S is the area of the vibrating surface. Real vibrating surfaces are less efficient at radiating sound power than large pistons. The radiation efficiency is defined as the acoustic power actually radiated by a vibrating surface divided by the power that would be radiated by a piston (all parts vibrating in phase) with the same rms velocity. In mathematical form, the radiation efficiency is:

$$\sigma_{rad} = W_{rad}/\rho c S v^2 \tag{A.18}$$

Putting all of these formulas together, the sound pressure level at a distance r from a vibrating plate can be determined. Assume that the radiation efficiency of the plate is σ_{rad}, the directivity index is DI, and the plate is radiating into a half-space. The sound pressure level is:

$$L_p = 10 \log \frac{S_\rho V^2 \sigma_{rad}}{p^2_{ref}} \quad -10 \log (2\pi r^2) + DI$$

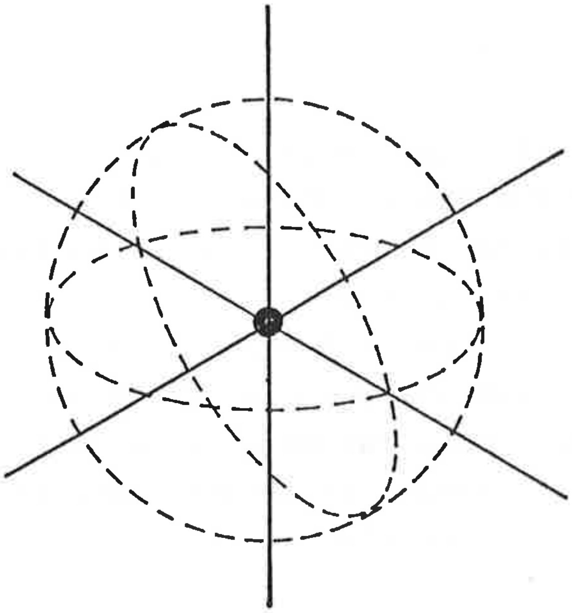
$$= L_v + 10 \log S\sigma_{rad}/2\pi r^2 + DI - 14 \qquad \text{(A.19)}$$

where L_v is vibration velocity level. The values of the reference quantities are given in Table A-1. Note that the pressure level is directly proportional to the average velocity level of the vibrating surface. Once the radiation efficiency of a vibrating surface is approximated or calculated, one can estimate the noise radiated from a vibrating surface by measuring the vibration level of the surface. This can be an important technique when trying to compare the amount of acoustic energy radiated from several simultaneously vibrating surfaces.

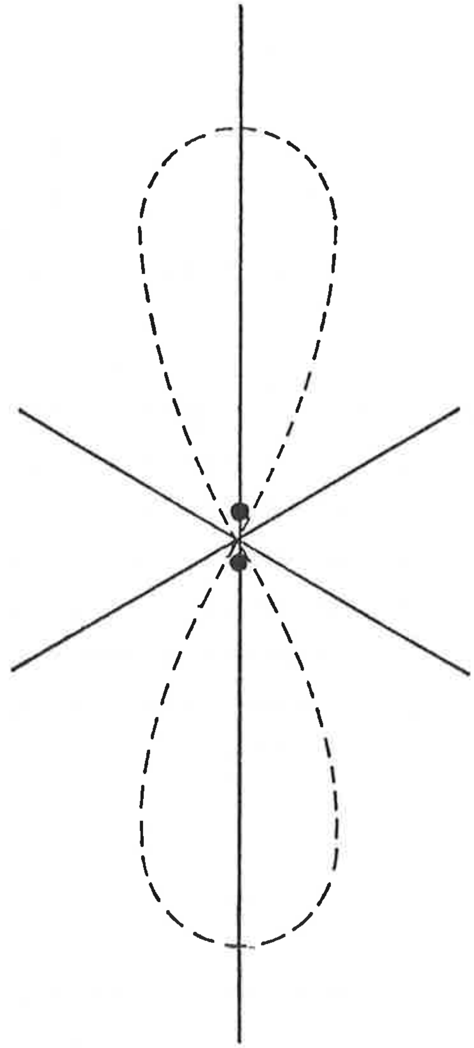
A.5.2 Sources of Sound

In most applications, sound sources can be idealized as either point sources or line sources. The most common idealization is that of a monopole point source radiating sound uniformly in all directions. This idealization can be used for any sound source which has dimensions that are small compared with the wavelength of sound. In addition, even a relatively large nondirectional sound source can be idealized as a point source at a large distance compared to the source dimensions. An example is a noisy factory building radiating sound uniformly through the walls. At large distances, the factory will act as a point source located at the center of the building.

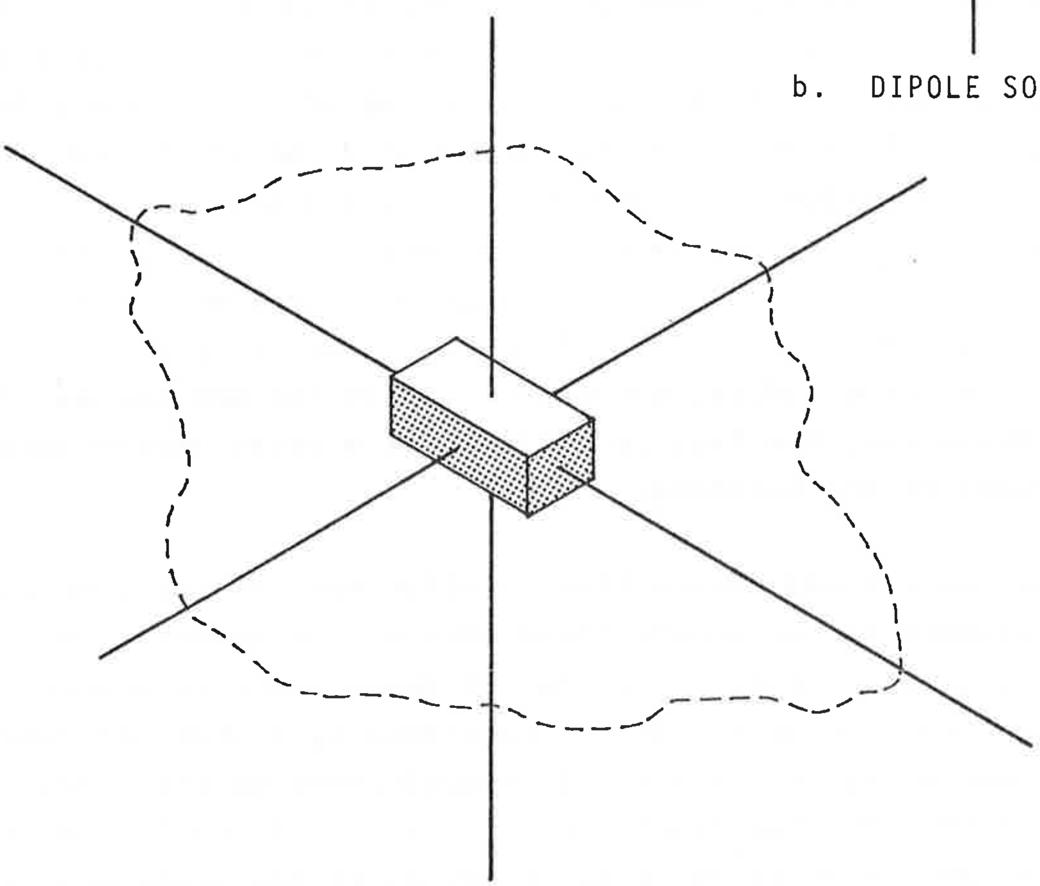
Another ideal sound source that is often used to model practical sound sources is the dipole sound source. Mathematically, the dipole source consists of two out-of-phase monopole sources with equal source strengths that are separated by a short distance. The directivities of dipole and monopole sources are compared in Figure A.14. For the dipole source, the sound level intensity is strongly dependent on the angle Φ, which is the angle measured from the axis joining the two sources. The maximum intensity



a. MONOPOLE SOURCE,
UNIFORM DIRECTIVITY



b. DIPOLE SOURCE



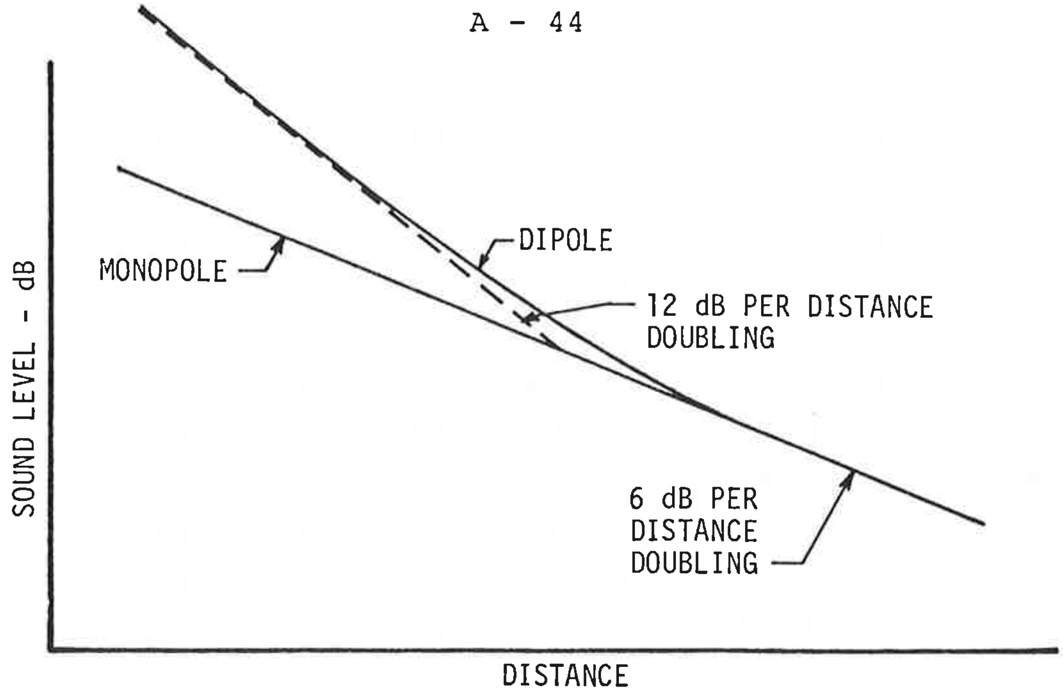
c. TYPICAL MACHINE NOISE SOURCE

FIGURE A.14 DIRECTIVITY OF REAL AND IDEAL NOISE SOURCES occurs at $\Phi=0$ and $\Phi=180^{o}$; at $\Phi=+90^{o}$, the two sources cancel each other out, and the intensity is equal to zero. Practical sources that behave like dipoles are tuning forks, pure tone fan noise, and unbaffled loudspeakers (e.g., loudspeakers without any enclosure). There is some evidence that vibrating railroad wheels behave like dipole sound sources. Note that typical real sound sources such as a shop machine will not radiate in a nondirectional manner and will not be well approximated as a dipole. In such cases, the source may be modeled as a higher order source (e.g., a quadrupole), but it is more common to describe the source in terms of an empirically obtained directivity index.

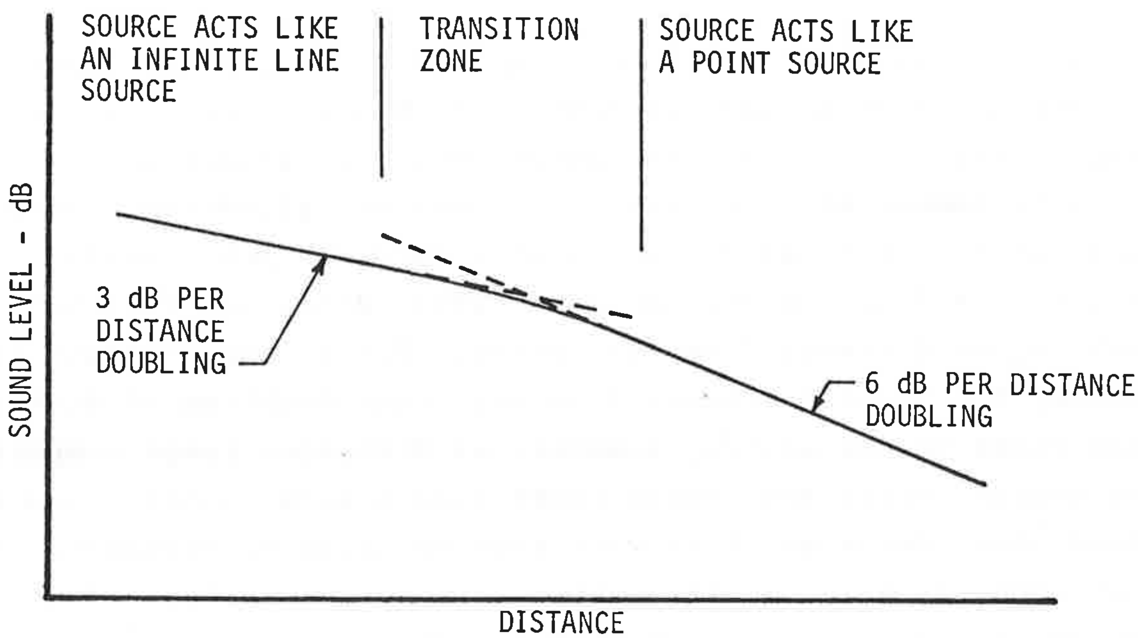
Figure A.15 compares the attenuation of sound with distance from dipole and monopole point sources. Note that close to the dipole source, the attenuation is -12 dB with each doubling of distance while further away the attenuation is -6 dB attenuation with each doubling of distance, the same as for a monopole source.

A transit train is often modeled as either a discrete number of incoherent point sources (monopoles or dipoles) or as a finite-length line source. A line source, which is essentially an infinite number of point sources, radiates cylindrical waves in contrast to the spherical waves of a monopole point source. For an infinite line source, the sound level decreases 3 dB with every doubling of distance from the source. For a finite-length line source, the level decreases 3 dB with each doubling of distance when close to the source, however, at distances large compared to the source length the source looks like a point source, and the sound level decreases 6 dB with each doubling of distance. Figure A.15 compares the geometric attenuation of sound from ideal point sources and a finite-length line source.

Note that Figure A.15 shows the geometric attenuation of sound with distance from the source. This ignores the effects of atmospheric absorption, barriers, reflecting surfaces, ground absorption, and climatic effects. Accurate predictions of sound



a. POINT SOURCES



b. FINITE-LENGTH LINE SOURCE

FIGURE A.15 GEOMETRIC ATTENUATION OF SOUND WITH DISTANCE FROM IDEAL SOURCES propagation over large distances must account for all these factors.

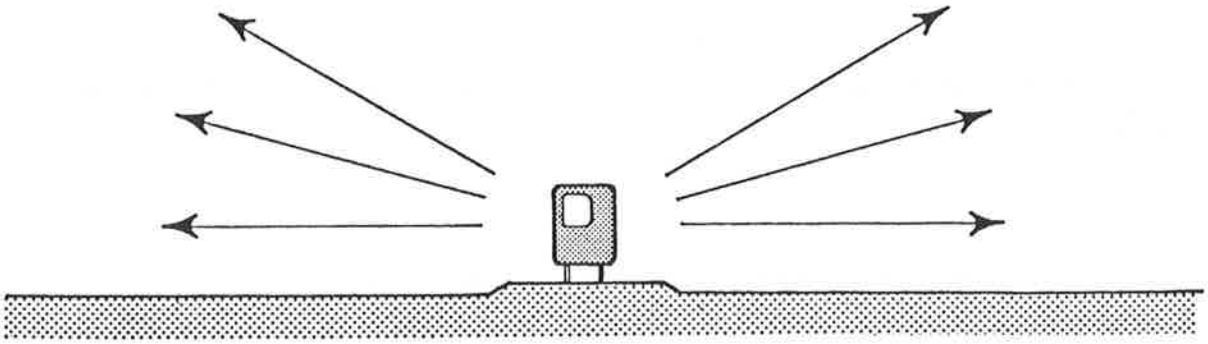
A.5.3 Environmental Effects on Sound Propagation

Unfortunately, when sound propagation outdoors is considered, there are the additional complications of refractions caused by wind or thermal gradients and excess attenuation caused by rail, fog, snow, and atmospheric absorption (Refraction is the bending of a wave at the boundary between media of different densities. A good example is the bending of light waves by a lens). These effects are all in addition to the geometric attenuation discussed in the previous section. The environmental effects on sound propagation are discussed briefly below.

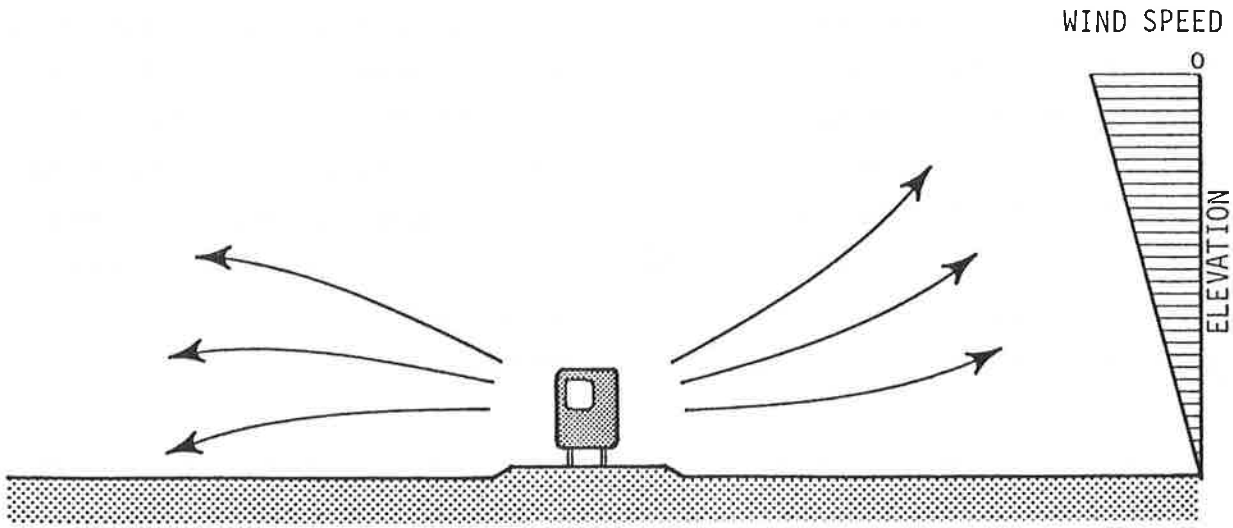
A.5.3.1 Effects of Wind -- Figure A.16b illustrates the effect of wind on sound propagation. Typically, the wind speed increases with elevation above the ground. The result is that when sound is propagating upwind, it is refracted (bent) upwards and when it is propagating downwind, it is bent downwards. The amount of refraction will depend on the rate of wind speed change with altitude.

Propagation of sound in wind can result in upwind shadowing and downwind reinforcement effects that can cause differences from the expected geometric attenuation of sound as large as 10 to 20 dB.

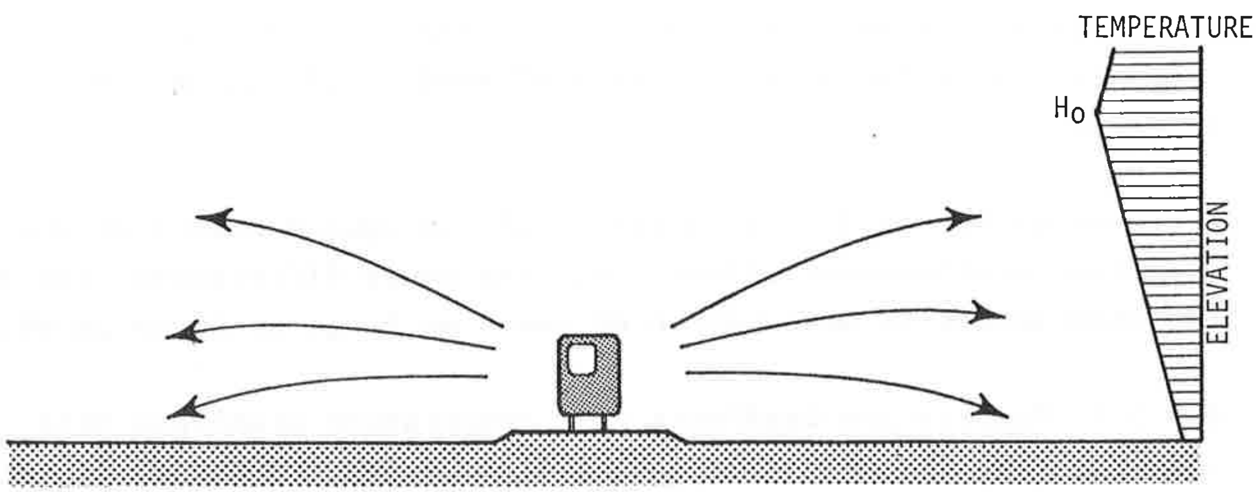
A.5.3.2 Temperature Gradients -- Temperature gradients will cause refractions in a manner similar to wind gradients. Under normal atmospheric conditions on a clear afternoon with air temperature decreasing with increasing altitude, the sound waves will be refracted upwards. However, when temperature inversions occur, there is an increase in air temperature with increasing elevation, and the sound is refracted downwards. Under such conditions strong noise level enhancements can occur. The effect of temperature gradients is shown in Figure A.16c.



a. NORMAL



b. WIND GRADIENT



c. INVERSION LAYER (TEMPERATURE INCREASING WITH HEIGHT UP TO AN ELEVATION OF H_0)

FIGURE A.16 ATMOSPHERIC EFFECTS ON SOUND PROPAGATION

A.5.3.3 Rain, Fog, and Snow -- Although it is commonly said that sound carries well on days of fog or light precipitation, the evidence indicates that this is due to thermal and wind gradients tending to be small during precipitation. Another factor contributing to the apparent ability of sound to carry well during light precipitation is lower levels of background noise at these times due to the normal reduction of outdoor activities.

Finally, although there is no evidence that lightly falling snow has a significant effect on the propagation of sound through air, in some cases snow on the ground will increase the attenuation of sound.

A.5.4 Vibration Propagation

The propagation of vibration through structures, soil, or other media is a very complex phenomenon. Consider a simple structural component such as a beam. Vibration can be transmitted through the beam as: longitudinal waves where, as in acoustic waves in air, the partical motion is parallel to the direction of wave propagation (longitudinal waves are sometimes referred to as compression waves); torsional waves resulting when the beam is excited by a torque; or as bending waves (also referred to as flexual waves). Of all the various wave types, bending waves are by far the most important for sound radiation because they result in large lateral deflections which can efficiently radiate sound waves.

Although most sound radiation results from bending waves, in complex structures compression waves excited in one component will often result in the excitation of bending waves in another component. Hence, even though the compression waves are very inefficient sound radiators, they can often result in extensive sound radiation due to secondary effects.

Figure A.17 illustrates a typical problem of vibration propagation. The forces at the wheel-rail interface result in vibration of the subway structure. The structure vibration excites the surrounding soil creating both compression waves and shear waves. At the surface, the compression and shear waves can combine to create a special class of ground surface waves called Rayleigh waves. The attenuation of the vibration amplitude with distance from the source is strongly dependent on the geologic properties of the soil and strongly frequency dependent. As indicated in the figure, the soil vibration excites vibration of adjacent structures which then propagates throughout the structure in the form of both bending and compression waves. The transverse vibration of room surfaces may be felt as mechanical motion by occupants of the room. It may radiate audible sound, or it may result in secondary sound radiation resulting from vibrating windows, dishes, etc.

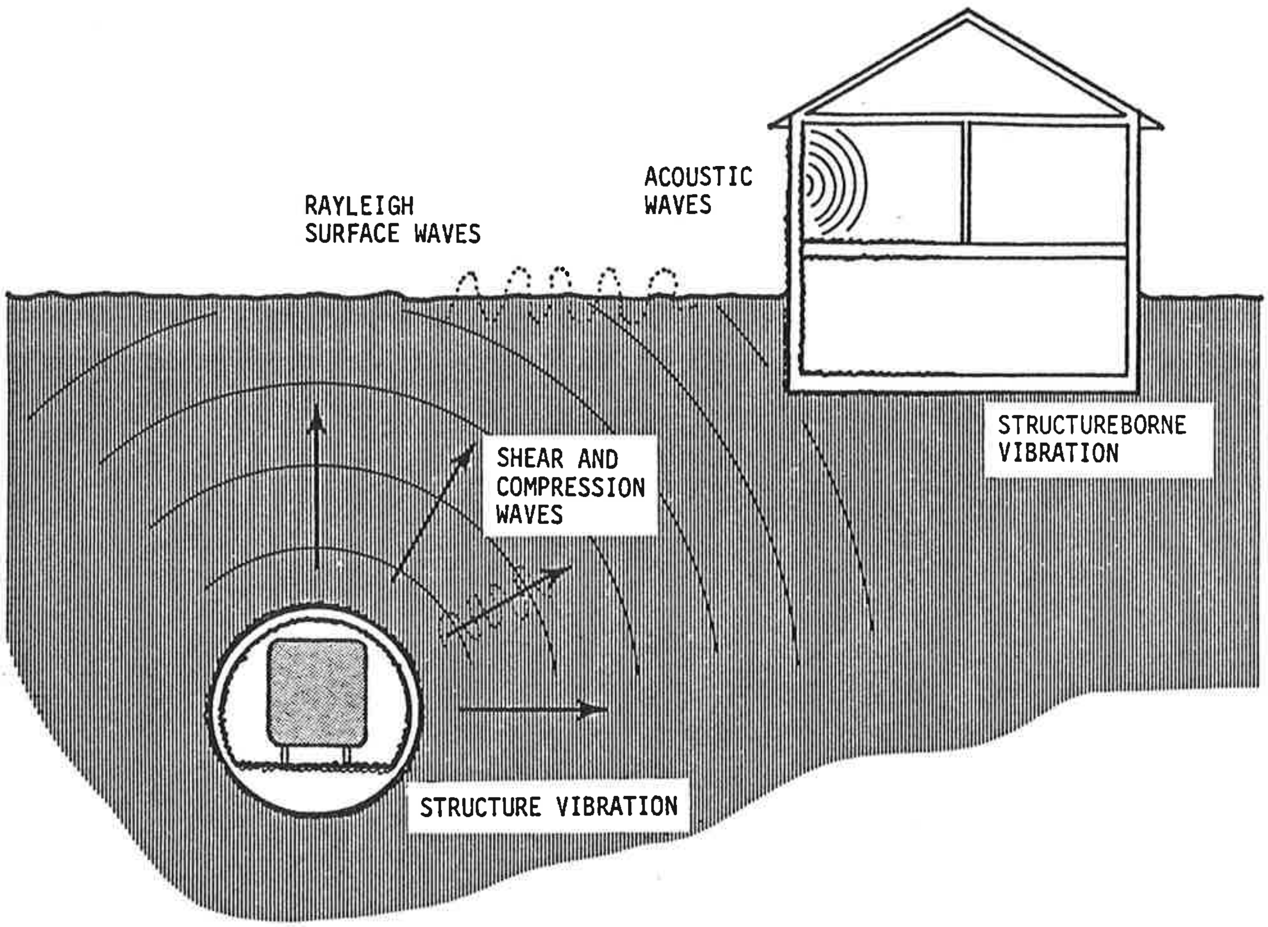


FIGURE A.17 EXAMPLE OF VIBRATION PROPAGATION

APPENDIX B

AMERICAN PUBLIC TRANSIT ASSOCIATION (APTA) NOISE AND VIBRATION GUIDELINES

Presented in this appendix is the noise and vibration section of the "1981 Guidelines for Design of Rail Transit Facilities," published in January 1979 by the Rail Transit Committee of APTA. These guidelines have been used extensively by various transit authorities to develop specifications for new transit vehicles, to develop design goals for community noise and vibration exposure, in the specification and design of stations, and in the specification of ancillary equipment.

2-7 NOISE AND VIBRATION

2-7.1 INTRODUCTION

Contemporary man is making decisions that reflect increasing awareness of his environment and his ability to control that environment. The choise of transportation modes, environmental legislation and public actions reflect this awareness. To gain acceptance by the public the design of the systems must respond to the physical, psychological and physiological demands of the public.

This document is intended to provide recommendations on guidelines and design goals for all noise and vibration control problems relating to rail rapid transit system operations.

The basic goals of these guidelines and the transit industry efforts to control noise and vibration are to:

- Provide system patrons with an acoustically comfortable environment by maintaining noise and vibration levels in vehicles and stations within acceptable limits.

- Reduce any adverse impact of the system operation on the community by minimizing transmission of noise and vibration to adjacent properties.

- Provide noise and vibration control consistent with economic constraints and appropriate technology.

2-7.2 PROBLEMS

The acoustical performance of a rapid transit system will depend on the consideration that has been given by the designer to noise and vibration problems during the design phase.

General problem areas to be considered during design are:

2-7.2.1 IMPROVING ACOUSTICAL ENVIRONMENT FOR PATRONS

This is achieved through control of noise and vibration in transit vehicles, in underground stations, in tunnels, and in above-ground stations. Particular attention should also be given to patron noise exposure caused by traffic at open stations adjacent to highways and streets or other noise sources.

2-7.2.2 REDUCING ADVERSE IMPACT ON THE COMMUNITY

Increased community acceptance requires control of airborne noise and vibration from surface and aerial transit operations, and from transit ancillary areas, such as yard operations, vent and fan shafts of the ventilation system, electrical substations, emergency service buildings and air conditioning chiller plants. The design may also provide control of ground-borne noise and vibration from subway operations.

2-7.3 NOISE AND VIBRATION SOURCES

The designer should identify the various potential sources of noise and vibration within the transit system. Noise generators may be subdivided into four general areas.

2-7.3.1 VEHICLE

Running noise including wheel-on-rail noise, such as rolling, side slippage of the wheel across the railhead, wheel slippage on curves, wheel flange bearing on restraining rail and side of running rail, and collector shoe running on third rail.

Undercar equipment noise including noise radiating from car trucks, wheels, drive gears, traction motors, compressors, cooling blowers, brake shoe bearing on wheel tread, disc brakes, motor alternators and air conditioning condenser fans and refrigerant compressors.

Car interior noise such as ventilating or air conditioning fan noise and noise from other auxiliary equipment.

Noise radiated by rattling components, such as doors, seats, coupling guard rails, air conditioning louvres and windows.

2-7.3.2 TRACKWORK

Particular noise sources are special trackwork, track curves, restraining rail, mechanical joints, insulated joints, poor quality welded joints, and rail irregularities and corrugation.

2-7.3.3 WAY STRUCTURES

Airborne noise and vibration radiation by way structures such as bridge or elevated structure girders and decks, and ground-borne noise and vibration radiated from way structures, particularly subway structures.

2-7.3.4 EQUIPMENT

Noise and vibration radiated by equipment such as ventilating fans, car noise from vent shafts, escalators, vibrating panels, metal ducts, doors and equipment covers, substation transformers, emergency service generators, transit maintenance facilities, air conditioning chiller plant fans, and cooling towers.

2-7.4 EFFECTS OF NOISE AND VIBRATION

An understanding of the effects of noise and vibration is necessary for the establishment of acceptable acoustical design guidelines or criteria. Acousticians have measured and substantiated a broad range of effects. The effects of noise and vibration include:

-PSYCHOLOGICAL EFFECTS, including annoyance, inteference with rest or sleep, interference with work performance and interference with speech communication.

-ECONOMIC EFFECTS, such as decreased property values or interference with commercial activities.

-PHYSIOLOGICAL EFFECTS, including discomfort levels, stress reaction, temporary hearing loss, permanent hearing loss and other general effects on health.

The Reference List includes a number of publications which provide details on the efects of noise and vibration. See particularly References 2, 3 and 4.

2-7.5 GUIDELINES FOR NOISE AND VIBRATION IN VEHICLES AND STRUCTURES

Noise and vibration control methods should be developed to insure compliance with design goals based on some measure of human response to noise. There are two basic types of noise assessments:

- RELATIVE ASSESSMENTS are based on the difference between the noise in question and the prevailing ambient noise. Design goals based on ambient noise levels are often used to set objectives for remedial design in existing systems and are generally applicable in problems relating to the system impact on the community.

  Ambient noise levels in communities have been rising rapidly in recent years. Therefore, design goals based on ambient levels should be reviewed periodically to determine whether or not the noise levels resulting from the derived design goals are appropriate and will result in any significant benefit to the public.

- ABSOLUTE ASSESSMENTS apply to specific design goals related to measurable physical quantities. They usually deal with hearing damage and protection, speech communication or interference with other tasks, and subjective reactions such as annoyance. Such design goals can be used to ensure an acceptable acoustical environment for passengers and employees of the system.

  The use of absolute limit design goals in design problems relating to system impact on the community is the best procedure but should be considered carefully. There is a wide variation in public reaction that can occur even at the same noise levels. Factors which may affect public reaction to noise, in addition to the fact that a noise is new and may be higher than existing ambient levels, include soci-economic status, property ownership, duration and frequency of noise, previous community exposure, nature of the community and its previous success with complaints. However, the primary objective in good transit design should be to reduce annoyance rather than simply reduce complaints.

2-7.5.1 DESIGN GOAL PURPOSE

The purpose is to establish good practice guidelines in modern rapid transit design for those designing and planning new facilities. Achievement of all the noise and vibration limits in the guidelines may not be possible with existing transit equipment. Also, the guidelines are not intended for, nor are they to be confused with, noise abatement controls of the type already enacted or propsed in various communities and jurisdictions. Where there are other applicable ordinances or criteria which are in conflict or more restrictive, these should be substituted for the guideline design goals.

Specifications for transit cars should include both definitions of the desirable limits for noise and vibration and the procedures and techniques to be used in testing for conformance with the specifications. The tests should include determination of acoustical performance of car components, equipment, and complete car assembly.

Studies have been made to evaluate the various physical measurement scales to determine those scales which most closely correlate with subjective evaluations of noise. For most typical noise sources such as street traffic, transit vehicles, and general community noise, it has been found that the sound level meter "A" weighting scale gives good and adequate correlation with subjective evaluation of response to noises. Thus, the "A" weighted sound level, which can be read directly from a sound level meter, is best for evaluating, on an engineering basis, the probable response of people to the noise created by transit systems operations.

To the human ear a random noise with audible pure tone is more annoying than the simple random noise even if both noises have the same A-weighted sound level. The transit car specifications should include controls over pure tone noise.

2-7.5.2 MEASUREMENT PROCEDURE AND ASSUMPTIONS

Noise limits indicated assume the following: Car interior noise guidelines apply to measurements taken in a complete but empty car and made 4.0 ft (1.2 m) above the car floor along the centerline of the car. Exterior noise guidelines are based on measurements taken in essentially a free-field environment away from reflective or shielding surfaces.

Noise or sound levels are measured with a sound level meter which meets the Type 2 requirements of American National Standard (ANSI) S1.4-1971, Specification for Sound Level Meters.

All noise levels are A-weighted sound levels, per (ANSI) S1.4-1971, in decibels referenced to 20uPa (0.0002 microbar).

The "slow" meter response is assumed for all noise level measurements except those involving measurements of moving and transient sources such as exterior train noise, train noise from vent shafts, and from car door operation. The latter sources should be measured using "fast" meter response.

Table 2-7-A, Summary of Transit Vehicle and Structure Noise and Vibration Design Goals, is a tabulation of design goals discussed in the subsequent sections of this guideline.

TABLE 2-7-A. SUMMARY OF TRANSIT VEHICLE AND STRUCTURE NOISE AND VIBRATION DESIGN GOALS

Section	Item	Goals
2-7.5.3	Vehicle Interior Noise Levels (Empty Car)	
	In open (ballast and tie) at maximum speed on welded rail (+5 dBA on jointed rail)	70 dBA
	In open (concrete trackbed) at maximum speed	74 dBA
	In tunnels at maximum speed	80 dBA
	All auxiliaries operating, car stationary	68 dBA
	One auxiliary system operation, car stationary	65 dBA
	Door operation (fast meter response)	72 dBA
2-7.5.4	Vehicle Exterior Noise Levels (50 ft (15 m) from track centerline in open with no reflecting surfaces within 100 ft (30 m) of test location)	
	Car stationary, all auxiliaries operating	60dBA

TABLE 2-7-A (continued)

Section	Item	2-car	4-car	Goals 6- or 8-car
	Ballast and Tie Track (fast meter response):			
	train operating at 80 mph	84 dBA	86 dBA	87 dBA
	train operating at 60 mph	80	82	83
	Concrete Trackbed (fast meter response):			
	train operating at 80 mph	88	90	91
	train operating at 60 mph	85	87	88
2-7.5.5	Vehicle Equipment Noise Levels (15 ft (4.5m) from car centerline)			
	Vehicles with self-ventilated motors:			
	Propulsion system at equivalent to 80 mph			90 dBA
	Propulsion system at equivalent to 60 mph			84 dBA
	Vehicles with ducted ventilation to propulsion motors:			
	Propulsion system at equivalent to 80 mph			84 dBA
	Propulsion system at equivalent to 60 mph			78 dBA
	All Vehicles:			
	Full service brake operation			75 dBA
	Car stationary, auxiliary equipment, operating			68 dBA
	Decrease in allowable levels for presence of pure tones			3 dBA

TABLE 2-7-A (continued)

Section	Item	Goals
2-7.5.6	Vehicle Interior Vibration Levels due to Auxiliary Equipment (with car stationary)	
	Measurements taken on car interior surfaces which passengers normally contact, such as floor, seat frames station, etc., except as noted:	
	Maximum displacement, 0 Hz to 1.4 Hz	0.10 in. peak-to peak
	Maximum acceleration, 1.4 to 20 Hz	0.01 g peak
	Maximum velocity, above 20 Hz	0.03 in/sec peak
	Maximum displacement on detached traction motors	0.0015 in. peak-to-peak

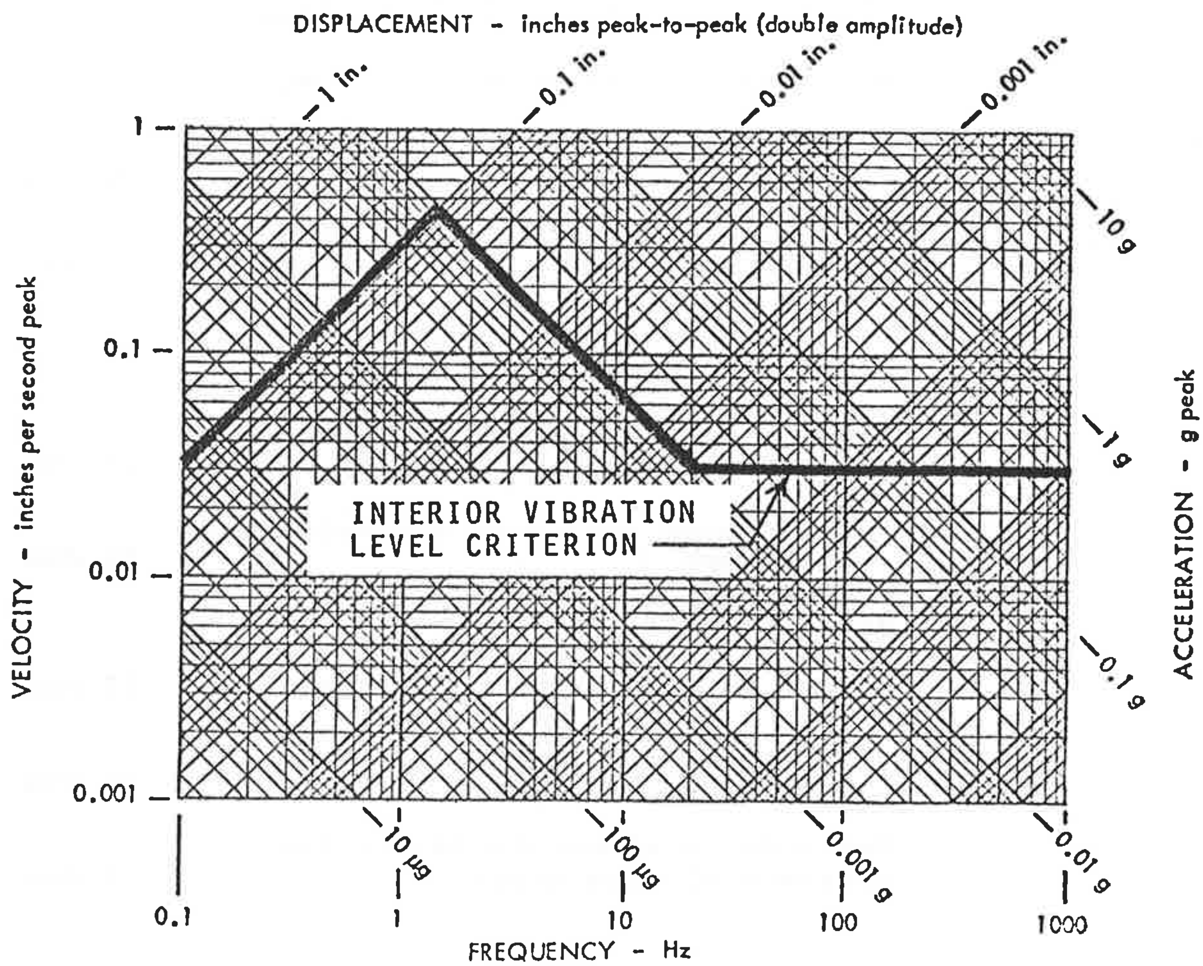


FIGURE 7-1 LIMITS FOR MAXIMUM CAR INTERIOR VIBRATION LEVELS DUE TO AUXILIARY EQUIPMENT

TABLE 2-7-A (continued)

Section	Item	Goals
2-7.5.7	Underground Stations	
	Platform noise level, trains entering and leaving	80-85 dBA
	Platform noise level, trains passing through	85 dBA
	Platform noise levels, trains stationary	68 dBA
	Platform area reverberation time	1.2 to 1.4 sec.
	(for large cross-section multi-track platform areas)	(1.4 to 1.6 sec)
	Platform noise level, only station ventilating system and escalators operating	55 dBA
	Noise level in station attendants' booths	50 dBA
2-7.5.8	Noise in Above-Ground Stations (at-grade or elevated)	
	Platform level, trains entering and leaving:	
	ballast and tie track	75-80 dBA
	concrete trackbed	80-85 dBA
2-7.5.9	Noise in Subway Tunnels	
	Minimum useful design reduction in reverberant noise levels with acoustic treatment	7 dBA
2-7.5.10	Shop Equipment Noise	
	Maximum noise level at operator's position	85 dBA
	Maximum noise level at any point 3 ft (1 m) from equipment	90 dBA

2-7.5.3 VEHICLE INTERIOR NOISE LEVELS

Surface Operations. For ease of communication and passenger comfort the sound level should not exceed about 70 dBA. Under normal operating conditions, at maximum speeds, on ballast and tie track in the open, this noise level represents a realistically attainable criterion and a datum upon which other noise design goals can be based. In all vehicles for public conveyance, it is desirable to maintain a background sound level which will afford some degree of speech privacy for passengers. Efforts to significantly reduce interior sound levels below this criterion would be undesirable. Standardizing noise measurements, both interior and exterior, with cars operating on ballast and tie track in the open, reduces the number of variables which arise when the trains are operated through or on the many varieties of structure used in mass transit systems today.

Interior noise levels when operating on concrete trackbed at-grade or on aerial structure should not exceed 74 dBA at maximum speeds.

Underground Operations. In underground structures a design goal of 80 dBA maximum should be established. This can be accomplished either by restrictive car body sound insulation requirements or by use of acoustical absorption treatment in tunnels or a combination of these two provisions. Difficulties with selection of absorption materials and durability of installations of tunnel absorption treatments have been encountered in some instances, indicating that providing the noise reduction via car body insulation may be preferable although this involves a potentially heavier and more costly vehicle. In other cases there have been successful and durable tunnel sound absorption installations -- a design arrangement which successfully allows for less stringent requirements on the car body.

Extra sound insulation of the car walls, doors, windows and roof are not needed for low interior noise during operation on surface tracks. Only the floor insulation is of primary importance for surface operation interior noise control. Therefore, the designer should consider the relative lengths of subway and surface tracks as one review parameter influencing the design choice of car body sound insulation or tunnel sound absorption.

Auxiliary Equipment. A consistent goal for background noise created by the car auxiliary equipment and the air conditioning and ventilating system when the car is stationary is 68 dBA. Similarly, with any one of the auxiliary systems operating, such as the air refrigeration or heating and distribution system, the motor alternator, the air compressor or equivalent, the motor control system, or the traction power motor cooling blowers or cooling system, the car interior noise level should not exceed 65 dBA.

Noise produced by operation of only the vehicle doors should not exceed 72 dBA measured 1 ft (.3m) or more from the door using the fast meter response.

2-7.5.4 VEHICLE EXTERIOR NOISE LEVELS

Ballast and Tie Track. With the vehicle stopped and all systems operating simultaneously under normal conditions, the noise level measured 50 ft (15 m) horizontally from the track centerline 4.0 ft (1.2 m) above grade or at the axle centerline elevation, whichever is higher, should not exceed 60 dBA at any point along the length of the car on either side.

When a 2-car train is moving at 80 mph (130 km/hr) on tangent, at-grade, ballast and tie track, with smooth ground rails and with all vehicle systems operating simultaneously, the noise level should not exceed 84 dBA measured 50 ft (15 m) from the car centerline as above. Similarly, at 60 mph (100 km/hr) noise levels should not exceed 80 dBA.

Aerial Structure and Concrete Trackbed. Aerial structures should be designed using a configuration which does not add significant structure radiated noise to the noise radiated by the vehicles. Because of the sound reflection from concrete decks the wayside noise from aerial structures -- or concrete trackbed on grade -- is higher than for ballast and tie but should not exceed 88 dBA at 80 mph (130 km/hr) and 85 dBA at 60 mph (100 km/hr) at 50 ft (15 m) from track centerline for 2-car trains. Aerial structure designs which result in generation of low frequency rumble due to vibration of girder plates or deck slabs should be avoided. Even though the low frequency noise may not significantly affect the outdoor A-weighted sound level, it can transmit easily into nearby buildings and create intrusion.

Wheel Squeal Noise. Short radius curves can result in intense wheel squeal noise produced by the wheel/rail interaction. Avoiding curves of radius less than 750 ft (330 m) will minimize squeal. Where there must be shorter radius curves or for existing systems with short radius curves, wheel damping is effective in reducing or eliminating wheel squeal. Simple ring dampers -- a steel ring in a groove in the wheel -- have been demonstrated to be very effective in reducing wheel squeal noise. Also, resilient wheels -- wheels with an elastomer between the hub and tire -- have been demonstrated to be effective in reducing or eliminating wheel squeal noise and can be used on vehicles not equipped with tread brakes.

2-7.5.5 VEHICLE EQUIPMENT NOISE LEVELS

When traveling at high speed the principal noise sources are the propulsion system, motors and gearing, and the wheel/rail system. At medium speeds the wheel/rail noise usually predominates while at higher speeds propulsion system noise predominates. Application of mechanical service brakes can also result in predominant noise at low speeds. Therefore, care must be exercised in the design and maintenance of the wheel/rail system, the propulsion system, and the braking system.

Experience has shown that the propulsion motors are the main noise source at high speeds and that the main component of the motor noise is the cooling fan noise for self-ventilated motors. Because motors with ducted ventilation are quieter, consideration should be given to the use of forced air-ducted cooling for the propulsion motors.

Propulsion Equipment. With self-ventilated motors the propulsion system noise level should be 90 dBA maximum with propulsion motors and wheels operating at rpm equivalent to 80 mph (130 km/hr). Similarly, 84 dBA should be the limit for operation at 60 mph (100 km/hr). With ducted ventilation the design guideline can be 84 dBA at 80 mph (130 km/hr) and 78 dBA at rpm equivalent to 60 mph (100 km/hr). These noise levels should be checked at 15 ft (4.5 m) from the car centerline by placing the car on jacks and allowing free-wheeling.

Measurements are to be taken at the level of the truck axles. To avoid excessive pure tone noise components, and because the gearing noise is predominantly pure tone in character, a separate requirement for the gearing should be included. The noise from a gearbox only under load should be 3 to 4 dBA less than the total noise from the complete propulsion system operating under free-wheeling conditions.

Service Brakes. Application of service brakes should not result in substantial increase in noise levels experienced on platforms when compared to the total noise from the auxiliary equipment, the propulsion system, and the wheel/rail interaction. To accomplish this goal, noise from the full application of service brakes at low speeds, 0 to 15 mph (0 to 25 km/hr), should be limited to 75 dBA at 15 ft (4.5 m).

Auxiliary Equipment. A limit of 68 dBA, at 15 ft (4.5 m) from the car centerline, should be established as appropriate for noise levels from auxiliary equipment when the car is stationary. The limit specification includes air brake noises such as the rapid release or "dumping" of air at terminals.

Effect of "Pure Tones". The equipment noise level should be reduced 3 dBA if significant pure tones in the range from 300 Hz to 4000 Hz are present. Pure tones are significant if any 1/3 octave band sound pressure level is 5 dB, or more, higher than the average of the two adjacent 1/3 octaves containing no pure tones.

2-7.5.6 VEHICLE INTERIOR VIBRATION LEVELS

The primary purpose for limiting vibration levels caused by equipment and auxiliaries mounted anywhere on the car is to prevent the car floor, seats, stanchions, etc., from vibrating at levels which would be annoying to passengers. In general, the response of people is approximately proportional to vibration ACCELERATION at frequencies below about 20 Hz and to vibration

VELOCITY for frequencies above 20 Hz. Human sensitivity to horizontal vibrations is 30% to 60% of sensitivity to vertical vibrations above about 1.0 Hz.

It is appropriate to limit the maximum displacement, at very low frequencies in particular, to prevent visual perception of vibration. The maximum displacement should, therefore, be limited to 0.10 inch, peak-to-peak, at any frequency up to 1.4 Hz. Similarly, vibration criteria should limit car vibrations to within the "barely perceptible" to "distinctly perceptible" ranges. Up to a frequency of 20 Hz vibration acceleration should be limited to 0.01 g peak (3.9 in./sec/sec) and above 20 Hz velocity should not exceed 0.03 in./sec peak. These limits pertain to measurements of vibration from car equipment made anywhere on the car floor, seat frames, stanchions, and other points normally contacted by passengers.

The vibration of any traction motor, detached and supported on resilient mountings which have at least 0.25 inch static deflection with equivalent horizontal and vertical stiffness, should not exceed a displacement of 0.0015 inch peak-to-peak measured anywhere on the motor when running at any speed between 50% and 100% of maximum operation speed.

2-7.5.7 UNDERGROUND STATIONS

Train Noise. Trains operating at top speeds of 80 mph (130 km/hr) and using maximum acceleration and braking could enter or leave stations at about 50 mph (80 km/hr) depending on platform length, approaching and leaving grades, station spacing, and other factors. Noise levels should be limited to maximums of 80 to 85 dBA by use of appropriate rail fixation and station absorption treatment. For express trains operating through stations, noise levels should be limited to 85 dBA. Resilient track fixation and absorption materials to control noise should be applied and for adequate noise reduction, coverage of the underplatform overhang surfaces and about 30% coverage of walls and ceilings will likely be necessary, depending on the size and shape of the platform space.

Stationary car noise should be limited to 68 dBA at 15 ft (4.5 m) from the train in the open on ballast and tie track and, to correspond with this, in stations the noise level should be limited to 68 dBA maximum anywhere on the platform with the train stationary.

Station Acoustics. For noise control and intelligibility of public address system announcements the reverberation time of station public areas should have a low value. The optimum acoustical/economic choice depends somewhat on the size and volume of the station space. For typical two-track stations a value of 1.2 to 1.4 seconds at mid-frequencies is appropriate. For very large or high ceiling spaces and for multi-track stations a value of 1.4 to 1.6 seconds is satisfactory. Such design guidelines are sufficient and appropriate to control noise, reduce speech interference and should provide for good intelligibility of public address system announcements and patron voice communication.

Ancillary Equipment Noise. Station ventilation system noise is probably the simplest to control by selection of fan locations and acoustical design of fan installations and ducts. Escalators and elevator equipment also produce noise which may affect station public space environment. Since the ventilation system and other mechanical equipment noise may be regarded as steady-state during lengthy periods of operation the design guideline noise level limit for station platforms should be 55 dBA except under emergency conditions. Escalator noise should be limited to 55 dBA maximum at the entrance combs. Ventilation system noise in station attendants' booths should be limited to 50 dBA.

2-7.5.8 NOISE IN ABOVE-GROUND STATIONS

In above-ground stations noise levels will be governed by train operations. A maximum noise level limit of 75 to 80 dBA as measured on the train platforms should be established for ballast and tie tracks. For concrete trackbed 80 to 85 dBA is the appropriate limit. Station location is a potential problem, particularly when train platforms are located in a highway median, adjacent to a street with high speed, high flow rate traffic, or adjacent to railroad right-of-way. An appropriate acoustical design with shielding can relieve patrons on platforms from an otherwise serious noise problem created by traffic or other noise sources. Design goals for maximum noise levels should be similar to those for the transit trains.

In some instances, where open station platforms are in close proximity to busy airports or railroads, it may be impractical and uneconomical to apply the above design goals to noise from aircraft or railroad operations. Some measure of protection should be afforded the passenger such as appropriately designed acoustical shelters to minimize directional noise. The upper noise level limit for train operations in underground stations of 85 dBA is suggested for uniformity.

2-7.5.9 NOISE IN SUBWAY TUNNELS

Appropriate noise abatement techniques should be used to reduce extreme noise levels from high speed train operation in tunnels to an acceptable level. A limit of 80 dBA for interior car noise at maximum tunnel operating speeds is recommended. An acoustical absorption system may be provided in the tunnels or additional sound insulation may be provided on the cars to meet this guideline. Tunnel sound absorption treatments can, for instance, provide 5 dB or more reduction of noise levels inside the car. Reducing tunnel noise by a sound absorption system improves the acoustical environment for system employees and aids in complying with the hearing conversation requirements of the Occupational Safety and Health Act.

2-7.5.10 SHOP EQUIPMENT NOISE

To avoid excessive noise exposure for employees and to comply with existing and proposed standards and requirements of the Occupational Safety and Health Act, shop equipment noise should not exceed 85 dBA at operator stations and should not exceed 90 dBA at any point 3 ft (1 m) from the equipment. Appropriate noise level limits should be included in all shop exposure limits in shop areas.

2-7.6 GUIDELINES FOR NOISE AND VIBRATION IN TRANSIT CORRIDOR COMMUNITIES

The purpose of this subsection is to consider the effect of noise and vibration on the community because of its importance in influencing public acceptance of the system and because of possible applicability of community noise ordinances or standards. Sources of wayside intrusion or annoyance due to noise and vibration created by a rail transit facility include:

- Airborne noise from surface and aerial train operations.
- Airborne noise from transit yard operations and maintenance facilities.
- Airborne noise from ventilating fans.
- Airborne noise from trains in subways, transmitted through ventilation shafts.
- Airborne noise from ancillary systems such as traction power substations, air conditioning chiller plants, cooling towers, and emergency service buildings.
- Ground-borne noise and vibration from subway operations.
- Ground-borne vibration from surface and aerial structure operations.

In defining community noise levels and corresponding appropriate design guidelines, urban or suburban areas may conveniently be considered in five general categories according to land usage and typical ambient noise levels as given by Table 2-7-B.

2-7.6.1 AIRBORNE NOISE FROM ABOVE-GROUND TRAIN OPERATIONS

Transit Train Noise Descriptors. Defining the acceptability of transient noises from surface and aerial transit operations in residential areas is difficult. Whereas overall transit noise levels are comparable to some existing community noises, such as street and highway traffic, transit operations can represent a new noise nuisance in the community. Therefore, acceptability guidelines should be carefully selected.

TABLE 2-7-B. GENERAL CATEGORIES OF COMMUNITIES ALONG TRANSIT SYSTEM CORRIDORS

Area Category	Area Description	Typical (Average or L_{50}*) Ambient Noise Level
I	Low density urban residential, open space park, suburban	40-50 dBA - day 35-45 dBA - night
II	Average urban residential, quiet apartment and hotels, open space, suburban residential, or occupied outdoor area near busy streets.	45-55 dBA - day 40-50 dBA - night
III	High density urban residential, average semi-residential/ commercial areas, parks, museum and non-commercial public building areas.	50-60 dBA - day 45-55 dBA - night
IV	Commercial areas with office buildings, retail stores, etc., primarily daytime occupancy. Central business district.	60-70 dBA
V	Industrial areas or freeway and highway corridors.	Over 60 dBA

*L_{50} is the median noise level.

Noise level guidelines for train operations should be derived considering the five general categories of community areas, defined in Table 2-7-B, and should also include consideration of the type of building. Single event maximum noise level design goal guidelines for airborne noise from trains in each of the area categories and for several types of buildings or occupancies are given in Table 2-7-C.

A number of "noise exposure level" evaluation schemes have been devised to provide a basis for determining noise level design goals and noise acceptability (3,4). Such evaluation procedures depend on several variables including maximum single event transient noise levels, number of events per hour of day or time of day. Since such factors are not necessarily available at the time of design and because the exposure level measures do not generally address maximum permissible single event noise levels, the use of a single event maximum level is more appropriate for transit design. Also, train noise levels, because of their short duration, may appear acceptable on a calculated exposure level basis but, because of the possible large differences between maximum passby levels and average community ambient noise, the train noise may be unacceptable because of its magnitude. Therefore, single event maximum noise levels should be used for the transit system facility design.

TABLE 2-7-C. GUIDELINES FOR MAXIMUM AIRBORNE NOISE FROM TRAIN OPERATIONS

Community Area Category	Single Event Maximum Noise Level Design Goal		
	Single Family Dwellings	Multi-Family Dwellings	Commercial Buildings
I Low Density Residential	70 dBA	75 dBA	80 dBA
II Average Residential	75	75	80
III High Density Residential	75	80	85
IV Commercial	80	80	85
V Industrial/Highway	80	85	85

These design goal guidelines are applied to nighttime operations because the sensitivity to noise is greater at night than during daytime hours. These guidelines should be applied outdoors and references to the building or area under consideration but not closer than 50 ft from track centerline. Because of the transient nature of train noise, community acceptance should be expected if the noise levels do not exceed these guidelines at night at the affected buildings or use areas.

For some types of buildings or occupancies maximum noise level limits should be applied regardless of the community area category and the designer should be particularly careful in locating surface or aerial transit lines adjacent to auditoriums, TV studios, schools, theatres, amphitheatres and churches. Table 2-7-D lists guidelines for maximum airborne noise from train operations in these areas.

TABLE 2-7-D. GUIDELINES FOR MAXIMUM AIRBORNE NOISE FROM TRAIN OPERATIONS

Building or Occupancy Type	Single Event Maximum Noise Level Design Goal
Amphitheatres	60 dBA
"Quiet" Outdoor Recreation Areas	65 dBA
Concert Halls, Radio and TV Studies, Auditoriums	70 dBA
Churches, Theatres, Schools, Hospitals, Museums, Libraries	75 dBA

2-7.6.2 GROUND-BORNE NOISE AND VIBRATION FROM TRAIN OPERATIONS

Ground-Borne Noise Characteristics. With modern transit cars and tracks, the ground-borne vibration levels are below the threshold of perception in most instances. However, the vibration levels produced at the wheel/rail interface are still sufficient to cause ground-borne noise -- a low frequency rumbling noise -- which can signal the passage of a train. This noise, which is radiated and heard only inside buildings near the transit line, is frequently of sufficient loudness to create a significant intrusion or annoyance.

By utilizing recent experience in track and vehicle design, vibration levels from normal transit operations can usually be reduced sufficiently to prevent significant intrusion of noise in buildings at distances of 100 to 200 ft (30 to 60 m) or more, depending on intervening geological strata and on building construction type and quality, occupancy, type of way structure, train speed, etc. However, where there is special trackwork, such as turnouts and crossovers, and at points where buildings are less than 100 to 200 ft (30 to 60 m) from a subway structure, there maybe need of additional vibration reduction measures within the subway such as floating slab trackbed construction. There are also some special cases where buildings, such as hospitals and medical laboratories next to tunnels contain equipment sensitive to mechanical vibration. These cases will require further investigation and the possible use of further vibration reduction measures such as isolation within the building itself.

The principal noise sources in modern buildings are the air conditioning and ventilating systems and background noises transmitted into the building from street and highway traffic. Noise and vibration from these sources will often exceed those generated by transit operations. However, the guidelines given in Table 2-7-C still apply to transit train noise outside the buildings.

The most critical locations where ground-borne noise could create intrusion are private residences, sleeping rooms in general, concert halls, auditoriums, and TV or recording studios. Since residences and sleeping rooms found in various classes of building are the most common, interior noise design guidelines relating to these areas should be considered using the same area categories and occupancies as for airborne noise from transit trains as given in Table 2-7-B.

Guidelines for Ground-Borne Noise. Table 2-7-E indicates the appropriate single event noise limit design goal guidelines for ground-borne noise for different building types in different area categories. Since the background or ambient noise in sleeping spaces is generally different in the different areas and building types, the allowable noise level can be greater in the noisier areas. Table 2-7-E indicates levels which should be acceptable if not exceeded. It would be unreasonable in most cases to design

TABLE 2-7-E.
GUIDELINES FOR MAXIMUM GROUND-BORNE NOISE FROM TRAIN OPERATIONS

Community Area Category	Maximum Single Event Ground-borne Noise Level Design Goal: 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

As with airborne noise, there are some types of buildings for which specific design goals can or should be applied, regardless of area category. Table 2-7-F presents design goals for generally acceptable levels of transient ground-borne noise levels in occupied spaces of various types of buildings and occupancies. This table is not intended to be all inclusive but may be a convenient general guide to the designer.

TABLE 2-7-F.
DESIGN GOAL FOR MAXIMUM GROUND-BORNE NOISE FROM TRAIN OPERATIONS

Type of Building or Room	Single Event Maximum Ground-borne Passby Noise Level Design Goal
Concert Halls and TV Studies	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	35-45 dBA
Commercial Buildings	45-55 dBA

for a noise level that is undetectable. Ground-borne noise which meets the design goals listed above will not be inaudible in all cases, however, the level will be sufficiently low that no significant intrusion or annoyance should occur. In most cases, there will be noise from street traffic, other occupants of a building, or other sources, which will create intrusion that is equivalent or greater in level than the noise from transit trains passing by.

A range for the maximum ground-borne noise limit is given in some cases to permit the designer to adjust the design goal to be suitable for the environment and location of the building. For example, at offices in a quiet, landscaped industrial park area the limit should be at the low end of the range, 35 dBA, whereas for offices located at a busy intersection or in a noisy central business district the limit can be at the upper end of the range, 45 dBA.

Ground-Borne Vibration. Experience with modern transit vehicles has indicated that truck design can influence the amplitude of low frequency ground-borne vibration, sometimes affecting the perceptibility or feelability of the vibration and sometimes affecting the audibility of the ground-borne noise. To minimize low frequency ground-borne vibration and noise, unsprung weight should be minimized and truck frame designs with frame on axle/wheel assembly resonance frequency in the range of 12 to 40 Hz should be avoided. In this frequency range there is the possibility of coincidence of the truck resonance frequency with the track support structure resonance frequency, with the floating slab resonance frequency, with nearby building resonance frequencies, and with the frequency range where maximum transmission of ground-borne vibration occurs. If these frequencies can be determined from design parameters and tests, then the specific truck resonance frequency range to be avoided can be determined.

As an alternative for existing systems with transit cars known to give satisfactory ground-borne vibration levels, new car specifications should include comparative ground-borne vibration tests with existing cars, requiring that the new cars not produce any greater amplitudes of low frequency range ground-borne vibration than the older cars under identical test conditions.

2-7.6.3 AIRBORNE NOISE FROM TRANSIT ANCILLARY FACILITIES

Ancillary Facility Noise Characteristics. In the case of noise from transit ancillary facilities it is appropriate to design for noise levels that are non-intrusive but which may be noticeable on occasion. The appropriate audible noise level design goal limit depends on the activities of occupants as well as background noise in the area. It is found that persons occupied with various tasks or recreational activities are not aware of short duration noise until its level is about 10 dB greater than the ambient. Conversely, it is possible for persons sitting and listening to detect an intruding transient sound when it is about 5 dB less than the background noise, particularly where pure tones are present.

There are two basic types of airborne noise from ancillary facilities: transient and steady-state. Transient noise occurs during passages of trains, for example. Steady-state noise may be characterized by fan noises or noises from electrical substations and chiller plants. The acceptable levels of the two types of noises are different. Transient noises are acceptable at higher levels than continuous or steady-state noises, particularly steady-state noises containing pure tones.

Guidelines for Ancillary Facility Noise. In defining noise levels from ancillary systems, the five general categories of urban or suburban area defined in Table 2-7-B are used. Table 2-7-G suggests noise levels in each of these categories which, if not exceeded by the transit system facility noises, should result in general community acceptance.

TABLE 2-7-G

GUIDELINES FOR NOISE FROM TRANSIT SYSTEM ANCILLARY FACILITIES

Community Area Category	Maximum Noise Level Design Goal: Transient Noises	Continuous Noises
I Low Density Residential	50 dBA	40 dBA
II Average Residential	55	45
III High Density Residential	60	50
IV Commercial	65	55
V Industry/Highway	75	65

The design goals in Table 2-7-G should be applied at 50 ft (15 m) from the shaft outlet or other ancillary facility or should be applied at the setback line of the nearest buildings or occupied area, whichever is the shorter distance. Transient noise design goals apply to short time duration events such as train passby noise transmitted from vent shaft openings.

Continuous noise design goals apply to noises such as fans, cooling towers or other long duration noises except electrical transformer hum. The design goals for transformer noise or hum should be 5 dBA less than given in the Table 2-7-G.

APPENDIX C

SAMPLE SPECIFICATIONS

Presented in this Appendix are three sample specifications that have been used by rail transit systems to ensure that new equipment and facilities will have appropriate acoustic properties. These specifications have been chosen as representative of good, thorough specifications. It should be possible to adopt these sample specifications to other transit systems, however, they should be carefully reviewed to ensure that all of the sections are appropriate for the specific application.

The three specifications are in or near the final format used by the transit systems and are presented here with the permission of the transit systems. The three specifications are:

C.1 Chicago Transit Authority Specification No. CTA 4000-78 for Three Hundred Rail Transit Cars (CTA 2400 Series Cars) Excerpts from Section 3 - Car Body and Section 15 - Tests and Adjustments, Noise and Vibration Criteria and Shipment of Cars
Pages 3-12, and 15-7 through 15-17

SECTION 3 - CAR BODY (Continued)

3.08 - Insulation (Continued)

C. Floor, Thermal

The floor shall be insulated with a layer of permanently fire-retardant insulation, having insulating properties equal to at least 1 inch of fiber-glass, with a maximum K factor of 0.24. The insulation shall be installed between the bottom of the floor panels and the stainless steel skin.

D. Acoustic

The entire car, sides, roof and floor shall be acoustically insulated, in addition to thermally insulated, to achieve interior noise levels as specified in Section 15.02. The materials used shall be suitable for the function required.

E. Fiberglass

Fiberglass insulation used in areas subjected to water in normal car service (under the floor and below the window sill level) requires precautions re-viewed by the Engineer to protect it from moisture and dirt.

F. Miscellaneous

1. Insulation used in any air stream shall have a smooth surface to prevent the build up of dust and dirt and shall be properly retained to prevent it from blocking the air flow.

2. Closed-cell insulation or sealed fiberglass shall be used wherever the insulation may be subjected to water in normal car service.

3. All insulation materials used must be reviewed by the Engineer.

G. Flammability

All insulation shall be tested in accordance with ASTM E162 and shall exhibit a flame spread index of 25 or less with no rapid running or dripping of flam-ing material.

3.09 - Ceiling and Side Lining

A. General

The general interior arrangement of the car shall conform to the CTA Print Sked-44, Page DR-1, and shall require the review by the Engineer.

The interior design shall provide for easy maintenance and cleaning by avoiding pockets and corners wherever possible.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.01 - Tests and Adjustments (Continued)

Q. Prototype Car Tests (Continued)

design or construction deficiency. If in the judgment of the Engineer, a design or construction deficiency is found, the prototype cars shall be modified, and the test period shall begin over again. When the four (4) cars have completed the 600-hour test period successfully, the Authority shall authorize the Contractor to start delivery of the remaining cars. Successful completion of this 600-hour test period shall not relieve the Contractor of responsibilities for compliance with all requirements of the specification.

15.02 - Noise and Vibration Criteria

A. General

The Contractor shall devote particular attention to the design of the car, equipment and trucks to obtain quiet operation and shall ensure that the noise and vibration criteria specified herein are not exceeded. Particular attention shall be given to the designing of all equipment to ensure minimum generation of noise and vibration, and to the attenuation of airborne and solid-borne noise and vibration along the path from source to passenger. Vibration isolators, enclosures or baffles, accoustical absorption, car body panels with adequate sound transmission loss or other appropriate methods shall be incorporated into the car design to adequately attenuate noise and vibration generated by wheels and rails and all car elements and equipment to ensure that the limitations on interior and wayside noise and vibration are not exceeded. Maximum permissible noise levels from some individual pieces of equipment are specified and maximum permissible interior and wayside noise and vibration levels are specified. Noise levels from equipment not specifically limited herein shall be controlled by the Builder to insure that the interior and wayside noise and vibration limits for the complete car are satisfied.

The Contractor shall perform noise and vibration tests on one complete "A" Car and one complete "B" Car to demonstrate compliance with all specifications stated herein. All test procedures, data and results shall be submitted to the Engineer for his review. The cars used for these tests shall be two of the prototype cars.

All equipment shall be designed to eliminate rattling and resonance at all speeds up to 10% above maximum normal running speed by the use of damping, gaskets, resilient mounts or similiar methods. Included in this requirement but not limiting the generality thereof are such accessories as:

- windows
- seats
- wiring
- piping
- ventilating ducts
- ventilating grilles
- wall panels
- lighting fixtures and covers
- stanchions
- barriers
- grab handles
- fire extinguishers
- doors

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

B. Noise Design Considerations

During the design and development of the car the Contractor shall make tests and calculations as required to substantiate that the noise levels of the completed car will meet these specifications. Noise levels of all individual pieces of equipment shall be determined by tests. The overall noise levels of the complete vehicle while operating shall be calculated on the basis of these tests, taking into account the characteristics of the car body structure, equipment mounts, and the additive effect of multiple noise sources.

C. Noise Measurements

1. Applicable Standards -

For test and measurements the Contractor shall use a Type 2 sound level meter meeting the requirements of the latest revision to ANSI S1.4 - 1971, Specification for General Purpose Sound Level Meters. Where octave band or 1/3 octave band measurements are specified, Contractor shall use an analyzer meeting the requirements for Class II Filters as given in the latest revision to ANSI S1.11 - 1971, Specification for Octave, Half-Octave and Third Octave Band Filter Sets.

Unless otherwise stated, noise herein means sound pressure level as defined in the latest revision to ANSI S1.4 - 1971. All noise levels listed are in decibels referred to 20 micro Pa as measured with the "A" weighting or the "C" weighting network of a standard sound level meter, abbreviated dBA and dBC, respectively. Unless otherwise specified, the "slow" meter response time shall be used. Narrow band or pure tone noise shall be evaluated using 1/3 octave band analysis.

Sound transmission losses specified for car body components such as doors, floors, walls and ceilings refer to sound insulation values obtained by measurement procedures outlined in ASTM E90-75 (for laboratory tests of body elements) or ASTM E336-71 (for field tests of the complete Car), except that octave band rather than 1/3 octave band measurements are specified herein. Laboratory tests of car body elements of limited size are to be used for developmental purposes only and may not be substituted for full scale transmission loss tests on the complete car.

2. Noise Measurement Conditions -

Noise measurements to specification shall be performed in an essentially free field environment, such as outdoors with no nearby structures or reflective surfaces which could influence the measurements by more than 2 dB, other than the standard ballast and tie road bed and the adjacent flat, clear ground.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

C. Noise Measurements (Continued)

For all tests, the levels of all sounds or vibrations other than those being evaluated shall be not less than 10 dB below the levels of the sounds or vibrations being evaluated, when measured with the same weighting network or octave band as that being used for the test.

Measurements of noise produced by equipment alone, prior to installation on the Car, shall be performed with the equipment supported above the floor or ground at the approximate elevation at which it will be mounted in the Car. Where auxiliary methods of driving or loading equipment, such as motors or dynamometers, are required, these devices shall be temporarily enclosed or baffled to eliminate their effect on the equipment noise being measured.

Interior noise shall be measured in a complete, fully finished car, unoccupied except for personnel required to perform the measurements and two observers. Exterior noise with the car moving or stationary shall be measured at a section of standard, at-grade, ballast and tie track free of railings, sound barriers or other wayside obstructions extending above the elevation of the bottom of the rail between the car and the microphone.

Interior and exterior noise measurements shall be performed with the car running on level, tangent, ballast and tie road with continuous welded rail, ground smooth. Welded rail joints along the test section shall be smooth and even on the running surface and on the gauge edge so that no abnormal noise or vibration is generated by wheels passing over the joints.

3. Narrow Band or Pure Tone Noise -

The exterior noise limits set forth in Paragraphs D,1 and D,3 for unmounted equipment and for conditions 1,a and b,1, of the specification table, limiting noise from the complete traction systems and the mounted undercar equipment, must be reduced by 3 dBA if significant pure tones in the range from 300 Hz to 4000 Hz are present in the noise. Pure tone noise shall be considered significant in this context if any 1/3 octave band sound pressure level is 5 dB or more higher than the average of the two adjacent 1/3 octaves containing no pure tones or "tonal" noise.

D. Noise Limits for Equipment Prior to Installation On Car

1. Traction Motors -

The noise produced by each traction motor shall not exceed 80 dBA at 15 feet in any direction from the center of the motor while operating at all speeds from zero to the equivalent of 60 mph car speed and at loads equivalent to maximum dynamic braking in either direction. Normal

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

D. Noise Limits for Equipment Prior to Installation on Car (Continued)

1. (Continued)
cooling air ducts to be employed on the motors on the car may be installed during this test. Normal cooling airflow for the motors must be provided during this test.

2. Propulsion System Gearing -

The noise produced by each propulsion system gearbox shall not create levels in excess of 81 dBA at 15 feet in any direction from the geometric center of each gearbox with gears rotating in either direction at all speeds from zero to the equivalent of 60 mph car speed and at loads equivalent to maximum dynamic braking.

3. Undercar Equipment -

The noise produced by the individual operation of all undercar equipment units, including refrigeration compressors, motors and alternators, blowers, brakes and other noise generating components, except traction motors and gears, shall not exceed 68 dBA at 15 feet in any direction from the center of the equipment while it is operating at normal conditions. All duct work, baffles or appurtenances which form a part of the installed assembly shall be included as part of the equipment for noise tests.

E. Noise Criteria for Equipment After Installation on Car and for Complete Car

1. Noise criteria for equipment installed on the complete car and for the complete car are shown in the following table:

MAXIMUM NOISE LIMITS FOR COMPLETE CAR

Condition	Interior	Exterior	Horizontal Distance from Track Centerline
a. Car Stationary on Jacks			
All traction motors, gearboxes and wheels rotating at all speeds up to 60 mph. All auxiliary equipment off except traction motor cooling blowers.	----	82 dBA	15 ft.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

E. Noise Criteria for Equipment after Installation on Car and for Complete Car (Continued)

1. (Continued)

	Interior	Exterior	Horizontal Distance from Track Centerline
b. Car Stationary on Ballast and Tie Track			
1) Each individual item of undercar equipment or each complete operating system, including the air conditioning system, with all system components operating at normal conditions. Includes starting and stopping transient noises.	65 dBA 78 dBC	68 dBA	15 ft.
2) Simultaneous operation of all vehicle systems including air conditioning and auxiliaries except traction motors, under normal conditions.	68 dBA 80 dBC	60 dBA	50 ft.
c. Car Moving			
1) Two car train moving at constant 60 mph with all vehicle systems operating simultaneously under normal conditions.	----	80 dBA	50 ft.
2) Car moving at any speed up to 60 mph and under all normal conditions of accelerations, deceleration and coasting with all vehicle systems operating simultaneously under normal conditions.	70 dBA	----	----
3) Full or partial application of friction brakes at low speeds, 0 to 15 mph.		75 dBA	15 ft.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

2. Noise Measurement Locations -

Interior noise limitations apply at any point along the car center line 4.5 ft. above the car floor and 2 ft. or more from the end walls.

All exterior noise measurements shall be made with the microphone on a horizontal plane passing through the axles on both sides of the car at the horizontal distances from the track centerline shown in the specification table. For measurements of noise produced by specific, individual pieces of equipment or operating systems, the microphone shall be located on a transverse line passing through the center of the equipment being measured. The limitation on exterior noise produced by all equipment operating simultaneously applies at the specified distance along the entire length of the car.

F. Noise Limits for Miscellaneous Equipment

1. Door Operation Noise

Noise produced by operation of all the doors on one side of the car only, shall be not exceed 72 dBA using "fast" meter damping, anywhere in the car 1 ft. or or more from the doors or door pockets and between 3 ft. and 6 ft. above the floor.

2. Public Address System

Noise generated by the car public address system in the standby condition shall not exceed 40 dBA when measured 12 inches away from any loudspeaker with P-A auxiliary equipment energized and operating and with the car electrical system energized.

3. Lighting System

Noise generated by fluorescent lamps, fixtures and ballasts installed in the car and energized at rated voltage and frequency, shall not exceed 48 dBA when measured 1 foot from each lighting fixture.

G. Car Body Transmission Loss

1. The sound transmission loss of the car body floor, wall and ceiling assemblies in completed form shall be adequate to achieve the interior noise level limits specified in 15.02, E, but in no case shall the average sound transmission loss of each characteristic section of the car body be less than specified in the following table.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

G. Car Body Transmission Loss (Continued)

1. (Continued)

SOUND TRANSMISSION LOSSES BY OCTAVE BANDS

Octave Band Center Frequency	Entire Floor	Walls Including Windows But Excluding Doors	Ceiling or Roof	Doors
250 Hz	27 dB	23 dB	23 dB	14 dB
500 Hz	35	31	31	22
1000 Hz	38	34	34	25
2000 Hz	38	34	34	25

The sound transmission loss shall be averaged over each characteristic section of the car body defined in the above table, and must include the influence of all sound energy which transmits through all weak areas such as apertures, door seals, air ducts, or openings for supply and return duct

2. The Contractor shall, using the procedures outlined in ASTM E336-71, Recommended Practice for Measurement of Airborne Sound Insulation In Buildings, perform tests before delivering the cars to CTA to determine the sound transmission loss of the car body sections. Evaluation of the sound transmission loss of each characteristic section of the completed car body shall be done using one or both of the following procedures:

   With the car located outdoors on ballast and tie, at-grade track or indoors in a space where reflected sound from nearby walls or floors will not influence the sound radiated from the car body by more than 2 dB, the Contractor shall, using portable loudspeakers in a manner reviewed by the Engineer, create a random noise of constant level, for the frequency range encompassing the 250 Hz to 2000 Hz center frequency octave bands, with sufficient Sound Pressure Level inside the car that the noise transmitted through the car body is at least 10 dB higher than the outside ambient SPL in each octave band and with sufficient diffusion or distribution that the sound level in the car is uniform within 3 dB at 12 inches or more from any body surface. (Achieving a uniform sound field over the car floor will require removal of the seats.) Using this procedure the car body section sound insulation can be evaluated by using a sound level meter and octave band analyzer to measure the space average SPL inside the car in the 250, 500, 1000, and 2000 Hz center

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

2. (Continued)

frequency octave bands and by also measuring the exterior SPL for each of these octave bands at a distance of 12 inches from all car surfaces at a sufficient number of locations to determine the average noise reduction for each characteristic body section, such as the floor, walls, roof and doors. The measurements must include the influence of any sound leakage such as at ducts, seals or openings and the influence of flanking sound transmission paths at locations such as the floor/wall juncture. The difference between the interior space average Sound Pressure Level and the average exterior Sound Pressure Level at each section is the Noise Reduction provided by the car body sections. Noise Reduction measured in this manner is 6 dB greater than the transmission loss. The measurements must be corrected to transmission loss in accordance with procedures given in ASTM E336-71 in order to determine compliance with the specified minimum sound insulation of each car body section.

Alternatively, with the car located near highly reflective surfaces, such as over a maintenance and inspection pit, the transmission loss may be measured in accordance with the two room reverberant sound field methods indicated in ASTM E336-71. To create a satisfactory reverberant condition outside the car, and to define the boundaries of the space, temporary baffles or barriers reviewed by the Engineer shall be placed between the car body exterior and the reflecting surfaces, such as between the car body exterior walls at the floor level and the edges of a maintenance pit, to define a closed space beneath the car for testing the car floor transmission loss. The temporary baffles both define the space exterior to the car and prevent flanking paths outside the car from influencing the measurements, for example, by preventing sound transmitted through the car walls or doors from bypassing the floor during a test of the floor.

H. Vibration

1. Traction Motors -

The vibration of any traction motor, detached and supported on resilient mountings providing at least 0.25 inch static deflection, shall not exceed 0.0015 inch peak-to-peak displacement anywhere on the motor while the motor is rotating at maximum normal operating speed and at 50% of maximum normal operating speed.

2. Equipment Installed on Car -

Equipment and auxiliaries mounted anywhere on the car or truck shall not cause vertical or horizontal vibrations anywhere on the floor, walls, ceiling or seat frames of the stationary car in excess of the following:

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

H. Vibration (Continued)

2. Equipment Installed on Car (Continued)

0 Hz to 1.4 Hz	0.1 in. peak-to-peak displacement
1.4 Hz to 20 Hz	0.01 g zero-to-peak acceleration
above 20 Hz	0.03 in./sec. zero-to-peak velocity

3. Wayside Ground Vibration -

Ground-borne vibration created by operation of a two-car train at constant speed on tangent track shall not exceed the ground-borne vibration created by operation of a reference two-car train of CTA 2401 - 2600 Series Cars.

Tests for vibration levels in the vertical direction shall be conducted by the Contractor at two sites selected by the Engineer--one site to be elevated structure with jointed rail, and one site to be ballast and tie tracks with continuous welded rail. The test and reference trains shall be operated at constant speeds of 30 MPH and 45 MPH in both directions on the same track. Measurements with the test and reference trains shall be done at the same time at each of the test sites to assure that conditions are the same for both trains.

The tests shall be accomplished with fully completed and equipped cars with loads not to exceed 1,500 pounds of personnel and instrumentation.

Measurements of vibration at the elevated structure test site shall include, but not be limited to, measurement at the base of structure column near track centerline and on an at-grade concrete slab or side-walk, an asphalt slab, a curb, or on a building structure at wayside distances (perpendicular to the track alignment) of 25 feet from the column centerline. At the ballast and tie track test site the tests shall include measurements of ground-borne vibration on an at-grade slab or curb, a concrete block buried in the ground or a firmly bedded tie at 25 feet and 50 feet from track centerline.

The vibration measurements shall be made with a lightweight piezo-electric accelerometer with amplifier and recording system giving uniform response, $\pm$ 1 dB, from 2.0 Hz to 600 Hz. The transuducer signal shall be analyzed by 1/3 and 1/1 octaves or shall be recorded in a manner permitting such analysis.

The vibration data from each train passby shall be analyzed in terms or rms acceleration for each 1/3 octave band from 2.5 Hz to 500 Hz center frequency, and for each octave band from 4 Hz to 250 Hz center frequency using a method reviewed by the Engineer.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

H. Vibration (Continued)

3. Wayside Ground Vibration - (Continued)

The average vibration levels shall be determined by arithmetically averaging the 1 second period maximum levels in decibels for all of 1/3 octave frequency bands within each of the following four (4) groups or sets of frequency ranges:

Group	1/3 Octave Center Frequencies - Hz
1	3.15 to 6.3
2	8.0 to 20
3	25 to 63
4	80 to 200

The test train average ground-borne vibration levels may not exceed those of the reference train, and the highest levels caused by the test train may not exceed those for the reference train for each of the groups of frequency ranges.

The highest levels shall be defined as the greatest or maximum 1-second period 1/3 octave band level for any 1/3 octave within each group or set and will not necessarily be the same 1/3 octave frequency for the test train and the reference train.

It shall be the Contractor's responsibility to perform the necessary preliminary tests and calculations during the design and development of the car as required to substantiate that the ground-borne vibration levels of the completed car will meet these requirements. It shall also be the Contractor's responsibility to conduct at his own expense the performance tests necessary to show that the ground-borne vibration produced by the cars does not exceed that produced by the operation of Cars 2401-2600.

I. Ride Quality

The ride quality shall be equal to or better than that of CTA Cars 2401-2600.

It shall be the Contractor's responsibility to test the same cars used in the above Noise Tests to insure that the ride quality is equal to or better than that found on Cars 2401-2600. CTA will supply data on the ride quality of Cars 2401-2600.

SECTION 15 - TESTS AND ADJUSTMENTS, NOISE AND VIBRATION CRITERIA AND SHIPMENT OF CARS (Continued)

15.02 - Noise and Vibration Criteria (Continued)

I. Ride Quality (Continued)

The ride quality shall be recorded as vertical, lateral and longitudinal accelerations measured in an appropriate manner at an appropriate location in the car at speeds from 0 to 60 mph. The tests shall be conducted at the same track sites used for the vibration tests.

15.03 - Shipment of Cars

The cars shall be shipped on flat cars to the property of the Authority at Skokie Shops, 3701 Oakton Street, Skokie, Illinois, 60076, FOB destination.

The Authority has an interchange with the Chicago and NorthWestern Transportation Co., at Skokie Shops. The cars and components must be adequately blocked and tied down to prevent damage to any bearings, pivots, shock mounts or other car parts.

Measures and procedures to protect the cars from damage during shipment shall be reviewed by the Engineer.

Flat cars must be chosen to insure that no part of the car can interfere with CTA fixed property, third rails, signals or switch stands.

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Pages 01561-1 through 01561-5

SECTION 01561

NOISE CONTROL

1.01 DEFINITIONS

A. Daytime refers to the period from 7:00 a.m. to 10:00 p.m. local time daily except Sundays and legal holidays. Nighttime refers to all other times including all day Sunday and legal holidays.

B. Construction limits are defined as the existing Right-of-Way lines and construction easement and work site property lines as shown on the drawings.

C. Construction Noise Zones are defined for the purpose of these noise and vibration control requirements as those delineated on the map contained herein.

1.02 CONSTRUCTION NOISE CONTROL

A. General:

1. Perform construction operations in a manner to minimize noise. Provide working machinery and equipment with efficient noise suppression devices and employ other noise abatement measures necessary for protection of both employees and the public.

2. Restrict working hours and schedule operations in a manner that will minimize to the greatest extent feasible the disturbance to the public areas adjacent to the work and to occupants of buildings in the vicinity of the work.

3. Protect employees against noise exposure in accordance with the requirements of the Occupational Safety and Health Act of 1972.

4. Compliance with the requirements of this Section will not offer any relief from responsibility for compliance with local ordinances, regulations, and other Sections.

B. Special Requirements:

1. Compliance with the requirements of this Section will require the use of machines with effective mufflers or enclosures and selection of quieter alternative procedures. Compliance may also require the use of completely closed boarding (tongue and groove plywood or sheathing) around work sites or a combination of closed boarding and effective mufflers or enclosures. It will be necessary to arrange haul routes to minimize noise at residential sites and it may be necessary to place operating limitations on machines and trucks.

2. Submit working drawings of work sites and haul routes showing provisions for control of construction noise.

C. Monitoring Procedures:

1. Monitor noise levels of work operations to assure compliance with the noise limitations contained herein and retain records of noise measurements for inspection by the Engineer. Promptly inform the Engineer of any complaints received from the public regarding noise. Describe the action proposed and the schedule for implementation and subsequently inform the Engineer of the results of the action.

D. Measurement Procedures:

1. Except where otherwise indicated, perform all noise measurements using A-weighting and slow response of an instrument complying with the criteria for a Type 2 General Purpose sound level meter as described in ANSI S1.4.

2. Estimate peak values of impulse or impact noises on a C-weighting and fast response.

3. Measure noise levels at buildings affected acoustically by the operations at points between three feet and six feet from the closest building face at five feet above grade level or at a distance of 200 feet from the construction limits at five feet above grade level, whichever is appropriate. At some buildings measurements at elevations higher than five feet above grade may be required, as determined by the building occupancy.

E. Noise Level Restrictions:

1. <u>Noise Level Restrictions in All Areas</u>:

   In no case expose the public to construction noise levels exceeding 90 dBA (slow) or to impulsive noise levels with a peak sound pressure level exceeding 125 dBC (fast).

2. <u>Noise Level Restrictions at Affected Structures</u>:

   Conduct construction activities in such a manner that the noise level 200 feet from the Construction Limits or at affected buildings outside the Construction Limits do not exceed the levels listed in the following schedules. Where more than one noise limit is applicable, use the more restrictive requirement for determining compliance.

   a. Continuous noise:

      Prevent noises from stationary sources, rock drills, parked mobile sources or any source or combination of sources producing repetitive or long-term noise lasting more than a few hours from exceeding the following limits:

ZONE	AFFECTED STRUCTURE OR NOISE ZONE	MAXIMUM ALLOWABLE CONTINUOUS NOISE LEVEL, dBA	
	Residential:	Daytime	Nighttime
-A-	Single family residence areas	60	50
-B-	Along an arterial or in multi-family residential areas, including hospitals	65	55
-C-	In semi-residential/commercial areas, including hotels	70	60
	Commercial:	At All Times	
-D-	In semi-residential/commercial areas, including schools	70	
-E-	In commercial areas with no nighttime residency	75	
	Industrial:		
-F-	All Locations	80	

b. Intermittent Noise:

Prevent noises from non-stationary mobile equipment operated by a driver or from any source of non-scheduled intermittent, non-repetitive, short-term noises not lasting more than an hour from exceeding the following limits:

Zone	Affected Structure or Noise Zone	Intermittent Noise Level, dBA	
	Residential:	Daytime	Nighttime
-A-	Single family residence areas	75	60
-B-	Along an arterial or in multi-family residential areas, including hospitals	80	65
-C-	In semi-residential/commercial areas, including hotels	85	70

Zone	Affected Structure or Noise Zone	Intermittent Noise Level, dBA
	Commercial:	At All Times
-D-	In semi-residential/commercial areas, including schools	80
-E-	In commercial areas with no nighttime residency	85
	Industrial:	
-F-	All locations	90

3. In the event of a conflict as to which limit is applicable, the Authority shall have the sole right to make the final determination as to which noise limit shall apply.

4. Refer to Specification Section 02211, Article 1.06B for limits on blasting (Tunnels only.)

F. Noise Emission Restrictions:

1. Use only equipment meeting the noise emission limits listed below, as measured 50 feet from the equipment in substantial conformity with the provisions of SAE J366a and SAE J952b and in accordance with this Section.

TYPE OF EQUIPMENT	NOISE LIMIT
All equipment other than highway trucks, including hand tools and heavy equipment	85 dBA
Highway trucks in any operating mode or location	83 dBA

G. Noise Abatement Measures:

1. Utilize the noise abatement measures listed below to minimize to the greatest extent feasible the noise levels in all areas outside the Construction Limits.

   a. Utilize shields, impervious fences or other physical sound barriers to inhibit transmission of noise.

   b. Utilize sound retardent housings or enclosures around noise producing equipment.

   c. Utilize effective intake and exhaust mufflers on internal combustion engines and compressors and rock drills.

   d. Line or cover hoppers, storage bins and chutes with sound deadening material.

e. Do not use air or gasoline driven saws.

f. Conduct truck loading, unloading and hauling operations so that noise is kept to a minimum.

g. Route construction equipment and vehicles carrying soil, concrete or other materials over street and routes that will cause the least disturbance to residents in the vicinity of the Work. Advise the Engineer in writing of the proposed haul routes and times prior to securing a permit from the local government, if permits are required.

h. Locate stationary equipment to minimize noise impact on the community, subject to approval of the Engineer.

i. Use augering or sonic hammer methods for installing piles in lieu of impact pile drivers.

j. No surface rock drilling will be permitted between the hours of 10:00 p.m. and 7:00 a.m.

k. Provide attenuators at the exhaust of ventilation fans to filter out the third harmonic sound level to a maximum of 45 dB. A 48" attenuator will be available to the contractor at no charge. (Add this section to Humboldt and La Salle only.)

1.03 MEASUREMENT AND PAYMENT

A. No separate measurement or payment will be made for work required under this Section. All costs in connection therewith will be considered incidental to the work under this contract.

E N D O F S E C T I O N

:mp
6.8.2

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3.1.12.11 Exterior Noise from Ancillary Facilities

The following criteria apply to noise from transit system ancillary facilities:

	Community Area Category	Maximum Noise Levels Transient	Continuous
I	Low Density Residential	50 dBA	40 dBA
II	Average Residential	55	45
III	High Density Residential	60	50
IV	Commercial	65	55
V	Industrial and Highway Corridor	75	65

These criteria should be applied at 50 feet from the shaft outlet or other ancillary facility or should be applied at the setback line of the nearest building or occupied area, whichever is closer. Transient criteria apply to short duration events such as train passby noise transmitted from a vent shaft opening. Continuous criteria apply to noises such as fans, cooling towers, or other long duration noises except electric transformer hum. The criteria for transformer noises or hum should be 5 dBA less than given in the table for continuous noise. Sound attenuation is not required on the outlet of emergency exhaust fans except in cases where the emergency exhaust fans are used as part of a station ventilation system.

3.1.12.12 Fare Collection and Vertical Circulation Equipment

3.1.12.12.1 Scope

- Fare gates

3.1.12.12.1 Scope (cont.)

- Change machines
- Emergency refund machines
- Escalators
- Elevators

3.1.12.12.2 General Considerations

For equipment located in public areas and for all normal operating conditions, the noise level at 3 feet from the above listed equipment shall not exceed 55 dBA for steady-state noise, and transient noise shall not exceed 60 dBA measured using the fast meter response.

3.1.12.12.3 Escalator Noise

Noise produced by escalators operating individually in either direction under no load and under maximum load in the station environment shall not exceed 55 dBA 5 feet above the tread at the entrance combs at both ends of the escalator.

3.1.12.12.4 Elevator Noise

Steady-state noise produced by elevators or associated equipment shall not exceed 55 dBA (Slow) in public spaces 3 feet or more from the elevator or associated equipment or within the elevator cab at any location 5 feet above the floor and 1 foot or more from any wall. Transient noise produced by elevators or associated equipment, not including entrance door operations, shall not exceed 60 dBA (Fast) in public spaces 3 feet or more from the elevator

3.1.12.12.4 Elevator Noise (cont.)

or associated equipment or within the elevator cab at any location 5 feet above the floor and 1 foot or more from any wall. Transient noise produced by operation of the elevator door shall not exceed 65 dBA (Fast) 3 feet or more from the elevator door inside or outside of the elevator cab.

3.1.12.13 Ventilating Equipment

3.1.12.13.1 Fan and Equipment Rooms

Spaces for fans and other potentially noisy equipment shall be separated from public areas insofar as possible. If direct access into such rooms from public areas cannot be avoided, provide doors having a suitable sound rating. Control sound transmission through other openings by appropriate means such as acoustically lined ducts or shafts.

3.1.12.13.2 Fan Equipment

The noise levels from fan shafts and other stationary equipment are dependent on the sound level radiated by the machinery. For station ventilation fans and subway emergency ventilation fans the sound power level should not exceed the values given in Figure 3.1.12 C.

3.1.12.13.3 Fan and Vent Shafts

For fan and vent shafts with surface gratings or openings the noise shall be limited in accordance with the criteria for exterior noise from ancillary facilities, paragraph 3.1.12.11.

Octave Band Center Frequency (Hz)	Sound Power Level (dB) Subway Emergency Ventilation Fans	Sound Power Level (dB) Underplatform Heat Removal Fans
63	87	104
125	96	107
250	98	109
500	99	110
1000	99	109
2000	94	104
4000	91	100
8000	90	96

Fans shall have certified sound power levels not to exceed the above decibel ratings (10^{-12} watts) when operating under specified load conditions and measured at the fan in accordance with the AMCA test code. Emergency ventilation fans shall be operated in both directions with inlet bell and outlet cone.

3.1.12.13.3 Fan and Vent Shafts (cont.)

Vent shaft noise reduction shall be achieved by absorption treatment in the shafts - applied to the walls and ceilings. Fan shaft noise reduction shall be achieved by use of standard duct attenuators in shafts where the fans are near the surface gratings. For shafts with fans located remotely from the grating the noise reduction shall be achieved by the use of standard attenuators and sound absorption treatment applied to the fan room and shaft walls and ceilings with the combination to achieve the total attenuation required. Sound absorption treatment shall consist of 2 to 4 inch thick mechanically attached panels, e.g. expanded cellular glass foam blocks.

APPENDIX D

INSTRUMENTATION COMPONENTS

When selecting instrumentation components, one must have at least a general understanding of the specific types of noise and vibration to be measured. For example, the equipment needed to obtain a frequency analysis of the ground vibration of train passbys must be selected with unusual care. Ground vibration is a relatively low-level, low-frequency phenomenon, lasting for a short period of time, generally less than ten seconds. Equipment used to measure and analyze industrial vibration is often designed for steady-state vibration, at levels high enough to cause fatigue failure. With such equipment, it is difficult or impossible to obtain a vibration signal level for groundborne vibration which is above the electronic noise floor of the equipment. Furthermore, because the analysis equipment is designed for steady-state signals, analysis of the low-frequency components of the vibration may not be practical.

D.1 SOUND LEVEL METERS

The sound level meter is the most basic measurement instrument for any type of noise. The selection and use of sound level meters is discussed in some detail in Chapter 4. The primary requirement in sound level meters is the necessary level for accuracy. Sound level meters are classified by the ANSI Standards (see Ref. 4.8). For transit noise measurements, either the Type 1, Precision, or Type 2, General Purpose should be used.

All sound level meters are equipped to give a visual indication of the rms sound level using either the slow or fast meter response settings. Slow and fast are rigorously defined in the ANSI Standards and can be related to the rms averaging time. To adjust for different noise environments, most sound level meters have attenuators; usually the attenuators change the level in 10 dB steps. Beyond these basics, sound level meters come equipped with a wide assortment of other features. Those likely to be useful in transit noise applications are octave or 1/3 octave filters, integrating circuits that allow direct measurements of L_{eq}, various weighting circuits, digital displays, and elements for measuring peak and impulse levels.

D.2 GRAPHIC LEVEL RECORDERS

A graphic level recorder, often abbreviated as GLR, is a strip chart recorder, used for the display of sound or vibration level as a function of time. It is second only to the sound level meter in usefulness. It is important to recognize that graphic level recorders are not the same as a standard strip chart recorder.

Graphic level recorders are usually equipped with attenuators; rms, peak, d.c., and perhaps average detectors; paper and pen driving mechanisms; and are available with numerous attachments for performing a variety of functions. Although most graphic level recorders are capable of recording d.c. voltages linearly, i.e., not in terms of dB levels, the dynamics of the pen driving control mechanism and electronics are not the same as used in conventional X-Y or d.c. recorders. Level recorders are specifically designed for the logarithmic display of signal voltages.

The averaging time for the rms detector and paper speed is adjustable over a wide range. On most level recorders, the full-scale to baseline range can be changed by exchanging a potentiometer. Cable, chain, or electronic drives are available for synchronizing the level recorder with an oscillator or narrowband frequency analyzer. Some level recorders can also generate polar plots of level as a function of direction. By displaying a level decay as a function of time, reverberation times may be calculated. Special protractors that allow reading reverberation times directly from the strip chart are available.

D.3 CRITERIA FOR MICROPHONE SELECTION

The microphone is a key component in a system for measuring and analyzing sound. Selection of the appropriate microphone is very important, since the microphone is often the weakest link in a measurement system. Things to consider when selecting a microphone are directivity, frequency response, and sensitivity.

Directivity: Microphones used for noise measurements must be of the omnidirectional type. Many microphones are designed to be sensitive in primarily one direction; these "directional" microphones are almost never used for noise measurements because the directionality makes it difficult or impossible to accurately define the sensitivity. Unfortunately, even omnidirectional microphones exhibit directional characteristics at high frequencies where the wavelength of sound becomes comparable to the microphone dimensions. This directional characteristic is related to the diffraction of sound waves by the microphone, an effect that begins to become significant when the diameter of the microphone is approximately one-fifth of the wavelength of sound. The directivity of microphones is almost always included in manufacturers' literature.

The ideal microphone for acoustic measurements would be omnidirectional over the entire frequency range. Unfortunately, this ideal is never achieved, although there are a number of microphones that approximate omnidirectionality or can be corrected to approximate omnidirectionality. There are basically two types of microphones for sound measurements: the "free field" response microphone and the "pressure response" microphone. The differences mainly concern application and frequency response characteristics required for a particular measurement. A "free field" type has been corrected for pressure doubling effects at wavelengths approaching the dimensions of the microphone capsule and has optimum uniform frequency response at 0 degree angle of incidence.

In the past, most transit system noise measurements have been performed with 1-inch-diameter "free field" microphones fitted with special random incidence correctors to improve the omni-directional characteristics. Pressure response microphones that are 1-inch-diameter or larger should never be used for measurements of a diffuse sound field. They are designed to provide uniform response to the sound pressure actually imposed on the diaphragm and were originally intended for use in small couplers for the purpose of developing calibration standards. For a sound field at a 0 degree angle of incidence to the microphone axis (i.e., perpendicular to the diaphragm), pressure response microphones have a rising response at high frequencies caused by the effects of diffraction and reflection.

Some of the current generation sound measurement devices are being equipped with microphones that are 1/2 or 1/4 inch in diameter. Since diffraction and reflection effects are dependent on the microphone diameter, they create considerably less frequency response deviation with the smaller diameter microphones. It has been found that for microphones 1/2-inch diameter or smaller, both pressure response and free field response microphones can be used for most indoor or outdoor measurements with little or no corrections up to frequencies of 10 to 12 kHZ. However, with equipment using 1-inch-diameter microphones, the free field correction versions should be used with a random incidence corrector placed over the diaphragm.

Frequency Response: The frequency response of a microphone determines the types of sound fields it can be used to measure. The ANSI standards of sound level measurements define the frequency response limits for various classes of microphones. The limits on Type 1 microphones are the most restrictive. Type 2 microphones are generally adequate for community noise surveys, measurements to evaluate employee noise exposure, and other noncritical measurements. When performing detailed measurements for diagnostic purposes or for evaluating noise control methods, Type 1 microphones should be used.

Sensitivity: The response of the microphone is proportional to mechanical motion of the diaphragm, which is caused by the sound pressure imposed on the microphone assembly. Diaphragm sensitivity, defined as the electrical output voltage divided by the sound pressure, is usually expressed in dB re 1 volt per Pascal. Typical sensitivities are 20 to 50 millivolts per Pascal (-26 to -36 dB re 1 volt per Pascal). The sensitivity of a microphone varies with the temperature, humidity, and atmospheric pressure. The temperature coefficient is the most important and can be corrected by the manufacturers' calibration charts usually supplied with the microphone. Temperature effects can usually be ignored at ordinary room or outdoor temperatures. When taking community noise measurements in cold temperatures or when large temperature variations occur, recalibration of the microphone and measurement system may be necessary. The effects of barometric pressure and humidity on the microphone sensitivity are not usually significant, although high humidity can cause problems with some microphones. Reduced atmospheric pressure at high altitudes can slightly affect some microphones, depending on design, and the manufacturer should be consulted if critical measurements are to be performed at altitudes of several thousand feet. Generally, for altitudes encountered at the majority of transit systems, the sensitivity variation of most microphones will be less than 1 dB.

D.3.1 Types of Microphones

Condensor Microphones: The condensor microphone is the most stable; indeed, this type is used as the laboratory calibration standard. This microphone consists of a tensioned diaphragm, separated and insulated from a perforated or solid backplate by an entrapped airspace. This forms a capacitor with one moveable plate, the diaphragm. A constant charge is supplied to the capacitor by means of a polarization circuit, and the output voltage is the result of variations of capacitance due to motion of the diaphragm induced by incident sound pressure waves. Condensor microphones require a special preamplifier with extremely high input impedance, capable of supplying the necessary polarization voltage and responding to the capacitance changes without drawing current. The advantages of condensor microphones are superior stability, low temperature coefficient, high sensitivity, and a uniform broad frequency response. Their major disadvantage is susceptibility to high humidity, which may produce noise and actual failure. The noise is produced when moisture collects on the diaphragm and perforated backplate. A heater and desiccator can prevent most of the problems associated with humidity.

<u>Electret Microphones</u>: Electret microphones have been developed that are similar to the condensor microphone except that a permanently polarized dielectric is used between the diaphragm and backplate to develop the polarization charge rather than requiring an external polarizing voltage. The response of an electret microphone is comparable to that achieved with a condensor microphone. The advantages of the Electret microphone are reduced cost, high sensitivity, uniform and broad frequency response, a low temperature coefficient, good stability in normal environments, and greater freedom from the effects of humidity-induced noise. However, the sensitivity is somewhat less stable than the condensor microphone over long periods of time, and electrets may exhibit less dynamic range, a limitation due to its slightly lower sensitivity. Also, the frequency response of electret microphones is more sensitive to humidity than is that of condensor microphones.

<u>Crystal Microphones</u>: Crystal (or piezoelectric) microphones consist of a diaphragm attached to a piezoelectric crystal. The crystal creates a charge proportional to the force applied to the diaphragm. The major advantages of crystal microphones are their

low cost and reliability. However, their sensitivity and frequency responses are not as stable as those of condensor and electret microphones, so that crystal microphones are rarely used for measurements requiring high precision. Although they rarely meet the Type 1 specifications, they are commonly used on Type 2 or Type 3 sound level meters and are often used for community and industrial noise surveys.

Dynamic Microphones: Dynamic microphones are commonly used for general purposes such as speech or music recording. They consist of a diaphragm attached to a moving coil in a magnetic field in which the voltage output is proportional to the coil velocity induced by the sound field. The frequency response characteristics of the dynamic microphone are less uniform than the microphones described above, and they are not generally suitable for noise measurement purposes. However, they are relatively inexpensive and often desirable for announcement or communication. The output of the dynamic microphone is of very low impedance. This allows the microphone to be used with very long cables, with only a minimal amount of noise pickup. A matching transformer should be used to enhance the signal level of the microphone prior to amplification.

D.3.2 Vibration Sensitivity of Microphones

All microphones are sensitive to vibration in some degree, which can cause problems when measuring noise in high vibration environments. Condensor and electret microphones typically will develop a signal equivalent to a sound pressure level of 85 or 90 dB, for a vibration acceleration of about 1 g. In piezoelectric microphones, vibration sensitivity is equivalent to a sound pressure level of about 100 to 150 dB for 1 g acceleration, a considerably higher level than that of condensor or electret microphones. Dynamic microphones are very sensitive to vibration, and this is another reason they are not generally recommended for noise measurements. There is a theoretical lower limit of vibration sensitivity for all microphones, due to the air mass effect on the microphone diaphragm. For a 1-inch-diameter microphone, the theoretical lower limit is equivalent to a sound pressure level of 67 dB for 1 g acceleration.

D.3.3 Signal Conditioning for Microphones

Electret and condensor microphones require extremely high input impedance preamplifiers to match the transducer to the input characteristics of the measurement instrumentation. The microphone is usually mounted directly on the preamplifier, forming a single unit, so that there is no cable between the microphone and preamplifier to cause a loss of signal. The output cable from the preamplifier becomes the microphone cable to the amplifier or recorder carrying both power for the preamp and the microphone signal. Lengths of cable up to 100 to 200 m can generally be used unless signal levels are high. Short cables are available for use between condensor microphones and their preamplifiers and are especially designed to prevent loading of the microphone output, but length is limited to about 3 m. Only preamplifiers specifically designed for use with condensor or electret microphones should be used. It is advisable to use sound level meters or preamplifiers supplied by the manufacturer of the specific microphone. Some manufacturers supply preamplifiers which can be used with other manufacturers' microphones and vice versa, but care must be exercised in selecting the appropriate microphone and preamplifier. When mixing manufacturers' equipment, one should consider their separate electrical noise floor and peak signal handling capabilities, in order to ensure maximum use of the dynamic range achievable by whatever type of microphone is used. An accelerometer preamplifier may be adapted for use with a piezoelectric microphone.

D.3.4 Wind Screens

Wind screens are available from microphone manufacturers for use with their specific microphones. These wind screens should be used whenever possible to reduce the effects of wind buffeting and to protect the microphone from shock or rough handling. Large wind screens are available for community noise measurements where the microphone may be placed in windy environments. Wind screens affect the microphone response by only a minimal amount, which may be neglected for most measurements.

D.4 VIBRATION TRANSDUCERS

Vibration transducers are needed for measurement of vibration in terms of acceleration, velocity, or displacement. The accelerometer is by far the most common vibration transducer, and most of the following discussion concerns piezoelectric accelerometers and associated signal conditioning.

Piezoelectric accelerometers are basically undamped, single degree of freedom, spring mass systems with a seismic mass and a piezoelectric crystal as spring. This assembly produces a very high resonance frequency, typically in the range of 10 to 30 kHz, and some special accelerometers have resonance frequencies as high as 100 kHz. Their very low-damping characteristics result in very little phase shift up to the resonance frequency. However, the mounting of the accelerometer will lower its resonance. The sensitivity of the mounted accelerometer is typically about 1.04 times its nominal sensitivity at about one-fifth of the mounted resonance frequency. A rule of thumb for selection of accelerometers is that the mounted resonance frequency should be two to five times the upper frequency of interest, depending on the need for linear response. The influence of various mounting techniques on accelerometer response is discussed in Chapter 4.

One very useful feature of the piezoelectric accelerometer is its very large dynamic range, which is limited at low levels by its sensitivity and the noise floor of the electronics and at high levels by the nonlinear response of the transducer. Some accelerometers, when coupled to the appropriate electronics, can measure accelerations from as low as 1×10^{-6} g's to as high as 1000 g's. The highest acceleration amplitude likely to be encountered on transit car truck and axles are 10 to 150 g's on aerial structures. Ground vibration measurements often involve very low vibration levels, on the order of 10^{-4} to 10^{-3} g's.

A number of piezoelectric materials are used for the construction of piezoelectric accelerometers. Lead-zirconate-titanate ceramic accelerometers offer the highest source capacitances and the highest sensitivities. Quartz accelerometers, on the other hand, have very low source capacitances and low sensitivities. The source capacitances and sensitivities of the so-called "Piezite 10" (Trademark) ceramic accelerometers lie between the lead-zirconate-titanate ceramics and the quartz varieties.

The sensitivity of the typical accelerometer often increases very slightly with increasing acceleration input, typically about one percent per thousand g's. However, in most vibration measurements on rapid transit systems, this nonlinear distortion may be ignored.

Accelerometer sensitivities are normally given in two ways: the charge sensitivity (e.g., 100 pico Couloumbs per g) and the voltage sensitivity (e.g., 50 mv/g). The voltage produced by a piezoelectric accelerometer is determined by the charge stored in the total shunt capacitance. This total consists of the source capacitance of the accelerometer in parallel with the cable capacitance and the input capacitance of the signal conditioning. The output voltage is given by $V=Q/C$, where Q is the charge generated and C is the total shunt capacitance.

Lead-zirconate-titanate accelerometers exhibit decreasing charge sensitivity with increasing frequency. However, the source capacitance of the accelerometer exhibits a similar frequency dependence, decreasing approximately one percent per octave increase in frequency. The voltage frequency response of the accelerometer thus becomes relatively flat, provided that the accelerometer source capacitance is large, compared with the cable and input capacitance of the signal conditioning. In contrast, quartz accelerometers typically carry a relatively flat charge and voltage sensitivity. This attribute does not usually outweigh the disadvantage of greatly reduced sensitivity, a property of the quartz type piezoelectric accelerometers. For most transit vibration measurements, the variation of the charge and voltage response is not significant and can be safely ignored. The resonance characteristic of the accelerometer is far more important.

All vibration transducers, including the piezoelectric accelerometer, exhibit some degree of sensitivity to transverse acceleration. This sensitivity is approximately $\pm$ 3% of the nominal sensitivity of the accelerometer along the axis of sensitivity. Like the charge sensitivity variation or the voltage sensitivity variation with frequency, the transverse sensitivity of the accelerometer may usually be ignored.

When measuring very low levels of acceleration, the acoustic sensitivity of the accelerometer may have to be considered, although this can be ignored in most circumstances. Acoustic sensitivity is sometimes significant for low-level ground vibration, in the presence of high ambient community noise. The acoustic sensitivity of the accelerometer is often less than the sensitivity of the measurement surface to the airborne noise and is typically equivalent to 0.1 to 1 g when exposed to a sound pressure level of 150 dB (Ref. 4.5).

A valuable asset of piezoelectric accelerometers is that they can be used under conditions of high humidity, salt spray, sand dust, and altitude, with no degradation of performance. Hermetically sealed models provide better resistance to moisture contamination than those using epoxy sealing, which is used with most accelerometers. If continual exposure of the accelerometer to very high levels of humidity or dampness is anticipated, hermetic models may be desirable (Ref. 4.10).

Long-term temperature effects upon the damping and resonance frequencies of piezoelectric accelerometers are negligible. Charge and voltage sensitivities change, by a very minor amount, with changing temperatures. However, over the range of temperatures likely to be encountered during rail transit vibration measurements, temperature sensitivities are negligible.

Transient temperature changes or pyroelectric effects are far more significant than long-term temperature changes. Transient temperature changes cause spurious output signals, which may be noticeable during ground vibration measurements at frequencies below 10 Hz. Such transient temperature effects add to the background noise of the measurement system employed and can cause blocking of the signal conditioning amplifier. Compression accelerometers are perhaps the most sensitive to pyroelectric effects; shear accelerometers are slightly less sensitive. Quartz accelerometers are less sensitive than those using ceramic crystals. Generally, an accelerometer with low sensitivity to pyroelectric effects should be chosen for ground vibration measurements. However, the selection of such an accelerometer must be balanced against the need for high sensitivity, normally obtained only with compressive accelerometers.

Although accelerometers are relatively durable, there are a number of possibilities for failure: increases in the transverse sensitivity, open or shorted circuits, changes in sensitivity, changes in capacitance, poor frequency response, and low source resistance can all occur and cause breakdown. Some accelerometers may even exhibit a reverse polarity, although this is usually caused by manufacturing defects. If an accelerometer is subjected to severe shock or heat, it should be checked. Excessive heat can alter the sensitivity of an accelerometer by destroying its charge-generating capability. This is rarely a problem at temperatures below about 100 degrees C.

To this point, the discussion of vibration transducers has focused on piezoelectric accelerometers, the transducer by far the most commonly used for transit system vibration measurements. However, several other types of vibration transducers are useful in specialized applications.

Piezoresistive Accelerometers: Piezoresistive accelerometers rely on the piezoresistive characteristics of certain ceramic materials. These accelerometers are essentially strain gauges with very high gauge factors. Because they have a frequency response down to 0 Hz and a very high resonance frequency, they are very useful under certain conditions. These accelerometers are rarely used for transit measurements because they require special strain gauge instrumentation. Reference 4.12 contains a very good description of piezoresistive accelerometers.

Velocity Transducers: Velocity transducers employ a voice coil and inertial mass for the measurement of vibration velocity and are similar in concept to the dynamic microphone. Velocity transducers have low source impedances, so that in contrast to accelerometers they require neither very high input impedance preamplifiers nor charge amplifiers. Velocity transducers are useful under certain circumstances, but are not widely used in the transit industry except during construction monitoring. If necessary, the acceleration signal obtained with a piezoelectric accelerometer may be integrated with an analog integrator to obtain the required vibration velocity signal without the use of a velocity transducer.

Displacement Transducers: There are a number of transducer types available for measuring vibration displacement. One displacement transducer, similar in concept to that of the condensor microphone, uses the variation of capacitance between a probe surface and a conductive measurement surface. Eddy current displacement transducers operate according to the principle of magnetic induction of surface currents in conducting surfaces. Another type of displacement gauge is the linear differential transformer gauge, based on the change in inductance caused by the displacement of a magnetic core within encircling coils. Displacement transducers and their uses are described in a number of handbooks (Ref. 4.6).

One particular advantage of the inductive and capacitive displacement transducers is that the measurement transducer need not touch the measurement surface, so that the transducer mass cannot alter the dynamic motion of the measurement surface. This is an asset in the measurement of vibration of thin panels. Furthermore, the fact that these transducers avoid contact with measurement surfaces allows them to measure the displacement of rotating shafts. In those transducers relying on the principle of magnetic induction, variation in the magnetic permeability of the measurement surface will produce a spurious output. As a result, in many applications such transducers cannot be used.

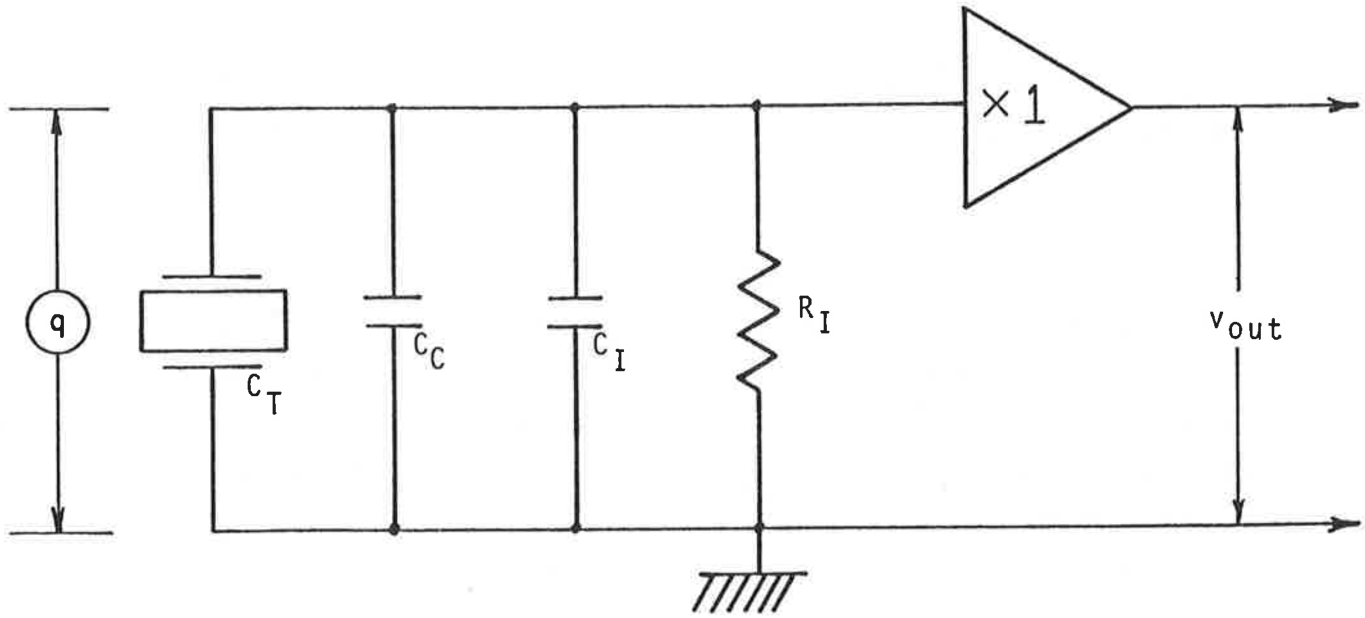
D.5 PREAMPLIFIERS FOR CAPACITIVE TRANSDUCERS

Preamplifiers are normally required as signal conditioning for such capacitive transducers as piezoelectric accelerometers, condensor, electret, piezoelectric microphones, and capacitive displacement gauges. These preamplifiers may be classified into two groups: high impedance voltage preamplifiers and charge amplifiers. Preamplifiers should have a wide dynamic range and broad frequency response for general noise and vibration measurements.

The low-frequency response characteristic of a voltage preamplifier is affected by the transducer's source capacitance and by the input resistance of the preamplifier. For shock or transient sound measurements, the RC time constant of the transducer and preamplifier combination should be 10 to 50 times longer than the duration of the transient (Ref. 5.13).

Ultrahigh Impedance Voltage Preamplifiers: Figure D.1 is a simplified circuit diagram of an ultrahigh input impedance voltage unity gain preamplifier. Such a preamplifier is used to transform the normally high source impedance of the transducer to a low impedance, suitable for driving long cables or amplifiers with relatively low input impedances. The transducer's voltage sensitivity is determined by the transducer charge sensitivity, divided by the total shunt capacitance (consisting of the sum of the transducer source capacitance, input capacitance of the preamplifier, and the cable capacitance). The low-frequency response and phase shift of the transducer preamplifier combination are affected by the total shunt capacitance and are given as f_o and ϕ in Figure D.1. Most good accelerometer voltage preamplifiers have input impedances on the order of 1000 megohm, so that the typical -3 dB point for an accelerometer and cable capacitance of 200 pico farads will be about 0.8 Hz.

Since the voltage frequency response of an accelerometer is usually flatter than that of its charge frequency response, the voltage preamplifier seems slightly more desirable than a charge amplifier. However, for many measurements, the differences between the voltage and charge response characteristics are insignificant. Of greater concern is the variation in the system sensitivity and/or the low-frequency cutoff frequency, caused by differences among various accelerometer source capacitances and cable lengths. Due to the low-frequency response variation, the voltage preamplifier may be less desirable than a charge amplifier for some applications.



R_I = Input Resistance

C = $C_T + C_c + C_I$

C_T = Transducer Capacitance

C_C = Cable Capacitance (≈30 pfd/ft)

C_I = Input Capacitance (≈5 to 10 pfd)

C = Total Capacitance

q = Charge

$$|v_{out}| = \frac{|q|/C}{[1 + (f_o/f)^2]^{1/2}}$$

$f_o = 1/(2\pi RC)$

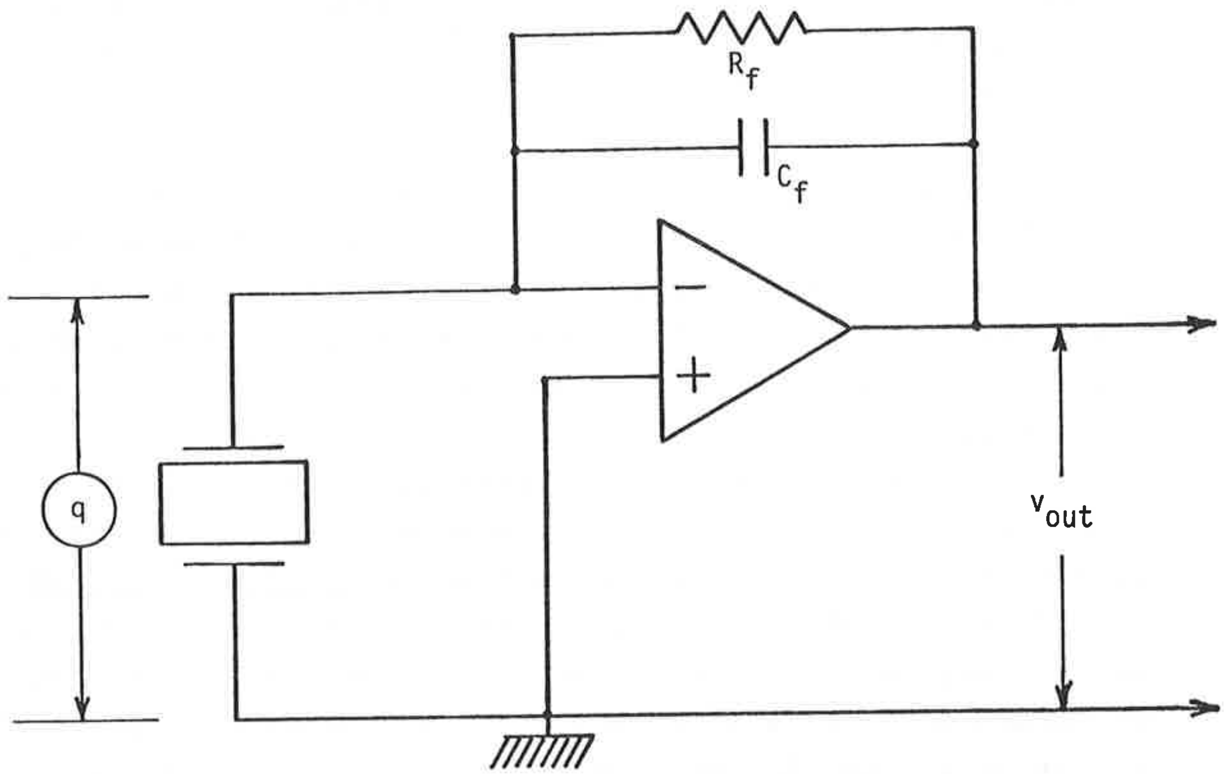
$\phi = \arctan f/f_o$

FIGURE D.1 SIMPLIFIED CIRCUIT DIAGRAM OF ACCELEROMETER AND UNITY GAIN VOLTAGE PREAMPLIFIER

The equivalent input noise voltage of a good quality accelerometer voltage preamplifier should be less than 3 microvolts for a bandwidth of 0 to 20,000 Hz and source capacitance of 500 pfd. In ground vibration measurements, the selection of accelerometer preamplifiers with very low noise characteristics is mandatory, but measurement of transit equipment shock or vibration does not usually require this.

Charge Amplifiers: Figure D.2 is a simplified circuit diagram of a charge amplifier. A charge amplifier produces an output voltage equivalent to the charge generated by a piezoelectric transducer. The feedback capacitor shown in Figure D.2 determines the charge sensitivity of the amplifier, and the feedback resistor in parallel with the feedback capacitor determines the low-frequency cutoff point. Many commercial charge amplifiers do not have feedback resistors, but use a reset switch to return the output voltage to zero prior to a measurement. These amplifiers should be avoided for general noise and vibration measurements, unless very low-frequency response characteristics are required. The transducer sensitivity and its low-frequency response characteristics are unaffected by the piezoelectric transducer capacitance or cable shunt capacitance. In this respect, a charge amplifier is more desirable than a voltage preamplifier, whose sensitivity and low-frequency response are affected by the transducer and cable capacitances. With a charge amplifier, the accelerometer can be calibrated with a short connecting cable and then installed in a measurement system with a much longer connecting cable. The only effect is increased electrical noise.

A charge amplifier has effectively zero input impedance, so that problems related to noise pickup and connector contamination are reduced. However, the equivalent input electrical noise of the charge amplifier is not necessarily greater or less than that associated with the voltage preamplifier given the same electronic component types. The signal-to-noise ratio is reduced as the effective source capacitance of the transducer and cable is increased as with longer cables. Cable noise, arising from



$$\left| v_{out} \right| = \frac{|q|/C_f}{[1 + (f_o/f)^2]^{1/2}}$$

ϕ = arctan f_o/f

f_o = $1/(2\pi R_f C_f)$

q = charge

FIGURE D.2 CHARGE AMPLIFIER TYPE ACCELEROMETER PREAMPLIFIER triboelectric effects, will also reduce the signal-to-noise ratio if relatively long transducer connecting cables are used. Cables should therefore be restrained and kept as short as possible. Low noise cables have been developed specifically for use with piezoelectric transducers.

A charge amplifier does not affect the dynamic characteristics of most piezoelectric transducers. However, when using a charge amplifier, the resonant frequency of lead-zirconate-titanate compression accelerometers is reduced by approximately 3% to 4%. All other effects are insignificant.

D.6 VOLTAGE AMPLIFIERS

Voltage amplifiers amplify low-level signal voltages to levels suitable for display or analysis. The input impedance of most voltage amplifiers varies, typically, from 100 kilohms to approximately 10 megohms, although input impedances outside this range are not uncommon. The bandwidth of a voltage amplifier will depend on the design and construction of the amplifier. Often, a bandwidth limiting filter is employed in the design to limit the response of the amplifier to a particular frequency range.

The output impedance of most modern solid state amplifiers is about 50 ohms. However, amplifiers usually have low output current capability: about 1 to 10 milliamps. The output current capability of any amplifier or preamplifier determines the peak output signal voltage for a given load, provided that the output voltage is within the maximum output voltage specified for the amplifier. Thus, if an amplifier is capable of 1 milliamp output of current, the peak output voltage into a 10 kilohm load will be 10 volts, while the peak output voltage with a 50 ohm load (such as a terminated cable) will be 50 millivolts.

The gain of a voltage amplifier refers to the ratio of the output voltage to the input voltage. Practical gains for most voltage amplifiers range from 0 dB to approximately 40 dB, and additional gain may be achieved by cascading amplifiers. Gain may be either fixed or variable and is determined by the feedback employed in the amplifier circuit. Practical signal level adjustment is usually accomplished with a step attenuator preceding the voltage amplifier stage.

D.7 ANALOG INTEGRATORS AND DIFFERENTIATORS

An acceleration signal may be integrated to obtain the vibration velocity, or conversely, vibration velocity may be differentiated to obtain the vibration acceleration. Analog integrators and differentiators can be used for these functions. The simplest type of analog integrator consists of a single RC network, with an RC time constant greater than the longest period of interest. Conversely, an analog differentiator may be constructed with an RC network, whose RC time constant is significantly lower than the shortest period of interest. The network is followed by a voltage amplifier to recover the signal.

Active analog integrators and differentiators are available which employ an operational amplifier with an appropriate feedback network to integrate or differentiate the signal. The advantage of an active analog integrator or differentiator is that the integrating network may be chosen to provide unity gain at a specific frequency. For example, given a signal voltage proportional to an acceleration signal in g's, an appropriately chosen integrator may be inserted to provide an output of voltage, proportional to the vibration velocity in meters per second. Such an integrator or differentiator may be inserted into a measurement system without recalibration.

Analog integration of acceleration signals to obtain an analog vibration velocity signal is the form of integration most likely to be required. However, some active integrators also provide a further stage of integration for obtaining vibration displacement. Analog differentiators are less likely to be needed in a typical measurement since most vibration measurements are performed with an accelerometer. In fact, use of differentiation is not advisable because of loss of accuracy, particularly at high frequencies. With integration, there is generally no loss of measurement accuracy.

Many active integrators do not have a built-in, low-frequency rolloff: they must be reset prior to each measurement. These integrators are not useful for most rapid transit measurements and should probably be avoided unless they have the option of selecting a low-frequency rolloff at about 1 Hz.

D.8 DETECTORS

Every measurement system requires a detector to convert the analog signal into a displayable form. Detectors may be classified as root mean square (rms), peak, absolute average, and quasi-rms. The output from these detectors may be an analog voltage proportional to the quantity being detected or a digital signal.

Most laboratory-quality noise and vibration instruments manufactured today employ "true" analog rms detectors that incorporate an exponentially weighted infinite averaging time. The effective averaging time is essentially the time constant of the detector, usually selected by the operator to correspond to "fast" or "slow" sound level meter response.

RMS detection with finite averaging or integration times is also available and may be in either analog or digital form. The digital rms detector may use a high speed analog-to-digital converter, with a sampling rate three or more times the maximum signal frequency of interest. Alternatively, a digital rms detector may rely on a statistical sampling technique in which the amplitude of the signal waveform is sampled at a rate significantly lower than the maximum signal frequency. This latter type is suitable for random noise and vibration data, but may not be appropriate for the analyses of transient events of durations comparable to or less than the sampling interval.

Peak detectors detect the greatest positive and/or negative amplitudes of a waveform. Peak detection is rarely used in noise and vibration measurements, although a good laboratory instrument is usually capable of such measurements. Peak detection is useful for transient or shock measurements.

Some instruments of older design and construction employ a so-called averaging detector, whose output is an analog voltage proportional to the average of the absolute value or "rectified average" of the input waveform. The averaging detector is essentially a rectifier circuit, calibrated to indicate the rms value of sine wave inputs only. These detectors are not accurate for rms measurement of complex periodic data with high crest factors or for general random vibration or noise. Their use should thus be avoided where critical measurements are concerned.

Some high-quality sound level meters and measuring instruments of older vintage use a quasi-rms detector which is essentially a more complex form of rectifier circuit than that used for an average detector. The quasi-rms detector is adequate to detect the rms amplitude of a wide class of signals.

D.9 DISPLAY AND READOUT DEVICES

The most common display device for noise and vibration data is the D'Arsonval meter, and more recently, the digital Panel meter. Some strip chart recorders produce continuous records of the amplitude or level of a noise or vibration signal as a function of time or frequency. The most practical strip chart recorder for the presentation of noise and vibration data is the graphic level recorder, described in Section D.2.

Other strip chart recorders simply display the instantaneous magnitude of applied voltage as a function of time. Such strip chart recorders are rarely, if ever, used to display noise and vibration waveforms because their writing speed is inherently limited. They may, however, prove useful for the display of various auxiliary measurement parameters, such as train speed, air pressure, or wind speed.

A very common and necessary display device is the oscilloscope, which displays the instantaneous magnitude of noise or vibration signals as a function of time and is particularly useful as a visual monitor during data analysis.

D.10 NARROWBAND ANALYZERS

The two basic types of nonreal-time narrowband analyzers are the constant percentage bandwidth analyzer and the constant bandwidth analyzer. The constant percentage bandwidth analyzer usually consists of a narrowband filter which is swept through the analysis range in synchronism with specially printed strip chart paper. The typical bandwidth of such a narrowband analyzer is approximately 1/10 octave and often may be adjustable to meet the particular needs of the measurement.

Constant bandwidth narrowband analyzers use a heterodyne technique, in conjunction with a very narrow bandpass filter, to achieve extremely fine spectral resolution. The analyzer frequency is controlled and swept through the frequency range by an external oscillator or control voltage. The spectral data obtained from such an analyzer are usually displayed with an X-Y recorder, with the frequency scale distributed linearly along the X axis, but may also be displayed with a log frequency scale on either an X-Y recorder or a graphic level recorder.

Both of these narrowband analyzers require steady-state data for meaningful analysis. These data are obtained by recording on magnetic tape and constructing a tape loop of the recorded sample for continuous reproduction.

D.11 REAL-TIME SPECTRAL ANALYZERS

Real-time analyzers perform very rapid spectral analysis and can generate large amounts of data in relatively short periods of time. The term "real-time" implies that the analysis may be performed directly, without using an intermediate tape loop.

For noise and vibration analysis, the 1/3 octave band real-time analyzer is the most common. This analyzer is essentially a constant percentage bandwidth analyzer with a bandwidth of 23% of the center frequency. One-third octave band spectra are relatively easy to interpret, and the 1/3 octave bandwidth corresponds closely with the bandwidth of moderately damped resonant systems. It also approximately corresponds with the human ear resolution of broadband noise over part of the frequency range. Greater resolution is rarely required in most noise and vibration problems.

A constant bandwidth real-time spectrum analyzer is useful for identifying harmonic components of noise or vibration, such as those produced by a gear train or transformer. There are two basic types of constant bandwidth real-time analyzers: the first and most common type "time compresses" the input signal and presents it to a narrowband filter of high resonant frequency in a repetitive fashion, with varying degrees of time compression. The output of the filter is then detected, with an absolute average or an rms detector, and displayed as a function of frequency. The second major type of constant bandwidth analyzer uses digital techniques and the fast Fourier transform (FFT) algorithm to determine the real and imaginary spectral components of the signal. The power spectral density is obtained from this type of analyzer by squaring and summing the real and imaginary parts to obtain the power spectrum.

Both of these devices analyze a sample only over a finite time interval, the length of which determines the resolution of the frequency analysis. The longer the time interval over which the analysis is performed, the greater the resolution. Both types of constant bandwidth analyzers can average a large number of successive samples to obtain a statistically significant result.

Most time compression analyzers use an absolute average detector and linear averaging of spectra to measure spectral densities. This is not technically correct and errors of 1 or 2 or even 3 dB must be expected. Indeed, there is some confusion and inconsistency between spectral densities measured with time compression analyzers and estimates based on 1/3 octave data from 1/3 octave analyzers using true rms detection. Some people argue that the type of detector, averaging or rms, is unimportant with the very narrow filters used in a time compression analyzer because the output of such a filter is essentially a sine wave. This is perhaps true if the filter coherence time is significantly longer than the averaging time of the detector. However, the filter output (signal amplitude) does vary with time in a random fashion, so that it is not strictly sinusoidal; if the detector's averaging time is longer than the coherence time of the filter, an absolute average detector can cause errors. A good design for a real-time spectrum analyzer that uses absolute average detectors should take these factors into account.

Some time compression analyzers use linear or arithmetic averaging of statistically independent spectral amplitudes to obtain an average spectrum. Proper computation, however, requires that the root mean square of the individual spectral amplitudes be taken or that their corresponding energies be averaged.

Many constant bandwidth analyzers provide options for calculating 1/3 octave band levels from the constant bandwidth spectra. However, this hybrid 1/3 octave band filter does not always conform to ANSI specifications because the lowest 1/3 octave bands will not be adequately resolved by the narrowband spectrum. For example, for a 400 line transform with a 10 kHz bandwidth, each spectral line represents 25 Hz. This can be used to calculate an accurate 1/3 octave band level at the high frequencies. However, the bandwidth of the 25 Hz 1/3 octave band is about 5 Hz, considerably less than the resolution of the constant bandwidth spectrum, which cannot be used to calculate a 1/3 octave level. A true 1/3 octave band real-time analyzer, however, can easily measure the 1/3 octave spectrum of the signal from 25 Hz or lower to as high as 20 kHz in one analysis.

If 1/3 octave band measurements are the primary concern, then a 1/3 octave band real-time analyzer should be chosen for this purpose rather than the hybrid system of a constant bandwidth analyzer with an option for constructing the 1/3 octave band spectra from the power spectral density. This is particularly important where low-frequency, groundborne vibration is concerned.

D.12 STATISTICAL DISTRIBUTION ANALYZERS FOR COMMUNITY NOISE AND VIBRATION LEVELS

Various statistical distribution analyzers for the statistical analysis of noise and vibration levels are available from major manufacturers. Most of these instruments are self-contained units, which calculate results and record data in the field. Alternatively, some consist of field data recording units and laboratory analysis units; the noise and vibration data are first digitized and recorded on magnetic tape for later reproduction and analysis in the laboratory by digital computers. The manufacturers of graphic level recorders may supply an attachment for statistical analysis of noise or vibration levels, using the level recorder. These latter units are relatively inexpensive means of obtaining statistical distribution data for community noise evaluations, but are cumbersome to use. Current technology field units, described above, can be used for short as well as long sampling periods, without requiring a level recorder or tape recording, and provide immediate results. Thus, their use is much less expensive than tape recording techniques.

D.13 OSCILLATORS

An oscillator that provides an undistorted and stable sinusoidal signal, whose frequency may be varied from approximately 10 Hz to 20 kHz, is of great value to any measurement system. An oscillator can often be interfaced with a graphic level recorder to measure and document the frequency response of the various measuring instruments used for noise and vibration measurements. This is particularly important in magnetic tape recorders, whose response characteristics can vary with time due to head wear and which vary with the type of magnetic tape used.

D.14 MAGNETIC TAPE RECORDERS

Magnetic tape recorders are essential for acquiring field noise and vibration data for later analysis in the laboratory. Tape recording noise and vibration data provide a permanent record, which may be easily filed for reference and reanalysis if necessary.

The most common tape recorders are full-track, single channel units or half-track, dual channel units as used widely in the motion picture industry. Some half-track machines have a third auxilliary channel between the two audio tracks, which can be used for various purposes. Less expensive tape recorders, such as those purchased for home recording systems, are usually quarter-track machines capable of recording in two directions, thus doubling the recording capacity of the tape for a given tape speed. However, home tape recorders usually lack the necessary portability required by noise and vibration data acquisition, and quarter-track machines have less signal-to-noise ratio than half-track or full-track machines.

Small, lightweight, battery-operated tape recorders can provide excellent recording and reproduction of noise and vibration data. Some of these recorders have been used in the motion picture industry and are designed for rugged field service and operation in extreme temperature environments. Many of these tape recorders provide the necessary power supply and connectors for a variety of laboratory microphones. Some of these tape recorders even have various weighting networks, such as A-weighting, so that they may be used as a sound level meter under certain conditions.

Portable cassette recorders have also been used to record noise and vibration data. Cassette recorder technology is rapidly changing, and some of the best recorders are suitable for some measurement purposes. However, cassette recorders are not recommended for measurements which require full frequency responses and high signal-to-noise ratios, which are normally obtainable only from the more expensive portable half-track or full-track standard format machines. With the improvements that are being incorporated into professional grade cassette recorders, they do represent a viable alternative to the expensive, instrumentation quality, reel-to-reel recorders.

Multichannel instrumentation tape recorders are available with four to fourteen channel configurations. Tape speeds for these recorders range from 15/16 in./sec to 120 in./sec. Either direct or frequency modulated carrier (FM) recording may be obtained. FM recorders allow recording of signals down to DC, with exceptionally flat frequency response.

Multichannel instrumentation recorders are useful where data from a large number of measurement positions must be recorded simultaneously or where auxiliary data, such as wind speed, train speed, or traction motive power, must be recorded simultaneously with noise and vibration data. Instrumentation recorders can be rented, but their frequency response characteristics should be closely checked prior to use. Selecting FM mode recorders will help to overcome some of the reliability and accuracy problems of rental units.

The record-reproduce response characteristics of the tape recorder affect the overall response characteristics of the measurement system. The variation of frequency response of the tape recorder makes it very difficult to fully meet the Type 1 sound level characteristics with recorded data. However, the better machines can be custom equalized to obtain a very well-controlled record-reproduce response of $\pm$ 0.5 dB from about 40 Hz to 12 kHz if necessary.

The typical specifications for a good magnetic tape recorder are outlined below:

- frequency response $\pm$ 1 dB from 25 to 20 kHz;
- wow and flutter, 0.1%;
- signal-to-noise ratio, 55 dB overall;
- crosstalk between channels, 50 dB;
- third harmonic distortion at maximum recording level, 1%.

Some very good recorders will, in fact, achieve even better performance characteristics. However, some tape recorders, although not fully satisfying the specifications, may nevertheless be satisfactory for noise and vibration measurements.

Equalization is employed by most audio tape recorders for the direct recording mode to enhance the signal-to-noise ratio of the record-reproduce process. Equalization normally emphasizes the high-frequency components of a signal during the recording and de-emphasizes the high frequencies during reproduction, thereby achieving an overall flat frequency response. De-emphasis during reproduction reduces the high-frequency "hiss" from the tape, thereby improving the signal-to-noise ratio. The equalization employed by most American-made tape recorders is the NAB equalization. Some of these recorders have a switch which allows the selection of either NAB or CCIR equalization. CCIR is used by most European manufacturers. Generally speaking, the NAB equalization involves somewhat greater high-frequency pre-emphasis and results in a slightly better signal-to-noise ratio than the CCIR equalization.

Equalization is not employed in multichannel instrumentation recorders meeting IRIG specifications, although some minor equalization is employed to correct for head loss characteristics and other minor effects. The result is that the instrumentation recorder usually has a lower signal-to-noise ratio than that obtained by the audio recorders which use NAB or CCIR equalization. The signal-to-noise ratio of the instrumentation recorder is usually 40 to 50 dB, depending on the tape speed and bandwidth, whereas good audio recorders achieve 55 to 65 dB signal-to-noise ratio.

FM recording is available on some audio and most instrumentation recorders. FM recording provides a very flat frequency response essentially from 0 Hz to about one-tenth of the maximum frequency capable of being recorded by direct means, given the same tape speed or from 0 Hz to about one-third the FM carrier frequency. Frequency modulation recording is particularly useful for recording vibration, especially ground vibration from transit trains, where the maximum spectral content of the vibration energy may be in the region of 10 to 60 Hz. Another attractive aspect of FM recording is that very little phase distortion is introduced in the record-reproduce process. Frequency modulation recording is thus particularly useful for recording transient events in which the actual waveform is of particular interest. Tape recorders used for frequency modulation recording must be of very high quality and exhibit very low levels of wow and flutter: less than 0.1% is necessary to guarantee adequate signal-to-noise ratio. Custom-manufactured, portable frequency modulators can be used for FM recording on standard audio tape recorders.

D.15 CALIBRATORS FOR NOISE AND VIBRATION TRANSDUCERS

There are two field techniques for overall measurement system calibration. One is the insert voltage technique in which a calibration voltage is inserted in the ground or return side of the transducer cable. With this technique, the electrical characteristics of the transducer along with those of the measurement system are included in the calibration. The disadvantages of this technique are that it requires an oscillator, computation of appropriate calibration voltages, interruption of the cable shield or signal return which can introduce extraneous noise and does not indicate any changes in the transducer sensitivity. The second technique, and by far the most practical, is direct calibration. The transducer is excited with a known sound pressure in the case of microphones or a known vibration amplitude in the case of an accelerometer. The excitation is provided with special, portable battery-operated calibrators.

Microphone Calibrators: The pistonphone is the most accurate of the portable, handheld calibrators. This unit uses a sealed cavity and an oscillating piston to apply a sinusoidal pressure to a microphone diaphragm at a known sound pressure level and frequency, e.g., at approximately 124 dB at 250 Hz. The actual sound pressure level generated depends upon the microphone size and its equivalent volume. The pistonphone is very stable and typically provides a known sound level within $\pm$ 0.1 dB, if appropriate corrections are made to account for microphone geometry and barometric pressure. A barometer, indicating the required sound pressure level correction, is normally supplied with the pistonphone. Calibration is effected simply by placing the pistonphone over the microphone end, sealing protective grid cap. Some microphones may be damaged by the pressure exerted against the diaphragm during the placing of the pistonphone, in which cases the pistonphone should not be used. Microphone manufacturers should be consulted about appropriate calibrating devices.

Acoustic calibrators generate an acoustic sound field through a piezoelectric transducer and electronic oscillator and are used in a fashion identical to that of the pistonphone. However, these calibrators do not use sealed cavities, so they are less likely to damage microphones through excessive pressure. Some acoustic calibrators allow calibration at a number of frequencies, although 1000 Hz is the most common. When calibrating at 1000 Hz or higher, the microphone free field response characteristic compared to pressure response must be known in order to determine its response characteristic at that frequency. Normally, the sound pressure level indicated for the acoustic calibrator, when used with a specific microphone model, takes the microphone response at the given calibration frequency into account. Acoustic calibrators do not require barometric corrections, so that a barometer is not necessary for calibration purposes.

Accelerometer Calibrators: Portable, battery-operated calibrators for accelerometer transducers consist of a shaker table and voice coil driven by internal or external electronics. The calibration acceleration is a sinusoid of either 1 g peak or 1 g rms at a frequency in the range of 50 to 100 Hz. These calibrators provide an adjustment for the accelerometer mass.

An accelerometer can be calibrated at a variety of frequencies by using an oscillator and comparing its output with a reference accelerometer of accurately known response characteristics.

This type of small, portable, battery-operated accelerometer calibrator is much more practical than the insert voltage calibration technique because the voltage sensitivity or charge sensitivity of the accelerometer need not be known and reference voltages need not be computed. When recorded, the calibration signal provides a convenient tone for calibration of the analysis equipment during data reproduction from magnetic tape. However, if the accelerometer transducer is in a position from which it may not easily be removed, or if its electrical characteristics must be checked after severe environmental stress, the insert voltage calibration technique is the most feasible.

The portable accelerometer calibrator may be used for the calibration of velocity and displacement transducers, provided that appropriate corrections for the frequency of the calibration are included.

APPENDIX E
REPORT OF NEW TECHNOLOGY

This handbook is a unique guide focusing on the prediction and control of all types of urban rail transit noise. For the first time, information on acceptability criteria, noise measurement, and control of noise from vehicles, surface, aerial, and subway tracks, stations, ancillary equipment, and yards and shops is assimilated in one document. In presenting various information related to noise and vibration within the transit environment, this aggregation of recent research in this area will significantly enhance future efforts in predicting and controlling noise and vibration.

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