

Fundamentals of Acoustics

FOURTH EDITION



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GLOSSARY OF SYMBOLS

This list identifies some symbols that are not necessarily defined every time they appear in the text.

a	acceleration; absorption coefficient (dB per distance); Sabine absorptivity	d'	detectability index
a_E	random-incidence energy absorption coefficient	D	directivity; dipole strength
A	sound absorption	DI	directivity index
AG	array gain	DNL	detected noise level
b	loss per bounce; decay parameter	DT	detection threshold
$b(\theta, \phi)$	beam pattern	$\mathcal{D}$	diffraction factor
B	magnetic field; susceptance	e	specific energy
BL	bottom loss	E	total energy
$\mathcal{B}$	adiabatic bulk modulus	E_k	kinetic energy
$\mathcal{B}_T$	isothermal bulk modulus	E_p	potential energy
c	speed of sound	EL	echo level
c_g	group speed	$\mathcal{E}$	time-averaged energy density
c_p	phase speed	$\mathcal{E}_i$	instantaneous energy density
C	electrical capacitance; acoustic compliance; heat capacity	f	instantaneous force; frequency (Hz)
C_p	heat capacity at constant pressure	f_r	resonance frequency
c_p	specific heat at constant pressure	f_u, f_l	upper, lower half-power frequencies
C_V	heat capacity at constant volume	F	peak force amplitude; frequency (kHz)
c_V	specific heat at constant volume	F_e	effective force amplitude
$CNEL$	community noise equivalent level (dBA)	g	spectral density of a transient function; sound-speed gradient; acceleration of gravity; aperture function
d	detection index	G	conductance
		$\mathcal{G}$	adiabatic shear modulus
		h	specific enthalpy
		$h(\theta, \phi)$	directional factor

$H(T_K)$	population function	L_{TPN}	tone-corrected perceived noise level
I	time-averaged acoustic intensity; current, effective current amplitude	L_x	x -percentile-exceeded sound level (dBA, fast)
I_{ref}	reference acoustic intensity	LNP	noise pollution level (dBA)
$I(t)$	instantaneous acoustic intensity	m	mass
IIC	impact isolation class	m_r	radiation mass
IL	intensity level	M	acoustic inertance; bending moment; molecular weight; acoustic Mach number, flow Mach number
ISL	intensity spectrum level	$\mathcal{M}$	microphone sensitivity
$\mathcal{I}$	time-averaged spectral density of intensity	$\mathcal{ML}$	microphone sensitivity level
$\mathcal{I}(t)$	instantaneous spectral density of intensity	$\mathcal{M}_{ref}$	reference microphone sensitivity
$\mathcal{J}$	impulse	N	loudness (sone)
k	wave number	NCB	balanced noise criterion curves
$\vec{k}$	propagation vector	NEF	noise exposure forecast
k_B	Boltzmann's constant	NL	noise level
k_c, k_m	coupling coefficients	NR	noise reduction
ℓ	discontinuity distance	NSL	noise spectrum level
L	inductance	p	acoustic pressure
L_A	A-weighted sound level (dBA)	P	peak acoustic pressure amplitude
L_C	C-weighted sound level (dBC)	P_e	effective acoustic pressure amplitude
L_d	daytime average sound level (dBA)	P_{ref}	reference effective acoustic pressure amplitude
L_{dn}	day-night averaged sound level (dBA)	PR	privacy rating
L_e	evening average sound level (dBA)	Pr	Prandtl number
L_{eq}	equivalent continuous sound level (dBA)	PSL	pressure spectrum level
L_{ex}	noise exposure level (dBA)	PTS	permanent threshold shift
L_{EPN}	effective perceived noise level	$\mathcal{P}$	hydrostatic pressure
L_h	hourly average sound level (dBA)	$\mathcal{P}_0$	equilibrium hydrostatic pressure
L_I	intensity level $re 10^{-12} \text{ W/m}^2$	q	charge; source strength density; thermal energy; scaled acoustic pressure ($p/\rho_0 c^2$)
L_N	loudness level (phon)	Q	quality factor; source strength (amplitude of volume velocity)
L_n	night average sound level (dBA)		

FUNDAMENTALS OF ACOUSTICS

Fourth Edition

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John Wiley & Sons, Inc.

New York Chichester Weinheim Brisbane Singapore Toronto

*With grateful thanks to our wives,
Linda Miles Coppens and Marilyn Sanders,
for their unflagging support and gentle patience.*

ACQUISITIONS EDITOR Stuart Johnson

MARKETING MANAGER Sue Lyons

PRODUCTION EDITOR Barbara Russiello

SENIOR DESIGNER Kevin Murphy

ELECTRONIC ILLUSTRATIONS Publication Services, Inc.

This book was set in 10/12 Palatino by Publication Services, Inc. and printed and bound by Hamilton Press. The cover was printed by Hamilton Press.

This book is printed on acid-free paper. ∞

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Library of Congress Cataloging-in-Publication Data:

Fundamentals of acoustics / Lawrence E. Kinsler. . . [et al.].—4th ed.

p.cm.

Includes index.

1. Sound-waves. 2. Sound—Equipment and supplies. 3. Architectural acoustics. I.
Kinsler, Lawrence E.

QC243 .F86 2000 534—dc21

99-049667

ISBN 0-471-84789-5

Printed in the United States of America

10 9 8 7 6 5 4 3 2

PREFACE

Credit for the longevity of this work belongs to the original two authors, Lawrence Kinsler and Austin Frey, both of whom have now passed away. When Austin entrusted us with the preparation of the third edition, our goal was to update the text while maintaining the spirit of the first two editions. The continued acceptance of this book in advanced undergraduate and introductory graduate courses suggests that this goal was met. For this fourth edition, we have continued this updating and have added new material.

Considerable effort has been made to provide more homework problems. The total number has been increased from about 300 in the previous editions to over 700 in this edition. The availability of desktop computers now makes it possible for students to investigate many acoustic problems that were previously too tedious and time consuming for classroom use. Included in this category are investigations of the limits of validity of approximate solutions and numerically based studies of the effects of varying the various parameters in a problem. To take advantage of this new tool, we have added a great number of problems (usually marked with a suffix "C") where the student may be expected to use or write computer programs. Any convenient programming language should work, but one with good graphing software will make things easier. Doing these problems should develop a greater appreciation of acoustics and its applications while also enhancing computer skills.

The following additional changes have been made in the fourth edition: (1) As an organizational aid to the student, and to save instructors some time, equations, figures, tables, and homework problems are all now numbered by chapter and section. Although appearing somewhat more cumbersome, we believe the organizational advantages far outweigh the disadvantages. (2) The discussion of transmitter and receiver sensitivity has been moved to Chapter 5 to facilitate early incorporation of microphones in any accompanying laboratory. (3) The chapters on absorption and sources have been interchanged so that the discussion of beam patterns precedes the more sophisticated discussion of absorption effects. (4) Derivations from the diffusion equation of the effects of thermal conductivity on the attenuation of waves in the free field and in pipes have been added to the chapter on absorption. (5) The discussions of normal modes and waveguides

have been collected into a single chapter and have been expanded to include normal modes in cylindrical and spherical cavities and propagation in layers. (6) Considerations of transient excitations and orthonormality have been enhanced. (7) Two new chapters have been added to illustrate how the principles of acoustics can be applied to topics that are not normally covered in an undergraduate course. These chapters, on finite-amplitude acoustics and shock waves, are not meant to survey developments in these fields. They are intended to introduce the relevant underlying acoustic principles and to demonstrate how the fundamentals of acoustics can be extended to certain more complicated problems. We have selected these examples from our own areas of teaching and research. (8) The appendixes have been enhanced to provide more information on physical constants, elementary transcendental functions (equations, tables, and figures), elements of thermodynamics, and elasticity and viscosity.

New materials are frequently at a somewhat more advanced level. As in the third edition, we have indicated with asterisks in the Contents those sections in each chapter that can be eliminated in a lower-level introductory course. Such a course can be based on the first five or six chapters with selected topics from the seventh and eighth. Beyond these, the remaining chapters are independent of each other (with only a couple of exceptions that can be dealt with quite easily), so that topics of interest can be chosen at will.

With the advent of the handheld calculator, it was no longer necessary for textbooks to include tables for trigonometric, exponential, and logarithmic functions. While the availability of desktop calculators and current mathematical software makes it unnecessary to include tables of more complicated functions (Bessel functions, etc.), until handheld calculators have these functions programmed into them, tables are still useful. However, students are encouraged to use their desktop calculators to make fine-grained tables for the functions found in the appendixes. In addition, they will find it useful to create tables for such things as the shock parameters in Chapter 17.

From time to time we will be posting updated information on our web site: www.wiley.com/college/kinsler. At this site you will also be able to send us messages. We welcome you to do so.

We would like to express our appreciation to those who have educated us, corrected many of our misconceptions, and aided us: our coauthors Austin R. Frey and Lawrence E. Kinsler; our mentors James Mcgrath, Edwin Ressler, Robert T. Beyer, and A. O. Williams; our colleagues O. B. Wilson, Anthony Atchley, Steve Baker, and Wayne M. Wright; and our many students, including Lt. Thomas Green (who programmed many of the computer problems in Chapters 1–15) and L. Miles.

Finally, we offer out heartfelt thanks for their help, cooperation, advice, and guidance to those at John Wiley & Sons who were instrumental in preparing this edition of the book: physics editor Stuart Johnson, production editor Barbara Russiello, designer Kevin Murphy, editorial program assistants Cathy Donovan and Tom Hempstead, as well as to Christina della Bartolomea who copy edited the manuscript and Gloria Hamilton who proofread the galleys.

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FUNDAMENTALS OF VIBRATION

1.1 INTRODUCTION

Before beginning a discussion of acoustics, we should settle on a system of units. Acoustics encompasses such a wide range of scientific and engineering disciplines that the choice is not easy. A survey of the literature reveals a great lack of uniformity: writers use units common to their particular fields of interest. Most early work has been reported in the CGS (centimeter–gram–second) system, but considerable engineering work has been reported in a mixture of metric and English units. Work in electroacoustics and underwater acoustics has commonly been reported in the MKS (meter–kilogram–second) system. A codification of the MKS system, the SI (Le Système International d’Unités), has been established as the standard. This is the system generally used in this book. CGS and SI units are equated and compared in Appendix A1.

Throughout this text, “log” will represent logarithm to the base 10 and “ln” (the “natural logarithm”) will represent logarithm to the base e .

Acoustics as a science may be defined as the generation, transmission, and reception of energy as vibrational waves in matter. When the molecules of a fluid or solid are displaced from their normal configurations, an internal elastic restoring force arises. It is this elastic restoring force, coupled with the inertia of the system, that enables matter to participate in oscillatory vibrations and thereby generate and transmit acoustic waves. Examples include the tensile force produced when a spring is stretched, the increase in pressure produced when a fluid is compressed, and the restoring force produced when a point on a stretched wire is displaced transverse to its length.

The most familiar acoustic phenomenon is that associated with the sensation of sound. For the average young person, a vibrational disturbance is interpreted as sound if its frequency lies in the interval from about 20 Hz to 20,000 Hz (1 Hz = 1 hertz = 1 cycle per second). However, in a broader sense acoustics also includes the *ultrasonic* frequencies above 20,000 Hz and the *infrasonic* frequencies below 20 Hz. The natures of the vibrations associated with acoustics are many, including

the simple sinusoidal vibrations produced by a tuning fork, the complex vibrations generated by a bowed violin string, and the nonperiodic motions associated with an explosion, to mention but a few. In studying vibrations it is advisable to begin with the simplest type, a one-dimensional sinusoidal vibration that has only a single frequency component (a pure tone).

1.2 THE SIMPLE OSCILLATOR

If a mass m , fastened to a spring and constrained to move parallel to the spring, is displaced slightly from its rest position and released, the mass will vibrate. Measurement shows that the displacement of the mass from its rest position is a sinusoidal function of time. Sinusoidal vibrations of this type are called *simple harmonic vibrations*. A large number of vibrators used in acoustics can be modeled as simple oscillators. Loaded tuning forks and loudspeaker diaphragms, constructed so that at low frequencies their masses move as units, are but two examples. Even more complex vibrating systems have many of the characteristics of the simple systems and may often be modeled, to a first approximation, by simple oscillators.

The only physical restrictions placed on the equations for the motion of a simple oscillator are that the restoring force be directly proportional to the displacement (Hooke's law), the mass be constant, and there be no losses to attenuate the motion. When these restrictions apply, the frequency of vibration is independent of amplitude and the motion is simple harmonic.

A similar restriction applies to more complex types of vibration, such as the transmission of an acoustic wave through a fluid. If the acoustic pressures are so large that they no longer are proportional to the displacements of the particles of fluid, it becomes necessary to replace the normal acoustic equations with more general equations that are much more complicated. With sounds of ordinary intensity this is not necessary, for even the noise generated by a large crowd at a football game rarely causes the amplitude of motion of the air molecules to exceed one-tenth of a millimeter, which is within the limit given above. The amplitude of the shock wave generated by a large explosion is, however, well above this limit, and hence the normal acoustic equations are not applicable.

Returning to the simple oscillator shown in Fig. 1.2.1, let us assume that the restoring force f in newtons (N) can be expressed by the equation

$$f = -sx \quad (1.2.1)$$

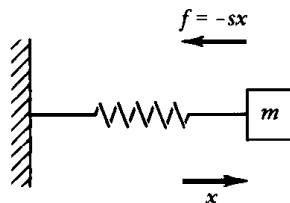


Figure 1.2.1 Schematic representation of a simple oscillator consisting of a mass m attached to one end of a spring of spring constant s . The other end of the spring is fixed.

REFLECTION AND TRANSMISSION

6.1 CHANGES IN MEDIA

When an acoustic wave traveling in one medium encounters the boundary of a second medium, reflected and transmitted waves are generated. Discussion of this phenomenon is greatly simplified if it is assumed that both the incident wave and the boundary between the media are planar and that all media are fluids. The complications that arise when one of the media is a solid will be left to Section 6.6. However, it is worthwhile to note that for normal incidence many solids obey the same equations developed for fluids. The only modification needed is that the speed of sound in the solid must be the *bulk speed of sound*, based on both the *bulk* and *shear* moduli since, unlike the bar of Chapter 3, an extended solid is not free to change its transverse dimensions. See Appendix A11. Values of the bulk speeds of sound in some solids are listed in Appendix A10.

The ratios of the pressure amplitudes and intensities of the reflected and transmitted waves to those of the incident wave depend on the characteristic acoustic impedances and speeds of sound in the two media and on the angle the incident wave makes with the interface. Let the incident and reflected waves travel in a fluid of characteristic acoustic impedance $r_1 = \rho_1 c_1$, where ρ_1 is the equilibrium density of the fluid (the subscript "0" has been suppressed for economy of notation) and c_1 the speed of sound in the fluid. Let the transmitted wave travel in a fluid of characteristic acoustic impedance $r_2 = \rho_2 c_2$. If the complex pressure amplitude of the incident wave is P_i , that of the reflected wave P_r , and that of the transmitted wave P_t , then we can define the *pressure transmission and reflection coefficients*:

$$T = P_t/P_i \quad (6.1.1)$$

$$R = P_r/P_i \quad (6.1.2)$$

Since the intensity of a harmonic plane progressive wave is $P^2/2r$, the *intensity transmission and reflection coefficients* are real and are defined by

$$T_I = I_t/I_i = (r_1/r_2)|T|^2 \quad (6.1.3)$$

$$R_I = I_r/I_i = |\mathbf{R}|^2 \quad (6.1.4)$$

Most real situations have beams of sound with *finite* cross-sectional area. As we have seen, a beam can be described locally by nearly parallel rays and thus can be approximated by a plane wave of finite extent. While there can be some anomalies resulting from diffraction at the edges of the beam, if the cross-sectional area is sufficiently great compared to a wavelength, they can be ignored and the equations developed in this chapter applied.

The power carried by a beam of sound is the acoustic intensity multiplied by the cross-sectional area of the beam. If an incident beam of cross-sectional area A_i is obliquely incident on a boundary, the cross-sectional area A_t of the transmitted beam generally is not the same as that of the incident beam. It will be shown later that the cross-sectional areas of the incident and reflected beams are equal under all circumstances. The *power transmission and reflection coefficients* are defined by

$$T_{II} = (A_t/A_i)T_I = (A_t/A_i)(r_1/r_2)|\mathbf{T}|^2 \quad (6.1.5)$$

$$R_{II} = R_I = |\mathbf{R}|^2 \quad (6.1.6)$$

From conservation of energy, the power in the incident beam must be shared between reflected and transmitted beams so that

$$R_{II} + T_{II} = 1 \quad (6.1.7)$$

Cases more complicated than those included in this chapter are available in specialized textbooks.¹

6.2 TRANSMISSION FROM ONE FLUID TO ANOTHER: NORMAL INCIDENCE

As indicated in Fig. 6.2.1, let the plane $x = 0$ be the boundary between fluid 1 of characteristic acoustic impedance r_1 and fluid 2 of characteristic acoustic impedance r_2 . Let there be an incident wave traveling in the $+x$ direction,

$$\mathbf{p}_i = \mathbf{P}_i e^{j(\omega t - k_1 x)} \quad (6.2.1)$$

which, when striking the boundary, generates a reflected wave

$$\mathbf{p}_r = \mathbf{P}_r e^{j(\omega t + k_1 x)} \quad (6.2.2)$$

and a transmitted wave

$$\mathbf{p}_t = \mathbf{P}_t e^{j(\omega t - k_2 x)} \quad (6.2.3)$$

All the waves must have the same frequency, but, since the speeds c_1 and c_2 are different, the wave numbers $k_1 = \omega/c_1$ in fluid 1 and $k_2 = \omega/c_2$ in fluid 2 are different.

¹Officer, *Introduction to the Theory of Sound Transmission*, McGraw-Hill (1958). Ewing, Jardetzky, and Press, *Elastic Waves in Layered Media*, McGraw-Hill (1957). Brekhovskikh, *Waves in Layered Media*, Academic Press (1960).

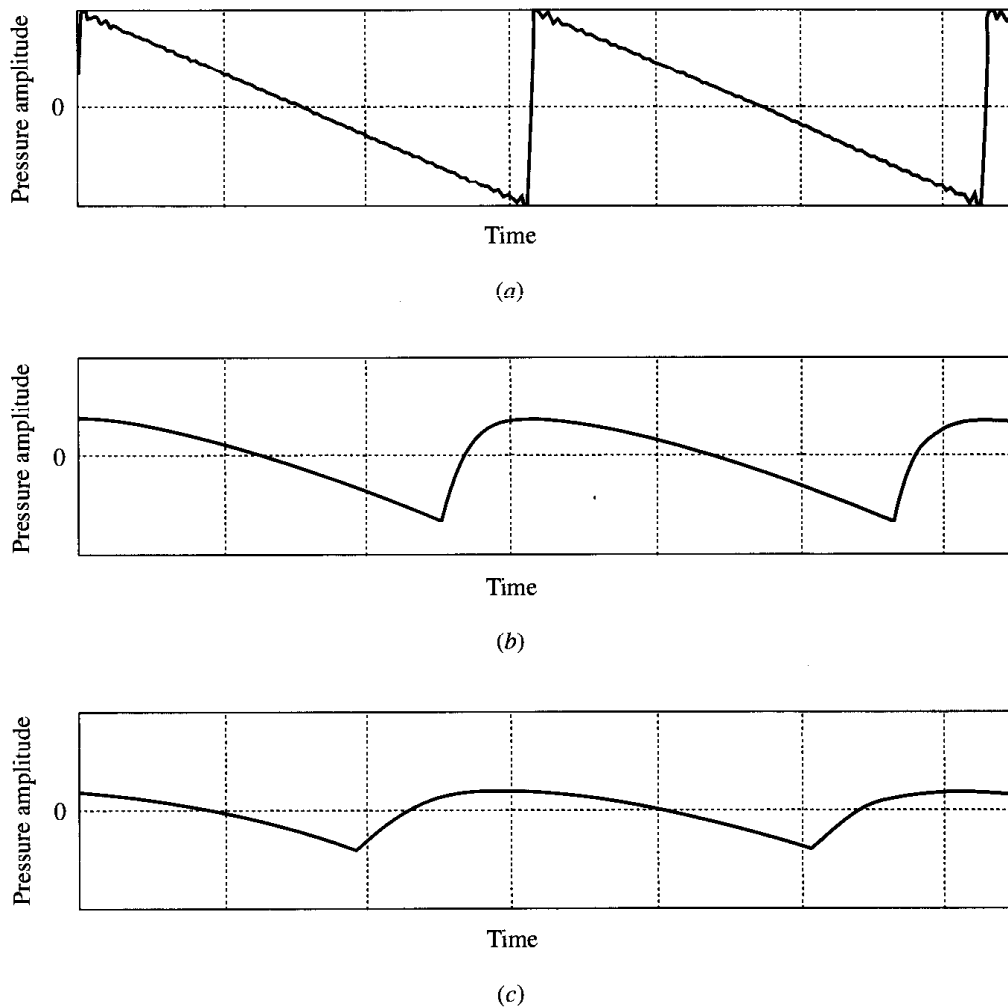


Figure 8.9.1 The time history of an initially sawtooth traveling wave at several positions down a pipe with wall losses predominating: (a) $\alpha x = 0$, (b) $\alpha x = 0.5$, and (c) $\alpha x = 1.5$, where α is the spatial decay coefficient of the fundamental. (One hundred harmonics are included in the calculation.)

waveform will have its peaks rounded off, but the maxima and minima of the harmonics migrate forward, leading to a sharper rise and gentler fall to the waveform. See Fig. 8.9.1.

*8.10 ATTENUATION IN SUSPENSIONS

When a fluid contains inhomogeneities such as fog droplets, suspended particles, bubbles, thermal microcells, or regions of turbulence, acoustic energy is lost from a sound beam faster than in the homogeneous medium. The excess attenuation in suspensions arises from (1) absorption—the conversion of acoustic energy into thermal energy—and (2) scattering—the reradiation of incident acoustic energy out of the incident beam. In this section, we will restrict discussion to the cases of liquid droplets in a gas and air bubbles in water, but the analysis will be general enough to cover many similar cases.

The combined effects of scattering and absorption can be described by an *extinction cross section* σ , where σ is an *effective* area whose product with the incident intensity is equal to the power lost from the sound beam. Assume that the particle density is N particles per unit volume, that they all have an extinction cross section σ , and that they do not “shadow” each other. In each incremental distance dx along the path of the incident sound beam, there

will be a total cross section of $N\sigma dx$ for each unit cross-sectional area of the beam. The fractional intensity that is "intercepted" by the particles is $dI/I = N\sigma dx$ so that

$$I = I_0 e^{-N\sigma x} \quad (8.10.1)$$

For the cases of interest, the intensity is proportional to the square of the pressure amplitude P , so $P = P_0 \exp(-\alpha x)$, where $\alpha = N\sigma/2$ Np/m or $\alpha = 4.35N\sigma$ dB/m. If, instead of only one population of particles all with the same cross section, we had N particles per unit volume, each with its own cross section σ_i , then $N\sigma$ would be replaced with the summation of the individual scatterers, each of cross section σ_i , within the volume. The absorption coefficient is generalized to

$$\alpha = \frac{1}{2} \sum_{i=1}^N \sigma_i \quad (8.10.2)$$

Depending on frequency, each of these cross sections may be either equal to, less than, or greater than the *geometric cross section* πa_i^2 , where a_i is the radius of the i th particle.

(a) Fogs

Fog and smoke particles can significantly affect acoustic propagation through the atmosphere. In the immediate neighborhood of suspended particles, thermoviscous losses occur in addition to those in the homogeneous fluid. The equilibrium between each droplet and the surrounding gas is altered by the sound wave and adjustment lags behind the perturbation. This leads to relaxation effects. We will consider two important relaxation effects that contribute to the absorption of a sound wave by a fog: viscous damping of the gross motion of the droplet and thermal damping from heat conduction into and out of the droplet.

Assume a fog consists of identical liquid droplets in a gas through which a plane wave travels in the $+x$ direction. At a coordinate x let the gas have complex particle speed $u(t) = U \exp(j\omega t)$. Let the gas have density ρ_0 , speed of sound c , and shear viscosity η . Let there be N droplets of water per unit volume of gas, each droplet with radius a , density ρ_d , and mass m_d . As the gas oscillates back and forth, viscous forces attempt to carry the droplet along with a speed $d\xi/dt$. In the limit of small *Reynolds number* Re , the viscous force is given by the *Stokes relation* $6\pi\eta a V$, where $V = [d\xi/dt - u(t)]$ is the relative speed of the droplet with respect to the gas.⁹ [The Reynolds number of a flow is defined by the dimensionless ratio $VL/(\eta/\rho)$, where V and L are the characteristic speed and length of the object in the flow and η/ρ is the *kinematic viscosity* of the fluid surrounding the object.] The equation of motion for the displacement ξ of the droplet when the Stokes relation holds is then

$$m_d \frac{d^2 \xi}{dt^2} + 6\pi\eta a \left(\frac{d\xi}{dt} - U e^{j\omega t} \right) = 0 \quad (8.10.3)$$

Solution for the velocity of the droplet in the x direction is immediate,

$$\begin{aligned} \frac{d\xi}{dt} &= \frac{1}{1 + j\omega\tau_{f\eta}} U e^{j\omega t} \\ \tau_{f\eta} &= \frac{2}{9} \rho_d a^2 / \eta \end{aligned} \quad (8.10.4)$$

where $\tau_{f\eta}$ is the relaxation time for viscous damping. The absorption coefficient for viscous losses is found by calculating the work done per unit time by the viscous forces on the N

⁹Stokes, *Trans. Cambridge Philos. Soc.*, 9(2), 8 (1851). Or, see any good book on fluid dynamics.

droplets in each unit volume divided by the energy density of the acoustic wave,

$$2\alpha_{f\eta}c = \frac{\langle 6\pi\eta aN [(d\xi/dt) - u(t)]^2 \rangle_t}{\frac{1}{2}\rho_0 U^2} = \frac{6\pi\eta aN}{\rho_0} \frac{(\omega\tau_{f\eta})^2}{1 + (\omega\tau_{f\eta})^2} \quad (8.10.5)$$

If we define r_m as the ratio of the total mass of water droplets per unit volume of gas to the density of the gas, and recognize that in all fogs of practical interest the density ρ_f of the fog is very similar to that of the gas alone, then

$$r_m = \frac{4}{3}\pi a^3 \rho_d N / \rho_0 \approx \rho_f / \rho_0 - 1 \quad (8.10.6)$$

With the approximation $\omega/c \approx k$,

$$\alpha_{f\eta}/k = \frac{1}{2}r_m\omega\tau_{f\eta}/[1 + (\omega\tau_{f\eta})^2] \quad (8.10.7)$$

which has the same dependence on $\omega\tau$ as has α_M/k in (8.6.14).

As the temperature of the gas fluctuates with the density fluctuations, the droplets heat and cool, disturbing the temperature field in the surrounding gas. The thermal conductivity of the liquid in the droplets is much greater than that of the gas, so we can assume that each droplet has uniform temperature that is a function of time. Let the ambient temperature of the gas at a large distance from a droplet be T_∞ and the internal temperature of the droplet be T_d . The change $\Delta q_d/\Delta t$ in the internal energy per unit volume of the droplet is related to the change of its internal temperature by (8.4.3),

$$\frac{\Delta q_d}{\Delta t} = c_{\mathcal{P}d}\rho_d \frac{dT_d}{dt} \quad (8.10.8)$$

where $c_{\mathcal{P}d}$ is the specific heat at constant pressure for the droplet. This rate of energy loss from the surrounding gas must determine its local temperature field T by (8.4.4),

$$c_{\mathcal{P}d}\rho_d \frac{dT_d}{dt} = \kappa \nabla^2 T \quad (8.10.9)$$

with κ the thermal conductivity of the gas. This is *Poisson's equation*. The left side is the equivalent of a charge density distributed uniformly throughout the volume of the droplet and the temperature T is the equivalent of the electrical potential. By analogy, solution for the temperature throughout the volume exterior to the droplet with the boundary condition that the temperature approach T_∞ at large distances is

$$T(r) = T_\infty - \frac{c_{\mathcal{P}d}\rho_d a^3}{3\kappa r} \frac{dT_d}{dt} \quad (r \geq a) \quad (8.10.10)$$

(The mass of the droplet has been expressed in terms of its density and volume.) The second boundary condition is $T(a) = T_d$ and this yields the differential equation for the droplet temperature,

$$\frac{dT_d}{dt} = \frac{3\kappa}{c_{\mathcal{P}d}\rho_d a^2} (T_\infty - T_d) \quad (8.10.11)$$

In the presence of a plane traveling wave of particle velocity amplitude U and angular frequency ω , the ambient temperature is

$$T_\infty = T_{eq} + T_{eq}(\gamma - 1)(U/c)e^{j\omega t} \quad (8.10.12)$$

and comparison of (8.10.11) and (8.10.12) with (8.6.3), (8.6.6), and (8.6.7) shows that the droplet temperature is

$$\mathbf{T}_d = T_{eq} + \frac{T_{eq}(\gamma - 1)U/c}{1 + j\omega\tau_{f\kappa}} e^{j\omega t} \quad (8.10.13)$$

$$\tau_{f\kappa} = \frac{1}{3} c_{\mathcal{P}d} \rho_d a^2 / \kappa$$

where $\tau_{f\kappa}$ is the relaxation time for the thermal losses. Calculation of the absorption coefficient measuring the thermal losses resulting from the droplets proceeds in much the same way as the calculation for bulk thermal losses in Section 8.4. The complex temperature field external to the droplet is found from substituting (8.10.12) and (8.10.13) into (8.10.10),

$$\mathbf{T}(r) = T_\infty - T_{eq}(\gamma - 1) \frac{U}{c} \frac{a}{r} \frac{j\omega\tau_{f\kappa}}{1 + j\omega\tau_{f\kappa}} e^{j\omega t} \quad (8.10.14)$$

The gradient of the real temperature $T(r)$ is substituted into (8.4.7) and the volume taken over all space *exterior* to the droplet and averaged over one period of the motion. This, multiplied by N , the number of droplets per unit volume, and then divided by the energy density of the plane wave, $\frac{1}{2} \rho_0 U^2$, gives

$$\frac{1}{\mathcal{E}} \frac{d\mathcal{E}}{dt} = -2\alpha_{f\kappa} c = -\frac{4\pi a N}{\rho_0} (\gamma - 1) \frac{\kappa}{c_{\mathcal{P}}} \frac{(\omega\tau_{f\kappa})^2}{1 + (\omega\tau_{f\kappa})^2} \quad (8.10.15)$$

Use of (8.10.6), the relaxation time from (8.10.13), and $T_{eq}(\gamma - 1) = c^2/c_{\mathcal{P}}$ for a perfect gas yields

$$\frac{\alpha_{f\kappa}}{k} = \frac{1}{2} r_m r_c (\gamma - 1) \frac{\omega\tau_{f\kappa}}{1 + (\omega\tau_{f\kappa})^2} \quad (8.10.16)$$

$$r_c = c_{\mathcal{P}d} / c_{\mathcal{P}}$$

The total absorption arising from these two mechanisms is simply the sum,

$$\alpha_f = \alpha_{f\eta} + \alpha_{f\kappa} \quad (8.10.17)$$

Calculation of the phase speed for frequencies well below the relaxation values can be accomplished fairly simply by finding the modified adiabat for the fog resulting from the effective density ρ_f and effective ratio of specific heats γ_f of the fog. Then, the phase speed is found by using $c^2 = \gamma \mathcal{P}_0 / \rho$ and $c_p^2 = \gamma_f \mathcal{P}_0 / \rho_f$. For low frequencies, the droplets are carried along with the motion and can be considered as simply a more massive constituent of the gas. Under this assumption, the effective density is simply the static value $\rho_f = \rho_0(1 + r_m)$. The modified ratio of specific heats can be obtained similarly. At frequencies much less than the relaxation values, the liquid in the droplets can be considered to be essentially in thermal equilibrium with the gas. With these two approximations, the effective specific heats of the fog are found from

$$\begin{aligned} \rho_f c_{\mathcal{P}f} &= \rho_0 c_{\mathcal{P}} + \rho_0 r_m c_{\mathcal{P}d} \\ \rho_f c_{Vf} &= \rho_0 c_V + \rho_0 r_m c_{Vd} \end{aligned} \quad (8.10.18)$$

In liquids the ratio of specific heats is nearly unity, so $c_{\mathcal{P}d}/c_{Vd} \approx 1$, and use of $r_m r_c \ll 1$ gives

$$\gamma_f / \gamma = 1 - (\gamma - 1) r_m r_c \quad (8.10.19)$$

for low frequencies. Then, the phase speed c_p is

$$c_p/c = [(\rho_0/\rho_f)(\gamma_f/\gamma)]^{1/2} \approx 1 - \frac{1}{2}r_m[1 + (\gamma - 1)r_c] \quad (\omega \ll \omega_\eta \text{ and } \omega_\kappa) \quad (8.10.20)$$

A more general derivation,¹⁰ of substantially greater difficulty, shows that the phase speed is given by

$$\frac{c_p}{c} = 1 - \frac{1}{2}r_m \left[\frac{1}{1 + (\omega\tau_{f\eta})^2} + \frac{(\gamma - 1)r_c}{1 + (\omega\tau_{f\kappa})^2} \right] \quad (8.10.21)$$

The phase speed c_p is less than c at low frequencies, having the asymptotic value obtained from (8.10.20). It increases relatively rapidly for frequencies around the relaxation values, until at higher frequencies it becomes asymptotic to c from below. This is consistent with simple physical arguments. For frequencies much less than the relaxation values the above discussion is valid. For frequencies above the relaxation values, (1) there is less thermal energy exchange between the droplets and the gas so that $\gamma_f \rightarrow \gamma$, and (2) the droplets cannot drift along with the surrounding gas and are effectively frozen out of the motion so that the effective density of the moving medium reduces to that of the gas alone. Thus, for frequencies far above resonance the behavior of the phase speed should approach that of the gas alone.

(b) Resonant Bubbles in Water

High attenuations are also produced in water containing air bubbles when the ensonifying acoustic wave causes the bubbles to resonate. A resonating bubble vibrates at such a large amplitude that an appreciable amount of energy is reradiated out of the sound beam and the temperature variations are sufficiently great to cause significant energy losses. The effects of viscosity are of much less importance. The presence of air bubbles also affects the density and compressibility of the medium, changing its speed of sound and resulting in a considerable amount of acoustic energy being reflected and refracted away from the direction of the initial sound beam. Thus, a beam of sound can be attenuated by reflection, refraction, absorption, and scattering as it enters water containing a high concentration of air bubbles. Although air bubbles do not occur in large numbers in the main body of the ocean, high concentrations are found in the wakes of ships, at shallow depths when waves are breaking at the surface, and at various depths where there are regions of intense biological activity.

When ensonified with an incident pressure p of frequency f , a bubble of radius a at depth z can be driven into radial oscillations. Let the *outward* displacement of the surface of the bubble be ξ . The overpressure within the bubble is given by the adiabat (5.5.6), $p = -\rho_b c_b^2 \Delta V/V$, where $\Delta V/V$ is the condensation within the bubble and ρ_b and c_b are for the enclosed air. For a perfect gas, (5.6.3) gives $\rho_b c_b^2 = \gamma \mathcal{P}_b$ with $\mathcal{P}_b$ the total hydrostatic pressure within the bubble. (For very small bubbles, surface tension increases the hydrostatic pressure, but for the sizes of interest here this effect can be ignored without introducing much error.) The condensation within the bubble is $\Delta V/V$ and the compressive force on the surface of the bubble is $f = -4\pi a^2 p$. Combination of these equations yields $f = -12\pi a \gamma \mathcal{P}_b \xi$, so that the effective stiffness s of the bubble is

$$s = 12\pi a \gamma \mathcal{P}_b \quad (8.10.22)$$

For bubbles of specified radius a , the bubble stiffness depends on the depth. The hydrostatic pressure in seawater increases by nearly 1 atm for each 10 m of depth, so we can write the

¹⁰Temkin and Dobbins, *J. Acoust. Soc. Am.*, **40**, 317 (1966).

equilibrium pressure within the bubble as

$$\mathcal{P}_b = \mathcal{P}_0(1 + z/10) \quad (8.10.23)$$

for z in meters.

When the bubble is oscillating, it acts as a small pulsating sphere and radiates sound uniformly in all directions. It therefore has the radiation impedance for a pulsating sphere in the low-frequency limit. As seen in Chapter 7, this is

$$\begin{aligned} \mathbf{Z}_r &= R_r + j\omega m_r \\ R_r &= 4\pi a^2 \rho_0 c (ka)^2 \\ m_r &= 4\pi a^3 \rho_0 \end{aligned} \quad (8.10.24)$$

with R_r and m_r the radiation resistance and mass, respectively, and ρ_0 , c , and k the density, speed of sound, and propagation constant for water. It is clear that the mass of the bubble can be ignored since it is much less than the radiation mass. Thus, a bubble of radius a at depth z acts like a harmonic oscillator with resonance angular frequency $\omega_0 = \sqrt{s/m_r}$, which gives

$$\omega_0 = \frac{1}{a} \left(\frac{3\gamma \mathcal{P}_b}{\rho_0} \right)^{1/2} \quad (8.10.25)$$

Evaluating (8.10.25) at the surface gives $k_0(0) a = 0.0136$, so the radius of the resonant bubble is clearly much less than the acoustical wavelength at resonance. In this frequency range, we see that $R_r \ll \omega m_r$.

In addition to the loss by radiation, there is also absorption of energy by the bubble. Because of the high thermal conductivity of water and the smallness of the bubble, the condensation of the air is not adiabatic. Analysis is involved,¹¹ but the result is that the bubble can be characterized by an additional (mechanical) resistance R_m given by

$$R_m / \omega m_r = 1.6 \times 10^{-4} \sqrt{\omega} \quad (8.10.26)$$

This is accurate within about 10% for bubbles resonant below about 50 kHz. Losses from viscosity are negligible in this frequency range. Thus, the total input mechanical impedance $\mathbf{Z}$ of the bubble is

$$\mathbf{Z} = (R_m + R_r) + j(\omega m_r - s/\omega) \quad (8.10.27)$$

This describes the bubble as a simple harmonic oscillator with two sources of energy loss, one arising from the radiation of acoustic energy and the other the result of thermal conduction between the bubble and the surrounding water. The driving force is $4\pi a^2 \mathbf{p}$, where $\mathbf{p}$ is the complex acoustic pressure provided by the incident plane traveling wave, and the particle speed $\mathbf{u}_r(a, t)$ describes the radial motion of the bubble surface. The quality factor of the bubble resonance is

$$Q = \frac{\omega_0 m_r}{R_m + R_r} = \frac{1}{k_0 a + 1.6 \times 10^{-4} \sqrt{\omega_0}} \quad (8.10.28)$$

For a bubble of radius $a = 0.065$ cm just beneath the ocean surface, the resonance frequency is 5 kHz and $Q = 24$.

¹¹Devin, *J. Acoust. Soc. Am.*, **31**, 1654 (1959).

walls of a rectangular enclosure is $L_M = 4V/S$, so the number of reflections per second is $N = cS/4V$. With each reflection, the sound is reduced in energy (on the average) by the factor $(1 - \bar{a}_E)$, where $\bar{a}_E$ is the *area-averaged random-incidence energy absorption coefficient*,

$$\bar{a}_E = \frac{1}{S} \sum_i S_i a_{Ei} \quad (12.4.1)$$

and a_{Ei} is the random-incidence energy absorption coefficient of the i th surface. The total attenuation of the energy over a time interval of one reverberation time T must therefore be $(1 - \bar{a}_E)^{NT}$. This must correspond to a reduction of 60 dB in the sound pressure level, so that $10 \log(1 - \bar{a}_E)^{NT} = -60$. Solution for T and use of the above expression for N yields the *Eyring–Norris reverberation formula*,

$$T = \frac{0.161V}{-S \ln(1 - \bar{a}_E)} \quad (12.4.2)$$

Expanding $\ln(1 - \bar{a}_E)$ for small $\bar{a}_E$ and comparing the Sabine and Eyring formulas show that for highly reverberant rooms ($\bar{a} \ll 1$) we can make the identification

$$a = \bar{a}_E \quad (12.4.3)$$

The Sabine absorptivity and the random-incidence energy absorption coefficient for a specific surface can be taken as the same. The Eyring and Sabine predictions for T are identical for small $\bar{a}$, but for large $\bar{a}$ the Eyring formula predicts a smaller value.

If the power reflection coefficient R_Π is known for each angle of incidence θ , then the relationship between $R_\Pi(\theta)$ and $a_E(\theta)$ is

$$a_E(\theta) = 1 - R_\Pi(\theta) \quad (12.4.4)$$

So that, from (6.1.7), $a_E(\theta)$ is the power transmission coefficient T_Π . With the help of (12.2.2) it is straightforward to see that for a diffuse incident field

$$a_E = 2 \int_0^{\pi/2} a_E(\theta) \sin \theta \cos \theta d\theta \quad (12.4.5)$$

While it is usual to assume a diffuse sound field, there is no guarantee that this is the case. If the angular dependence of the power reflection coefficient is known, the diffuse field absorptivity can be evaluated from (12.4.5). If the distribution of rays is not diffuse the integral can be modified appropriately. Otherwise, the absorptivity can be measured according to Sabine's method in a reverberant chamber. The results may not be in agreement. In particular, as mentioned earlier, the reverberation chamber measurement can lead to absorptivities in excess of unity. There is still discussion about the relative merits of the different methods of determining the effective a_E for a surface, but many investigators recommend the use of Sabine's equation and the experimentally determined Sabine absorptivities for most practical applications.⁵

⁵Young, J. *Acoust. Soc. Am.*, **31**, 912 (1959). *Noise and Vibration Control*, ed. Beranek, McGraw-Hill (1971).

Further discussions of these and other formulas for reverberation times are available.⁶

12.5 SOUND ABSORPTION MATERIALS

Table 12.5.1 provides absorptivities and absorptions for various materials. For more information, refer to the bulletin "Performance Data: Architectural—Acoustic Materials," published annually by the Acoustical and Insulating Materials Association, and to the sources cited in footnote 2.

Sound absorbers important in acoustic design can be loosely classified as (1) porous materials, (2) panel absorbers, (3) cavity resonators, and (4) individual people and items of furniture.

1. *Porous materials*, such as acoustic tiles and plasters, mineral wools (fiberglass), carpets, and draperies are networks of interconnected pores within which viscous losses convert acoustic energy into heat. The absorptivities of such materials are strong functions of frequency, being relatively small at the lower frequencies and increasing to relatively high values above about 500 Hz. The absorptivities increase with increasing material thickness. Low-frequency absorption can be increased by mounting the material away from the wall. Painting acoustic plasters and tiles will invariably result in a substantial reduction in effectiveness.
2. A *nonporous panel* mounted away from a solid backing vibrates under the influence of an incident sound, and the dissipative mechanisms in the panel convert some of the incident acoustic energy into heat. Such absorbers (gypsum sheetrock, plywood, thin wooden paneling, etc.) are quite effective at low frequencies. The addition of a porous absorber in the space between the panel and the wall will further increase the efficiency of the low-frequency absorption.
3. A *cavity resonator* consists of a confined volume of air connected to the room by a narrow opening. It acts like a Helmholtz resonator, absorbing acoustic energy most efficiently in a narrow band of frequencies near its resonance. These absorbers may be in the form of individual elements, such as concrete blocks with slotted cavities. Other forms consist of perforated panels and wood lattices spaced away from a solid backing with absorption blankets in between. Besides allowing for free architectural expression, these provide useful absorption over a wider frequency range than is possible with individual cavity elements.
4. Table 12.5.1 also includes the *sound absorption per item* for clothed people, upholstered seats, and wooden furniture. Wooden furniture includes chairs with very little upholstery, school desks, and tables (a table providing work space for five people counts as five tables). For widely dispersed audiences with wooden desks, tables, or chairs (as are found in sparsely filled classrooms and many lecture halls), it may be more appropriate to use the absorption per body and per article of furniture rather than the audience absorptivity.

⁶A good starting point is the *Encyclopedia of Acoustics*, ed. Crocker, Wiley (1997); in particular, the chapters by Tohyama (Chap. 77), Kuttruff (Chap. 91), and Bies and Hansen (Chap. 92).

Table 12.5.1 Representative Sabine absorptivities and absorptions

Description	Frequency (Hz)					
	125	250	500	1000	2000	4000
	Sabine Absorptivity α					
Occupied audience, orchestra, chorus	0.40	0.55	0.80	0.95	0.90	0.85
Upholstered seats, cloth-covered, perforated bottoms	0.20	0.35	0.55	0.65	0.60	0.60
Upholstered seats, leather-covered	0.15	0.25	0.35	0.40	0.35	0.35
Carpet, heavy on undercarpet (1.35 kg/m ² felt or foam rubber)	0.08	0.25	0.55	0.70	0.70	0.75
Carpet, heavy on concrete	0.02	0.06	0.14	0.35	0.60	0.65
Acoustic plaster (approximate)	0.07	0.17	0.40	0.55	0.65	0.65
Acoustic tile on rigid surface	0.10	0.25	0.55	0.65	0.65	0.60
Acoustic tile, suspended (false ceiling)	0.40	0.50	0.60	0.75	0.70	0.60
Curtains, 0.48 kg/m ² velour, draped to half area	0.07	0.30	0.50	0.75	0.70	0.60
Wooden platform with airspace	0.40	0.30	0.20	0.17	0.15	0.10
Wood paneling, 3/8–1/2 in. over 2–4 in. airspace	0.30	0.25	0.20	0.17	0.15	0.10
Plywood, 1/4 in. on studs, fiberglass backing	0.60	0.30	0.10	0.09	0.09	0.09
Wooden walls, 2 in.	0.14	0.10	0.07	0.05	0.05	0.05
Floor, wooden	0.15	0.11	0.10	0.07	0.06	0.07
Floor, linoleum, flexible tile, on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Floor, linoleum, flexible tile, on subfloor	0.02	0.04	0.05	0.05	0.10	0.05
Floor, terrazzo	0.01	0.01	0.02	0.02	0.02	0.02
Concrete (poured, unpainted)	0.01	0.01	0.02	0.02	0.02	0.02
Gypsum, 1/2 in. on studs	0.30	0.10	0.05	0.04	0.07	0.09
Plaster, smooth on lath	0.14	0.10	0.06	0.04	0.04	0.03
Plaster, smooth on lath on studs	0.30	0.15	0.10	0.05	0.04	0.05
Plaster, 1 in. damped on concrete block, brick, lath	0.14	0.10	0.07	0.05	0.05	0.05
Glass, heavy plate	0.18	0.06	0.04	0.03	0.02	0.02
Glass, windowpane	0.35	0.25	0.18	0.12	0.07	0.04
Brick, unglazed, no paint	0.03	0.03	0.03	0.04	0.05	0.07
Brick, smooth plaster finish	0.01	0.02	0.02	0.03	0.04	0.05
Concrete block, no paint	0.35	0.45	0.30	0.30	0.40	0.25
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Concrete block, smooth plaster finish	0.12	0.09	0.07	0.05	0.05	0.04
Concrete block, slotted two-well	0.10	0.90	0.50	0.45	0.45	0.40
Perforated panel over isolation blanket, 10% open area	0.20	0.90	0.90	0.90	0.85	0.85
Fiberglass, 1 in. on rigid backing	0.08	0.25	0.45	0.75	0.75	0.65
Fiberglass, 2 in. on rigid backing	0.21	0.50	0.75	0.90	0.85	0.80
Fiberglass, 2 in. on rigid backing, 1 in. airspace	0.35	0.65	0.80	0.90	0.85	0.80
Fiberglass, 4 in. on rigid backing	0.45	0.90	0.95	1.00	0.95	0.85
	Sound Absorption A in m ²					
Single person or heavily upholstered seat (± 0.10 m ²)	0.40	0.70	0.85	0.95	0.90	0.80
Wooden chair, table, furnishing, for one person	0.02	0.03	0.05	0.08	0.08	0.05

Proper choice of the amounts and distributions of these classes of absorbers can tailor the behavior of the reverberation time with frequency to obtain almost any desired acoustic environment. Since the optimum reverberation time depends on the use of the room, it is possible to design multipurpose rooms with sliding or rotating panels that expose surfaces of different absorption properties. However, artificial reverberation introduced electronically can be a less expensive and more flexible solution, especially in large rooms.

13.3 SPEECH INTERFERENCE

Noise decreases the intelligibility of speech by raising the listener's threshold of hearing while, at the same time, masking the information. This loss of information may be partially compensated for by moving closer, talking louder, or using electronic amplification.

Fortunately, speech is highly redundant. Usually, much of a sentence can be lost without seriously affecting intelligibility; meaning can still be extracted from context. To measure intelligibility, trained talkers recite, clearly and distinctly, specially selected words or sentences to trained listeners. Intelligibility is then rated according to the percent of correct responses. The intelligibility of isolated words is more strongly affected by noise but improves markedly with increasing numbers of syllables; disyllabic words are understood about twice as easily as monosyllabic words with the same background noise. Figure 13.3.1 shows the intelligibility of sentences and words as a function of the relative A-weighted sound levels of speech and noise. For a *sentence* intelligibility of better than 95%, the signal level must at least equal the noise level. Since it is difficult to reconstruct proper names from contexts, a paging system should have *word* intelligibility over 85%. This requires a signal-to-noise ratio of 6 dB—a specification seldom achieved in airport lobbies let alone bus depots.

The *voice level* (VL) is the *A-weighted SPL* recorded at a distance of 1 m in front of the speaker. For untrained voices, it appears that (within uncertainties of a few dBA) a quiet conversational voice corresponds to about $VL = 57$ dBA, a normal speaking voice to 64 dBA, a raised voice to 70 dBA, a very loud voice to 77 dBA, and a shout to about 83 dBA.

A measure of intelligibility suitable for field use is the *speech interference level* (SIL). This is the arithmetic average of the noise levels (in dB, *not* dBA) in the four octave bands centered at 500, 1000, 2000, and 4000 Hz. In the absence of all other information, the A-weighted sound level can be used to get a *rough* estimate of the intelligibility of speech under most commonly encountered noise conditions. A SIL estimated by

$$SIL \approx L_A - 7 \quad (13.3.1)$$

will be in error by less than 4 dB for all but the most pathological noise spectra.

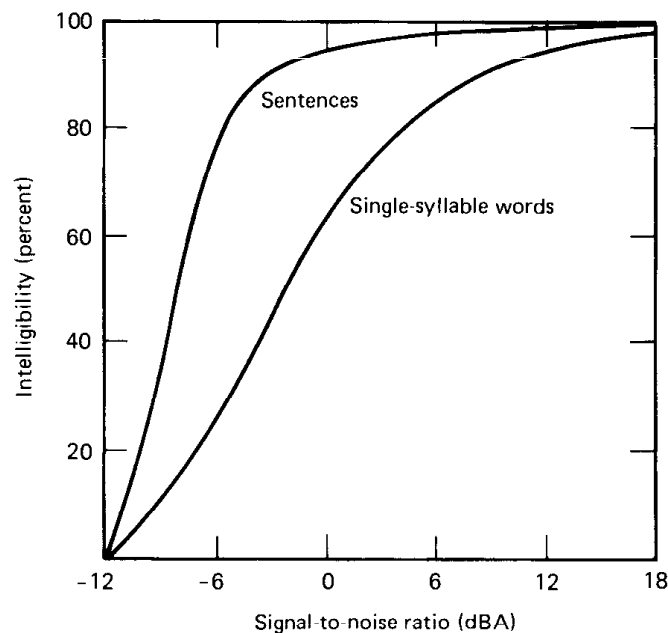


Figure 13.3.1 Percent of words and sentences correctly identified in the presence of background noise.

For people speaking at voice levels natural for their environment, and not trying to achieve more than *just reliable* communication (approximately 1 out of 20 sentences unintelligible), a set of criteria relating distance and the maximum *SIL* have been established.⁵ For separation distances $r > 8$ m

$$SIL \leq -20 \log r + 58 \quad (13.3.2)$$

is required for the male voice. For smaller separations, disproportionately higher *SILs* can be accepted since people naturally progressively raise their voices as the *SIL* grows beyond 40 dB. For $r < 8$ m, the relationship for just-reliable communication at natural voice levels is

$$SIL \leq -29 \log r + 66 \quad (13.3.3)$$

with the voice level rising from normal for a $SIL = 40$ dB to very loud for a $SIL = 82$ dB. Approximate expressions for the minimum required voice levels to ensure just-reliable communication are

$$\begin{aligned} VL &\geq SIL + 20 \log r + 6 & r > 8 \text{ m} \\ VL &\geq \frac{4}{3}(SIL + 20 \log r) - 13 & r < 8 \text{ m} \end{aligned} \quad (13.3.4)$$

For female voices, the *SILs* should be decreased by around 5 dB as noted in Section 11.10. If the required voice level is below 64 dBA, communication conditions are satisfactory or better. If the *VL* lies between 64 and 70 dBA then conditions are acceptable, between 70 and 77 dBA difficult, and above 77 dBA impractical to impossible.

13.4 PRIVACY

In addition to speech intelligibility, speech privacy in multifamily dwellings, private offices, open-plan offices, and classrooms is important not only to protect confidentiality but also to prevent intrusion on the privacy of others. The degree of privacy depends on the sound insulation of intervening walls and barriers and also on the background noise. The insulative properties of partitions will be discussed in Section 13.13; here we will consider only the effect of background noise. Figure 13.3.1 shows that if the A-weighted noise in the receiving room is 9 dBA above the A-weighted speech level transmitted through the wall, only about 10% of the words and 30% of the sentences will be understood. This should ensure adequate privacy with little distraction in an office, but an even lower speech level would be required in a dwelling unit.

Privacy can be enhanced by (1) decreasing the amount of sound that passes through the wall (build a better wall), (2) decreasing the speech level in the source room (add acoustic absorption), or (3) increasing the noise level in the receiving room (mask by unobtrusive noise like flow noise through a ventilator screen). A procedure for calculating the degree of privacy is discussed in Beranek.⁶

⁵ANSI S-3.14-1977(R-1986). See Tocci, *Encyclopedia of Acoustics*, Chap. 94, and Crocker, *Encyclopedia of Acoustics*, Chap. 80, for more detailed discussion.

⁶Beranek and Ver, *op. cit.*

13.5 NOISE RATING CURVES

A noise that is acceptable for speech interference may be annoying for other activities (reading, listening to music, or sleeping). While two continuous noises with the same frequency spectrum can be rated by comparing their A-weighted sound levels, time-varying noises with different spectra require a rating procedure that accounts not only for the intermittent nature of the noise but also for its subjective qualities.

The procedure generally accepted for rating steady broadband noise containing no significant tones, such as produced by ventilating systems and distant road traffic, consists of comparing measured octave-band levels with a family of criteria curves.

Several procedures for rating steady noise are in general use, differing only in the shapes of their criteria curves. One of the currently accepted methods uses the *balanced noise criterion* (NCB) curves suggested by Beranek.⁷ These curves are shown in Fig. 13.5.1.

To determine the rating of a noise, its octave-band levels are plotted on the same graph as the criteria curves. The highest curve reached by any octave-band level of the noise gives the NCB rating of the noise. The dashed line in Fig. 13.5.1 is the spectrum of a noise that is rated at $NCB = 40$.

Determining the sonic quality of the noise spectrum requires two other procedures: (1) Calculate the *SIL* of the noise as described earlier. If the level of any octave band having center frequency at or below 1 kHz exceeds the NCB curve of value $SIL + 3$, the noise will appear “rumbly.” (2) Calculate the average level of the octave bands centered at 125, 250, and 500 Hz and identify the NCB curve that intersects that level at 250 Hz. If any band level for the octaves with center frequencies equal to or greater than 1 kHz lies above that NCB curve, the noise will sound “hissy.”

The maximum noise rating recommended for a room depends not only on the intended use of the room but also on such factors as the expectations of the users. For example, an urban dweller will tolerate significantly higher ambient noise levels than will a rural dweller. In fact, the urban dweller may be bothered by low noise levels because sounds otherwise masked may be clearly heard and prove

Table 13.5.1 Recommended acceptable noise levels in unoccupied rooms

<i>Location</i>	<i>Noise Criteria (NCB)</i>
Concert hall, recording studio	10–15
Music room, legitimate theater	25–30
Church, courtroom, conference room, hospital, bedroom	25–35
Library, private office, living room, classroom	30–40
Restaurant, movie theater, retail shop, bank	35–45
Gymnasium, clerical office	40–50
Shops, garage	50–60

distracting. Table 13.5.1 shows recommended acceptable noise ratings for a variety of different room uses.

⁷Beranek, J. *Acoust. Soc. Am.*, **86**, 650 (1989).

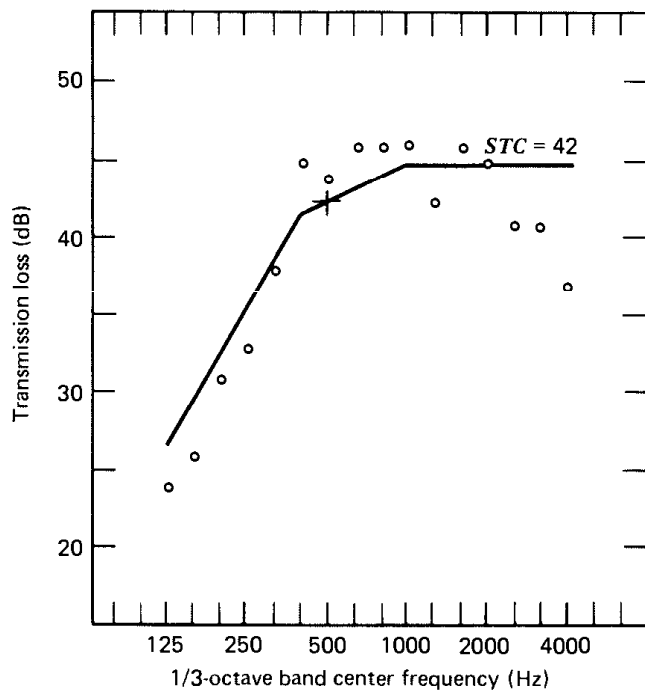


Figure 13.13.1 Determination of the sound transmission class (STC) from transmission loss (TL) measurements. The rating for this wall is $STC = 42$. (The TL at 4 kHz is 8 dB below the $STC = 42$ curve and the total deficiency is 30 dB.)

Table 13.13.1 Sound transmission class (STC) for representative wall constructions^a

Construction	Mass per Unit Area (kg/m ²)	STC
1. 4 in. hollow block, $\frac{1}{2}$ in. plaster on both sides	115	40
2. 4 in. brick, $\frac{1}{2}$ in. plaster on both sides	210	40
3. 9 in. brick, $\frac{1}{2}$ in. plaster on both sides	490	52
4. 24 in. stone, $\frac{1}{2}$ in. plaster on both sides	1370	56
5. $\frac{3}{8}$ in. gypsum wallboard	8	26
6. $\frac{1}{2}$ in. gypsum wallboard	10	28
7. $\frac{5}{8}$ in. gypsum wallboard	13	29
8. Two $\frac{1}{2}$ in. gypsum wallboards bonded together	22	31
9. 2 × 4 studs on 16 in. centers, $\frac{1}{2}$ in. gypsum wallboard on both sides	21	33
10. Same as 9 but with $\frac{5}{8}$ in. gypsum wallboard on both sides	26	34
11. Same as 10 but with two sheets of $\frac{5}{8}$ in. gypsum wallboard on one side and one sheet on the other side	42	36
12. Same as 10 but with $\frac{1}{2}$ in. plaster over wallboard	68	46
13. Same as 9 but with a 2 in. isolation blanket	23	36
14. Same as 10 but with a 2 in. isolation blanket	29	38
15. Same as 11 but with a 2 in. isolation blanket	44	39
16. Same as 14 but with one side resiliently mounted	29	47
17. Same as 14 but with both sides resiliently mounted	29	49
18. Double row of 2 × 4 studs on 16 in. centers, $\frac{5}{8}$ in. gypsum wallboard on both sides, and 2 in. isolation blanket	37	57
19. Double row of 2 × 4 studs on 16 in. centers, two $\frac{5}{8}$ in. gypsum wallboard on both sides, no isolation blanket	60	58
20. Same as 19 but with 2 in. isolation blanket	60	62

^a Numbers 1 to 15 are single-leaf construction. Numbers 16 to 20 are double-leaf construction.

Because of the different spectra associated with external noise, *STC* ratings cannot be used directly to predict isolation values of outside walls. However, a technique does exist that allows the *STC* to be used for this purpose. See Harris (*op. cit.*).

Besides their *STC*, floor/ceiling partitions should be considered for their *impact isolation class (IIC)*. This is a measure of their ability to isolate a room from noise produced by impact on the floor above. Again, see Harris (*op. cit.*).

13.14 RECOMMENDED ISOLATION

The amount of acoustic isolation required between two rooms depends on the noise level in the source room and the level of intrusive noise acceptable in the receiving room. Both these levels depend on the designated use of the rooms and the latter level depends on the ambient noise that will tend to mask the intrusive noise.

The recommendations of the Federal Housing Administration account for the difference in background noise by defining three grades of buildings.

Grade 1. Buildings with exterior nighttime noise levels lower than 40 dBA and recommended interior noise levels lower than 35 dBA.

Grade 2. Buildings with recommended interior noise levels of 40 dBA or lower.

Grade 3. Buildings with exterior nighttime noise levels of 55 dBA or higher and interior noise levels of 45 dBA or higher.

For walls separating different apartments, they recommend an *STC* of 55, 52, and 48 for building Grades 1, 2, and 3, respectively. For walls between rooms of the same dwelling unit, they recommend for a Grade 1 building the following: bedroom to bedroom *STC* = 48; living room to bedroom 50; bathroom to bedroom, kitchen to bedroom, and bathroom to living room 52. The recommendations for Grade 2 are 4 dB lower and those for Grade 3 are another 4 dB lower.

For walls separating apartment rooms from commonly shared service spaces (garages, laundries, party rooms), the following minimum requirements have been suggested: bedroom *STC* = 70, living room 65, and kitchen and bathroom 60.

Recommendations also exist for the impact isolation class for floors separating dwellings.

13.15 DESIGN OF PARTITIONS

Ignoring flanking and leakage, the basic mechanism of sound transmission through a wall is that sound in the source room forces the exposed surface to vibrate; this vibration is transmitted through the structure of the wall to the other surface, which in turn vibrates producing sound in the receiving room. If the two surfaces of the wall are rigidly connected so that they vibrate as a unit (a single-leaf partition), the transmission loss depends only on the frequency and the mass per unit area, stiffness, and intrinsic damping of the wall. If the partition consists of two unconnected walls separated by a cavity (a double-leaf partition), the transmission loss depends on the properties of the two walls and on the size of the cavity and its absorption.

(a) Single-Leaf Partitions

For a plane, nonporous, homogeneous, flexible wall the transmission loss is dependent on the density of the partition and the frequency of the noise. The energy ΔE_i that strikes area ΔS of the partition in the time interval Δt was found to be (12.2.2)

$$\Delta E_i = \frac{\mathcal{E} \Delta S \Delta r}{2} \int_0^{\pi/2} \sin \theta \cos \theta d\theta \quad (13.15.1)$$

The energy ΔE_t transmitted through the partition is the same integral, but with $T_{\Pi}(\theta)$ multiplying the integrand,

$$\Delta E_t = \frac{\mathcal{E} \Delta S \Delta r}{2} \int_0^{\pi/2} T_{\Pi}(\theta) \sin \theta \cos \theta d\theta \quad (13.15.2)$$

where the transmission coefficient is given by (6.7.4),

$$T_{\Pi}(\theta) = \frac{1}{1 + a^2 \cos^2 \theta} \quad (13.15.3)$$

$$a = \omega \rho_s / 2 \rho_0 c = \pi f \rho_s / \rho_0 c$$

where ρ_s is the surface density of the partition and ρ_0 and c are for the air. Taking the ratio of (13.15.1) and (13.15.2) and integrating (13.15.1) gives an integral expression for the random incidence transmission coefficient,

$$\langle T_{\Pi} \rangle_{\theta} = \frac{\Delta E_t}{\Delta E_i} = 2 \int_0^{\pi/2} \frac{\sin \theta \cos \theta}{1 + a^2 \cos^2 \theta} d\theta \quad (13.15.4)$$

The change of variable $u = a \cos \theta$ allows immediate evaluation,

$$\langle T_{\Pi} \rangle_{\theta} = \frac{2}{a^2} \int_0^a \frac{u}{1 + u^2} du = \frac{1}{a^2} \ln(1 + a^2) \quad (13.15.5)$$

The transmission loss TL , defined in (13.13.1), is then

$$TL = -10 \log \langle T_{\Pi} \rangle_{\theta} = 20 \log a - 10 \log[\ln(1 + a^2)] \quad (13.15.6)$$

As seen from Fig. 13.15.1, values of ρ_s range from about 10 to 1000 kg/m² but cluster around 20 to 50 kg/m². Frequencies of interest run from about 60 Hz to 4 kHz, and $\rho_0 c$ for air is 415 Pa · s/m. Values of a thus span $4 < a < 3 \times 10^4$. Substitution shows that the last term on the right in (13.15.6) has values between about -5 and -13 dB, with the majority falling around -8 dB. As a conservative estimate, it is appropriate to assume the smallest deduction, so that the acoustic transmission through the partition is overestimated. Taking -5 dB and expressing $\rho_0 c$ and π numerically gives

$$TL = 20 \log(f \rho_s) - 47 \quad (13.15.7)$$

with frequency in Hz and surface density in kg/m². This *mass law* predicts that doubling the mass per unit area at a given frequency or doubling the frequency for a given mass per unit area increases the TL by 6 dB.

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GLOSSARY OF SYMBOLS

(continued from front endpapers)

r	specific gas constant; characteristic acoustic impedance; specific acoustic resistance	SSL	source spectrum level
		STC	sound transmission class
		S_V	scattering strength per unit volume
r_t	transition range	SWR	standing wave ratio
r_s	skip distance	$\mathcal{S}$	transmitter sensitivity
R	resistance (acoustic, electrical, mechanical); reflection coefficient; radius of curvature	$\mathcal{S}\mathcal{L}$	transmitter sensitivity level
		$\mathcal{S}_{ref}$	reference transmitter sensitivity
Re	Reynolds number	T	period of motion; temperature; tension; transmission coefficient; reverberation time
$\dot{R}$	range rate		
R_m	mechanical resistance		
R_r	radiation resistance	T_I	intensity transmission coefficient
R_I	intensity reflection coefficient	T_K	temperature in kelvin
R_{Π}	power reflection coefficient	T_{em}, T_{me}	transduction coefficients
RL	reverberation level	T_{Π}	power transmission coefficient
ROC	receiver operating characteristic	TL	transmission loss
$\mathcal{R}$	universal gas constant	TNI	traffic noise index (dBA)
s	spring constant; condensation	TS	target strength
sL	apparent source level	TS_R	target strength for reverberation
S	cross-sectional area; surface area; salinity	TTS	temporary threshold shift
S_A, S_B	scattering strengths per unit area	$\mathcal{T}$	membrane tension per unit length
$SENEL$	single event noise exposure level L_{ex} (dBA)	$\vec{u}$	particle velocity
SIL	speech interference level	u	particle speed
SL	source level	U	peak particle velocity amplitude; volume velocity
SPL	sound pressure level	U_e	effective particle velocity amplitude
SS	sea state		

$\vec{v}$	scaled particle velocity ($\vec{u}/c$)	Θ	phase angle of impedance
V	volume; voltage; effective voltage amplitude; volume displacement	κ	thermal conductivity; radius of gyration; transverse component of the propagation vector
VL	voltage level; voice level (dBA)	λ	wavelength
V_{ref}	reference effective voltage amplitude	Γ	c_0 times the eikonal; Goldberg number
ω	angular frequency; bandwidth	ξ	longitudinal particle displacement
W	explosive yield	Ξ	amplitude of ξ
x	specific acoustic reactance	Π	time-averaged power
X	electrical reactance	Π_i	instantaneous power
X_m	mechanical reactance	ρ	instantaneous density (kg/m^3); probability density function
X_r	radiation reactance	ρ_0	equilibrium density
y	transverse displacement	ρ_L	linear density (kg/m)
Y	admittance; Young's modulus	ρ_S	surface density (kg/m^2)
z	specific acoustic impedance	σ	Poisson's ratio; standard deviation; extinction cross section
Z	impedance (acoustic, electrical, mechanical)	σ_S	scattering cross section
Z_m	mechanical impedance	τ	relaxation time; pulse duration; processing time
Z_r	radiation impedance	ϕ	transformation factor; phase angle; torsion angle; turns ratio; inverse turns ratio
α	spatial absorption coefficient	Φ	velocity potential
β	temporal absorption coefficient; one-half the flair constant; a nonlinearity coefficient	ω	angular frequency (rad/s)
γ	ratio of heat capacities; attenuation coefficient; a nonlinearity coefficient	ω_0	natural angular frequency
δ	boundary layer thickness; skin depth	ω_d	damped angular frequency
η	coefficient of shear viscosity; efficiency	ω_u, ω_l	upper, lower half-power angular frequencies
η_B	coefficient of bulk viscosity	Ω	solid angle
η_e	effective coefficient of viscosity	Ω_{eff}	effective solid angle
θ	angle of incidence; phase angle; grazing angle; horizontal beam width; angle of elevation or depression	$\delta(v)$	Dirac delta function of argument v
		δ_{nm}	Kronecker delta
		$1(v)$	Heaviside unit function of argument v (unit step function)