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TCRP Report 23

Wheel/Rail Noise Control Manual

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Report 23

Wheel/Rail Noise Control Manual

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TRANSIT COOPERATIVE RESEARCH PROGRAM

The nation's growth and the need to meet mobility, environmental, and energy objectives place demands on public transit systems. Current systems, some of which are old and in need of upgrading, must expand service area, increase service frequency, and improve efficiency to serve these demands. Research is necessary to solve operating problems, to adapt appropriate new technologies from other industries, and to introduce innovations into the transit industry. The Transit Cooperative Research Program (TCRP) serves as one of the principal means by which the transit industry can develop innovative near-term solutions to meet demands placed on it.

The need for TCRP was originally identified in *TRB Special Report 213—Research for Public Transit: New Directions*, published in 1987 and based on a study sponsored by the Urban Mass Transportation Administration—now the Federal Transit Administration (FTA). A report by the American Public Transit Association (APTA), *Transportation 2000*, also recognized the need for local, problem-solving research. TCRP, modeled after the longstanding and successful National Cooperative Highway Research Program, undertakes research and other technical activities in response to the needs of transit service providers. The scope of TCRP includes a variety of transit research fields including planning, service configuration, equipment, facilities, operations, human resources, maintenance, policy, and administrative practices.

TCRP was established under FTA sponsorship in July 1992. Proposed by the U.S. Department of Transportation, TCRP was authorized as part of the Intermodal Surface Transportation Efficiency Act of 1991 (ISTEA). On May 13, 1992, a memorandum agreement outlining TCRP operating procedures was executed by the three cooperating organizations: FTA; the National Academy of Sciences, acting through the Transportation Research Board (TRB); and the Transit Development Corporation, Inc. (TDC), a nonprofit educational and research organization established by APTA. TDC is responsible for forming the independent governing board, designated as the TCRP Oversight and Project Selection (TOPS) Committee.

Research problem statements for TCRP are solicited periodically but may be submitted to TRB by anyone at any time. It is the responsibility of the TOPS Committee to formulate the research program by identifying the highest priority projects. As part of the evaluation, the TOPS Committee defines funding levels and expected products.

Once selected, each project is assigned to an expert panel, appointed by the Transportation Research Board. The panels prepare project statements (requests for proposals), select contractors, and provide technical guidance and counsel throughout the life of the project. The process for developing research problem statements and selecting research agencies has been used by TRB in managing cooperative research programs since 1962. As in other TRB activities, TCRP project panels serve voluntarily without compensation.

Because research cannot have the desired impact if products fail to reach the intended audience, special emphasis is placed on disseminating TCRP results to the intended end users of the research: transit agencies, service providers, and suppliers. TRB provides a series of research reports, syntheses of transit practice, and other supporting material developed by TCRP research. APTA will arrange for workshops, training aids, field visits, and other activities to ensure that results are implemented by urban and rural transit industry practitioners.

The TCRP provides a forum where transit agencies can cooperatively address common operational problems. The TCRP results support and complement other ongoing transit research and training programs.

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NOTICE

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The members of the technical advisory panel selected to monitor this project and to review this report were chosen for recognized scholarly competence and with due consideration for the balance of disciplines appropriate to the project. The opinions and conclusions expressed or implied are those of the research agency that performed the research, and while they have been accepted as appropriate by the technical panel, they are not necessarily those of the Transportation Research Board, the National Research Council, the Transit Development Corporation, or the Federal Transit Administration of the U.S. Department of Transportation.

Each report is reviewed and accepted for publication by the technical panel according to procedures established and monitored by the Transportation Research Board Executive Committee and the Governing Board of the National Research Council.

Special Notice

The Transportation Research Board, the National Research Council, the Transit Development Corporation, and the Federal Transit Administration (sponsor of the Transit Cooperative Research Program) do not endorse products or manufacturers. Trade or manufacturers' names appear herein solely because they are considered essential to the clarity and completeness of the project reporting.

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FOREWORD

*By Staff
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This manual will be of interest to engineers responsible for wheel/rail noise control in the design, construction, and operation of rail transit systems. It provides practical step-by-step procedures for identifying wheel/rail noise control technologies with demonstrated effectiveness. Procedures are included for identifying wheel/rail noise sources, developing mitigation designs, and estimating probable costs and effectiveness. The manual covers noise generated on tangent track, curved track, and special trackwork. Mitigation measures include onboard, track, and wayside treatments. Accompanying the manual is a user-friendly software package that assists in identifying appropriate noise mitigation techniques for various types of wheel/rail noise. The user is presented with several screens to navigate a decision tree until a set of possible mitigation options is reached. Several sound “clips” are included to assist the user in determining the type of noise that most closely resembles that which is to be controlled. The software package also provides several calculation worksheets to estimate life-cycle costs and expected noise attenuation for various mitigation measures.

In today’s climate of environmental consciousness, transit systems are being called upon to reduce noise, which previously was considered an intrinsic part of their operations. Wheel/rail noise generated at either sharp radius curves or on tangent track is considered objectionable, and transit agencies have implemented numerous mitigation techniques of varying effectiveness to reduce or control this noise. Documenting the successes and failures of these mitigation practices is useful to transit agencies and designers.

Under TCRP Project C-3, research was undertaken by Wilson, Ihrig & Associates, Inc., to assess existing wheel/rail noise-mitigation techniques, classify and evaluate them, and provide the transit industry with tools to select the most appropriate proven solutions for wheel/rail noise problems. To achieve the project objectives, the researchers conducted a comprehensive literature review of wheel/rail noise control practices; surveyed all North American and selected foreign heavy and light rail transit agencies to ascertain their current wheel/rail noise-mitigation techniques and their related experiences—both good and bad; compiled wheel/rail noise mitigation field test reports from transit agencies, product manufacturers, and suppliers; and field tested noise mitigation measures at several transit agencies. Based on these activities, this *Wheel/Rail Noise Control Manual* and accompanying software tool were developed.

An unpublished companion report, prepared under this project and entitled *Wheel/Rail Noise Control for Rail Transit Operations—Final Report*, provides a summary of the various tasks undertaken during the project and includes results of wheel/rail mitigation techniques field tested during the project. Field tests conducted during the project were used to assess the effectiveness of dry-stick lubricants (high positive friction [HPF] dry-stick friction modifiers and low coefficient of friction [LCF] flange lubricants) in controlling rail corrugation and wayside noise at tangent track and wheel squeal at curves in Los Angeles and Sacramento; and the effectiveness of rail vibration dampers in controlling wheel squeal at curves in Boston. The results of these field tests have been incorporated in this *Wheel/Rail Noise Control Manual*. The companion document is available on request through the TCRP, 2101 Constitution Avenue, N.W., Washington, D.C. 20418.

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Dr. James T. Nelson, Vice President of Wilson, Ihrig & Associates, Inc., served as Principal Investigator and Dr. George Paul Wilson, President of Wilson, Ihrig & Associates, Inc., served as Project Director. Dr. Allan Zaremski, President of Zeta-Tech, Inc., provided a detailed review of the literature, and contributed to the Manual with respect to rail lubrication, corrugation control, and grinding practices. Mr. David Coate of Acentech, Inc., provided material concerning sound barrier wall performance, and provided copies of technical reports prepared by Bolt Beranek and Newman Inc., for the U.S. Department of Transportation. Dr. Paul J. Remington of BBN Systems and Technologies, Inc., provided a detailed technical report summarizing wheel/rail rolling noise theory and recent advances thereof with respect to high speed rail. Dr. David Malam of W.S. Atkins, Ltd., in the United Kingdom provided a summary of foreign

literature and experience, and conducted a survey of several systems in the United Kingdom. Mr. Pablo A. Daroux of Wilson, Ihrig & Associates, Inc., designed and developed the software package for supporting the Wheel/Rail Noise Control Manual, including digitizing numerous audio samples of wheel/rail noise. Mr. Derek Watry, of Wilson, Ihrig & Associates, Inc., provided a model for economic analysis of noise control provisions. Mr. Frank Iacovino of Acentech, Inc., provided technical support in the measurement of wheel squeal noise reduction effectiveness of rail vibration dampers at the MBTA, for which Phoenix, USA, provided rail vibration dampers and hardware for testing. Valuable comments and suggestions were provided by members of various transit agencies and by the TCRP project C-3 Panel. The MBTA, Sacramento RTD, and LACMTA provided access to facilities for testing and evaluation. Finally, many substantial advances in wheel/rail noise control were made during the late 1970s and early 1980s under Federal programs administered by the Transportation Systems Center, now the Volpe Transportation Center in Cambridge, MA. Under this program, many individuals contributed valuable expertise and analysis which greatly benefited the current study.

CHAPTER 1 INTRODUCTION

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

This manual identifies mitigation procedures for wheel/rail noise produced by rail transit systems. The material incorporated herein is based on an extensive review of the literature and survey of rail transit systems.

The intended audience of this Manual includes transit system engineers responsible for design, construction, and operation of rail transit systems. The user of this Manual assumes all risks and responsibilities for selection, design, construction, and implementation of mitigation measures. No warranties are provided to the user, either expressed or implied. The data and discussions presented herein are for information only.

1.2 PURPOSE

This Manual provides the user with practical step-by-step procedures for controlling wheel/rail noise, based on proven technologies with demonstrated effectiveness. Procedures are included for identifying wheel/rail noise sources, developing mitigation designs, and estimating probable costs.

1.3 SCOPE

The scope of the Manual is wheel/rail noise generated by the interaction of wheel and rail, and radiated by the wheel and rail to the vehicle interior and wayside. Structure radiated and groundborne noise are not specifically included, though noise from aerial or steel elevated structures is discussed primarily from the standpoint of trackwork design. Groundborne noise is not considered, even though its genesis involves wheel/rail interaction. Groundborne noise and vibration is discussed in the *Handbook of Urban Rail Noise and Vibration Control (1)* and in the *State-of-the-Art Review: Prediction and Control of Groundborne Noise and Vibration from Rail Transit Trains (2)*.

1.4 MANUAL ORGANIZATION

Chapter 2 summarizes the fundamentals of acoustics that apply to wheel/rail noise control. The reader's background is

assumed to be that of a mechanical or civil engineer who is capable of understanding and applying concepts in dynamics, mechanical vibration, and acoustics. Concepts and nomenclature familiar to the noise control engineer working in transit noise control and transportation noise impact analysis are presented and discussed. The reader is referred to the literature for detailed discussions.

Guidelines and goals for wayside and interior noise levels are discussed in Chapter 3, drawing heavily on the American Public Transit Association (APTA) *Guidelines for Rail Transit System Design (3)*, and on the Federal Transit Administration (FTA) *Transit Noise and Vibration Impact Assessment (4)*.

Chapter 4 discusses the theory of wheel/rail noise generation, including the most recent models of rolling noise that have been proposed or investigated with respect to high-speed rail as well as conventional transit. Wheel squeal noise generation is discussed separately of rolling noise, as the processes are dissimilar, involving noise generation at curved track. The principal purpose of this chapter is to provide the reader a basis for identifying types of noise and the potential effectiveness of various noise mitigation procedures. The wheel/rail noise generating mechanisms are several in number; they include random rail and wheel roughness, rail corrugation, stick-slip generated squeal and howl, and impact noise. Identification of the nature of the type of noise is critical in identifying appropriate noise control measures. Included are discussions of normal or optimally low noise levels for vehicles and equipment in good running condition.

Chapter 5 is perhaps the principal chapter of the Manual, where the user is led through a step-by-step process of identifying appropriate noise mitigation provisions. The Manual stops short of recommending a specific treatment, providing only noise reduction estimates and probable costs. Treatments are categorized in terms of tangent track, curved track, and special trackwork. The treatments are further categorized for onboard, trackwork, and wayside application. This chapter is supported by a computer program that provides aural examples of various types of noise, algorithms of computing noise reductions, and a cost model.

Chapter 6 provides a methodology for assessing the costs of various mitigation options. Cost comparisons are made on the basis of the annuity cost, taking into account the present

value, and discounted future fixed and periodic costs. The methods of this chapter are supported by the accompanying software package.

Chapters 7, 8, and 9 present detailed discussions of various treatment designs for onboard, trackwork, and wayside application, respectively. The chapter concerning onboard treatments includes material on vehicle interior noise control, and the chapter on wayside application includes sections on subway station and vent and fan shaft noise control.

Chapter 10 discusses rail corrugation and possible methods for control. Rail corrugation is perhaps the most significant cause of community reaction and complaints concerning rolling noise, due to the particularly raucous nature and high level of the sound. No discussion of wheel/rail noise control would be complete without including rail corrugation generation and control. The discussion provided here is intended to acquaint the user with certain theoretical aspects and observations concerning rail corrugation, together with possible methods for control, of which the only reliable treatment identified thus far is aggressive rail grinding. However, this chapter does not provide solutions to the rail corrugation problem.

Each chapter contains its own table of contents and reference list, so that each chapter is reasonably self-contained.

1.5 ANNOTATED BIBLIOGRAPHY

The Manual is supported by an annotated bibliography assembled as part of the literature review. The annotated bibliography attempts to include all references to published documents, and is implemented in a bibliographic software package for updating and generation of key word lists.

1.6 SOFTWARE SUPPORT

A software package is provided with the Manual. The software was developed with Microsoft, Inc.'s Visual Basic as an application under Microsoft, Inc.'s Windows® 3.1 operating system. The software should run properly on Microsoft, Inc.'s Windows 95 operating system.

The elements of the software package include the following:

- Audio samples of wheel/rail noise, including rolling noise, corrugated rail noise, wheel flat noise, and wheel squeal.
- A decision tree to help the user select noise control treatments.
- A cost analysis algorithm to estimate the annuity cost of a treatment, given fixed and periodic costs.
- Noise reduction routine for sound barriers.
- A “help” facility which provides references to the Manual, including text imported from the Manual.

The software package is intended to be user-friendly, providing the user a means of rapidly identifying appropriate mit-

igation techniques. However, the user should refer to the Manual for detailed discussion of each noise control treatment. Further, manufacturers' costs, specifications, and performance data should be reviewed carefully prior to selection.

1.7 WHEEL/RAIL NOISE CONTROL SOFTWARE

The purpose of this software package is to aid the user in determining the cause of abnormal noise levels in the transit system and identify mitigation options. The user is presented with several screens to navigate a decision tree until a set of possible mitigation options are reached. Several “sound clips” and two “sound demo” screens are available to help the user determine the type of noise that most closely resembles that which is to be controlled.

The software package contains three Calculation Worksheets, i.e., spreadsheet-like screens to

1. Estimate the expected attenuation of rail transit vehicle noise by sound barrier walls,
2. Predict L_{dn} and CNEL levels at any nearby receiver due to a typical light rail train, and
3. Calculate the Equivalent Uniform Annual Cost of mitigation options so that the true cost of mitigation alternatives over their lifetime can be compared.

The software has a comprehensive set of “help screens” which can be activated at any time by pressing the Windows® help key (F1). The help screens provide mostly operational information to assist the user in the basics of the software. The user can also search the help system for key words by selecting the “Help” menu option on top of the screen and then “Search for help on...”

In addition, the text and tables of most of the Wheel/Rail Noise Control Manual have been incorporated in the help file provided with the software. The text has been enriched by the use of hypertext “links,” shown as underlined words in green, which can be “clicked” to obtain definitions of technical terms from the Glossary or used to navigate to other related topics. The help window also contains a series of buttons on its top row which allow immediate access to the Glossary of terms or the Contents page, and navigation through a Chapter of the Manual (“< <” and “> >” browsing keys).

System Requirements

The following are system requirements:

1. 386 or better Processor (486-33 or better recommended)
2. Microsoft Windows® 3.1 or higher
3. VGA video or better (640×480 pixel resolution minimum). The screens have been optimized for the

“lowest common denominator” resolution of 640×480, and that is, therefore, the recommended resolution to which Windows should be set. Please see your system administrator to adjust your screen to this resolution.

4. 5.5 Megabytes of free hard disk space.
5. 4 Megabytes of RAM memory (8 recommended).

In addition, the following is recommended:

6. Windows compatible sound system card and loudspeakers (only necessary to play back sound clips).

Installation

Insert disk #1 in the 3¹/₂" diskette drive of your computer. From the Windows File Manager screen select:

File | **R**un

then type:

A:SETUP <enter> (type B:SETUP if your 3¹/₂" drive is B:)

The software will be installed by default in the directory “C:\TCRPC3”. However, the setup program will allow installation in any other directory or drive.

Follow the instructions on the screen to swap diskettes when needed.

The setup program will create a new program group and insert the program icon for the software (a picture of a wheel on a rail emanating sound waves).

To run the software, double-click on the icon.

Operation

Once the software is run by double-clicking on the icon, the user will be presented with a Copyright banner screen which will remain on for about 5 sec. After that, the main screen will appear and the user will be asked to select one of the three noise classes: Tangent Track noise, Curved Track noise, and noise associated with Special Trackwork such as frogs. Most of the screens have an “information line” at the bottom providing further information about the current screen or option.

By single-clicking on one of the rectangular buttons on the screen current at the time, the user will move further down the decision tree until a final screen displaying the section of text of Chapter 5 of the Wheel/Rail Noise Control Manual pertinent to the noise mitigation option selected by the user

is displayed. If there is a sample calculation screen for that particular mitigation option, a button will appear which, when “clicked” will display the appropriate worksheet.

Also, if the user selects Sound Barriers, Absorptive Sound Barriers or Earth Berms as mitigation options, another button will become available which leads to a Sound Barrier Insertion Loss worksheet. The computations there conform to the design guidelines for sound barrier attenuation described in the *Handbook of Urban Rail Noise and Vibration Control (1)*.

To go back up the decision tree, the user can single-click the left pointing arrow at the bottom left of each screen or press the “Escape” key, usually located at the top-left corner of most keyboards.

To start again at the top of the decision tree, select the menu option **F**ile | **S**tart Again.

To adjust the playback volume of the sound clips, use the utility program provided with your sound board. This is usually a “mixer” utility that adjusts the playback level of Windows programs.

To **END** the run, select **F**ile | **E**xit from the menu bar at the upper left corner of the screen.

For questions or difficulties installing the software, call Pablo Daroux at (510) 658–6719 between the hours of 9 a.m. and 5 p.m. PST.

The software is a Microsoft Windows® application; therefore a personal computer running Windows 3.1 or Windows 95 is required. The code was written by Pablo Daroux at Wilson, Ihrig & Associates using Microsoft Visual Basic version 4.0 on a 486–66 personal computer. It has been tested in machines running Windows 3.1, 3.11, Windows for Workgroups 3.11, and NT Workstation Version 3.51. CPUs tested under include 486DX2/50, 486DX2/66, Pentium 90 and Pentium Pro 200 series with no compatibility problems.

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CHAPTER 2

FUNDAMENTALS OF ACOUSTICS

2.1 INTRODUCTION

This chapter discusses certain fundamental aspects of acoustics and noise to acquaint the user with basic concepts and terminology that may be otherwise unfamiliar. The fundamentals are covered in both technical and descriptive terms. The following topics are covered:

- Physics of sound,
- Parameters that are used to measure and characterize acoustic wave phenomena,
- Descriptors that are used to characterize environmental noise exposure, and
- The basic theory of sound absorption and sound isolation.

A number of reference books are available that provide an introduction to the field for the nontechnically oriented reader, along with more detailed information for engineers. In addition to the references cited throughout this manual, the following are recommended:

- The *Handbook of Noise Control (1)*: This is a general reference on noise control. The Handbook contains 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.
- Beranek, *Noise and Vibration Control (2)*: This widely used text provides substantial detail and is sufficiently mathematically oriented to allow evaluation of parametric effects. Although a valuable source for acoustical engineers, it is probably not appropriate for those who do not have backgrounds in engineering or physics.
- Rettinger, *Acoustic Design and Noise Control (3)*: This book (Volume 1, Acoustic Design and Volume 2, Noise Control) presents a wealth of practical information on the design of noise control features.
- Kinsler and Frey, *Fundamentals of Acoustics (4)*: This widely used text is standard for undergraduate acoustics courses and is, therefore, readily available. The text provides a thorough discussion of the theory of acoustics and environmental noise, though the material tends to be very theoretical for direct application to noise control problems.

- “Transportation Noise and Its Control” (5): This pamphlet, prepared by the U.S.DOT, describes transportation noise (aircraft, highway, and rapid transit) to the general public. It also includes a general description of the physics of sound.
- Lutz, *Theory and Practical Application of Noise and Vibration Abatement for Railway Vehicles (6)*: This monograph focuses on noise abatement procedures that can be applied to diesel locomotives, but most of the information is sufficiently general to also be applied to transit system noise control.
- *Handbook of Urban Rail Noise and Vibration Control (7)*: This document, prepared for the Urban Mass Transit Administration (UMTA), provides detailed design guidelines for both noise and vibration produced by urban rail transit systems.

This list of references is meant to be representative and is by no means exhaustive.

2.2 PHYSICS OF SOUND

Sound is a fluctuating disturbance of the air (or other gas) caused by propagating pressure waves. Sound travels through air in the form of small waves similar to the way circular waves created by a tossed stone spread on the surface of a pond. The most common source of disturbance is a vibrating object, such as a tuning fork. The vibrating object disturbs the air molecules by alternately causing compression (squeezing together) and rarefaction (pulling apart) of the air molecules. The compression and rarefaction result in a pressure wave that travels (propagates) away from the vibrating object at a constant speed. As for waves on the surface of a quiet pond, there is no net transfer of matter by the wave when averaged over time.

2.2.1 Amplitude and Spectra

Sound is characterized by the amplitude and the frequency of the pressure fluctuations. Typically, sound will contain many different frequency components, and together form the spectrum of the sound. There are some sounds that consist of only one frequency; the sound produced by a tuning fork is

an example of a single discrete frequency sound. The sound of a train passby consists of a wide frequency spectrum which is smooth, without discrete frequency components.

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° C increase in temperature.

2.2.2 Geometric Spreading

By considering a stone thrown into a pond and the resulting circular surface wave, several salient features of noise generation and noise control can be visualized. However, the pond wave is a *surface wave* radiating in two dimensions whereas sound is a *body wave* radiating in three dimensions. Consider first the size of the stone thrown into the pond. Clearly, the larger the stone, the larger the resulting wave will be. In a noise scenario, the “size of the stone” corresponds to the amplitude of the vibrating object which, in turn, determines the amplitude of the pressure wave. The pressure wave amplitude is related to the perceived *loudness* of the sound. Hence, the loudness of unwanted noise can be reduced by limiting (damping) the vibration amplitude of the object creating the sound, i.e., by throwing a smaller stone. “Large stones” in rail transit include things such as jointed rails, wheel flats, and rough rails.

Imagine for the moment that the stone hits the water in the middle of a very large pond (so that the wave can travel a long way before it encounters the shore). As the circular wave propagates outward, the amplitude decreases for two reasons, both of which involve the amount of *energy* in the wave.

Every wave contains a certain amount of energy which is distributed throughout the wave. Using the water wave analogue, the energy per unit surface area of the wave is called the *energy density*, and the energy density is related to the amplitude of the wave. The energy consists of roughly equal parts of kinetic and potential energy when averaged over a cycle in time, or period. As the circular wavefront in the pond becomes larger, the energy is spread out over a greater area, or volume, and, hence, the amplitude of the wave decreases. This is called *geometric attenuation*. With geometric attenuation, the energy of the wave is spread over an increasingly large area as the wave propagates away from the source, but the total amount of energy in the wave remains constant in the absence of absorption or damping. If geometric attenuation were the only mechanism by which the wave amplitude decreased, the wavefront would eventually reach the shore, although its height might be very small by the time it got there.

2.2.3 Atmospheric Absorption

Friction is another mechanism which decreases the wave amplitude by actually decreasing the amount of energy in the

wave. The internal friction of a fluid is measured by its *viscosity*. Most people think of honey or oil when they hear the word viscosity because these are common examples of viscous (i.e., high internal friction) fluids. However, water and air also have viscosities. In other words, they have internal friction. When a wave passes through a fluid, the internal friction of that fluid converts some of the mechanical wave energy to heat energy. This reduction in wave energy can be significant. Returning to the analogy of a circular wave on a pond, the absorption of high-frequency components will cause an initially “rough” wave to become smooth as it spreads. Ultimately, the viscosity of the pond water could prevent the wavefront from reaching the shore by dissipating all of the wave energy before it arrived.

To review the previously discussed attenuation mechanisms, consider them in the context of rail transit passby noise. As the train passes, acoustic waves which compose the passby noise propagate away from the train. As they spread, the energy contained within them is distributed over a large area, which might be visualized as an imaginary cylinder with its center along the track. The spreading reduces the energy density of the noise, manifested as a decrease in loudness. At the same time, some of the acoustic wave energy is being converted into heat energy by the viscosity of the air, especially at the higher frequencies. This will further decrease the loudness of the noise and, more noticeably, will alter the character of the passby noise, similar to turning down the treble adjustment on a car radio. Although both of these mechanisms are effective in reducing the amplitude (loudness) of a propagating acoustic wave, the distance required to attenuate train passby noise to an acceptable level is often much further than the distance to the nearest affected receptor. Therefore, other means of controlling train noise may be necessary.

2.2.4 Diffraction Due to Barriers

Harbors and marinas often have walls or piles of boulders placed at their entrances to protect them from the undulating force of incoming waves. Likewise, sound barrier walls are often used to reduce loudness in the space behind them by reflecting sound waves which impinge upon them (assuming they are sufficiently massive). Walls are not, however, completely effective at blocking acoustic pressure waves because of a phenomena called *diffraction*.

As everyone who routinely calls out to people in adjacent rooms knows, sound waves can travel around corners, up stairs, and down hallways, because acoustic pressure waves can be both reflected and diffracted. Diffraction is the process by which the direction of a sound wave “bends” around a corner or over the top of a wall. Surface waves on a pond diffract around the hulls of ships or breakwaters. Diffraction is frequency dependent; low-frequency waves bend around corners much more readily than high-frequency waves. As with the energy absorption mechanism introduced above, this changes the character of broadband noise which

is diffracted by disproportionately reducing the high-frequency components of the sound, or “treble.”

2.2.5 Sound Absorption

Sound barrier walls made of concrete or brick reflect practically all of the energy contained in the acoustic waves which strike them directly. However, these reflected waves may still find their way over the wall by being subsequently reflected off the train car and then diffracted. To prevent this, absorptive material may be fixed to the surface of the wall. The absorbing mechanism is due to friction between moving air and the loose fibers or pore walls in the sound absorbing treatment. The friction between the air and porous or fibrous absorptive material converts the acoustic energy to heat, thereby attenuating the sound.

2.2.6 Wind and Temperature Gradients

When sound propagation outdoors is considered, there are the additional complications of refraction caused by wind or thermal gradients and excess attenuation caused by rain, fog, snow, and atmospheric absorption. (Refraction is the phenomenon by which the direction of propagation of a sound wave is changed due to spatial variation in the speed of sound.) These environmental effects on sound propagation are discussed briefly below.

Figure 2–1 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, its path is refracted (bent) upwards, and when it is propagating downwind, its path 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 that can cause large deviations from the expected geometric attenuation of sound.

Temperature gradients will cause refractions in a manner similar to wind gradients. Under normal atmospheric conditions, a clear afternoon with air temperature decreasing with increasing altitude, the sound waves will be refracted upwards. However, when the air temperature increases with elevation, the sound is refracted downwards. This condition is called an *inversion* and may occur just after dusk, persisting throughout the night and into the following morning. Under such conditions, strong noise level enhancements can occur. The effect of a temperature inversion layer is shown in Figure 2–1.

Although it is commonly said that sound carries well on days of fog or light precipitation, the evidence indicates that this is due to a lack of strong thermal and wind gradients 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 because of 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 by acting as a sound absorption treatment, producing a so-called “ground effect.” Ground effects may also be produced by grass-covered surfaces or other soft ground cover, and are usually considered in noise predictions for relatively large distances. For typical rail transit noise, ground effects are not normally considered, because of the relatively narrow noise impact corridors associated with rail transit noise. However, this is not a rule, as certain situations may dictate that ground effects be considered, especially where noise reductions at large distances from the track are considered. In such cases, introduction of a sound barrier wall may effectively elevate the source height, thus reducing the ground effect and circumventing, perhaps partially, the noise reduction effectiveness of the wall.

2.3 PARAMETERS USED TO CHARACTERIZE SOUND

In this section, some basic mathematical aspects of sound waves are described to explain how sound can be measured and analyzed with scientific instruments to yield meaningful design data. Initially, the simplest type of waveform, the sinusoidal wave, is considered. An understanding of this simple wave form is of fundamental importance because even the most complex waveforms encountered in the real world can be thought of, and analyzed, as being composed of a large number of these simple waves.

2.3.1 Sinusoidal Waves

Figure 2–2 illustrates the instantaneous amplitude as a function of time of a sinusoidal wave (sine wave), a wave or motion consisting of a single frequency. A single-frequency sound wave is generally referred to as a “pure tone.” Figure 2–2 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 an undamped 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.
- Wheel squeal produced at curves

The oscillatory motion of the wave in Figure 2–2 is completely described by the frequency f , the amplitude, A , and the initial phase angle. In mathematical terms, the amplitude, $p(t)$ is given by

$$p(t) = A \cos(2\pi f t + \phi)$$

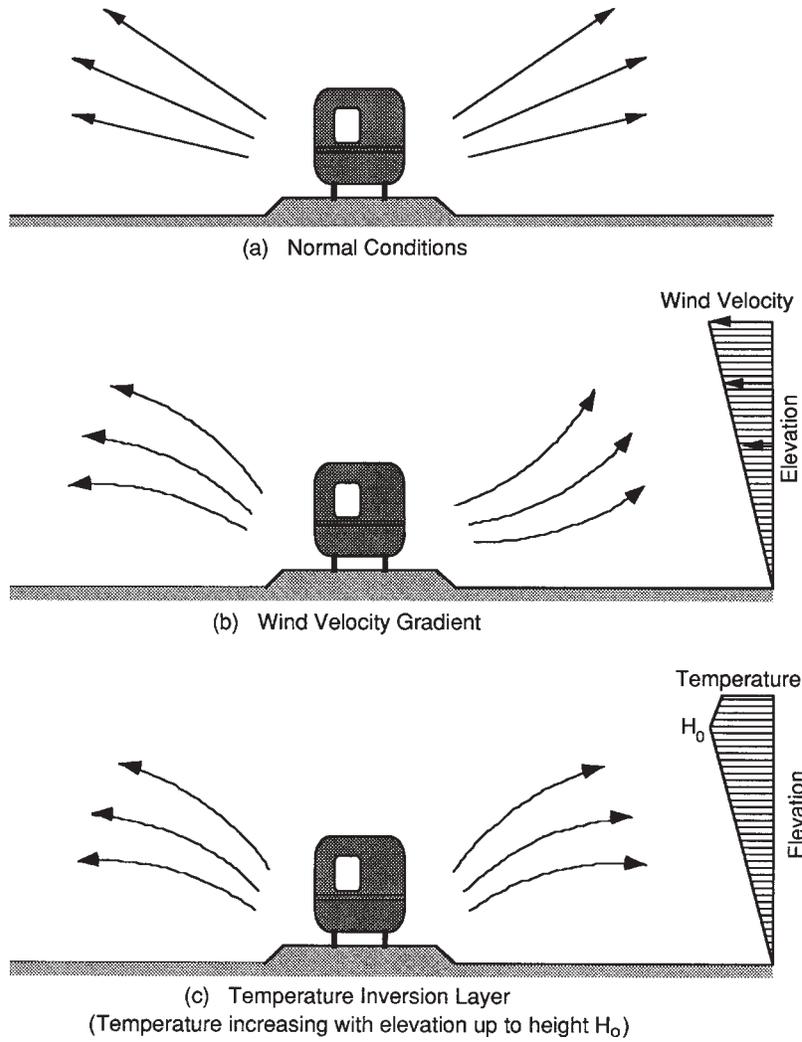


FIGURE 2-1 ATMOSPHERIC EFFECTS ON SOUND PROPAGATION

The frequency is the number of cycles of the motion that occur in 1 sec and, for sound waves, is related to its pitch. Frequency is denoted in Hertz (Hz), where one Hertz is defined as one cycle per second.

Widely used descriptors of simple, oscillatory waves are:

Period (for one cycle) = $T = 1/f$

Peak-to-peak amplitude = $2A$

Root-mean-square amplitude =

$$p_{\text{rms}} = \sqrt{\frac{1}{T} \int_0^T p^2(t) dt}$$

The integration time, T , is an important parameter, often defined as 1 sec (“slow meter response”), or 0.1 sec (“fast meter response”). The slow- and fast-meter responses, or rms detector time constants, are employed in sound level meters to determine the rms sound level as a function of time. How-

ever, the time, T , can be set to any length, such as an hour, day, year, or simply the period, $1/f$, of the wave. The discussion of any rms amplitude should include a specification of the “integration time,” or sound-level meter response characteristic, used to arrive at the amplitude.

In the limit as T approaches infinity, the rms amplitude of a sinusoidal sound pressure wave of amplitude, A , is:

$$p_{\text{rms}} = A/\sqrt{2} = 0.707A$$

Thus, the *mean-square pressure*, $\langle p^2 \rangle$, is simply $1/2$ the square of the amplitude, A^2 .

Fortunately, the condition that T approach infinity for the calculation of the root-mean-square amplitude and the rectified average can be approximated by a relatively short sample time. As a worst case example, the rms amplitude of a 20 Hz sine wave is within one percent of its theoretical limit after 0.2 sec. The rms amplitudes of higher frequency waves converge even faster. Thus, sound level meters

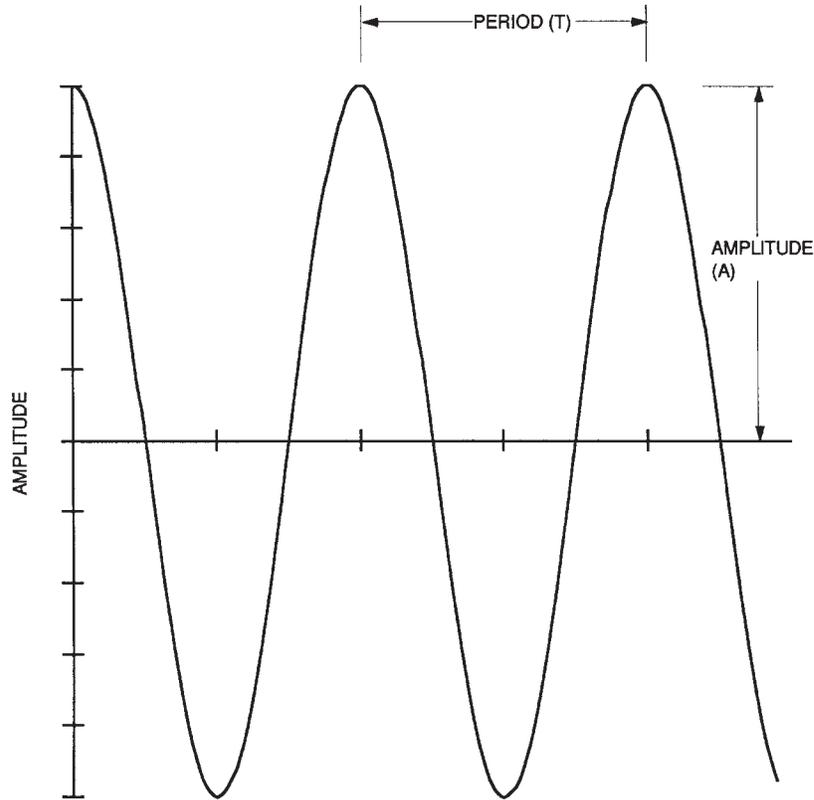


FIGURE 2-2 SINGLE FREQUENCY WAVEFORM

typically use averaging times on the order of 0.1 sec or 1 sec, corresponding to the fast- and slow-meter responses, respectively.

2.3.2 Superposition of Two Sinusoidal Waves

The next level of complexity in wave analysis is the summation of two sinusoidal waves, as illustrated in Figure 2-3. For the special case where two waves have the same frequency, the result will be a third wave oscillating at the given frequency with an amplitude and phase angle dependent on the amplitudes and phase angles of the constituent waves. If the amplitudes are similar and the phase angles are close, there will be constructive interference and the resulting amplitude will be approximately double that of the constituents. On the other hand, if the phase angles are roughly 180 degrees apart, there will be destructive interference and the resulting amplitude will be small. In any case, the rms amplitude is determined in the same manner as described above for a sine wave.

When the frequencies of the two waves are different, the rms value of the combined signal is given by the formula

$$P_{\text{rms}} = \sqrt{P_{1\text{rms}}^2 + P_{2\text{rms}}^2}$$

Theoretically, the above formula is exact only as the integration time period considered approaches infinity. However,

the rms amplitude of the combined signal is within 1% of this limit for averaging time greater than $50/\Delta f$, with the absolute difference of the two frequencies, Δf , given in Hertz. This is the time necessary to account for the effect of alternating constructive and destructive interference which occurs as the relative phase between the two waves slowly varies. As an example, if two noise sources differ in frequency by 1 Hz, such as fans running at 120 Hz and 121 Hz, approximately 1 min of the resulting sound would need to be analyzed before the rms converged to the above limit. If the fans were operating at 120 Hz and 180 Hz, less than 1 sec would be required. These results are primarily relevant to the combination of sound sources with strong tonal components such as fans and other machinery.

2.3.3 Random Noise

Most noise is comprised not simply of several tonal components, but very complex waveforms which have continuous frequency distributions. Such sounds are often called "broadband," if the frequency distribution covers a wide range of frequencies. Figure 2-4 is an illustration of broadband random noise such as that of a waterfall or wheel/rail noise. The term "random" indicates that the magnitude of the noise cannot be precisely predicted for any instant of time. The rms value, usually determined by measurement, is the

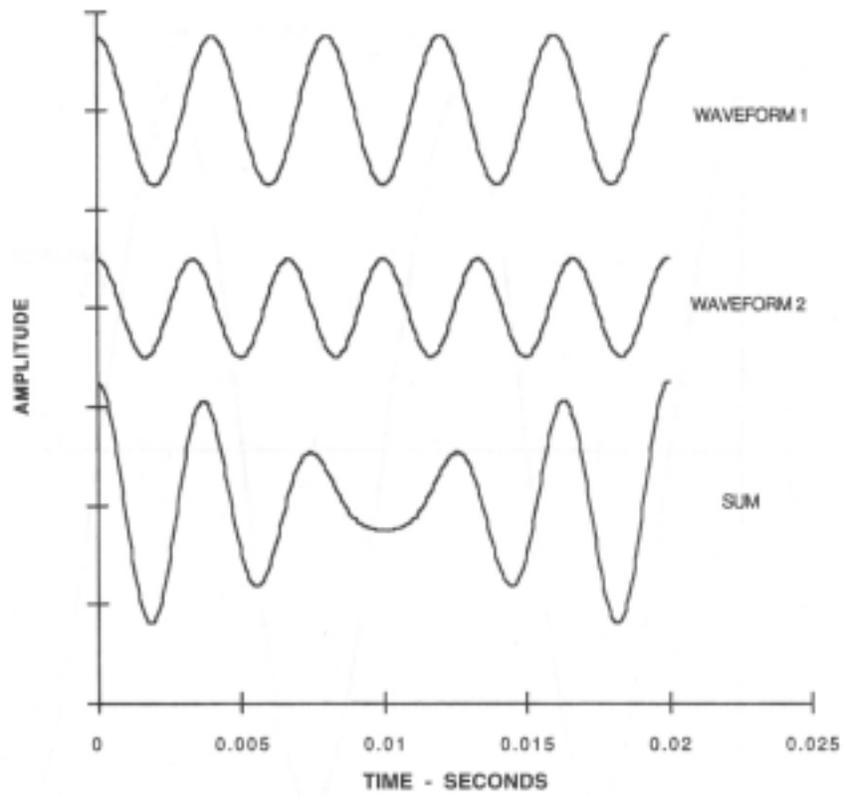


FIGURE 2-3 SUPERPOSITION OF TWO SINE WAVES OF FREQUENCY 250 HZ (WAVEFORM 1) AND 300 HZ (WAVEFORM 2)

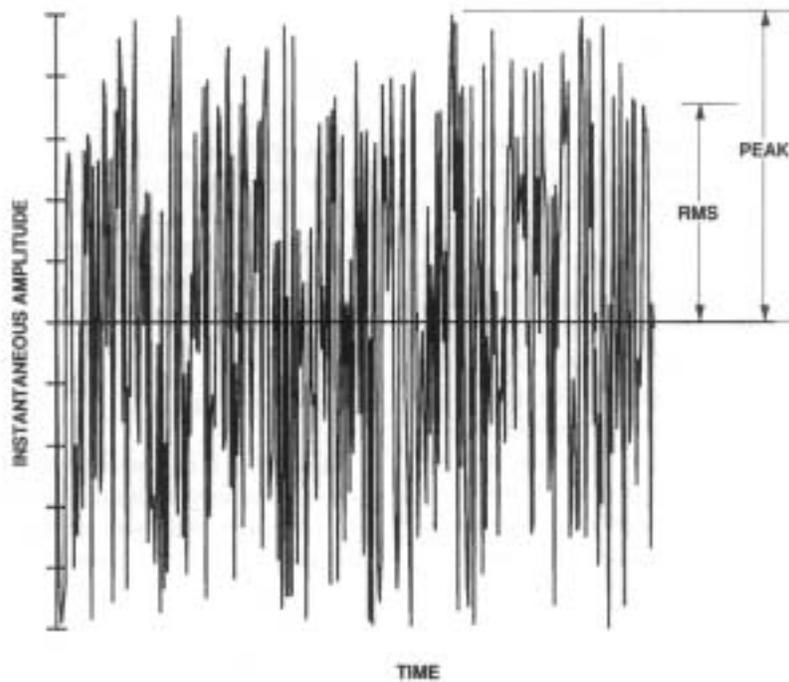


FIGURE 2-4 RANDOM WAVEFORM EXAMPLE

most common descriptor used for the amplitude of random, complex signals. The rms amplitude is the preferred metric for two reasons. First, the square of the rms amplitude of a sound is a measure of the sound energy. Second, the rms amplitude of a sound or vibration is well correlated with human response. The peak level is often an instantaneous event that occurs so rapidly the human perception mechanisms cannot respond.

As was previously mentioned, broadband sounds are composed of infinitely many simple sinusoidal waves. Thus, by extension of the formula for the rms amplitude of combined sine waves, the rms amplitude of combined broadband sounds is given by the above formula also. In the case of broadband signals, there is so much constructive and destructive interference that the opposing effects tend to cancel each other and the above expression is correct for even short periods of time (fractions of a second).

2.3.4 Decibel Levels

The human ear is capable of responding to a very wide range of sound pressures; at the threshold of pain the sound pressure is roughly 1 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.

The general definition of the decibel is

$$L_w = 10 \text{ Log}_{10}(W/W_{\text{ref}})$$

where W is a quantity proportional to power, W_{ref} is a reference power, and L_w is the level in decibels, abbreviated as “dB.” 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.

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. The definition of Sound Pressure Level (SPL) is

$$L_p = \text{SPL} = 20 \log_{10}(p/p_0) = 10 \text{ Log}_{10}(p^2/p_0^2)$$

where p is the rms pressure for the sound in question. The square of the sound pressure, p^2 , is used because it is proportional to power, and decibels are typically used to indicate the ratio of two values of “power-like” quantities. The factor 20 is used when expressing the sound in terms of its pressure, or amplitude. The rms amplitude is employed most commonly. One may also employ a peak, a zero-to-peak, or a peak-to-peak amplitude. However, it is not reasonable to use the deci-

bel scale to describe the instantaneous time varying amplitude of a wave, since it crosses through “0”, and may be negative, for which the amplitude would not be defined. When using the decibel scale, the user is implying that the signal has been “detected” in some manner, such as by a sound-level meter.

Table 2–1 presents standard reference quantities used for noise and vibration measurements. For example, the reference sound pressure is 20 micro-Pascal, and the reference sound power is 10^{-12} Watt. The reference quantity for vibration is less clearly defined, but it is customary to employ 1 micro-g and 1 micro-in./sec for acceleration and vibration velocity, respectively, where “g” is the earth’s gravitational acceleration.

Since the decibel scale is logarithmic, the levels for two sounds are not added arithmetically. When adding two signals, their energies are first obtained and then added, and the resulting level is ten times the logarithm (base ten) of the energy sum. For example, if the power of one signal is 6×10^{-7} watts, and the power of the second signal is 7×10^{-7} watts, combining signals 1 and 2 creates a new signal with a power of 13×10^{-7} watts. The power *level* of the combined signals (with a reference level of 10^{-12} watts) is

$$\begin{aligned} LW = PWL &= 10 \text{ Log}_{10} [(6 \times 10^{-7} + 7 \times 10^{-7})/10^{-12}] \\ &= 61.1 \text{ dB} \end{aligned}$$

Thus, for example, the sound level for the sum of two sound pressures of level 90 dB is 93 dB.

Figure 2–5 presents a simple chart that can be used to perform decibel addition when the levels 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. Referring to Figure 2–5, the level difference 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 sound level, L_{total} , can also be calculated using the following relationship:

$$L_{\text{total}} = 10 \log(10_1^{(L/10)} + 10_2^{(L/10)})$$

Using this relationship, the example given above of adding 68 dB and 65 dB gives a combined level of 69.76 dB.

2.3.5 Weighted Sound Levels

The human ear does not respond in a uniform manner to different frequency sounds. For example, a sound pressure with level 70 dB will be perceived as much louder at 1,000 Hz than at 100 Hz. To account for this, various frequency weighting filters have been developed to reflect human sensitivity to the noise spectrum. The weighting filters de-emphasize the frequency ranges in which the human ear is

TABLE 2-1 REFERENCE QUANTITIES FOR SOUND AND VIBRATION

Name	Definition	Reference Quantities	
		SI	Typical US Practice
Sound Pressure Level	$20 \text{ Log}_{10}(p/p_0)$	20×10^{-6} Pascal	
Sound Power Level	$10 \text{ Log}_{10}(W/W_0)$	10^{-12} Watt	
Sound Intensity Level	$10 \text{ Log}_{10}(I/I_0)$	10^{-12} Watt/m ²	
Vibration Acceleration Level	$20 \text{ Log}_{10}(a/a_0)$	10^{-2} m/second	10^{-6} g
Vibration Velocity Level	$20 \text{ Log}_{10}(v/v_0)$	10^{-8} m/second	10^{-6} in/second
Vibration Displacement Level	$20 \text{ Log}_{10}(d/d_0)$	10^{-11} m	
Vibration Force Level	$20 \text{ Log}_{10}(F/F_0)$	10^{-6} N	1 lb
Energy	$10 \text{ Log}_{10}(E/E_0)$	10^{-12} J	

less sensitive. Figure 2-6 illustrates the weighting filter response curves that are commonly used with 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 one most widely used for transit-related noise measurements and community noise descriptions. The A-weighted curve shown in Figure 2-6 is almost universally accepted as a standard metric for quantifying acoustic data in terms that can be related to the subjective effects of the noise. Although a number of relatively more complex techniques have been devel-

oped to describe human perception of sounds, psychoacoustic studies have shown these methods to be only marginally better than the use of A-weighted levels. For example, the *sones* and *phons* are measures of the *loudness* and *loudness level*, but, because of the complex steps required to calculate sones and phons, they are rarely used to evaluate transportation and community noise.

The A-weighted sound level, given in dBA relative to 20 micro-Pascal, is reasonably well correlated with human response to noise. As a general rule of thumb, a 2 dBA change in noise level is barely detected by the human ear.

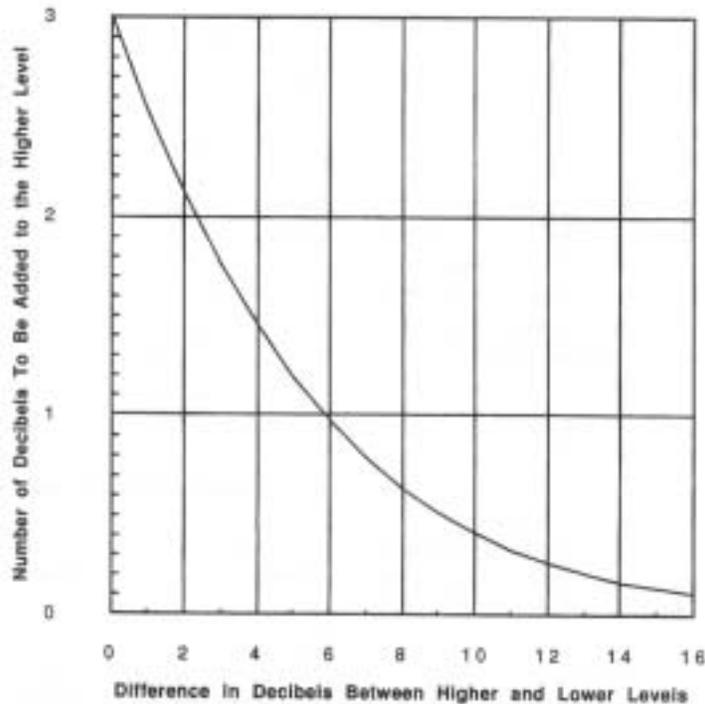


FIGURE 2-5 ENERGY SUMMATION OF LEVELS

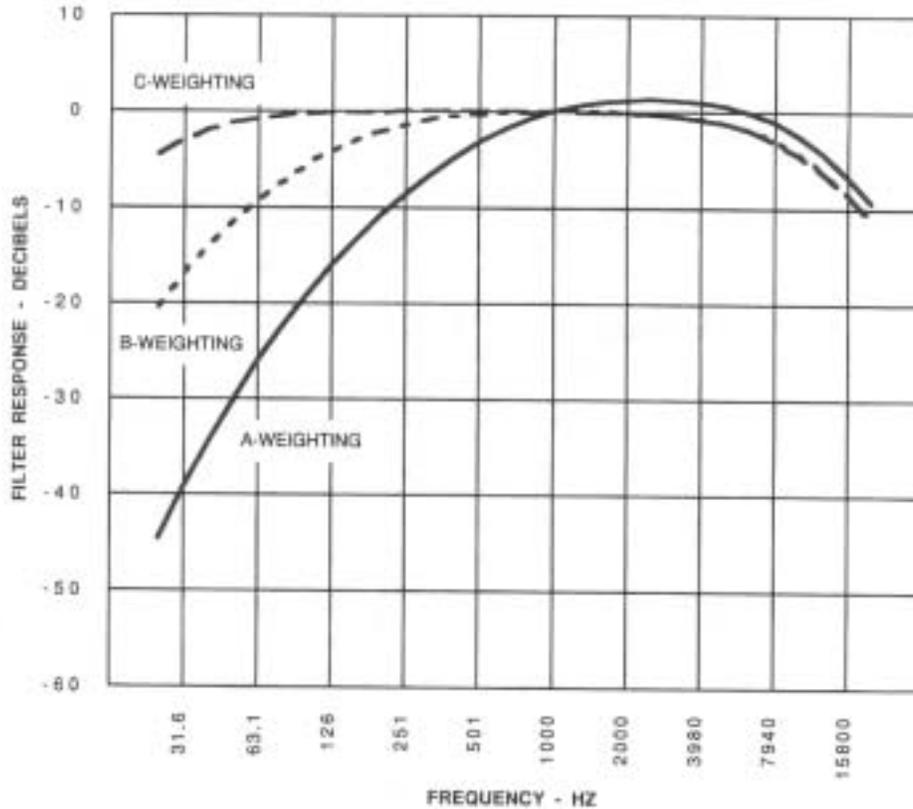


FIGURE 2-6 RESPONSE OF A-, B-, AND C-WEIGHTING NETWORKS

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 perceived loudness.

One notable disadvantage of the A-weighted sound level is that it does not accurately reflect the annoyance of audible pure tones, clicks, buzzes, rattles, and so on that may be part of a sound. Although the A-weighted levels of two sounds may be the same, sound with identifiable components or components which contain information will be considerably more distracting, annoying, and intrusive to most people than sounds which do not. 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, and attract more attention, because of the joint impact noise. To account for this phenomenon, community noise ordinances typically include a 5 dBA penalty to be added to the measured level of noise that has identifiable pure tones or other annoying components.

2.3.6 Frequency Analysis

Additional information is often needed concerning the frequency content or spectral distribution of a sound. Spectral

analyses provide estimates of the sound energy as a function of frequency and are particularly useful to characterize a noise or vibration source and for designing noise control measures. There are several methods of analyzing the frequency distribution of a complex signal: octave band, 1/3-octave band, and constant bandwidth analyses. All of these charts represent analyses of the same signal, which consists of broadband noise with several tonal components (the strongest pure tone is at 1,000 Hz).

2.3.6.1 Octave and 1/3-Octave Analyses

Figure 2-7 is an example of a 1/3-octave band analyses. The vertical and horizontal scales follow standard conventions for plotting octave and 1/3-octave band data. These conventions specify that the vertical axis be scaled as 5 dB per centimeter, and that the horizontal axis be scaled as one 1/3 octave per 0.5 cm, or one octave per 1.5 cm. This standard format for presenting octave or 1/3-octave band data facilitates comparing different analyses by overlaying on a light table, or by holding up to a light. The left-hand scale indicates the bandwidth of the filters used for the analysis, for example, 1/3-octave band. Also indicated in Figure 2-7 are the overall and A-weighted sound levels corresponding to the 1/3-octave band spectrum.

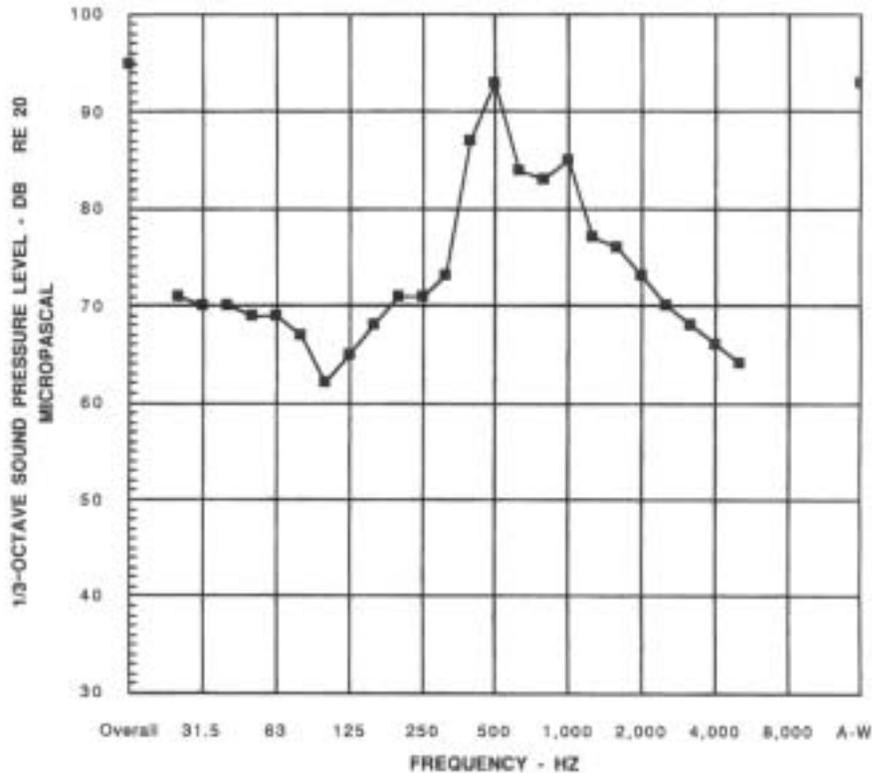


FIGURE 2-7 EXAMPLE OF 1/3-OCTAVE BAND ANALYSIS

Figure 2-8 shows the frequency response characteristic per American National Standards Institute (ANSI) specifications (8) for 3rd Order octave and 1/3-octave band filters. An octave covers a 2 to 1 ratio of frequencies, and each octave can be further subdivided into three 1/3-octaves to obtain more detail. Octave and 1/3-octave band filters are “constant percentage bandwidth” filters. The bandwidth of a 1/3-octave band filter is 23% of its nominal center frequency. The ideal octave band filter would pass the portion of the signal within the frequency band of interest and remove all other frequency components. Perfect (rectangular) filters are not possible, because realizable band pass filters do not have vertical skirts. However, they are reasonably well approximated by 6-pole analog (3rd Order) or digital filters, as employed by modern commercial 1/3-octave band analyzers.

The preferred frequency limits used for octave band and 1/3-octave bands are defined (9) by ANSI; these frequency limits are presented in Table 2-2. The center frequencies are generally used to reference specific octave and 1/3-octave bands.

Octave and 1/3-octave band analyses consist of contiguous non-overlapping frequency bands. For N contiguous frequency bands, the energy sum of the 1/3- or 1/1-octave band noise levels is the same as would be measured with one filter that covered the entire frequency range of the N filters. (If the individual filter response bandwidths did

overlap, or gaps existed between contiguous bands, as may be the case with narrow band constant bandwidth analyzers using weighting functions, the energies of the individual bands may not be so easily summed to obtain the total energy without inclusion of a correction.) In practice, three contiguous 1/3-octave band levels can be combined to obtain an equivalent octave band level for the octave band containing the three 1/3 octaves, and all of 1/3-octave band levels can be combined to obtain the equivalent overall level.

Octave band or 1/3-octave band analyses are relatively easy to perform with any number of commercial analyzers, and these analyses are often performed when simple frequency analysis is required. The use of 1/3-octave bands is usually preferable to octave band analysis, since 1/3-octave analysis provides more detail than octave band analysis. In Figure 2-7 the 1000 Hz pure tone could easily be overlooked in the octave band analysis, but it is clearly evident in the 1/3-octave band analysis.

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 octave band levels for comparison. However, another approach that is often used is to shift the octave band chart down by 5 dB to approximately convert from octave band levels to 1/3-octave band levels, assuming a constant energy per 1/3-octave band. The 5 dB factor (actu-

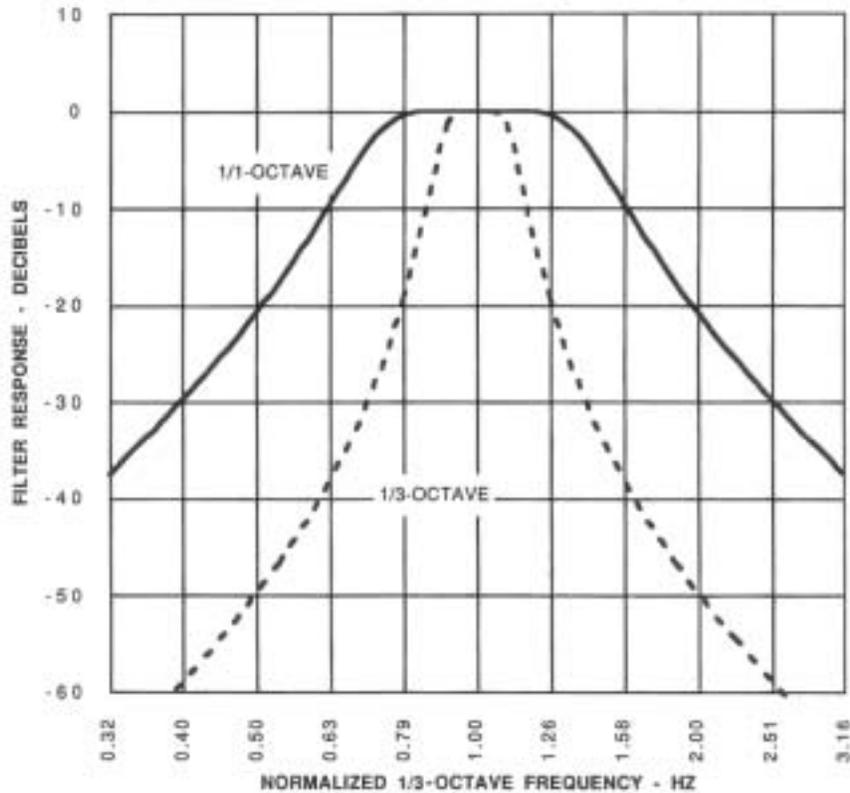


FIGURE 2-8 RESPONSE OF 1/3-OCTAVE AND 1/1-OCTAVE BAND FILTERS (ANSI SPECIFICATION S1.11-1986)

ally, 4.7 dB) 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 2-7, this assumption is reasonable except at the 1,000 Hz octave. Because the level of the 1,000 Hz 1/3-octave dominates the 1,000 Hz octave band, the level of the 1,000 Hz octave band is only 1 dB above the level of the 1,000 Hz 1/3-octave band.

2.3.6.2 Constant Bandwidth Analyses

Figure 2-9 is an example of constant bandwidth narrow-band analysis of wayside passby noise using a Fast Fourier Transform (FFT) analyzer, referred often to as a *Fourier analyzer*. The advantage of this type of frequency analysis is that each of the spectral components of the acoustical data is clearly described and identified, so that sources of specific discrete frequency components may be more easily identified. Thus, the frequency of wheel squeal may be precisely defined and correlated with natural modes of vibration of the wheel. In Figure 2-9, the wayside noise contains a series of discrete frequency components between 500 and 900 Hz which may related to corrugation,

anti-resonance in the wheel, or other mechanical characteristic. A broadband peak exists between 1,000 and 2,000 Hz, which might be related to resonances or anti-resonances of the wheel. Thus, narrowband analyses provide a means of characterizing types of noise and identifying noise sources.

The left-hand scale of Figure 2-9 indicates the quantity being displayed. In this case, the scale indicates that the noise levels are of the acoustic energies passed by filters with an effective noise bandwidth of 15 Hz. Filters with other bandwidths can be employed for analyzing noise, or vibration, as discussed below. The analyses actually includes 400 spectral "lines" separated by 10 Hz, with each line representing a filter with bandwidth 15 Hz. Thus, there is some overlap of the spectral lines, which must be taken into account when summing the spectral energies to obtain the total energy of the analyses. In this case, the energy sum of the spectral components, or lines, must be reduced by about 1.8 dB to give the total energy.

The 1/3-octave bands at 40, 400, and 4,000 Hz have effective noise bandwidths of 9.26, 92.6, 926 Hz, respectively. Thus, a constant bandwidth analyzer with a 20 Hz resolution bandwidth has much higher resolution than the 1/3-octave band filter at 4,000 Hz. At very low frequencies, however, the constant bandwidth analysis has much less resolution than the 1/3-octave analysis.

TABLE 2-2 ONE-THIRD OCTAVE BAND CENTER AND EDGE FREQUENCIES

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	63	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	11360	7079	8000	8913
40				8913	10000	11220
41				11220	12500	14130
42	11360	16000	22720	14130	16000	17780
43				17780	20000	22390

2.3.6.3 Spectrum Level and Power Spectral Density

The results of narrowband frequency analyses are commonly presented in terms of *spectrum level* or *power spectral density* (PSD). The spectrum level of a noise is the level that would be measured if an analyzer had an ideal filter 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 spectrum in Figure 2-9 can be converted to a power spectral density level, or spectrum level, plot by subtracting 11.8 dB ($10 \log_{10}[\text{Effective Noise Bandwidth}]$). However, if the effective noise bandwidth exceeds the “line” width of the actual discrete frequency component, the result will be in error, since the power spectral density is not defined for discrete frequency components. *Normalized levels*, such as the spectrum level, or power spectral density, are not accurate for characterizing discrete frequency components. Mathematically, the *spectral density* of a discrete frequency component of noise is infinite. Therefore, when displaying spectral data with discrete frequency components, the data should not be normalized for filter bandwidth. The main uses of the power spectral density or spectrum level for-

mat are (1) comparing data taken with analyzers with different analysis bandwidths and (2) checking compliance with specifications given in terms of spectrum level.

2.4 CHARACTERIZING NOISE ENVIRONMENTS

Noise produced by any transit system must be characterized to determine the need for mitigation. The field of acoustics is replete with descriptors and methods for doing this, largely because of the many types of noise and the difficulty in assessing human responses to noise. Some of the descriptors that are used are described below.

2.4.1 Noise Criterion Curves

The Noise Criterion (NC) curves are commonly used for noise criteria indoors and are widely used in architectural specifications for HVAC systems. The NC curves are illustrated in Figure 2-10. The NC curves are used by plotting the octave band levels against the NC curves, and the NC level for a par-

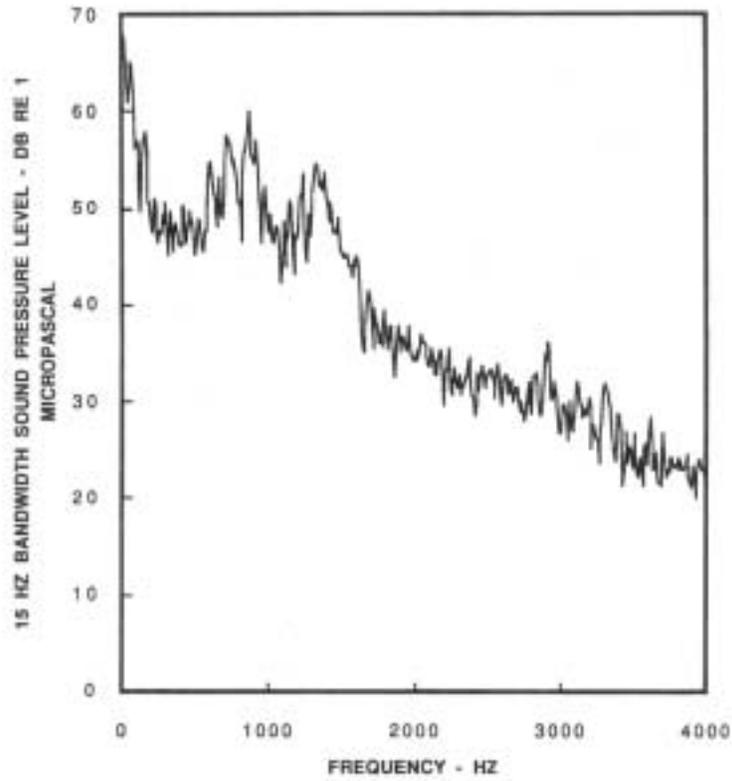


FIGURE 2-9 EXAMPLE OF FOURIER SPECTRAL ANALYSIS

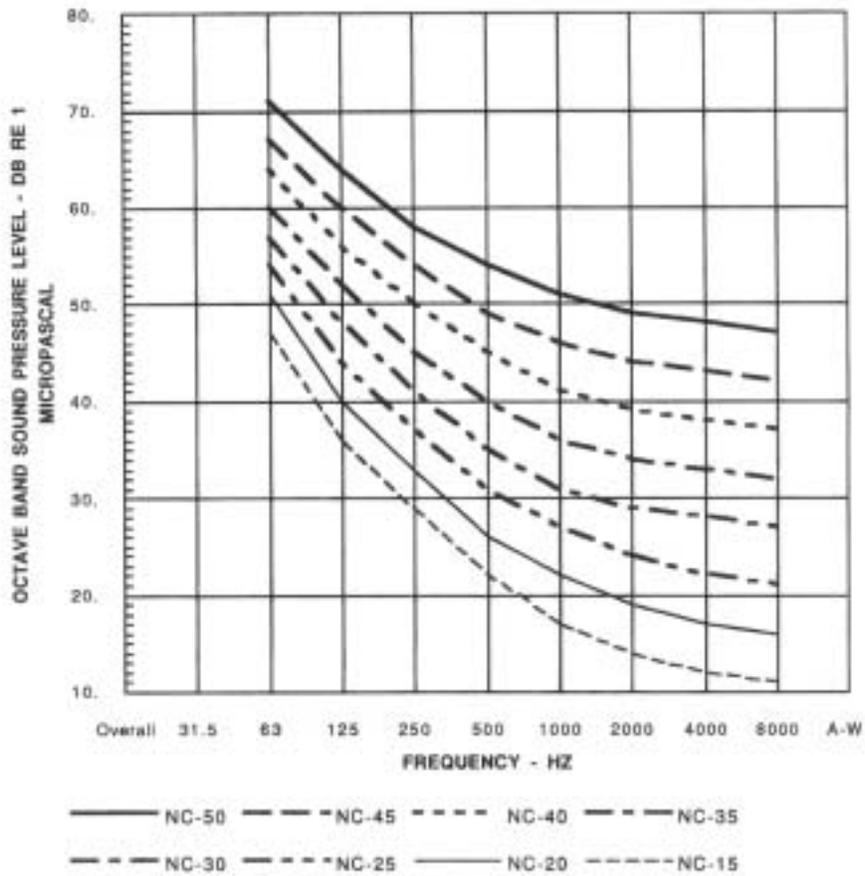


FIGURE 2-10 NOISE CRITERIA CURVES

ticular 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 octave band that dominates the overall rating. Another form of the NC curves called the preferred noise criteria (PNC) curves, designed to overcome some objections to the NC curves, are also occasionally encountered. The NC curves can be used to specify noise environments or evaluate noise consisting of broadband and pure tones, and have been applied to rail transit groundborne noise in buildings.

2.4.2 Energy Equivalent Levels

Because noise, particularly outdoor noise, varies with time, there is a need for measures of noise that account for these variations. The most popular measures are based on an energy dose, or equivalent noise energy, principle. An energy dose or equivalent energy level is mathematically attractive for combining the sound energies of differing sources or events. Several of these noise descriptors are defined below:

Equivalent Sound Level— L_{eq} . Sometimes referred to as the energy average sound level over the period of interest, L_{eq} is 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 time-varying noise during the same time period. Mathematically, it is defined as

$$L_{eq} = 10 \text{Log}_{10} \left[\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} \frac{p^2(t)}{p_0^2} dt \right]$$

where $p(t)$ is the time varying pressure and p_0 is the standard reference pressure of 20 micro-Pa (2×10^{-5} N/m²). This is mathematically equivalent to the definition of rms level given above. However, “rms” is typically used for much shorter periods of time than L_{eq} . The average sound level over a period of time T is often symbolized as $L_{eq}(T)$. Commonly used descriptors are

- $L_{eq}(h)$ = hourly averaged sound level
- $L_{eq}(8h)$ = 8-hour averaged sound level
- $L_{eq}(d)$ = average daytime sound level
- $L_{eq}(n)$ = average nighttime sound level

The term “average” indicates energy average.

Day-Night Average Sound Level— L_{dn} . L_{dn} is a 24-hour average sound level in which the nighttime noise levels occurring between 10:00 P.M. and 7:00 A.M. are increased by 10 dBA before calculation of the 24-hour average. The 10 dBA *penalty* is included 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.

Community Noise Equivalent Level (CNEL). CNEL is similar to the L_{dn} except that the 24-hour period is broken into three periods: day (0700 to 1900), evening (1900 to 2200), and night (2200 to 0700). Penalties of 5 dBA are applied to the evening period and 10 dBA to the nighttime period. In most cases, CNEL will be less than 0.5 dBA higher than the L_{dn} , an insignificant difference which is often ignored.

2.4.3 Histograms

Figure 2–11 illustrates the statistical distribution, or histogram, of fluctuating community noise. The vertical axis is the percentage of time that the noise level indicated on the horizontal axis is exceeded. Adding a transient noise source such as rapid transit trains that create relatively high noise levels for relatively short periods of time will modify the histogram as indicated in the figure. The train noise increased the level exceeded 1% of the time, but left the level exceeded 10% of the time unchanged.

Histograms may be represented by *levels exceeded “n” percent of the time, or L_n* . The level exceeded n-percent 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 can range from a full 24-hour period to a several minute spot check. To avoid confusion, the time period should be clearly specified. The statistical levels are usually determined from histograms developed with the “fast” sound-level meter response, or rms averaging time of 1 sec. In practice, the “slow” meter response is often used. This will affect the extreme ends of the histogram, but should not affect the L_{10} or L_{50} significantly.

The levels exceeded 50% of the time are largely unaffected by transient events which, together, occur over a period of time less than 30 min in any hour. For this reason, the median noise level, L_{50} , is a *robust descriptor* of time varying community noise levels against which transient noise levels may be projected. The L_1 is often used to describe the “maximum level,” and the L_{90} is often used to describe the background sound level.

Histograms are usually developed for A-weighted sound levels, but can be developed for 1/3-octave, and octave sound levels, as well as for vibration levels. Histograms and the related L_n ’s are one of the most practical tools available for describing nonstationary, or time varying, community noise.

2.5 SOUND ABSORPTION AND TRANSMISSION

Sound absorption and sound isolation are commonly misunderstood parameters. Sound absorption refers to the ability of materials, such as fiberglass blankets or acoustical tile, to convert sound energy into heat. Sound absorbing materi-

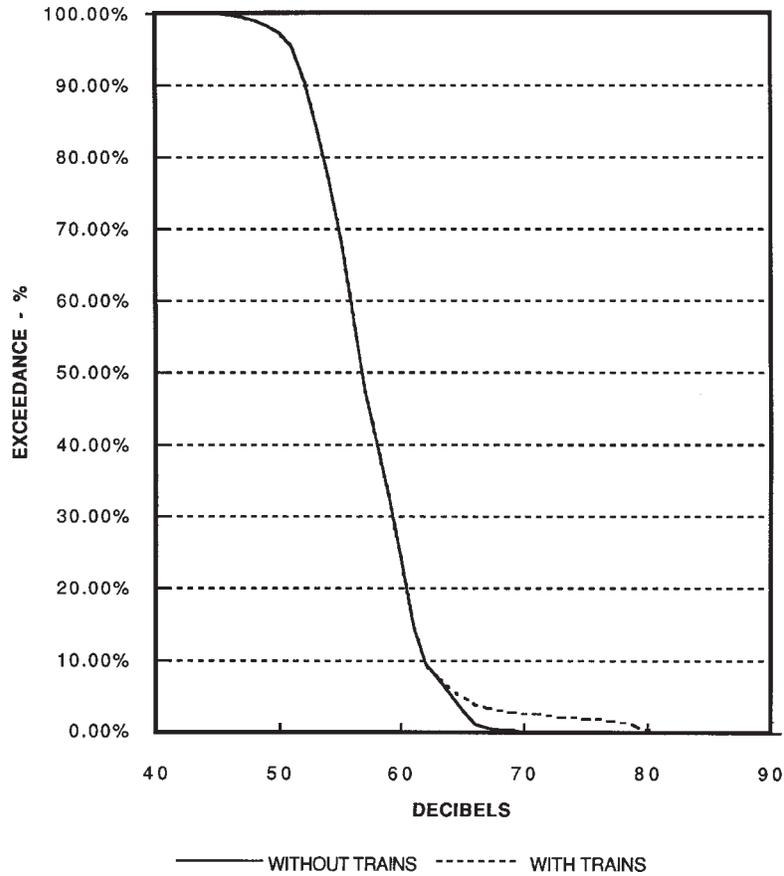


FIGURE 2-11 HISTOGRAM OF COMMUNITY NOISE LEVELS

als reduce sound reflected from a surface by dissipating acoustical energy in the material. Porous materials do not stop, block, or contain sound. If air passes through or around a material, so can sound. Sound transmission loss refers to the ability of a partition or barrier to attenuate sound as it transmits through. A material must be massive and airtight to effectively isolate sound.

Figure 2-12 illustrates the reflection and transmission of sound energy. The incident sound intensity impinges on the wall. Part of the sound intensity is transmitted through the wall, and part of the sound power is reflected from the wall. Another part is absorbed in the wall. The absorption coefficient of the wall determines the intensity of the reflected sound relative to the intensity of the incident sound, includes the effect of both the sound intensity transmitted through the wall as well as the sound intensity absorbed by the material in the wall. The sound absorption coefficient is always between 0 and 1, though published values of the coefficient may exceed 1 because of the methods of measurement. The transmission coefficient is the ratio of transmitted sound intensity to incident sound intensity, and is always between 0 and 1. The sound transmission loss in decibels is a convenient descriptor of the sound isolation qualities of the wall. These quantities are mathematically defined as

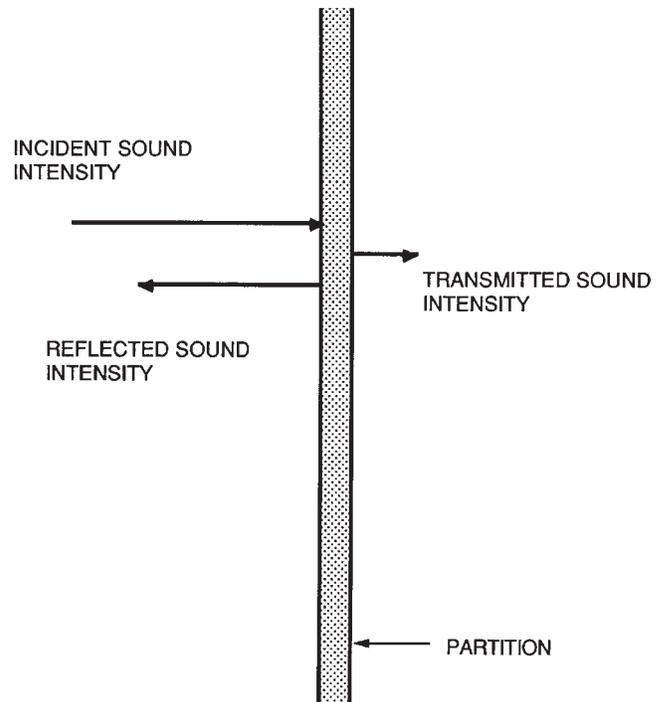


FIGURE 2-12 REFLECTION AND TRANSMISSION OF ACOUSTIC ENERGY

$$\text{Sound Absorption Coefficient } (\alpha) = 1 - I_{\text{reflected}}/I_{\text{incident}}^2$$

$$\text{Transmission Coefficient } (\tau) = I_{\text{Transmitted}}/I_{\text{Incident}}$$

$$\text{Transmission Loss in Decibels (TL)} = 10 \text{ Log } (1/\tau)$$

An absorption coefficient of zero indicates perfect reflection of incident sound energy. An absorption coefficient of one indicates a perfectly nonreflecting wall, which may be due to complete absorption of sound energy, perfect transmission of sound energy through the wall, or a combination of both which produces a nonreflecting wall. For example, an opening in a wall would have an absorption coefficient of one. (Actually, at long wavelengths relative to the dimension of the opening, the absorption coefficient may exceed one.)

The sound absorptive properties of a material depend on the porosity and thickness, or dynamic response, of the material, and the angle at which the sound waves strike the material. For convenience, the reported absorption coefficient is generally the average over all angles of incidence, sometimes referred to as the random incidence absorption coefficient.

2.5.1 Sound Transmission Loss

The sound transmission loss (TL) represents the loss in decibels of the sound intensity as it transmits through the wall or barrier. Sound transmission loss is given in decibels. A wall with a large sound transmission loss is a very effective sound isolator, and a wall with zero transmission loss is acoustically transparent.

2.5.2 Sabine Absorption Coefficient

The Sabine absorption coefficient is very similar to the absorption coefficient defined above, but is the absorption coefficient determined for a patch of material placed on the floor of a reverberation chamber. The Sabine absorption coefficient is often greater than one, owing to the method of measurement. More discussion of the Sabine absorption coefficient can be found in Beranek's work (2).

2.5.3 Noise Reduction Coefficient–NRC

The noise reduction coefficient, or NRC, is a single-number descriptor of the sound absorbing properties of various materials. The NRC is the arithmetic average of the "Sabine absorption coefficients" at 250, 500, 1,000 and 2,000 Hz.

2.5.4 Sound Transmission Class–STC

The sound transmission class, or STC, is a single-number descriptor of the sound transmission losses of partitions. Information on determining the STC can be found in Beranek's work, referenced above. As a general rule, partitions with high values of STC have high sound transmission losses.

2.5.5 Materials and Mounting

Materials such as glass-fiber insulation have high absorption coefficients but very low transmission loss, while solid materials such as concrete have relatively high transmission loss but very low absorption coefficients. Mounting glass-fiber blankets on a concrete wall will result in both high transmission loss and high absorption. Manufacturers provide data on the sound absorption coefficients and sound transmission losses of various materials and partitions. More discussion is provided concerning sound transmission loss and sound absorption coefficient where appropriate, specifically in the chapters concerning station and vent shaft treatments, and vehicle noise control.

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