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Noise Control

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Contents

1. INTRODUCTION

Most people think acoustics applies only to rooms with special functions, such as concert halls or churches. Actually, any space has acoustical qualities, and if these qualities are inappropriate, the utility of the space may be compromised. Normally, noise problems are associated with sounds that people can hear. However, ultrasonic and infrasonic sounds can also produce psychological effects and, under certain conditions, definite physiological effects.

Several examples are appropriate and will serve to illustrate the range of acoustical problems often encountered. Most of us are aware that some spaces must be quiet to be useful. If intrusive noise were present in a bedroom, school study hall, or library reading room, for example, it would be difficult to use that facility as it was intended. Sometimes, even "quiet" can be inappropriate. A sports arena or nightclub would be dull places to visit if no noise were present. There is a middle ground too. Open plan office spaces need to be moderately quiet to allow for pleasant working conditions, but not so quiet that private conversation would be impossible.

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The acoustic properties of these spaces, and indeed of any space, are determined by the geometry of the space and of the materials within it. The factors governing the behavior of sound within spaces are well understood and can be used to ensure that the spaces function well acoustically. This function falls into the professional disciplines of architectural acoustics or noise control. Because noise control rests on the factors affecting the behavior of sound, it is pertinent to discuss some of the physics involved.

2. THE PHYSICS OF SOUND

2.1. *Sound*

Sound is a disturbance that propagates through an elastic medium (air, water, etc.) at a speed characteristic of that medium. Noise and its control can encompass a wide range of mediums such as underwater noise from ocean traffic or unwanted vibrations in mechanical structures. For the purposes of this text, noise will refer to airborne disturbances unless otherwise noted. When a sound source in air vibrates, it causes the air to oscillate, which, in turn, produces extremely small changes in the pressure of the surrounding air. The pressure waves spread out like ripples on a pond when a stone is dropped into it, except that sound waves fill the whole volume of air, whereas ripples are confined to the surface of the pond.

2.2. *Speed of Sound*

In a free field, sound propagates with the velocity c defined by

$$c = 20.05\sqrt{T_K}\ (m/s) \tag{1}$$

or

$$c = 49.03\sqrt{T_R}\ (f/s)$$

where T_K and T_R are the temperature in Kelvin and Rankine, respectively (1).

A simpler formula for the velocity of sound in air sufficiently accurate at normal temperatures, 0–30ºC, is

$$c = 331 + 0.6T_C\ (m/s) \tag{2}$$

where T_C is the temperature in centigrade.

Example 1

Determine the speed of sound at 20ºC (68ºF) in both metric and English units.

Solution:

The Kelvin temperature is T_K = 273.2 + 20 = 293.2 K and the Rankine temperature is T_R = 459.7 + 68 = 527.7ºR

The speed of sound c is then

$$c = 20.05\sqrt{293.2} = 343\ \ m/s$$

or

$$c = 49.03\sqrt{527.7} = 1125\ \ ft/s$$

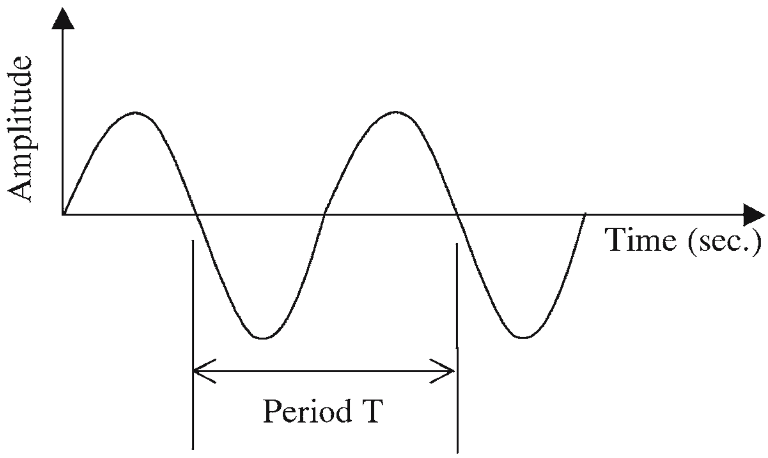

Fig. 1. Harmonic oscillation of pressure.

2.3. *Sound Pressure*

Sound waves produce changes in the density of the medium (air) as they travel through it. These changes in air density cause pressure fluctuations around the ambient static pressure. If air particles oscillate in a harmonic mode (sinusoidally varying with time), sound pressure will also change harmonically and cause a pure tone.

For a pure tone, the sound pressure p can be described as

$$p = a \sin(\omega t) = a \sin(2\pi f t) \text{ Pascals} \tag{3}$$

where a is the amplitude in Pascals, ω is the angular frequency in radians per second, t is the time in seconds, and f is the frequency in hertz.
The angular frequency is defined as

$$\omega = 2\pi f \text{ rad/s} \tag{4}$$

Figure 1 shows a pure tone oscillation, although pure tones do not often exist in nature. Even musical instruments do not produce pure tones. Instead, the sounds they emit consist of a fundamental tone and a number of harmonics. The harmonics occur at integer multiples of the fundamental frequency and they confer on an instrument its special character. A buzz saw, too, is rich in harmonics.

In most situations, the disturbances created in the air that we call sound cannot easily be expressed mathematically in time and space. This results from the fact that over an extended portion of a vibrating surface creating the airborne disturbance, some portions are compressing the surrounding air while other portions are causing rarefactions. This results in a phase difference in the pressure in either time or space. Complicating the situation further, physical boundaries cause reflections and allow the disturbances to interact with each other. Fortunately, the sound pressure in a sound field will in most cases vary in a random fashion, and statistical techniques can therefore be used to deal with the phenomenon.

The field of acoustics and noise control has nearly uniformly adopted the metric system throughout and, as such, the unit used for measuring sound pressure is the Pascal (Pa). In earlier years, the units bar, microbar, and dyne per square centimeter were used to measure sound pressure.

The following conversion factors apply:

1 bar = 10^5 Pa
1 μbar = 10^{-6} Pa = 1 dyn/cm^2
1 dyn/cm^2 = 10^{-1} Pa
1 Pa = 1 N/m^2

The sound pressure in a sound wave can be measured with a microphone. The electric signal generated by the microphone is typically amplified and recorded onto an oscilloscope or other recording medium. A detailed picture of the sound wave can be produced by an oscilloscope; however, sufficient information can usually be obtained by a continuous display of the instantaneous value of the sound pressure as a function of time only. Consequently, it is possible to use simpler equipment such as a sound-level meter, available at electronics stores, rather than an oscilloscope to analyze noise.

2.4. *Frequency*

The frequency of a sound indicates the number of cycles performed in 1 s:

$$f = 1/T \text{ Hz} \tag{5}$$

where T is the period of one full cycle. The unit for frequency is the hertz (Hz):

1 Hz = 1 cycle/sec = 1 cps
1000 Hz = 1 kilohertz = 1 kHz

The audible frequency range to humans, 16–20,000 Hz, has been divided into a series of octave bands and one-third (1/3) octave bands. Just as with an octave on a piano keyboard, an octave in sound analysis represents the frequency interval between a given frequency and twice that frequency. The interval is identified by the center frequency, representing the geometric mean of the bounds of that interval. The internationally agreed upon 1000-Hz center frequency determines the center frequencies of the remaining bands. The center frequencies and approximate cutoff frequencies are listed in Table 1.

2.5. *Wavelength*

The wavelength λ is equal to the distance the oscillations have propagated in the time period T:

$$\lambda = cT = c/f \tag{6}$$

This shows that the wavelength is inversely proportional to the frequency. In the audio frequency range (16–20,000 Hz), the low frequencies have wavelengths of several meters (or feet), whereas the wavelengths for the high frequencies are only a few centimeters (or fractions of an inch).

Example 2

Determine the wavelength of a 125-Hz and an 8000-Hz tone at 20ºC (68ºF) in both metric and English units.

Table 1
Center and Approximate Frequency Limits for Octave and One-Third Octave Bands Covering the Audio Frequency Range

Octave			One-Third Octave		
Lower limit (Hz)	Center frequency (Hz)	Upper limit (Hz)	Lower limit (Hz)	Center frequency (Hz)	Upper limit (Hz)
11	16	22	14.1	16	17.8
			17.8	20	22.4
			22.4	25	28.2
22	31.5	44	28.2	31.5	35.5
			35.5	40	44.7
			44.7	50	56.2
44	63	88	56.2	63	70.8
			70.8	80	89.1
			89.1	100	112
88	125	177	112	125	141
			141	160	178
			178	200	224
177	250	355	224	250	282
			282	315	355
			355	400	447
355	500	710	447	500	562
			562	630	708
			708	800	891
710	1,000	1,420	891	1,000	1,122
			1,122	1,250	1,413
			1,413	1,600	1,778
1,420	2,000	2,840	1,778	2,000	2,239
			2,239	2,500	2,818
			2,818	3,150	3,548
2,840	4,000	5,680	3,548	4,000	4,467
			4,467	5,000	5,623
			5,623	6,300	7,079
5,680	8,000	11,360	7,079	8,000	8,913
			8,913	10,000	11,220
			11,220	12,500	14,130
11,360	16,000	22,720	14,130	16,000	17,780
			17,780	20,000	22,390

Source: Data from refs. 1 and 2.

Solution

$$125 \text{ Hz:} \qquad \lambda = \frac{343}{125} = 2.74 \text{ m}$$

$$\lambda = \frac{1125}{125} = 9 \text{ ft}$$

$$8000\text{ Hz:}\qquad \lambda = \frac{343}{8000} = 0.0043\text{ m} = 4.3\text{ cm}$$
$$\lambda = \frac{1125}{8000} = 0.14\text{ ft} = 1.7\text{ in.}$$

2.6. rms Sound Pressure

Most common sounds consist of a rapid, irregular series of positive-pressure disturbances (compressions) and negative-pressure disturbances (rarefactions) measured from the static pressure. Typically, there is no net increase in the ambient pressure as a result of the presence of acoustic disturbances except in certain exotic problems such as sound produced by explosions. Thus, the mean value of any series of sound pressure disturbances is not meaningful, as there are as many compressions as rarefactions and the net acoustic pressure is zero.

The root-mean-square (rms) sound pressure yields a nonzero value to describe the pressure disturbance, and this is more meaningful. Physically, the rms value is indicative of the energy density of the disturbance. Mathematically, the rms value is obtained by squaring the sound pressures at any instant of time and then integrating over the sample time and averaging the results. The rms value is then the square root of this time average:

$$p_{\text{rms}} = \sqrt{\lim_{T\to\infty} \frac{1}{2T} \int_{-T}^{T} p(t)^2 dt} \tag{7}$$

2.7. Sound Level Meter

The rms value of the sound pressure can be measured by a sound-level meter that can typically display overall sound levels or octave band levels in either a linear format or with various weighting functions (dBA, dBC, etc.). Specific details on sound-level meters are covered by various ANSI and IEC standards (3,4).

2.8. Sound Pressure Level

The sound pressures that are normally measured with a sound-level meter cover an extremely large range. The sound pressure of the faintest sound a human ear can hear is equivalent to about 2×10^{-5} Pa and sound pressures that can be measured close to jet engines are equivalent to about 2×10^{2} Pa. The ratio of these two sound pressures is 10^7. In order to handle in a simple fashion such a large measurement range, a logarithmic measurement scale is used. There are other benefits to using such a scale because the human response to sensations (experiences, sounds, fragrances, pains, etc.) corresponds to a logarithmic intensity scale rather than to a linear scale.

The sound pressure level then is a logarithmic ratio L_p defined as

$$L_p = 10\log\frac{p_{\text{rms}}^2}{p_{\text{ref}}^2} = 20\log\frac{p_{\text{rms}}}{p_{\text{ref}}} \tag{8}$$

where p_{rms} is the sound pressure of interest (in Pa) and p_{ref} is a reference sound pressure (in Pa) usually chosen as the limit of hearing of 20 µPa. The unit for the sound pressure level, SPL or L_p, is the decibel (dB) (5).

Table 2
Relation Between Sound Pressure and Sound Pressure Level (SPL or L_p)

Sound pressure (p)	Sound pressure level (L_p)
p_{ref}	0 dB
$1.12p_{\text{ref}}$	1 dB
$1.26p_{\text{ref}}$	2 dB
$2p_{\text{ref}}$	6 dB
$3.16p_{\text{ref}}$	10 dB
$10p_{\text{ref}}$	20 dB
$100p_{\text{ref}}$	40 dB
$10{,}000p_{\text{ref}}$	80 dB
$1{,}000{,}000p_{\text{ref}}$	120 dB

Source: ref. 1.

The relationship between sound pressure and sound pressure level (with 20 μPa as the reference sound pressure) is shown in Table 2.

Example 3

Determine the sound pressure level for sound pressures of p = 1 Pa and p = 1 atm (1.013 × 10^5 Pa) (reference to 20 μPa).

Solution:

$$L_p = 20\log\frac{1}{20\times10^{-5}} = 20\log\frac{10^5}{2} = 94 \text{ dB}$$
$$L_p = 20\log\frac{1.103\times10^5}{20\times10^{-5}} = 20\log\frac{1.103\times10^{10}}{2} = 194 \text{ dB}$$

2.9 Loudness

Sound waves cause the membrane in the ear to vibrate, and these vibrations are transmitted through interconnected small bones to the inner ear, the cochlea. Thousands of hair cells in the cochlea retransmit the sound information to the brain through nerves. A young unspoiled ear can hear pure tones if they have sound pressure levels as a function of frequency as illustrated in the lowest curve in Fig. 2.

For example, as can be seen in Fig. 2, the threshold of hearing at 1000 Hz is about 4 dB. It can also be seen in Fig. 2 that the ear is most sensitive to sound in the frequency range 300–6000 Hz. The sensitivity drops at lower and higher frequencies. The family of curves in Fig. 2 (Fletcher and Munson curves) (6,7) indicate the sound pressure level a tone must have in order to be as loud as the 1000-Hz tone with a sound pressure level as indicated on the curve. A 20-Hz tone must have a sound pressure level about 70 dB higher than a tone at 1000 Hz in order for a person just to hear the tone. The curves also indicated the loudness—the subjective interpretation of the magnitude of sound—for pure tones. The units describing loudness are called phons. By definition, the phon is equal to the sound pressure level (in dB) reference to 20 μPa of an equally loud 1000-Hz tone. Pain will occur when the loudness exceeds 120 phons.

An approximate measurement of the loudness of a pure tone can be made by using a sound-level meter with a weighting network. The weighting network approximates an

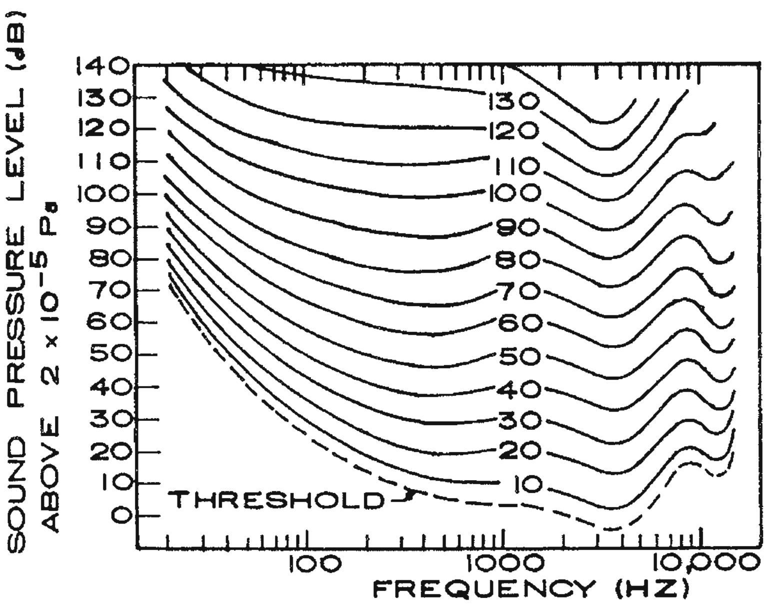

Fig. 2. Normal equal loudness contours for pure tones. (From ref. 6.)

average loudness curve. Most sound-level meters are equipped with weighting networks, A, B, and C (4). The sensitivity of these networks is shown in Fig. 3.

Initially the intent was that for loudnesses in the 30–60-phon range, the A-weighting should be used, whereas in the 60–90-phon range, the B-weighting applies, and for loudnesses above 90 phons, the C-weighting is appropriate.

Because the Fletcher and Munson curves are based on pure tones, measurement of a complicated noise spectrum by a sound-level meter often gives results that vary from the loudness measured in accordance with the definition of the phon. This occurs because the brain sums the loudness of the individual components of the spectrum dif-

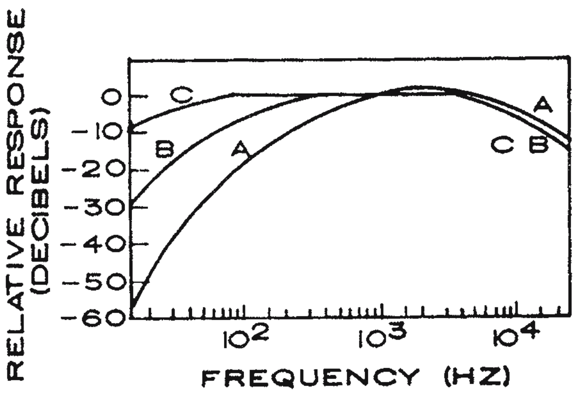

Fig. 3. Frequency response of A-, B-, and C-weighting networks. (From ref. 4.)

Table 3
Sample Sound Pressure Levels (dBA)

Sound pressure level (dBA)	Event
100–110	Jet fly at 300 m (1000 ft)
90–100	Power mower
80–90	Heavy Truck 64 km/h (40 mph) at 15 m (50 ft),food blender (at receiver), motorcycle at 15 m (50 ft)
60–70	Vacuum cleaner (at receiver), air conditioner at 6m (20 ft)
40–50	Quiet residential—daytime
20–30	Wilderness

Source: ref. 7.

ferently from the straight summation performed by the sound-level meter. The sound-level meter gives a value that approximates the rms value of the sound signal, but the function of the brain is much more complicated. For that reason, it is not strictly correct to say that measurements made with a sound-level meter indicate accurately the loudness in phons. All that the reading indicates is the sound level with the A-, B-, or C-weighting. Table 3 shows some typical sound levels of common sound sources in dBA, where dBA indicates that the A-weighting network has been used.

2.10. Sound Power Level

The sound power radiated by a sound source covers an extremely wide range. It is more convenient to express the sound power level L_w using a logarithmic scale based on an internationally selected sound power as a reference. Thus,

$$L_w = 10\log(W/W_0)\text{dB} \tag{9}$$

where $W_0 = 10^{-12}$ watts (W) (5). For many practical problems, the sound power, W, can be expressed as

$$W = \frac{P_{\text{rms}}^2}{\rho c}\, dA \tag{10}$$

where ρ is the ambient density, c is the speed of sound, and dA is the area around the source. One can note that as one moves away from a pointlike source, p_{rms} decays as $1/r^2$, but the sphere (or hemisphere) dA that encloses the source increases as $4\pi r^2$ so that W remains constant. It is for this reason that much noise legislation is worded for power rather than pressure so that an accurate portrait of the noise from a source can be properly specified and verified.

2.11. Sound Energy Density

The sound energy density D is the energy that arises from the sound field present in a small volume of the air in a room. The relation between the space-average mean-square sound pressure and the space-average sound energy density is

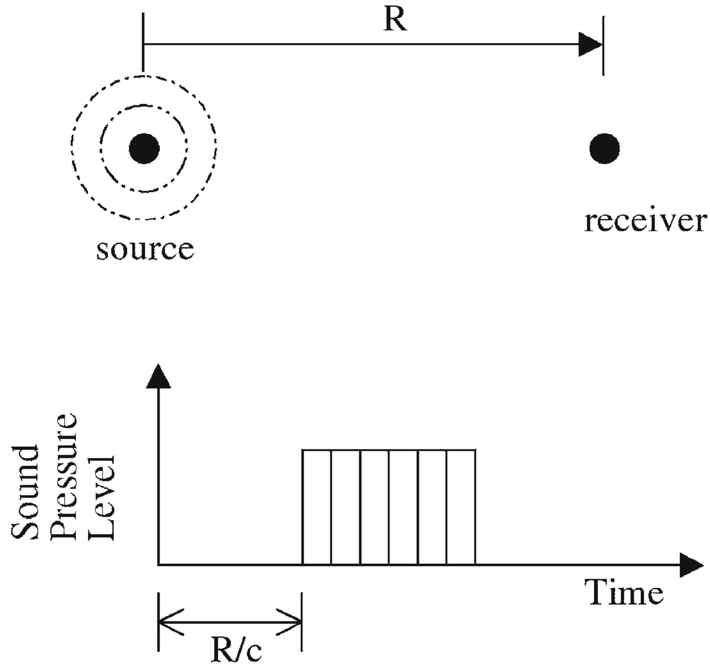

Fig. 4. Sound field out-of-doors. (From ref. 1.)

$$D = \frac{P_{\mathrm{rms}}^2}{\rho c^2} \tag{11}$$

where ρ is the density of the air (kg/m^3).

3. INDOOR SOUND

3.1. Introduction

The acoustical environment in a room depends greatly on the size, shape, and other properties of the confining walls, floor, and ceiling. Before examining how the properties of the room affect the sound field, it is pertinent to examine a simpler situation. What happens when a sound source such as a pistol is fired out-of-doors in the absence of any nearby reflecting surfaces as is seen in Fig. 4. The sound waves will travel through the atmosphere from the pistol to the listener (which acousticians typically call the receiver), who will only hear one crack. The farther the receiver is away, the later the crack will arrive and the weaker it will be. The direct sound wave is the only sound heard.

The sound wave arrives at time t:

$$t = R/c \tag{12}$$

where R is the distance between the source and the receiver and c is the speed of sound.

If there is a large wall or building at a distance near the firing position as evidenced in Fig. 5, the receiver will hear a different sound. The first sound wave will be from the

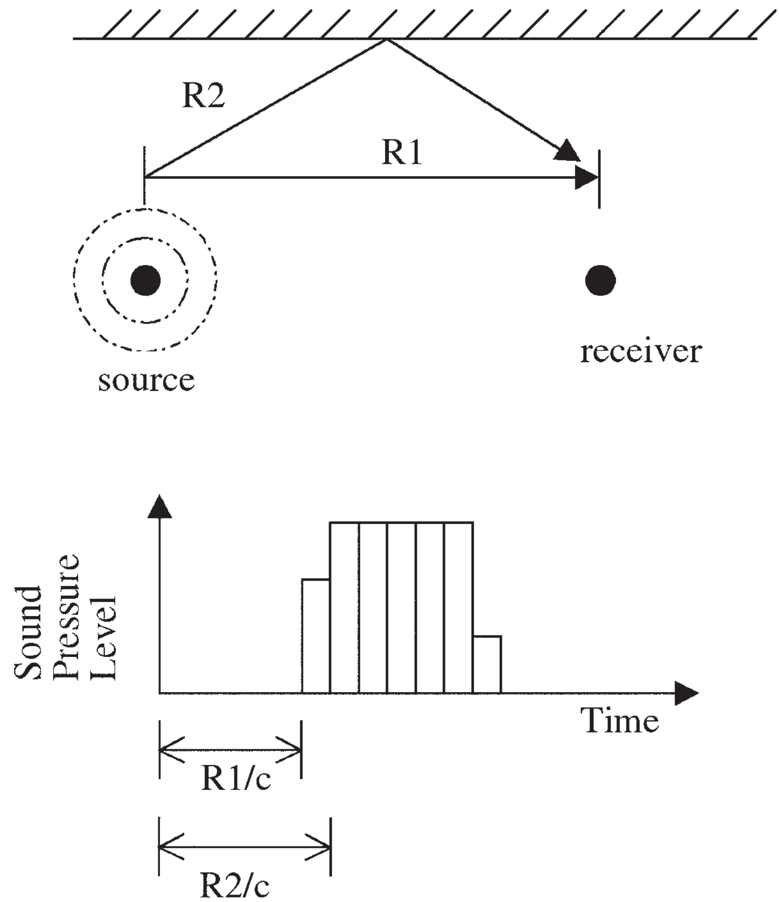

Fig. 5. Sound field out-of-doors near reflecting wall. (From ref. 1.)

direct arrival and the second sound wave from the reflected arrival off of the wall. The two sound waves, will arrive at

$$t_1 = R_1/c \tag{13}$$

and

$$t_2 = R_2/c \tag{14}$$

The human ear is able to recognize individual sound impulses if they are separated by a time period of about 50 ms. If the time difference between the direct sound and the reflected sound is greater, an echo will be heard; if the time difference is less, only one modified crack will be heard, as evidenced in Fig. 5.

Example 4

Determine the time a receiver will hear a pistol shot when the distance between the pistol and the receiver is 200 m and there is a building behind the pistol, 40 m away. Make the calculation for the two temperature conditions −10ºC and 25ºC.

Solution:

−10ºC:

$$c = 20.05\sqrt{T_K} = 20.05\sqrt{263.2} = 325 \text{ m/s}$$
$$t_1 = \frac{200}{325} = 0.615 \text{ s}$$
$$t_2 = \frac{200}{325} = 0.872 \text{ s}$$

25ºC

$$c = 20.05\sqrt{T_K} = 20.05\sqrt{298.2} = 346 \text{ m/s}$$
$$t_1 = \frac{200}{346} = 0.578 \text{ s}$$
$$t_2 = \frac{280}{346} = 0.810 \text{ s}$$

The reflected sound wave is weaker than the direct sound ways for two reasons:

1. It will have traveled farther than the direct sound wave, and the sound intensity decreases with distance.
2. Some sound energy will be lost in the process of reflection.

It is often advantageous to think of the reflected sound as coming from the mirror image of the sound source. This can be done if the reflecting surface is large compared to the wavelength of the sound and if any irregularity on the reflecting surface is small compared to the wavelength.

If the pistol is fired in an enclosed space and the receiver is inside the room, he or she will then receive a number of impulses one after the other. In a room, however, the individual impulses tend to blend together because they occur in such rapid succession. The individual impulses are difficult to separate for a second reason to. We can imagine that the reflections are caused by fictitious mirror sound sources of first, second, and higher order, with the higher orders representing successive reflections. The higher the order of the mirror image, the weaker the sound source because source energy will be lost every time the sound wave reflects.

3.2. Sound Buildup and Sound Decay

Until now, only impulsive sound has been discussed. What happens when sounds of longer duration are present? Figure 4 can again be used to examine the outdoor situation previously described. If a sound source is placed in a position where it emits sound for 6 s, a receiver in position *R* will receive sound for a period of 6 s. However, the greater the distance between source and receiver, the weaker the sound and the later the receiver starts hearing the sound. If there is a large wall near either the sound source or the receiver, a different picture will be seen. As shown in Fig. 5, the receiver will initially only hear the direct sound. Then a weaker reflected sound will arrive, and the sound level will increase. After 6 s, the direct sound will disappear and only the reflected sound will be heard until that also disappears.

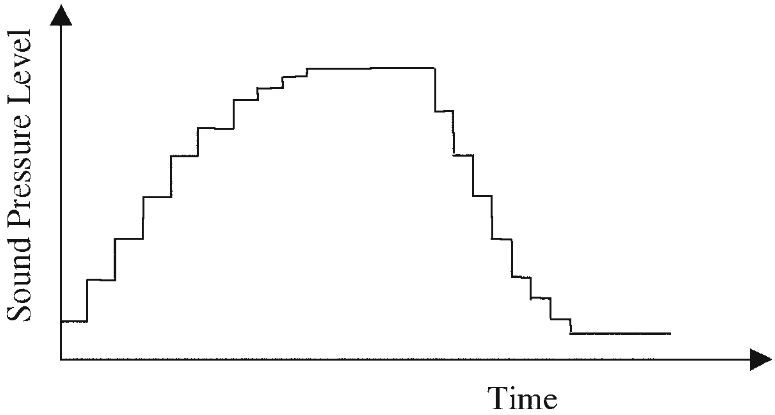

Fig. 6. Buildup and decay of sound field in a room. (From ref. 1.)

The changes in the sound field in an enclosed space, when a continuous sound source is turned on and off, are shown in Fig. 6. The situation is similar to the outdoor example, only many more reflecting surfaces are present.

In the first half of Fig. 6, every step up results from the arrival of a new reflection. Expressed another way, every step up corresponds to the contribution from one of the mirror images. The farther away the mirror images, the later the contribution arrives.

The individual steps on the curve gradually become smaller and smaller because the mirror images lie farther and farther away and the intensity of the sound wave gets reduced at each reflection. A stationary or steady-state situation represented by the horizontal portion of the curve in Fig. 6 is rapidly reached.

When the sound source is switched off, the sound level drops off, following a similar step function because the contribution from the individual mirror images gradually disappears. The time it takes to reach the stationary situation and the time it takes from the moment the sound source is switched off until the sound has disappeared depend solely on the acoustical properties of the room. Thus, the rise time or the decay time can be utilized to characterize a room acoustically. It should be noted that Fig. 6 is presented for qualitative purposes and that actual build up and decay curves may be smoother or quite spikey indeed based on the geometry and contents of the room such as furniture or people.

The equations used to determine the acoustical properties are usually developed according to statistical techniques; thus, several equations exist, each differing slightly as a result of the different statistical assumptions made. However, all equations are based on the energy balance that will always exist.

Before developing the equations determining the sound buildup and sound decay in a room, it will be necessary to calculate how much sound energy will be incident on a wall element dS in 1 s. To do so, consider any volume element dV at a distance R from the wall element dS at such a direction that the perpendicular to dS makes the angle θ with R, as seen in Fig. 7.

Because the surface area of the sphere (radius R) centered at dS is $4\pi R^2$ and because the projection of dS on the surface of the sphere is $dS \cos\theta$, then the fraction

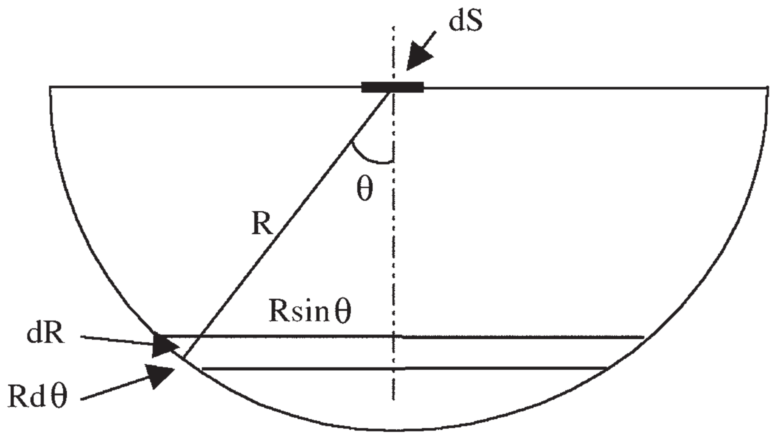

Fig. 7. Figure for use in the calculation of sound energy incident on a wall element per second. (From ref. 1.)

$(dS\cos\theta)/4\pi R^2$ of the total sound energy passing through the infinitesimal volume toward the wall ($D\,dV$) will be incident on dS. (It is assumed that the sound energy in the volume element dV will propagate equally well in all directions.)

All volume elements dV located within a hemisphere of radius c and centered at dS will transmit sound energy to the wall element dS during a 1-s time period. Thus, τ, the sound energy incident on the wall element dS, can be obtained by integrating over the hemisphere.

For the volume element dV, we will use the rotational element at a distance R from dS at the angle θ between R and dS's perpendicular. From Fig. 7, it can be seen that $dV = 2r\pi\sin\theta\, d\theta\, R\, dR$

$$\begin{aligned}\tau &= \int_0^c \int_0^{\pi/2} \frac{dS\cos\theta}{4\pi R^2} D2\pi R\sin\theta \;d\theta\, dR \\ &= \int_0^c \int_0^{\pi/2} \frac{1}{2} DdS\;\cos\theta\;\sin\theta\, d\theta dR \\ &= \frac{1}{4} DcdS\end{aligned} \tag{15}$$

The absorption coefficient α_n for a wall element S_n is defined as the proportion of incident sound energy that is absorbed. This means that the expression for absorbed energy is $(1/4)Dc\alpha_n S_n$ .The energy balance is

Added energy − Absorbed energy = Change in energy

Mathematically, this becomes

$$W - \frac{1}{4} Dc\sum \alpha_n S_n = V\frac{dD}{dt} \tag{16}$$

where W is the emitted sound power in (watts), D is the energy density (watts-s/m^3), c is the speed of sound (m/s), V is the room volume (m^3) and α is the absorption coefficient defined as the percentage of impinging energy absorbed on any given wall element of area S.

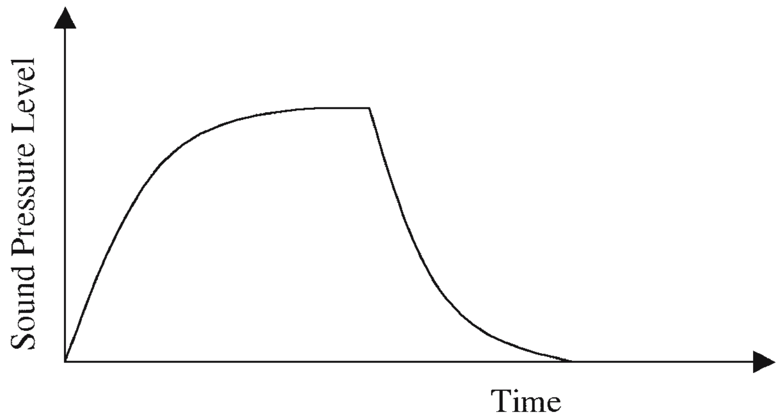

Fig. 8. Sound buildup and decay in room (exponential curve). (From ref. 1.)

Statistical techniques must be applied to the middle term of the above equation that expresses the amount of energy absorbed per unit time. If we assume that at any time the sound field is diffuse (that the energy intensity is everywhere equal), then during sound buildup and decay, the solution to Eqn. (16) is as follows:

Sound buildup:

$$D = D_0(1 - \exp(-cAt/4V)) \text{ (W-s/m}^3\text{)} \tag{17}$$

Sound decay:

$$D = D_0 \exp(-cAt/4V) \text{ (W-s/m}^3\text{)} \tag{18}$$

where D_0 represents the energy density once equilibrium is reached and A is the total absorption in the room equal to $\Sigma \alpha_n S_n$. During the stationary (equilibrium) period,

$$D_0 = 4W/cA \text{ (W-s/m}^3\text{)} \tag{19}$$

In the next section, the way in which these equations are put to use will be amplified, but before doing so, it is pertinent to make some additional comments. First, examine Fig. 8, which represents both the sound buildup and decay according to the exponential expressions just developed.

These curves approximate the physical process involved; however, they do not represent the way we perceive the process. The ears seldom hear the sound buildup but do hear the sound decay. Because the human perception is more strongly related to the sound decay, the main emphasis has been placed on that phenomenon, (i.e., sound decay). The second comment pertains to the conditions under which the diffuse sound field assumption is valid.

3.3. *Diffuse Sound Field*

Only large rooms, where the number of normal modes of vibration is large, can be considered to have a diffuse sound field. In the diffuse sound field, the average energy density will be the same at all points. This means that there would be no net flow of power in any direction. Because we will always have a net flow of power away from a source to the places where the energy is absorbed, we will actually never have a diffuse

Table 4
Values for Air Absorption, 4*m*.

		4*m* (dB/1000 m)							
Relative humidity(%)	Temperature (ºC)	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000Hz	8000 Hz
25	15	0.2	0.6	1.3	2.4	5.9	19.3	66.9	198
	20	0.2	0.6	1.5	2.6	5.4	15.5	53.7	180.5
	25	0.2	0.6	1.6	3.1	5.6	13.5	43.6	153.4
	30	0.1	0.5	1.7	3.7	6.5	13.0	37.0	128.2
50	15	0.1	0.4	1.2	2.4	4.3	10.3	33.2	118.4
	20	0.1	0.4	1.2	2.8	5.0	10.0	28.1	97.4
	25	0.1	0.3	1.2	3.2	6.2	10.8	25.6	82.2
	30	0.1	0.3	1.1	3.4	7.4	12.8	25.4	72.4
75	15	0.1	0.3	1.0	2.4	4.5	8.7	23.7	81.6
	20	0.1	0.3	0.9	2.7	5.5	9.6	22.0	69.1
	25	0.1	0.2	0.9	2.8	6.5	11.5	22.4	61.5
	30	0.1	0.2	0.8	2.7	7.4	14.2	24.0	68.4

Note: The evaluation of 4m in linear terms can be found from the above values divided by 4340.
Source: ref. 9.

sound field. However, the concept of a diffuse sound field is useful in rooms with not too much absorption and where the measurement positions are not close to either highly absorption surfaces or the sound source.

3.4. Reverberation Time

The relationship expressed in the preceding discussion is incorporated into the concept of reverberation time. The reverberation time t in a room is defined as the time it takes the sound pressure level to decrease to a value 60 dB below its original value. The pioneer of room acoustics. W.C. Sabin (8) developed this definition and derived the following formula based on the preceding equations.

$$t = 0.163\frac{V}{A + 4\text{m}V} \quad \text{metric}$$
$$t = 0.049\frac{V}{A + 4\text{m}V} \quad \text{English} \tag{20}$$

where V is the room volume (m^3[ft^3]), A is the total absorption ($\Sigma\alpha_n S_n$), and m is an attenuation coefficient relating to the absorption. The 4m term is often expressed in units of dB/1000 m to highlight its relative influence on sound levels. For use in Eq. (20), the values in Table 4 must be divide by 4340 to provide 4m in units of 1/m.

The total absorption A is defined as

$$A = \Sigma\alpha_n S_n = \alpha_1 S_1 + \alpha_2 S_2 + \cdots \tag{21}$$

where α_1, α_2, · · · are the absorption coefficients of the different wall elements of surface area S_1, S_2, · · ·, respectively. A has the unit m^2-Sabin (ft^2-Sabin) if the unit for S is m^2 (ft^2).

3.5. *Optimum Reverberation Time*

From an examination of the reverberation time formula, it can be seen that the total absorption in a room can be calculated from measured reverberation time. Optimum values exist for the reverberation time for various functional spaces. Thus, if a room is to be modified to change its function, the change in the amount of absorption materials in the room can be determined. Similarly, if a space is to be designed, knowledge of the total amount of absorption within the room can be used to calculate reverberation time and hence determine whether the room will be acoustically satisfactory. Changes can be made in the materials used if necessary. Table 5 presents some examples of optimum reverberation time for different functional spaces.

Table 5
Reverberation Times(s) for Some Common Spaces

Room size (m^3)	Speech or lecture hall	Opera house	Churches
100	0.65	0.85	1.15
1000	0.8	1.25	1.5
10,000	0.95	1.6	1.8
100,000	1.1	1.9	2.2

Source: ref. 1.

3.6. *Energy Density and Reverberation Time*

It is important to recognize, however, that changing the reverberation time also affects energy density. The energy density in a room (away from the individual sound sources, where the sound field is more likely to be diffuse) is directly proportional to the reverberation time. From Eqs. (19) and (20),

$$D_0 = 4W/cA \tag{22}$$

and

$$t = 0.16\ V/A \text{ (assuming } m = 0) \tag{23}$$

The equation representing the relationship can be determined to be

$$D_0 = \frac{W_t}{13.6V} \tag{24}$$

where c is taken as 340 m/s.

This formula clearly shows that the energy density in the room can only be reduced (and a noise-reduction achieved) either by reducing the sound power emitted to the room or by a reduction of the reverberation time—the last being equivalent to an increase in the total absorption.

In rooms, such as industrial areas, where short reverberation times are desirable, the energy density will be low also. In such a case, there is no problem, because both characteristics are desirable. In music rooms, however, where high energy density and medium reverberation time are assets, some compromise may be necessary.

3.7. Relationship Between Direct and Reflected Sound

It was determined that the energy density in the steady-state situation was

$$D_0 = 4W/cA \text{ W-s/m}^3 \tag{19}$$

Close to the sound source, however, the energy density will be larger than D_0 because of the contribution D_d from the sound source. Assuming the sound source radiates evenly in all directions, then

$$D_d = W/c4\pi R^2 \text{ W-s/m}^3 \tag{25}$$

The ratio of D_d to D_0 becomes

$$D_d/D_0 = A/16\pi R^2 \tag{26}$$

and as long as that ratio is greater than 1, we are in the direct sound held. This means that D_d is dominant when

$$R^2 \leq A/16\pi R^2 \tag{27}$$

This distance R can be used to determine how far away from a machine an operator must be in order to achieve a noise reduction in a room solely through the use of absorption materials. The total energy density in a room is given by

$$D = D_0 + D_\mathrm{d} = \frac{4W}{cA} + \frac{W}{c4\pi R^2} = \frac{W}{c}\left[\frac{4}{A} + \frac{1}{4\pi R^2}\right] \tag{28}$$

The sound pressure level, SPL reference to 20 μPa, can be determined by inserting Eq. (28) into Eq. (11) and utilizing Eq. (8):

$$\mathrm{SPL} = \mathrm{PWL} + 10\log\left[\frac{1}{4\pi R^2} + \frac{4}{A}\right] \tag{29}$$

where

$$\mathrm{PWL} = 10\log W/W_0 \text{ [see Eq. (9)]}$$

Example 5

Determine the distance R for which the direct energy density is the main contributor in two different rooms.

Room 1: Volume $V = 300$ m^3
Reverberation time $t = 1.0$ s

Room 2: Volume $V = 20{,}000$ m^3
Reverberation time $t = 2.5$ s

Solution

Room 1:

$$A = \frac{0.16 \times V}{t} = \frac{0.16 \times 300}{1.0} = 48 \text{ m}^2\text{-Sabin}$$

$$R^2 \leq \frac{48}{16\pi} = 0.95 \text{ m}$$

$$R \leq 1 \text{ m}$$

Room 2:

$$A = \frac{0.16 \times V}{t} = \frac{0.16 \times 20000}{2.5} = 1280 \text{ m}^2\text{-Sabin}$$

$$R^2 \le \frac{1280}{16\pi} = 25.5 \text{ m}$$

$$R \le 5 \text{ m}$$

4. SOUND OUT-OF-DOORS

4.1. Sound Propagation

One major difference between sound inside an enclosed space and sound outdoors is that the effect of multiple reflecting surfaces outdoors is usually not as significant as indoors.

At a distance R from a point source, the energy density will be equal to

$$D = \frac{W}{c \times 4\pi R^2} \tag{30}$$

The sound pressure level SPL reference to 20 µPa can be determined by inserting Eq. (30) into Eq. (11) and utilizing Eq. (8):

$$\text{SPL} = \text{PWL} - 10 \log 4\pi R^2 \text{ dB} \tag{31}$$

Implicit in this equation is the inverse square law, which states that for each doubling of distance the sound pressure level will drop off by 6 dB. This is only true if there are no winds, no temperature gradients, and no reflecting surfaces.

4.2. Wind and Temperature Gradients

Another major difference between sound inside an enclosed space and sound outdoors is the influence of meteorology. Under normal circumstances, the wind velocity will change as a function of distance above the ground. If the wind velocity changes with the height above the ground, the sound velocity is also changed. This causes the sound waves to be turned toward the ground in the downwind direction and away from the ground in the upwind direction, as indicated in Fig. 9.

A temperature change above the ground also influences the propagation of the sound waves. If the temperature increases with height, the sound waves will be turned toward

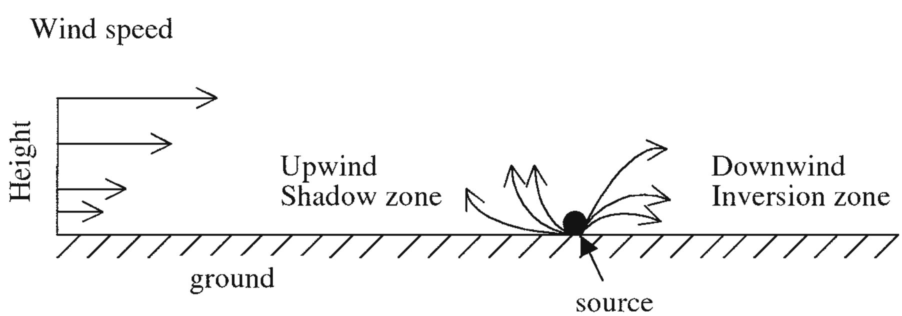

Fig. 9. The influence of wind direction and wind velocity on sound propagation. (From ref. 1.)

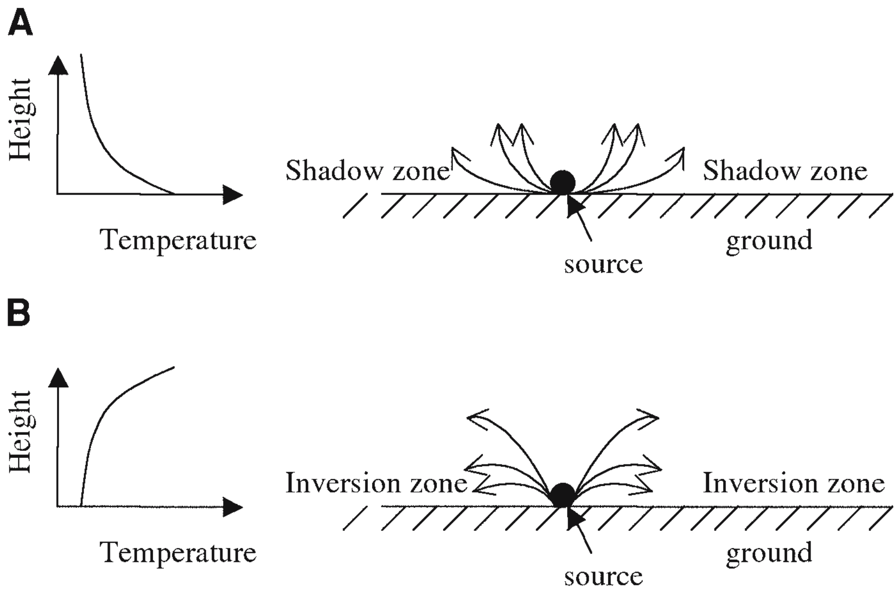

Fig. 10. The influence of temperature change on sound propagation: **(A)** increasing temperatures by height; **(B)** decreasing temperatures by height. (From ref. 1.)

the ground although the opposite is true for temperatures, which decrease with height, as shown in Fig. 10.

4.3. Barriers

The propagation of sound outdoors is affected by land forms such as mountains, hills, dikes, and so forth. These natural as well as artificial barriers can be effective in reducing received sound, provided that the barriers have no holes and that they block the direct path of sound from the source to the receiver.

Four primary parameters determine how effective a barrier is in reducing sound: (1) source-to-barrier distance, (2) barrier-to-receiver distance, (3) the height of the barrier and (4) the length of the barrier. Figure 11 shows a method that can be used to estimate the reduction obtainable from a barrier (10). To use the figure, the four parameters involved are combined into a single descriptor—the increase in sound path length. This descriptor is then related directly to sound reduction. The method is limited to barriers that can be regarded as rigid and infinitely long. Figure 11 gives the average noise reduction that can be obtained for broadband noise. For low frequencies the attenuation is lower, and for high frequencies the attenuation is higher.

As shown in Fig. 11, the source-to-receiver distance is D. With the barrier in place, the shortest source-to-receiver distance becomes $a + b$. The difference $(a + b) - D$ is the parameter δ, the sound path length difference. For a situation in which $\delta = 1.0$, for example, the sound reduction for an infinitely long barrier would be about 17 dB when metric units are used and about 12 dB when English units are used.

Recent research has focused on absorptive and partial length barriers (9). A more flexible, and complicated, treatment of barriers comes from Kurze and Anderson (11,12). In this

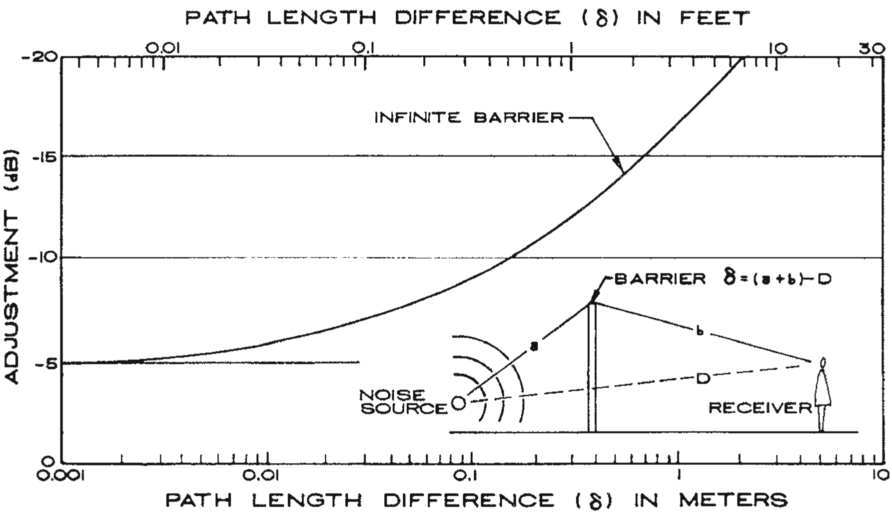

Fig. 11. Average noise reduction for an acoustic barrier of infinite length. (From refs. 1 and 10.)

model, four parameters combine into one variable, the Fresnel number N, to determine how effective a barrier is in reducing sound. They are (1) the source-to-barrier distance, (2) the barrier-to-receiver distance, (3) the height of the barrier and (4) the frequency of interest. Figure 12 is a sketch of the barrier problem to be utilized in Eq. 32 that describes the insertion loss IL or the change in sound level before and after the barrier is added:

$$
\begin{aligned}
&N = \frac{2f(A+B-C)}{c} \\
&\mathrm{IL} = 0, \qquad N < -0.1916 - 0.635b' \\
&\mathrm{IL} = 5(1+0.6b') + 20\log\frac{\sqrt{-2\pi N}}{\tan\sqrt{-2\pi N}}, \qquad -0.1916 - 0.635b' \le N \le 0 \\
&\mathrm{IL} = 5(1+0.6b') + 20\log\frac{\sqrt{2\pi N}}{\tanh\sqrt{2\pi N}}, \qquad 0 < N < 5.03 \\
&\mathrm{IL} = 20(1+0.15b'), \quad N \ge 5.03
\end{aligned}
\tag{32}
$$

where b' is 0 for a wall and 1 for a berm (earthen barrier). This method is also limited to barriers that can be regarded as infinitely long but it has the advantage of being frequency dependent. One can check whether a barrier is sufficiently long to qualify as infinite by looking at the flanking paths as a separate diffraction problem. If the insertion loss of the flanking path is sufficiently larger than the path over the top, it can be neglected.

5. NOISE REDUCTION

5.1. Absorptive Materials

A reduction of the sound pressure level in a room can often be efficiently accomplished by reducing the sound power emitted by the sound source. By reducing the

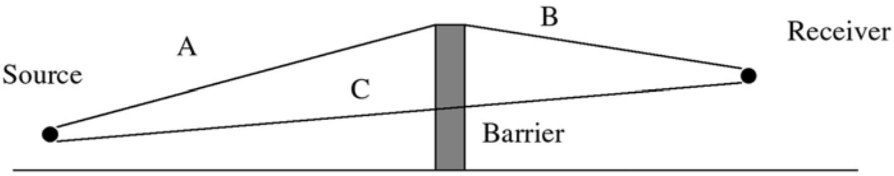

Fig. 12. Geometry for calculation of insertion loss using the method of Kurze and Anderson. (From ref. 12.)

emitted sound power, both the direct and the reflected sound will be reduced. However, it is often not appropriate to modify the sound source because modification may have adverse effects, such as reduced efficiency, interference with access to the machine, and so forth (13–15). In such cases, the energy density can be changed by modifying the sound field. One commonly used modification is obtained by installing absorption materials at strategic places such as the walls and ceiling of the room. The beneficial result of this treatment is reduction of the intensity of the reflected sound (15,16).

In Section 3, we developed the formulas governing energy density, sound power, reverberation time, room volume, and total absorption. In this section, we will consider how much noise reduction can be achieved by the use of absorptive materials.

Let us look at an example where an existing room has a total absorption of A_0. If S square meters of absorptive material is installed with an absorption coefficient α, then the total absorption after the installation will be

$$A = A_0 + \alpha S \text{ m}^2\text{-Sabin} \tag{33}$$

However, only the surface exposed to the incident sound determines the absorption, so if the new material covers any of the older material

$$A = A_0 + S\,(\alpha - \alpha_0) \text{ m}^2\text{-Sabin} \tag{34}$$

where α_0 is the absorption coefficient for the wall that will be covered with the absorptive treatment from Eq. (19). The energy density of the reflected sound will be reduced by

$$\Delta = 10 \log A/A_0 \text{ dB} \tag{35}$$

Since noise is a combination of direct and reflected signals, the noise reduction that can be achieved is highly dependent on the initial amount of absorption in the space. In most practical installations, it will not be possible to achieve more than 5–l0 dB noise reduction. A 10-dB reduction is equivalent to approximately half of the sound loudness, rated subjectively.

By placing absorption materials on room surfaces, the direct sound will not be reduced; therefore, machine operators who are located a few feet from the noisy machine will receive little benefit. However, the contribution to their noise exposure from adjacent machines will be reduced.

Example 6

Determine the noise reduction of the reflected sound that can be obtained in a $20 \times 25 \times 35\,\text{m}^3$ space where the reverberation time is 4.0 s and the existing ceiling material having an absorption coefficient of 0.25 is exchanged with a material having an absorption coefficient of 0.90.

Solution

$$V = 20 \times 25 \times 35 = 17{,}500 \text{ m}^3$$
$$t = \frac{0.16 \times 17{,}500}{A_0}, t = 4.0 \text{ s}$$
$$A_0 = \frac{0.16 \times 17{,}500}{4} = 700 \text{ m}^2\text{-Sabin}$$
$$A = A_0 + S(\alpha - \alpha_0)$$
$$A = 700 + 25 \times 35 \times (0.90 - 0.25)$$
$$A = 700 + 570 = 1270 \text{ m}^2\text{-Sabin}$$

The noise reduction Δ is

$$\Delta = 10 \log \frac{1270}{700} = 10 \log 1.82$$
$$\Delta = 2.6 \text{ dB}$$

When using absorption materials in an attempt to reduce the energy density of the reflected sound, it is beneficial to place the materials as close to the sound source as possible. Good results often occur when a noisy machine is placed near a wall covered with an effective absorption material. Another alternative would be to cover the machine with a barrier furnished with absorptive material. A canopy with absorptive material can be effective if it is suspended close to the machine. It is important to recognize that the absorption coefficient varies with frequency, so that before a selection is made, it will be necessary to know the frequency content of the disturbing noise. In most practical situations, noise-reduction techniques should be aimed at the middle- and high-frequency regions (500–8000 Hz), which is the range where human perception is the greatest. Sounds in this frequency range are typically the most tiring, cause the greatest task interference, and are the most damaging to hearing.

The acoustics of a room are not affected by whether the sound energy is absorbed by a real or an imaginary surface. In this respect, an open window is a very effective sound absorber because it acts as a sink for all of the arriving sound energy. For real surfaces, the sound energy can either be converted to heat or transmitted as sound waves in the building structure. Sound-absorbing materials can, on the basis of the physical process involved, be separated into two main groups: (1) the porous absorbers and (2) the resonant absorbers.

5.1.1. Porous Absorbers

Porous absorbers are complicated structures often manufactured from glass or mineral fibers pressed into boards held together by suitable adhesives. Owing to the complexity of the construction details of the porous materials (such as length of channels, side branches, irregularity in shape of fibers, etc.), it is difficult to predict accurately the absorption values from calculations although great strides have been made in recent years (17–19). In general, though, the acoustical properties of these materials depend on the air particles within the materials being set into motion (oscillating) when a sound

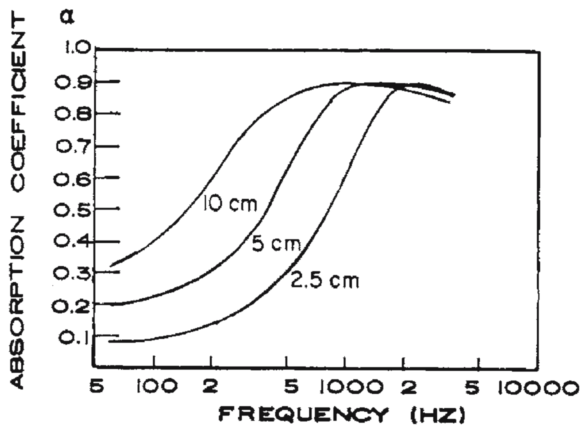

Fig. 13. Absorption coefficients of porous materials with 2.5, 5, and 10 cm thicknesses. (From ref. 1.)

wave strikes the material. Furthermore, this motion, resisted by viscous forces near the surfaces of the fibers, results in part of the sound energy being transformed into heat.

The porosity of the materials is of great importance. If the porosity is low, then the sound waves will have difficulty in penetrating the material, and most of the sound will be reflected from the surface. If the porosity is large, the sound wave will easily penetrate the material and the reflection from the surface will be very small. This, however, does not necessarily mean that the absorption will be great. Substantial absorption will be obtained only if the sound wave that penetrates into the material is reduced to a great extent before it leaves the material.

One valid generalization about porous absorbers is that the first maximum of absorption will occur when the thickness of the porous material equals one-quarter of a wavelength of the incident sound, for example, if $d = \lambda/4$. Therefore, if it is important to obtain high absorption values at low frequencies, the thickness of the material must be large. However, low-frequency absorption can be obtained with thin materials having high flow resistance values as long as they are installed with an airspace between them and the wall.

Figure 13 shows how the absorption coefficient theoretically should change when the thickness is changed. The first maximum on the curves should occur at a frequency having a wavelength equivalent to four times the thickness of the material. However, remember that the complexity of these porous materials requires that one should only use data that have been obtained experimentally.

If the absorption material is covered with thin plastic film to prevent contamination or spilling, then the absorption coefficient will change. The greatest change will take place at high frequencies, at which the film will be impermeable to sound, thus decreasing the absorption coefficient. The least change will occur when the film is no thicker than 50 μm (1–2 mil).

5.1.2. Resonance Absorbers

Resonance absorbers can be thought to consist of a simple mechanical system containing a mass *m* and a stiffness *s* (20). Each resonance absorber will thus possess a frequency at which sound energy impinging on it will be absorbed. The formula for the resonance frequency of such a system (1) is

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{s}{m}}\ \text{Hz} \tag{36}$$

5.1.2.1. Membrane Absorbers

A membrane absorber consists of an impermeable plate placed at a distance from a wall. The air in the cavity between the plate and the wall acts as a spring with a characteristic stiffness. The resonance frequency for such a system (1) is

$$f_0 = 600/\sqrt{md}\ \text{Hz} \tag{37}$$

where m is the weight of the front plate (in kg/m^2) and d is the depth of the cavity (in m). In English units, the resonance frequency will be

$$f_0 = 1340/\sqrt{md}\ \text{Hz} \tag{38}$$

where m is the weight of the front plate (in lb/ft^2), and d is the depth of the cavity (in ft) (1).

The above formula does not consider the stiffness of the front plate itself and it naturally must be included if it is significant. The stiffness of a panel or plate depends on the support system utilized. This is especially true if the distance between the wall and the plate is large, because the air has little stiffness and the plate stiffness becomes more important. If the distance between panel or plate supports is small, the panel or plate has great stiffness.

The resonance frequency of a panel system is normally in the range 100–400 Hz for absorbers with panel thicknesses of 5–15 mm (¼–⅝ in.) and cavity depths of 10–25 mm (⅜–1 in.). Figure 14 shows a typical membrane absorber and associated absorption coefficient curve.

When a sound wave strikes a membrane absorber, the front plate starts to oscillate. The oscillations are the greatest if the incoming sound frequency is close to the resonance frequency of the membrane absorber. The absorption of a membrane absorber can be increased by filling the cavity behind the plate with a porous absorber. However, the maximum absorption that can he obtained that membrane absorbers seldom exceeds a coefficient of 0.5.

5.1.2.2. Helmholtz Resonators

A Helmholtz resonator is an enclosed space with stiff walls, having only a single opening to a room, normally called the throat. Such a resonator can be represented by a simple mechanical system consisting of an oscillating mass m (the "plug" of air in the throat) resisted by a spring with a stiffness s (the air in the enclosed space).

A Helmholtz resonator, like any mechanical system, has a resonance frequency determined by

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{l_a V}}\ \text{Hz} \tag{39}$$

where c is the speed of sound (in m/s), S is the area of the throat opening (in m^2), V is the volume of the enclosed space (in m^3), and l_a is the adjusted length of the throat (in m) (1).

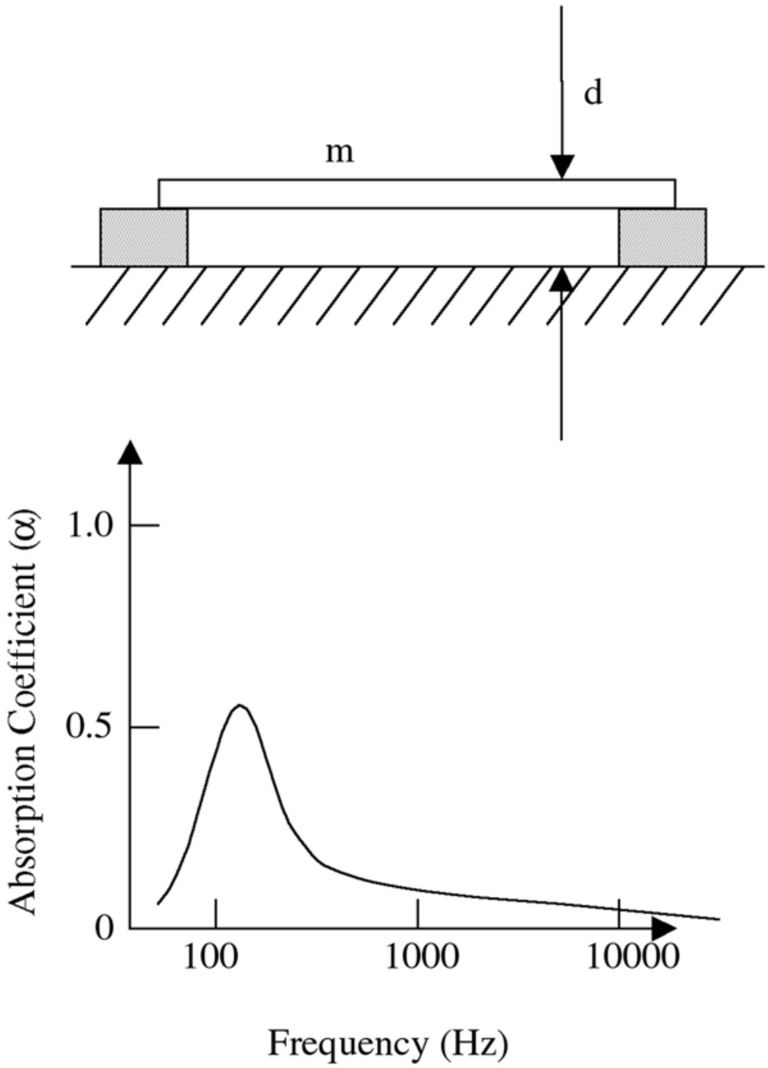

Fig. 14. Qualitative description of absorption coefficients for a membrane absorber as a function of frequency. (From ref. 1.)

For most practical purposes, the adjusted length of the throat is determined (7) by

$$\begin{aligned} l_a &= l + 1.7a \quad \text{flanged} \\ l_a &= l + 1.4a \quad \text{unflanged} \end{aligned} \tag{40}$$

where a is the diameter of the throat and l is the length of the throat. An opening consisting of a circular hole in the thin wall of a resonator will have an adjusted length of approx $1.6a$ (7).

The formula for the resonance frequency is only valid if the dimensions of the resonator are small compared to the resonance frequency wavelength and if the dimensions of the throat are small compared to the enclosed space. A Helmholtz resonator absorbs sound effectively only in the vicinity of the resonance frequency. The absorption can be large provided that the internal damping of the resonator is minimal; filling the enclosed space with a porous material will increase damping. For optimum performance, the Helmholtz resonator should be empty and the walls of the enclosed space should be very stiff.

The highest absorption that can be obtained with a Helmholtz resonator (1) is

$$A_0 = \frac{\lambda^2}{2\pi} \tag{41}$$

The Helmholtz resonator is very selective, so that the absorption falls off rapidly as the sound frequency shifts away from the resonance frequency. Helmholtz resonators are often utilized to solve special problems, such as reducing standing waves in music rooms or concert halls.

5.1.2.3. PERFORATED PANE/ABSORBERS

If the front plate in a membrane absorber is perforated, one may visualize the perforated plate as consisting of a large number of small Helmholtz resonators with a common cavity. The common cavity can be imagined to be divided into a number of smaller spaces by fictitious walls so that each hole in the front plate has one space behind it.

Most standard perforated plates contain between 6% and 25% open area. Most holes are made circular in shape with diameters between 1 and 4 mm. The thickness of the front plates depends on the material and usually varies between 1 and 10 mm.

The resonance frequency for such a system is given by (21)

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{p}{100dl_a}}\ \text{Hz} \tag{42}$$

where p is the perforation percentage, d is the distance between the plate and the wall (in m), and l_a is the adjusted length of the throat (in m).

This formula shows that there is greater latitude in the possible resonance frequency for which the perforated plate absorber can be designed than for the single Helmholtz resonator. Practical experience has shown that the perforated panel absorbers have the additional benefit of possessing good absorption over a wider frequency range than a single Helmholtz resonator, especially if porous material is placed behind the plate.

If the open area of the perforations exceeds about 25%, then the Helmholtz resonator part of the acoustic properties can be neglected and the acoustic properties can be predicted solely on the basis of the porous material behind the plate.

5.1.2.4. SLOT RESONATORS

A slot resonator is a variation of a Helmholtz resonator, the only differences being that one dimension of the cavity space is much larger than the other and the opening to the space is a slot. The resonance frequency for a slot resonator is much more complicated than for a Helmholtz resonator; the procedure will not be discussed here.

5.2. Nonacoustical Parameters of Absorptive Materials

Absorptive materials should always be thought of as an integral part of a building design—not as an afterthought. In this way, their appearance can be used positively to provide the most pleasing environment.

Absorptive materials should be protected from abuse, especially if located within 2 m from the floor, and should be noncombustible and flame retardant and should not shed or collect dust. The acoustical properties should not change as a result of maintenance and cleaning. Porous materials cannot be painted because the absorptive characteristics will be completely ruined; they need to be cleaned on a regular basis or replaced if they become unattractive.

5.3. *Absorption Coefficients*

Design values for absorption coefficients should be obtained from qualified laboratory measurements. Reputable laboratories follow the guidelines of the International Standard Organization recommendations for measuring the reverberation time and calculating the absorption coefficients (22). Such standardizations have been found to be necessary to obtain reliable and consistent coefficients.

One comment should be made regarding reported absorption coefficients. They are all derived from measurements made under idealized conditions as near as possible to diffuse sound fields. A diffuse sound, however, is rarely obtainable in real life. The net result is that absorption coefficients in practice will be effectively lower than given. The laboratory values still have their use because they permit comparisons between different products that have been tested under similar conditions. Values for most commercially available absorption materials can be found from vendors and manufacturers (1,9,12). Materials do not have to be "acoustical" to possess absorption coefficients.

Some typical absorption coefficient values are shown in Table 6 for common construction materials. As can be seen, even these materials have acoustical properties.

6. SOUND ISOLATION

6.1. *Introduction*

Sound isolation refers to the process of insulating an area adjacent to or near a sound source in order to achieve an acceptable acoustical environment in the insulated area. An acceptable acoustical environment can mean that people within the insulated area are not seriously disturbed by intrusive sounds; in the opposite sense, it could mean that people are confident that they cannot be overheard by others outside the room—they have acoustical privacy.

When dealing with room acoustic problems, it is important to know the amount of sound energy removed from an area. When dealing with sound isolation, it is important to know how the sound energy leaves one area and enters another. Airborne sound can travel from one room to another in several ways:

1. Through a dividing wall (by causing the wall to vibrate and reradiate sound on the other side)
2. Through cracks, holes, and so forth
3. By means of flanking paths (other structural paths or through a ceiling plenum)

Direct transmission through the dividing wall is often responsible for most of the intrusive sound.

Table 6
Absorption Coefficients at Various Frequencies for Some Common Construction Materials

	Frequency (Hz)					
	125	250	500	1000	2000	4000
Wood flooring	0.15	0.11	0.10	0.07	0.07	0.07
Linoleum flooring	0.02	0.03	0.03	0.03	0.03	0.02
Padded carpet	0.11	0.14	0.20	0.33	0.52	0.82
Brickwall	0.01	0.01	0.02	0.02	0.03	0.03
Brick (unglazed)	0.03	0.03	0.03	0.04	0.05	0.07
Concrete	0.01	0.01	0.02	0.02	0.02	0.04
Plaster, gypsum	0.13	0.15	0.02	0.03	0.04	0.05
Windows	0.4	0.3	0.2	0.17	0.15	0.1
Doors	0.18	0.12	0.10	0.09	0.08	0.07
Drapes	0.06	0.1	0.38	0.63	0.7	0.73
Water surface	0.008	0.008	0.013	0.015	0.02	0.025
Rock wool, 2.5 cm	0.09	0.23	0.53	0.72	0.75	0.77
Rock wool, 5.0 cm	0.20	0.53	0.74	0.78	0.75	0.77
Rock wool, 10.0 cm	0.68	0.84	0.82	0.78	0.75	0.77

Source: Data from refs. 1 and 9.

6.2. Transmission Loss

When dealing with room acoustic problems, it is important to know the amount of sound energy that will be absorbed, although it is less important to know what percentage of the absorbed sound will be converted to heat or will be transmitted away from the room through openings or other paths. When it comes to sound isolation, however, it definitely is of importance to know how much sound energy will be transmitted through the building's structural elements.

Transmission loss is a term describing the sound-attenuating properties of a material or system. The transmission loss (TL) of a wall or slab construction is defined as the difference between the sound power impinging on the surface in the source room and the sound power radiated from the surface in the receiving room. In logarithmic form, this is expressed as

$$\mathrm{TL} = 10 \log W_1/W_2 \,\mathrm{dB} \tag{43}$$

This can be written as

$$\mathrm{TL} = 10 \log 1/\tau \,\mathrm{dB} \tag{44}$$

where τ is the transmission coefficient. The transmission loss can only be measured accurately when great care has been taken to eliminate the indirect transmission paths.

6.2.1. Measurement of Transmission Loss

In order to calculate the transmission loss of a material, it is necessary to make measurements in properly designed rooms separated by the material for which the TL quantification is desired. When making measurements of transmission loss of structures,

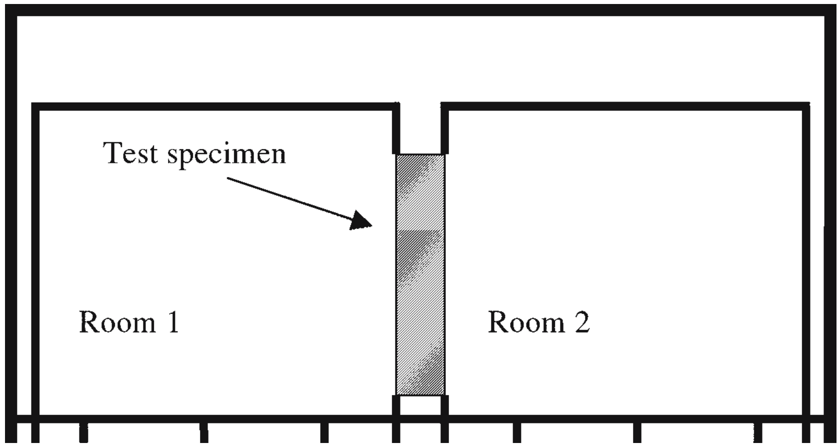

Fig. 15. Reverberation rooms for measurement of transmission loss. (From ref. 1.)

it is of utmost importance that the sound transmission between source and receiver room primarily takes place through the wall of interest (23). Transmission of sound through floor, side walls, and so forth must be negligible. This can he accomplished by constructing the two rooms on separate foundations and inserting resilient layers—such as sheets of cork, glass fiber, or mineral wool—where the two rooms abut each other. Some rooms have a separate frame for the installation of the test specimen between the two rooms, as shown, in Fig. 15.

The sound power which impinges on a surface S can in the stationary situation be determined by

$$W_1 = \frac{1}{4} cSD_1 \tag{15}$$

Recall that the energy density in the steady-state situation could be expressed by

$$D_2 = 4W_2 / cA_2 \tag{19}$$

The energy density in the diffuse field also relates to the rms sound pressure:

$$\begin{aligned} p_{1,\mathrm{rms}}^2 &= \rho c^2 D_1 \\ p_{2,\mathrm{rms}}^2 &= \rho c^2 D_2 \end{aligned} \tag{11}$$

Substituting these formulas in the formula for the transmission loss, we get

$$\mathrm{TL} = 20 \log \left(p_{1,\mathrm{rms}} / p_{2,\mathrm{rms}} \right) - 10 \log (A_2 / S) \tag{45}$$

$$\mathrm{TL} = \mathrm{SPL}_1 - \mathrm{SPL}_2 - 10 \log (A_2 / S) \tag{46}$$

where SPL_1 and SPL_2 are the sound pressure levels in the source and receiver room, respectively. Measurements of the SPL can be directly made, but A_2 should be calculated from reverberation time data. Although some time was spent in the preceding sections detailing methods for calculating A determining it from the reverberation time removes any uncertainty in the assumed values for α.

6.2.2. Calculation of Transmission Loss

There are occasions when estimates of specific wall construction TL are needed. A calculating procedure, rather than a measurement procedure, is needed for these occasions and is discussed in the following subsections.

6.2.2.1. Transmission Loss of a Single-Thickness Wall

An accurate calculation of the transmission loss of even a simple single wall is extremely difficult to perform and is beyond the intent of this text. If some assumptions are made, then it is possible to derive a simplified formula that approximates actual transmission loss fairly well. The following assumptions are made:

1. Only plane waves are incident on the walls.
2. The wall radiates plane waves.
3. The wall is infinitely large.
4. No energy losses occur inside the wall.
5. The surfaces in the receiving room are totally absorptive.

The simplified formula is (12,24)

$$\mathrm{TL} = 20 \log f + 20 \log m - C \tag{47}$$

where f is the frequency (in Hz), m is the weight of the wall (in kg/m^2), and C is a constant depending on the units used (for metric and English units, the values are 43 and 29, respectively). If one relaxes assumption 5 and has a diffuse field impinging on the incident wall, the transmission loss is lowered by 5 dB to account for the possible varying angles of incidence (12). The formula indicates that the TL will increase by 6 dB for each doubling of frequency and doubling of surface weight (or thickness for a constant material). This relationship is normally called the "mass law."

Figure 16 shows a graphic presentation of Eq. (47) (dashed line) and an empirically determined curve (solid line). As can be seen from Fig. 16, the empirically determined curve is lower than the theoretical curve. It can also be seen that for walls with surface weight less than about 200 kg/m^2 (40 lb/ft^2), each doubling of the weight only results in a TL increase of approx 5 dB, whereas above 200/kg m^2, the ''mass law'' is valid again.

One of the assumptions made was that the wall was only affected by plane waves normal to the wall. This, however, is not realistic. Sound waves will strike the wall from all directions, at all angles. An incident sound wave just grazing the wall will deform the wall and cause bending modes to be generated. At some frequency, dependent on physical properties of the wall, the bending modes will be easily excited. This is the critical frequency *fc*. At the critical frequency, the transmission loss theoretically becomes zero. The critical frequency *fc* can be determined from

$$f_c = \frac{c^2}{2\pi}\sqrt{\frac{m}{B}}\ \mathrm{Hz} \tag{48}$$

where c is the speed of sound (in m/s), m is the surface weight (in kg/m^2), and B is the bending stiffness (in Nm) (1).

The bending stiffness B can be determined from

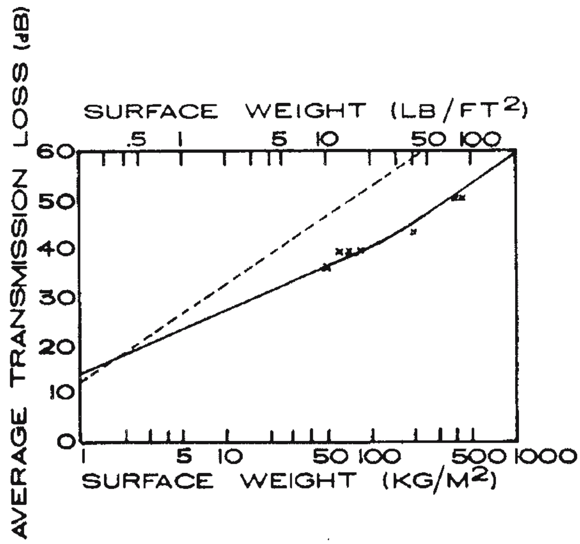

Fig. 16. Average transmission loss in the frequency range 100–3000 Hz as a function of the surface weights of walls. (From ref. 1.)

$$B = \frac{E \times d^3}{12 \times (1 - \sigma^2)} \tag{49}$$

where d is the thickness of the wall (in m), E is Young's modulus (in Pa), and σ is Poisson's ratio.

By utilizing the relationship $m = d\rho$, where ρ is the density of the wall, the critical frequency becomes

$$f_c = \frac{c^2}{2\pi d} \sqrt{\frac{12\rho(1 - \sigma^2)}{E}} \text{ Hz} \tag{50}$$

Poisson's ratio will be equal to about 0.3 for many structural materials. Attention to the critical frequency is important because, if the sound to be isolated is significant at that frequency, alternate construction may be needed.

Example 7

Determine the critical frequency for a 6-mm-thick window pane. The density of glass is about 2500 kg/m^3. Young's modulus is 3.5×10^{10} kg/m-s^2

Solution

$$f_c = \frac{340^2}{2\pi \times 0.006} \sqrt{\frac{12 \times 2500 \times (1 - 0.09)}{3.5 \times 10^{10}}}$$

$$f_c = 2720 \text{ Hz}$$

The transmission loss curves for a 6-mm-thick window is shown in Fig. 17 (as calculated according to the above discussion). Note in Fig. 17 that the transmission loss

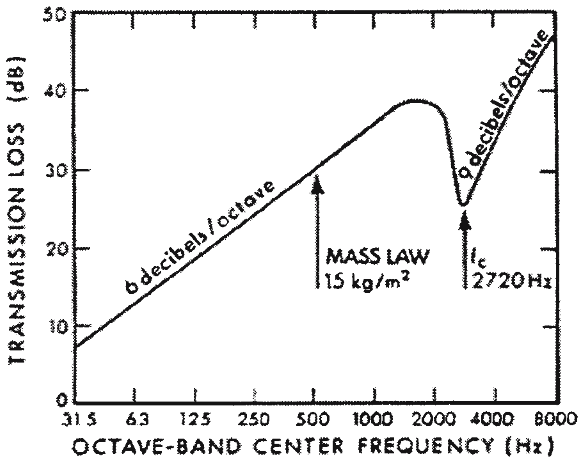

Fig. 17. Transmission loss curves for a 6-mm-thick window as a function of frequency. (From ref. 1.)

at the critical frequency is not zero, as is suggested by theory. Laboratory studies have been undertaken and they seem to indicate that the minimum TL should be around 25 dB.

6.2.2.2. Transmission Loss of Double Wall

The transmission loss of a double wall depends on the following:

1. The material between the individual walls
2. The thickness of the cavity between the individual walls
3. The structural ties between the individual walls
4. The materials of which the walls are made
5. The thickness of each wall

In the idealized situation where the two wall sections are completely independent of each other, the transmission loss at very low frequencies will be approximately equivalent to the TL of a single wall with the same total surface weight. However, at the resonance frequency of the wall, the transmission loss will be less than expected. The resonance frequency for a double wall is determined by

$$f_r = \frac{c}{2\pi}\sqrt{\frac{(m_1 + m_2)\times\rho}{m_1 \times m_2 \times d}}\ \text{Hz} \tag{51}$$

where m_1, and m_2 are the surface weight (in kg/m^2), ρ is the density of the air (1.2 kg/m^3), and d is the distance between wall elements (in m) (1).

If the two wall elements are of equal density and thickness, the formula for the resonance frequency becomes

$$f_r = \frac{c}{2\pi}\sqrt{\frac{2\rho}{m_1 d}}\ \text{Hz} \tag{52}$$

The transmission loss at frequencies lower than f_r will be lower than that of a single wall with total weight equal to (m_1, $+m_2$). At higher frequencies however, the increase in transmission loss will be greater than the characteristic 6 dB per octave for a single wall. It is therefore important to keep the resonance frequency as low as possible.

In order to obtain maximum utilization of the individual wall elements (to get as high a transmission loss as possible), it will also be necessary to place absorptive material in the cavity between the two wall elements. This can be accomplished by placing thermal isolation fiberglass blankets in the cavity. It is necessary to cover only half the wall area in order to increase the transmission loss by 3–5 dB. The function of the blankets is to reduce standing waves that may be set up between the walls, rather than to reduce the direct sound transmission.

6.3. Noise Reduction

The noise reduction (NR) is the difference in decibels between the sound pressure levels in two rooms. The NR accounts for all sound paths and thus is more inclusive than the TL.

The NR can be expressed as

$$\mathrm{NR} = \mathrm{TL} - 10\log S/A_2 - (C_1 + C_2)\ \mathrm{dB} \tag{53}$$

where S is the area of the common wall (in m^2), A_2 is the total absorption in the receiving room (in m^2-Sabin). (1), C_1 is a correction that depends on air leaks, and C_2 is a correction that depends on flanking transmission, which will be described in subsection 6.3.2 (C_1 and C_2 are zero for no air leaks and no flanking transmission).

6.3.1. Air Leaks

No aspect of a wall construction requires more attention to detail than ensuring that no air path leaks exist. The seriousness of air leaks can be illustrated by the following. A wall with an air leak can be considered as consisting of two elements with different transmission losses. The sound power transmitted by the two elements with a common incident sound power is

$$W_{\mathrm{trans}} = \tau_a S_a + \tau_w S_w \tag{54}$$

where τ_a and τ_w are average transmission coefficients for the air leak and for the wall, respectively S_a and S_w are the areas of the air leak and the wall, respectively (in m^2). The composite transmission loss is then

$$\mathrm{TL} = 10\log\frac{S_a + S_w}{\tau_a S_a + \tau_w S_w} \tag{55}$$

If an infinitely "good" wall (no transmission at all, $\tau = 0$) is placed between two rooms—and there is no flanking transmission—the formula indicates that the maximum NR between the two spaces will be no greater than 10 dB if 10% of the wall area is not closed off ($\tau = 1$). If 1% of the wall area is not closed off, the noise reduction will be 20 dB. If only 0.1% of the wall area is not closed off, the noise reduction will be 30 dB. If only 0.01% of the wall area is not closed off, the noise reduction will not exceed 40 dB.

A typical wall between apartments may have an area of 30–40 m^2. In such a wall, 0.01% of the wall area may be the equivalent to an 8–10-m-long crack no greater than 0.5 mm. In order to achieve sound isolation between apartments, it will be necessary to have an average NR of about 50 dB. If that much sound isolation is provided, then the 70–80 dBA sound levels caused by such common sources as radios and television sets will be reduced to a sound level of about 20–30 dBA in an adjacent apartment. A sound level in the 20–30 dBA range is equivalent to a typical ambient sound level: thus, the intrusion will be minimal.

6.3.2. Flanking Transmission

A flanking path is any other path the sound can take in getting from one room to another than directly through the separating wall (the direct path). Although a sound path through an air leak is also a flanking path by the definition, it has been discussed separately because of the ease of handling those transmission problems. In this sub section, the remaining types of flanking transmission are considered.

The determination of the actual value of the flanking transmission is one of the least investigated and possibly one of the least understood. One explanation for this lack of understanding is that in situations where no more than 45–50 dB NR is required, the contribution from flanking transmission is minimal. In the situation where higher NR values are necessary, precautions have to be taken to reduce the flanking transmission in addition to increasing the TL of the dividing wall.

Flanking transmission can generally be reduced by breaking the structural vibration pathway. This can be accomplished by physically separating the source side from the receiver side. Sufficient separation is often obtained by ensuring that a small space exists between the side walls (or floor and/or ceiling) and the dividing wall. In such cases, the space is filled with a resilient material such as cork, glass fiber, or mineral wool. These materials are used to eliminate the undesirable attributes of the space and, fortunately, they do not short circuit the vibration pathway.

In more critical cases, the source room walls or ceilings may be treated with resiliently mounted skins or suspension ceilings. In the most severe cases, the entire source area may be "detached" from the main structure by "floating" the space on resilient material.

6.4. Noise Isolation Class (NIC)

The Noise Isolation Class is a metric that is used to specify a desired sound isolation between two spaces (9). It is determined by measurements of the sound pressure level in each of the spaces, say on opposite sides of a door. Much like the noise reduction (NR), it incorporates all transmission paths in a de facto manner since it is a direct measurement of the space. It is evaluated by using a broad band, omnidirectional source in one space to ensonify that space and then measuring the sound pressure level on both sides of the interface between the spaces to determine the loss of sound level. It is important to not measure too close to either surface so as to not preferentially measure any given sound path. If possible, a 1 m spacing between interface and measurment location should suffice. Sound pressure levels in one-third octave bands between 125 Hz and 4000 Hz are used to evaluate the NIC. The NIC of the interface is determined by comparing the measured loss values to a contour generated by specifying a loss at 500 Hz and then correcting it by the following values at the other frequencies of interest:

Freq	125	160	200	250	315	400	500	630	800	1k	1250	1.6k	2k	2.5k	3.15k	4k
Correction	−16	−13	−10	−7	−4	−1	0	1	2	3	4	4	4	4	4	4

This value at 500 Hz is adjusted upward or downword and the NIC is determined when either (1) any individual predicted one-third octave value exceeds the actual measured one-third octave loss by 8 dB or (2) the sum total of predicted values exceed the measured values by 32 dB (deficiencies only). The lower, or more conservative, value of NIC is to be used.

Example 8

Sound levels were measured on opposite sides of a doorway between two rooms and the following values were obtained. Determine the NIC of the door.

Freq (Hz) 1/3 Oct. Bd.	Source	Receiver
125	65.5	42.5
160	61.7	43.5
200	55.3	34.1
250	48.7	29.8
315	49.0	25.8
400	43.6	21.7
500	49.4	23.4
630	44.4	21.2
800	49.9	24.5
1000	50.2	23.3
1250	49.1	21.6
1600	48.1	23.8
2000	52.8	33.6
2500	55.1	33.5
3150	55.2	28.5
4000	54.7	24.6

Solution

At a value of NIC = 24, the measured loss at 2 kHz is 19.2 dB while the predicted loss is 28 dB (24 + 4 correction). Thus, the predicted level exceeds the measured level by more than 8 dB. Note that the 32 dB criterion is exceeded at NIC = 25, so the lower value is used to describe the door's effectiveness.

7. VIBRATIONS

7.1. Introduction

Structural vibrations are always present even though sensitive instruments may be needed to detect those caused by minuscule forces such as footfalls or distant traffic. When vibrations are more severe, they can be sensed. If the vibrations occur in the low part of our audibility range, we may not be able to distinguish between feeling and hearing them.

Vibrations can become a problem in a number of ways. For example. they can cause unwanted motion in equipment. Little motion can be tolerated for sensitive scientific equipment such as electron microscopes. More motion can be tolerated for industrial equipment, but each has its limits. Continuous vibrations can cause metal fatigue. Sufficiently intense vibrations can cause physiological disorders in personnel handling equipment such as grinders or chain saws. Vibrations can cause noise. Because vibrations are of such significance, it is of interest to determine how vibratory energy can be prevented from traveling through and within building structures and how equipment can be insulated from vibrations.

7.2. Vibration Isolation

Vibrations are prevented from reaching machinery or passing from the machinery to surrounding structures by incorporating specially designed devices into their supports. These devices, collectively termed “vibration isolators”, act to reduce the intensity of the vibrations that pass through them.

Most machinery vibration problems deal with harmonic oscillations caused by some cyclical function of the machine (gear rpm, machinery imbalance, etc.) The vibratory response of the isolating system (i.e., the forces transmitted through the system) is shown in . Figure 18 shows the transmissibility plotted as a function of the ratio of forcing frequency, ω, to the resonance frequency, ω_0, of the isolator. The resonance frequency ω_0 of the isolator is

$$\omega \cong \sqrt{s/M} \tag{56}$$

where s is the stiffness of the isolator (in N/m) and M is the weight of the machine supported by the isolator (in kg) (1).
The transmissibility (ε) is defined as

$$\text{Transmissibility} = \text{transmitted force/impressed force} \tag{57}$$

It can be shown that if there is no damping in the support system, then the formula can be expressed by

$$\varepsilon = \frac{1}{1-\left(\omega/\omega_0\right)} \tag{58}$$

when ω/ω_0 becomes zero, the transmissibility equals 1.0 and no isolation will be obtained. If ω/ω_0 equals 1.0, the theoretical amplitude of the transmitted force goes to infinity, because the driving frequency equals the natural frequency of the system. Therefore, if it is desired to reduce the magnitude of the driving force, the transmissibility must be less than 0 dB. This occurs if the driving frequency is greater than $\omega_0\sqrt{2}$ as determined from the mathematical derivation. In practice, the driving frequency must be two or four times greater than the resonance frequency of the vibration isolator.

There is a family of curves shown in Fig. 18 because the actual transmissibility is dependent on energy losses that occur within the isolator. Collectively, these losses are expressed in terms of the loss factor η, which is directly proportional to the damping coefficient ξ of the isolator:

$$\eta = \xi/\omega_0 M \tag{59}$$

There is a convenient relationship between the resonance frequency of the isolator and its static deflection that enables us easily to choose an appropriate device on the basis of static deflection. This is fortunate for two reasons. First, the static deflection of the isolator may affect the physical stability of the machinery isolated (especially during

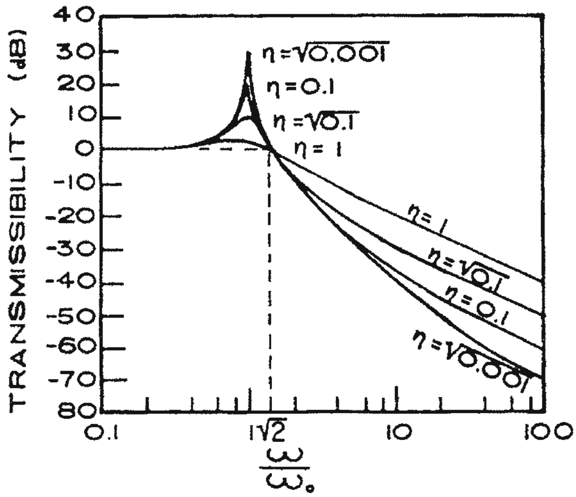

Fig. 18. Transmissibility as a function of loss factor. (From ref. 1.)

startup or shutdown) and, second, physical constraints often control the situation. The relationship occurs because the static deflection is determined by

$$x = Mg/s \tag{60}$$

where g is the gravity acceleration (in m/s^2). Recall that

$$\omega_0 = 2\pi f_0 \tag{4}$$

$$\omega_0 = \sqrt{s/M} \tag{56}$$

then we get

$$\omega_0 = 2\pi f_0 = \sqrt{s/M} = \sqrt{g/x} \tag{61}$$

$$f_0 = \frac{1}{2\pi}\sqrt{g/x} \tag{62}$$

and because $g = 9.81$ m/s^2, f_0 becomes

$$f_0 = \frac{1}{2\sqrt{x}} \tag{63}$$

where x is the static deflection (in m). In English units,

$$f_0 = \frac{3.13}{\sqrt{x}} \tag{64}$$

Corresponding values of static deflection and resonance frequency are shown as follows:

X (m)	0.001	0.003	0.01	0.03	0.1
f_0 (Hz)	16	9	5	3	1.6

The resonance frequency for the foundation of the support system should preferably be less than one-third that of the driving frequency.

8. ACTIVE NOISE CONTROL

The bulk of the work presented here has examined passive noise control by means of either blocking sound transmission from the source or into the receiver or by absorbing it along its propagation route. An alternate means of noise control is active noise control whereby energy is added to a system in an attempt to reduce the net pressure at a receiver. There are two main avenues of this research that can be loosely described as either sound cancellation or source reduction. The first methodology attempts to measure sound that is propagating past a receiver, say in the tail pipe of a car, and to inject noise of an equal magnitude and opposite phase in order to cancel the propagating sound wave. The second methodology attempts to alter the vibration pattern of a structure in an attempt to force the structure into vibration modes that posses lower radiation efficiencies. The treatment of this relatively new line of research is beyond the scope of this text, but the reader is directed to refs. 25–27.

9. DESIGN EXAMPLES

9.1. Indoor Situation

In an industrial plant the noise from one machine was disturbing all operators in the 12-m × 17-m × 5-m (40 ft × 50 ft × 15 ft) production area. The machine was unattended for most of the day and the suggestion was made to enclose it completely.

Because the machine is unattended during most of the day, an enclosure seems a reasonable approach. The physical demands of the situation must be examined before the type of enclosure is chosen:

Does the enclosure need inspection windows?
Will an access door be necessary?
What are the material requirements'?

In this case we have the following,

No inspection windows are necessary
A 0.5-m × 0.5-m access door is needed

Material requirements are as follows:

- Easy cleaning
- Low cost
- Ease of assembly and disassembly
- Structural viability

On the basis of the material requirements, plywood is selected as the material for the enclosure.

The acoustical requirements can now be examined. First, we will select a noise criterion. In many cases, people will select the ambient noise, with the machine turned off, as a criterion, or select a sound level that bears a relationship to hearing damage. The latter will often be chosen if the ambient criterion places too great strains on the construction of the noise control features.

Table 7
Sound Pressure Levels in Decibels at Octave Band Center Frequencies

	Frequency (Hz)					
	250	500	1000	2000	4000	8000
Ambient	66	54	50	44	40	36
Machine noise	92	100	104	107	103	99
Criterion	91	86	82	80	80	81
Noise reduction		14	22	27	23	18
Reverberation time (s)	4.1	2.7	2.0	1.8	1.7	1.2

Source: ref. 1.

In order to determine the criterion, ambient measurements were made with the machine turned off; next, sound pressure level measurements were made at a distance of 2 m (approx 6 ft) from the sound source. In addition to these measurements, we also made reverberation time measurements for the purpose of determining the acoustical characteristics of the processing area. The results of the measurements are shown in Table 7. Because the ambient is extremely low, a noise criterion relating to the prevention of hearing damage is selected and noted in Table 7.

Using Eq. (20), we can calculate the total sound absorption in the production room:

$$t = 0.163 \frac{V}{A + 4mV} \text{ metric}$$
$$t = 0.049 \frac{V}{A + 4mV} \text{ english} \tag{20}$$

The total absorption $A + 4mV$ (in m^2-Sabin) becomes

40 60 82 90 96 136

It is now possible to calculate the power level (PWL) emitted by the machine by using Eq. (29). The vaule in the brackets in the equation becomes

0.030 0.027 0.025 0.024 0.024 0.023

and 10 log becomes (in dB)

−15 −16 −16 −16 −16 −16

and the PWL finally becomes (dB)

107 116 120 123 119 115

In this particular situation, the material selected is plywood about 1/2-in.-thick, which as indicated in Fig. 16, appears to meet the acoustical criterion (7 kg/m^2; 28-dB transmission loss, average [500 Hz]). After the plywood has been painted, it is easily cleanable. The cost of plywood makes this solution economically attractive. Easy assembly is achieved through the use of machine screws, and it is structurally viable in the necessary size.

Can the enclosure be constructed by the plywood alone or do we need absorption inside? To answer this question, we need to calculate the average sound pressure level in the enclosure, with finished dimensions of $2.0 \times 2.0 \times 1.5$ m^3. A determination must also be made of the transmission loss as a function of frequency.

From Table 6, we can select the absorption coefficient (α) for plywood (use door)

0.12 0.10 0.09 0.08 0.07 0.06 (estimated)

Because the wall area is around 20 m^2, the total absorption (A) becomes

2.5 2.0 1.8 1.6 1.4 1.2 m^2-Sabin

An average distance away from the sound source inside the enclosure will be about 0.5 m, so the value in the bracket in Eq. (29) becomes

1.9 2.3 2.5 2.8 3.2 3.6
and log becomes
3 4 4 4 5 6
and the PWL is
110 120 124 127 124 121

By using Fig. 16, we find that the average transmission loss of 1/2-in. plywood is about 28 dB. (The average transmission loss does for most homogeneous materials correspond to the transmission loss at 500 Hz.)

Next, we need to determine the critical frequency f_c. Young's modulus for plywood is 0.675×10^{10} Pa. The critical frequency f_c can be determined from Eq. (50):

$$f_c = \frac{340^2}{2\pi 0.012}\sqrt{\frac{12\times 600\times(1-0.09)}{0.675\times 10^{10}}}\ \text{Hz} \qquad (50)$$

$$f_c = 1510\ \text{Hz}$$

The total transmission loss spectrum can now be determined—remember that the transmission increases by about 6 dB for doubling of frequency. Figure 19 shows the transmission loss as a function of frequency. The transmission loss (in dB) that can be determined from that curve is

22 28 33 28 37 46

Because no gasketing is envisioned between individual wall panels and around the access door, the full potential of the plywood will not be obtained. For calculation purposes, it is expected that all cracks that may be in the enclosure will amount to about 0.01 m^2.

It will, therefore, be necessary to calculate the composite transmission loss. In order to do this, we first use Eq. (44) to determine the transmission coefficients (τ) for the plywood panels:

6.2 1.6 0.5 1.6 0.2 0.25 ($\times 10^{-4}$)

and τ for the cracks is 1.0.

Then, we use Eq. (55) to determine the composite transmission loss:

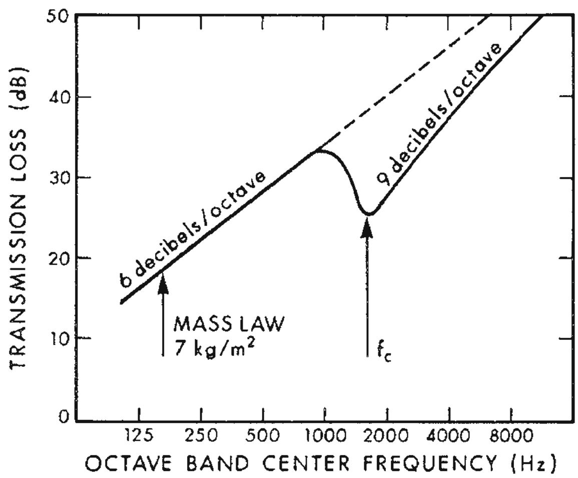

Fig. 19. Transmission loss of ½-in. plywood. (From ref. 1.)

$$\mathrm{TL} = 10 \log \frac{S_a + S_w}{\tau_a S_a + \tau_w S_w} \tag{55}$$

and the results (in dB) become

22 27 30 27 31 33

The NR of the enclosure can now be determined using Eq. (53):

$$\mathrm{NR} = \mathrm{TL} - 10 \log S/A_2 - (C_1 + C_2) \text{ dB} \tag{53}$$

C_1 can be considered zero because cracks were made part of the TL, and because there is no flanking, $C_2 = 0$. The NR (in dB) becomes

25 32 36 34 38 41

This NR is then subtracted from the SPL calculated to be inside the enclosure.

By subtracting the NR from the calculated inside SPL, we get the outside $\mathrm{SPL}_{\mathrm{out}}$:

$\mathrm{SPL}_{\mathrm{out}}$	85	88	88	93	86	80 dB
Criterion	91	86	82	80	80	81 dB

This result shows clearly that this "bare" enclosure cannot meet the criterion in the 500–4000 Hz frequency range and that absorption inside the enclosure will be necessary. We then decided to pad all enclosure walls (approx 20 m^2) with 2.5-cm-thick rockwool. From Table 6, we get the absorption coefficients (α)

0.23 0.53 0.72 0.75 0.77 0.80 (estimated)

and by utilizing Eq. (21), we get A_2 (in m^2-Sabin):

4.6 10.6 14.4 15.0 15.4 16.0

To calculate the reduction in the sound pressure level inside the enclosure, we utilize the absorption characteristics of the "bare" plywood, A_2, and Eq. (35):

$$\Delta = 10\log A/A_0 \text{ dB} \tag{35}$$

The NR (Δ) (in dB) becomes

3 7 9 10 10 11

Because the outside SPL will drop with the equivalent amount as the inside, the outside SPL becomes

SPL_{out}	82	81	79	83	76	69 dB
Criterion	91	86	82	80	80	81 dB

The expected SPL, therefore, will only exceed our criterion in the 2000-Hz octave band by about 3 dB.

In addition to lowering the inside SPL, the rockwool has a second benefit. It will reduce the area of the open crack to perhaps one-tenth of what was expected in the "bare" enclosure. This means that the composite TL of the enclosure will be greater than previously determined.

Using a crack area of 0.001 m^2, we get a composite TL (in dB) of the enclosure walls of

22 28 32 28 36 41

which differs from the previous calculated composite TL (in dB) as shown

— +1 +2 +1 +5 +8

This means that the calculated outside SPL_{out} can be reduced by these amounts:

SPL_{out}	82	80	77	82	71	61 dB
Criterion	91	86	82	80	80	81 dB

Once again, we see that our criterion can only be expected to be exceeded in the 2000-Hz octave band (about 2 dB). The decision was made to build the enclosure, and after the installation, the measurements made outside the enclosure (SPL_{meas}, in dB) were

85 80 76 75 70 63

and the difference (in dB) from the calculated result becomes

+3 — −1 −7 −1 +2

The only major difference was in the 2000-Hz octave band, the octave band containing the critical frequency. The measurements showed that the dip in the transmission loss at that frequency may not have been as pronounced as expected and that the TL curve might look somewhat like the curve shown in Fig. 20. Beyond that, the correlation between calculations and measurements is extremely good.

9.2. Outdoor Situation

Noise, an increasing harassment in our daily life, is largely the result of modern technology. However,what technology created—noise pollution—technology can correct.

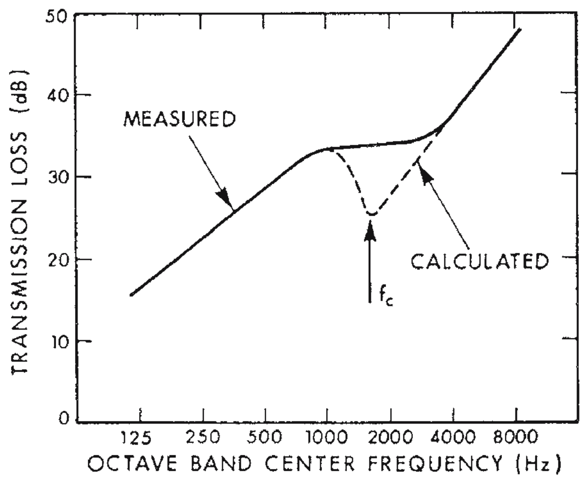

Fig. 20. Transmission loss of ½ in. plywood (corrected). (From ref. 1.)

Noise can be controlled, and much of the technology required for solving most noise problems is available today.

As more people have responded to noise intrusions, various government bodies have developed some type of noise regulation governing environmental noise. The implementation of noise standards is by no means uniform across the United States. A good review of various ways to handle noise ordinances varying from subjective nuisance to objective defined levels can be found in ref. 28. A typical implementation is the use of the day-night level (L_{DN}):

$$L_{DN} = 10\log\left[\frac{1}{24}\left(15\times 10^{L_{eqD}/10} + 9\times 10^{(L_{eqN}+10)/10}\right)\right] \quad (65)$$

where the L_{eqD} is the equivalent sound level during daylight hours of 7 AM to 10 PM and the L_{eqN} is the nighttime level from 10 PM to 7 AM. This equivalent level can be the average of several spot measurements or, more typically, a running average from a recording sound-level meter. The +10 added to the L_{eqN} represents a penalty for nighttime noise because there is a greater expectation of quiet at night. Although values vary across municipalities, typical design values suggest L_{DN} less than 55 dBA for outdoor situations and L_{DN} less than 45 dBA for indoor situations to avoid complaints.

The following example reflects a community's recognition of a noise intrusion from a neighboring industrial plant. Our problem was to determine the likely causes of the noise complaint and develop solutions for its control. Figure 21 shows a layout of the plant, as well as the location of the surrounding residential area. Measurements of noise were performed both close to equipment and out in the community. To determine the severity of the noise problem, it is necessary to have a noise criterion. At the time of the problem, the state in which this situation existed had no quantified regulation, just a common nuisance law. For this example, the Illinois Noise Pollution Control Regulation was adopted to control community noise levels. Under the Illinois regulation, allowable noise emission levels vary, depending on whether the source may be categorized as an industrial,

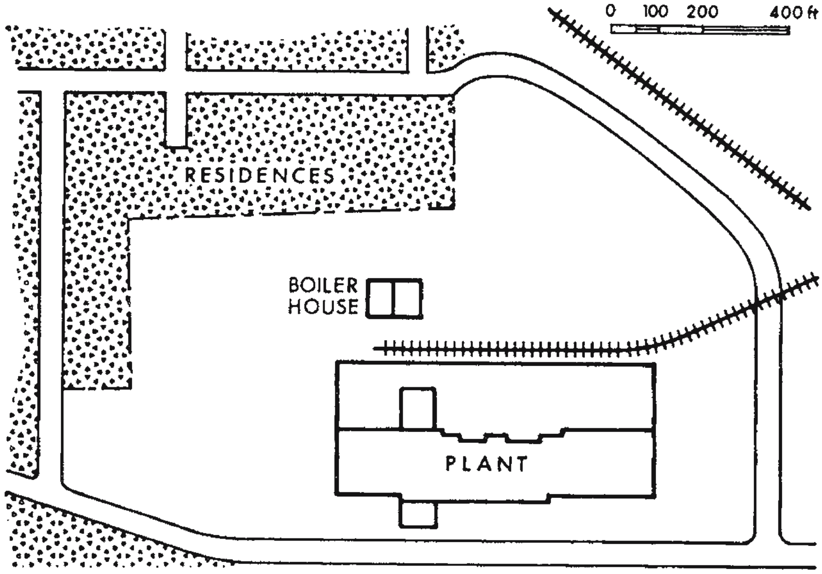

Fig. 21. Plant layout for the exemplary problem studied. (From ref. 1.)

commercial, or residential. Emission criteria are dependent also on the category of the area that receives these noise levels.

Briefly, the Illinois coding system identified industrial land areas as Class "C" land, commercial areas as Class "B" land, and residential areas as Class "A" land. According to the Illinois regulation, the plant in this example was categorized as Class "C" land and the community as Class "A" land.

Figure 22 indicates the criterion that applies when a Class "C" land (i.e., the plant) radiates noise to an adjacent Class "A" land. The regulation distinguishes between daytime (7 AM to 10 PM) and nighttime (10 PM to 7 AM) criteria, and because the plant is operating on a round-the-clock basis, it is the more severe nighttime criterion that was selected. Figure 22 is used for evaluating the noise radiated by the plant, and a violation exists if the noise measured in the community exceeds the criterion curves.

Although the criterion curve is necessary to evaluate whether or not there is a violation of the noise code, it is also useful in identifying (1) those noise sources that are major contributors to the violation and (2) the amount of noise reduction needed to comply with the regulation. The range of sound pressure level measured in the community, as well as the criterion curve, is shown in Fig. 23 and it is quite obvious that a violation exists. More than 90 data samples were measured and analyzed in the course of this project; a representative sample is included in Table 8.

Figure 24 shows the layout of the plant with some of the noise sources indicated. With the measurement made 5 ft from the cooling tower and using Eq. (31), we can determine the sound pressure level expected approx 250 ft away.

$$SPL_5 = PWL - 10\log 4\pi\, 5^2$$

$$SPL_{250} = PWL - 10\log 4\pi\, 250^2$$

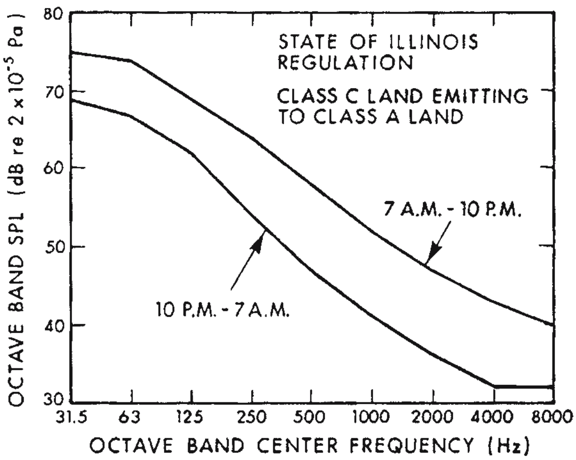

Fig. 22. Noise criterion. (From ref. 1.)

Subtraction results in

$$SPL_5 - SPL_{250} = 10\log 4\pi\, 250 - 10\log 4\pi\, 5^2 = 20\log 250/5$$

$$SPL_5 - SPL_{250} = 20\log 50 = 34\,\mathrm{dB}$$

$$SPL_{250} = SPL_5 - 34\,\mathrm{dB}$$

Figure 25 shows the sound pressure level measured 5 ft from the cooling tower, the calculated SPL at 450 ft solely contributed by the cooling tower, and the measured SPL at that position. The nighttime criterion is also shown. This figure shows without any doubt that the cooling tower is a major contributor to the sound pressure level measured at that location.

The protrusion of the criterion curve indicates the required noise reduction. These facts raise a question: Will the plant be able to meet the noise criterion if the cooling

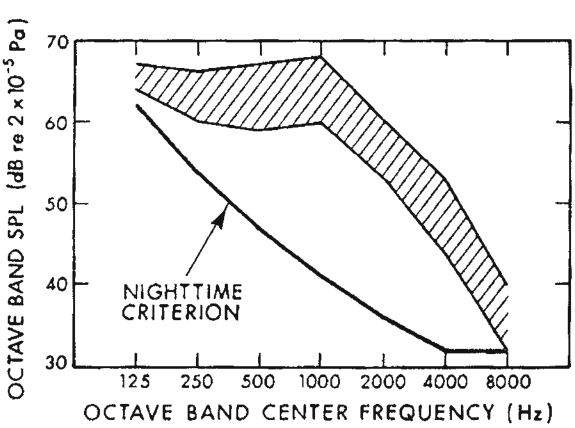

Fig. 23. Range of noise level in the community studied. (From ref. 1.)

Table 8
Measured Sound Pressure Levels in the Community Under Study

	Sound Pressure Level (dB) at Octave Band Center Frequency (Hz)						
Source location	125	250	500	1000	2000	4000	8000
5 ft from cooling tower	89	95	95	93	88	79	72
10 ft from smoke eliminator	92	100	96	89	80	72	66
3 ft from bean tank	84	84	83	89	95	97	96
3 in. from steam ejector	81	78	80	86	88	85	74
3 ft from exhaust fan	105	97	94	91	84	76	80
3 ft from steam vent	96	94	88	83	78	72	70
3 ft from cooling tower	82	87	84	86	84	73	62
6 in. from stack	83	80	75	71	65	63	56
3 ft from blower	98	92	88	86	78	72	73

Source: ref. 1.

tower is quieted or do some of the other sources also have to be quieted? To determine the answer, we will have to treat the other potential noise sources in the same fashion as the cooling tower.

The three bean tanks are each 6 ft tall and have a diameter of 5 ft, and because sound pressure level measurements have been performed very close (3 in.) to the tanks, the total sound power of the three tanks can be determined from

$$\mathrm{PWL} = \mathrm{SPL}_{3''} + 10 \log S - 10$$

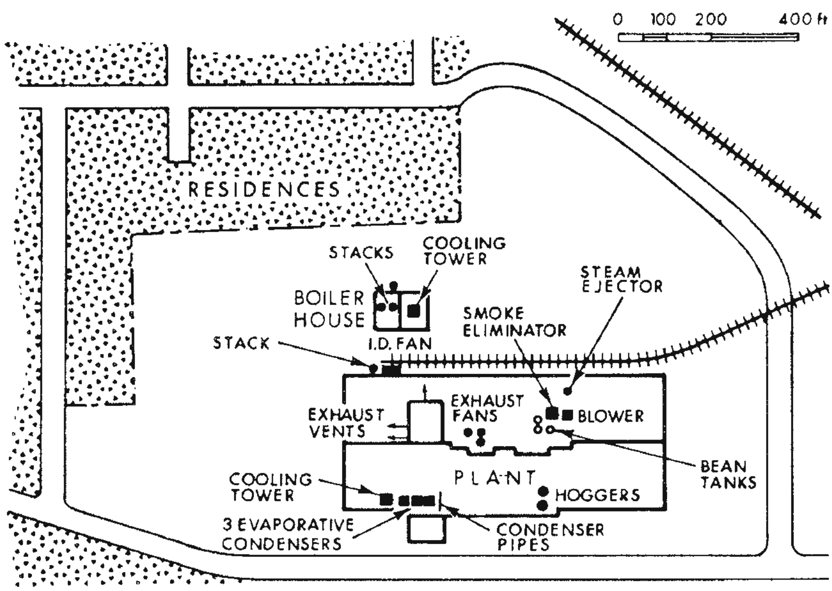

Fig. 24. Location of noise sources. (From ref. 1.)

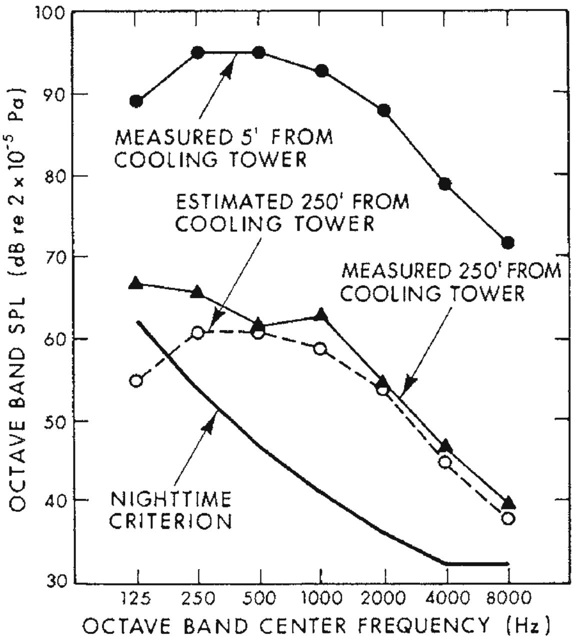

Fig. 25. Noise levels in connection with cooling tower. (From ref. 1.)

$$
\begin{aligned}
&= \mathrm{SPL}_{3''} + 10\log 3 \times \pi \times 5 \times 6 - 10 \\
&= \mathrm{SPL}_{3''} + 10\log 283 - 10 \\
&= \mathrm{SPL}_{3''} + 14\,\mathrm{dB}
\end{aligned}
$$

Because the distance to the receiver position is about 800 ft, we use Eq. (31) again to obtain the expected SPL_{800}:

$$
\begin{aligned}
\mathrm{SPL}_{800} &= \mathrm{PWL} - 10\log 4\pi R^2 \\
&= \mathrm{PWL} - 10\log 4\pi 800^2 \\
&= \mathrm{SPL}_{3''} + 14\,\mathrm{dB} - 10\log 8 \times 10^6 \\
&= \mathrm{SPL}_{3''} + 14\,\mathrm{dB} - 69\,\mathrm{dB} \\
&= \mathrm{SPL}_{3''} - 55\,\mathrm{dB}
\end{aligned}
$$

Figure 26 shows the sound pressure levels associated with the bean tanks and the criterion curve. Because the calculated sound levels protrude above the criterion curve in the 2000–8000-Hz octave-band range, a noise problem will still exist after the cooling tower noise has been controlled.

The same procedure applied to all other noise sources allows us to categorize them as follows:

Major Noise Sources
- Cooling tower on main roof
- Steam ejector on main roof
- Bean tanks on main roof

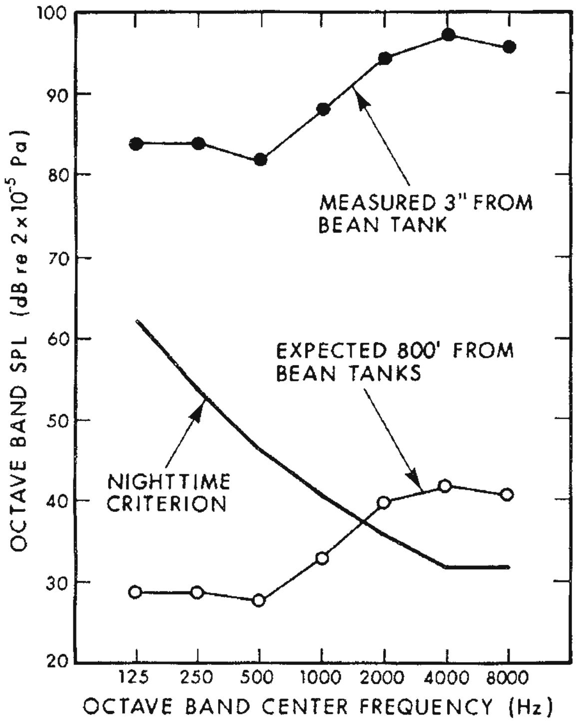

Fig. 26. Noise levels in connection with bean tanks. (From ref. 1.)

Exhaust fan in wall

Secondary Noise Sources
- Stacks
- Induced draft fan
- Cooling tower on boiler roof
- Evaporative condensers
- I-logger mufflers
- Condenser pipes
- Smoke eliminators
- Blower

Tertiary Noise Sources
- Openings in walls
- Small exhaust fans
- Miscellaneous other openings in buildings

For the major noise sources, we must now determine noise control. We decide that for the cooling tower, a barrier is most feasible. How high must the wall or barrier be?

Figure 11 shows the noise reduction that can be expected from a barrier. As shown in Fig. 25, the necessary noise reduction is about 28 dB at 500 Hz (roughly in the center of the frequency range); thus, we find that no barrier can yield such a phenomenal noise reduction. We decide, then, to develop possible alternatives of different height barriers in connection with standard cooling tower mufflers. How high will the barrier have to be under the following circumstances?

Muffler performance: 8 dB at 500 Hz
13 dB at 500 Hz
18 dB at 500 Hz
23 dB at 500 Hz

To determine the barrier performance, it is necessary to select a barrier position between the cooling tower and the receiver. We choose to place the barrier no closer than 4 ft from the side of the cooling tower in order not to disturb the airflow too much, thereby reducing the capacity of the cooling tower. The cooling tower is 8 ft wide.

For the following calculations, it can be assumed that all the noise is radiating out from the center of the cooling tower, located 8 ft from the wall and 8 ft above the roof level. Using English units in connection with Fig. 11, we find that δ has to be at a minimum.

Muffler performance (dB)	Remaining attenuation (dB)	δ	Barrier height (H)
8	20	7.0	20.7
13	15	2.3	14.5
18	10	0.5	10.7
23	5	0.006	8.3

The last column is derived from simple geometrical calculation, based on the equation given in Fig. 11:

$$\delta = a + b - D$$
$$\delta = \sqrt{h^2 + 442^2} + \sqrt{h^2 + 8^2} - 450$$

Because $442^2 >> h^2$, the equation becomes

$$\delta \cong 442 + \sqrt{h^2 + 8^2} - 450$$
$$\delta \cong \sqrt{h^2 + 8^2} - 8$$

The barrier height H is

$$H = h + 8\,\text{ft}$$

The final solution to this problem—a decision made by management personnel of the plant and based on these alternatives—was to use a muffler with an insertion loss of 13 dB and to build a 15-ft-high barrier that should give a noise reduction of approximately 15 dB.

The development of feasible noise control principles in controlling ejector noise, bean tank noise, and muffler design is beyond the scope of this chapter, but we believe that the example shows some of the complexity involved in determining the offending sources in a community noise situation. In all cases, the extrapolation technique only holds for an ideal situation in which there are no adverse thermal gradients as well as no wind. So rarely does that situation occur that the noise sources should be divided into at least four categories:

Major noise sources—definitely need treatment
Secondary noise sources—need some treatment
Tertiary noise sources—may need treatment
Minor noise sources—do not need any treatment

Noise control steps can then be taken in an orderly manner, with the worst offenders attenuated first, and the least disruptive last.

GLOSSARY

In this chapter we have used terms, the definitions for which are indicated below. Most of these definitions can be found in "Glossary of Terms Frequently Used in Acoustics," issued by the American Institute of Physics as well as refs. 9 and 12.

Absorption coefficient (acoustical ahsorptivity) (α). The sound-absorption coefficient of a surface exposed to a sound field is the ratio of the sound energy absorbed by the function of both angle of incidence and frequency. Tables of absorption coefficients given in the literature usually list the absorption coefficients at various frequencies, the values being those obtained by averaging overall angles of incidence.

Acoustic, acoustical. The qualifying adjectives acoustic and acoustical mean containing, producing, arising from, actuated by, related to, or associated with sound. Acoustic is used when the term being qualified designates something that has the properties, dimensions, or physical characteristics associated with sound waves; acoustical is used when the term being qualified does not designate explicitly something that has such properties, dimensions, or physical characteristics.

Acoustics. Acoustics is the science of sound, including (1) its production transmission and effects and (2) the qualities that determine the value at a room or other enclosed space with respect to distinct hearing.

Ambient noise. Ambient noise is the all-encompassing noise associated with a given environment, being usually a composite of sounds from many sources near and far.

Angular frequency (ω). The angular frequency of a periodic quantity is its frequency in radians per unit time, usually radians per second. Thus, it is the frequency multiplied by 2π.

Band pressure level. The band pressure level of a sound for a specified frequency band is the effective sound pressure level for the sound energy contained within the band. The width of the band and the reference pressure must be specified. The width of the band may be indicated by the use of a qualifying adjective: for example, octave-band (sound pressure) level, half-octave-band level, third-octave-band level, 50-cps band level. If the sound pressure level is caused by thermal noise, the standard deviation of the band pressure level will not exceed 1 dB if the product of the bandwidth in cycles per second by the integration time in seconds exceeds 20.

Decibel (dB). The decibel is a unit of level which denotes the ratio between two quantities that are proportional to power; the number of decibels corresponding to the ratio of two amounts of power is 10 times the logarithm to the base 10 of this ratio. In many sound fields, the sound pressure ratios are not proportional to the square root of the corresponding power ratios, so that strictly speaking, the term "decibel" should not be used in such cases; however, it is common practice to extend the use of the unit to these cases (*see*, e.g., *Sound pressure level*).

Diffuse sound field (Random-incidence sound field). A diffuse sound field is a sound field such that the sound pressure level is everywhere the same, and all directions of energy flux are equally probable.

Effective sound pressure (p) (root-mean-square sound pressure). The effective sound pressure at a point is the root-mean-square value of the instantaneous sound pressures, over a time interval at the point under consideration. In the case of periodic sound pressure, the interval must be an integral number of periods or an interval long compared to a period. In the case of nonperiodic sound pressures, the interval should be long enough to make the value obtained essentially independent of small changes in the length of the interval. The term "effective sound pressure" is frequently shortened to "sound pressure."

Free field. A free sound field is a field in a homogeneous, isotropic medium free from boundaries. In practice, it is a field in which the effects of the boundaries are negligible over the region of interest. The actual pressure impinging on an object (e.g., a microphone) placed in an otherwise free sound field will differ from the pressure that would exist at that point with the object removed, unless the acoustic impedance of the object matches the acoustic impedance of the medium.

Frequency (f). The frequency of a function that is periodic in time is the reciprocal of the period. The unit is the cycle per unit time (e.g., cycles per second [cps] or hertz [Hz]).

Fundamental frequency. The fundamental frequency of a periodic quantity is equal to the reciprocal of the shortest period during which the quantity exactly reproduces itself.

Harmonic. A harmonic is a sinusoidal quantity having a frequency that is an integral multiple of the fundamental frequency of a periodic quantity to which it is related.

Hertz (Hz). A unit of frequency; formerly called cycles per second (cps).

Level. In communication and acoustics, the level of a quantity is the logarithm of the ratio of that quantity to a reference quantity of the same kind. The base of the logarithm, the reference quantity, and the kind of level must be specified.

Loudness. Loudness is the intensive attribute of an auditory sensation, in terms of which sounds may be ordered on a scale extending from soft to loud. Loudness depends primarily on the sound pressure of the stimulus, but it also depends on the frequency and waveform of the stimulus.

Loudness level. The loudness level, in phons, of a sound is numerically equal to the sound pressure level in decibels, relative to 2×10^{-5} Pa, of a pure tone of frequency 1000 Hz, consisting of a plane-progressive sound wave coming from directly in front of the observer, which is judged by normal observers to be equivalent in loudness.

Noise. Unwanted sound.

Noise level. The acoustical noise level is the sound level.

Normal mode of vibration. In an undamped multi-degree-of-freedom system undergoing free vibration, a normal mode of vibration is a pattern of motion assumed by the system in which the motion of every particle is simple harmonic with the same period and phase. Thus, vibration in a normal mode occurs at a natural frequency of the system. In general, any composite motion of a system is analyzable into a summation of normal modes. (The terms *natural mode, characteristic mode*, and *eigen mode* are synonymous with *normal mode*.)

Octave-band pressure level. The octave-band pressure level of a sound is the band pressure level for a frequency band corresponding to a specified octave. (The location of the octave-band pressure level on a frequency scale is usually specified as the geometric mean of the upper and lower frequencies of the octave.)

Oscillation. Oscillation is the variation, usually with time, of the magnitude of a quantity with respect to a specified reference when the magnitude is alternate1y greater and smaller than the reference.

Period. The period of a periodic quantity is the smallest value of the increment of the independent variable for which the function repeats itself.

Phon. The phon is the unit of loudness level.

Power level (PWL). The power level, in decibels, is 10 times the logarithm to the base 10 of the ratio of a given power to a reference power. The form of power (e.g., acoustic) and the reference power must be indicated. The reference power used in this chapter for sound power level is 10^{-12} W.

Pure tone. A pure tone is a sound wave, the sound pressure of which is a simple sinusoidal function of the time.

Rate of decay. The rate of decay is the time rate at which the sound pressure level (or velocity level) is decreasing at a given point and at a given time. The commonly used unit is the decibel per second.

Resonance. The resonance of a system under forced vibration exists when any small increase or decrease in the frequency of excitation causes a decrease in the response of the system.

Resonant frequency (f_r). A resonant frequency is a frequency at which resonance exists.

Reverberation. Reverberation is the sound that persists at a given point after direct reception from the source has stopped.

Reverberation room. A reverberation room is an enclosure in which all the surfaces have been made as sound reflective as possible. Reverberation rooms are used for certain acoustical measurements.

Reverberation time (t). The reverberation time for a given frequency is the time required for the average sound pressure level, originally in a steady state, to decrease 60 dB after the source is stopped. Usually, the pressure level for the upper part of this range is measured and the result extrapolated to cover 60 dB.

Root-mean-square sound pressure. See *Effective sound pressure.*

Sabin (square meter unit of absorption). The Sabin is a measure of the sound absorption of a surface; it is the equivalent of 1 m^2 (ft^2) of perfectly absorptive surface.

Sound. (a) Sound is an alteration in pressure, stress, particle displacement, shear, or so forth in an elastic medium or (b) sound is an auditory sensation evoked by the alterations described in (a). In case of possible confusion, the term “sound wave” or “elastic wave” may be used for concept (a) and the term “sound sensation” for concept (b). Not all sound waves evoke an auditory sensation. The medium in which the sound exists is often indicated by an appropriate adjective (e.g., airborne, structure-borne).

Sound absorption. Sound absorption is the process by which sound energy is diminished in passing through a medium or in striking a surface.

Sound-absorption coefficient. See *Absorption coefficient.*

Sound energy. The sound energy of a given part of a medium is the total energy in this part of the medium minus the energy that would exist in the same part of the medium with no sound waves present.

Sound energy density. The sound energy density at a point in a sound field is the sound energy contained in a given infinitesimal part of the medium divided by the volume of that part of the medium. The commonly used unit is W-s/m^3.

Sound field. A sound field is a region containing sound waves.

Sound level. The sound level, in decibels, is the weighted sound pressure level obtained by use of a sound-level meter whose weighting characteristics are specified in the latest revision of the American Standards Association standard on sound-level meters. The reference pressure is 2×10^{-5} Pa, unless otherwise specified.

Sound-level meter. A sound-level meter is a device used to measure sound pressure level or weighted sound pressure level, constructed in accordance with the standard specifications for sound-level meters set up by the American Standards Association. The sound-level meter consists of a microphone, an amplifier to raise the microphone output to useful levels, a calibrated attenuator to adjust the amplification to values appropriate to the sound levels being measured, and an instrument to indicate the measured sound level; optional weighting networks are included to adjust the overall frequency characteristics of the response and provision is made for an output connection to additional measuring equipment.

Sound power of a source (W). The sound power of a source is the total sound energy radiated by the source per unit of time.

Sound power level (PWL or L_w). The sound power level of a sound source, in decibels, is 10 times the logarithm to the base 10 of the ratio of the sound power radiated by the source to a reference power. The reference power is 10^{-12} W.

Sound pressure level (SPL or L_p). The sound pressure level, in decibels, of a sound is 20 times the logarithm to the base 10 of the ratio of the pressure of this sound to the reference pressure. The reference pressure employed in this chapter is 2×10^{-5} Pa. In many sound fields, the sound pressure ratios are not proportional to the square root of corresponding power ratios and hence cannot be expressed in decibels in the strict sense; however, it is common practice to extend the use of the decibel to these cases.

Sound reduction between rooms. The sound reduction, in decibels, between two rooms is the amount by which the mean-square sound pressure level in the source room exceeds the level in the receiving room. If a common partition separates two rooms, the first of which contains a sound source, the sound reduction between the two rooms is equal to the transmission loss of the partition plus a function of the total absorption in the second room and the area of the common partition.

Spherical wave. A spherical wave is a wave in which the wavefronts are concentric spheres.

Threshold of audibility (threshold of detectability). The threshold of audibility for a specified signal is the minimum effective sound pressure of the signal that is capable of evoking an auditory sensation in a specified fraction of the trials. The characteristics of the signal, the manner in which it is presented to the listener, and the point at which the sound pressure is measured must be specified. The ambient noise reaching the ears is assumed to be negligible, unless otherwise stated.

Transmission coefficient. The sound transmission coefficient of a partition is the fraction of incident sound transmitted through the partition. The angle of incidence and the characteristic of sound observed must be specified (e.g., pressure amplitude at normal incidence).

Transmission loss. Transmission loss is the reduction in the magnitude of some characteristic of a signal between two stated points in a transmission system. The characteristic is often some kind of level, such as power level or voltage level; in acoustics, the characteristic that is commonly measured is sound pressure level. If the levels are expressed in decibels, then the transmission loss is likewise in decibels.

Transmission loss of a partition (TL). The sound transmission loss of a partition, in decibels, is −10 times the logarithm to the base 10 of the transmission coefficient for the partition. It is equal to the number of decibels by which sound incident on a partition is reduced in transmission through it. Thus, a measure of the airborne sound installation of the partition. Unless otherwise specified, it is to be understood that the sound fields on both sides of the partition are diffuse.

Velocity of sound. The speed of sound in air is given by $v = \sqrt{1.4 \times 287 \times T_K}$ where the 1.4 represents the ratio of specific heats for air, 287 is the universal gas constant for air, and T_K is the temperature in Kelvin (ºC +273.15).

Wave. A wave is a disturbance propagated in a medium in such a manner that at any point in the medium the quantity serving as a measure of the disturbance is a function of the time, while at any instant the quantity serving as a measure of the disturbance at a point is a function of the position of the point. Any physical quantity that has the same relationship to some independent variable (usually time) that a propagated disturbance has, at a particular instant, with respect to space, may be called a wave.

Wavelength (λ). The wavelength of a periodic wave in an isotropic medium is the perpendicular distance between two wavefronts in which the displacements have a difference in phase of one complete period.

NOMENCLATURE

a	Amplitude (m)
c	Speed of sound (m/s)
d	Depth (m)
f	Frequency (Hz)
g	Gravity acceleration (m/s^2)
l	Length (m)
m	Surface weight (kg/rn^2)
n	Attenuation coefficient
p	Sound pressure (dB)
s	Stiffness (N/m)
t	Time (s)
x	Static deflection (m)
A	Total absorption (m^2-Sabin)
B	Bending stiffness (Nm)
C	Correction factor (dB)
D	Sound energy density (W-s/m^3)
E	Young's modulus (Pa)

IL	Insertion loss (dB)
L_p	Sound pressure level (dB)
M	Total weight (kg)
NR	Noise reduction (dB)
PWL	Sound power level (dB)
S	Area (m^2)
T_c	Celsius temperature (ºC)
T_K	Kelvin temperature (K)
T_R	Rankine temperature (ºR)
TL	transmission loss (dB)
V	Volume (m$^{3)}$
W	Sound power (W)
α	Absorption coefficient
δ	Path length difference (m)
ε	Transmissibility
η	Loss factor
θ	Geometric angle
λ	Wavelength (m)
ξ	Damping coefficient (kg rad/s)
ρ	Density (kg/rn^3)
σ	Poisson's ratio
τ	Transmission coefficient
ω	Angular frequency (rad/s)
Δ	Noise reduction (dB)
$4m$	Air absorption (1/m or dB/km)

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