MASTER HANDBOOK OF ACOUSTICS

SEVENTH EDITION

McGraw Hill

F. ALTON EVEREST & KEN C. POHLMANN

Master Handbook of Acoustics

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Dedicated to the memory of F. Alton Everest

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Introduction

You hold in your hands, either physically or electronically, the seventh edition of the *Master Handbook of Acoustics*. Mr. F. Alton Everest was the original author of this book. In 1981 he devised the formula for an acoustics book that balanced theory and practice. Many engineering books sprinkle examples and problems throughout the text, to inform the reader of practical applications. He improved on that model by presenting basic theory combined with a significant quantity of pragmatic information, then attaching entire chapters, comprising a substantial portion of the book, that are purely devoted to practical examples. These chapters are particularly essential for anyone building a room with similar characteristics.

Mr. Everest understood that this was the perfect way to teach introductory acoustics while simultaneously providing practical guidance to anyone undertaking a construction project. He thus created a valuable tool that we know and trust, a book that has become a classic. The acoustical engineering community grieved when Mr. Everest passed away in 2005 at the age of 95.

I was honored when McGraw-Hill asked me to prepare a fifth, a sixth, and now this seventh edition of the *Master Handbook of Acoustics*. I had used the handbook since it was first published, and was well familiar with its value as a teaching text and reference handbook. Readers who are familiar with another of my books, *Principles of Digital Audio*, may be surprised to learn that my passion for digital technology is equaled by my enthusiasm for acoustics. I taught courses in architectural acoustics (in addition to classes in digital audio) for 30 years at the University of Miami, where I directed the Music Engineering Technology program. Throughout that time, I also consulted on many acoustics projects, ranging from recording studio to listening room design, from church acoustics to community noise intrusion. As with many practitioners in the field, it was important for me to understand the fundamentals of acoustical properties, to be able to articulate those principles to clients, and also to stay current with the practical applications and solutions to today's acoustical problems. This essential equilibrium was the guiding principle of Mr. Everest's original vision for this book, and I have continued to seek that same balance. Further, through Mr. Everest's four editions, and my three editions, this book has improved steadily to reach a high level of refinement.

Occasionally, and particularly among newbies to the field of acoustics, the question arises, "Why is it important to study acoustics?" One reason, among many, is that you will be joining in, and hopefully contributing to, a noble scientific undertaking. Since antiquity, some of the world's greatest scientists and engineers have studied acoustics and its elegant complexities. Greek philosophers including Pythagoras, Aristotle, and

Euclid began the exploration of the nature of musical harmonics and how we hear sound. The great Roman engineer and architect Vitruvius carefully analyzed echo and reverberation in his building projects. Over the years, heavyweights such as Ptolemy, Galileo, Mersenne, Kircher, Hooke, Newton, Laplace, Euler, D'Alembert, Bernoulli, Lagrange, Poisson, Faraday, Helmholtz, Ohm, Doppler, and Sabine all made contributions. In all, countless men and women have worked to evolve the science of acoustics to a high degree of sophistication.

But, pressing the question, in today's binary world, is acoustics still important? Consider this: We rely on our eyes and ears. Our eyes close when we sleep; we cannot see in the dark; someone can sneak up on us unseen from behind. But from birth to death, awake or asleep, in light and in dark, our ears are always sensitive to our world around us. Whether we are hearing sounds that give us pleasure, or sounds that alert us to danger, whether they are sounds of nature, or sounds of technology, the properties of acoustics and the way that architectural spaces affect those sounds are woven into every moment of our lives. Is acoustics important? I think it is. And I'm pretty sure Mr. Everest would agree.

Ken C. Pohlmann

CHAPTER 1

Fundamentals of Sound

Sound can be considered as wave motion in air or other elastic media. In this case, sound acts as a stimulus. Sound can also be considered as an excitation of the hearing mechanism that results in its perception. In this case, sound is a sensation. This duality of sound is familiar to those interested in audio and music. The type of problem at hand dictates our approach. If we are interested in the physical disturbance of the air in a room, it is a problem of physics. If we are interested in how that disturbance is perceived by a person listening in the room, psychoacoustical methods must be used. Because this book addresses acoustics in relation to people, both aspects of sound will be considered. That being said, because we are primarily interested in how room materials and geometry affect the disturbance, our investigations will mainly deal with physics.

Sound can be characterized by objective phenomena. For example, frequency is an objective property of sound; it specifies the number of waveform repetitions per unit of time (usually 1 second). Frequency can be readily measured on an oscilloscope or a frequency counter. From a physics standpoint, the concept of frequency is straightforward. We will have much more to say about the objective qualities of sound, particularly in the way that the properties of sound are dictated by the rooms we inhabit.

On the other hand, that rate of repetition can be characterized subjectively. Frequency is then considered in terms of pitch, which is a subjective property of sound. Perceptually, we hear different pitches for soft and loud 100-Hz tones. As intensity increases, the pitch of a low-frequency tone goes down, while the pitch of a high-frequency tone goes up. Harvey Fletcher found that playing pure tones of 168 and 318 Hz at a modest level produces a very discordant sound. At a high intensity, however, the ear hears pure tones in the 150- to 300-Hz octave relationship as a pleasant sound. We cannot equate frequency and pitch, but they are analogous. Another objective/subjective duality exists between intensity and loudness. Similarly, the relationship between waveform (or spectrum) and perceived quality (or timbre) is not linear. A complex waveform can be described in terms of a fundamental and a series of harmonics of various amplitudes and phases. But perception of timbre is complicated by the frequency-pitch interactions in the human hearing mechanism as well as other factors.

The interaction between the physical properties of sound, and our perception of them, poses delicate and complex issues. It is this complexity in audio and acoustics that creates such interesting problems. On one hand, the design of a loudspeaker or a concert hall should be a straightforward and objective engineering process. But in practice, that objective expertise must be carefully tempered with purely subjective wisdom. As has often been pointed out, loudspeakers are not designed to play sine waves into calibrated microphones placed in anechoic chambers. Instead, they are designed to play music in our listening rooms. In other words, the study of audio and acoustics involves both art and science. To learn the complexities of audio and acoustics, we begin with the science, keeping in mind that our ears will ultimately determine the success or failure of our projects.

Simple Harmonic Motion and the Sine Wave

The weight (mass) and the spring shown in Fig. 1-1 comprise a vibrating system. Moreover, the weight moves in what is called simple harmonic motion. When the weight is at rest, the system is said to be in equilibrium. If the weight is pulled down to the –5 mark and released, the spring pulls the weight back toward 0. However, the weight will not stop at 0; its inertia will carry it beyond 0 almost to +5. The displacement of the weight defines the amplitude of the motion.

The weight will continue to vibrate, or oscillate. Each up/down repetition is called a cycle, and the motion is said to be periodic. In the arrangement of a mass and a spring, vibration or oscillation is possible because of the elasticity of the spring and the inertia of the weight. Elasticity and inertia are two things all media must possess to be capable of conveying sound. In this practical example, the amplitude of motion will slowly decrease due to frictional losses in the spring and the air around it.

Harmonic motion is a basic type of oscillatory motion, and it yields an equally basic wave shape in sound and electronics. To illustrate this, if a pen is fastened to the weight's pointer, as shown in Fig. 1-2, and a strip of paper is moved past it at a uniform speed, the resulting trace is a sine wave. The sine wave is a pure waveform closely related to simple harmonic motion. In this figure, the sine wave traced by the pen has completed one full period and is more than halfway through a second period. The periodic motion of the weight will continue to trace the sine wave indefinitely. (For a moment, we are ignoring the frictional losses that would decrease amplitude.) This simple oscillatory system will always create sinusoidal motion; without outside forces, no other motion is

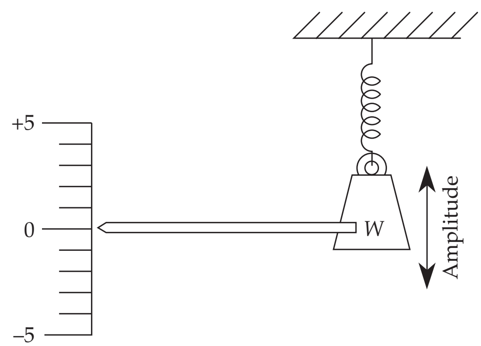


Figure 1-1 A weight on a spring vibrates at its natural frequency because of the elasticity of the spring and the inertia of the weight.

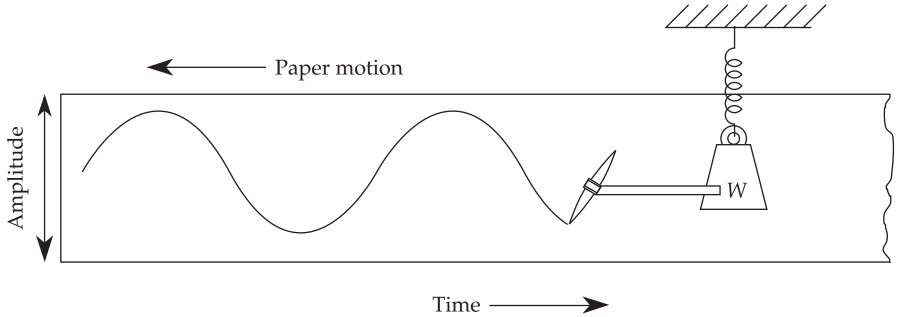


Figure 1-2 A pen fastened to the vibrating weight traces a sine wave on a paper strip moving at a uniform speed. This shows the basic relationship between simple harmonic motion and the sine wave.

possible with this system. However, this graph of a sine wave, showing amplitude versus time, sets precedence for plotting many different wave shapes.

As another example of oscillatory motion, consider a piston in an internal-combustion automobile engine that is connected to the crankshaft by a connecting rod. The rotation of the crankshaft and the up-and-down motion of the pistons illustrate the relationship between rotary motion and linear simple harmonic motion. As with the weight on a spring, the piston position plotted against time produces a sine wave.

Sound in Media

The weight and spring system in the previous example models the motion of air molecules. If an air particle is displaced from its original position, elastic forces of the air tend to restore it to its original position. Because of the inertia of the particle, it overshoots the resting position, bringing into play elastic forces in the opposite direction, and so on.

An elastic medium is essential to the existence of sound waves. Because air is such a common agent for the conduction of sound, it is easy to forget that other media are also conductors of sound. Thus, sound is readily conducted in gases, liquids, and solids such as air, water, steel, concrete, and so on, which are all elastic media. Imagine a railroad track; a friend stationed a distance away strikes a rail with a rock. You will hear two sounds, one sound coming through the rail and one through the air. The sound through the rail arrives first because the speed of sound in steel is faster than in air. Similarly, liquids can be very efficient conductors of sound; underwater sounds can be detected after traveling thousands of miles through the ocean.

Without a medium, sound cannot be propagated. In the laboratory, an electric buzzer is suspended in a heavy glass bell jar. As the button is pushed, the sound of the buzzer is readily heard through the glass. As the air is pumped out of the bell jar, the sound becomes fainter and fainter until it is no longer audible. The sound-conducting medium, the air inside the jar, has been removed between the source and the ear. Outer space is an almost perfect vacuum; no sound can be conducted except in the tiny island of atmosphere within a spaceship or a spacesuit.

Particle Motion

Waves created by the wind travel across a field of grain, yet the individual stalks remain firmly rooted as the wave travels on. In a similar manner, particles of air propagating a sound wave do not move far from their undisplaced positions, as shown in Fig. 1-3. The disturbance travels on, but the propagating particles move only in localized regions (with perhaps a maximum displacement of a few ten-thousandths of an inch). Note also that the velocity of a particle is maximum at its equilibrium position, and zero at the points of maximum displacement (a pendulum has the same property). The maximum velocity is called the velocity amplitude, and the maximum displacement is called the displacement amplitude. The maximum particle velocity is very small, less than 0.5 in/sec for even a loud sound. As we will see, to lower the level of a sound, we must reduce the particle velocity.

There are three distinct forms of particle motion. For sound traveling in a gaseous medium such as air, the particles move in the direction the sound is traveling. This motion is described as longitudinal waves, which expand and contract in the direction of propagation, as shown in Fig. 1-4A. As we will see, this oscillation causes high- and low-pressure regions. The instantaneous pressure on opposite sides of a pressure minimum has opposite polarity. The pressure on one side is increasing, whereas the pressure on the other side is decreasing. A second type of wave motion is illustrated by a violin string, as shown in Fig. 1-4B. The tiny elements of the string move transversely, or at right angles to the direction of travel of the waves along the string. Thirdly, if a stone is dropped on a calm water surface, concentric waves travel out from the point of impact, and the water particles trace circular orbits (for deep water, at least), as shown in Fig. 1-4C.

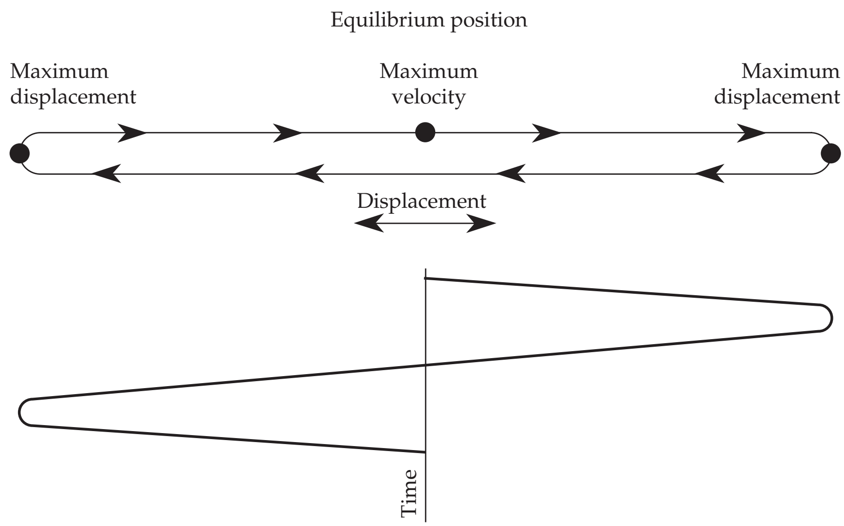


Figure 1-3 An air particle is made to vibrate about its equilibrium position by the energy of a passing sound wave because of the interaction of the elastic forces of the air and the inertia of the air particle.

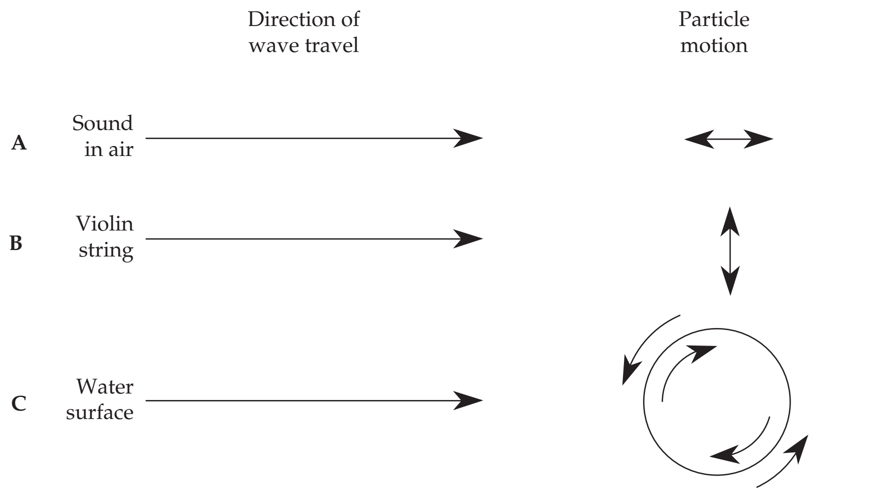


Figure 1-4 Particles involved in the propagation of sound waves can move with (A) longitudinal motion in air, (B) transverse motion on a string, or (C) circular motion on the water surface.

Propagation of Sound

How are air particles, moving slightly back and forth, able to carry music from a loudspeaker to our ears? The dots of Fig. 1-5 represent air molecules with different density variations. The molecules crowded together represent areas of compression (crests in the wave shape) in which the air pressure is slightly greater than the prevailing atmospheric pressure (typically about 14.7 lb/in^2 at sea level). The sparse areas represent rarefactions (troughs in the wave shape) in which the pressure is slightly less than atmospheric pressure. The arrows (see Fig. 1-5) indicate that, on average, the molecules are moving to the right of the compression crests and to the left in the rarefaction troughs between the crests. Any given molecule, because of elasticity, after an initial displacement, will return toward its original position. It will move a certain distance to the right and then approximately the same distance to the left of its undisplaced position as the sound wave progresses uniformly to the right. Sound propagates because of the transfer of momentum from one particle to another.

In this example, why does the sound wave move to the right? The answer is revealed by a closer look at the arrows (see Fig. 1-5). The molecules tend to bunch up where two arrows are pointing toward each other, and this occurs a bit to the right of each compression region. When the arrows point away from each other, the density of molecules decreases. Thus, the movement of the higher-pressure crest and the lower-pressure trough accounts for the progression of the sound wave to the right.

As mentioned previously, the pressure at the crests is higher than the prevailing atmospheric barometric pressure and lower than the atmospheric pressure at the troughs, as shown in the sine wave of Fig. 1-6. These fluctuations of pressure are very small indeed. The faintest sound the ear can hear (20 μPa) exists at a pressure some 5,000 million times smaller than atmospheric pressure. To summarize, typical sounds such as speech and music are represented by correspondingly small ripples in pressure superimposed on the atmospheric pressure.

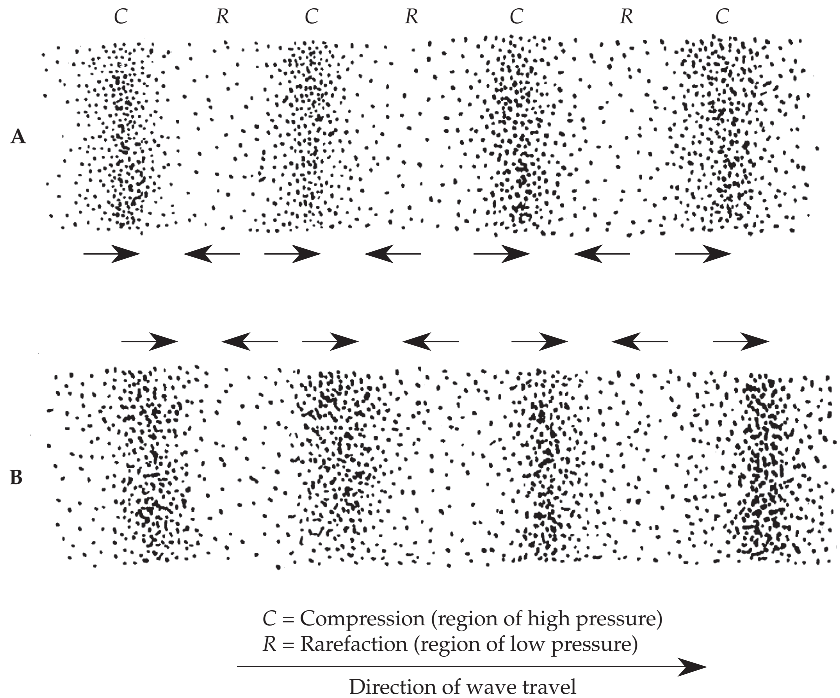


FIGURE 1-5 Sound waves traveling through a medium change the localized air particle density. (A) A sound wave causes the air particles to be pressed together (compression) in some regions and spread out (rarefaction) in others. (B) An instant later the sound wave has moved slightly to the right.

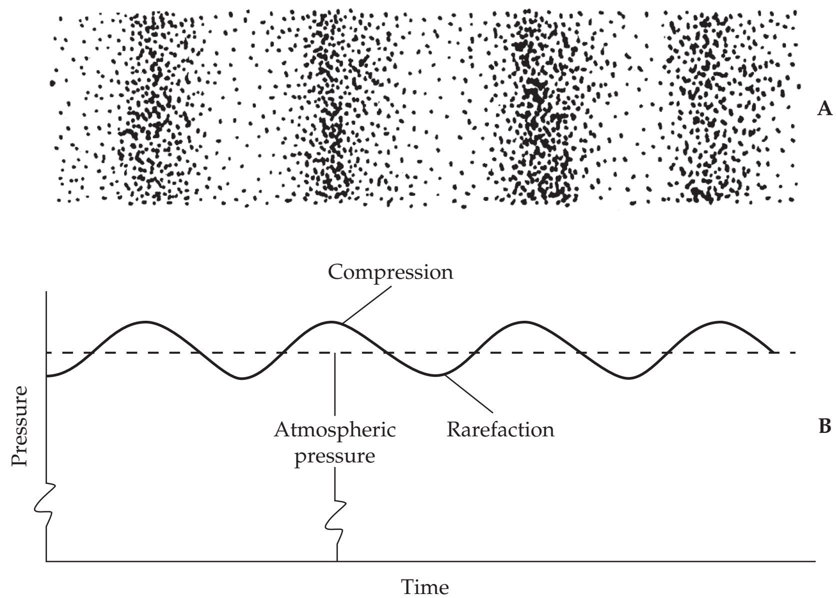


FIGURE 1-6 Pressure variations of sound waves are superimposed on prevailing barometric pressure. (A) An instantaneous view of the compressed and rarefied regions of a sound wave in air. (B) The compressed regions are very slightly above and the rarefied regions very slightly below atmospheric pressure.

Medium	Speed of Sound (ft/sec)	Speed of Sound (m/sec)
Air	1,130	344
Distilled water	4,915	1,498
Seawater	5,023	1,531
Wood, fir	12,470	3,800
Steel bar	16,570	5,050
Gypsum board	22,310	6,800

Table 1-1 Examples of Speed of Sound in Different Materials

Speed of Sound

The speed of sound in air is about 1,130 ft/sec (344 m/sec) at 70°F (21°C). This is about 770 mi/hr (1,239 km/hr). In the field of aerodynamics, this speed is known as Mach 1.0 (technically, it is air speed relative to the local speed of sound). This speed is not particularly fast in relation to familiar things. For example, commercial aircraft routinely travel at speeds that approach the speed of sound; for example, a Boeing 787 jetliner has a cruising speed of 561 mi/hr (Mach 0.85). The speed of sound is dramatically slower than the speed of light (670,616,629 mi/hr). It takes about 5 seconds for sound to travel 1 mile. You can gauge the distance of a thunderstorm by counting the time between the sight of the lightning flash and the sound of its thunder; if you count to 5 seconds, the storm is about a mile away. The speed of sound in the audible range is appreciably affected by temperature and slightly affected by humidity. It is not appreciably affected by the intensity of sound, its frequency, or by changes in atmospheric pressure. In some cases, some factors that would otherwise affect the speed of sound are offset by other factors, yielding insignificant changes.

Sound will propagate at a certain speed that depends on the medium and other factors. Other properties being equal, the stiffer or more rigid a medium, or the less compressible it is, the faster the speed of sound in it. Generally, sound travels faster in liquids than in air, and it travels faster in solids than in liquids. For example, sound travels at about 5,023 ft/sec in seawater and about 16,570 ft/sec in steel. Other examples are shown in Table 1-1. As noted, sound also travels faster in air as temperature increases (an increase of about 1.1 ft/sec for every degree Fahrenheit). Finally, humidity slightly affects the speed of sound in air; the more humid the air, the faster the speed. It should be noted that the speed (velocity) of sound is different from the particle velocity. The speed (velocity) of sound determines how fast sound energy moves through a medium. Particle velocity is determined by the loudness of the sound.

Wavelength and Frequency

A sine wave is illustrated in Fig. 1-7. The wavelength λ is the distance a wave travels in the time it takes to complete one cycle. A wavelength can be measured between successive peaks or between any two corresponding points on the cycle. This also holds for periodic waves other than the sine wave. The frequency f specifies the number of

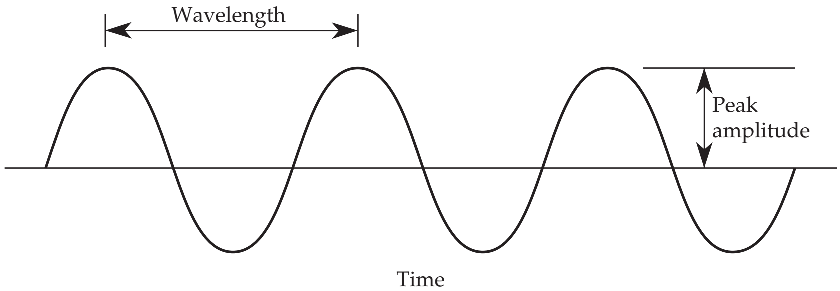


Figure 1-7 Wavelength is the distance a wave travels in the time it takes to complete one cycle. It can also be expressed as the distance from one point on a periodic wave to the corresponding point on the next cycle of the wave.

cycles per second, measured in hertz (Hz). Frequency and wavelength are related as follows:

$$\text{Wavelength (ft)} = \frac{\text{Speed of sound (ft/sec)}}{\text{Frequency (Hz)}} \tag{1-1}$$

which can also be written as

$$\text{Frequency (Hz)} = \frac{\text{Speed of sound (ft/sec)}}{\text{Wavelength (ft)}} \tag{1-2}$$

As noted, the speed of sound in air is about 1,130 ft/sec at normal conditions. For sound traveling in air, Eq. (1-2) becomes

$$\text{Wavelength (ft)} = \frac{1{,}130}{\text{Frequency (Hz)}} \tag{1-3}$$

This relationship is perhaps the most fundamentally important relationship in audio. Figure 1-8 gives two approaches for a graphical solution to Eq. (1-3).

Complex Waveforms

Speech and music wave shapes depart radically from the simple sine wave and are considered as complex waveforms. However, no matter how complex the waveform is, as long as it is periodic, it can be reduced to sine components. The obverse of this states that any complex periodic waveform can be synthesized from sine waves of different frequencies, different amplitudes, and different time relationships (phase). Joseph Fourier was the first to prove these relationships. The idea is simple in concept but often complicated in its application to specific speech or musical sounds. Let us see how a complex periodic waveform can be reduced to simple sinusoidal components.

Harmonics

A simple sine wave of a given amplitude and frequency, f_1, is shown in Fig. 1-9A. Figure 1-9B shows the second harmonic sine wave f_2 that is twice the frequency and half the amplitude of f_1. Combining f_1 and f_2 at each point in time, the wave shape of

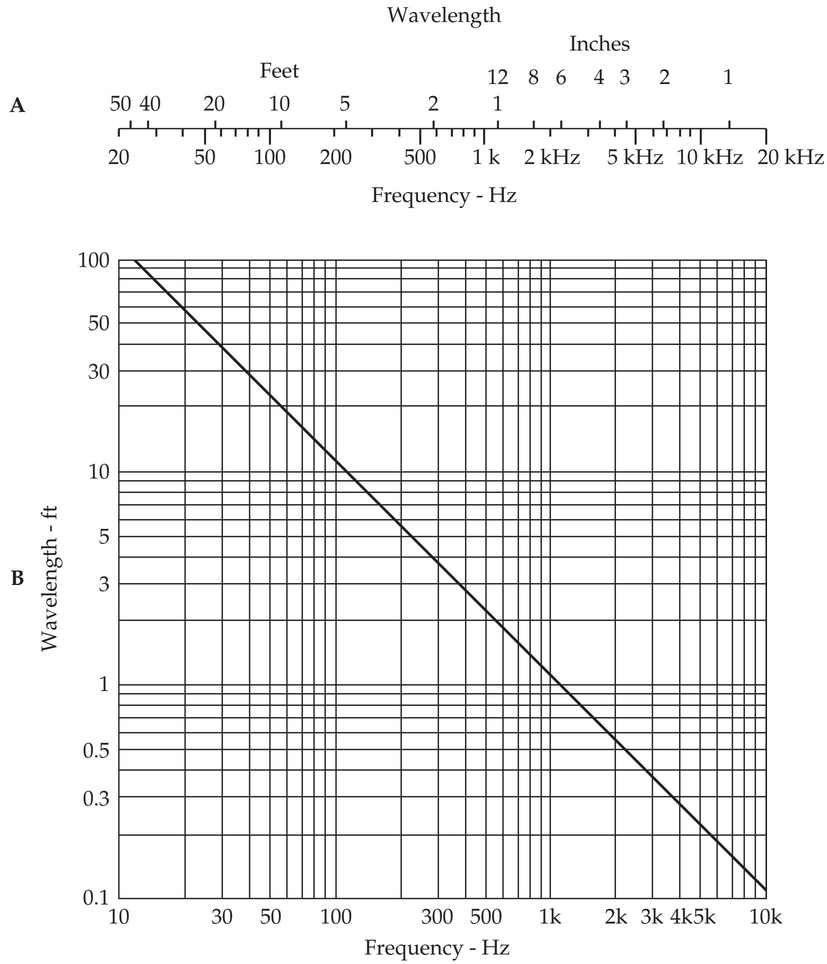


FIGURE 1-8 Wavelength and frequency are inversely related. (A) Scales for approximately determining wavelength of sound in air from a known frequency or vice versa. (B) A chart for determining the wavelength in air of sound waves of different frequencies. (Both are based on the speed of sound of 1,130 ft/sec.)

Fig. 1-9C is obtained. Figure 1-9D shows the third harmonic sine wave f_3 that is three times the frequency and half the amplitude of f_1. Adding this to the $f_1 + f_2$ wave shape of C, Fig. 1-9E is obtained. The simple sine wave of Fig. 1-9A has been progressively changed as other sine waves have been added to it; this is valid for both acoustic waves and electronic signals. The process can be reversed. The complex waveform of Fig. 1-9E can be disassembled, as it were, to the simple f_1, f_2, and f_3 sine components by either acoustic or electronic filters. For example, passing the waveform of Fig. 1-9E through a filter permitting only f_1 and rejecting f_2 and f_3, the original f_1 sine wave of Fig. 1-9A emerges in pristine condition.

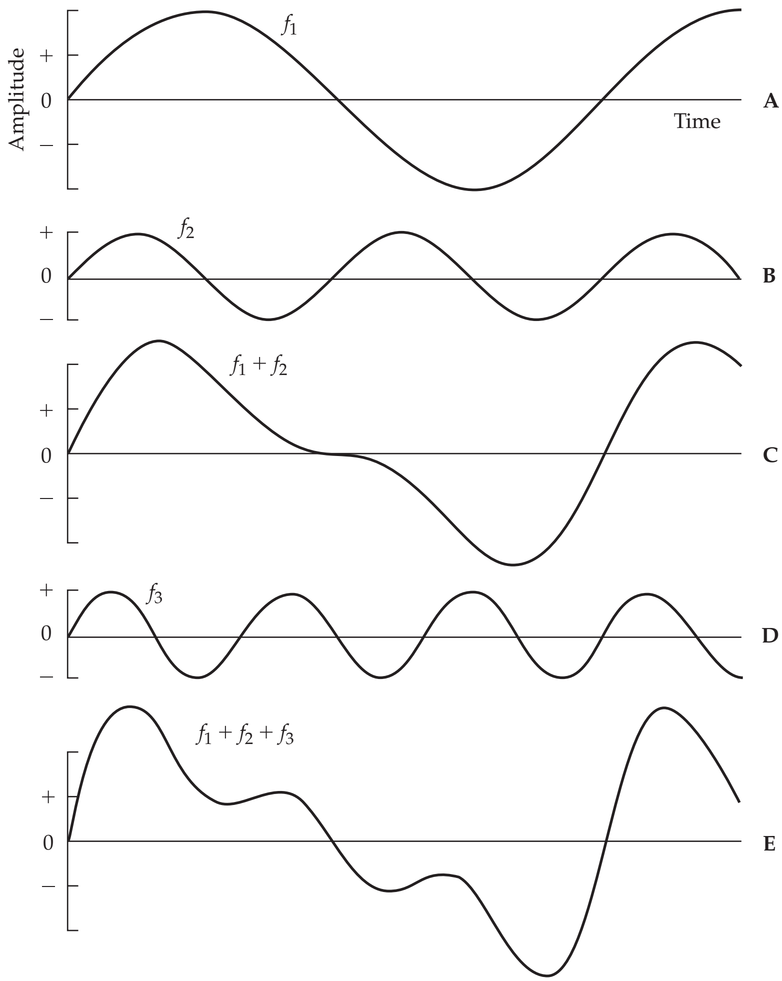


Figure 1-9 A study in the combination of sine waves. (A) The fundamental frequency f_1. (B) The second harmonic f_2 that is twice the frequency and half the amplitude of f_1. (C) The sum of f_1 and f_2 obtained by adding ordinates point by point. (D) The third harmonic f_3 that is three times the frequency and half the amplitude of f_1. (E) The waveform resulting from the addition of f_1, f_2, and f_3. All three components are in phase; that is, they all start from zero at the same instant.

The sine wave with the lowest frequency (f_1) of Fig. 1-9A is called the fundamental, the sine wave that is twice the frequency (f_2) of Fig. 1-9B is called the second harmonic, and the sine wave that is three times the frequency (f_3) of Fig. 1-9D is the third harmonic. The fourth harmonic and the fifth harmonic are four and five times the frequency of the fundamental, and so on.

Phase

In Fig. 1-9, all three components, f_1, f_2, and f_3, start together at the same time. This is called an in-phase condition. In many cases, the time relationships between harmonics and the fundamental, and between individual harmonics, are quite different from this. We observed that one revolution (360°) of the crankshaft of an automobile engine was equated to one cycle of simple harmonic motion of the piston. The up-and-down travel of the piston spread out in time traces a sine wave such as that in Fig. 1-10. One complete sine-wave cycle represents 360° of rotation. If another sine wave of identical frequency is delayed 90°, its time relationship to the first sine wave is a quarter wave late (time increasing to the right). A half-wave delay would be 180°, and so on. For the 360° delay, the waveform at the bottom of Fig. 1-10 synchronizes with the top one, reaching positive peaks and negative peaks simultaneously and again producing the in-phase condition.

Referring again to Fig. 1-9, all three components of the complex waveform of Fig. 1-9E are in phase. That is, the f_1 fundamental, the f_2 second harmonic, and the f_3 third harmonic all start at zero at the same time. What happens if the harmonics are out of

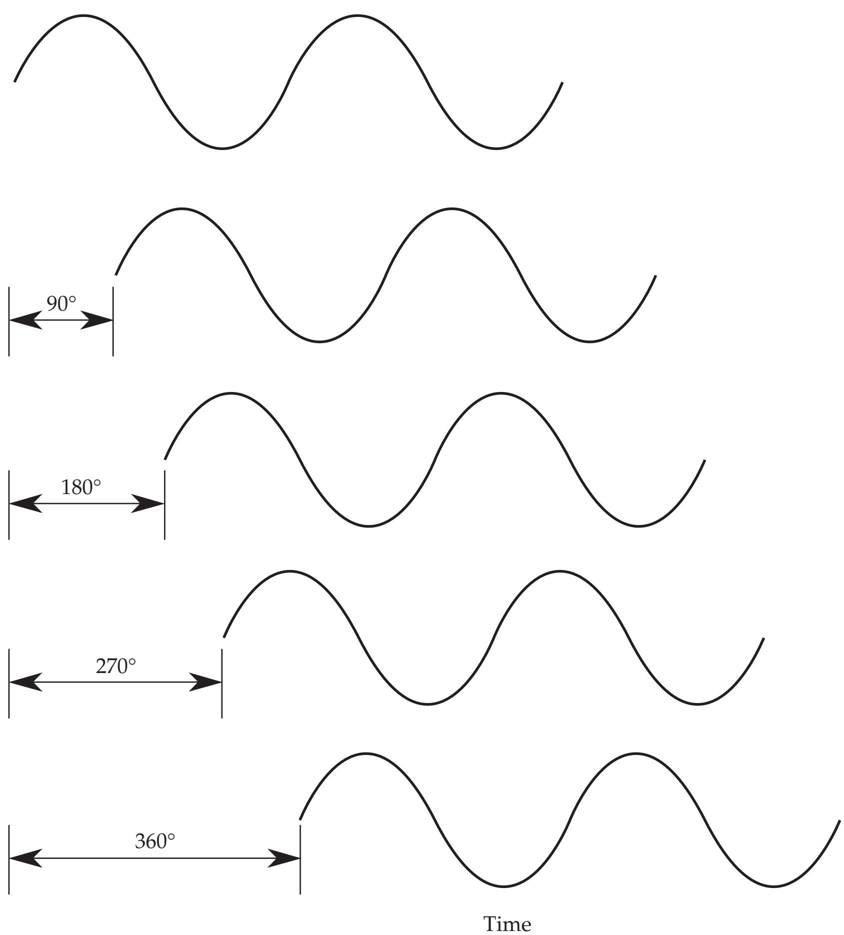


Figure 1-10 Illustration of the phase relationships between waveforms with the same amplitude and frequency. A rotation of 360° is analogous to one complete sine-wave cycle.

phase with the fundamental? Figure 1-11 illustrates this case. The second harmonic f_2 now leads f_1 by 90°, and the third harmonic f_3 lags f_1 by 90°. By combining f_1, f_2, and f_3 for each instant of time, with due regard to positive and negative signs, the contorted waveform of Fig. 1-11E is obtained.

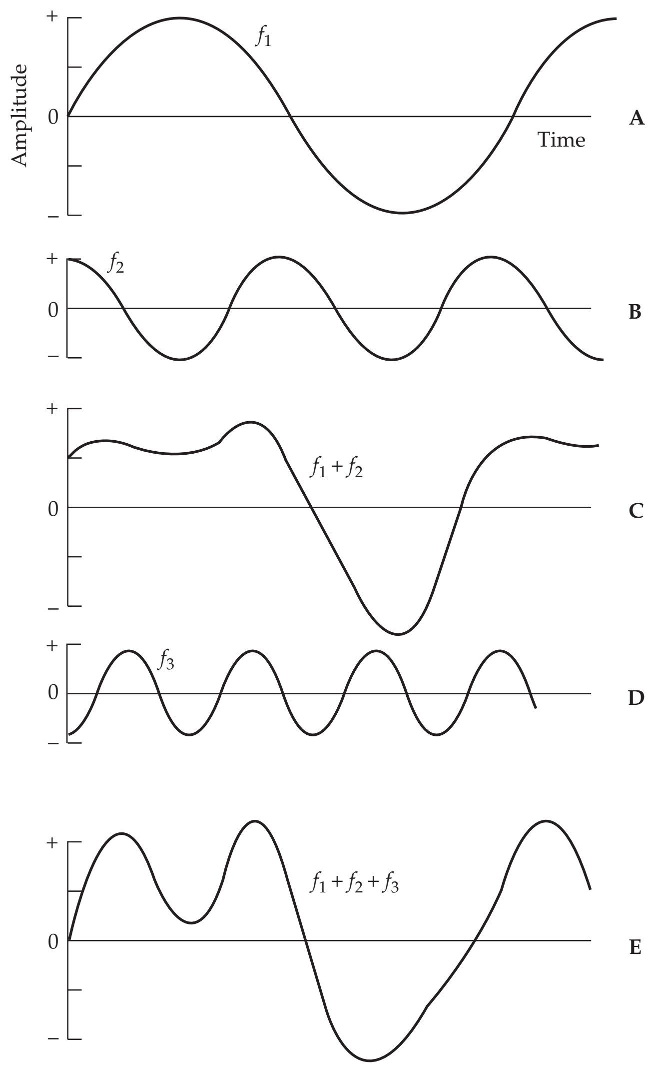


FIGURE 1-11 A study of the combination of sine waves that are not in phase. (A) The fundamental frequency f_1. (B) The second harmonic f_2 that is twice the frequency and half the amplitude of f_1 with phase leading f_1 by 90°. (C) The combination of f_1 and f_2 obtained by adding ordinates point by point. (D) The third harmonic f_3 that is three times the frequency and half the amplitude of f_1 with phase lagging f_1 by 90°. (E) The sum of f_1, f_2, and f_3. Compare this resulting waveform with that of Fig. 1-9E. The difference in waveforms is due entirely to the shifting of the phase of the harmonics with respect to the fundamental.

The difference between Figs. 1-9E and 1-11E is that a phase shift has been introduced between harmonics f_2 and f_3, and between the fundamental f_1. That is all that is needed to produce drastic changes in the resulting wave shape. Curiously, even though the shape of the waveform is dramatically changed by shifting the time relationships of the components, the ear is relatively insensitive to such changes. In other words, although waveforms E of Figs. 1-9 and 1-11 are visually very different, they would sound very much alike.

A common error is to confuse polarity with phase. Phase is the time relationship between two signals, while polarity is the +/− or the −/+ relationship of a given pair of signal leads.

Partials

Musicians may be inclined to use the term partial instead of harmonic, but it is best that a distinction be made between the two terms because the partials of some musical instruments are not harmonically related to the fundamental. That is, partials might not be exact multiples of the fundamental frequency, yet richness of tone can still be imparted by such deviations from the true harmonic relationship. For example, the partials of percussion instruments such as bells, cymbals, and chimes are in a nonharmonic relationship to the fundamental.

Octaves

Audio engineers and acousticians frequently use the integral multiple concept of harmonics, closely allied as it is to the physical aspect of sound. Musicians often refer to the octave, a logarithmic concept that is firmly embedded in musical scales and terminology because of its relationship to the ear's characteristics. Audio people are also involved with the human ear; hence their common use of logarithmic scales for frequency, logarithmic measuring units, and various devices based on octaves.

Harmonics and octaves are compared in Fig. 1-12. Harmonics are linearly related; each next harmonic is an integer multiple of the preceding one. An octave is defined as a 2:1 ratio of two frequencies. For example, middle C (C4) has a frequency close to 261 Hz. The next highest C (C5) has a frequency of about 522 Hz. Ratios of frequencies

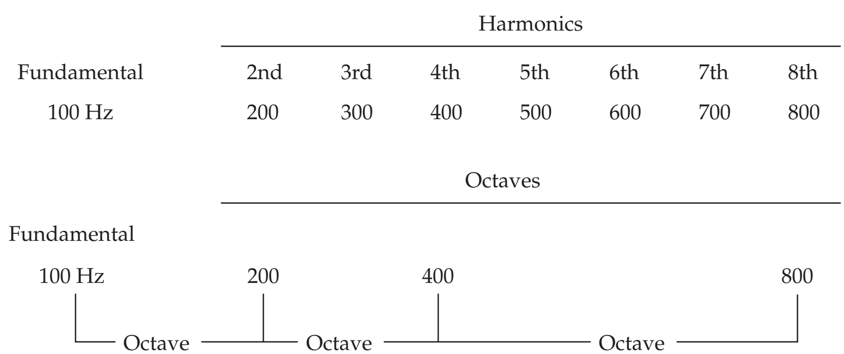


Figure 1-12 Comparison of harmonics and octaves. Harmonics are linearly related; octaves are logarithmically related.

are very much a part of the musical scale. The frequency ratio 2:1 is the octave; the ratio 3:2 is the fifth; 4:3 is the fourth; and so on.

The interval from 100 to 200 Hz is an octave, as is the interval from 200 to 400 Hz. The interval from 100 to 200 Hz is perceived as being larger than the interval from 200 to 300 Hz; this demonstrates that the ear perceives intervals as ratios rather than arithmetic differences. In particular, our perception of frequency is logarithmic. Because the octave is important in acoustical work, it is useful to consider the mathematics of the octave.

As the ratio of 2:1 is defined as the octave, its mathematical expression is

$$\frac{f_2}{f_1} = 2^n \tag{1-4}$$

where f_2 = frequency of the upper edge of the octave interval, Hz
f_1 = frequency of the lower edge of the octave interval, Hz
n = number of octaves

For one octave, $n = 1$ and Eq. (1-4) becomes $f_2/f_1 = 2$, which is the definition of the octave. Other applications of Eq. (1-4) are as follows:

Example 1 If the low-frequency limit of a band is 20 Hz, what is the high-frequency limit of a band that is 10 octaves wide?

$$\frac{f_2}{20} = 2^{10}$$
$$f_2 = (20)\,(2^{10})$$
$$f_2 = (20)\,(1{,}024)$$
$$f_2 = 20{,}480\ \text{Hz}$$

Example 2 If 446 Hz is the lower limit of a 1/3-octave band, what is the frequency of the upper limit?

$$\frac{f_2}{446} = 2^{1/3}$$
$$f_2 = (446)\,(2^{1/3})$$
$$f_2 = (446)\,(1.2599)$$
$$f_2 = 561.9\ \text{Hz}$$

Example 3 What is the lower limit of a 1/3-octave band centered on 1,000 Hz? The f_1 is 1,000 Hz but the lower limit would be 1/6-octave lower than the 1/3-octave, so $n = 1/6$:

$$\frac{f_2}{f_1} = \frac{1{,}000}{f_1} = 2^{1/6}$$
$$f_1 = \frac{1{,}000}{2^{1/6}}$$
$$f_1 = \frac{1{,}000}{1.12246}$$
$$f_1 = 890.9\ \text{Hz}$$

Example 4 What is the frequency of the lower limit of an octave band centered on 2,500 Hz?

$$\frac{2,500}{f_1} = 2^{1/2}$$

$$f_1 = \frac{2,500}{2^{1/2}}$$

$$f_1 = \frac{2,500}{1.4142}$$

$$f_1 = 1,767.8 \text{ Hz}$$

What is the upper limit?

$$\frac{f_2}{2,500} = 2^{1/2}$$

$$f_2 = (2,500)\,(2^{1/2})$$

$$f_2 = (2,500)\,(1.4142)$$

$$f_2 = 3,535.5 \text{ Hz}$$

In many acoustical applications, sound is considered as falling in eight octave bands, with center frequencies of 63; 125; 250; 500; 1,000; 2,000; 4,000; and 8,000 Hz. In some cases, sound is considered in terms of 1/3-octave bands, with center frequencies falling at 31.5; 50; 63; 80; 100; 125; 160; 200; 250; 315; 400; 500; 630; 800; 1,000; 1,250; 1,600; 2,000; 2,500; 3,150; 4,000; 5,000; 6,300; 8,000; and 10,000 Hz.

Spectrum

The commonly accepted range of the audible spectrum is 20 Hz to 20 kHz. This is the approximate range of frequencies that fall within the perceptual limits of the human ear; the range is one of the specific characteristics of the human ear. Here, in the context of sine waves and harmonics, we may establish the concept of spectrum. The visible spectrum of light has its counterpart in sound in the audible spectrum. We cannot see far-ultraviolet light because the frequency of its electromagnetic energy is too high for the eye to perceive. Nor can we see far-infrared light because its frequency is too low. There are likewise sounds that are too low (infrasound) and sounds that are too high (ultrasound) in frequency for the ear to hear.

Figure 1-13 shows several waveforms that typify the infinite number of different waveforms encountered in audio. These waveforms were produced by a signal generator and captured on an oscilloscope. To the right of each capture is the spectrum of that particular signal. The spectrum shows how the energy of the signal is distributed in frequency. In all but the signal of Fig. 1-13D, the audible range of the spectrum was processed with a waveform analyzer having a very sharp filter with a passband 5 Hz wide. In this way, concentrations of energy were located and measured.

For a sine wave, all the energy is concentrated at one fundamental frequency; there are no harmonics. As we have seen, this is the defining characteristic of a sine wave. In Fig. 1-13A, the sine wave produced by this particular signal generator is not a pure sine wave. There is some extraneous harmonic content, which is a distortion of the sine wave, but in scanning its spectrum, the harmonics detected were too low in amplitude to appear on the graph.

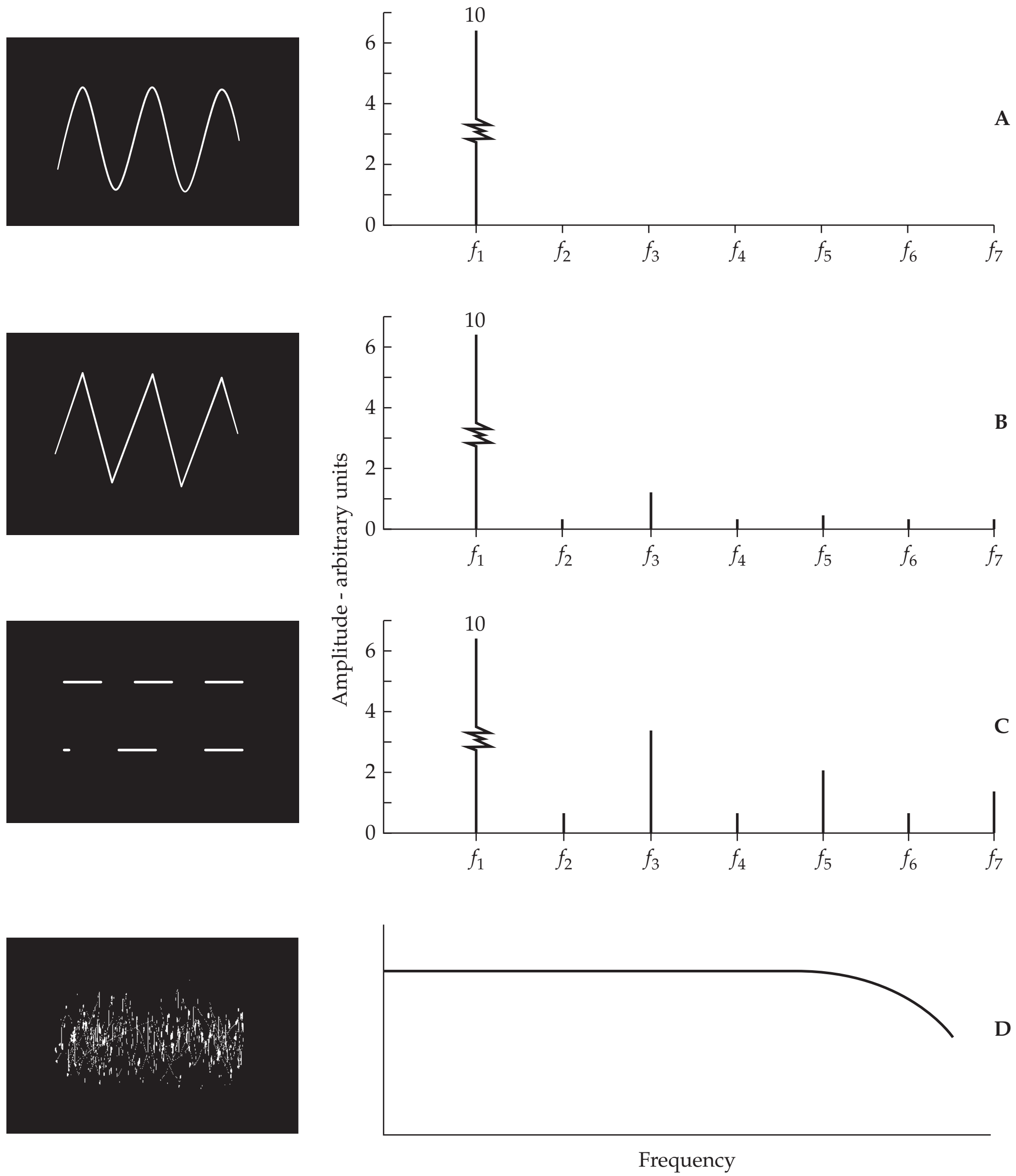


Figure 1-13 A comparison of basic waveforms and noise. (A) The spectral energy of a pure sinusoid is contained entirely at a single frequency. The triangular (B) and square (C) waveforms each have a prominent fundamental and numerous harmonics at integral multiples of the fundamental frequency. (D) White random noise has energy distributed uniformly throughout the spectrum up to some point at which energy begins to fall off. Random noise may be considered as statistically distributed signals with a continuous frequency spectrum.

The triangular waveform shown in Fig. 1-13B has a major fundamental component of 10 unit's magnitude. The waveform analyzer has detected a significant second harmonic component at f_2, twice the frequency of the fundamental with a magnitude of 0.21 units. The third harmonic shows an amplitude of 1.13 units, the fourth of 0.13 units, and so on. The seventh harmonic has an amplitude of 0.19 units and the fourteenth harmonic (about 15 kHz in this case) has an amplitude of only 0.03 units. We see that this triangular waveform has both odd and even components of modest amplitude through the audible spectrum. If we know the amplitude and phase of each of these, the original triangular waveform can be synthesized by combining them.

A comparable analysis reveals the spectrum of the square wave shown in Fig. 1-13C. It has harmonics of far greater amplitude than the triangular waveform with a distinct tendency toward more prominent odd than even harmonics. The third harmonic shows an amplitude of 34% of the fundamental. The fifteenth harmonic (not shown) of the square wave is 0.52 units. Figure 1-14A shows a square wave; it can be synthesized by adding harmonics to a fundamental. However, many harmonics would be needed. For example, Fig. 1-14B shows the waveform that results from adding two nonzero harmonic components, and Fig. 1-14C shows the result from adding nine nonzero harmonic components. This demonstrates why a bandlimited "square wave" does not have a square appearance.

The spectra of sine, triangular, and square waves reveal energy concentrated at harmonic frequencies, but nothing between. These are all periodic waveforms, which repeat themselves cycle after cycle. The fourth example in Fig. 1-13D is a random-noise signal; in particular, a white-noise signal with equal intensity at different frequencies.

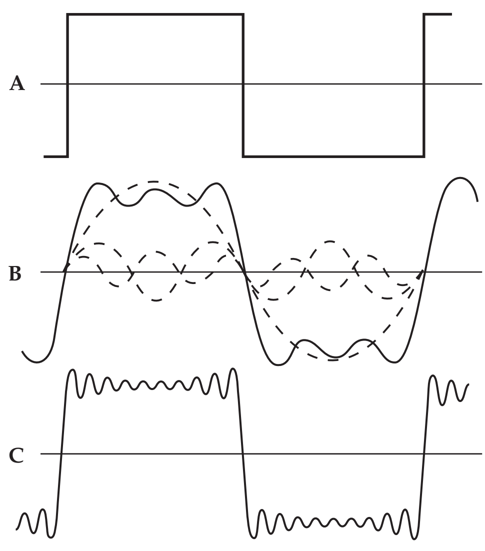


FIGURE 1-14 A square wave can be synthesizing by adding harmonics to the sine-wave fundamental. (A) A square wave (with an infinite number of harmonic components). (B) The summation of the fundamental and two nonzero harmonic components. (C) The summation of the fundamental and nine nonzero harmonic components. Clearly, many components would be needed to smooth the ripples and produce the square edges of (A).

The spectrum of this signal cannot be measured satisfactorily by a waveform analyzer with a 5-Hz passband because the fluctuations are so great that it is impossible to get an accurate reading. Analyzed by a wider passband of fixed bandwidth and with the help of various integrating devices to get a steady indication, the spectral shape shown is obtained. This spectrum tells us that the energy of the random-noise signal is equally distributed throughout the spectrum; the roll-off at high frequencies indicates that the upper frequency limit of the random-noise generator has been reached. The signal is still considered to be white noise because its energy spectrum is flat across the bandwidth of interest.

There is little visual similarity between the sine and the random-noise signals as revealed by an oscilloscope, yet there is a hidden relationship. Even random noise can be considered as comprising sine-wave components constantly shifting in frequency, amplitude, and phase. If we pass random noise through a narrow filter and observe the filter output on an oscilloscope, we will see a restless, sine-like wave continually shifting in amplitude. Theoretically, an infinitely narrow filter would sift out a pure sine wave.

Key Points

- Sound exists as wave motion in elastic media such as gases, solids, and liquids.
- Simple harmonic motion is a basic type of oscillatory motion, and yields a sine wave.
- Sound propagates because of the transfer of momentum from one particle to another.
- The speed of sound in air is nominally about 1,130 ft/sec, and varies according to the properties of the medium and other factors.
- The wavelength of sound is the distance a wave travels in the time it takes to complete one cycle.
- Speech and music wave shapes are complex waveforms. If they are periodic, they can be reduced to sine components known as harmonics.
- Phase describes the time relationships between waveforms.
- The commonly accepted range of the audible spectrum is 20 Hz to 20 kHz; this is the approximate range of frequencies that fall within the perceptual limits of the human ear.

CHAPTER 2

Sound Levels and the Decibel

The decibel is one of the most important units of measure in the field of audio. The decibel provides an extraordinarily efficient way to describe audio phenomena and our perception of them. In this chapter, we explore the decibel concept and see how decibels can be used to measure sound levels in various applications. Several numerical examples demonstrate the mathematics needed to use decibel measurements.

Sound levels expressed in decibels clearly demonstrate the wide range of sensitivity in human hearing. The threshold of hearing matches the ultimate lower limit of perceptible sound in air, that is, the noise of undisturbed air molecules bumping against the eardrum. At the other end of the range, the ear can tolerate high-intensity sound levels; at very high intensities, hearing damage is a very real possibility. A sound level expressed in decibels is a convenient way of handling the billion-fold range of sound pressures to which the ear is sensitive.

Ratios versus Differences

There are innumerable ways to measure changes in any phenomenon. For example, we might express a change as an absolute difference, or we might use a ratio. Which is better for expressing changes in the perceived loudness of sound?

Imagine a sound source set up in a room completely protected from interfering noise. The sound source is adjusted for a weak sound with a sound pressure of 1 unit, and its loudness is carefully noted. In observation A, to double the perceived loudness, the sound pressure must be increased from 1 to 10 units. The source pressure is now adjusted to 10,000 units. For observation B, to again double the perceived loudness, we find that the sound pressure must be increased from 10,000 to 100,000 units. The results of this experiment can be summarized as follows:

Observations	Two Pressures	Ratio of Two Pressures
A	10–1	10:1
B	100,000–10,000	10:1

Observations A and B accomplish the same doubling of perceived loudness. In observation A, this was accomplished by an increase in sound pressure of only 9 units, whereas in observation B, an increase of 90,000 units was required. We conclude that ratios of pressures seem to describe loudness changes better than absolute differences in pressure.

Early acoustics researchers including Ernst Weber, Gustaf Fechner, and Hermann von Helmholtz demonstrated the importance of using ratios in such measurements. Ratios apply equally well to sensations of vision, vibration, and even electric shock. Ratios of stimuli come closer to matching human perception than do absolute differences of stimuli. This matching is not perfect, but close enough to make a strong case for expressing sound levels in decibels, which are based on ratios.

Ratios of powers, or ratios of intensities, or ratios of sound pressure, voltage, current, or anything else are dimensionless. For example, the ratio of 1 W to 100 W is 1 W/100 W, and the watt unit in the numerator and the watt unit in the denominator cancel, leaving a pure number without dimension. This is important in our discussion of decibels because decibels use logarithms and logarithms can only be taken of nondimensional numbers.

Expressing Numbers

Table 2-1 illustrates three different ways numbers can be expressed. The decimal and arithmetic forms are familiar in everyday activities. The exponential form, while not as commonly used, has an almost unique ability to simplify the expression of many relationships. When writing "one hundred thousand" watts, we can express the number as 100,000 W or 10^5 W. When writing a "millionth of a millionth" of a watt, the string of zeros behind the decimal point is clumsy, but writing 10^{-12} W is easy. Engineering calculators display the exponential form in scientific notation, by which very large or very small numbers can be expressed. Moreover, the prefix pico means 10^{-12}, so the value can be expressed as 1 pW (shown later in Table 2-4).

Decimal Form	Arithmetic Form	Exponential Form
100,000	$10 \times 10 \times 10 \times 10 \times 10$	10^5
10,000	$10 \times 10 \times 10 \times 10$	10^4
1,000	$10 \times 10 \times 10$	10^3
100	10×10	10^2
10	10×1	10^1
1	10/10	10^0
0.1	1/10	10^{-1}
0.01	$1/(10 \times 10)$	10^{-2}
0.001	$1/(10 \times 10 \times 10)$	10^{-3}
0.0001	$1/(10 \times 10 \times 10 \times 10)$	10^{-4}
100,000	(100)(1,000)	$10^2 + 10^3 = 10^{2+3} = 10^5$
100	10,000/100	$10^4/10^2 = 10^{4-2} = 10^2$
10	100,000/10,000	$10^5/10^4 = 10^{5-4} = 10^1 = 10$
10	$\sqrt{100} = \sqrt[2]{100}$	$100^{1/2} = 100^{0.5}$
4.6416	$\sqrt[3]{100}$	$100^{1/3} = 100^{0.333}$
31.6228	$\sqrt[4]{100^3}$	$100^{3/4} = 100^{0.75}$

TABLE 2-1 Expressing Numbers in Decimal, Arithmetic, and Exponential Form

The softest sound intensity we can hear (the threshold of audibility) is about 10^{-12} W/m^2. A very loud sound (causing a sensation of pain) might be 10 W/m^2. (Acoustic intensity is acoustic power per unit area in a specified direction.) This range of intensities from the softest sound to a painfully loud sound is 10,000,000,000,000. Clearly, it is more convenient to express this range as an exponent, 10^{13}. Furthermore, it is useful to establish the intensity of 10^{-12} W/m^2 as reference intensity I_{ref} and express other sound intensities I as a ratio I/I_{ref} to this reference. For example, the sound intensity of 10^{-9} W/m^2 would be written as 10^3 or 1,000 (the ratio is dimensionless). We see that 10^{-9} W/m^2 is 1,000 times the reference intensity.

Logarithms

Representing 100 as 10^2 simply means that $10 \times 10 = 100$. Similarly, 10^3 means $10 \times 10 \times 10 = 1{,}000$. But what about 267? This is where logarithms are useful. Logarithms are proportional numbers, and a logarithmic scale is one that is calibrated proportionally. It is agreed that 100 equals 10^2. By definition, we can say that the logarithm of 100 to the base 10 equals 2, commonly written $\log_{10} 100 = 2$, or simply $\log 100 = 2$, because common logarithms are to the base 10. The number 267 can be expressed as 10 to some power between 2 and 3. Avoiding the mathematics, we can use a calculator to enter 267, push the "log" button, and 2.4265 appears. Thus, $267 = 10^{2.4265}$ and $\log 267 = 2.4265$. Logarithms are handy because, as Table 2-1 demonstrates, they reduce multiplication to addition and division to subtraction.

Logarithms are particularly useful to audio engineers because they can correlate measurements to human hearing, and they also allow large ranges of numbers to be expressed efficiently. Logarithms are the foundation for expressing sound levels in decibels where the sound level is a logarithm of a ratio. In particular, a sound level in decibels is 10 times the logarithm to the base 10 of the ratio of two power-like quantities as described below.

Decibels

We observed that it is useful to express sound intensity in ratios. Furthermore, we can express the intensities as logarithms of the ratios. An intensity I can be expressed in terms of a reference I_{ref} as follows:

$$\log_{10} \frac{I}{I_{ref}} \text{ bels} \tag{2-1}$$

The intensity measure is dimensionless, but to clarify the value, we assign it the unit of a bel (from Alexander Graham Bell). However, when expressed in bels, the range of values is somewhat small. To make the range easier to use, we usually express values in decibels. The decibel is 1/10 bel. A decibel (dB) is 10 times the logarithm to base 10 of the ratio of two quantities of intensity (or power). Thus, the intensity ratio in decibels becomes:

$$\text{IL} = 10 \log_{10} \frac{I}{I_{ref}} \text{ decibels} \tag{2-2}$$

This value is called the sound intensity level (IL in decibels) and differs from intensity (I in watts/m^2). Using decibels is convenient, and decibel values more closely follow the way we hear the loudness of sounds.

Questions sometimes arise when levels other than intensity need to be expressed in decibels. Equation (2-2) applies equally to acoustic intensity, as well as acoustic power, electric power, or any other kind of power. For example, we can write the sound-power level as:

$$\text{PWL} = 10 \log_{10} \frac{W}{W_{\text{ref}}} \text{ decibels} \tag{2-3}$$

where PWL = sound-power level, dB
W = sound power, watts
W_{ref} = a reference power, 10^{-12} W

Sound intensity is difficult to measure. Sound pressure is usually the most accessible parameter to measure in acoustics (just as voltage is for electronic circuits). For this reason, the sound-pressure level (SPL) is often used. SPL is a logarithmic value of the sound pressure, in the same way that the sound intensity level (IL) corresponds to sound intensity. SPL is approximately equal to IL; both are often referred to as the sound level. Acoustic intensity (or power) is proportional to the square of the acoustic pressure p. This slightly alters the defining equation that we use. When the reference pressure is 20 µPa (micropascals), a sound pressure p measured in micropascals has an SPL of:

$$\begin{aligned} \text{SPL} &= 10 \log_{10} \frac{p^2}{p_{\text{ref}}^2} \\ &= 20 \log_{10} \frac{p}{20\ \mu\text{Pa}} \text{ decibels} \end{aligned} \tag{2-4}$$

where SPL = sound-pressure level, dB
p = acoustic pressure, µPa or other
p_{ref} = acoustic reference pressure, µPa or other

The tabulation of Table 2-2 shows whether Eq. (2-2) or (2-3), or Eq. (2-4) applies.

	Equation (2-2) or (2-3)	**Equation (2-4)**
Parameter	$10 \log_{10}(a_1/a_2)$	$20 \log_{10}(b_1/b_2)$
Acoustic		
Intensity	X	
Power	X	
Air particle velocity		X
Pressure		X
Electric		
Power	X	
Current		X
Voltage		X
Distance (from source-SPL; inverse square)		X

Table 2-2 Use of 10 log and 20 log Equations

Reference Levels

As we have seen, reference levels are widely used to establish a baseline for measurements. For example, a sound-level meter is used to measure a certain sound-pressure level. If the corresponding sound-pressure level is expressed in normal pressure units, a wide range of very large and very small numbers results. As we have seen, by expressing levels in decibels, we compress the large and small ratios into a more convenient and comprehensible range. Basically, a sound-level meter reading is a certain sound-pressure level, 20 log (p/p_{ref}), as in Eq. (2-4). The sound-pressure reference p_{ref} must be standardized so that ready comparisons can be made. Several such reference pressures have been used over the years, but for sound in air the standard reference pressure is 20 μPa. This might seem quite different from the reference pressure of 0.0002 microbar or 0.0002 dyne/cm^2, but it is the same standard merely written in different units. This is a very small sound pressure (0.0000000035 lb/in^2) and corresponds closely to the threshold of human hearing at 1 kHz. The relationship between sound pressure in pascals, pounds/square inch, and sound-pressure level is shown in Fig. 2-1.

When a statement is encountered such as, "The sound-pressure level is 82 dB," the 82-dB sound-pressure level is normally used in direct comparison with other levels.

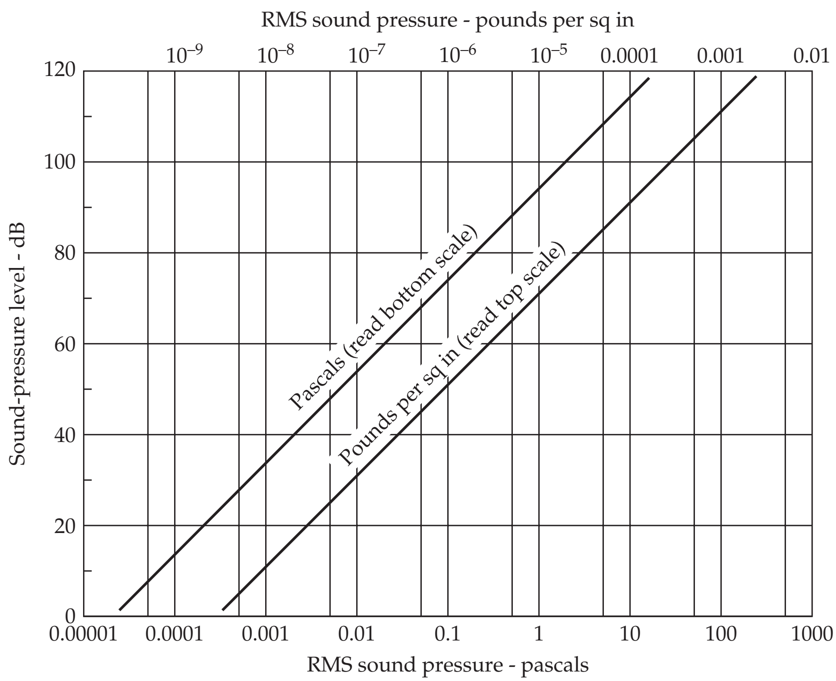


Figure 2-1 This graph shows the relationship between sound pressure in pascals or pounds/square inch, and sound-pressure level (referred to 20 μPa). This is a graphical approach to the solution of Eq. (2-2).

However, if the sound pressure were needed, it can be computed readily by working backward from Eq. (2-4) as follows:

$$82 = 20 \log \frac{p}{20\ \mu\text{Pa}}$$

The y^x button on a calculator helps us evaluate $10^{4.1}$. Enter 10, enter 4.1, press the y^x power button, and the value 12,589 appears.

$$p = (20\ \mu\text{Pa})(12{,}589)$$

$$p = 251{,}785\ \mu\text{Pa}$$

There is another lesson here. The 82 has two significant figures. The 251,785 has six significant figures and implies a precision that is not there. Just because a calculator says so doesn't make it so. A better answer is 252,000 μPa, or 0.252 Pa.

Logarithmic and Exponential Forms Compared

The logarithmic and exponential forms of numerical expression are equivalent as can be seen in Table 2-1. When working with decibels, it is important that this equivalence be understood.

Let's say we have a power ratio of 5:

$$10 \log_{10} 5 = 6.99 \text{ is exactly equivalent to}$$

$$5 = 10^{6.99/10}$$

There are two 10s in the exponential statement, but they come from different sources as indicated by the arrows. Now let us treat a sound pressure ratio of 5:

$$20 \log_{10} 5 = 13.98$$

$$5 = 10^{13.98/20}$$

Remember that sound-pressure level in air means that the reference pressure (p_{ref}) in the pressure ratio is 20 μPa. There are other reference quantities; some of the most common ones are listed in Table 2-3. The prefixes of Table 2-4 are often employed when dealing with very small and very large numbers. These prefixes are the Greek names for the power exponents of 10.

Level in Decibels	Reference Quantity
Acoustic	
Sound-pressure level in air (SPL, dB)	20 μPa
Power level (Lp, dB)	1 pW (10^{-12} W)
Electric	
Power level re 1 mW	10^{-3} W (1 mW)
Voltage level re 1 V	1 V
Volume level, VU	10^{-3} W

TABLE 2-3 Reference Quantities in Common Use

Prefix	Symbol	Multiple
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}

TABLE 2-4 Prefixes, Symbols, and Exponents

Acoustic Power

It doesn't take many watts of acoustic power to produce very loud sounds. This is fortunate for music playback because loudspeaker efficiency (output for a given input) is very low, perhaps on the order of 10%. On the other hand, increasing amplifier power to achieve higher acoustic levels can be frustrating. Doubling amplifier power from 1 to 2 W is a 3-dB increase in power level (10 log 2 = 3.01), but this yields a very small increase in loudness. Similarly, an increase in amplifier power from 100 to 200 W or 1,000 to 2,000 W yields the same 3-dB increase in level.

Table 2-5 lists sound pressure and sound-pressure levels of various sounds. There is a vast difference in sound pressure from 0.00002 Pa (20 μPa) to 100,000 Pa, but this range is reduced to a convenient form when expressed in sound levels. The same information is present in graphical form in Fig. 2-2.

The 194-dB sound-pressure level of a Saturn V rocket launch is considerable (even higher levels were variously reported during launches). (For perspective, detonating 50 lb of TNT 10 ft away would generate a similar SPL.) That sound-pressure level approaches the atmospheric pressure and hence is a disturbance on the order of magnitude of atmospheric pressure. The 194-dB sound pressure is an RMS (root-mean-square) value. A peak sound pressure 1.4 times as great would modulate the atmospheric pressure completely.

Sound Source	Sound Pressure (Pa)	Sound-Pressure Level* (dB, A-Weighted)
Saturn V rocket	100,000	194
Ram jet	2,000	160
Propeller aircraft	200	140
Riveter	20	120
Heavy truck	2	100
Noisy office or heavy traffic	0.2	80
Conversational speech	0.02	60
Quiet residence	0.002	40
Leaves rustling	0.0002	20
Hearing threshold, excellent ears at frequency maximum response	0.00002	0

*Reference pressure (these are identical):
20 μPa (micropascals)
0.00002 Pa (pascals)
2×10^{-5} N/m^2
0.0002 dyne/cm^2 or μbar

Table 2-5 Examples of Sound Pressures, and Sound-Pressure Levels

Using Decibels

As we have seen, a sound level is expressed as a logarithm of a ratio of two power-like quantities. When levels are computed from other than power ratios, certain conventions are observed. The convention for Eq. (2-4) is that sound power is proportional to sound-pressure squared. The voltage-level gain of an amplifier in decibels is 20 log (output voltage/input voltage), which holds true regardless of the input and output impedances. However, for power-level gain, the impedances must be considered if they are different. It is important to clearly indicate what type of level is intended, or else label the gain in level as "relative gain, dB." The following examples illustrate the use of the decibel.

Example 1: Sound-Pressure Level A sound-pressure level is 78 dB. What is the sound pressure?

$$78 \text{ dB} = 20 \log p/(20 \times 10^{-6})$$
$$\log p/(20 \times 10^{-6}) = 78/20$$
$$p/(20 \times 10^{-6}) = 10^{3.9}$$
$$p = (20 \times 10^{-6})(7,943.3)$$
$$p = 0.159 \text{ Pa}$$

Note that the reference level in SPL measurements is 20 μPa.

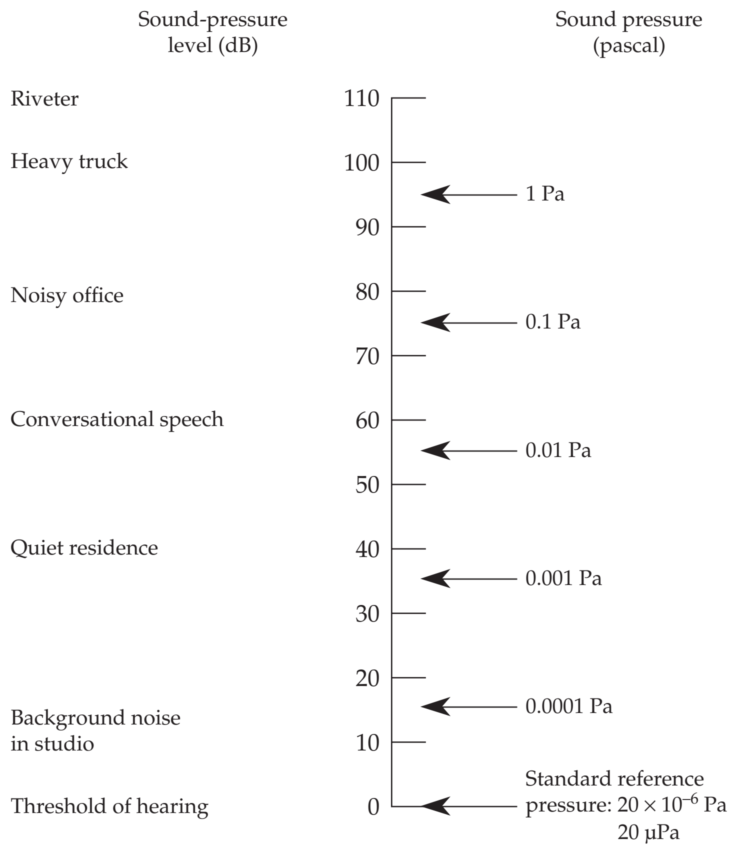


FIGURE 2-2 An appreciation of the relative magnitude of a sound pressure of 1 Pa can be gained by comparison to known sounds. The standard reference pressure for sound in air is 20 μPa, which corresponds closely to the minimum audible pressure.

Example 2: Loudspeaker Sound-Pressure Level A loudspeaker with an input of 1 W into 8 Ω produces an output sound-pressure level (SPL) of 115 dB on axis at 1 m (3 ft). What is the SPL at 6.1 m (20 ft)?

$$\begin{aligned} \text{SPL} &= 115 - 20 \log (6.1/1) \\ &= 115 - 15.7 \\ &= 99.3 \text{ dB} \end{aligned}$$

The assumption made in the 20 log 6.1 factor is that the loudspeaker is operating in a free field and that the inverse square law is valid in this case. This is a reasonable assumption for a 20-ft distance if the loudspeaker is operating away from reflecting surfaces. The free field is described in more detail in Chap. 3.

This loudspeaker is rated at an SPL of 115 dB at 1 m with 1 W. If the input were decreased from 1 W to 0.22 W, what would be the SPL at 1-m distance?

$$\begin{aligned} \text{SPL} &= 115 - 10 \log (0.22/1) \\ &= 115 - 6.6 \\ &= 108.4 \text{ dB} \end{aligned}$$

Note that 10 log is used because two powers are being compared.

Example 3: Microphone Voltage An omnidirectional dynamic microphone has an open-circuit voltage specified as –80 dB for the 150-Ω case. It is also specified that 0 dB = 1 V/μbar. What is the open-circuit voltage ν in volts?

$$
\begin{aligned}
-80 \text{ dB} &= 20 \log \nu/1 \\
\log \nu/1 &= -80/20 \\
\nu &= 0.0001 \text{ V} \\
&= 0.1 \text{ mV}
\end{aligned}
$$

Example 4: Line Amplifier Output Voltage A line amplifier has an input impedance of 600 Ω and output impedance of 600 Ω. It has a gain of 37 dB. With an input of 0.2 V, what is the output voltage?

$$
\begin{aligned}
37 \text{ dB} &= 20 \log (\nu/0.2) \\
\log(\nu/0.2) &= 37/20 \\
&= 1.85 \\
\nu/0.2 &= 10^{1.85} \\
\nu &= (0.2)(70.79) \\
\nu &= 14.16 \text{ V}
\end{aligned}
$$

Example 5: General-Purpose Amplifier Gain An amplifier has a bridging input impedance of 10,000 Ω and an output impedance of 600 Ω. With a 50-mV input, an output of 1.5 V is observed. What is the voltage gain of the amplifier?

$$
\begin{aligned}
\text{Voltage gain} &= 20 \log (1.5/0.05) \\
&= 29.5 \text{ dB}
\end{aligned}
$$

It must be noted that this is not a power-level gain because of the differences in impedance. However, a calculation of voltage gain may serve a practical purpose in certain cases.

Example 6: Concert Hall Calculations A seat in a concert hall is 84 ft from the tympani. The tympanist strikes a single note. The sound-pressure level of the direct sound of the note at the seat is measured to be 55 dB. The first reflection from the nearest sidewall arrives at the seat 105 msec after the arrival of the direct sound. (A) How far does the reflection travel to reach the seat? (B) What is the SPL of the reflection at the seat, assuming perfect reflection at the wall? (C) How long will the reflection be delayed after arrival of the direct sound at the seat?

(A) $\text{Distance} = (1{,}130 \text{ ft/sec})\ (0.105 \text{ sec})$
$= 118.7 \text{ ft}$

(B) First, the level L, 1 ft from the tympani, must be estimated:

$$55 = L - 20 \log (84/1)$$

$$L = 55 + 38.5$$

$$L = 93.5 \text{ dB}$$

The SPL of the reflection at the seat is:

$$\text{dB} = 93.5 - 20 \log (118.7/1)$$

$$= 93.5 - 41.5$$

$$= 52 \text{ dB}$$

(C) The reflection will arrive after the direct sound at the seat after:

$$\text{Delay} = (118.7 - 84)/1{,}130 \text{ ft/sec}$$
$$= 30.7 \text{ msec}$$

A free field is also assumed here. The 30.7-msec reflection might be called an incipient echo.

Example 7: Combining Decibels A studio has a heating, ventilating, and air-conditioning (HVAC) system, as well as a floor-standing fan. If both the HVAC and fan are turned off, a very low noise level prevails, low enough to be neglected in this calculation. If the HVAC alone is operating, the sound-pressure level at a given position is 55 dB. If the fan alone is operating, the sound-pressure level is 60 dB. What will be the sound-pressure level if both are operating at the same time? The answer, most certainly, is not 55 + 60 = 115 dB. Rather,

$$\text{Combined dB} = 10 \log \left(10^{\frac{55}{10}} + 10^{\frac{60}{10}}\right)$$
$$= 61.19 \text{ dB}$$

We see that the 55-dB sound level only slightly increases the overall sound-pressure level established by the 60-dB level.

In another example, if the combined level of two noise sources is 80 dB and the level with one of the sources turned off is 75 dB, what is the level of the remaining source?

$$\text{Difference dB} = 10 \log \left(10^{\frac{80}{10}} + 10^{\frac{75}{10}}\right)$$
$$= 78.3 \text{ dB}$$

In other words, combining the 78.3-dB level with the 75-dB level gives a combined level of 80 dB. Both the HVAC and the fan are assumed to yield a wideband noise signal. It is important to remember that different sound levels can only be compared when the sounds have the same or similar spectral properties.

Measuring Sound-Pressure Level

A sound-level meter is designed to give readings of sound-pressure level (SPL). Sound pressure in decibels is referenced to the standard reference level, 20 μPa. The human hearing response is not flat across the audio band. For example, our hearing sensitivity particularly rolls off at low frequencies, and also at high frequencies. Moreover, this roll-off is more pronounced at softer listening levels. For this reason, to emulate human hearing, sound level meters usually offer a selection of weighting networks designated A, B, and C, having frequency responses shown in Fig. 2-3. The networks reduce the measured sound-pressure level at low and high frequencies. The A network is an inversion of the 40-phon hearing response, the B network is an inversion of 70-phon response, and the C network is an inversion of 100-phon response. The phon is described in Chap. 4. A particular network is selected based on the general level of sounds to be measured (background noise, jet engines, and so on). For example:

- For sound-pressure levels of 20 to 55 dB, use network A.
- For sound-pressure levels of 55 to 85 dB, use network B.
- For sound-pressure levels of 85 to 140 dB, use network C.

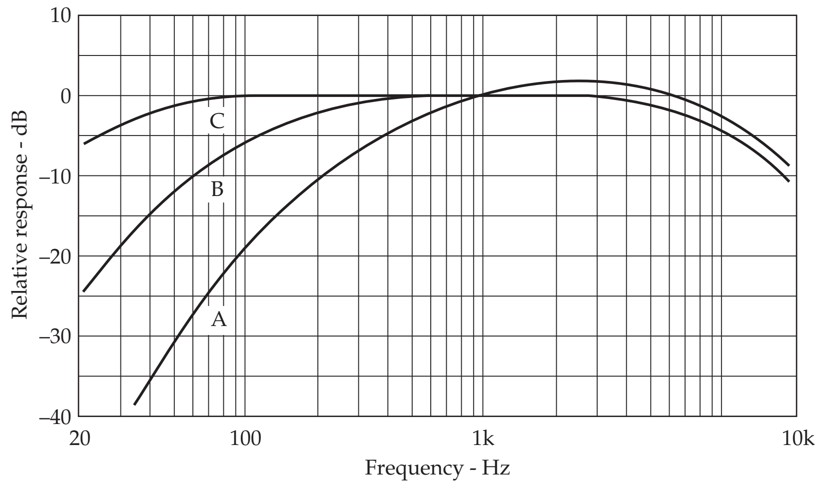


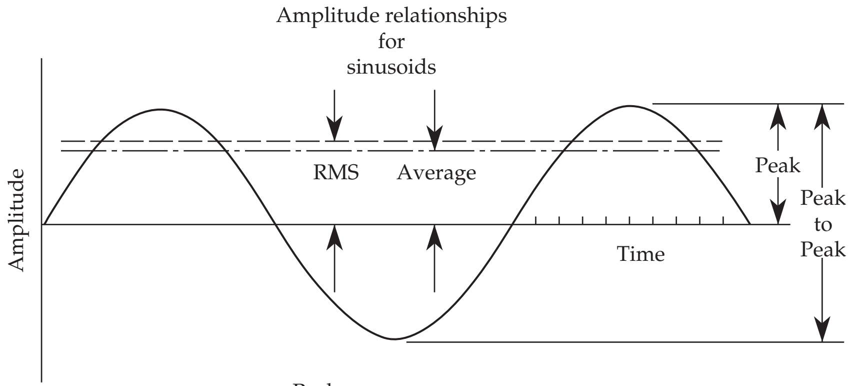
Figure 2-3 The A, B, and C weighting response characteristics for sound-level meters. (ANSI S1.4-1981.) A weighting is most commonly used.

These network response shapes were designed to bring the sound-level meter readings into closer conformance to the relative loudness of sounds. However, the B and C weightings often do not correspond to human perception; the use of B weighting is no longer recommended. The Z (zero) weighting describes a flat frequency response; it is described in the IEC 61672 standard. The A weighting is widely used for many acoustical noise measurements including environmental measurements. When a measurement is made with A weighting, the value is designated as dBA or dB(A). When making dBA measurements with a calibrated microphone, the dB SPL units are referenced to 20 μPa = 0 dB SPL. Typically, dBA readings are lower than unweighted dB readings. Since A weighting is essentially flat above 1 kHz, any difference between a dBA and unweighted reading primarily shows differences in the low-frequency content of the signal. For example, a large difference in readings shows that the signal has significant low-frequency content. The A weighting network is also described in the IEC 61672 standard.

Simple frequency weightings such as these cannot accurately represent loudness. Measurements made with simple weightings are not accepted as measurement loudness levels, but are only used for comparison of levels. Frequency analysis of sounds is recommended, using octave or 1/3-octave bands. More advanced measurement techniques are described in App. A.

Sine-Wave Measurements

The sine wave (or sinusoid) is a specific kind of alternating signal and is described by its own set of specific terms. Viewed on an oscilloscope, the easiest value to read is the peak-to-peak value (of voltage, current, sound pressure, or whatever the



$$\text{RMS} = \frac{\text{Peak}}{\sqrt{2}} = (0.707)\,(\text{Peak})$$

$$= \left(\frac{\pi}{2\sqrt{2}}\right)(\text{Average}) = (1.1)\,(\text{Average})$$

$$\text{Peak} = (\sqrt{2})\,(\text{RMS}) = (1.414)\,(\text{RMS})$$

$$= \left(\frac{\pi}{2}\right)(\text{Average}) = (1.57)\,(\text{Average})$$

Figure 2-4 Amplitude relationships for sinusoids apply to sine waves of electrical voltage or current, as well as to acoustical parameters such as sound pressure. Another term used in the audio field is crest factor, or peak divided by RMS. These mathematical relationships apply to sine waves and cannot be applied to complex waveforms.

sine wave represents), as shown in Fig. 2-4. If the wave is symmetrical, the peak-to-peak value is twice the peak value.

The common ac (alternating current) voltmeter is, in reality, a dc (direct current) instrument fitted with a rectifier that changes the alternating sine-wave current to pulsating unidirectional current. The dc meter then responds to the average value, as indicated in Fig. 2-4. However, such meters are almost universally calibrated in terms of RMS (root-mean-square, described as follows). For pure sine waves, this is quite an acceptable fiction, but for nonsinusoidal waveforms, the reading will be in error.

An alternating current of 1 A (ampere) RMS (or effective) is exactly equivalent in heating power to 1 A of direct current as it flows through a resistance of known value. After all, alternating current can heat a resistor or do work no matter which direction it flows; it is just a matter of evaluating it. In the right-hand positive curve of Fig. 2-4, the ordinates (height of lines to the curve) are read for each marked increment of time. Then (a) each of these ordinate values is squared, (b) the squared values are added together, (c) the average is found, and (d) the square root is taken of the average (or mean). Taking the square root of this average gives the RMS (root-mean-square) value of the positive curve of Fig. 2-4. The same can be done for the negative curve (squaring a negative ordinate gives a positive value), but simply doubling the positive curve of a symmetrical waveform is easier. In this way, the RMS or "heating-power" value of any alternating or periodic waves can be determined, whether the wave represents voltage, current, or sound pressure. Figure 2-4 also provides a summary of

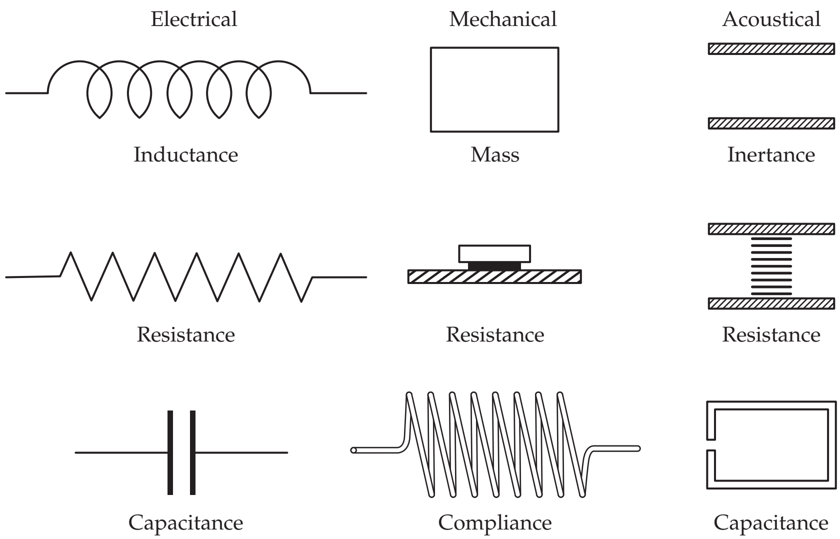


Figure 2-5 The three basic elements of electrical systems and their analogs in mechanical and acoustical systems.

relationships pertaining only to the sine wave. It is important to remember that simple mathematical relationships properly describe sine waves, but they cannot be applied to complex sounds.

Electrical, Mechanical, and Acoustical Analogs

An acoustical system such as a loudspeaker can be represented in terms of an equivalent electrical or mechanical system. An engineer uses these equivalents to set up a mathematical approach for analyzing a given system. For example, the effect of a cabinet on the functioning of a loudspeaker is clarified by thinking of the air in the enclosed space as acting like a capacitor in an electric circuit, absorbing and giving up the energy imparted by the cone movement.

Figure 2-5 shows the graphical representation of the three basic elements in electrical, mechanical, and acoustical systems. Inductance in an electrical circuit is equivalent to mass in a mechanical system and inertance in an acoustical system. Capacitance in an electrical circuit is analogous to compliance in a mechanical system and capacitance in an acoustical system. Resistance is resistance in all three systems, whether it is the frictional losses offered to air-particle movement in glass fiber, frictional losses in a wheel bearing, or resistance to the flow of current in an electrical circuit.

Key Points

- In many cases, ratios and logarithms are used to more closely correlate acoustical measurements to human hearing.
- Decibels can be used to express sound intensity and other phenomena as the logarithm of a ratio.

- Sound-pressure level (SPL) is a logarithmic value of sound pressure; depending on usage, slightly different equations are used to define it.
- An SPL measurement is made relative to a reference level, such as 20 μPa.
- To more closely conform to human perception, SPL measurements are often weighted with a particular response curve. An A-weighted curve is often used.
- Many sine-wave measurements are described using a set of specific relationships; these cannot be applied to complex waveforms.

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CHAPTER 3

Sound in the Free Field

Many practical acoustical problems are invariably associated with structures such as rooms and buildings, and vehicles such as automobiles and airplanes. These can generally be classified as problems in physics. These acoustical problems can be very complex in a physical sense; for example, a sound field might comprise thousands of reflected components, or temperature gradients might bend sound in an unpredictable manner. In contrast to practical problems, sound can also be considered in a free field, where its behavior is very predictable. In a free field, analysis is straightforward. This analysis is useful because it helps us understand the basic nature of sound waves. Then, these basic characteristics can be adapted to more complex physical problems.

The Free Field

Simply put, a free field is an open space. Sound in a free field travels in straight lines and is unimpeded. Unimpeded sound is not subject to the many influences that we will consider in later chapters. Sound in a free field is unreflected, unabsorbed, undeflected, undiffracted, unrefracted, undiffused, and not subjected to resonance effects. In most practical applications, these are all factors that could (and do) affect sound leaving a source and traveling to a listener. An approximate free field can exist in anechoic chambers, special rooms where all the interior surfaces are covered by sound absorbers. Also, approximate free-field conditions can exist close to a sound source. But generally, a free field is a theoretical invention, a free space that allows sound to travel without interference.

A free space must not be confused with cosmological space. Sound cannot travel in a vacuum; it requires a medium such as air. Here, free space means any space in which sound acts as though it is in a theoretical free field. In this unique environment, we can consider how sound diverges from a source, and how its intensity varies as a function of distance from the source.

Sound Divergence

Consider the point source of Fig. 3-1, radiating sound at a fixed power. The source can be considered as a point because its largest dimension is small (perhaps one-fifth or less) compared to the distances at which it is measured. For example, if the largest

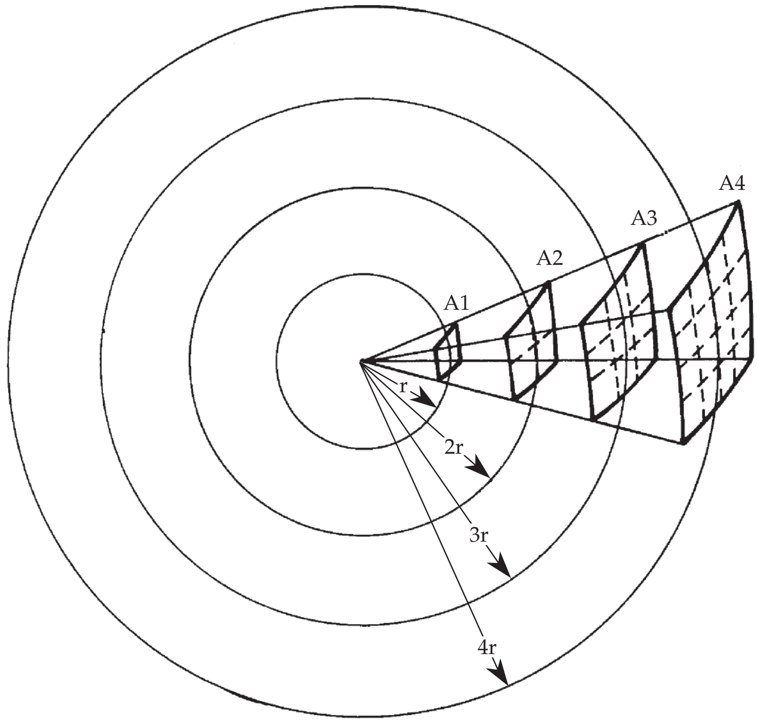

Figure 3-1 In the solid angle shown, as radius *r* is increased, the same sound energy is distributed over spherical surfaces of increasing area. The intensity of sound is inversely proportional to the square of the distance from the point source.

dimension of a source is 1 ft, it can be considered as a point source when measured at 5 ft or farther. Looked at in another way, the farther we are from a sound source, the more it behaves like a point source. In a free field, far from the influence of interfering objects, sound from a point source propagates spherically and uniformly in all directions. In addition, as described in the following text, the intensity of sound decreases as the distance from the source increases.

This sound is of uniform intensity (power per unit area) in all directions. The circles represent spheres having radii in simple multiples. All of the sound power passing through the small square area A1 at radius *r* also passes through the areas A2, A3, and A4 at radii 2*r*, 3*r*, and 4*r*, respectively. The same sound power flows out through A1, A2, A3, and A4, but an increment of the total sound power traveling in this single direction is spread over increasingly greater areas as the radius is increased. Thus, intensity decreases with distance. This decrease is due to the geometric spreading of the sound energy and is not a loss in the strict sense of the word.

Sound Intensity in the Free Field

Building on the preceding discussion (and referring again to Fig. 3-1), we observe that sound from a theoretical point source travels outward spherically. We also note that the area of a sphere is $4\pi r^2$. Therefore, the area of any small segment on the surface of the sphere also varies as the square of the radius. This means that the sound intensity (sound power per unit area) decreases as the square of the radius. This is an inverse square law. The intensity of a point-source sound in a free field is inversely proportional to the square of the distance from the source. In other words, intensity is proportional to $1/r^2$. More specifically:

$$I = \frac{W}{4\pi r^2} \tag{3-1}$$

where I = intensity of sound per unit area
W = power of source
r = distance from source (radius)

In this equation, since W and 4π are constants, we see that doubling the distance from r to $2r$ reduces the intensity I to $I/4$; this is because at twice the distance, the sound passes through an area that is four times the previous area. Likewise, tripling the distance reduces the intensity to $I/9$, and quadrupling the distance reduces intensity to $I/16$. Similarly, halving the distance from $2r$ to r increases the intensity to $4I$.

Sound Pressure in the Free Field

The intensity of sound (power per unit area) is a difficult parameter to measure. However, sound pressure is easily measured, for example, by using ordinary microphones. When using sound pressure, the free-field equation must be modified. Because sound intensity is proportional to the square of sound pressure, the inverse square law (for sound intensity) becomes the inverse distance law (for sound pressure). In other words, sound pressure is inversely proportional to distance r. In particular:

$$P = \frac{k}{r} \tag{3-2}$$

where P = sound pressure
k = a constant
r = distance from source (radius)

For every doubling of distance r from the sound source, sound pressure will be halved (not quartered). In Fig. 3-2, the sound-pressure level in decibels is plotted against distance. This illustrates the basis for the inverse distance law: when the distance from the source is doubled, the sound-pressure level decreases by 6 dB. This applies only to a free field. This law provides the basis for estimating the sound-pressure level in many practical circumstances.

Free-Field Sound Divergence

When the sound-pressure level L_1 at distance r_1 from a point source is known, the sound-pressure level L_2 at another distance r_2 can be calculated:

$$L_2 = L_1 - 20\log\frac{r_2}{r_1}\ \text{decibels} \tag{3-3}$$

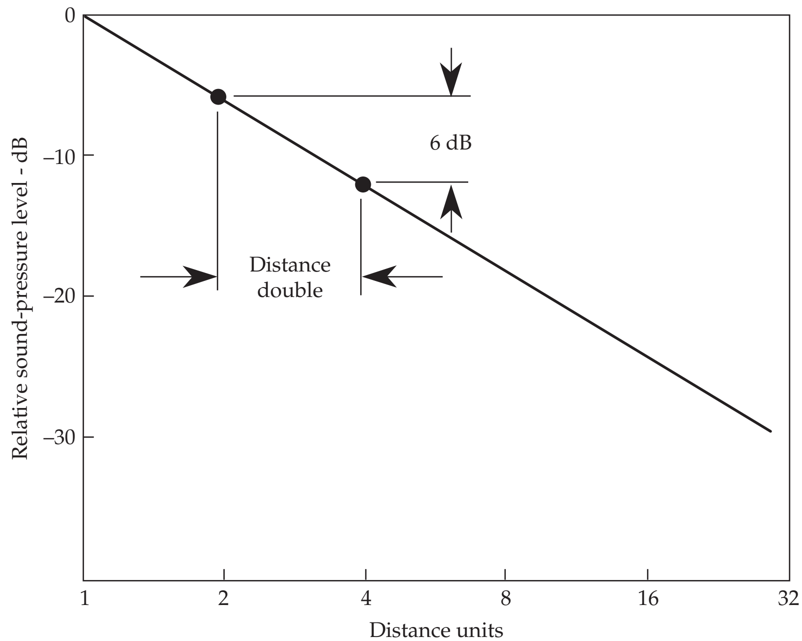

FIGURE 3-2 The inverse square law for sound intensity equates to the inverse distance law for sound pressure. This means that sound-pressure level is reduced by 6 dB for each doubling of the distance.

In other words, the difference in sound-pressure level between two points that are distances r_1 and r_2 from the source is:

$$L_2 - L_1 = 20 \log \frac{r_2}{r_1} \text{ decibels} \tag{3-4}$$

For example, if a sound-pressure level of 80 dB is measured at 10 ft, we can determine the sound-pressure level at 15 ft:

$$20 \log 10/15 = -3.5 \text{ dB, and the level is } 80 - 3.5 = 76.5 \text{ dB.}$$

Similarly, we can determine the sound-pressure level at 7 ft:

$$20 \log 10/7 = +3.1 \text{ dB, and the level is } 80 + 3.1 = 83.1 \text{ dB.}$$

This is only true for a free field in which sound diverges spherically, but this procedure may be helpful for rough estimates under other conditions.

If a microphone is 5 ft from a singer and a VU meter in the control room peaks at +6 dB, moving the microphone to 10 ft would bring the reading down approximately 6 dB. The 6-dB figure is approximate because these distance relationships hold true only for free-field conditions. In practice, the effect of sound energy reflected from walls would change the doubling of distance relationship to something less than 6 dB.

An awareness of these relationships helps estimate acoustical situations. For example, a doubling of the distance from 10 to 20 ft would, in free space, be accompanied by the same sound-pressure level decrease, 6 dB, as for a doubling from 100 to 200 ft. This accounts for the great carrying power of sound outdoors.

Even at a distance, not all sound sources behave as point sources. For example, traffic noise on a busy road can be better modeled as many point sources cumulatively acting as a line source. Sound spreads in a cylinder around the line. In this case, sound intensity is inversely proportional to the distance from the source. Doubling the distance from r to $2r$ reduces the intensity I to $I/2$. This is a decrease of 3 dB for every doubling of distance from the line source, or an increase of 3 dB for every halving of distance from the line source.

Sound Fields in Enclosed Spaces

Free fields exist in enclosed spaces in anechoic circumstances. However, in most rooms, in addition to the direct sound, reflections from the enclosing surfaces affect the way sound level decreases with distance. No longer does the inverse square law or the inverse distance law describe the entire sound field. In a free field, we can calculate sound level in terms of distance. In contrast, in a perfectly reverberant sound field, the sound level is equal everywhere in the sound field. In practice, rooms yield a combination of these two extremes, with both direct sound and reflected sound.

For example, assume that there is a loudspeaker in an enclosed space that is capable of producing a sound-pressure level of 100 dB at a distance of 4 ft. This is shown in the graph of Fig. 3-3. In the region very close to the loudspeaker, the sound field is in considerable disarray. The loudspeaker, at such close distances, cannot be considered a point source. This region is called the near field. In the near field, sound level decreases about 12 dB for every doubling of distance. This near-field region (not to be confused with "near-field" or "close-field" studio monitoring) is of limited practical interest.

After moving several loudspeaker dimensions away from the loudspeaker, significant measurements can be made in the far field. This far field consists of the free field and the reverberant field, and a transition region between them. Approximate free-field conditions exist near the loudspeaker where it operates as a point source. Direct sound is predominant, spherical divergence prevails in this limited space, and reflections from the surfaces are of negligible comparative level. In this region, sound-pressure level decreases 6 dB for every doubling of distance.

Moving away from the loudspeaker, sound reflected from the surfaces of the room affects the result. We define the critical distance as the location in a room where the direct and reflected sound levels are equal. The critical distance is useful as a rough single-figure description of the acoustics of the environment. Still farther away from the source, the reverberant sound field dominates. Its level remains constant, even at greater distances from the source. This level depends on the amount of absorption in the room. For example, in a very absorptive room, the reverberant level will be low.

Hemispherical Field and Propagation

True spherical divergence requires an absence of reflecting surfaces. Because this is difficult to achieve, it is sometimes approximated by placing a sound source such as a loudspeaker face-up, on a hard, reflective surface. This creates a hemispherical sound

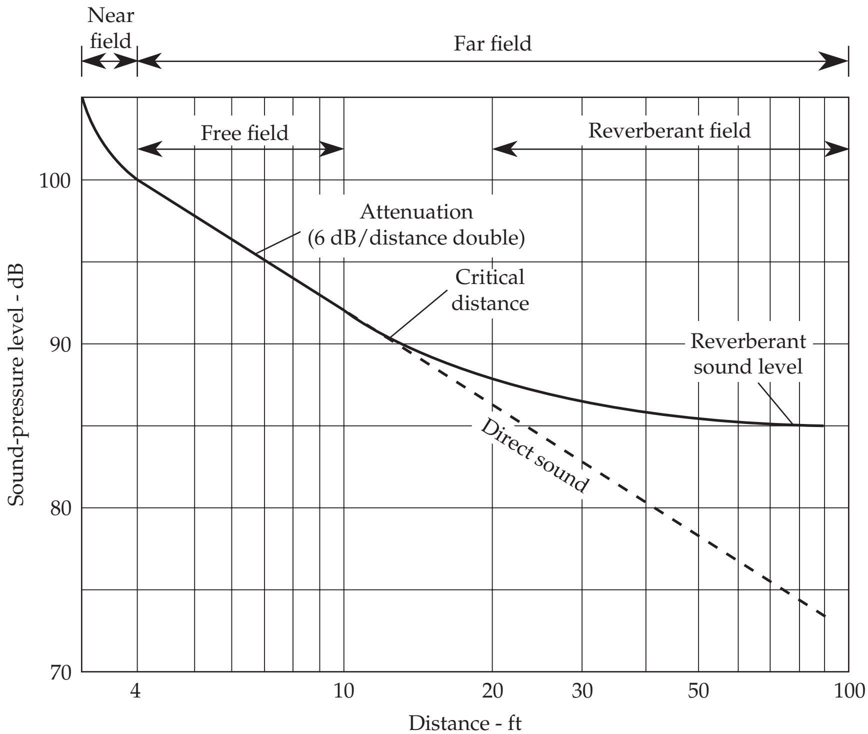

FIGURE 3-3 Even in an enclosed space, an approximate free field can exist close to the source, where the sound-pressure level is attenuated by 6 dB for every doubling of distance. By definition, the critical distance is that distance at which the direct sound-pressure level is equal to the reverberant sound-pressure level. Farther from the source, the sound-pressure level becomes constant and depends on the amount of absorption in the room.

field, radiating upward. This can be used, for example, to measure the response of the loudspeaker. In this case, the sound field radiates over a surface that is $2\pi r^2$ in area. Therefore, the intensity varies as $I = W/2\pi r^2$. As in a spherical sound field, the sound-pressure level in a hemispherical field attenuates by 6 dB for every doubling of distance. However, in a hemispherical sound field, the level begins 3 dB higher than in a spherical sound field.

How do we characterize hemispherical sound propagation over the earth's surface? Estimates made by the "6 dB per doubling distance" rule are only rough approximations. Reflections from the earth outdoors usually tend to make the sound level with distance something less than that indicated by the 6 dB approximation. The reflective efficiency of the earth's surface varies from place to place. Consider the sound level of a sound at 10 ft and again at 20 ft from the source. In practice, the difference between the two will probably be closer to 4 dB than 6 dB. For such outdoor measurements, the distance law must be taken at "4 or 5 dB per doubling distance." General environmental noise can also influence the measurement of specific sound sources.

Key Points

- The free field can be considered as an open space; sound travels in straight lines and is unimpeded.
- Sound from a theoretical point source travels outward spherically.
- In a free field, sound intensity (sound power per unit area) decreases as the square of the radius. This is an inverse square law.
- In a free field, when the distance from the source is doubled, the sound-pressure level decreases by 6 dB. This is an inverse distance law.
- In enclosed spaces, in addition to the direct sound, reflections from the enclosing surfaces affect the way sound level decreases with distance.
- The critical distance is the location in a room where the direct and reflected sound levels are equal.

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CHAPTER 4

The Perception of Sound

The study of the physical structure of the ear is a study in physiology. The study of human perception of sound is one of psychology and psychoacoustics. Psychoacoustics is an inclusive science embracing the physical structure of the ear, the sound pathways and their function, the human perception of sound, and the interrelationships of these. In many ways, psychoacoustics is the human basis for the entire field of audio engineering. Its role in the design of perceptual codecs such as MP3 and AAC is obvious, but psychoacoustics is also vital in architectural acoustics, telling us, for example, how a room's sound field is interpreted by the listener. Every good room design must carefully account for the perceptual acuity of the listeners in the room.

A stimulus wave striking the eardrum sets in motion mechanical movements that result in electrical discharges that are sent to the brain. The brain recognizes and interprets those discharges, creating the sensation that we call sound. That process is far from simple. Even after decades of vigorous research, our knowledge of human hearing is still incomplete.

Listening to a symphony orchestra, concentrate first on the violins, then the cellos, and the double basses. Now focus your attention on the clarinets, then the oboes, bassoons, and flutes. The ability of the ear–brain to separate individual parts of a complex sound wave is only one of the remarkable powers of the human auditory system. Moreover, a keen observer can listen to the sound of a violin and pick out the various overtones apart from the fundamental. The human ear is, by far, the most sophisticated and complex device in all of audio engineering.

Sensitivity of the Ear

The sensitive nature of our hearing can be underscored by a thought experiment. The bulky door of an anechoic chamber is opened, revealing extremely thick walls, and 3-ft wedges of glass fiber, pointing inward, lining all walls, ceiling, and what could be called the floor, except that you walk on an open steel grillwork.

You sit on a chair. This experiment takes time, and you lean back, patiently waiting. It is very eerie in here. The sea of sound and noises of life and activity in which we are normally immersed and of which we are ordinarily scarcely conscious are now conspicuous by their absence.

The silence presses down on you in the tomblike silence as the minutes pass. New sounds are discovered, sounds that come from within your body. The beating of your heart and the blood coursing through the vessels become audible. If your ears are keen, your patience is rewarded by a hissing sound between the thumps of the heart and the slushing of blood—it is the sound of air particles striking your eardrums.

The human ear cannot detect sounds softer than the random motion of air particles against the eardrum. This is the threshold of hearing. There would be no reason to have ears more sensitive because any lower-level sound would be drowned by the air-particle noise. This means that the ultimate sensitivity of our hearing just matches the softest sounds possible in an air medium.

Leaving the anechoic chamber, imagine being thrust into the loudest imaginable acoustical environments. At this extreme, our ears can respond to the roar of a cannon, the noise of a rocket blastoff, or a jet aircraft under full power. At loud levels, physiological features of the ear partially help protect the sensitive mechanism from damage. At the threshold of feeling (where a tingling sensation is felt), both prolonged exposure or sudden or intense noises can easily cause temporary shifts in the hearing response. At these loud levels, some permanent hearing damage is probably inevitable.

Ear Anatomy

The three principal parts of the human auditory system are the outer ear, middle ear, and inner ear, as shown in Fig. 4-1. The outer ear is composed of the pinna and the

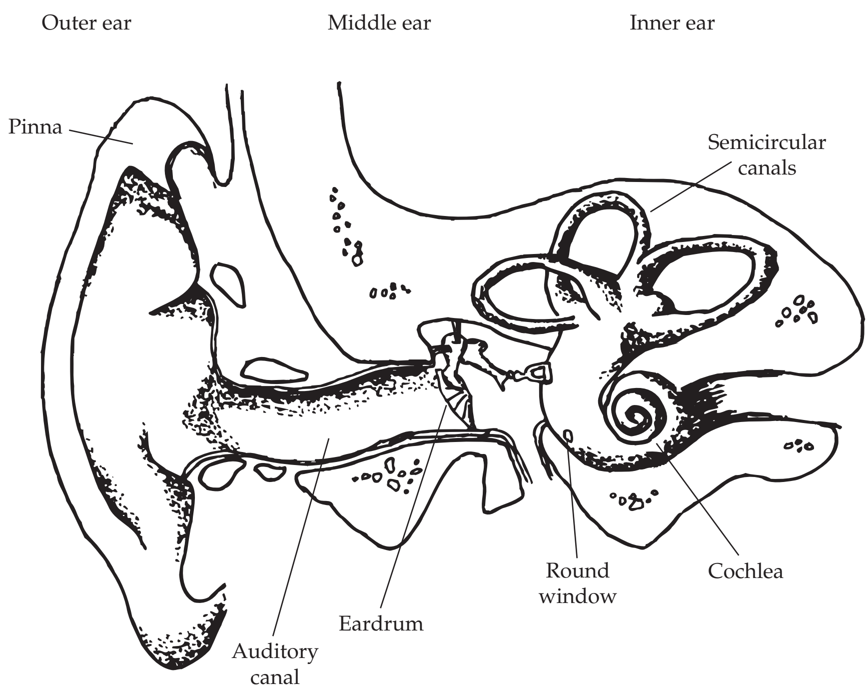

Figure 4-1 The human ear receives sound from the outer ear and it passes through the auditory canal. The middle ear connects the eardrum in air to the cochlea in fluid. The cochlea in the inner ear converts sound energy to electrical impulses that are transmitted to the brain.

auditory canal. The auditory canal is terminated by the tympanic membrane or the eardrum. The middle ear is an air-filled cavity spanned by three tiny bones collectively called the ossicles and comprising the malleus, incus, and stapes. These three bones are also sometimes called the hammer, anvil, and stirrup, respectively, because of their shapes. About the same size of a grain of rice, the stapes is the smallest bone in the body. The malleus is attached to the eardrum and the stapes is attached to the oval window of the inner ear. Together these three bones form a mechanical, lever-action connection between the air-actuated eardrum and the fluid-filled cochlea of the inner ear. The inner ear is terminated in the auditory nerve, which sends electrical impulses to the brain.

The Outer Ear—Pinna

Among its other functions, the external part of the outer ear, known as the pinna, operates as a sound-gathering device that focuses sound into the ear canal and provides amplification. For speech frequencies (2,000 to 4,000 Hz), the pinna increases sound pressure at the eardrum by about 5 dB. Cupping your hand behind the pinna increases the effective size of the pinna and thus the apparent loudness by an amount varying with frequency. Some animals are able to move their outer ears to point at the source, to help amplify the sound; humans lack this ability, but it is common (and often unconscious) to slightly move one's head to better hear a sound and to determine its location.

The pinna also performs a crucial function in imprinting directional information on all sounds entering the ear. This means that information concerning the direction of the source is superimposed on the sound content itself so that the resultant sound pressure on the eardrum enables the brain to interpret both the content of the sound and the direction from which it comes. The pinna provides a differentiation of sound from the front as compared to sound from the rear as well as allowing differentiation of sound from all around the listener. Both ears work together to provide additional spatial cues. At least in a free field, it is easy to close your eyes, and accurately point to sound sources.

A Demonstration of Directional Cues

A simple psychoacoustics demonstration can show how changes in sounds falling on the ear can yield subjective directional impressions. Listen with a headphone on one ear to an octave bandwidth of random noise with an adjustable notch filter in the signal path. Adjusting the filter to attenuate at 7.2 kHz will cause the noise to seem to come from a source on the level of the observer. With the notch adjusted to 8 kHz, the sound seems to come from above. With the notch at 6.3 kHz, the sound seems to come from below. This experiment demonstrates that the human hearing system extracts directional information from the shape of the sound spectra at the eardrum.

The Outer Ear—Auditory Canal

The auditory canal, commonly called the ear canal, also increases the loudness of the sounds traversing it. In Fig. 4-2, the ear canal, with an average diameter of about 0.3 in (0.7 cm) and length of about 0.9 in (2.3 cm), is idealized by straightening it and giving it a uniform diameter throughout its length. Acoustically, this is a reasonable approximation; the ear canal acts like a pipe-like duct, open at one end and closed at the inner end by the eardrum.

The acoustical similarity of this ear canal to an organ pipe is apparent. As with a pipe closed at one end, the resonance effect of the ear canal increases sound pressure at the eardrum relative to the opening of the canal. Moreover, the pressure increase varies

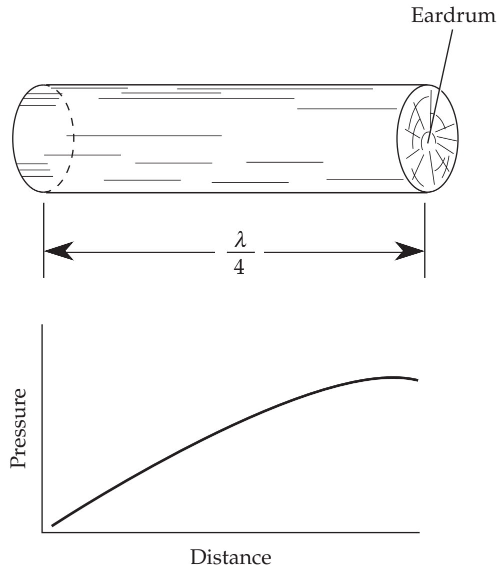

Figure 4-2 The auditory canal, closed at one end by the eardrum, acts as a quarter-wavelength pipe. Resonance provides acoustical amplification for midrange frequencies.

with frequency. The ear canal, closed at one end, will fundamentally support a quarter-wavelength resonant frequency; this can be used to model the ear canal's response. In practice, the resonant frequency of the canal differs from one person to another and depends on its length as well as other factors such as volume and curvature of the canal. Also, the canal walls are not solid, and the eardrum is concave. Very generally, the resonant frequency in an adult is about 3,000 Hz.

In addition, a plane wave striking the front of the head will be diffracted; this further increases sound pressure at the eardrum and broadens the overall boost in sound pressure. These effects combine to make the ear most sensitive to a range of midrange frequencies of approximately 2,000 to 4,000 Hz. Not surprisingly, this is the same frequency range occupied by speech. Unfortunately, this same boost in sound pressure also makes the ear relatively more susceptible to hearing loss at these important midrange frequencies.

The Middle Ear

Transmitting sound energy from a tenuous medium such as air into a dense medium like water poses challenges. Without some transfer mechanism, sound originating in air reflects on water like light on a mirror. For efficient energy transfer, the two impedances must be matched; in this case, the impedance ratio is about 4,000:1. It would be very unsatisfactory to drive the 1-Ω voice coil of a loudspeaker with an amplifier having an output impedance of 4,000 Ω; not much power would be transferred. Similarly, the ear must provide a way for energy in air to enter the ear's fluid interior.

The object is to transfer, as efficiently as possible, the energy represented by the vibratory motion of the eardrum diaphragm to the fluid of the inner ear. The twofold solution is suggested in Fig. 4-3. The three ossicles [malleus, incus, and stapes (hammer, anvil,

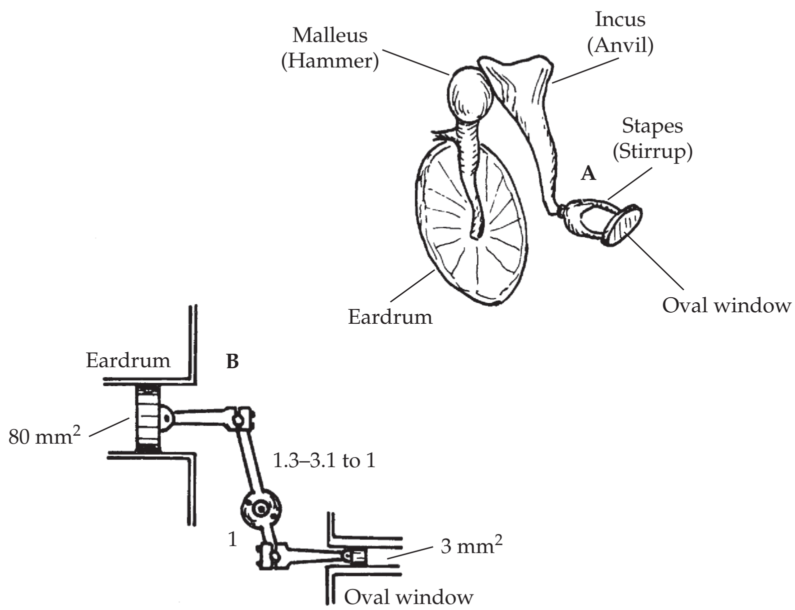

FIGURE 4-3 The middle ear provides impedance matching. (A) The ossicles comprising malleus, incus, and stapes (hammer, anvil, and stirrup) of the middle ear transmit mechanical vibrations of the eardrum to the oval window of the cochlea. (B) A mechanical analog of the impedance-matching function of the middle ear. The difference in area between the eardrum and the oval window, coupled with the step-down mechanical linkage, matches the motion of the air-actuated eardrum to the fluid-loaded oval window.

and stirrup), as shown in Fig. 4-3A] form a mechanical linkage between the eardrum and the oval window, which is in intimate contact with the fluid of the inner ear. The first of the three bones, the malleus, is fastened to the eardrum. The third, the stapes, is actually a part of the oval window. There is a lever action in this linkage with a ratio leverage ranging from 1.3:1 to 3.1:1. That is, the eardrum motion is reduced by this amount at the oval window of the inner ear.

This is only one part of the mechanical-impedance-matching device. The area of the eardrum is about 0.125 in^2 (80 mm^2), and the area of the oval window is only 0.0045 in^2 (3 mm^2). Hence, a given force on the eardrum is reduced by the ratio of 80/3 or about 27-fold.

In Fig. 4-3B, the action of the middle ear is likened to two pistons with area ratio of 27:1 connected by an articulated connecting rod having a lever arm ranging from 1.3:1 to 3.1:1, yielding a total mechanical force increase of between 35 and 80 times. The acoustical impedance ratio between air and water being on the order of 4,000:1, the pressure ratio required to match two media would be $4{,}000^{1/2}$ or about 63.2. This falls within the 35 to 80 range obtained from the mechanics of the middle ear illustrated in Fig. 4-3B. It is interesting that the ossicles in infants are fully formed and do not grow significantly larger over time; any change in size would only decrease the efficiency of the energy transfer.

The problem of matching sound in air to sound in the fluid of the inner ear is solved by the mechanics of the middle ear. Impedance matching plus ear-canal resonance is very efficient; diaphragm motion comparable to molecular dimensions gives a threshold perception.

A schematic of the ear is shown in Fig. 4-4. The conical eardrum at the inner end of the auditory canal forms one side of the air-filled middle ear. The middle ear is vented to the upper throat behind the nasal cavity by the Eustachian tube. The eardrum operates as an "acoustical suspension" system, acting against the compliance of the trapped air in the middle ear. The Eustachian tube is suitably small and constricted so as not to destroy this compliance. The round window separates the air-filled middle ear from the practically incompressible fluid of the inner ear.

The Eustachian tube fulfills a second function by equalizing the static air pressure of the middle ear with the outside atmospheric pressure so that the eardrum and the delicate membranes of the inner ear can function properly. Whenever we swallow, the Eustachian tubes open, equalizing the middle ear pressure. Changes in external air pressure (such as when an aircraft without a pressurized cabin undergoes rapid changes in altitude) may cause momentary deafness or pain until the middle ear pressure is equalized by swallowing. Finally, the Eustachian tube has a third emergency function of drainage if the middle ear becomes infected.

The Inner Ear

The acoustical amplifiers and mechanical impedance matching features of the middle ear, discussed so far, are relatively well understood. The intricate operation of the inner ear, containing the cochlea, is not as explicit.

The cochlea is the sound-analyzing organ. It is in close proximity to the three mutually perpendicular, semicircular canals of the vestibular mechanism, the balancing organ

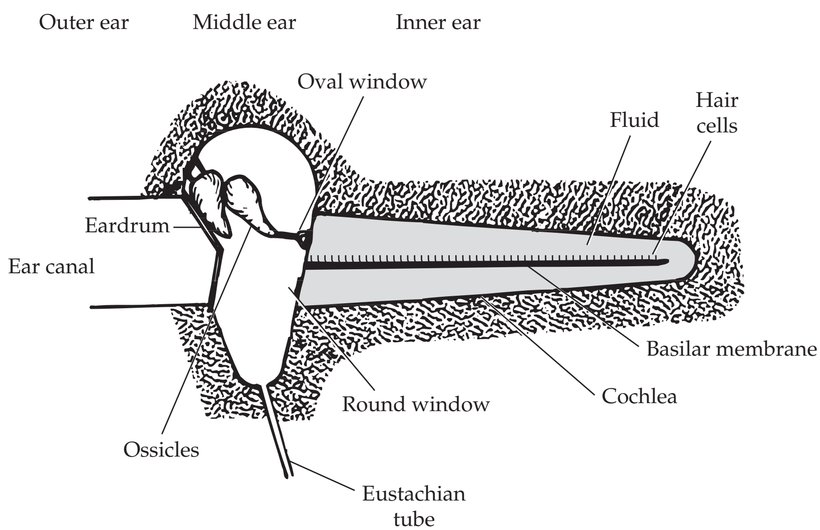

Figure 4-4 Idealized sketch of the human ear showing the uncoiled fluid-filled cochlea. Sound entering the ear canal causes the eardrum to vibrate. This vibration is transmitted to the cochlea through the mechanical linkage of the middle ear. The sound is analyzed through standing waves set up on the basilar membrane.

(see Fig. 4-1). The same fluid permeates both organs but their functions are independent. The cochlea is about the size of a pea and is encased in solid bone. It is coiled up like a cockleshell from which it gets its name. For the purposes of illustration, as shown in Fig. 4-4, this 2-3/4 turn coil has been stretched out its full length, about 1 in. The fluid-filled inner ear is divided lengthwise by two membranes: Reissner's membrane and the basilar membrane. Of immediate interest are the basilar membrane and its response to sound vibrations in the fluid.

Vibration of the eardrum activates the ossicles. The motion of the stapes, attached to the oval window, causes the fluid of the inner ear to vibrate. An inward movement of the oval window results in a flow of fluid around the basilar membrane, causing an outward movement of the membrane of the round window; the round window thus provides pressure release. Sound actuating the oval window results in standing waves being set up along the basilar membrane. The position of the amplitude peak of the standing wave on the basilar membrane changes as the frequency of the exciting sound is changed.

Low-frequency sound results in maximum amplitude nearer the distant end of the basilar membrane; high-frequency sound produces peaks nearer the oval window; mid frequencies are in between. For a complex signal such as music or speech, many momentary peaks are produced, constantly shifting in amplitude and position along the basilar membrane. These resonant peaks on the basilar membrane were originally thought to be so broad as to be unable to explain the sharpness of frequency discrimination displayed by the human ear. Subsequent research showed that at low sound intensities, the basilar membrane tuning curves are very sharp, broadening only for intense sound. It appears that the sharpness of the basilar membrane's mechanical tuning curves is comparable to the scale of single auditory nerve fibers, which innervate it.

Stereocilia

Waves set up on the basilar membrane in the fluid-filled duct of the inner ear stimulate hairlike nerve terminals that convey signals to the brain in the form of neuron discharges. There is one row of inner hair cells and three to five rows of outer hair cells. Each hair cell contains a bundle of tiny hairs called stereocilia. As sound causes the cochlear fluid and the basilar membrane to move, the stereocilia vibrate according to the vibrations around them. Stereocilia at various locations along the basilar membrane are stimulated by characteristic frequencies corresponding to that location. The inner hair cells can be thought of as microphones, transducers that convert mechanical vibration to electrical signals that initiate neural discharges to the auditory nerve and the brain. The outer hair cells provide additional gain or attenuation to more sharply tune the output of the inner hair cells and to make the hearing system more sensitive.

Bending of the stereocilia triggers the nerve impulses that are carried by the auditory nerve to the brain. A single nerve fiber is either firing or not firing, in binary fashion. When a nerve fires, it causes an adjoining one to fire, and so on. Physiologists liken the process to a burning gunpowder fuse. The rate of travel bears no relationship to how the fuse was lighted. Presumably, the loudness of the sound is related to the number of nerve fibers excited and the repetition rates of such excitation. When all the nerve fibers are excited, this is the maximum loudness that can be perceived. The threshold sensitivity would be represented by a single fiber firing. The sensitivity of the system is remarkable; at the threshold of hearing, the faintest sound we can hear, tiny filaments associated with the stereocilia move about 0.04 nm.

A well-accepted theory of how the inner ear and the brain function has not yet been formulated. The presentation here is a highly simplified explanation of a very complex mechanism. Some of the theories discussed here are not universally accepted.

Loudness versus Frequency

The seminal work on loudness was done by Fletcher and Munson at Bell Laboratories and reported in 1933. Since that time, refinements have been added by others. The family of equal-loudness contours shown in Fig. 4-5, more recent work by Robinson and Dadson, has been adopted as an international standard (ISO 226).

Each equal-loudness contour is identified by its value at the reference frequency of 1 kHz, and the term, loudness level in phons, is thus defined. For example, the equal-loudness contour passing through 40-dB sound-pressure level at 1 kHz is called the 40-phon contour. Similarly, the 100-phon contour passes through 100 dB at 1 kHz. The line of each contour shows how sound-pressure level must be varied at different frequencies to sound equally loud as the 1-kHz reference loudness of 40 phons. Each contour was obtained experimentally by asking subjects to state when levels at different frequencies sounded equally loud as the reference level at 1 kHz, at each of the 13 different reference levels. These data are for pure tones and do not apply directly to music and

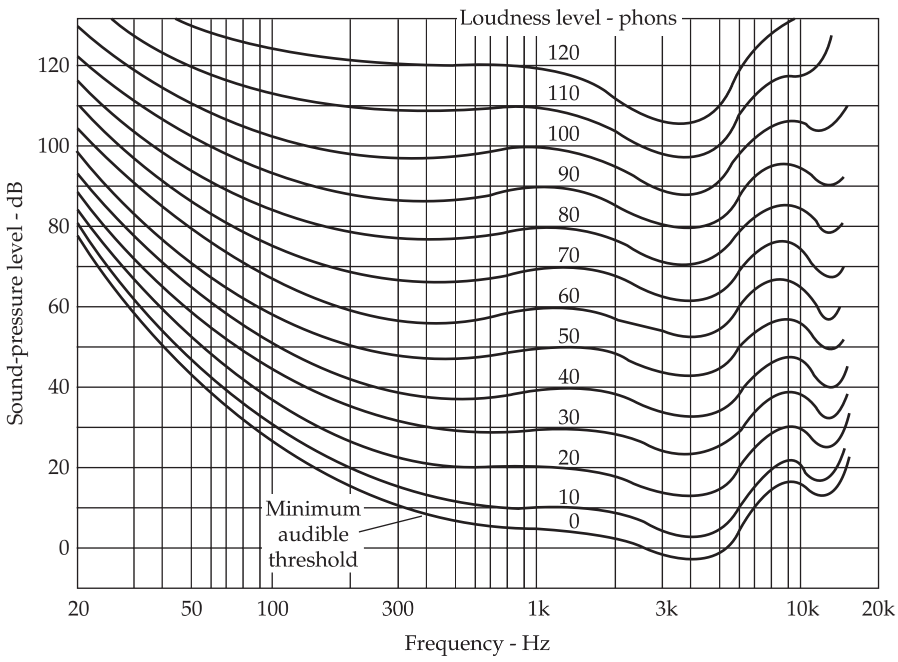

Figure 4-5 Equal-loudness contours of the human ear for pure tones. These contours reveal the relative lack of sensitivity of the ear to bass tones, especially at lower sound levels. Inverting these curves gives the frequency response of the ear in terms of loudness level. These data are taken for a sound source directly in front of the listener, pure tones, binaural listening, and subjects aged 18 to 25. (*Robinson and Dadson.*)

other audio signals. Loudness is a subjective term; sound-pressure level is strictly a physical term. Loudness level is also a physical term that is useful in estimating the loudness of a sound (in units of sones) from sound-level measurements. However, a loudness level is not the same as an SPL reading. The shapes of the equal-loudness contours contain subjective information because they were obtained by a subjective comparison of the loudness of a tone to its loudness at 1 kHz.

The curves of Fig. 4-5 reveal that perceived loudness varies greatly with frequency and sound-pressure level. For example, a sound-pressure level of 30 dB yields a loudness level of 30 phons at 1 kHz, but it requires a sound-pressure level of an additional 58 dB to sound equally loud at 20 Hz, as shown in Fig. 4-6. The curves tend to flatten at greater loudness, showing that the ear's response is more uniform at high levels. For example, the 90-phon curve rises only 32 dB between 1 kHz and 20 Hz. Note that inverting the curves of Fig. 4-6 gives the frequency response of the ear in terms of loudness level. We see that at low volume levels the ear is less sensitive to bass notes than mid-band notes. This bass deficiency of the ear means that the frequency fidelity of reproduced music depends on the volume-control setting. Listening to background music at low levels would require a different playback frequency response than listening at higher levels. There are also deviations in the ear's high-frequency response, but these are relatively less noticeable.

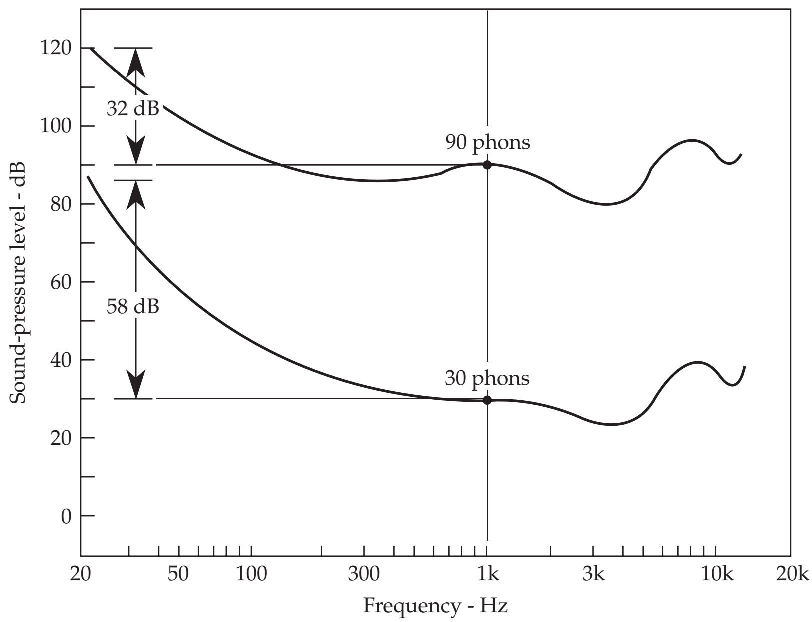

FIGURE 4-6 A comparison of the ear's response at 20 Hz, to the response at 1 kHz. At a loudness level of 30 phons, the sound-pressure level of a 20-Hz tone must be 58 dB higher than that at 1 kHz to have the same loudness. At a 90-phon loudness level, an increase of only 32 dB is required. The ear's response is somewhat flatter at high loudness levels. Loudness level is only an intermediate step to true subjective loudness.

Loudness Control

Suppose you want to play back a recording at a quiet level (assume 50 phons). If the music was originally played and recorded at a higher level (assume 80 phons), you would need to increase bass as well as treble for proper balance. The loudness control found on some audio devices is intended to compensate for the change in frequency response of the ear for different loudness levels by boosting low and high frequencies. But the EQ curve corresponding to a given setting of the loudness control might apply only to a specific loudness level of reproduced sound. This is an incomplete solution to the problem. Consider the many things that affect the volume-control setting: Loudspeakers vary in acoustical output for a given input power. The gain of power amplifiers differs. Listening-room conditions vary from dead to highly reverberant, affecting sound-field levels. For a loudness control to function properly, the system would have to be calibrated and the loudness control would have to adaptively vary the frequency response relative to the level at the listener. This demonstrates how the ear's nonlinear level response introduces complexity to the tasks of recording and playback.

Area of Audibility

The results of tests administered to trained listeners are shown in Fig. 4-7. The listeners faced the sound source and judged whether a tone at a given frequency was barely audible (contour A) or elicited a sensation or feeling in the ears that may begin to be painful (contour B). These two contours therefore represent the extremes of our perception of loudness.

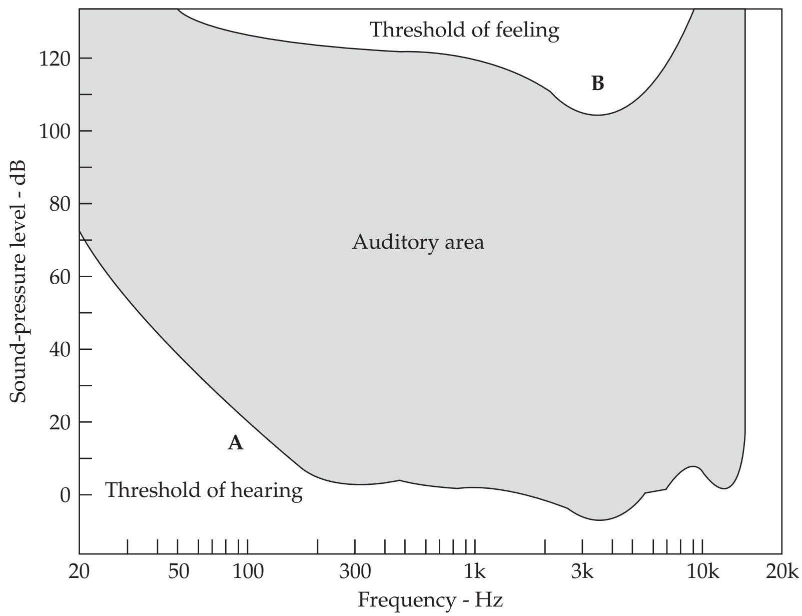

FIGURE 4-7 The auditory area of the human ear is bounded by two threshold contours. (A) The threshold of hearing delineates the lowest level sounds the ear can detect. (B) The threshold of feeling defines the upper extreme. All of our auditory experiences occur within this sound level and frequency area.

Contour A, the threshold of hearing, represents the level at each frequency where sounds are just barely audible. The contour also shows that human ears are most sensitive around 3 kHz. Another way to state this is that around 3 kHz a lower-level sound elicits a greater threshold response than higher or lower frequencies. At this most sensitive region, a sound-pressure level defined as 0 dB can just barely be heard by a person of average hearing acuity. The reference level of pressure of 20 μPa was selected to establish this 0-dB level.

Contour B, the threshold of feeling, represents the level at each frequency where a tickling sensation is felt in the ears. At 3 kHz, this occurs at a sound-pressure level of about 110 dB. A further increase in level results in an increase in feeling until a sensation of pain is produced. The threshold tickling is a warning that the sound is becoming dangerously loud and that hearing damage is either imminent or has already taken place.

Between the threshold of hearing and the threshold of feeling is the area of audibility. This area has two dimensions: the vertical range of sound-pressure level and the horizontal range of frequencies that the ear can perceive. The sounds that humans experience are of such a level and frequency as to fall within this auditory area.

The area of audibility for humans is quite different from that of many animals. Bats specialize in sonar cries that are far above the upper frequency limit of our ears. The hearing of dogs extends higher than humans, hence the usefulness of ultrasonic dog whistles. Sound in the infrasonic and ultrasonic regions, as related to the hearing of humans, is no less true sound in the physical sense, but it does not result in human perception.

Loudness versus Sound-Pressure Level

The phon is a unit of physical loudness level that is referenced to a sound-pressure level at 1 kHz. This is useful but it tells us little about human reaction to the loudness of sound. Some sort of subjective unit of loudness is needed. Many experiments conducted with hundreds of subjects and many types of sound have yielded a consensus that for a 10-dB increase in sound-pressure level, the average person reports that loudness is doubled. Likewise, for a 10-dB decrease in sound level, subjective loudness is cut in half. The sone is a unit of subjective loudness. One sone is defined as the loudness (not loudness level) experienced by a person listening to a 1-kHz tone of 40-phon loudness level. A sound of 2 sones is twice as loud as 1 sone, and 0.5 sone is half as loud.

Figure 4-8 shows a graph for translating sound-pressure levels to loudness in sones. One point on the graph is the definition of the sone, the loudness experienced by a person hearing a 1-kHz tone at 40-dB sound-pressure level, or 40 phons. A loudness of 2 sones is then 10 dB higher; a loudness of 0.5 sone is 10 dB lower. A straight line can be drawn through these three points, which can then be extrapolated for sounds of higher and lower loudness. This graph only applies to 1-kHz tones.

The idea of subjective loudness has great practical value. For example, a consultant might be required by a court to give an opinion on the loudness of an industrial noise that bothers neighbors. The consultant can make a 1/3-octave analysis of the noise, translate the sound-pressure levels of each band to sones (using graphs such as Fig. 4-8), add together the sones of each band, and arrive at an estimate of the loudness of the noise. It is convenient to be able to add component sones, and less convenient to combine sound-pressure levels in decibels.

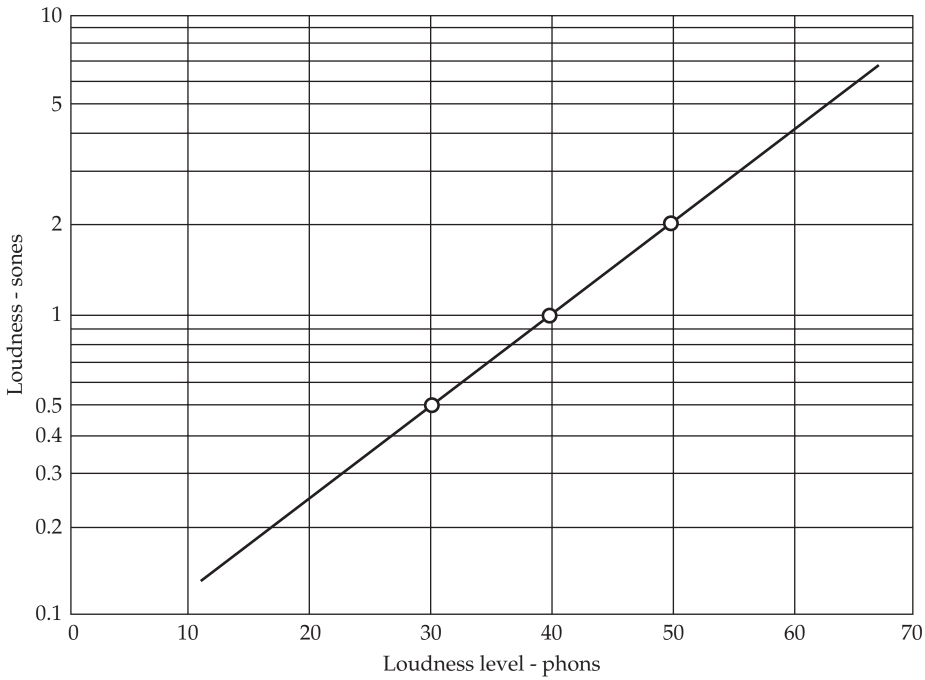

Figure 4-8 The graphical relationship between subjective loudness in sones and physical loudness level in phons. This graph only applies to 1-kHz tones.

Table 4-1 shows the relationship between loudness level in phons and the subjective loudness in sones. Although most audio engineers have little use for phons or sones, it is good to realize that a true subjective unit of loudness (sone) is related to loudness level (phon), which in turn is related by definition to what we measure with a sound-level meter. There are empirical methods for calculating the loudness of sound as perceived by humans from purely physical measurements of sound spectra, such as those measured with a sound-level meter and an octave or 1/3-octave filter.

Loudness Level (Phons)	Subjective Loudness (Sones)	Typical Examples
100	64	Heavy truck passing
80	16	Talking loudly
60	4	Talking softly
40	1	Quiet room
20	0.25	Very quiet studio

Table 4-1 Loudness Level in Phons versus Loudness in Sones

Loudness and Bandwidth

Thus far, we have discussed loudness in terms of single-frequency tones, but tones do not give all the information we need to relate subjective loudness to meter readings. The noise of a jet aircraft taking off, for example, sounds much louder than a tone of the same sound-pressure level. The bandwidth of the noise affects the loudness of the sound, at least within certain limits.

Figure 4-9A represents three sounds (noise) having the same sound-pressure level of 60 dB. Their bandwidths are 100, 160, and 200 Hz, but their heights (representing sound intensity per hertz) vary so that their areas are equal. In other words, the three sounds have equal intensities. (Sound intensity has a specific meaning in acoustics and is not to be equated to sound pressure. Sound intensity is proportional to the square of sound pressure for a plane progressive wave.) However, the three sounds of Fig. 4-9A do not have the same loudness. Figure 4-9B shows how a bandwidth of noise having a constant 60-dB sound-pressure level and centered on 1 kHz is related to loudness as experimentally determined. The 100-Hz bandwidth noise has a loudness level of 60 phons and a loudness of 4 sones. The 160-Hz bandwidth noise has the same loudness. But something unexpected happens as the bandwidth is increased beyond 160 Hz. From 160 Hz upward, increasing bandwidth increases loudness. For example, the loudness of the 200-Hz bandwidth noise is louder. Why the sharp change at 160 Hz?

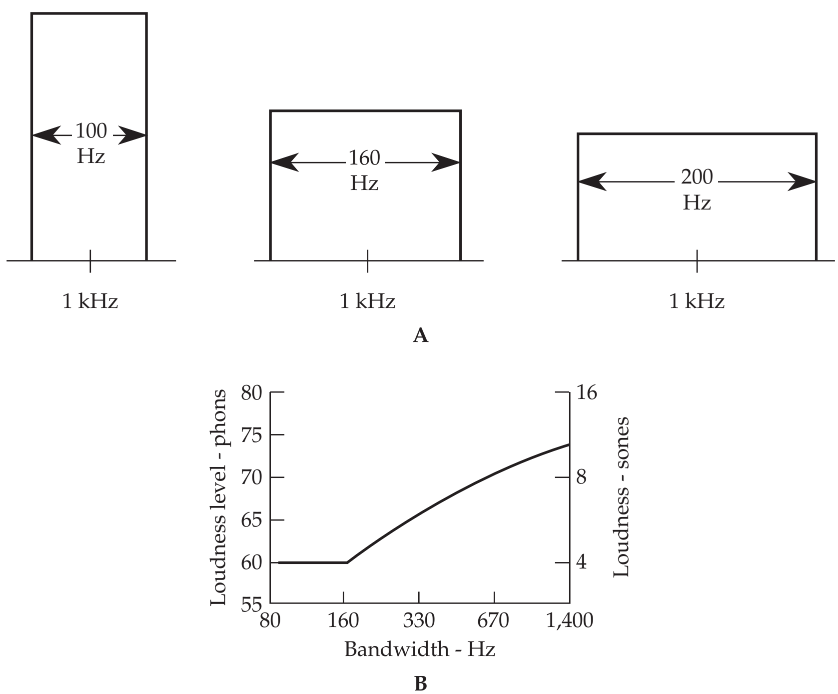

FIGURE 4-9 Bandwidth affects the loudness of sounds. (A) Three noise signals of different bandwidths, but all having the same sound-pressure level of 60 dB. (B) The subjective loudness of the 100- and 160-Hz noise is the same, but the 200-Hz band sounds louder because it exceeds the 160-Hz critical bandwidth of the ear at 1 kHz.

The reason is that 160 Hz is the width of the ear's critical band at 1 kHz. This is demonstrated by the fact that if a 1-kHz tone is presented to a listener along with random noise, only the noise in a band 160 Hz wide is effective in masking the tone. In other words, the ear acts like an analyzer composed of a set of band-pass filters stretching throughout the audible spectrum. This filter set is not like that found in an electronics laboratory. The common 1/3-octave filter set may have 28 adjacent filters fixed and overlapping at the –3 dB points. The ear's critical band filters are continuous; no matter what frequency we choose, there is a critical band centered on that frequency.

Research shows how the width of the critical bands varies with frequency. This bandwidth function is shown in Fig. 4-10. In particular, critical bands become much wider at higher frequencies. There are other methods for measuring critical bandwidth; they provide different estimates particularly below 500 Hz. For example, the equivalent rectangular bandwidth (ERB) (that applies to young listeners at moderate sound levels) is based on mathematical methods. The ERB offers the convenience of being calculable from the equation:

$$\text{ERB} = 6.23f^2 + 93.3f + 28.52 \text{ Hz} \tag{4-1}$$

where f = frequency, kHz

One-third-octave filter sets have been justified in certain measurements because the filter bandwidths approach those of the critical bands of the ear. For comparison, a plot of 1/3-octave bandwidths is included in Fig. 4-10. One-third-octave bands are 23.2% of

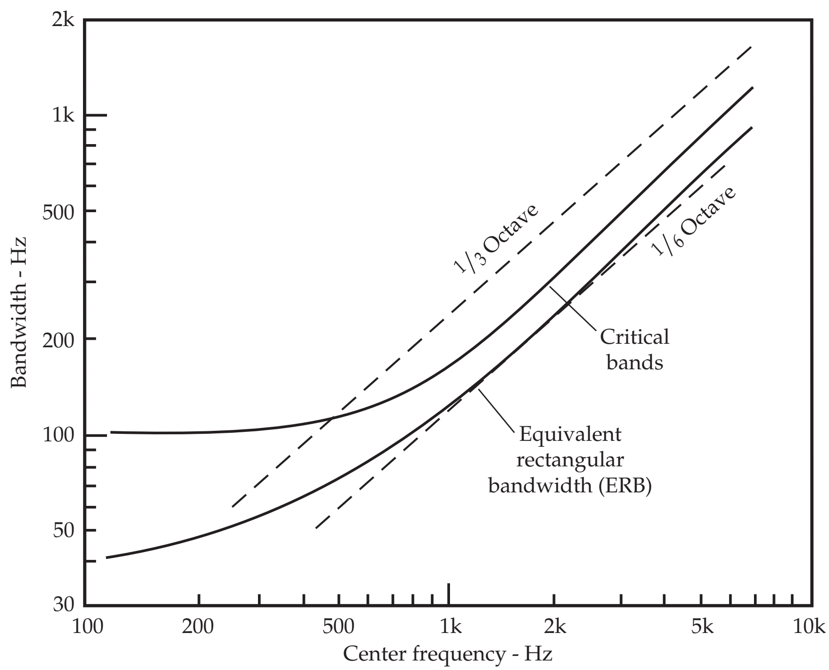

FIGURE 4-10 A comparison of bandwidths of 1/3- and 1/6-octave bands, critical bands of the ear, and equivalent rectangular bandwidth (ERB).

the center frequency. The critical-band function is about 17% of the center frequency. The ERB function is about 12%; this is close to that of 1/6-octave bands (11.6%). This suggests that 1/6-octave filter sets may be at least as relevant as 1/3-octave filters.

Critical bands are important in many audio disciplines. For example, perceptual codecs such as MP3 and AAC are based on the principle of masking. A tone (a music signal) will mask quantization noise that lies within a critical band centered at the tone's frequency. However, if the quantization noise extends outside the critical band, it will not be masked by the tone. Codecs thus attempt to contain noise within the critical-bandwidth masking curves created by loud tones. One critical band is defined as having a width of 1 bark (named after German physicist Heinrich Barkhausen).

Loudness of Impulses

The examples discussed so far have been concerned with steady-state tones and noise. How does the ear respond to transients of short duration? This is important because music and speech are replete with transients. To focus attention on this aspect of speech and music, play some audio tracks backward. The initial transients now appear at the ends of syllables and musical notes and stand out prominently.

As a 1-sec tone burst, a 1-kHz tone sounds like 1 kHz. But an extremely short burst of the same tone sounds like a click. The duration of such a burst also influences the perceived loudness. Short bursts do not sound as loud as longer ones. Figure 4-11 shows how much the level of shorter pulses must be increased to have the same loudness as a long pulse or steady tone. For example, a 3-msec pulse must have a level about 15 dB higher to sound as loud as a 0.5-sec (500-msec) pulse. Tones and random noise follow roughly the same relationship in loudness versus pulse length.

The region less than 100 msec in Fig. 4-11 is significant. Only when the tones or noise bursts are shorter than this amount must the sound-pressure level be increased to

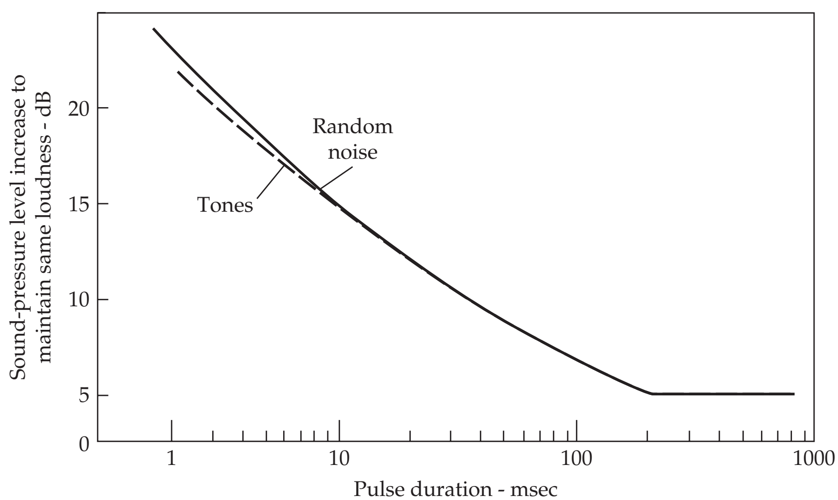

FIGURE 4-11 Short pulses of tones or noise are less audible than longer pulses. The discontinuity of the 100- to 200-msec region is related to the integrating time of the ear.

produce a loudness equal to that of long pulses or steady tones or noise. This 100 msec appears to be the maximum integrating time or the time constant of the human ear. In particular, events occurring within 35 msec, such as reflections from walls, are integrated by the ear with regard to level. This shows that the ear responds to sound energy averaged over time.

Figure 4-11 shows that our ears are less sensitive to short transients such as peaks in sound level. This has a direct bearing on understanding speech. The consonants of speech determine the meaning of many words. For instance, the only differences between bat, bad, back, bass, ban, and bath are the consonants at the end. The words led, red, shed, bed, fed, and wed have the all-important consonants at the beginning. No matter where they occur, consonants are transients having durations on the order of 5 to 15 msec. Figure 4-11 shows that transients this short must be louder to be comparable to longer sounds. In the above words, each consonant is not only much shorter than the rest of the word; it is also at a lower level. Thus we need good listening conditions to distinguish between such sets of words. Too much background noise or too much reverberation can seriously impair the intelligibility of speech because they can mask important, lower-level consonants.

Audibility of Loudness Changes

As we have seen, the ear is sensitive to a wide dynamic range of sounds from the softest to loudest. Within that range, the ear is relatively sensitive to small changes in loudness. For example, steps of 5 dB are definitely audible, whereas steps of 0.5 dB may be inaudible, depending on circumstances; detecting differences in intensity varies somewhat with frequency and also with sound level. For example, at 1 kHz, for very low levels, a 3-dB change is the least change detectable by the ear, but at high levels the ear can detect a 0.25-dB change. (The former statistic is why many recording engineers mix at loud playback volumes.) As another example, a very low-level, 35-Hz tone requires a 9-dB level change to be detectable. For the important mid-frequency range and for commonly used levels, the minimum detectable change in level that the ear can detect is about 2 dB. In most cases, at least in acoustical design, making level changes in increments less than these is usually unnecessary.

Pitch versus Frequency

Pitch is a subjective term. It is chiefly a function of frequency but it is not linearly related to it. Because pitch is somewhat different from frequency, it requires another subjective unit, the mel. Frequency is a physical term measured in hertz. Although a soft 1-kHz signal is still 1 kHz if you increase its level, the pitch of a sound may depend on sound-pressure level. A reference pitch of 1,000 mels is defined as the pitch of a 1-kHz tone with a sound-pressure level of 60 dB. The relationship between pitch and frequency, determined by experiments with juries of listeners, is shown in Fig. 4-12. On the experimental curve, 1,000 mels coincides with 1 kHz; thus the sound-pressure level for this curve is 60 dB. The shape of the curve of Fig. 4-12 is similar to a plot of position along the basilar membrane as a function of frequency. This suggests that pitch is related to action on this membrane.

Researchers tell us that the human ear can detect about 280 discernible steps in intensity and some 1,400 discernible steps in pitch. As changes in intensity and pitch are

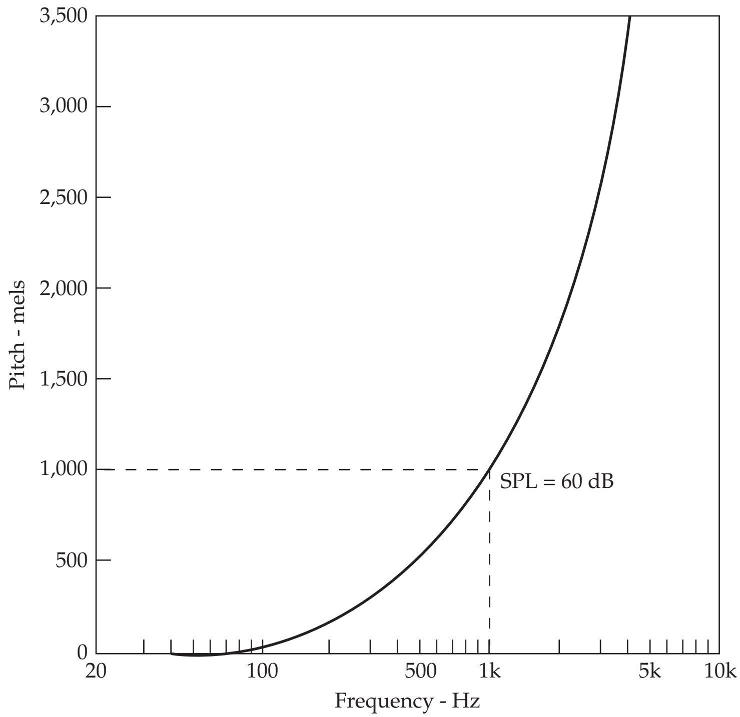

FIGURE 4-12 Pitch (in mels, a subjective unit) is related to frequency (in hertz, a physical unit) according to this curve obtained by juries of listeners. (*Stevens and Volkman.*)

vital to audio communication, it would be interesting to know how many combinations are possible. Offhand, it might seem that there would be $280 \times 1{,}400 = 392{,}000$ combinations detectable by the ear. This is overly optimistic because the tests were conducted by comparing two simple, single-frequency sounds in rapid succession and bears little resemblance to the complexities of commonly heard sounds. Other experiments suggest that the ear can detect only about 7 degrees of loudness and 7 degrees of pitch or only 49 pitch-loudness combinations. Perhaps not coincidentally, this is not too far from the number of phonemes (the smallest unit in a language that distinguishes one utterance from another) which can be detected in a language.

An Experiment in Pitch

The level of sound affects the perception of pitch. For low frequencies, the pitch goes down as sound level is increased. At high frequencies, the reverse takes place; the pitch increases with sound level.

The following is an experiment suggested by Fletcher. Two audio oscillators are required, as well as a frequency counter. One oscillator is applied to the input of one channel of a playback system, the other oscillator to the other channel. The frequency of one oscillator is adjusted to 168 Hz and the other to 318 Hz. At low level these two tones are quite discordant. If the level of the 168-Hz and 318-Hz tones is increased until the (perceived) pitches decrease to 150 Hz and 300 Hz, this will yield an octave relationship which gives a pleasant sound. This illustrates the decrease of pitch at lower frequencies. A similar test would show that the pitch of higher frequency tones increases with sound level.

The Missing Fundamental

The auditory system can sometimes play tricks on our perception of sound. If tones such as 1,000, 1,200, and 1,400 Hz are reproduced together, a pitch of 200 Hz is heard. This can be interpreted as the fundamental with 1,000 Hz as the fifth harmonic; 1,200 Hz as the sixth harmonic; and so on. The auditory system recognizes that the upper tones are harmonics of the 200-Hz tone and perceptually supplies the missing fundamental that would have generated them.

Timbre versus Spectrum

Timbre describes our perception of the tonal quality of complex sounds. The term is applied chiefly to the sound of musical instruments. A flute and oboe sound different even though they are both playing the same pitch. The tone of each instrument has its own timbre. Timbre is determined by the number and relative strengths of the instrument's partials.

Timbre is a subjective term. The analogous physical term is spectrum. A musical instrument produces a fundamental and a set of partials (or harmonics) that can be analyzed with a wave analyzer. Suppose, for example, the fundamental frequency is 200 Hz, the second harmonic frequency is 400 Hz, the third harmonic is 600 Hz, and so on. The subjective pitch that the ear associates with our measured 200 Hz varies slightly with the level of the sound. The ear also has its own subjective interpretation of the harmonics. Thus, in an intricate way, the ear's perception of the overall timbre of an instrument's note might be considerably different from the measured spectrum. In other words, timbre (a subjective description) and spectrum (an objective measurement) are not the same.

Localization of Sound Sources

The perception of the location of a sound source begins at the external ear, with the pinna. Sound reflected from the ridges, convolutions, and surfaces of the pinna combines with the unreflected direct sound at the entrance to the auditory canal. This combination, now encoded with directional information by the pinna, passes down the auditory canal to the eardrum and then to the middle and inner ear, and finally to the brain for interpretation.

This directional encoding process of the sound signal is shown in Fig. 4-13. The sound wavefront can be considered as a multiplicity of sound rays coming from a specific source at a specific horizontal and vertical angle. As these rays strike the pinna, they are reflected from the surfaces, some of the reflections going toward the entrance to the auditory canal. At that point these reflected components combine with the unreflected component.

For a sound coming directly from the front of the listener (azimuth and vertical angle = 0°), the "frequency response" of the combination sound at the opening of the ear canal is shown in Fig. 4-14. A curve of this type is called a transfer function; it represents a vector combination involving phase angles.

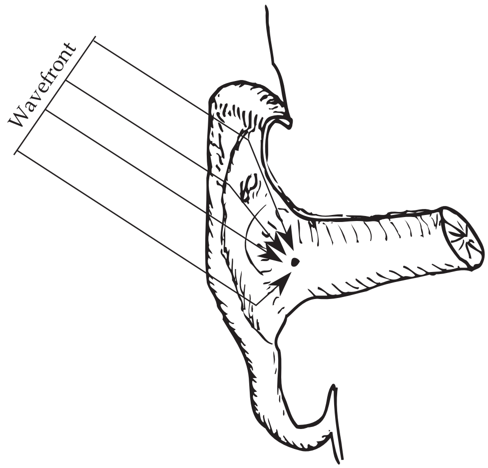

Figure 4-13 A wavefront of a sound can be considered as numerous rays perpendicular to that wavefront. Such rays, striking a pinna, are reflected from the various ridges and convolutions. Those reflections directed to the opening of the ear canal combine vectorially (according to relative amplitudes and phases). In this way the pinna encodes all sound falling on the ear with directional information, which the brain decodes as a directional perception.

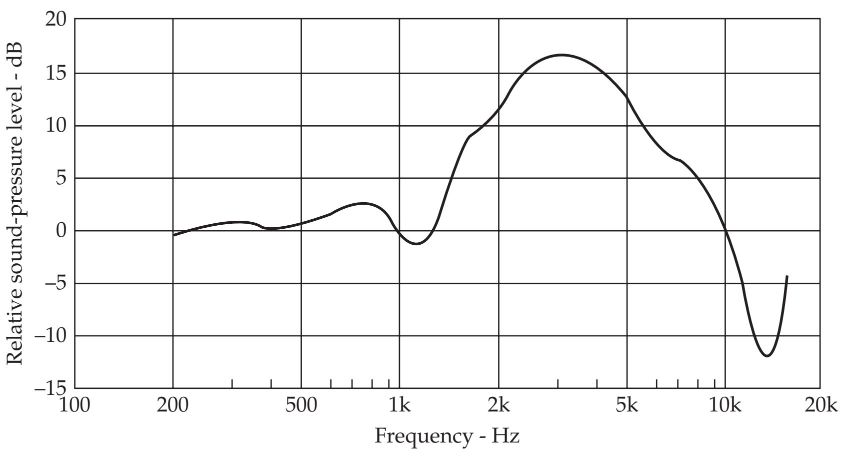

Figure 4-14 A measured example of the sound pressure (transfer function) at the opening of the ear canal corresponding to sound arriving from a point directly in front of the listener. The shapes of such transfer functions vary with the horizontal and vertical angles at which the sound arrives at the pinna. (*Mehrgardt and Mellert.*)

For the sound at the entrance of the ear canal to reach the eardrum, the auditory canal must be traversed. As the transfer function at the entrance to the ear canal (see Fig. 4-14) and that of the ear canal itself are combined, the shape of the resulting transfer function impinging on the eardrum is radically changed. The transfer function of the ear canal alone is a static, fixed function that does not change with direction of arrival of the sound. As we have seen, the ear canal acts like a quarter-wave pipe closed at one end by the eardrum and exhibiting prominent resonance.

The transfer function representing the specific direction to the source (see Fig. 4-14) combining with the fixed transfer function of the ear canal gives the transfer function at the eardrum of Fig. 4-15. In this example, the brain translates this to a perception of sound coming from directly in front of the listener.

The transfer function at the entrance to the ear canal is shaped differently for different horizontal and vertical directions. This is how the pinna encodes all arriving sound enabling the brain to yield different perceptions of direction. The sound arriving at the eardrum is the raw material for all directional perceptions. The brain neglects the fixed component of the ear canal and translates the differently shaped transfer functions to directional perceptions.

Another more obvious directional function of the pinna is that of forward–backward discrimination, which does not directly depend on spatial encoding. At higher frequencies (shorter wavelengths), the pinna is an effective barrier; sounds from behind have relatively lower high-frequency levels. The brain uses this front–back differentiation to convey a general perception of direction.

The ear can also weakly perceive vertical localization. The median plane is a vertical plane passing symmetrically through the center of the head and nose. Sources of sound in this plane present identical transfer functions to the two ears. The auditory mechanism uses another technique for such localization, that of giving a certain place identity to different frequencies. For example, signal components near 500 and 8,000 Hz are perceived as coming from directly overhead, components near 1,000 and 10,000 Hz as coming from the rear.

Sound arriving from directly in front of a listener results in a peak in the transfer function at the eardrum in the 2- to 3-kHz region. This is partly the basis of the technique of imparting “presence” to a recorded voice by adding an equalization boost in this frequency region. A voice can also be made to stand out from a musical background by adding such a peak to the voice response.

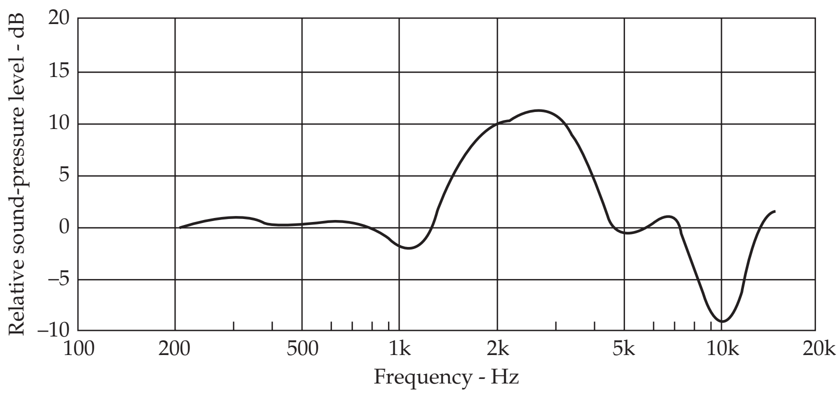

Figure 4-15 The transfer function of Fig. 4-14 at the opening of the ear canal is altered to this shape at the eardrum after being combined with the transfer function of the ear canal. In other words, a sound arriving at the opening of the ear canal from a source directly in front of the observer (see Fig. 4-14) describes this response at the eardrum because it has been combined with the characteristics of the ear canal itself. The brain discounts the fixed influence of the ear canal from every changing arriving sound.

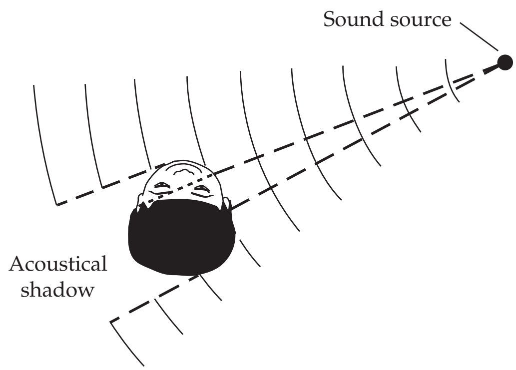

Figure 4-16 Our binaural directional sense depends in part on the difference in intensity and time of arrival of the sound falling on two ears.

Binaural Localization

Two ears function together in binaural hearing to allow localization of sound sources in the horizontal plane. Signals from both ears are combined in the brain; thus localization largely takes place in the brain, and not in the individual ears. Two factors are involved, the difference in intensity and the difference in time of arrival (phase) of the sound falling on the two ears. In Fig. 4-16, the ear nearest the source receives a greater intensity than the far ear because it is closer, and because the head casts an acoustical shadow. Because of diffraction, acoustical shadows are much weaker at lower frequencies. However, at higher frequencies, the acoustical shadow, combined with the path-length difference, results in a higher intensity at the nearest ear.

Because of the difference in distance from the source, the near ear receives sound somewhat earlier than the far ear. Below 1 kHz, the phase (time) effect dominates while above 1 kHz the intensity effect dominates. There is one localization blind spot. A listener cannot tell whether sounds are coming from directly in front or from directly behind because the intensity of sound arriving at each ear is the same and in the same phase. Using these cues, the ear can localize sound sources in the horizontal plane to within 1° or 2°.

Law of the First Wavefront

The sound that arrives first creates in the listener the perception of direction; this is sometimes called the law of the first wavefront. Imagine two people in a small room, one person speaking and the other listening. The first sound to reach the listener is that traveling a direct path because it travels the shortest distance. This direct sound establishes the perception of the direction from which the sound came. Even though it is immediately followed by a multitude of reflections from the various surfaces of the room, this directional perception persists and tends to diminish the effects of later reflections insofar as direction is concerned. This identification of the direction to the source of sound is accomplished within a small fraction of a millisecond.

The Franssen Effect

The ear is relatively adept at identifying the locations of sound sources. However, it also employs an auditory memory that can sometimes confuse direction. The Franssen effect demonstrates this. Two loudspeakers are placed to the left and right of a listener in a live room. The loudspeakers are about 3 ft from the listener at about 45° angles. A sine wave is played through the left loudspeaker, and the signal is immediately faded out and simultaneously faded in at the right loudspeaker, so there is no appreciable change in overall level. Most listeners will continue to locate the signal in the left loudspeaker, even though it is silent and the sound location has changed to the right loudspeaker. They are often surprised when the cable to the left loudspeaker is disconnected, and they continue to "hear" the signal coming from the left loudspeaker. This demonstrates the role of auditory memory in sound localization.

The Precedence (Haas) Effect

Our hearing mechanism integrates spatially separated sounds over short intervals, and under certain conditions tends to perceive them as coming from one location. For example, in an auditorium, the ear and brain have the ability to gather all reflections arriving within about 35 msec after the direct sound and combine (integrate) them to give the impression that the entire sound field is from the direction of the original source, even though reflections from other directions are involved. The sound that arrives first establishes the perceptual source location of later sounds. This is variously called the precedence effect, or Haas effect, and obeys the law of the first wavefront. The sound energy integrated over this period also gives an impression of added loudness.

It is not too surprising that the human ear fuses sounds arriving during a certain time window. After all, at the cinema, our eyes fuse a series of still pictures, giving the impression of continuous movement. The rate of presentation of the still pictures is important; there must be at least 16 pictures per second (62-msec interval) to avoid seeing a series of still pictures or a flicker. Auditory fusion similarly is a process of temporal fusion. Auditory fusion works best during the first 35 msec after the onset of sound; beyond 50 to 80 msec the integration breaks down, and with long delays, discrete echoes are heard.

Haas placed his subjects 3 m from two loudspeakers arranged so that they subtended an angle of 45°, the listener's line of symmetry splitting this angle (there is some ambiguity in the literature about the angle). The rooftop conditions were approximately anechoic. Both loudspeakers played the same speech content and at the same level, but one loudspeaker was delayed relative to the other. Clearly, sound from the undelayed loudspeaker arrived at the listening position slightly before the sound from the delayed loudspeaker. Haas studied the effects of varying the delay on speech signals. As shown in Fig. 4-17, Haas found that in the 5- to 35-msec delay range, the sound from the delayed loudspeaker was perceived as coming from the undelayed loudspeaker. In other words, listeners localized both sources to the location of the undelayed source.

Moreover, the level of the delayed sound had to be increased by more than 10 dB over the undelayed sound before its location was heard as being separate. In a room, reflected energy arriving at the ear within 35 msec is spatially integrated with the direct sound and is perceived as being spatially part of the direct sound as opposed to reverberant sound. This is sometimes called the fusion zone or Haas zone. These integrated early reflections

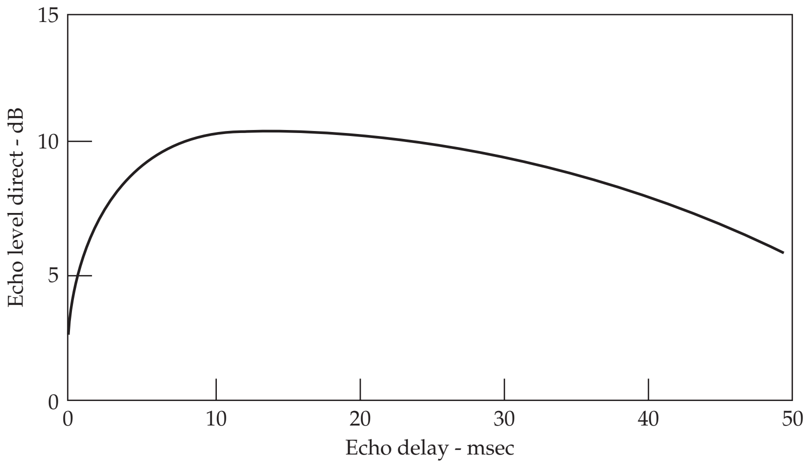

FIGURE 4-17 The precedence effect, or Haas effect, in the human auditory system describes temporal fusion. In the 5- to 35-msec region, the echo level must be about 10 dB higher than the direct sound to be discernible as an echo. In this region, reflected components arriving from many directions are integrated by the ear. The resulting sound appears to come from the direct source and seems louder because of the reflections. For delays 50 to 100 msec and longer, reflections are perceived as discrete echoes. (*Haas.*)

increase the loudness of the direct sound and can change its timbre. As Haas said, they result in ". . . a pleasant modification of the sound impression in the sense of broadening of the primary sound source while the echo source is not perceived acoustically."

The transition zone between the integrating effect for delays less than 35 msec and the perception of delayed sound as spatially discrete is gradual, and therefore somewhat indefinite. Some researchers set the dividing line at 62 msec (1/16 sec), some at 80 msec, and some at 100 msec beyond which there is no question about the discrete location of the delayed sound. If the delayed sound is attenuated, the fusion zone is extended. For example, if the delayed sound is –3 dB relative to the first sound, integration extends to about 80 msec. Room reflections are lower in level than direct sound, so we expect that integration will extend over a longer time. However, with very long delays of perhaps 250 msec or more, the delayed sound is clearly heard as a discrete echo.

Other researchers had previously found that very short delays (<1 msec) were involved in our discerning the direction of a source by slightly different times of arrival at our two ears. Delays greater than this do not affect our directional sense.

The precedence effect is easily demonstrated. Stand 100 ft from a concrete wall and clap your hands—a distinct echo will be heard (177 msec). As you move closer and continue to clap your hands, the echo will arrive sooner and will be louder. But as you enter the fusion zone, your ears will spatially integrate the echo into the direct sound; you do not perceive the echo.

Perception of Reflected Sound

In the preceding section, "reflected" sound was considered in a rather limited way. In this section, a more general approach is taken. The loudspeaker arrangement used by Haas was also used by other researchers; in this familiar stereo setup the listener is located symmetrically between two separated loudspeakers. The sound from one loudspeaker is designated as the direct sound, that from the other loudspeaker, the delayed sound (the reflection). The delay injected between the two signals and their relative levels is adjustable.

With the sound of the direct loudspeaker set at a comfortable level, and with a delay of, say 10 msec, the level of the reflected, or delayed, loudspeaker sound is slowly increased from a very low level. The sound level of the reflection at which the observer first detects a difference in the sound is the threshold of reflection detection. For levels less than this, the reflection is inaudible; for levels greater than this, the reflection is clearly audible.

As the reflection level is gradually increased above the threshold value, a sense of spaciousness is imparted to the combined sound. This sense of spaciousness prevails even though the experiment is conducted in an anechoic space. As the level of the reflection is increased about 10 dB above the threshold value, another change is noticed in the sound; a broadening of the sound image and possibly a shifting of the image toward the direct loudspeaker added to the increasing sense of spaciousness. As the reflection level is increased another 10 dB or so above the image broadening threshold, another change is noted; discrete echoes are heard.

What practical value does this have? Consider the example of a listening room used for playback of recorded music. Figure 4-18 shows the effect of lateral reflections added to the direct sound from the loudspeakers. Speech is used as the signal. Reflections below the threshold of perception are unusable; reflections perceived as discrete echoes are also unusable. The usable area is the unshaded area between those two threshold curves, A and C. Calculations can give estimates of the level and delay of any specific reflection, knowing the speed of sound, the distance traveled, and applying the inverse square law. Figure 4-18 also shows the subjective reactions listeners will probably have to the combination of reflected and direct sound.

To assist in the calculations mentioned previously, the following equations can be applied:

$$\text{Reflection delay} = \frac{(\text{Reflected path}) - (\text{Direct path})}{1{,}130} \tag{4-2}$$

This assumes 100% reflection at the reflecting surface. Both path lengths are measured in feet; the speed of sound is measured in feet per second.

$$\text{Reflection level at listening position} = 20 \log \frac{\text{Direct path}}{\text{Reflected path}} \tag{4-3}$$

This assumes inverse square propagation. Both path lengths are measured in feet.

In an auditorium, for example, the room geometry can be designed so that time delays are less than 50 msec and thus fall within the fusion zone. Consider a direct sound with a path length of 50 ft, and an early reflection with a path length of 75 ft, both reaching a listener. The resulting time delay of 22 msec is well within the fusion zone. Similarly, by limiting the time delay between the direct sound and early reflections to less than 50 msec (a path length difference of about 55 ft), the listener will not hear the reflections as discrete echoes. If the typical attenuation of reflected sound is accounted

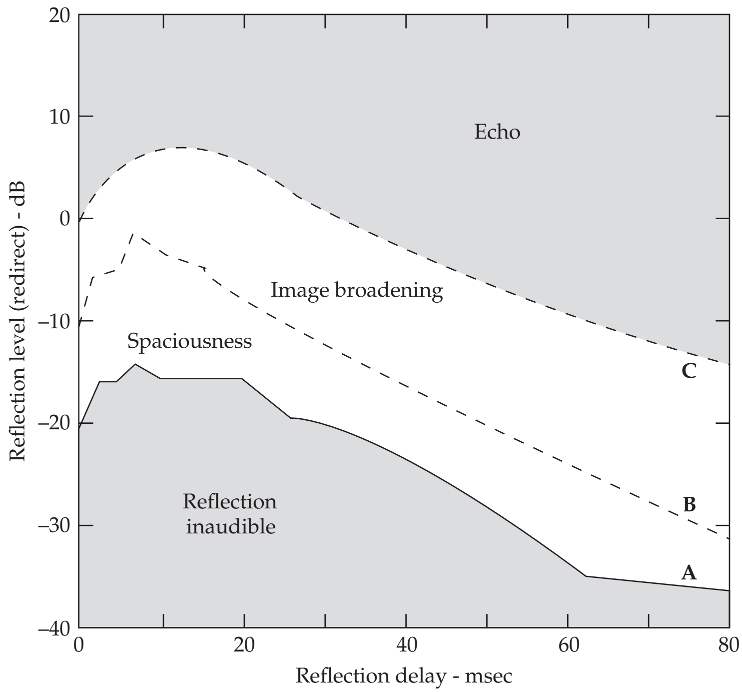

Figure 4-18 The effects of lateral reflections on the perception of the direct sound in a simulated stereo arrangement. These measurements were made in anechoic conditions, lateral angles 45° to 90°, with speech as the signal. Curve A is the absolute threshold of audibility of the reflection. Curve B is the image shift/broadening threshold. Curve C is the lateral reflection perceived as a discrete echo. (*A and B, Olive and Toole, and Toole; C, Meyer and Schodder, and Lochner and Burger.*)

for, differences slightly greater than 50 msec are permissible. Very generally, maximum differences of 50 msec are allowable for speech, while a maximum of 80 msec is allowable for music. Shorter differences are preferred. As we will see later, the precedence effect can also be used in the design of live end-dead end (LEDE) control rooms.

The Cocktail-Party Effect

The human auditory system possesses a powerful ability to distinguish between different sounds and to direct our attention to one sound amid many. This is sometimes called "the cocktail-party effect" or "auditory scene analysis." Imagine yourself at a crowded party with many talkers and music playing. You are able to listen to one talker while excluding many other conversations and sounds. If someone across the room speaks your name, you will be alert to that. There is evidence that musicians and conductors are highly skilled at this auditory segregation; they can independently follow the sounds of multiple musical instruments simultaneously.

This ability to distinguish between particular sounds is greatly assisted by our localization abilities. If the voices of two talkers are played over one loudspeaker, it can be difficult to differentiate them. However, if two physically separated loudspeakers are set up, and one voice is played over one loudspeaker, and the other voice is played over the other loudspeaker, it is quite easy to follow both voices (factors such as relative language, gender, and pitch of the talkers also play a role). While humans function well at differentiating sources at cocktail parties, electronic signal processing systems have a more difficult time. This field of signal processing is referred to as source separation or blind source separation.

Aural Nonlinearity

When multiple frequencies are input to a linear system, the same frequencies are output. However, the ear is a nonlinear system. When multiple frequencies are input, the output can contain additional frequencies. This is a form of distortion that is introduced by the auditory system and it cannot be measured by ordinary instruments. It is a subjective effect requiring a different approach. The following experiment demonstrates the nonlinearity of the ear and the output of aural harmonics. It can be performed with a stereo playback system and two audio oscillators. Plug one oscillator into the left channel and the other into the right channel, and adjust both channels for an equal and comfortable volume level at some midband frequency. Then tune one oscillator to 23 kHz and the other to 24 kHz without changing the level settings. With either oscillator alone, nothing is heard because the signal is outside the range of the ear. However, if the tweeters are good enough, you might hear a distinct 1-kHz tone.

The 1-kHz tone is the difference between 23 kHz and 24 kHz. The sum, 47 kHz, is another sideband. Such sum and difference sidebands are generated whenever two pure tones are mixed in a nonlinear element. The nonlinear element in this case is the middle and inner ear. In addition to intermodulation products, the nonlinearity of the ear generates new harmonics that are not present in the sound falling on the eardrum.

Another demonstration of auditory nonlinearity can be performed with the same equipment used above, with the addition of headphones. First, a 150-Hz tone is applied to the left earphone channel. If the hearing mechanism were perfectly linear, no aural harmonics would be heard as the exploratory tone in the right earphone channel is swept upward near the frequencies of the second, third, and other harmonics. However, since it is nonlinear, the presence of aural harmonics is indicated by the generation of beats. When 150 Hz is applied to the left ear and the exploratory tone of the right ear is slowly varied about 300 Hz, the second harmonic is indicated by the presence of beats between the two. If we change the exploratory oscillator to a frequency around 450 Hz, the presence of a third harmonic will also be revealed by beats. Researchers have estimated the magnitude of the harmonics by the strength of such beats. Conducting this experiment with tones of a higher level will make the presence of aural harmonics even more obvious.

Subjective versus Objective Evaluation

There remains a great divide between subjective assessments of sound quality and objective measurements. Consider the following descriptive words which are often applied to concert-hall acoustics: warmth, bassiness, definition, reverberance, fullness of tone, liveness, sonority, clarity, brilliance, resonance, blend, and intimacy. There is no instrument to directly measure qualities such as warmth or brilliance. However, in some cases, subjective terms can be related to objective measurements. For example, consider the term “definition.” German researchers have adopted the term “deutlichkeit” which literally means clearness or distinctness. It is measured by taking the energy in an echogram during the first 50 to 80 msec and comparing it to the energy of the entire echogram. This compares the direct sound and early reflections, which are integrated by the ear, to the entire extent of reverberant sound. This is a straightforward measurement of an impulsive sound from a pistol or other source.

Measurements are vitally important, but the ear is the final arbiter. Observations by human subjects provide valuable input to any acoustical evaluation. For example, in a loudness investigation, panels of listeners are presented with various sounds, and each observer is asked to compare the loudness of sound A with that of B. The data submitted by the jury of listeners is then subjected to statistical analysis; the dependence of human sensory factors, such as loudness, upon physical measurements of sound level is assessed. If the test is conducted properly and sufficient observers are involved, the results are trustworthy. In this way, for example, we discover that there is no linear relationship between sound level and loudness, pitch and frequency, or timbre and sound quality.

It is desirable to correlate subjective impressions of listeners with objective design parameters. This allows designers to know where audio fidelity limitations exist and thus know where improvements can be made. For example, this knowledge would allow optimization of the acoustics of a concert hall. The correlation between the listener’s impressions and the objective means to measure the phenomenon is a difficult problem. Correlations are not always known. One way to correlate subjective impressions with objective data is with research, in particular through critical listening. Over time, it is possible that patterns will emerge that will provide correlation. Though correlation is desirable, critical listening plays an important role without it.

Occupational and Recreational Hearing Loss

Hearing damage is a serious occupational hazard. Industrial factory workers, truck drivers, and many others may be subject to noise levels that are potentially harmful. Over time, with repeated exposure, hearing loss may occur. Audiologists determine what noise exposure workers are subjected to in different environments. This is not easy because noise levels fluctuate and workers move about; wearable dosimeters are often used to integrate the exposure over the workday. Companies often install noise shields around offending equipment and mandate ear plugs or ear muffs for workers.

The hearing of workers in industry is protected by law. The federal Occupational Safety and Health Administration (OSHA) in the Department of Labor maintains noise exposure limits in the workplace. The higher the occupational noise, the less time of exposure is allowed. Noise exposure is measured in daily noise doses for an 8-hour day. Table 4-2 lists the permissible daily noise exposure, measured with the slow response of a standard sound-level meter. The maximum allowed dose is 100% of the daily limit. A dose is calculated as the time a worker is exposed to different noise levels, relative to the maximum exposure time permitted at that level. For example, a worker may be exposed to a maximum of a 90-dBA

Sound-Pressure Level, dBA	Maximum Daily Exposure, Hours
85	16
90	8
92	6
95	4
97	3
100	2
102	1.5
105	1
110	0.5
115	0.25 or less

Table 4-2 OSHA Permissible Noise Exposure Times

noise for 8 hours, a 100-dBA noise for 2 hours, or a 115-dBA noise for 15 minutes. When the daily exposure is due to two or more noise levels, the total noise dose is given by:

$$D = \frac{C1}{T1} + \frac{C2}{T2} + \frac{C3}{T3} + \cdots \qquad (4\text{-}4)$$

where C = duration of exposure, hours
T = noise exposure limit, hours

For example, when a worker is exposed to a noise level of 100 dBA for 1.5 hours and a 95-dBA level for 0.5 hour, the noise dose is $D = 1.5/2 + 0.5/4 = 0.90$. Thus, the worker has been exposed to 90% of the maximum permissible noise.

The time-weighted and A-weighted sound level in dBA, sometimes referred to as TWA, may also be computed as TWA = 105 – 16.6 log (T). A subsequent hearing conservation measure calls for a TWA given as 100 – 16.6 log (T).

Other noise exposure regulations have been devised by the Environmental Protection Agency, Department of Housing and Urban Development, Workers' Compensation, and other agencies and nongovernment groups. These regulations are subject to frequent change. Professional audio engineers operating with high monitoring levels are risking irreparable injury to their hearing; in most cases, their work is not subject to occupational noise-protection laws.

Dangerous noise exposure is more than an occupational problem; it is also a recreational problem. An individual might work all day in a high-noise environment, then enjoy motorcycle or automobile racing, listen to music playback at high level, or spend hours in a nightclub. As high-frequency loss creeps in, volume controls are turned up to compensate, and the rate of deterioration is accelerated.

The audiogram is an important tool in the conservation of hearing. Comparing a current audiogram with earlier ones establishes the trend; if downward, steps can be taken to address the causes. The audiogram of Fig. 4-19 is that of a mixing engineer in a recording studio and shows serious hearing loss. This hearing loss, centered on 4 kHz, may be the accumulation of many years of listening to high-level sounds in the control room.

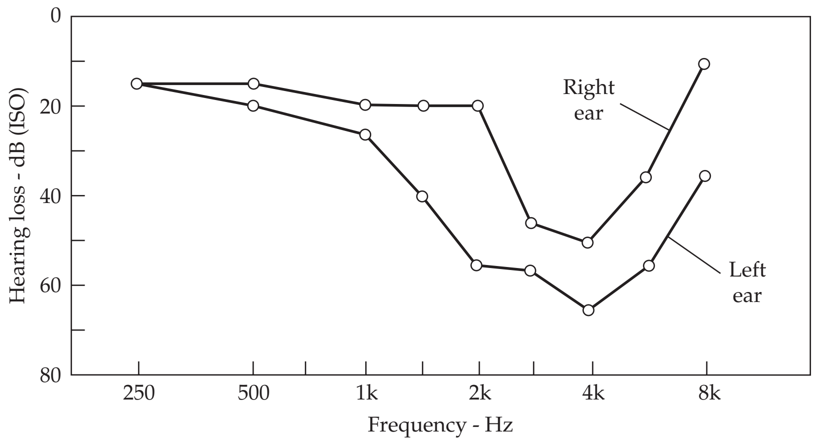

Figure 4-19 Audiograms showing serious hearing loss centered on 4 kHz, perhaps resulting from years of exposure to high-level sound in the control room of a recording studio.

Key Points

- The outer-ear pinna and the auditory canal, acting as a quarter-wave pipe closed at one end by the eardrum, contribute acoustical amplification in a region near 3 kHz. Vital speech frequencies lie in this region.
- The leverage of the ossicles bones of the middle ear and the ratio of areas of the eardrum and oval window efficiently match the impedance of air to the fluid of the inner ear.
- Waves set up in the inner ear by vibration of the oval window excite the sensory hair cells, which direct signals to the brain. There is a place effect, the peak of hair cell agitation for higher frequencies being nearer the oval window and low frequencies at the distal end.
- The area of audibility is bounded by two threshold curves, the threshold of audibility at the soft extreme and the threshold of feeling or pain at the loud extreme. Our auditory experience occurs within these two extremes.
- The loudness of tone bursts decreases as the length of the burst is decreased. Bursts greater than 200 msec have full loudness, indicating a time constant of the ear of about 100 msec.
- Our ears are capable of accurately locating the direction of a source in the horizontal plane. In a vertical median plane, however, localization ability is less accurate.
- Pitch is a subjective term. Frequency is the associated physical term, and the two have only a general relationship.
- Subjective timbre or quality of sound and the physical spectrum of the sound are related, but not equivalent.
- The nonlinearity of the ear generates intermodulation products and spurious harmonics.

- The precedence effect, or Haas effect, describes the ability of the ear to integrate all sound arriving within the first 35 msec, giving the perception that the sounds originate from the earlier source, and making the sounds seem louder.
- The "cocktail-party effect" demonstrates that our ability to distinguish between particular sounds is greatly assisted by our localization abilities.
- Although the ear is generally not effective at evaluating absolute sound parameters, it is very keen in comparing frequencies, levels, and sound quality.
- Occupational and recreational noises can cause temporary and permanent hearing loss. Precautionary steps to minimize this type of environmentally caused deafness are recommended.

CHAPTER 5

Signals, Speech, Music, and Noise

Signals such as speech, music, and noise are within the common experience of most people. With the gift of hearing, we are profoundly familiar with the sound of speech; we hear speech virtually every day of our lives. Speech is the key to human communication, and poor speech intelligibility is extremely frustrating. If we are lucky, we also hear music every day. Music can be one of the most pleasurable and needful of human experiences. It would be hard to imagine a world without music. Noise is usually considered as an intrusive and unwanted sound, often chaotic and disruptive to speech, music, or silence. The close relationship of speech, music, and noise is explored in this chapter.

Sound Spectrograph

A consideration of speech sounds is necessary to understand how the sounds are produced. Speech, like music, is highly variable and transient in nature, comprising energy moving through the three-dimensional scales of frequency, sound level, and time. A sound spectrograph can show all three variables. Every sound has its spectrographic signature that reveals the energy that characterizes it. The spectrographs of several commonly experienced sounds are shown in Fig. 5-1. In these spectrographs, time progresses horizontally to the right, frequency increases from the origin upward, and the sound level is indicated roughly by the density of the trace—the darker the trace, the more intense the sound at that frequency and at that moment of time. Random noise on such a plot shows up as gray, slightly mottled rectangles as all frequencies in the audible range and all intensities are represented as time progresses. The snare drum approaches random noise at certain points, but it is intermittent. The wolf whistle opens on a rising note followed by a gap, and then a similar rising note that then falls in frequency. The police whistle is a tone, slightly frequency modulated.

The human voice mechanism is capable of producing many sounds other than speech. Figure 5-2 shows a number of these sound spectrographs. Harmonic series appear on a spectrograph as more or less horizontal lines spaced vertically in frequency. These are particularly noticeable in the trained soprano's voice and the baby's cry, but traces are also evident in other spectrographs.

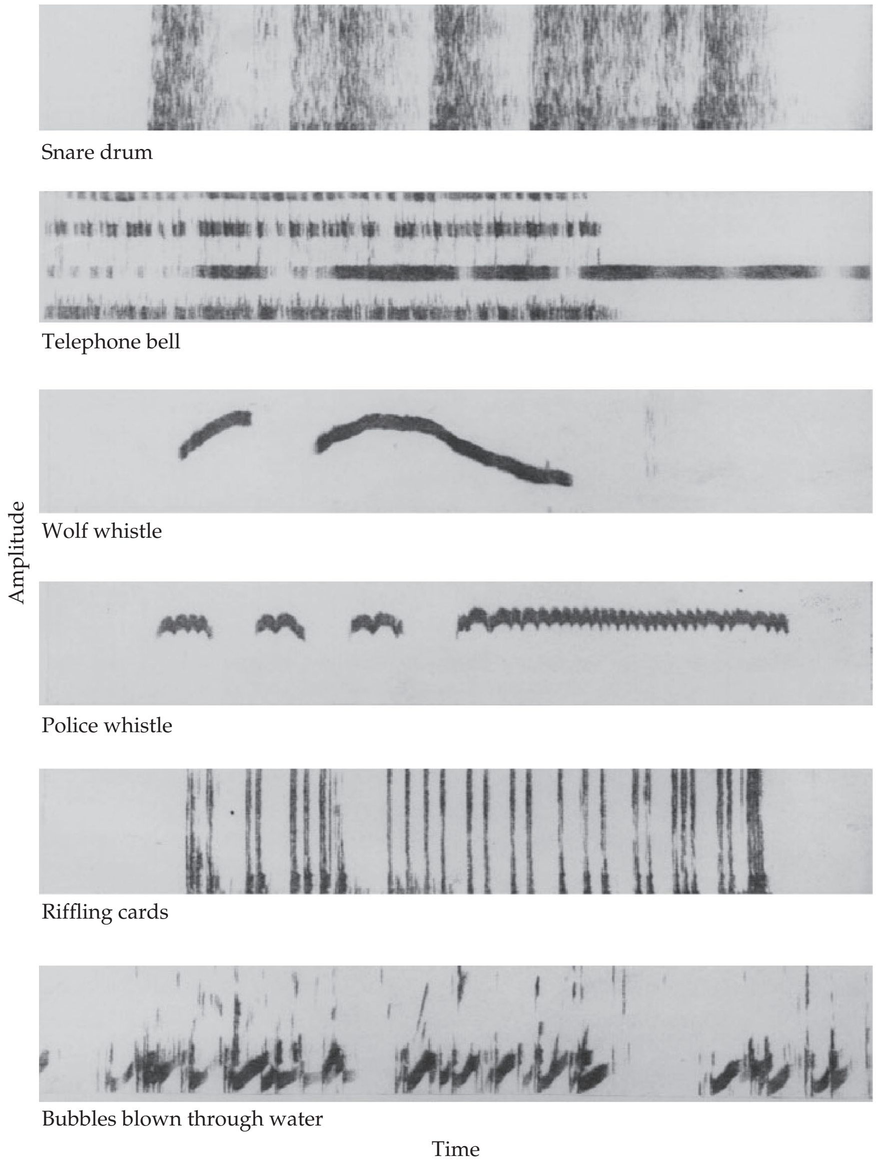

Figure 5-1 Spectrographs of common sounds. Time progresses to the right, the vertical axis is frequency, and the intensity of components is represented by the intensity of the trace. (*AT&T Bell Laboratories.*)

Speech

There are two quasi-independent components in the generation of speech sounds: the sound source and the vocal system. In general, speech is a two-stage process, as pictured in Fig. 5-3A, in which the raw sound is produced by a source and subsequently shaped in the vocal tract. To be more exact, three different sources of sound are shaped by the vocal

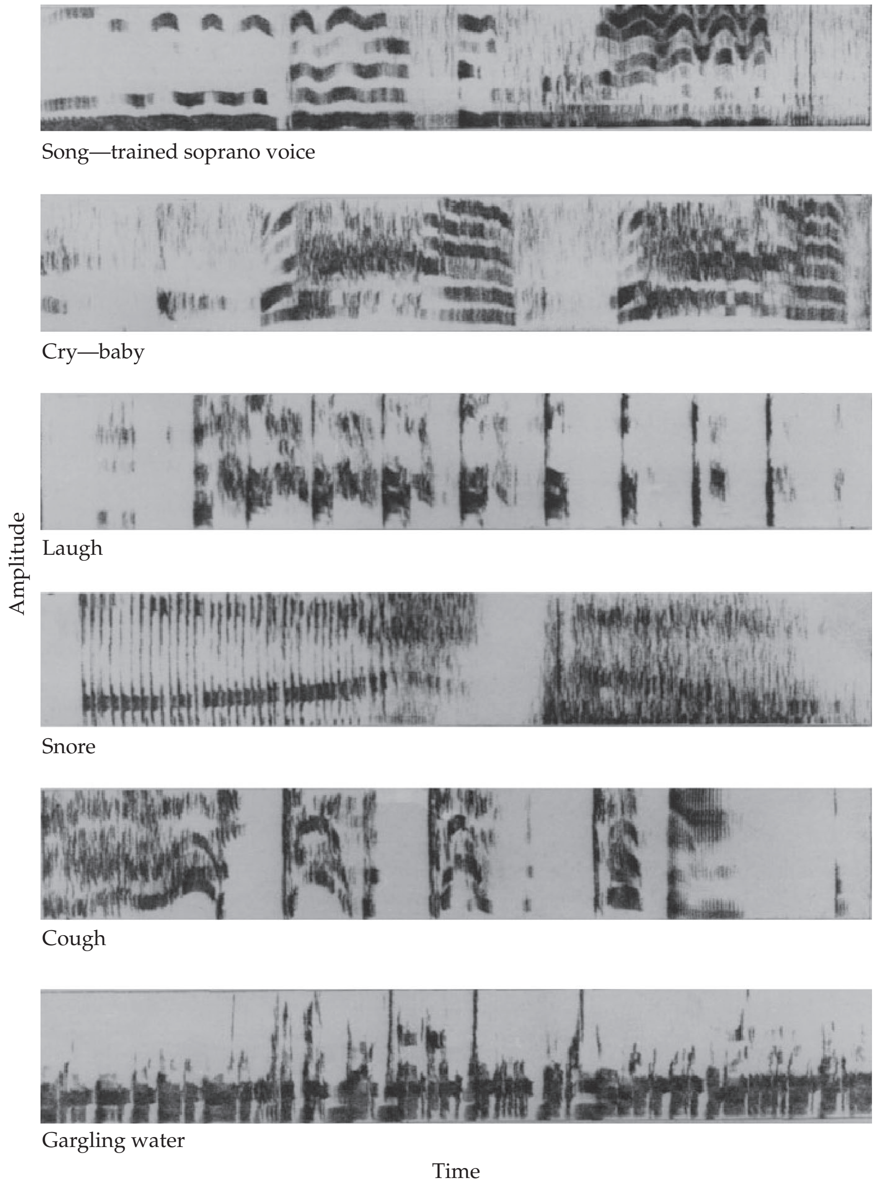

Figure 5-2 Spectrographs of human sounds other than speech. (*AT&T Bell Laboratories.*)

tract, as shown in Fig. 5-3B. First, there is the sound we naturally think of—the sounds emitted by the vocal cords. These are formed into the voiced sounds. They are produced by air from the lungs flowing through an open vocal tract, past the slit between the vocal cords (the glottis), which causes the cords to vibrate. This air stream, broken into pulses of air, produces a sound that can almost be called periodic, that is, repetitive in the sense that one cycle follows another. The result is vowel sounds such as *a, e, i, o,* and *u.*

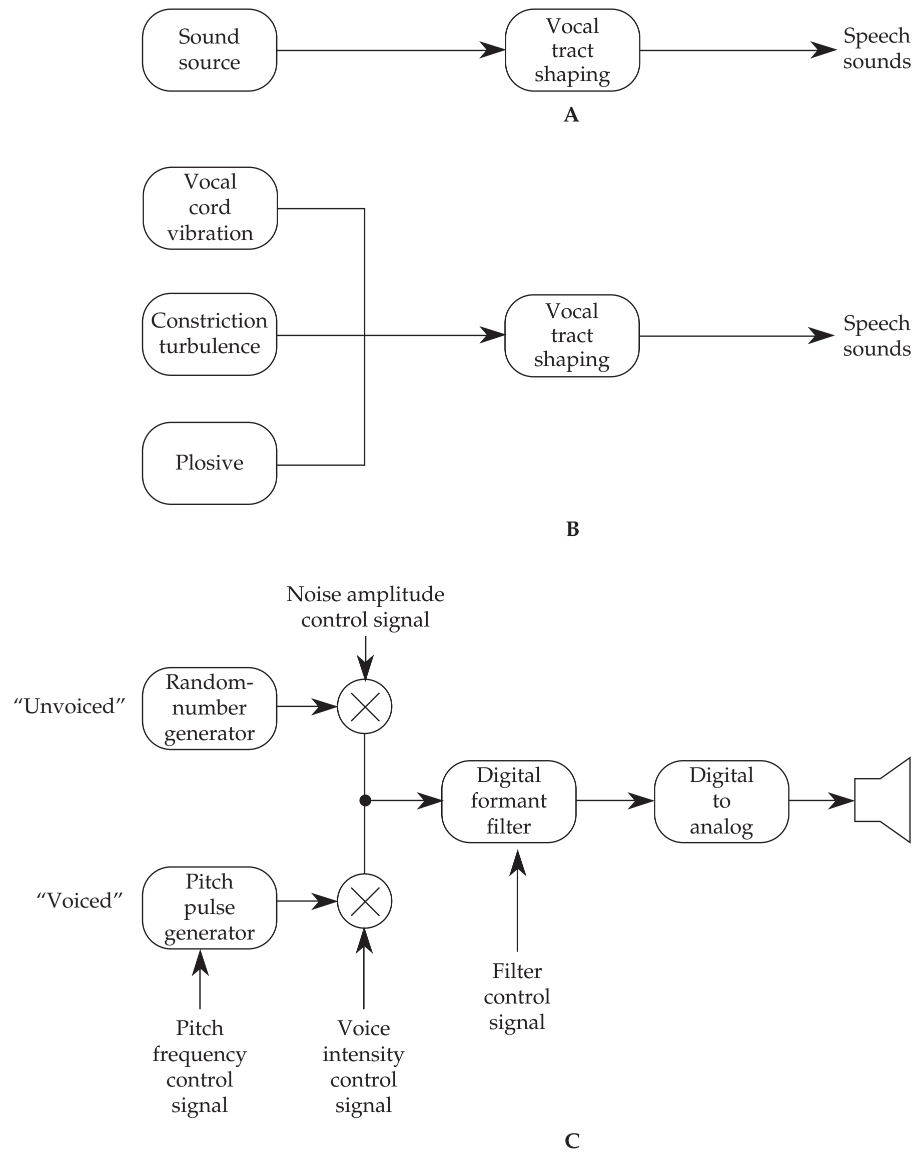

Figure 5-3 Three models of human speech. (A) The human voice is produced through the interaction of two essentially independent components: a sound source, and a time-varying filter action of the vocal tract. (B) The sound source is composed of vocal-cord vibration for voiced sounds, the fricative sounds resulting from air turbulence, and plosive sounds. (C) A digital system used to synthesize human speech.

The second source of speech sound is that made by forming a constriction at some point in the vocal tract with the teeth, tongue, or lips, and forcing air through it under pressure high enough to produce significant turbulence. Turbulent air creates noise. This noise is shaped by the vocal tract to form the fricative sounds of speech such as the consonants *f*, *s*, *v*, and *z*. Try making these sounds, and you will see that high-velocity air is very much involved.

The third source of speech sound is produced by the complete stoppage of the breath, usually toward the front of the mouth, a building up of pressure, and then the sudden release of the breath. Try speaking the consonants *k*, *p*, and *t*, and you will sense the force of such plosive sounds. They are usually followed by a burst of fricative or turbulent sound. These three types of sounds—voiced, fricative, and plosive—are the raw sources that are shaped into the words we speak.

Sound sources and signal processing can be implemented in digital hardware or software. A simple speech synthesis system is shown in Fig. 5-3C. A random-number generator produces the digital equivalent of *s*-like sounds for the unvoiced components. A counter produces pulses simulating the pulses of sound of the vocal cords for the voiced components. These are shaped by time-varying digital filters simulating the varying resonances of the vocal tract. Signals control each of these to form digitized speech, which is then converted to analog form.

Vocal Tract Molding of Speech

The vocal tract can be considered as an acoustically resonant system. This tract, from the vocal cords to the lips, has a length of about 6.7 in (17 cm). Its cross-sectional area is determined by the placement of the lips, jaw, tongue, and velum (a sort of trapdoor that can open or close off the nasal cavity) and varies from 0 to about 3 in^2 (20 cm^2). The nasal cavity has a length of about 4.7 in (12 cm) and has a volume of about 3.7 in^3 (60 cm^3). These dimensions help determine the resonances of the vocal tract and their effect on speech sounds.

Formation of Voiced Sounds

If the components of Fig. 5-3 are elaborated into source spectra and modulating functions, we arrive at something of importance in audio—the spectral distribution of energy in the voice. We also gain a better understanding of the aspects of voice sounds that contribute to the intelligibility of speech in the presence of reverberation and noise. Figure 5-4 shows the steps in producing voiced sounds. First, sound is produced by the vibration of the vocal cords; these are pulses of sound with a fine spectrum that falls off at about 10 dB/octave as frequency is increased, as shown in Fig. 5-4A. The sounds of the vocal cords pass through the vocal tract, which acts as a time-varying filter. The peaks in the contour of Fig. 5-4B are due to the acoustical resonances, called formants, of the vocal tract, which acts as a pipe that is essentially closed at the vocal cord end and open at the mouth end. Such an acoustical pipe 6.7-in long has resonances at odd quarter wavelengths; these peaks occur at approximately 500, 1,500, and 2,500 Hz. The output sound, shaped by the resonances of the vocal tract, is shown in Fig. 5-4C. This analysis applies to the voiced sounds of speech.

Formation of Unvoiced Sounds

Unvoiced sounds are shaped as shown in Fig. 5-5. Their production is similar to that of voiced sounds. Unvoiced sounds start with the distributed, random-noise–like spectrum of the turbulent air as fricative sounds are produced. The distributed spectrum of Fig. 5-5A is generated near the mouth end of the vocal tract, rather than the vocal cord end; hence the resonances of Fig. 5-5B are of a somewhat different shape. Figure 5-5C shows the sound output shaped by the time-varying filter action of Fig. 5-5B.

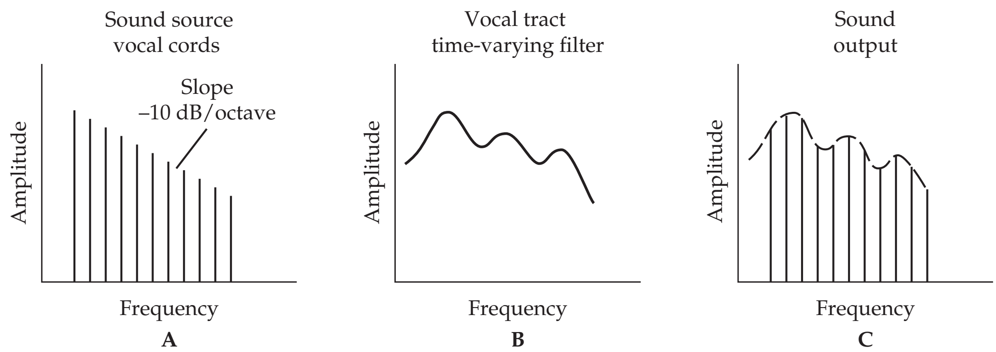

Figure 5-4 The production of voiced sounds can be considered as several steps. (A) Sound is first produced by the vibration of the vocal cords; these are pulses of sound with a spectrum that falls off with frequency. (B) The sounds of the vocal cords pass through the vocal tract, which acts as a time-varying filter. Acoustical resonances, called formants, are characteristic of the vocal pipe. (C) The output voiced sounds of speech are shaped by the resonances of the vocal tract.

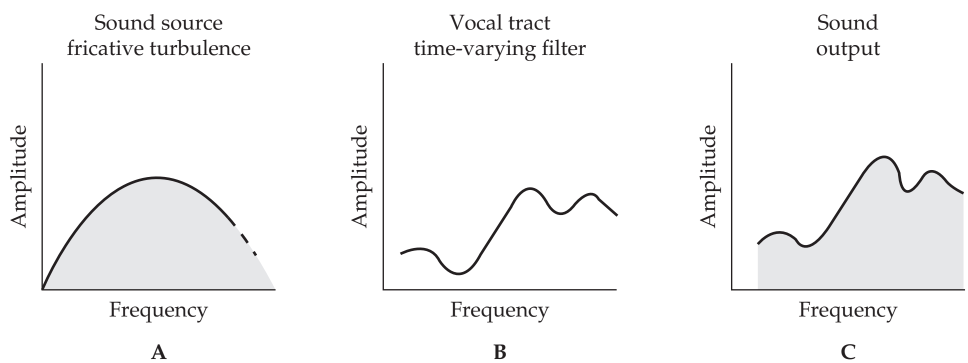

Figure 5-5 A diagram of the production of unvoiced fricative sounds such as *f*, *s*, *v*, and *z*. (A) The distributed spectrum of noise due to air turbulence resulting from constrictions in the vocal tract. (B) The time-varying filter action of the vocal tract. (C) The output sound resulting from the filter action of the distributed sound of (A).

Frequency Response of Speech

The voiced sounds originating in vocal cord vibrations, unvoiced sounds originating in turbulences, and plosives that originate near the lips, together form our speech sounds. As we speak, the formant resonances shift in frequency as the lips, jaw, tongue, and velum change position to generate the desired words. The result is the complexity of human speech evident in the spectrograph of Fig. 5-6. Information communicated via speech is a pattern of frequency and intensity that shifts rapidly with time. Note in Fig. 5-6 that there is little speech energy above 4 kHz. Although it is not shown by the spectrograph, there is also relatively little speech energy below 100 Hz. It is understandable why presence filters peak in the 2- to 3-kHz region; that is where human speech sounds resonate.

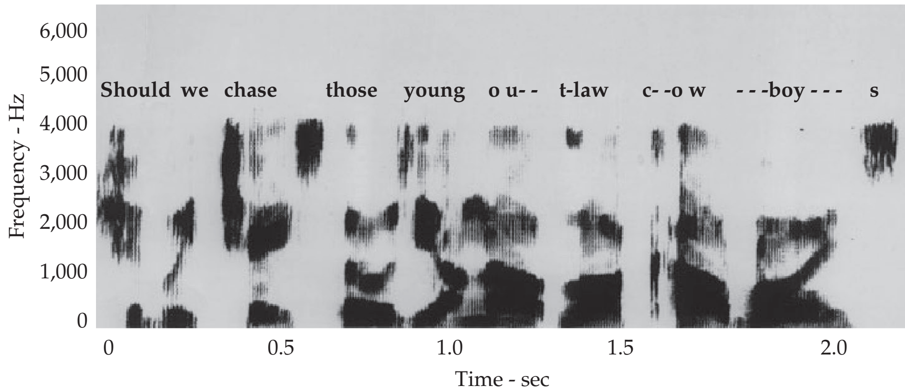

FIGURE 5-6 Sound spectrograph of a sentence spoken by a male voice. (*AT&T Bell Laboratories.*)

Directionality of Speech

Speech sounds do not have the same volume level in all directions. This is primarily due to the directionality of the mouth and the sound shadow cast by the head and torso. Two measurements of speech-sound directionality are shown in Fig. 5-7. Because speech sounds are variable and complex, averaging is necessary to give an accurate measure of directional effects.

The horizontal directional effects shown in Fig. 5-7A demonstrate only a modest directional effect of about 5 dB in the 125- to 250-Hz band. This is expected because the head is small compared to wavelengths of 4.5 to 9 ft associated with this frequency band. However, there are significant directional effects for the 1,400- to 2,000-Hz band. For this band, which contains important speech frequencies, the front-to-back difference is about 12 dB.

In the vertical plane shown in Fig. 5-7B, the 125- to 250-Hz band shows about a 5-dB front-to-back difference again. For the 1,400- to 2,000-Hz band, the front-to-back difference is also about the same as the horizontal plane, except for the torso effect. The discrimination against high-speech frequencies picked up on a lapel microphone is obvious (see Fig. 5-7B), although the measurements were not carried to angles closer to 270°.

Music

Musical sounds are extremely variable in their complexity. Sounds can range from the near sine-wave simplicity of a single instrument or voice to the highly intricate tonalities of a symphony orchestra where each instrument has a different tonal texture for each note.

String Instruments

Musical instruments such as the violin, viola, cello, bass, or guitar produce their tones by the vibration of strings. On a stretched string, the overtones are all exact multiples of the fundamental, the lowest tone produced. These overtones may thus be properly called harmonics. If a string is bowed in the middle, odd harmonics are emphasized because the fundamental and odd harmonics have maximum amplitude there. Because the even

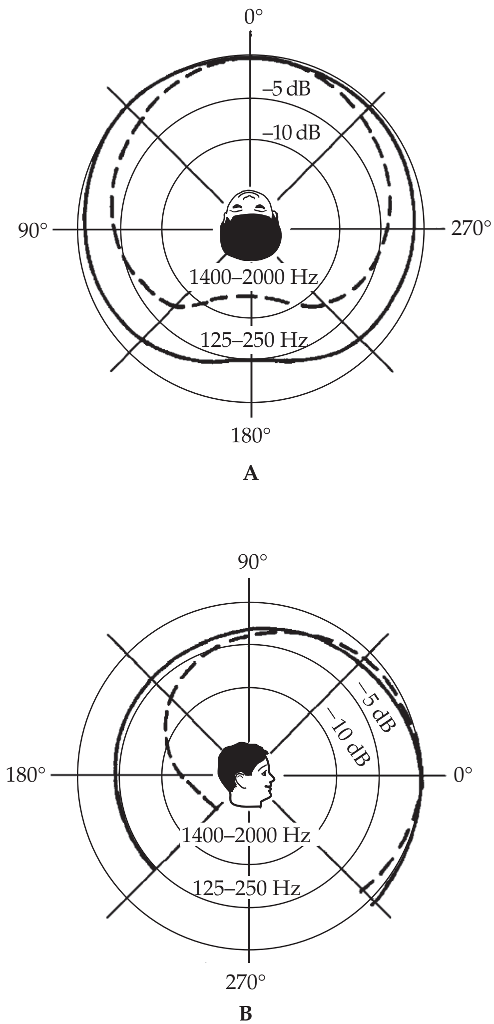

Figure 5-7 The human voice is directional. (A) Front-to-back directional effects of about 12 dB are found for critical speech frequencies. (B) In the vertical plane, the front-to-back directional effects for the 1,400- to 2,000-Hz band are about the same as for the horizontal plane. (*Kuttruff*.)

harmonics have nodes in the center of the string, they will be subdued if bowed there. The usual place for bowing is near one end of the strings, which gives a better blend of even and odd harmonics. The "unmusical" seventh harmonic is anathema in most music (musically, it is a very flat minor seventh). By bowing (or striking or plucking) 1/7 or 2/7 of the distance from one end, this harmonic is decreased. For this reason, the hammers in a piano are located near a node of the seventh harmonic.

The harmonic content of the E and G notes of a violin is shown in Fig. 5-8. Harmonic multiples of the higher E tone are spaced wider and hence have a thinner timbre.

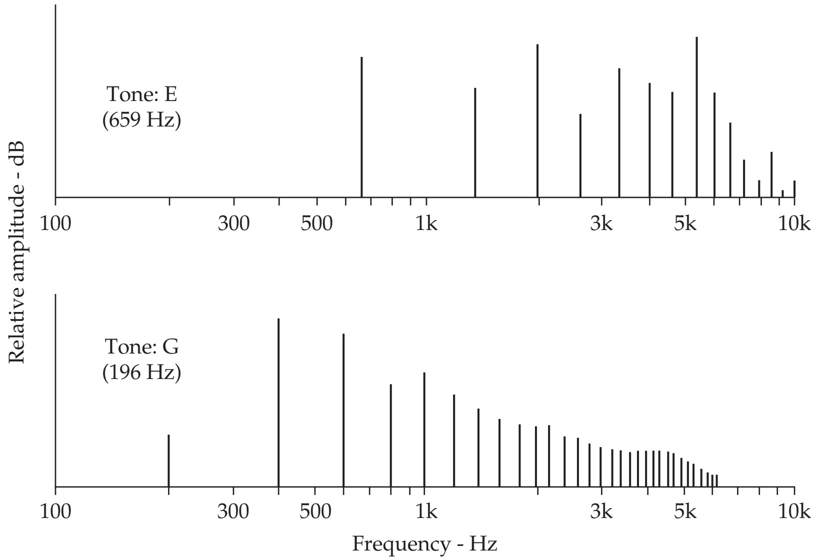

Figure 5-8 Spectra showing the harmonic content of the open strings of the violin. The lower-frequency tones sound richer because of the closely packed harmonics.

The lower G tone, on the other hand, has a closely spaced spectral distribution and a richer timbre. The small size of a violin relative to the low frequency of the G string means that the resonating body cannot produce a fundamental at as high a level as the higher harmonics. The harmonic content and spectral shape depend on the shape and size of the resonating violin body, the type and condition of the wood, and even the varnish. Why there are so few superb violins among the many good ones is a question that has not yet been completely answered.

Wind Instruments

In many musical instruments, resonance in pipes or tubes can be considered to be primarily one-dimensional. (Resonances in three-dimensional rooms are discussed in later chapters.) Standing-wave effects are dominant in pipes. If air is enclosed in a narrow pipe closed at both ends, the fundamental (twice the length of the pipe) and all its harmonics will be formed. Resonances are formed in a pipe open at only one end at the frequency at which the pipe length is 4 times the wavelength, and results in odd harmonics. Wind instruments form their sounds this way; the length of the column of air is continuously varied, as in the slide trombone, or in jumps as in the trumpet or French horn, or by opening or closing holes along its length as in the saxophone, flute, clarinet, and oboe.

The harmonic content of several wind instruments is compared to that of the violin in the spectrographs of Fig. 5-9. Each instrument has its characteristic timbre as determined by the number and strength of its harmonics and by the formant shaping of the train of harmonics by the structural resonances of the instrument.

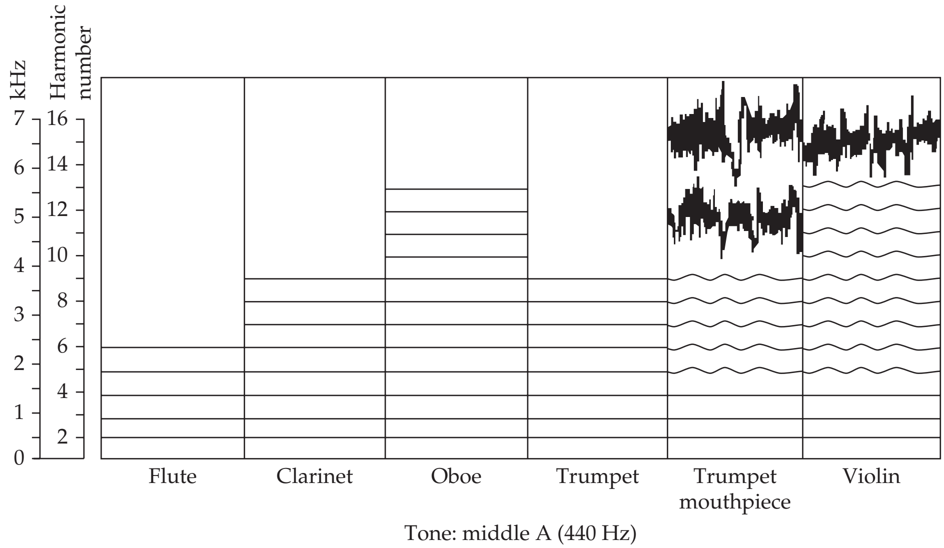

Figure 5-9 Spectrograph comparison of the harmonic content of woodwind instruments and a violin playing middle A (440 Hz). The differences displayed account for the differences in timbre of various instruments. (*AT&T Bell Laboratories.*)

Nonharmonic Overtones

Some musical instruments generate a complex type of nonharmonic overtones. The overtones of drums are not harmonically related, although they add richness to the drum sound. Triangles, bells, and cymbals give a mixture of overtones that blend reasonably well with other instruments. Piano strings are stiff strings and vibrate like a combination of solid rods and stretched strings. Thus piano overtones are not strictly harmonic. Nonharmonic overtones are responsible for the differences between organ and piano sounds, and give variety to musical sounds in general.

Dynamic Range of Speech and Music

The dynamic range of speech is relatively limited. From the softest to the loudest speech sounds, a voice spoken with normal effort might have a dynamic range of 30 to 40 dB. With more effort, the range of loud speech might be 60 to 70 dB. Even this range can be easily accommodated by audio recording technologies. Music, however, has historically posed a more difficult challenge for recording and transmission.

In the concert hall, a full symphony orchestra is capable of producing very loud sounds, but also soft, delicate passages. Seated in the audience, one can fully appreciate this sweep of sound due to the great dynamic range of the human ear. The dynamic range between the loudest and the softest passage may be 100 dB (the ear's dynamic range is about 120 dB). To be effective, the soft passages must still be audible above the ambient background noise in the hall, hence the emphasis on adequate acoustical

Number of Binary Digits	Dynamic Range (dB)
4	24
8	48
12	72
16	96
24	144

Table 5-1 Theoretical Dynamic Ranges for Digital Word Lengths

isolation to protect against traffic and other outside noises, and precautions to ensure that noise from air-handling equipment is low.

For those not present in the music hall, live radio or television broadcast, or recordings must suffice. Conventional analog radio broadcasts such as analog FM transmission are unable to handle the full dynamic range of an orchestra. Noise at the lower extreme and distortion at the upper extreme introduce limitations. In addition, broadcast regulatory restrictions that prohibit interference with adjacent channels also introduce limitations in dynamic range.

Digital audio, ideally, has the dynamic range and signal-to-noise ratio needed to fully capture music. The dynamic range in a digital system is directly related to the word length of binary digits (bits) as shown in Table 5-1. A Compact Disc, for example, stores 16-bit words and thus is capable of storing music with a 96-dB dynamic range; this can be extended if the signal is properly dithered. Consumer formats such as Blu-ray Audio discs as well as professional audio recorders can provide 24-bit resolution and avoid the audibility of digital artifacts resulting from subsequent signal processing. Digital techniques, when taking full advantage of the technology, largely transfer dynamic range limitations from the recording medium to the playback environment. On the other hand, digital formats such as AAC, MP3, and WMA provide dynamic range and fidelity that can be quite variable; their quality depends on factors such as the bit rate of the recorded or streaming file.

Power in Speech and Music

In many applications, one must consider the power of sound sources. For conversational speech, the average power is perhaps 20 μW, but peaks might reach 200 μW. Most of the power of speech is at mid-to-low frequencies, with 80% below 500 Hz, yet there is very little power below 100 Hz. On the other hand, the small amount of power at high frequencies is where consonants are and determines the intelligibility of speech. Higher and lower frequencies outside this range add a natural quality to speech but do not contribute to intelligibility.

Musical instruments can produce more power than the human voice. For example, a trombone might generate a peak power of 6 W, and a full symphony orchestra's peak power might be 70 W. The peak power levels of various musical instruments are listed in Table 5-2.

Instrument	Peak Power (W)
Full orchestra	70
Large bass drum	25
Pipe organ	13
Snare drum	12
Cymbals	10
Trombone	6
Piano	0.4
Trumpet	0.3
Bass saxophone	0.3
Bass tuba	0.2
Double bass	0.16
Piccolo	0.08
Flute	0.06
Clarinet	0.05
French horn	0.05
Triangle	0.05

Table 5-2 Peak Power of Musical Sources (*Sivian et al*.)

Frequency Range of Speech and Music

It is instructive to compare the frequency range of various musical instruments with that of speech. This is best done graphically. Figure 5-10 shows the ranges of various musical instruments and voices. It is important to note that this figure only shows the fundamental pitches and not the harmonic overtones. Very low organ notes are perceived mainly by their harmonics. Certain high-frequency noise accompanying musical instruments is not included, such as reed noise in woodwinds, bowing noise of strings, and key clicks and thumps of piano and percussion instruments.

Auditory Area of Speech and Music

The frequency range and dynamic range of speech, music, and other sounds place varying demands on the human ear. Speech uses only a small portion of the ear's auditory capability. The portion of the auditory area used in speech is shown by the shaded area of Fig. 5-11. This area is located centrally in the auditory range; neither extremely soft or extremely loud sounds nor sounds of very low or very high frequency are present in common speech sounds. The speech area of Fig. 5-11 is derived from long time averages, and its boundaries should ideally be represented by gradients to represent the transient excursions in level and frequency. The speech area, as represented, shows an average dynamic range of about 42 dB. The 170- to 4,000-Hz frequency range covers about 4.5 octaves.

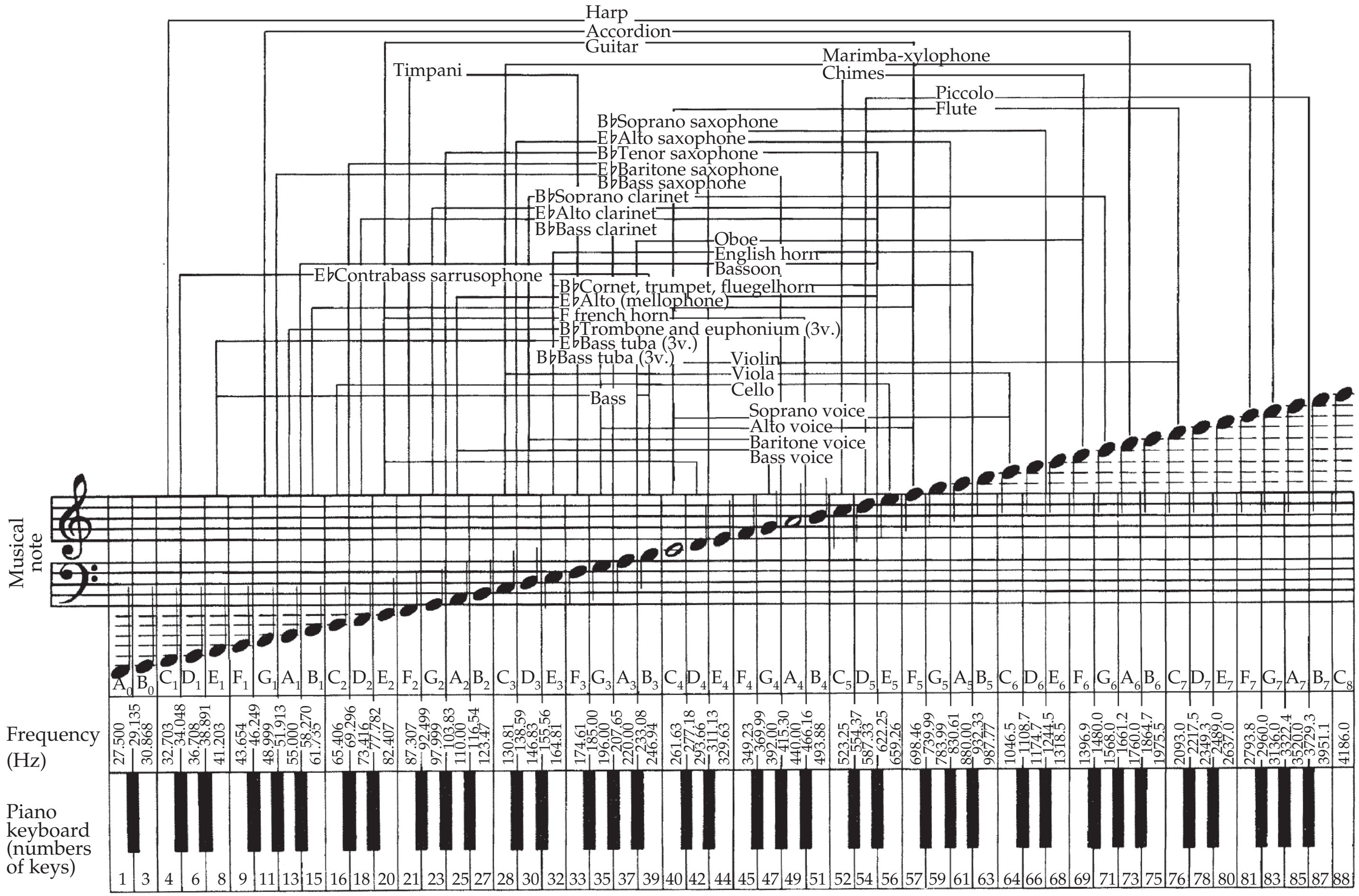

FIGURE 5-10 The audible frequency range of various musical instruments and voices. Only the fundamental tones are included; the partials (not shown) extend much higher. Also not shown are the many high-frequency incidental noises produced. (*C. G. Conn, Ltd.*)

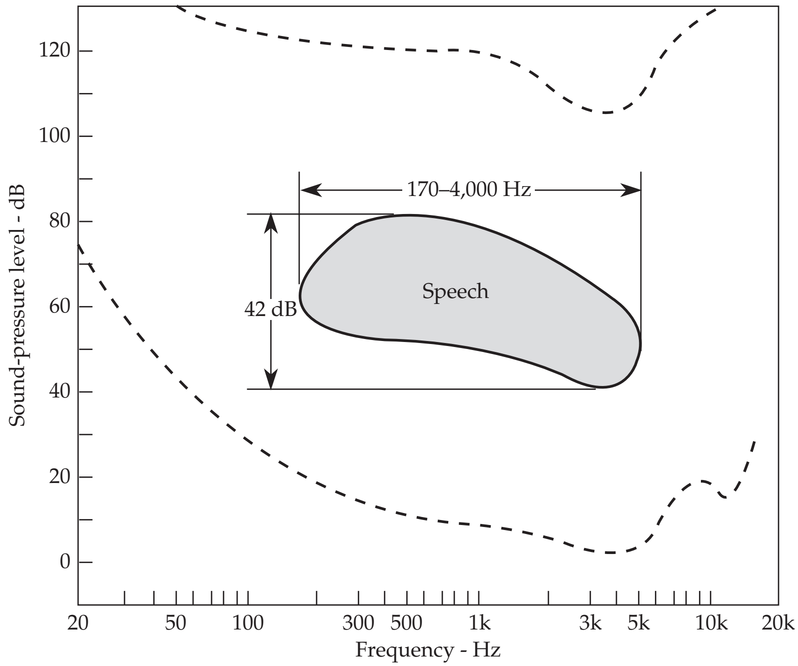

FIGURE 5-11 The portion of the auditory region utilized for speech sounds.

The music area shown in Fig. 5-12 is much greater than the speech area of Fig. 5-11. Music uses a greater proportion of the auditory range of the ear. Its excursions in both level and frequency are correspondingly greater than speech, as would be expected. Here again, long time averages are used to establish the boundaries of the music area, and the boundaries should use gradients to account for extremes. The music area shown is very conservative; it has a dynamic range of about 75 dB and a frequency range of about 50 to 8,500 Hz. This frequency span is about 7.5 octaves, compared to the 10-octave range of the human ear. High-fidelity standards demand a much wider frequency range than this. Without the averaging process involved in establishing the speech and music areas, both the dynamic range and the frequency range would be greater to accommodate the short-term transients that contribute little to the overall average but are still of great importance.

Noise

The word "signal" implies that information is being conveyed. Noise can also be considered an information carrier. For example, interrupting noise to form dots and dashes is one way to shape noise into communication. We will also see how a decaying band of noise can give information on the acoustical quality of a room. On the other hand, there are many types of noise that are undesirable, such as ringing cellphones and busy traffic. Sometimes it is difficult to distinguish between objectionable noise and a legitimate carrier of information. For example, the noise of an automobile conveys considerable

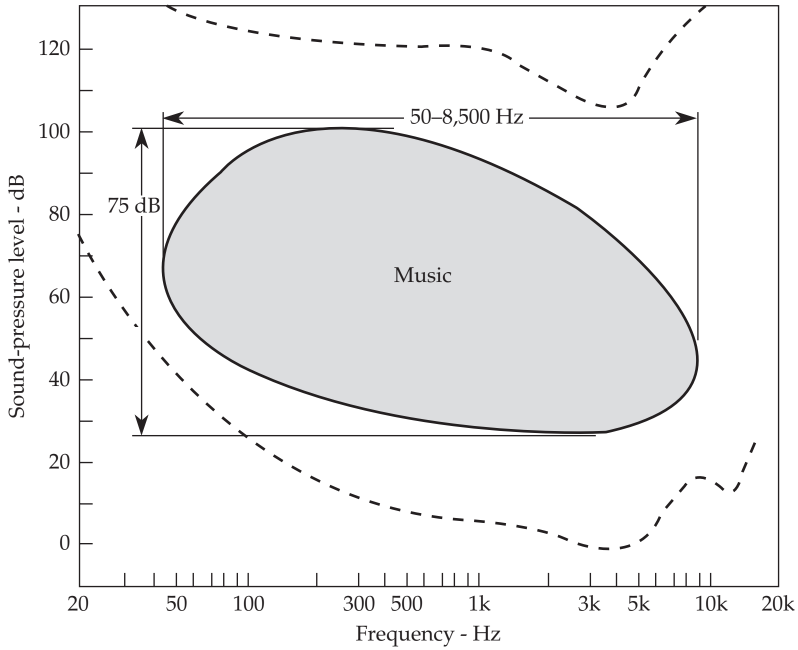

Figure 5-12 The portion of the auditory region utilized for music sounds.

information on how well it is running. An audio playback system can produce sounds deemed very desirable by the owner, but to a neighbor they might be considered intrusive. A loud ambulance or fire truck siren is specifically designed to be both objectionable and to convey an important alert. Society establishes limits to keep objectionable noise to a minimum while ensuring that information-carrying sounds can be heard by those who need to hear them.

Our evaluation of noise is very much a subjective response. Generally, high-frequency noise is more annoying than low-frequency noise. Intermittent noise is more annoying than steady or continuous noise. Moving and nonlocalized noise is more annoying than fixed and localized noise. No matter how it is evaluated, noise intrusion may be a minor annoyance, or it may cause serious consequences such as hearing damage.

Noise Measurements

Defining noise as unwanted sound fits many kinds of noise, but noise is also an important tool for measurements in acoustics. This noise is not necessarily different from unwanted noise; it is just that the noise is put to a beneficial use.

In acoustical measurements, pure tones are often difficult to use, while a narrow band of noise centered on the same frequency can make satisfactory measurements possible. For example, a studio microphone picking up a pure tone signal of 1 kHz from a loudspeaker will have an output that varies greatly from position to position due to room resonances. If, however, a band of noise one octave wide centered at 1 kHz is radiated from the same loudspeaker, the level from position to position would tend to be more uniform, yet the measurement would contain information on what is occurring in the region of 1 kHz. Such measuring techniques make sense because we are usually interested in how a studio or listening room reacts to the complex sounds being recorded or reproduced, rather than to steady, pure tones.

Random Noise

Random noise is generated in any analog electrical circuit and minimizing its effect often poses a difficult problem. Figure 5-13 shows a sine wave and a random noise signal as viewed on an oscilloscope. The regularity of the one is in stark contrast to the randomness of the other. If the horizontal sweep of the oscilloscope is expanded sufficiently and a capture is taken of the random noise signal, it would appear as in Fig. 5-14.

Noise is said to be purely random in character if it has a normal or Gaussian distribution of amplitudes. This means that if we sampled the instantaneous voltage at many equally spaced times, some readings would be positive, some negative, some greater,

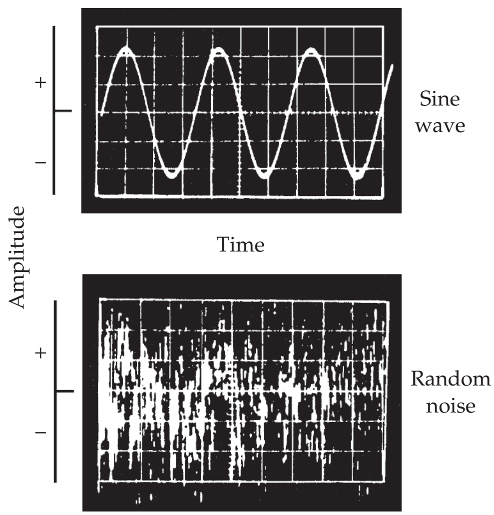

Figure 5-13 Oscillograms of a sine wave and of random noise. Random noise is continually shifting in amplitude, phase, and frequency.

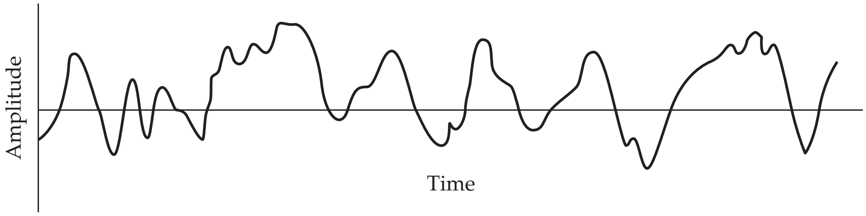

Figure 5-14 A section of the random noise signal of Fig. 5-13 spread out in time. The nonperiodic nature of a noise signal is evident; the fluctuations are random.

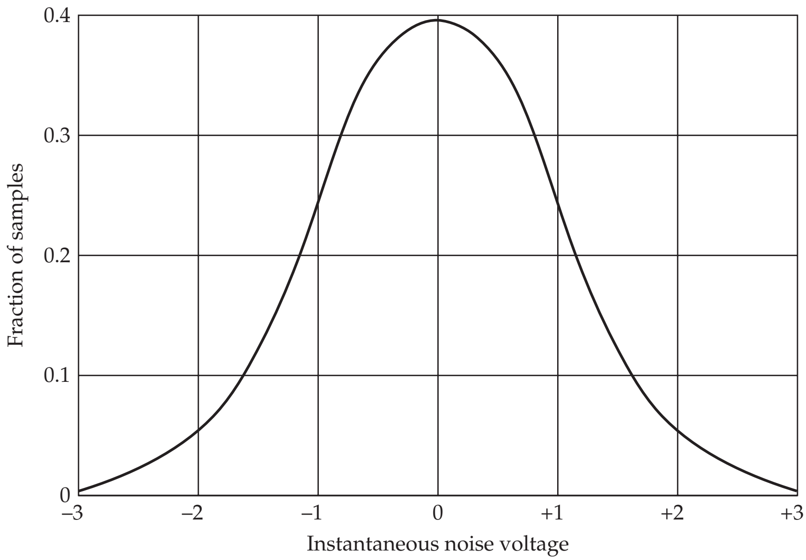

FIGURE 5-15 The proof of randomness of a noise signal lies in the sampling of instantaneous voltage, for example, at 1,000 points equally spaced in time, and plotting the results. If the noise is random, the familiar bell-shaped Gaussian distribution curve results.

some smaller, and a plot of these samples would approach the familiar Gaussian distribution curve of Fig. 5-15.

White and Pink Noise

Both white noise and pink noise are often used as test signals. White noise is analogous to white light in that the energy of both is distributed uniformly throughout the spectrum. In other words, white-weighted random noise has the same average power in each 1-Hz frequency band. It is sometimes said that white noise has equal energy per hertz. Therefore, when plotted on a logarithmic frequency scale, white noise exhibits a flat distribution of energy with frequency, as shown in Fig. 5-16A. Since each higher octave contains twice as many 1-Hz bands as the previous octave, white noise energy doubles in each higher octave. Because of this, white noise is heard as a high-frequency hissing sound.

White light sent through a prism is broken down into a range of colors. The red color is associated with longer wavelengths of light, that is, light in the lower frequency region. Pink-weighted random noise has the same average power in each octave (or 1/3-octave) band. Since successive octaves encompass progressively larger frequency ranges, pink noise has relatively more energy at low frequencies. Because of this, pink noise has a more prominent low-frequency sound than white noise. Pink noise is identified specifically as noise exhibiting more energy in the low-frequency region with a specific downward slope of –3 dB/octave, as shown in Fig. 5-16C. Very generally, pink noise is often used for acoustical measurements, whereas white noise is used for electrical measurements. This is because the energy distribution of pink noise more closely matches the way the ear subjectively hears sound. When using pink noise, a flat response results when constant percentage bandwidth filters are used such as octave

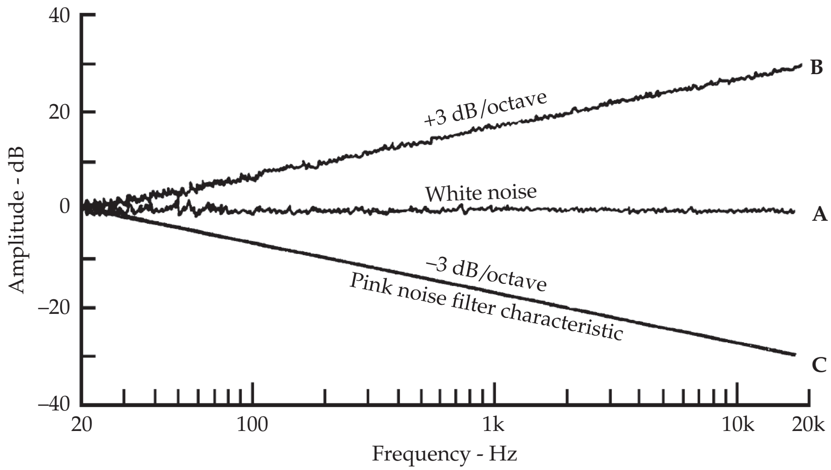

Figure 5-16 Spectrum of noise signals. (A) Random white noise has constant energy per hertz. If the spectrum of random white noise is measured with an analyzer of fixed bandwidth, the resulting spectrum will be flat with frequency. (B) If white noise is measured with an analyzer whose passband width is a given percentage of the frequency to which it is tuned, the spectrum will slope upward at 3 dB/octave. (C) Pink noise is obtained by low-pass filtering white noise with a characteristic that slopes downward at –3 dB/octave.

or 1/3-octave filters. In measuring a system, pink noise is applied to the input and, if the system is flat, the output response will be flat if 1/3-octave filters, for example, are used.

These white and pink terms arose because two types of spectrum analyzers are used. One type is a constant-bandwidth analyzer, which has a passband of fixed width as it is tuned through the spectrum. For example, a bandwidth of 5 Hz may be used. If white noise with its flat spectrum is measured with a constant-bandwidth analyzer, another flat spectrum would result because the fixed bandwidth would measure a constant energy throughout the band (see Fig. 5-16A).

In contrast, in a constant-percentage bandwidth analyzer, the bandwidth changes with frequency. An example of this is the 1/3-octave analyzer, commonly used because its bandwidth follows reasonably well with the critical bandwidth of the human ear throughout the audible frequency range. At 100 Hz the bandwidth of the 1/3-octave analyzer is 23 Hz, but at 10 kHz the bandwidth is 2,300 Hz. Obviously, it intercepts much greater noise energy in a 1/3-octave band centered at 10 kHz than one centered at 100 Hz. Measuring white noise with a constant-percentage analyzer would give an upward-sloping result with a slope of +3 dB/octave (see Fig. 5-16B).

In many audio-frequency measurements, a flat response throughout the frequency range is the desired characteristic of many instruments and rooms. Assume that the system to be measured has a frequency characteristic that is almost flat. If this system is driven with white noise and measured with a constant-percentage analyzer, the result would have an upward slope of +3 dB/octave. It would be more desirable if the measured result was nominally flat so that deviations from flatness would be very apparent. This can be accomplished by using a noise with a downward slope of –3 dB/octave, that is, pink noise. By passing white noise through a low-pass filter, such as that of Fig. 5-17, such a downward sloping noise can be obtained. A close-to-flat system

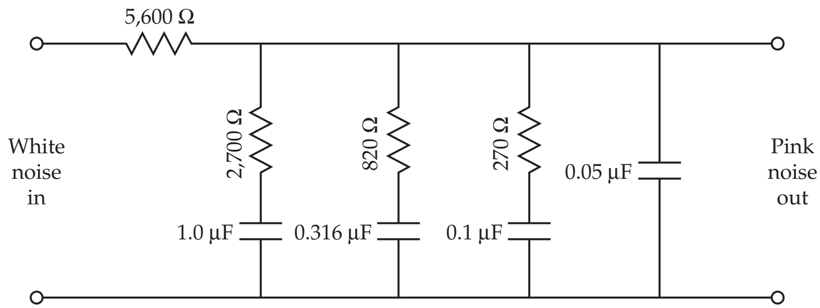

FIGURE 5-17 A simple filter for converting white noise to pink noise. It changes white noise of constant energy per hertz to pink noise of constant energy per octave. Pink noise is most useful in acoustical measurements utilizing analyzers having passbands with bandwidth of a constant percentage of the center frequency.

(such as an amplifier or room) driven with this pink noise would yield a close-to-flat response, which would make deviations from flatness very obvious.

Signal Distortion

Our discussion of the signals encountered in audio is incomplete without acknowledging what can happen to the signal in passing through transducers, amplifiers, and various forms of signal processing gear. Here is a list of some possible forms of distortion:

- **Bandwidth limitation** If the passband of an amplifier attenuates low or high frequencies, the signal output differs from the input by this bandwidth reduction.
- **Nonuniform response** Peaks and valleys within the passband alter the signal wave shape.
- **Phase distortion** Any introduced phase shifts affect the time relationship between signal components.
- **Dynamic distortion** A compressor or expander changes the original dynamic range of a signal.
- **Crossover distortion** In class-B amplifiers, in which the output devices conduct for only half of the cycle, any discontinuities near zero output result in crossover distortion.
- **Nonlinear distortion** If an amplifier is truly linear, there is a one-to-one relationship between input and output. Amplifier feedback helps control nonlinear tendencies. The human ear is not linear. When a pure tone is impressed on the ear, harmonics can be heard. If two loud tones are presented simultaneously, sum and difference tones are generated in the ear itself, and these tones can be heard as can their harmonics. A cross-modulation test on an amplifier does essentially the same thing. If the amplifier (or the ear) were perfectly linear, no sum or difference tones or harmonics would be generated. The production of frequency elements that were not present in the input signal is the result of nonlinear distortion.

- **Transient distortion** Strike a bell and it rings. Apply a transient wavefront signal to an amplifier and its response might ring too. For this reason, signals such as piano notes are difficult to reproduce. Tone burst test signals analyze the transient response characteristics of equipment. Transient intermodulation distortion, slew-induced distortion, and other measuring techniques evaluate transient forms of distortion.

Harmonic Distortion

The harmonic distortion method of evaluating the effects of circuit nonlinearities is universally accepted. In this method the device under test is driven with a sine wave of high purity. If the signal encounters any nonlinearity, the output wave shape is changed; that is, harmonic components appear that were not in the pure sine wave. A spectral analysis of the output signal measures these harmonic distortion products.

For example, a wave analyzer may use a constant passband width of 5 Hz, which is swept through the audio spectrum. Figure 5-18 shows results of such a measurement. The wave analyzer is first tuned to the fundamental, $f_0 = 1$ kHz, and the level is set for a convenient 1.00 V. The wave analyzer shows that the $2f_0$ second harmonic at 2 kHz measures 0.10 V. The $3f_0$ third harmonic at 3 kHz gives a reading of 0.30 V, the fourth a reading of 0.05 V, and so on. The data are assembled in Table 5-3.

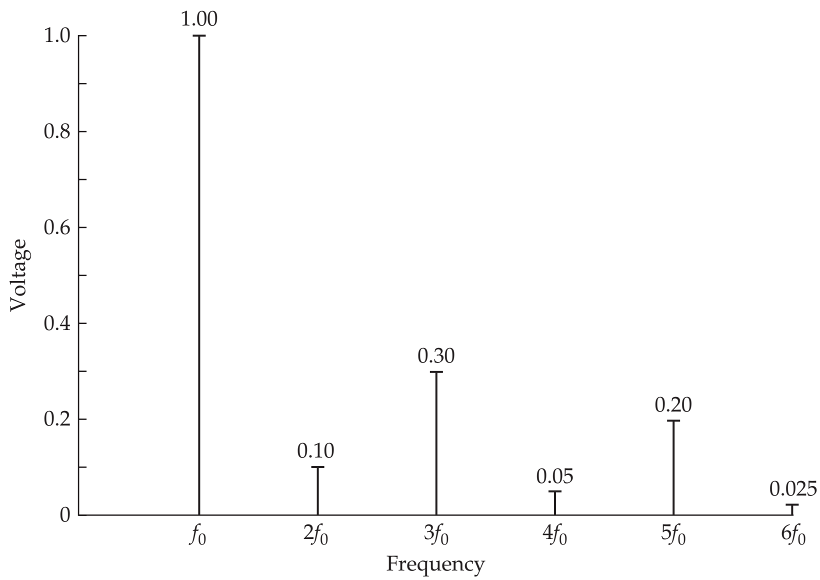

FIGURE 5-18 A distorted periodic wave is measured with a constant-bandwidth analyzer. The fundamental f_0 is set for some reference voltage, taken here as 1.00 V. Using a wave analyzer, the amplitude of the second harmonic at $2f_0$ is measured as 0.10 V. The wave analyzer similarly yields the amplitudes of each harmonic as shown. The root-mean-square (RMS) of the harmonic voltages is then compared to the 1.00-V fundamental to find the total harmonic distortion (THD) expressed in percentage.

Harmonic	Volts	(Volts)^2
Second harmonic $2f_0$	0.10	0.01
Third harmonic $3f_0$	0.30	0.09
Fourth harmonic $4f_0$	0.05	0.0025
Fifth harmonic $5f_0$	0.20	0.04
Sixth harmonic $6f_0$	0.025	0.000625
Seventh and higher	(negligible)	—
		Sum 0.143125

TABLE 5-3 Harmonic Distortion Products (Fundamental f_0 = 1 kHz, 1.00 V amplitude)

The total harmonic distortion (THD) may be found from the expression:

$$\mathrm{THD} = \frac{\sqrt{(e_2)^2 + (e_3)^2 + (e_4)^2 \ldots (e_n)^2}}{e_0} \times 100 \tag{5-1}$$

where $e_2, e_3, e_4, \ldots, e_n$ = voltages of second, third, fourth, etc., harmonics
e_0 = voltage of fundamental

In Table 5-3, the harmonic voltages have been squared and added together. Using the equation:

$$\mathrm{THD} = \frac{\sqrt{0.143125}}{1.00} \times 100$$
$$= 37.8\%$$

A THD of 37.8% is a very high distortion that would make any amplifier sound poor on any type of signal, but the example has served our purpose.

A simple adaptation of the THD method can also be used. Consider Fig. 5-18 again. If the f_0 fundamental were adjusted to some known value and then a notch filter were adjusted to f_0 and essentially eliminated it, only the harmonics would be left. Measuring these harmonics together with a root-mean-square (RMS) meter accomplishes what was done in the square root portion of Eq. (5-1). Comparing this RMS measured value of the harmonic components with that of the fundamental and expressing it as a percentage would give the THD.

An undistorted sine wave is sent through an amplifier, which clips positive peaks, as shown in Fig. 5-19. On the left, the flattening of the positive peaks with 5% THD is evident, and shown below that is the combined total of all the harmonic products with the fundamental rejected. On the right is shown the effect of greater clipping to yield 10% THD. Figure 5-20 shows a sine wave passing through the amplifier and symmetrically clipped on both positive and negative peaks. The combined distortion products for symmetrical clipping have a somewhat different appearance, but they measure the same: 5% and 10% THD.

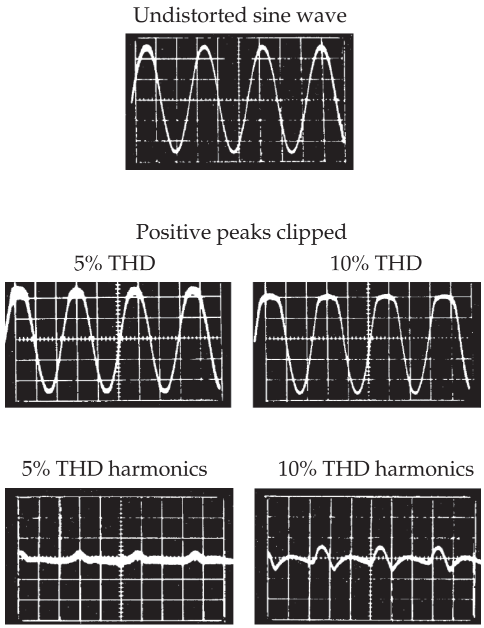

Figure 5-19 Oscillograms show an undistorted sine wave, which is applied to the input of an amplifier which clips the positive peaks of the signal. The appearance of the clipped sine wave is shown for 5% and 10% THD. If the fundamental is rejected by a notch filter, the summed harmonics appear as shown.

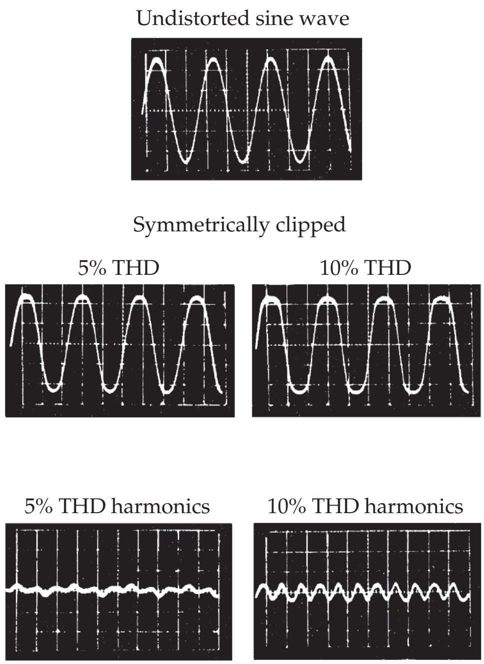

Figure 5-20 Oscillograms show an undistorted sine wave, which is applied to the input of an amplifier which clips both the positive and negative peaks in a symmetrical fashion. The appearance of the clipped sine wave is shown for 5% and 10% THD. The appearance of the harmonics alone, with the fundamental filtered out, is also shown.

Consumer power amplifiers commonly have specifications listing THD nearer 0.05% rather than 5% or 10%. In a series of double-blind subjective tests, Clark found that 3% distortion was audible on different types of sounds. With carefully selected material (such as a flute solo), detecting distortions down to 1% or 2% might be possible. A distortion of 1% with sine waves is readily audible.

Resonance

The amplitude of vibration of any resonant system is maximum at the natural or resonant frequency f_0 and is less at frequencies below and above that frequency. A modest excitation signal at that resonant frequency will result in high amplitude. As shown in Fig. 5-21, the amplitude of vibration changes as the frequency of excitation is varied, going through a peak response at the frequency of resonance. Perhaps the simplest example of a resonant system is a weight on a spring.

Such resonance effects appear in a wide variety of systems: the interaction of mass and stiffness of a mechanical system such as a tuning fork, or the acoustical resonance of the air in a bottle, as the mass of the air in the neck of the bottle reacts with the springiness of the air entrapped in the body of the bottle. Resonance is particularly important in architectural acoustics. Most rooms can be considered as enclosed spaces; they are essentially containers of air. As such, they exhibit modal resonances. As we shall see in later chapters, the resonant frequencies are a function of the interior dimensions of the room.

Resonance effects are also present in electronic circuits as the inertia effect of an inductance reacts with the storage effect of a capacitance. An inductor (its electrical symbol is L) is commonly a coil of wire, and a capacitor (C) is made of sheets of conducting material separated by nonconducting sheets. Energy can be stored in the magnetic field of an inductor as well as in the electrical charges on the plates of a capacitor. The interchange of energy between two such storage systems can yield a resonance effect.

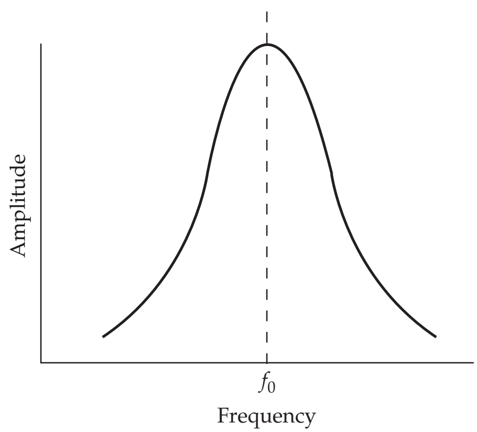

FIGURE 5-21 The amplitude of vibration of any resonant system is maximum at the natural or resonant frequency f_0 and is less at frequencies below and above that frequency.

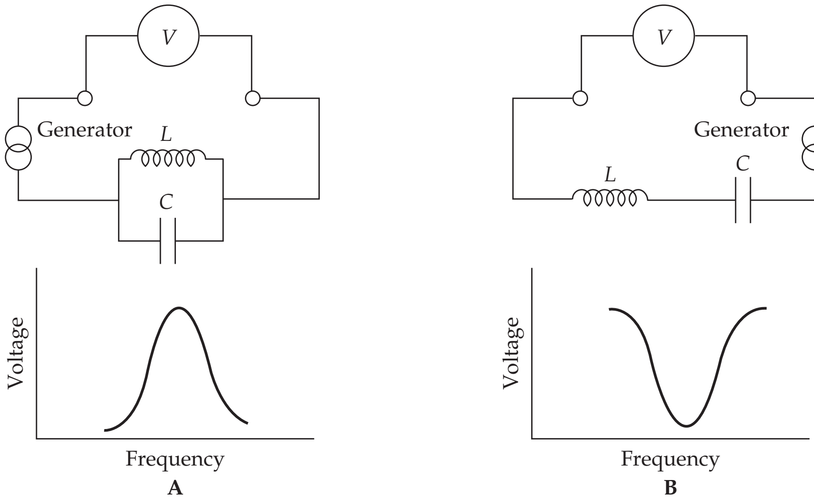

FIGURE 5-22 A comparison of (A) parallel resonance and (B) series resonance. For a constant alternating current flowing, the voltage across the parallel resonant circuit peaks at the resonance frequency, while that of the series resonant circuit is a minimum.

Figure 5-22 shows two circuits in which an inductor and a capacitor can exhibit resonance. Let us assume that an alternating current of constant amplitude but varying frequency is flowing in a parallel resonant circuit (see Fig. 5-22A). As the frequency is varied, the voltage at the terminals reaches a maximum at the natural frequency of the *LC* system, falling off at lower and higher frequencies. In this way the typical resonance curve shape is developed. Another way of saying this is that the parallel resonant circuit exhibits maximum impedance (opposition to the flow of current) at resonance.

A series resonant circuit (see Fig. 5-22B) also uses an inductor *L* and a capacitor *C*. As the alternating current of constant magnitude and varying frequency flows in the circuit, the voltage at the terminals describes an inverted resonance curve in which the voltage is minimum at the natural frequency and rises at both lower and higher frequencies. It can also be said that the series resonant circuit presents minimum impedance at the frequency of resonance.

Audio Filters

Filters are used in many applications including audio equalizers and loudspeaker crossovers. In analog filter design, by choosing the values of the resistors, inductors, and capacitors, almost any kind of frequency and impedance matching characteristic can be obtained. Common forms of filters include the low-pass, high-pass, band-pass, and band-reject. The characteristic frequency responses of these filters are shown in Fig. 5-23. Figure 5-24 shows how inductors and capacitors may be arranged in various passive circuits to form simple high- and low-pass filters. The filters shown in Fig. 5-24C will have much sharper cutoffs than the simpler ones in Figs. 5-24A and B. There are many other highly specialized filters with specific features. With such filters, a wideband signal such as speech or music can be altered at will.

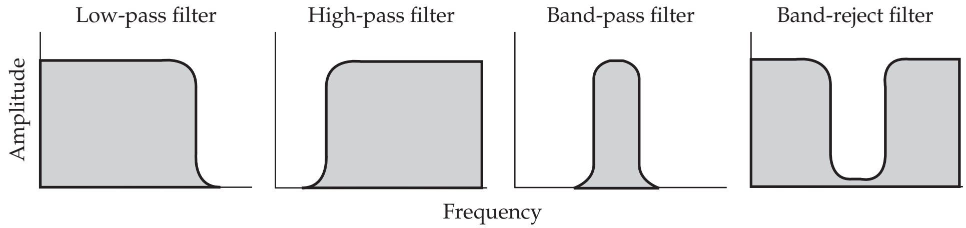

FIGURE 5-23 Frequency response characteristics for low-pass, high-pass, band-pass, and band-reject filters.

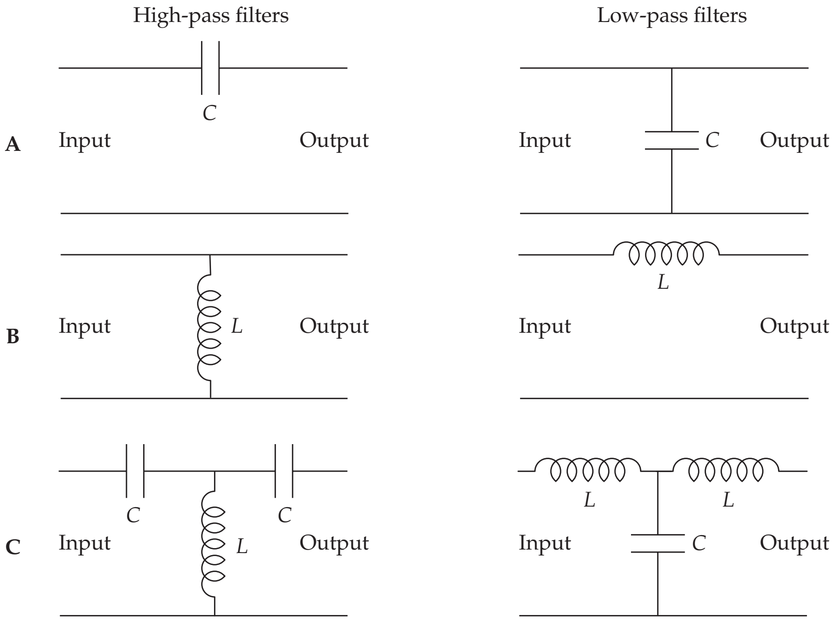

FIGURE 5-24 Inductors and capacitors can be used to form passive high- and low-pass filters. (A) Filters using capacitors. (B) Filters using inductors. (C) Filters using both capacitors and inductors; these have a sharper cutoff than those in (A) or (B).

Adjustable filters can be shifted to any frequency within their design band. One type is the constant bandwidth filter that offers the same bandwidth at any frequency. For example, a spectrum analyzer may have a 5-Hz bandwidth, whether it is tuned to 100 Hz or 10 kHz, or any other frequency within its operating band. Another type of adjustable filter offers a pass bandwidth that is a constant percentage of the frequency to which it is tuned. The 1/3-octave filter is such a device. If it is tuned to 125 Hz, the 1/3-octave bandwidth is 112 to 141 Hz. If it is tuned to 8 kHz, the 1/3-octave bandwidth is 7,079 to 8,913 Hz. In either case, the bandwidth is about 23% of the frequency to which it is tuned.

Passive filters do not require any power source. Active filters depend on powered electronics such as discrete transistors or integrated circuits for their operation. A passive low-pass filter assembled from inductors and capacitors is shown in Fig. 5-25A. An active

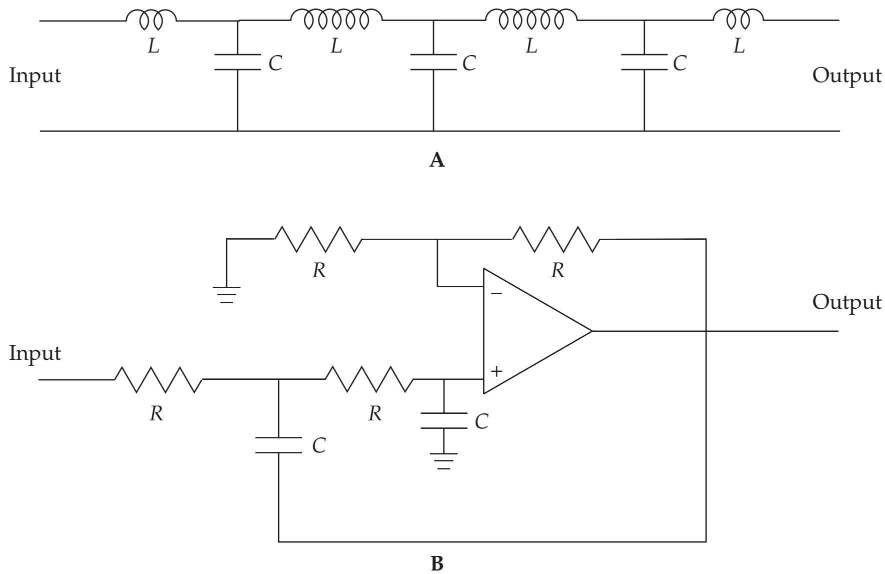

Figure 5-25 Two analog low-pass filters are shown. (A) A passive analog filter. (B) An active analog filter utilizing an op-amp integrated circuit.

low-pass filter based on an operational amplifier integrated circuit is shown in Fig. 5-25B. Both passive and active filters are widely used; both offer advantages depending on the application.

Filters can be constructed in analog or digital form. All the filters discussed to this point have been of the analog type; they operate on a continuous analog signal. Digital filters perform numerical operations on digital audio signals sampled in discrete time. In many cases, digital filters are implemented as software programs running on microprocessors; analog-to-digital converters and digital-to-analog converters are used to transfer analog audio signals through the digital filter. An example of a digital filter is shown in Fig. 5-26. This example is a finite impulse response (FIR) filter, sometimes

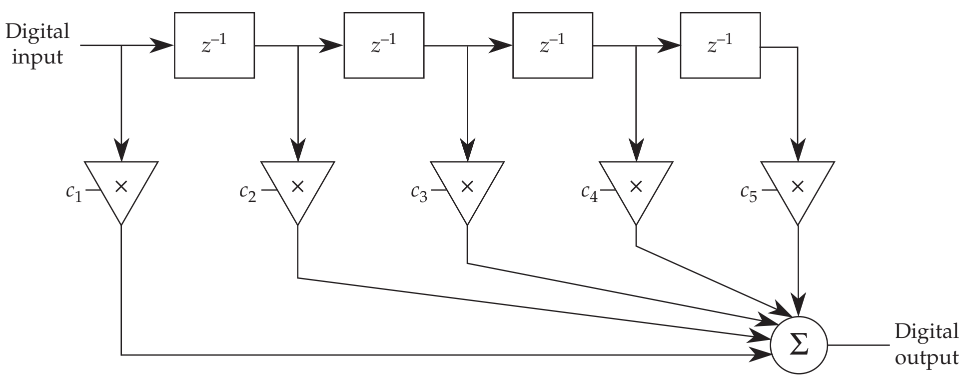

Figure 5-26 A finite impulse response (FIR) filter showing delay, multiplication, and summing to implement signal filtering.

called a transversal filter. Digital samples are input to the filter and applied to the top section of z^{-1} blocks comprising a tapped delay line; the middle section shows multiplication by filter coefficients; these outputs are summed to produce the filter's output samples. In this way, the signal's frequency response can be altered.

Digital filters are part of a larger technology known as digital signal processing (DSP). DSP is widely used throughout the audio industry in many different applications ranging from the processing of music signals for playback to signal analysis of room acoustics. For example, DSP can be applied to loudspeaker-room-listener problems. A microphone placed at the listening position measures the frequency and phase response of a loudspeaker's output in a listening room, and from that data, inverse equalization can be created to compensate for anomalies caused by the loudspeaker and the room acoustics.

Key Points

- Speech, like music, is highly variable and transient in nature, comprising energy moving through the three-dimensional scales of frequency, sound level, and time.
- The vocal tract can be considered as an acoustically resonant system. The dimensions help determine the resonances of the vocal tract and their effect on speech sounds.
- The spectral distribution of energy in the voice helps determine the intelligibility of speech with respect to the spectral distribution of reverberation and noise.
- Musical sounds are extremely variable in their complexity, ranging from near sine-wave simplicity to highly intricate tonalities. The dynamic range of music may be 100 dB (the ear's dynamic range is about 120 dB).
- Musical instruments can produce relatively high levels of acoustic power. For example, a full symphony orchestra's peak power might be 70 W.
- Many kinds of noise are undesirable and much attention is paid to reducing noise levels; however, noise is also an important tool for acoustical measurements. In many cases, a narrow band of noise centered on some frequency is used.
- White noise and pink noise are often used as test signals. White-weighted random noise has the same average power in each 1-Hz frequency band. Pink-weighted random noise has the same average power in each octave (or 1/3-octave) band. Since successive octaves encompass progressively larger frequency ranges, pink noise has relatively more energy at low frequencies.
- The amplitude of vibration of any resonant system is maximum at the natural or resonant frequency and is less at frequencies below and above that frequency. A modest excitation signal at that resonant frequency can result in high amplitude.
- Most rooms can be considered as enclosed spaces; they are essentially containers of air. As such, they exhibit modal resonances.
- Filters are used in many applications including audio equalizers and loudspeaker crossovers. Common forms of filters include the low-pass, high-pass, band-pass, and band-reject.

CHAPTER 6
Reflection

Imagine a sound source in a free field, or an approximation of a free field, such as an open meadow. Sound emanates from the source radially in all directions. Direct sound from the source moves past you and never returns. Now consider the same source in a room. Direct sound moves past, but when it strikes room boundaries, it reflects from the boundaries. Thus, a sound moves past you once in a direct path, and many additional times as many reflected paths, until it dies out. Sound comprising many reflections sounds dramatically different than sound in a free field. The reflections convey significant information about the room's size, shape, and boundary composition. Reflections help define the sonic characteristics of a room. Not insignificantly, reflections can add pleasing qualities to a sound. Conversely, reflections can also degrade sound quality. The success or failure of any room of acoustical importance largely depends on the characteristics of the sound reflections as dictated by the room's boundary conditions.

Specular Reflections

The essential mechanism of reflection from a flat surface is simple. Figure 6-1 shows the reflection of point-source sound waves from a rigid, plane wall surface. The spherical wavefronts (solid lines) strike the wall and the reflected wavefronts (broken lines) are returned toward the source. This is called a specular reflection and behaves the same as light reflecting from a mirror, described by Snell's law.

Sound follows the same rule as light: The angle of incidence is equal to the angle of reflection, as shown in Fig. 6-2. Geometry reveals that the angle of incidence θ_i equals the angle of reflection θ_r. Moreover, just as an image in a mirror, the reflected sound acts as though it originated from a virtual sound image. The virtual source is acoustically located behind the reflecting surface, just as viewing an image in a mirror. This image source is located the same distance behind the wall as the real source is in front of the wall. This is the simple case of a single reflecting surface.

When sound strikes more than one surface, multiple reflections will be created. For example, images of the images will exist. Consider the example of two parallel walls as shown in Fig. 6-3. Sound from the source will strike the left wall; this can be modeled as a virtual source located at I_L (a first-order image). Similarly, there is a virtual source located at I_R. Sound will continue to reflect back and forth between the parallel walls. For example, sound will strike the left wall, right wall, and left wall again, sounding as if there is a virtual source at location I_{LRL} (a third-order image). We observe that in this example, the walls are 15 distance units apart; thus the first-order images are 30 units apart, the second-order images are 60 units apart, the third-order images

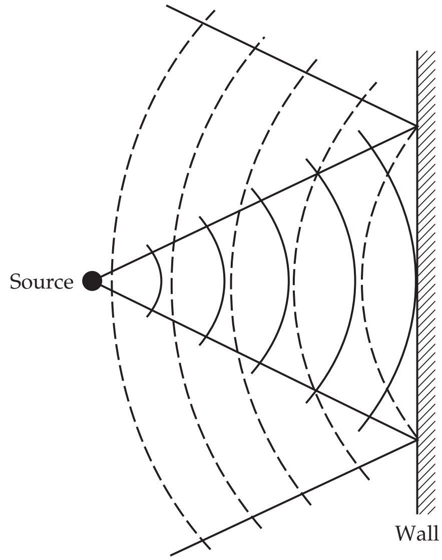

Figure 6-1 Reflection of a point-source sound from a flat surface (incident sound, solid lines; reflected sound, broken lines).

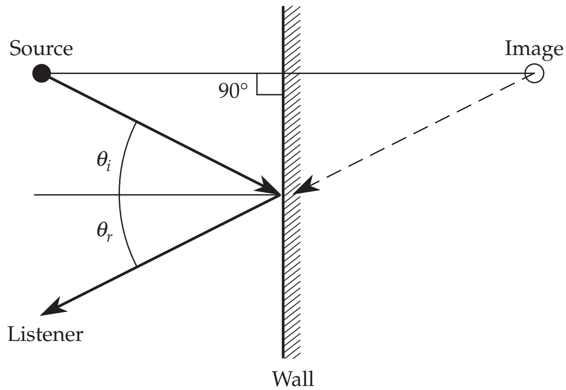

Figure 6-2 In a specular reflection, the angle of incidence (θ_i) is equal to the angle of reflection (θ_r). The reflected sound appears to be from a virtual image source.

are 90 units apart, and so on. Using this modeling technique, we can ignore the walls themselves, and consider sound as coming from many virtual sources spaced away from the actual source, arriving with time delays based on their distance from the source. For example, in a room with a single loudspeaker, the resultant sound at a listening position would be equivalent to the sound heard from the single loudspeaker, and from other more distant loudspeakers.

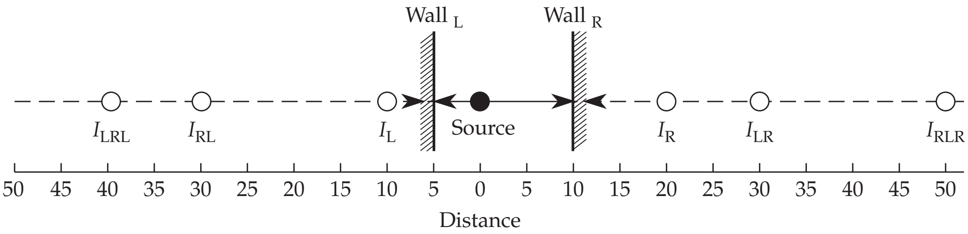

Figure 6-3 When sound strikes more than one surface, multiple reflections will be created. These can be viewed as virtual sound sources. Parallel walls can pose acoustical problems such as flutter echoes; these reflections are highly audible because of the regularity of the echo.

In a rectangular room, there are six surfaces and the source has an image in all six surfaces, sending energy back to the receiver, resulting in a highly complex sound field. In computing the total sound intensity at a given receiving point, the contributions of all these images must be taken into consideration.

Flutter Echoes

Returning again to Fig. 6-3, we note that parallel walls such as these, or other parallel surfaces, present an acoustical problem. If the distance between surfaces is large enough so the time between reflections is outside the Haas fusion zone, a flutter echo is created as sound bounces back and forth from one surface to the other. Because of the regularity of these reflections, the ear is very sensitive to the effect. In fact, even if the time delays are otherwise in the fusion zone, the effect may still be audible as a recurring echo. This echo can be very prominent in an otherwise diffuse sound field and is highly undesirable. In theory, with perfectly reflective surfaces, there would be an infinite number of images. The acoustical effect is the same as being between two mirrors and seeing a series of images. In practice, successive images are attenuated because of absorption or diffusion at the surfaces. Where possible, parallel walls or any parallel surfaces should be avoided in room design, and when unavoidable, they should be covered by absorbing or diffusing material. Splaying walls of other surfaces by a small amount of perhaps 5° or 10° can also avoid flutter echoes.

When sound strikes a boundary surface, some sound energy is transmitted or absorbed by the surface and some is reflected. The reflected energy is always less than the incident energy. Surfaces that are made of heavy materials (measured by surface weight) are usually more reflective than lighter materials that tend to absorb or transmit sound. Sound may undergo many reflections as it travels around a room. The energy lost at each reflection results in the eventual demise of that sound.

Reflection depends partly on the size of the reflecting object. Sound is reflected from objects that are large compared to the wavelength of the impinging sound. Generally speaking, sound will be reflected from a rectangular panel if each of its two dimensions is 5 times the wavelength of sound. Thus, objects act as frequency-dependent reflectors. A printed copy of this book would be a good reflector for 10-kHz sound (wavelength about an inch). While facing a sound source, moving the book in front of and away from your face would result in significant differences in high-frequency response because of acoustical shadowing. At the low end of the audible spectrum, 20-Hz sound (wavelength about 56 ft) would sweep past the book and the person holding it as though they did not exist, and without appreciable shadows.

Doubling of Pressure at Reflection

The sound pressure on a surface normal to an incident wave is equal to the energy density of the radiation in front of the surface. If the surface is a perfect absorber, the pressure equals the energy density of the incident radiation. If the surface is a perfect reflector, the pressure equals the energy density of both the incident and the reflected radiation. Thus the pressure at the face of a perfectly reflecting surface is twice that of a perfectly absorbing surface. In the study of standing waves, this pressure doubling takes on great significance as described later.

Reflections from Convex Surfaces

In a simplified view, sound can be considered as rays. Each ray should be considered as a beam of diverging sound with a spherical wavefront to which the inverse square law applies. Spherical wavefronts from a point source become plane waves at greater distance from the source. For this reason, impinging sound on various surfaces can usually be thought of as plane wavefronts. Reflection of plane wavefronts of sound from a solid convex surface scatters the sound energy in many directions, as shown in Fig. 6-4. The size of the irregularity must be large compared to the wavelength of the sound. This reflecting sound returns and diffuses the impinging sound. Polycylindrical sound-absorbing modules both absorb sound and contribute diffusion in the room. The latter is created by reflection from the cylindrically shaped convex surface of the modules.

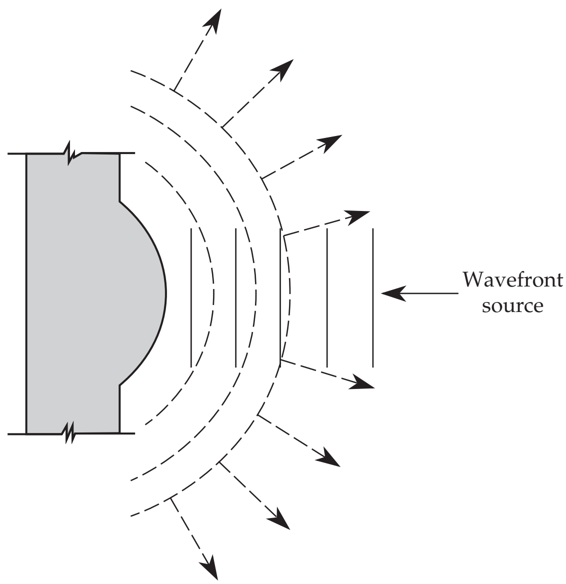

FIGURE 6-4 Plane sound waves impinging on a convex irregularity tend to be dispersed through a wide angle if the size of the irregularity is large compared to the wavelength of the sound.

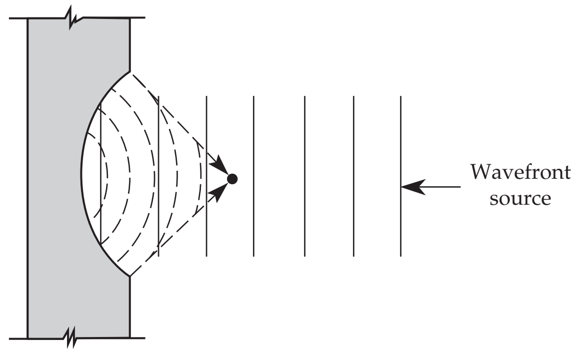

Figure 6-5 Plane sound waves impinging on a concave irregularity tend to be focused if the size of the irregularity is large compared to the wavelength of the sound.

Reflections from Concave Surfaces

Plane wavefronts striking a concave surface tend to focus to a point as illustrated in Fig. 6-5. The precision with which sound is focused is determined by the shape and relative size of the concave surface. Spherical concave surfaces are common. For example, they may be used to make a microphone highly directional by placing it at the focal point. Such microphones are frequently used to pick up field sounds at sporting events or record animal sounds in nature. The effectiveness of such concave reflectors depends on the size of the reflector with respect to the wavelength of sound. For example, a 3-ft-diameter spherical reflector will give good pickup directivity at 1 kHz (wavelength about 1 ft), but it is practically nondirectional at 200 Hz (wavelength about 5.5 ft). Concave architectural surfaces, for example, domed ceilings and archways in churches or auditoriums, can be the source of serious problems as they produce concentrations of sound in direct opposition to the goal of uniform distribution of sound in rooms.

Reflections from Parabolic Surfaces

A parabola generated by the equation $y = x^2$ has the characteristic of focusing sound precisely to a point. A very "deep" parabolic surface, such as that of Fig. 6-6, exhibits far greater directional properties than a shallow one. Again, the directional properties depend on the size of the opening in terms of wavelengths.

Plane waves striking such a reflector are brought to a focus at the focal point. Conversely, sound emitted at the focal point of the parabolic reflector generates plane wavefronts. For example, the parabola of Fig. 6-6 is used as a directional sound source with a small, ultrasonic Galton whistle pointed inward at the focal point. Standing waves are produced by reflections from a heavy glass plate. In a Galton whistle, the force exerted by the vibration of the air particles on either side of a node is sufficient to hold slivers of cork in levitation.

Whispering Galleries

St. Paul's Cathedral in London, St. Peter's Basilica in Vatican City, the Temple of Heaven in Beijing, Statuary Hall in the U.S. Capitol Building, Grand Central Station in New York City (in front of the Oyster Bar & Restaurant), and other structures contain

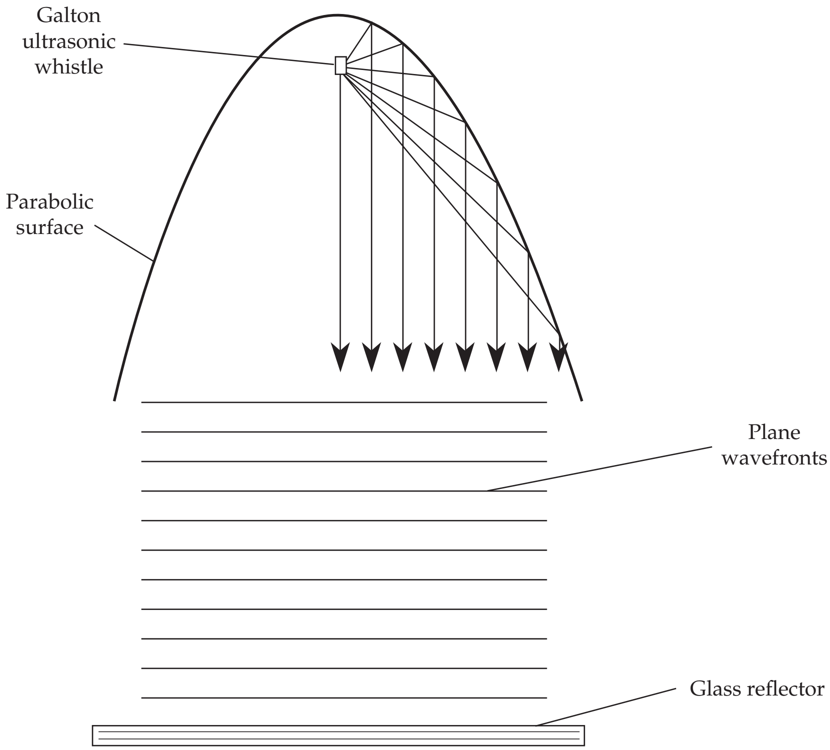

Figure 6-6 A parabolic surface can focus sound precisely at a focal point or, the converse, a sound source placed at the focal point can produce plane, parallel wavefronts. In this case, the source is an ultrasonic Galton whistle driven by compressed air.

a whispering gallery; the acoustical mechanism is diagrammed in Fig. 6-7. The source whisperer and the receiver may be separated from each other by a great distance; both are located at the focal points of hard-surfaced, paraboloid-shaped walls, and face the walls. At the source, a whisper directed tangentially to the surface is clearly heard on the receiver side. The phenomenon is assisted by the fact that the

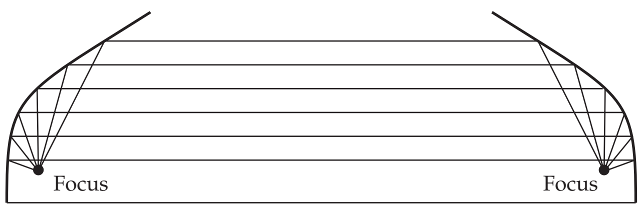

Figure 6-7 Graphic example of a whispering gallery showing symmetrical sound focusing points. A whisper directed tangentially to the paraboloid-shaped surface is readily heard by the receiver on the far side of the room. More generally, concave surfaces pose acoustical problems.

walls are paraboloid-shaped. This means that upward-directed components of the whispered sounds tend to be reflected downward and are conserved and transmitted rather than lost above. Although acoustically interesting, architectural configurations such as this are usually undesirable. Except for unique applications, concave surfaces such as sections of cylinders, spheres, paraboloids, and ellipsoids should not be used in acoustically important architecture. The sound focusing effects are exactly counter to the usual goal of providing uniform, diffuse sound throughout a space.

Standing Waves

The concept of standing waves directly depends on the reflection of sound. Assume two flat, solid parallel walls separated a given distance (as in Fig. 6-3). A sound source between them radiates sound of a specific frequency. As we observed, the wavefront striking the right wall is reflected back toward the source, striking the left wall where it is again reflected back toward the right wall, and so on. One wave travels to the right, the other toward the left. The two traveling waves interact to form a standing wave. Only the standing wave, the interaction of the two, is stationary. The frequency of the radiated sound establishes this resonant condition between the wavelength of the sound and the distance between the two surfaces. This phenomenon is entirely dependent on the reflection of sound at the two parallel surfaces. As discussed in other chapters, standing waves require careful design scrutiny, particularly in terms of a room's low-frequency response.

Corner Reflectors

We generally think of reflections as normal (perpendicular) reflections from surrounding walls, but reflections also occur at the corners of a room. Moreover, the reflections follow the source around the room. The corner reflector of Fig. 6-8 receives sound from

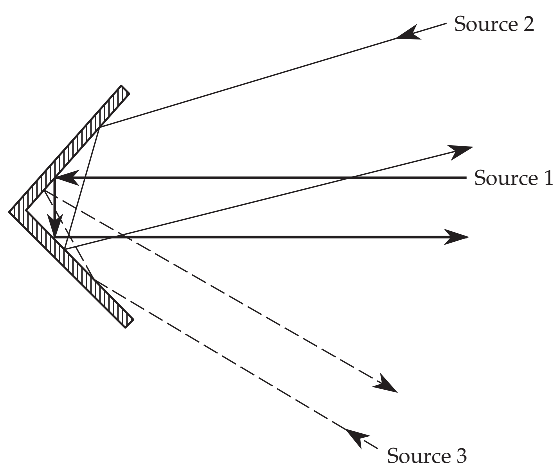

Figure 6-8 A corner reflector has the property of reflecting sound back toward the source from any direction.

the source at location 1, reflects the sound from two surfaces, and sends a reflection directly back toward that source. If the angles of incidence and reflection are carefully noted, a sound at source location 2 will also send a direct, double-surface reflection back to that source. Similarly, a source at location 3 on the opposite side of the normal is subject to the same effect. A corner reflector thus has the property of reflecting sound back toward the source from any direction. Corner reflections suffer losses at two surfaces, tending to make them somewhat less intense than normal reflections at the same distance.

The corner reflector of Fig. 6-8 involves only two surfaces. The same follow-the-source principle applies to the four upper tri-corners of a room formed by ceiling and walls and another four formed by floor and wall surfaces. Following the same principles, sonar and radar systems have long employed targets made of three circular plates of reflecting material assembled so that each is perpendicular to the others.

Mean Free Path

The average (mean) distance sound travels between successive reflections is called the mean free path. This distance is given by the expression:

$$MFP = \frac{4V}{S} \tag{6-1}$$

where MFP = mean free path, ft or m
V = volume of the space, ft^3 or m^3
S = surface area of the space, ft^2 or m^2

For example, in a room measuring 25 × 20 × 10 ft, on average a sound travels a distance of 10.5 ft between reflections. Sound travels 1.13 ft/msec. At that speed it takes 9.3 msec to traverse the mean-free distance of 10.5 ft. Viewed another way, about 107 reflections take place in the space of a second.

Figure 6-9 shows echograms of reflections occurring during the first 0.18 sec in a recording studio having a volume of 16,000 ft^3 and a reverberation time of 0.51 sec at 500 Hz. The microphone was successively placed in four different locations in the room. The impulsive sound source was in a fixed position. The sound source was a pistol that punctured a paper with a blast of air, giving an intense pulse of sound of less than 1 msec in duration. The reflection patterns at the four positions show differences; in each, scores of individual reflections are clearly resolved. These echograms define the transient sound field of the room during the first 0.18 sec as contrasted to the steady-state condition. These early reflections play a significant role in the perception of room acoustics.

Perception of Sound Reflections

When reproducing recorded sound in a listening room, during the enjoyment of live music in a concert hall, or in any activity in any acoustical space, the sound falling on the ears of the listener is very much affected by reflections from the surfaces of the room. Our perception of these reflections is an important manifestation of sound reflection.

The Effect of Single Reflections

Research studies on the audibility of simulated reflections often use an arrangement of loudspeakers shown in Fig. 6-10; it is similar to a traditional stereophonic playback

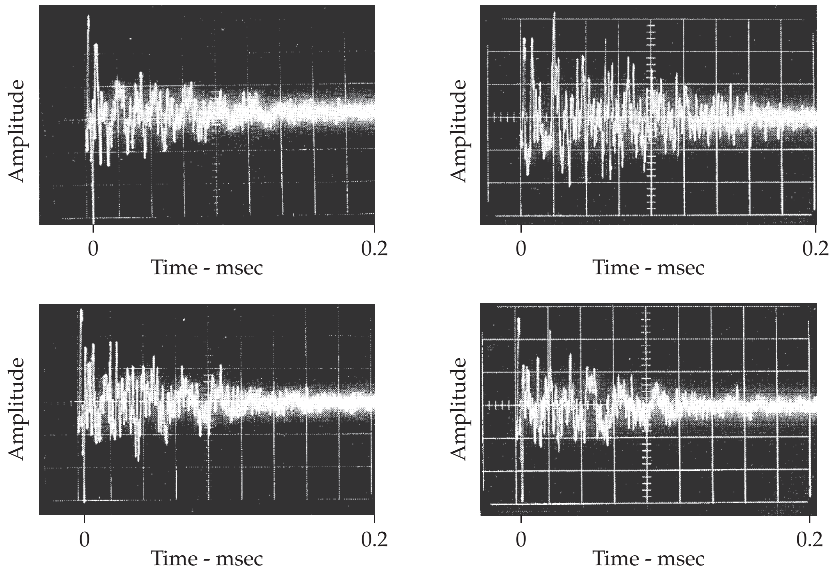

Figure 6-9 Individual reflections are resolved in these echograms taken at four different positions in a studio of 16,000 ft^3 volume and having a reverberation time of 0.51 sec. The horizontal time scale is 20 msec/div.

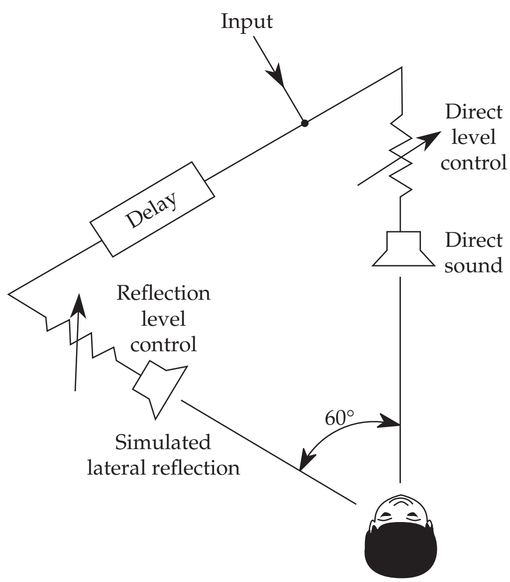

Figure 6-10 Typical equipment configuration used by investigators to study the audible effect of a simulated lateral reflection on the direct signal. Reflection level and delay (with respect to the direct sound) are the variables under control.

configuration. The observer is seated at the apex so that lines drawn to the two loudspeakers are approximately 60° apart (this angle varies with the investigator). A monaural signal is fed to both loudspeakers. The signal to one loudspeaker represents the direct sound. The signal to the other loudspeaker can be delayed by any amount; it represents a lateral reflection. The two variables under study are the level of the lateral reflection compared to that of the direct sound, and the time delay of the reflection with respect to the direct sound.

Olive and Toole investigated listening conditions in small rooms, such as recording studio control rooms and home listening rooms. In one experiment, they investigated the effect of simulated lateral reflections in an anechoic environment with speech as the test signal. The work is summarized in Fig. 6-11. This graph plots reflection level against reflection delay, the two variables specified above. A reflection level of 0 dB means that the reflection is the same level as the direct sound. A reflection level of −10 dB means that the reflection level is 10 dB below the direct sound. In all cases, reflection delay is specified in milliseconds, the reflection lagging the direct sound.

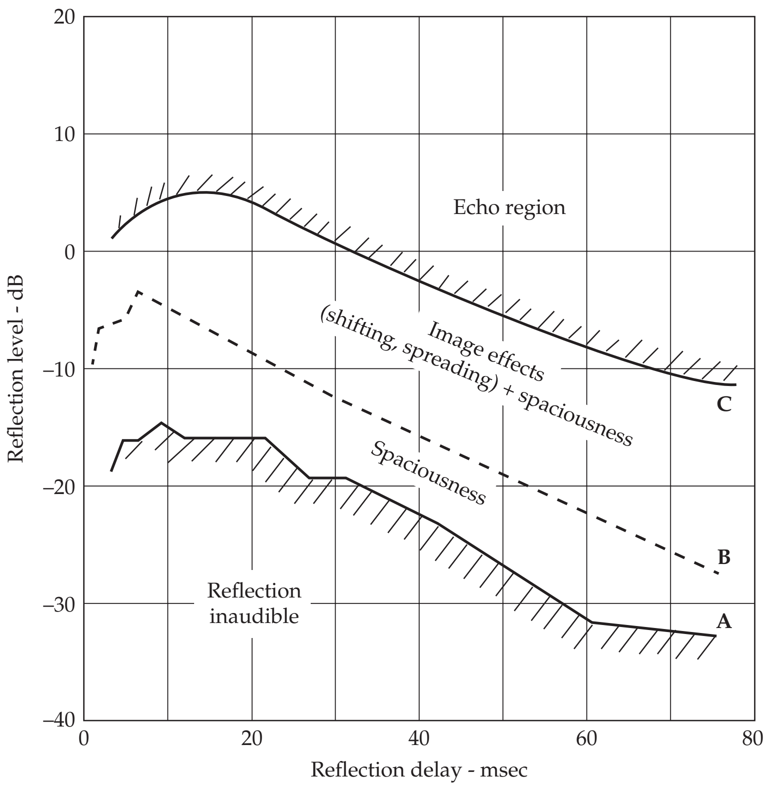

Figure 6-11 Results of investigations into the effect of a simulated lateral reflection in an anechoic environment with speech as the test signal. (A) The absolute threshold of detection of the reflection. (B) The image-shift threshold. (C) The threshold at which the reflection is heard as a discrete echo. (*Composite of results: Curves A and B from Olive and Toole, Curve C from Meyer and Lochner.*)

In Fig. 6-11, curve A is the absolute threshold of audibility of the echo. This means that at any particular delay, the reflection is not heard for reflection levels below this contour. Note that for the first 20 msec, this threshold is essentially constant. At greater delays, progressively lower reflection levels are required for a just-audible reflection. For small rooms, delays in the 0- to 20-msec range are of greatest significance. In this range the reflection audibility threshold varies little with delay.

Perception of Spaciousness, Images, and Echoes

Assume a reflection delay of 10 msec with the reflection coming from the side. At a very low level, the reflection is completely inaudible. As the level of the reflection is increased, it becomes audible as its level reaches about 15 dB below the direct sound. As the reflection level is increased beyond this point, the room takes on a sense of spaciousness; the anechoic room in which the tests were made sounds more like a normal room. The listener is not aware of the reflection as a discrete event, or of any directional effect, only this sense of spaciousness.

As the level of the reflection is increased further, other effects become audible. At about 10 dB above the threshold of audibility of the reflection, changes in the apparent size and location of the front auditory image become apparent. At greater delays, the image becomes smeared toward the reflection.

Reviewing what happens in the 10- to 20-msec delay range, as the reflection level is increased above the threshold of audibility, spatial effects dominate. As the reflection level is increased roughly 10 dB above the audibility threshold, image effects begin to enter, including image size and shifting of the position of the image.

Reflections having a level of another 10 dB above the image-shift threshold introduce another perceptual threshold. The reflections are now discrete echoes superimposed on the central image. Such discrete echoes are damaging to sound quality. For this reason, reflection level/delay combinations that result in such echoes must be minimized in practical designs.

Lateral reflections provide important perceptual cues in a sound field. Lateral reflections can affect spaciousness and the size and position of the auditory image. Olive and Toole investigated a two-loudspeaker installation and found that the effects obtained from a single loudspeaker are correlated to the stereo case. This suggests that single-loudspeaker data can be applied to stereo playback.

Those interested in the reproduction of high-fidelity audio will see the practicality of the results of these reflection studies. The spaciousness of a listening room as well as the stereo image definition can be adjusted by careful manipulation of lateral reflections. However, lateral reflections can be used only after interfering early reflections are reduced. This suggests practical room design techniques that will be explored in later chapters.

Effect of Angle of Incidence, Signal Type, and Spectrum on Audibility of Reflection

Researchers have shown that the direction from which a reflection arrives (angle of incidence) has practically no effect on the perception of the reflection, with one important exception. When the reflection arrives from the same direction as the direct sound, it can be up to 5 to 10 dB louder than the direct sound before it is detected. This is due to masking of the reflection by the direct sound. If the reflection is recorded along with the direct sound and reproduced over a loudspeaker, it will be masked by this 5- to 10-dB amount.

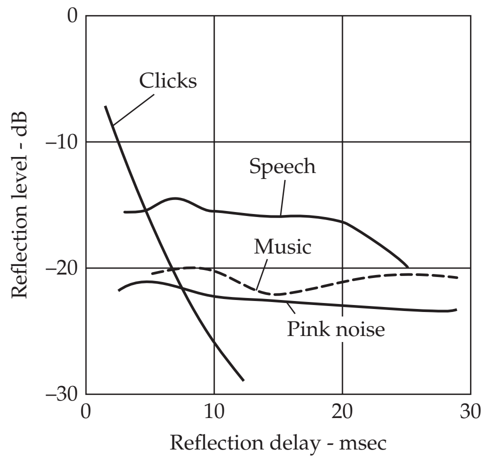

Figure 6-12 Absolute thresholds of perception of reflections of different types of signals: Clicks (2 clicks/sec) (noncontinuous), pink noise (continuous), speech, and classical music (Mozart). The similar thresholds for pink noise and classical music give assurance that pink noise is a reasonable surrogate for music in such tests. (*Olive and Toole.*)

The type of sound has a major effect on the audibility of reflections. Consider the difference between continuous and noncontinuous sounds. Impulses, in the form of 2 clicks/sec, are of the noncontinuous type. Pink noise is an example of the continuous type. Speech and music lie between the two types. Figure 6-12 shows the differences in audibility thresholds of continuous and noncontinuous sounds. Anechoic speech is closer to a noncontinuous sound than either music or pink noise. At delays of less than 10 msec, the level of impulses for threshold detection must be much higher than continuous sounds. The threshold curves for classical music (Mozart) and pink noise are very close together. This confirms the belief that pink noise is a reasonable surrogate for music in these types of experiments.

Most researchers use sounds having the same spectrum for both the direct and reflected simulations. In real life, reflections depart from the original spectrum because sound-absorbing materials invariably absorb high frequencies more than low frequencies. In addition to this, off-axis loudspeaker response lowers the high-frequency content even more. Threshold audibility experiments show that radical low-pass filtering of the reflection signal produces only minor differences in thresholds. The conclusion is that alteration of reflection spectrum does not appreciably change audibility thresholds.

Key Points

- The success or failure of any room of acoustical importance greatly depends on the characteristics of sound reflections as dictated by the room's boundary conditions.
- Acoustical specular reflection behaves the same as light reflecting from a mirror, described by Snell's law.

- A flutter echo is created as sound bounces back and forth between parallel surfaces. Where possible, parallel surfaces should be avoided in room design.
- When sound strikes a boundary surface, some sound energy is transmitted or absorbed by the surface and some is reflected.
- Sound can be considered as rays where each ray is viewed as a beam of diverging sound with a spherical wavefront.
- Convex architectural surfaces can usefully scatter sound energy in many directions. Concave architectural surfaces can cause unwanted concentrations of sound.
- Two opposing traveling waves between parallel surfaces interact to form a stationary standing wave. This is a resonant condition between the wavelength of the sound and the distance between the two surfaces.
- The mean free path is the average (mean) distance sound travels between successive reflections.
- Lateral reflections provide important perceptual cues in a sound field; the spaciousness of a listening room as well as the size and position of the auditory image can be adjusted by careful manipulation of lateral reflections.
- The angle of incidence (on-axis or not) of reflections and the type of sound (continuous or noncontinuous) affect the audibility of reflections.

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CHAPTER 7
Diffraction

From observation, we know that sound travels around obstacles and corners. For example, music reproduced in one room of a home can be heard down the hall and in other rooms. In part, this is due to reflections from walls and other surfaces, but it is also due to diffraction. Diffraction causes sound, which normally travels in straight lines, to bend and travel in other directions; diffraction occurs even in a free field without reflecting surfaces. However, the character of the music heard in distant parts of the home is different from the sound at the source. In particular, the bass notes are more prominent than treble notes. This is partly because their longer wavelengths more readily diffract (bend) around corners, past obstacles, and through openings. In contrast, treble notes, with short wavelengths, exhibit comparatively less diffraction. Diffraction thus varies according to the frequency of the sound in relation to the physical dimensions of the objects causing it.

Diffraction and Wavefront Propagation

Sound wavefronts normally travel rectilinearly, that is, in straight lines. Sound rays, a concept applicable to mid and high audible frequencies, can be considered as beams of sound that travel in straight lines perpendicular to the wavefront. Sound wavefronts and sound rays travel in straight lines, except when something gets in the way. Obstacles can cause sound to change direction from its original rectilinear path. The mechanism by which this change of direction takes place is called diffraction. Incidentally, the word diffract is from the Latin word *diffringere*, which means to break into pieces.

Isaac Newton pondered the relative merits of the corpuscular and wave theories of light. He decided that the corpuscular theory was the correct one because light is propagated rectilinearly. Later it was demonstrated that light is not always propagated rectilinearly, that diffraction can cause light to change its direction of travel. In fact, all types of wave motion, including sound, are subject to diffraction, a phenomenon caused by phase interference.

Christiaan Huygens formulated a principle that is the basis of the mathematical analysis of diffraction. The same principle also gives a simple explanation of how sound energy is diverted from a main beam into a shadow zone. Huygens' principle can be paraphrased: Every point on the wavefront of sound that has passed through an aperture or passed a diffracting edge is considered a new point source radiating energy into the shadow zone. The sound energy at any point in the shadow zone can be obtained mathematically by summing the contributions of all of these point sources on the wavefront.

Diffraction and Wavelength

For a given obstacle size, low-frequency sounds (long wavelengths) diffract more readily than high-frequency sounds (short wavelengths). Conversely, for a given obstacle, high frequencies diffract less readily than low frequencies. Diffraction is less noticeable for light than it is for sound because of the comparatively short wavelengths of light. As a result, optical shadows are more distinct than acoustical shadows. You might easily hear the diffracted music from a room down the hallway, but you might not see the diffracted light from there.

Looked at in another way, the effectiveness of an obstacle in diffracting sound is determined by the acoustic size of the obstacle. Acoustic size is measured in terms of the wavelength of the sound. An obstacle is acoustically small if wavelengths are long, but the same object is acoustically large if wavelengths are short. This relationship is described in more detail in the following text.

Diffraction by Obstacles

If an obstacle is acoustically small relative to wavelength, sound will easily diffract around it. The sound will bend around a small obstacle with only slight disturbance, creating little or no acoustical shadow. When an obstacle's dimensions are smaller or equal to the wavelength, virtually all sound will be diffracted. As noted, each wavefront passing the obstacle becomes a line of new point sources radiating sound into the shadow zone by diffraction. On the other hand, if an obstacle is acoustically large relative to wavelength, diffraction is less pronounced and a larger acoustical shadow is created. In addition, sound will tend to be reflected by a large obstacle; thus we see that obstructions act as frequency-dependent reflectors.

As noted, the effectiveness of an obstacle in diffracting sound is determined by the acoustic size of the obstacle. Consider two objects in Fig. 7-1 and their behavior with the same wavelength of sound. In Fig. 7-1A, the obstacle is small compared to the wavelength of the sound, thus sound strongly diffracts around it; the obstacle has no appreciable effect on the passage of sound. In Fig. 7-1B, however, if the obstacle is several

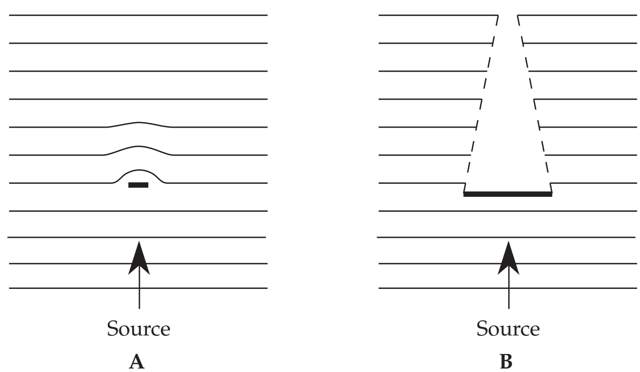


FIGURE 7-1 Diffraction varies according to the wavelength of sound and the size of an obstacle. (A) An obstacle very much smaller than the wavelength of sound allows the wavefronts to pass essentially undisturbed. (B) An obstacle larger than the wavelength casts an acoustical shadow.

wavelengths across, diffraction is less pronounced and the obstacle casts an acoustical shadow. Although not shown in this figure, some sound waves are also reflected from the face of the obstacle.

Another way of looking at the relationship of wavelength and diffraction is to change our basis of comparison, remembering that acoustic size depends on the wavelength of the sound. Consider Fig. 7-2A where the obstacle is the same physical size as that in Fig. 7-2B. However, the frequency of the sound in Fig. 7-2A is higher than that in Fig. 7-2B. We observe that the same obstacle casts a considerable acoustical shadow for high frequencies (the obstacle is relatively large), but casts much less of a shadow for low frequencies (the obstacle is relatively small).

As a final example, imagine an obstacle that is 0.1 ft across and an obstacle that is 1 ft across; if the frequency of the sound impinging on the first obstacle is 1,000 Hz (wavelength 1.13 ft) and the frequency impinging on the second is 100 Hz (wavelength 11.3 ft), both would exhibit the same diffraction.

A highway noise barrier is a common sight along busy roadways, and an example of an obstacle that is designed to isolate listeners (e.g., in homes) from impinging traffic sounds, as shown in Fig. 7-3. The spacing of the sound wavefronts indicates a relatively higher or lower frequency from the source (such as traffic). At higher frequencies, as shown in Fig. 7-3A, the barrier becomes effectively large and successfully shields a listener on the other side. Even if there is not complete isolation, at least higher-frequency traffic noise will be attenuated. At lower frequencies, as shown in Fig. 7-3B, the barrier becomes acoustically smaller. Low frequencies diffract over the barrier and are audible to the listener. Because of the change in the frequency response of the noise, the listener essentially hears bass-heavy traffic noise.

The wavefronts passing the top edge of the wall can be considered as lines of point sources radiating sound. This is the cause of the sound penetrating the shadow zone. The sound reflected from the wall should also be noted. It is as though the sound is coming from a virtual image on the far side of the wall.

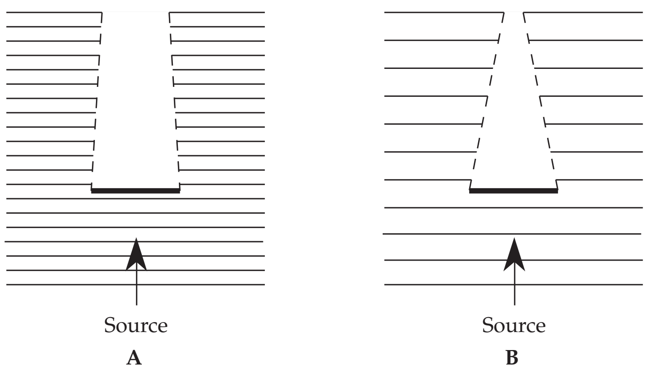


Figure 7-2 The same-sized obstacle will exhibit different degrees of diffraction, depending on the frequency (wavelength) of the sound. (A) When a high-frequency sound impinges on the obstacle, there is relatively little diffraction. (B) When a low-frequency sound impinges on the same obstacle, there is considerable diffraction. The edges act as sources, radiating sound into the shadow zone.

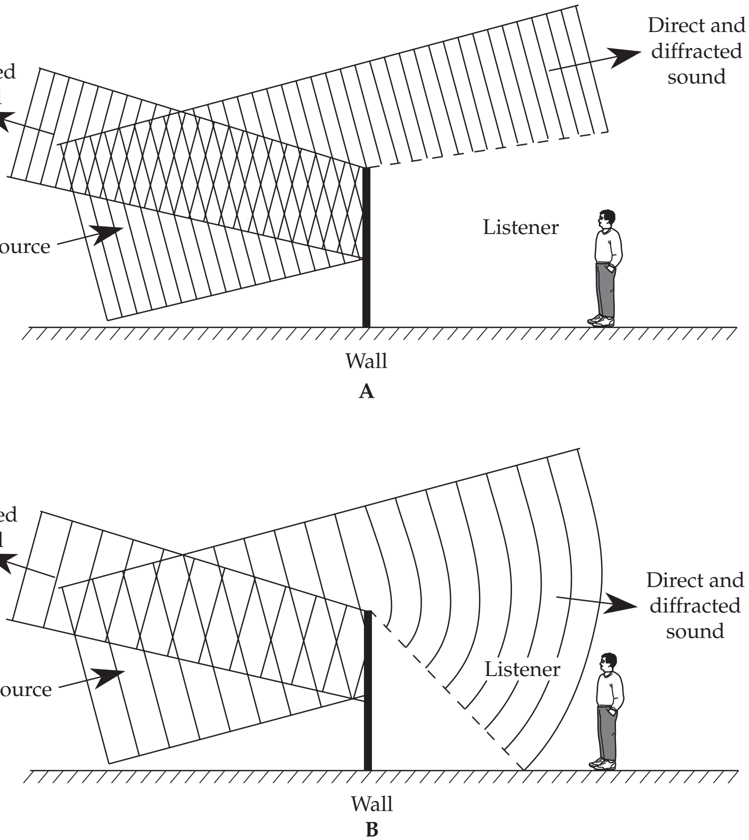


Figure 7-3 The sound striking a solid traffic barrier will be partly diffracted and partly reflected. (A) High-frequency traffic sounds are successfully attenuated on the other side of the barrier because of limited diffraction. (B) Low-frequency traffic sounds are less attenuated because of more prominent diffraction. Sound passing the top edge of the barrier acts as though the wavefronts are lines of sources, radiating sound energy into the shadow zone.

Figure 7-4 illustrates the effectiveness of highway barriers, showing the attenuation of sound in the shadow of a high, massive wall. The center of the highway is taken to be 30 ft from the wall on one side, and the home or other sensitive area is considered to be 30 ft on the other side of the wall (the shadow side). A 20-ft-high wall yields about 25 dB of attenuation from the highway noise at 1,000 Hz. However, the attenuation of the highway noise at 100 Hz is only about 15 dB. The wall is less effective at lower frequencies than at higher frequencies. The shadow zone behind the wall tends to be effective for the high-frequency components of the highway noise. The low-frequency components penetrate the shadow zone by diffraction. To be effective, barriers must be sufficiently tall, and also long enough to prevent flanking around the end of the barrier. The effectiveness of any barrier is strictly frequency-dependent.

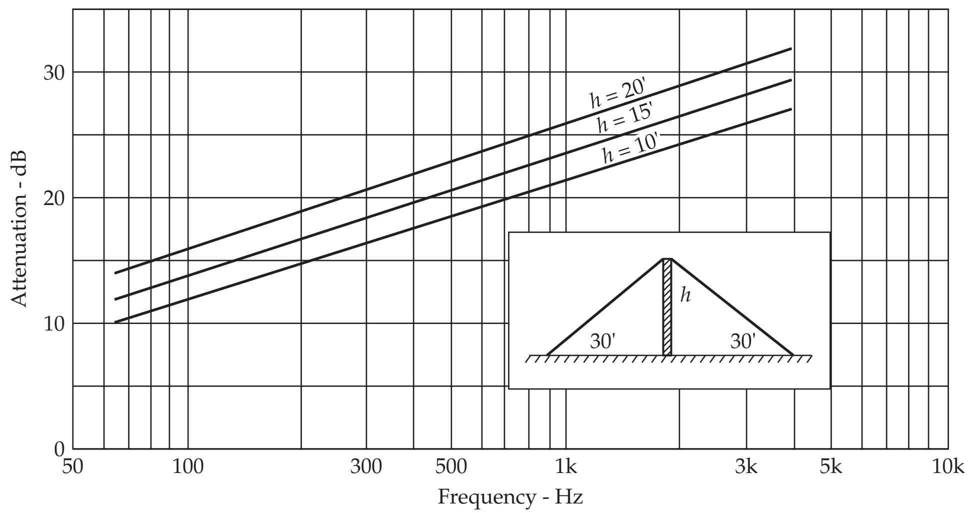


FIGURE 7-4 An estimation of the effectiveness of a traffic barrier in terms of sound (or noise) attenuation as a function of frequency and barrier height. (*Rettinger.*)

Diffraction by Apertures

Diffraction by apertures depends on the size of the opening and the wavelength of the sound. The amount of diffracted sound relative to total sound passing through an aperture increases as the opening size decreases. As with diffraction around obstacles, diffraction through apertures is wavelength-dependent. Diffraction increases as frequency decreases. Thus, an opening is more acoustically transparent at low frequencies.

Figure 7-5A illustrates the diffraction of sound by an aperture that is many wavelengths wide. The wavefronts of sound strike a solid obstacle; some sound passes

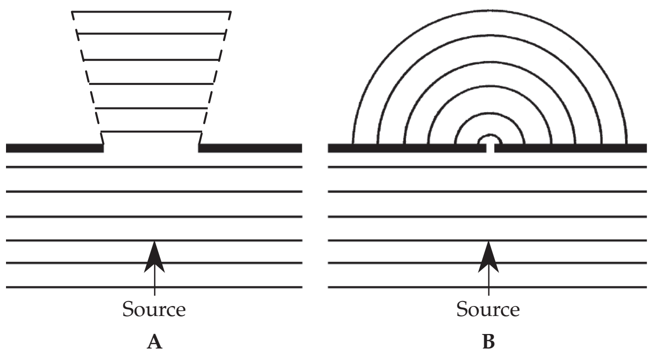


FIGURE 7-5 When plane waves of sound strike a barrier with an aperture, diffraction depends on the wavelength of sound and the relative size of the opening. (A) An aperture large in terms of wavelength allows wavefronts to go through with little directional disturbance. (B) If the aperture is small compared to the wavelength, a hemispherical field of sound radiates on the other side of the barrier.

through the wide aperture and although not shown, some sound is reflected. By diffusion, some of the energy in the main beam is diverted into the shadow zone. Each wavefront passing through the aperture becomes a row of point sources radiating diffracted sound into the shadow zone. The same principle holds for Fig. 7-5B, except that the aperture is small compared to the wavelength of sound. Most of the sound energy is reflected from the wall surface and only a small amount of energy passes through it. The points on the limited wavefront going through the aperture are so close together that their radiations take the form of a hemisphere. Following Huygens' principle, the sound emerges from the aperture as a hemispherical wave that diverges rapidly. Most of the energy passing through a small aperture is through diffraction. Because of diffraction, even a small opening in a barrier can transmit a relatively large amount of sound energy, particularly at low frequencies.

Diffraction by a Slit

Figure 7-6 diagrams a classic experiment first performed by Pohl, and described more recently by Wood. The equipment layout of Fig. 7-6A is very approximate. The source/slit arrangement rotated about the center of the slit and the measuring radiometer was at a distance of 8 m. The slit width was 11.5 cm wide, and the wavelength of the measuring sound was 1.45 cm (23.7 kHz). Figure 7-6B shows the intensity of the sound versus the angle of deviation. The dimension *X* indicates the geometrical boundaries of the ray. Any response wider than *X* is caused by diffraction of the beam by the slit. A narrower slit would yield correspondingly more diffraction and a greater width of the beam. The increase in width of the beam, that is, the extent of diffraction, is the striking feature of this experiment.

Diffraction by a Zone Plate

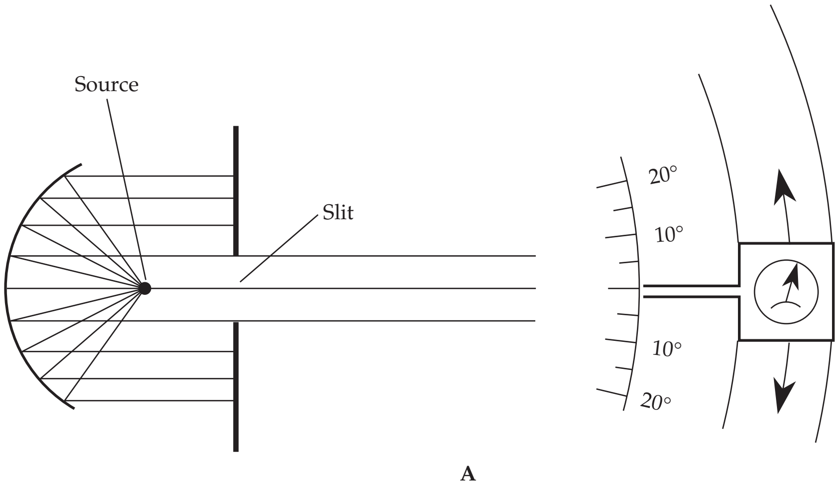
The zone plate shown in Fig. 7-7 can be considered an acoustic lens. It consists of a circular plate with a set of concentric, annular slits of specific radii. If the focal point is at a distance of r from the plate, the next longer path must be $r + \lambda/2$, where λ is the wavelength of the sound falling on the plate from the source. Successive path lengths are $r + \lambda$, $r + 3\lambda/2$, and $r + 2\lambda$. These path lengths differ by $\lambda/2$, which means that the sound through all the slits will arrive at the focal point in phase which, in turn, means that they add constructively, intensifying the sound at the focal point.

Diffraction around the Human Head

Figure 7-8 illustrates the diffraction caused by a sphere roughly the size of the human head. This diffraction by the head as well as reflection and diffraction from the shoulders and upper torso influences human perception of sound. In general, for sound of frequency 1 to 6 kHz arriving from the front, head diffraction tends to increase the sound pressure in front and decrease it behind the head. As expected, for lower frequencies, the directional pattern tends to become circular.

Diffraction by Loudspeaker Cabinet Edges

Loudspeaker cabinets are notorious for diffraction effects. If a loudspeaker is placed near a wall and aimed away from the wall, the wall is still illuminated with sound diffracted from the corners of the cabinet. Reflections of this sound can affect the quality of



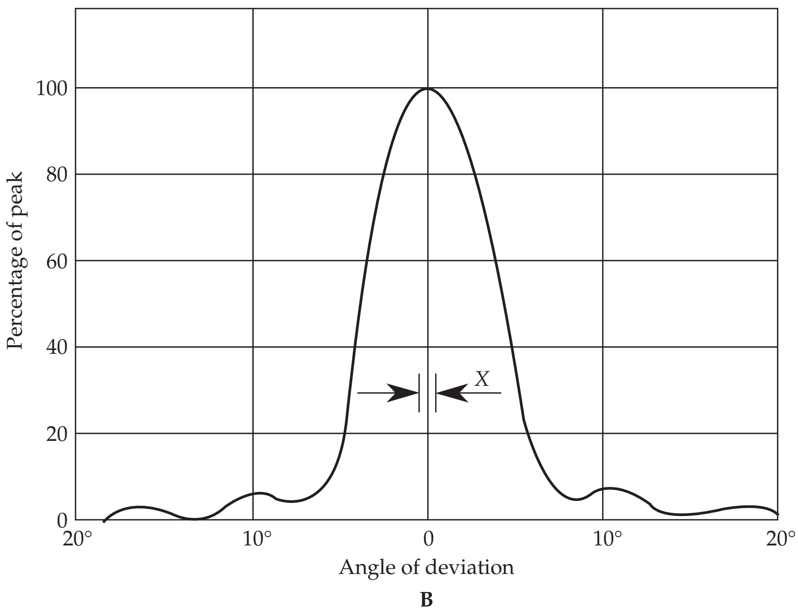


FIGURE 7-6 A consideration of Pohl's classic experiment in diffraction. (A) An approximation of the equipment arrangement showing source and slit. (B) Diffraction causes the beam to broaden with a characteristic pattern. The narrower the slit, the greater this broadening of the beam. (*Wood.*)

the sound at the listener's position. Vanderkooy and Kessel computed the magnitude of loudspeaker cabinet edge diffraction. The computations were made on a box loudspeaker cabinet with front baffle having the dimensions 15.7 × 25.2 in, and a depth of 12.6 in. A point source of sound was located symmetrically at the top of the baffle, as shown in Fig. 7-9. The response from this point source was computed at a distance from

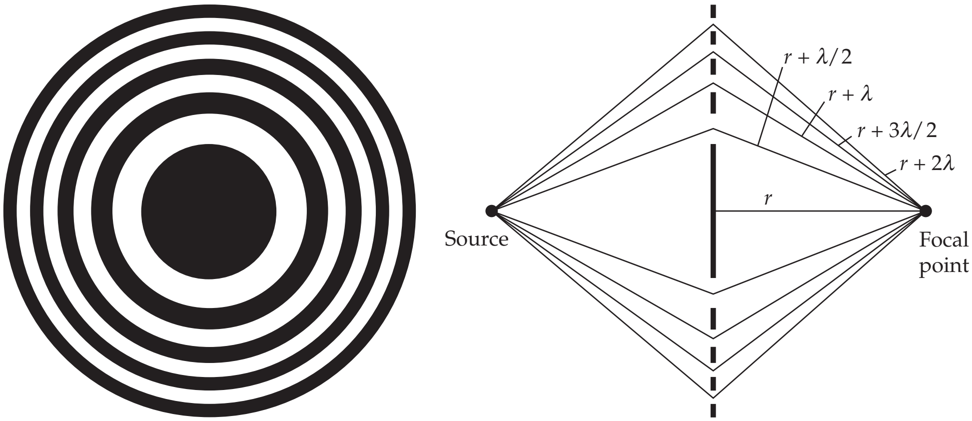


Figure 7-7 The zone plate acts as an acoustic lens. The slits are arranged so that the several path lengths differ by multiples of a half wavelength of the sound so that all diffracted rays arrive at the focal point in phase, combining constructively. (*Olson.*)

the box. The sound arriving at the observation point is the combination of the direct sound plus the edge diffraction. The resulting response is shown in Fig. 7-10. Fluctuations due to edge diffraction for this particular experiment approached ±5 dB. This is a significant change in overall frequency response of a reproduction system.

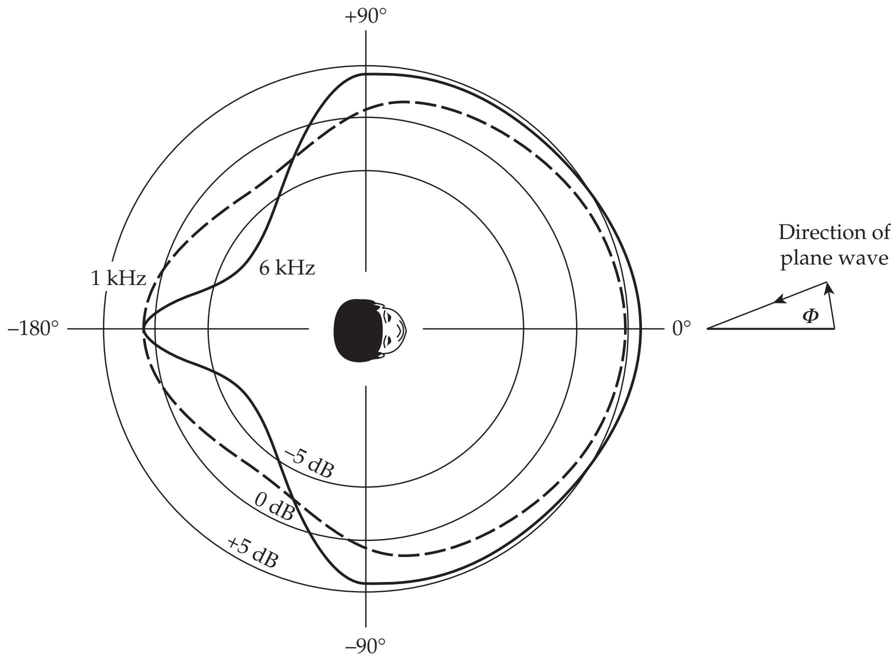


Figure 7-8 Diffraction occurs around a solid sphere about the size of a human head. For sound in the 1- to 6-kHz range, sound pressure is generally increased in the front hemisphere and reduced in the rear. (*Muller, Black, and Davis, as reported by Olson.*)

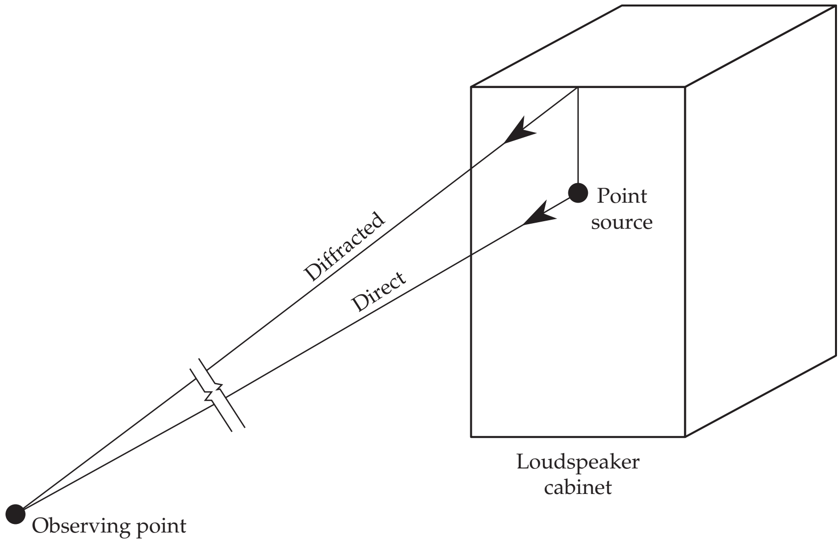


Figure 7-9 Experimental setup for the measurement of loudspeaker cabinet edge diffraction, as shown in Fig. 7-10. Sound arriving at the observation point is the combination of the direct sound plus the edge diffraction. (*Vanderkooy, Kessel.*)

This cabinet diffusion effect can be controlled by setting the loudspeaker face flush in a much larger baffling surface. Diffusion can also be reduced by rounding cabinet edges and using foam or other absorbing material around the front of the cabinet.

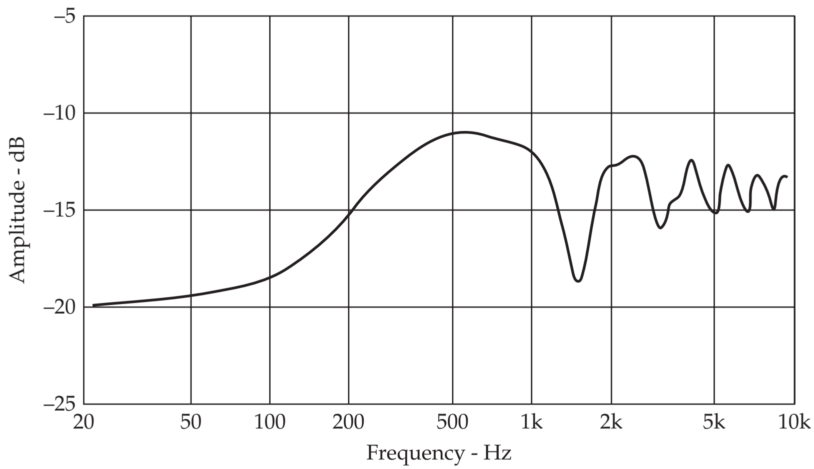


Figure 7-10 The calculated effects of loudspeaker edge diffraction on the direct signal in the setup of Fig. 7-9. There is a significant change in overall frequency response of the system. (*Vanderkooy, Kessel.*)

Diffraction by Various Objects

Early sound-level meters were simply boxes with a microphone. Diffraction from the edges and corners of the box seriously affected the accuracy of the readings at high frequencies. Modern sound-level meters have cases with carefully rounded contours, and the microphone is mounted on a smooth, slender, rounded stalk to remove it from the meter case.

Similarly, diffraction from the casing of a studio microphone can cause deviations from the desired flat response. This must be taken into account when designing a microphone.

When measuring sound absorption in large reverberation chambers, the common practice is to place the material to be measured in an 8- × 9-ft frame on the floor. Diffraction from the edges of this frame may result in absorption coefficients greater than unity. In other words, diffraction of sound makes the sample appear larger than it really is. Similarly, in practice, the absorption of materials is increased because of diffraction at their edges. For this reason, and others, it is preferable to space absorbing panels apart from each other, rather than join them into one section. This takes advantage of increased absorption because of edge diffraction.

Small cracks around observation windows or back-to-back microphone or electrical service boxes in partitions can destroy the hoped-for isolation between studios or between a studio and control room. The sound emerging on the other side of the hole or slit is spread in all directions by diffraction. For this reason, when designing for sound isolation, any opening in a partition must be sealed.

Key Points

- When sound encounters an obstacle, diffraction causes sound to bend and travel in other directions; diffraction varies according to the frequency of the sound in relation to the physical dimensions of the objects causing it.
- According to Huygens' principle, every point on a wavefront of sound that has passed through an aperture or passed a diffracting edge can be considered a new point source radiating energy into the shadow zone.
- For a given obstacle size, low-frequency sounds (long wavelengths) diffract more readily than high-frequency sounds (short wavelengths).
- If an obstacle is acoustically large relative to wavelength, diffraction is less pronounced and a larger acoustical shadow is created.
- Because of diffraction, the effectiveness of any barrier is frequency-dependent. For example, a highway noise barrier is less effective at lower frequencies than at higher frequencies.
- Because of diffraction, even a small opening in a barrier can transmit a relatively large amount of sound energy, particularly at low frequencies. To preserve isolation, any openings in a partition must be sealed.
- Because of diffusion, objects such as the human head and torso, and loudspeaker cabinets significantly affect the frequency response of a sound field.
- Absorbing panels should be spaced apart from each other, rather than joined into one section, to take advantage of increased absorption from edge diffraction.

CHAPTER 8
Refraction

At the turn of the twentieth century, Lord Rayleigh was puzzled because some powerful sound sources, such as cannon fire, could be heard only short distances sometimes and very great distances at other times. He calculated that if all the power from a siren driven by 600 horsepower was converted into sound energy and spread uniformly over a hemisphere, the sound should be audible to a distance of 166,000 miles, more than six times the circumference of the earth. However, such sound propagation is never experienced and a maximum range of a few miles is typical.

When dealing with sound propagation, particularly outdoors, refraction plays a large role. Refraction is a change in the direction of sound propagation that occurs when there is a change in the transmission medium. In particular, the change in the medium changes the speed of sound propagation, and therefore the sound bends.

There are many reasons why Rayleigh's estimate was wrong, and why sound is not heard over great distances. For one thing, refraction in the atmosphere will profoundly affect the propagation of sound over distance. In addition, the efficiency of sound radiators is usually quite low; not much of that 600 horsepower would actually be radiated as sound. Energy is also lost as wavefronts drag across the rough surface features of the earth. Another loss is dissipation in the atmosphere, particularly affecting high frequencies. Early experiments by Rayleigh and others accelerated research on the effects of temperature and wind gradients on the transmission of sound. This chapter will discuss the effects of refraction.

The Nature of Refraction

The difference between absorption and reflection of sound is obvious, but sometimes there is confusion between refraction and diffraction (and possibly diffusion, the subject of Chap. 9).

Refraction is the change in the direction of travel of sound because of differences in the velocity of propagation. Diffraction is the change in the direction of travel of sound as it encounters physical obstructions, edges, and openings. Of course, in practical situations, it is entirely possible for both effects to simultaneously affect the same sound.

Figure 8-1 recalls a common observation of the apparent bending of a pencil as one end is immersed in water. This is an illustration of the refraction of light, caused by different speeds of propagation in air and water and hence different refractive indices. The refraction of sound, which is another wave phenomenon, is similar. In the case of air and water, the change in the refractive indices is abrupt, as is the bending of light. Sound refraction can occur abruptly, or gradually, depending on how the transmission mediums affect its speed.

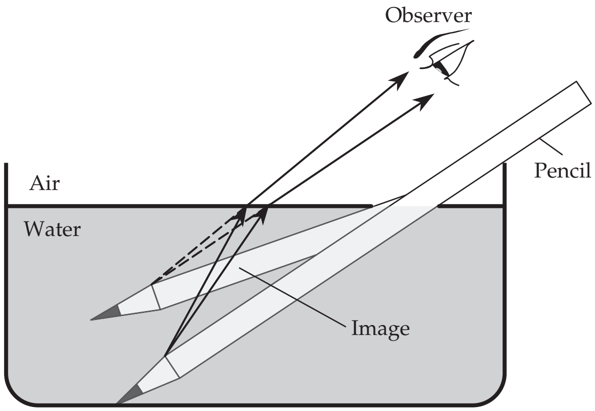


Figure 8-1 Partially immersing a stick in water illustrates refraction of light, created because the speeds of propagation are different in air and water. Sound is another wave phenomenon that is also refracted because of changes in the speed of sound in media.

Refraction in Solids

The sound-ray concept is helpful in considering the direction of propagation. Rays of sound are always perpendicular to sound wavefronts. Figure 8-2 shows two sound rays passing from one medium to another; further, the sound speed in the first medium is lower than that in the second medium. As one ray reaches the boundary between the

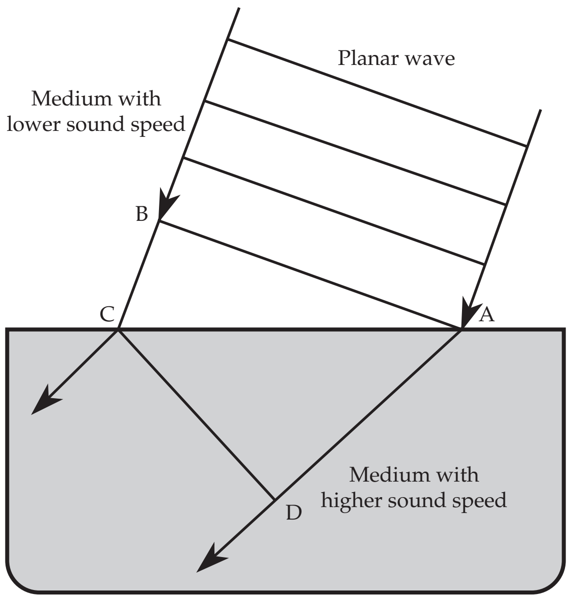


Figure 8-2 Rays of sound in this example are refracted when traveling from a medium having a lower speed of sound into a medium having a higher speed of sound. The wavefront A-B is not parallel to wavefront C-D because the direction of the wave is changed due to refraction.

two media at A, the other ray still has some distance to go. In the time it takes one ray to travel from B to C, the other ray has traveled a farther distance from A to D in the second medium. Wavefront A-B represents one instant of time as does wavefront C-D an instant later. But these two wavefronts are no longer parallel. The rays of sound have been refracted at the interface of the two media having unequal sound speeds. In particular, as noted, in this example, sound travels slower in the first medium and faster in the second medium. Generally, the stiffer or more rigid a medium, or the less compressible it is, the faster sound travels through it.

An analogy may be useful. Assume that the unshaded area of Fig. 8-2 is unpaved ground and the shaded area is a paved surface. Assume also that the wavefront A-B is a line of soldiers. The line of soldiers A-B, marching in military order, makes slow progress on the unpaved ground. As soldier A reaches the paved surface, the soldier speeds up and begins striding over the paved surface. Soldier A travels to D on the paved surface in the same time it takes soldier B to travel the distance B to C on the unpaved ground. This tilts the column of soldiers off in a new direction, which is the definition of refraction. In any homogeneous medium, sound travels rectilinearly (in the same direction). If a medium with another sound speed is encountered, the sound is refracted.

Refraction in the Atmosphere

The earth's atmosphere is anything but a stable, uniform medium for the propagation of sound. Sometimes the air near the earth is warmer than the air at greater heights; sometimes it is colder. At the same time this vertical layering exists, other changes may be taking place horizontally. It is a wondrously intricate and dynamic system, challenging acousticians (and meteorologists) to make sense of it.

In the absence of thermal gradients, a sound ray may be propagated rectilinearly, as shown in Fig. 8-3A. As noted, rays of sound are perpendicular to sound wavefronts.

In Fig. 8-3B, a thermal gradient exists between the cool air near the surface of the earth and the warmer air above. This affects the wavefronts of sound. Sound travels faster in warm air than in cool air causing the tops of the wavefronts to go faster than the lower parts. The tilting of the wavefronts directs the sound rays downward. Under such conditions, sound from source S is continually bent down toward the surface of the earth and may follow the earth's curvature and be heard at relatively great distances.

In Fig. 8-3C, the thermal gradient is reversed as the air near the surface of the earth is warmer than the air higher up. In this case, the lower parts of the wavefronts travel faster than the tops, resulting in an upward refraction of the sound rays. The same sound energy from source S would now be dissipated in the upper reaches of the atmosphere, reducing the chances of it being heard at any great distance at the surface of the earth.

Figure 8-4A presents a distant view of the downward refraction scenario of Fig. 8-3B. Sound traveling directly upward from the source S penetrates the temperature gradient at right angles and would not be refracted. It would speed up and slow down slightly as it penetrates the warmer and cooler layers, but would still travel in a vertical direction. All sound rays, except the vertical, would be refracted downward. The amount of this refraction varies materially: The rays closer to the vertical are refracted much less than those more or less parallel to the surface of the earth.

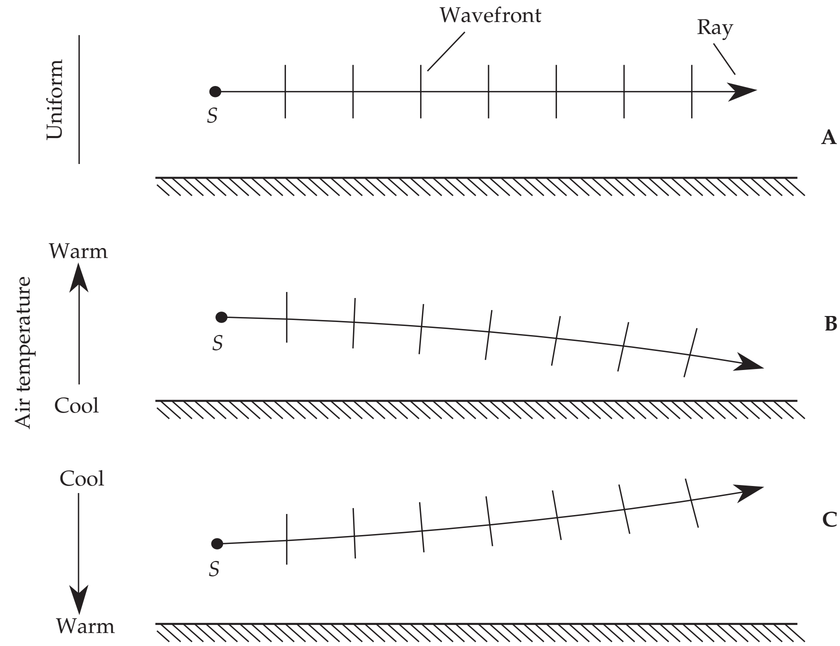


Figure 8-3 Refraction of sound paths resulting from temperature gradients in the atmosphere. (A) Air temperature constant with height. (B) Cool air near the surface of the earth and warmer air above. (C) Warm air near the earth and cooler air above.

Figure 8-4B is a distant view of the upward refraction scenario of Fig. 8-3C. Shadow zones are to be expected in this case. Again, the vertical ray is the only one escaping refractive effects.

Although it is not a case of refraction, wind can have a significant effect on sound propagation and plays an important role in analyzing noise pollution over large distances. For example, it is a common experience to hear sound better downwind than upwind. As a result, for example, on a windy day, sound can pass over a traffic barrier that would normally block traffic noise. However, this is not because wind is blowing sound toward the listener. Rather, wind speed is generally slower close to the ground than at greater heights. This wind gradient may affect the propagation of sound. Plane waves of sound from a distant source traveling with the wind will bend down toward the earth. Plane waves traveling against the wind will bend upward. Figure 8-5 illustrates the effect of wind on the downward refraction case of Fig. 8-4A. An upwind shadow is created while downwind sound is helpfully redirected downward. As noted, this is not true refraction, but the effect is the same, and it can be as significant as temperature-related refraction.

To a lesser extent, if wind moves the air at a certain speed, it is to be expected that the speed of sound will be affected. For example, if sound travels 1,130 ft/sec and a 10 mi/hr (about 15 ft/sec) wind prevails, upwind the sound speed with respect to the earth would be increased about 1%, and downwind it would be decreased by the same amount. This is a small change, but it may also affect sound propagation.

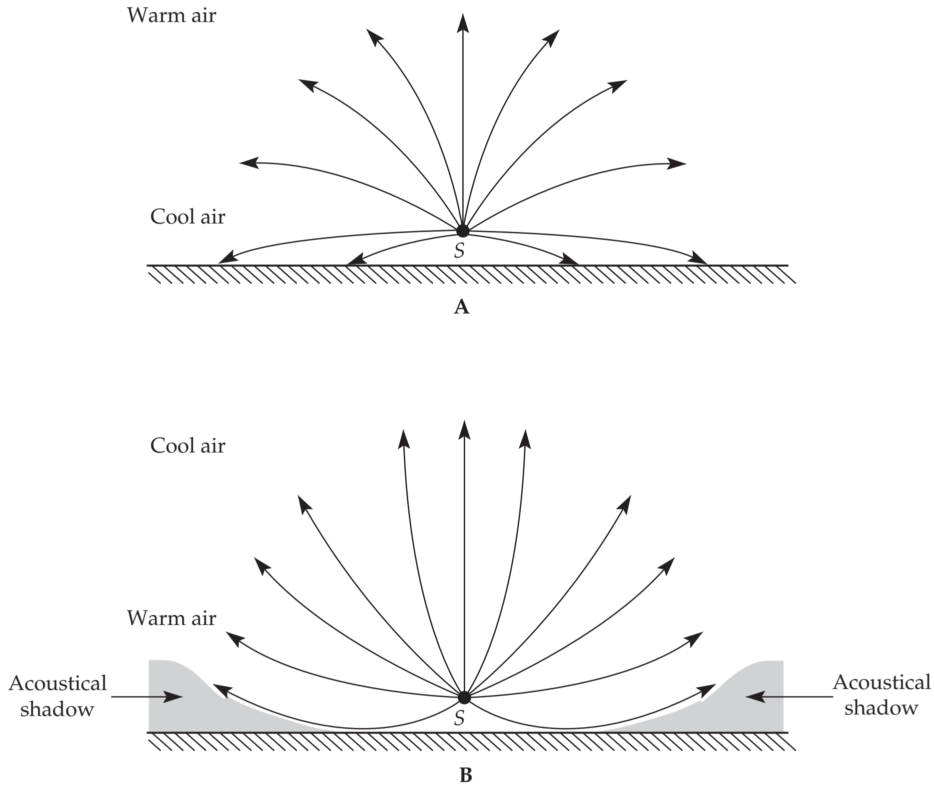


Figure 8-4 Comprehensive illustration of refraction of sound from a source. (A) Cool air near the ground and warmer air above. (B) Warm air near the ground and cooler air above; note that acoustical shadow areas result from the upward refraction.

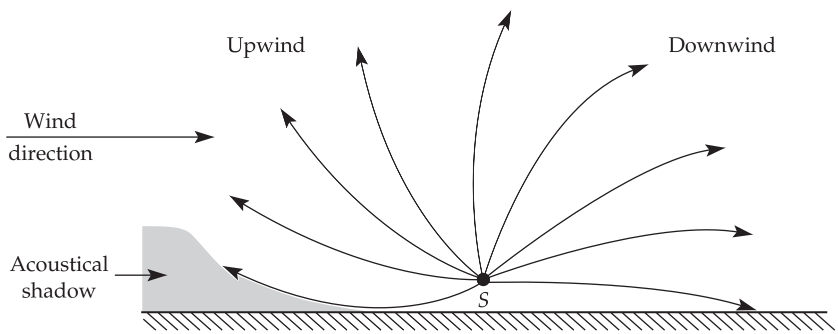


Figure 8-5 Although not a refraction effect, wind gradients can affect the propagation of sound. Shown here is the effect of wind on the downward refraction case of Fig. 8-4A. A shadow sound is created upwind while good listening conditions exist downwind.

It is possible, under unusual circumstances, that sound traveling upwind may actually be favored. For instance, upwind sound is kept above the surface of the ground, minimizing losses at the ground surface.

In some cases, there may be both temperature and wind changes that simultaneously affect sound propagation. They might combine to produce a larger effect, or tend to negate one another. Thus, results are unpredictable. For example, a lighthouse foghorn might be clearly audible near the lighthouse and also at a distance far away, but mysteriously, not be audible at points between. Interestingly, in 1862, during a Civil War battle in Iuka, Mississippi, it is believed that two divisions of Union soldiers did not enter battle because a wind-related acoustical shadow prevented them from hearing the sounds of the conflict raging 6 miles downwind.

Refraction in Enclosed Spaces

Refraction has an important effect on sound outdoors and sound traveling great distances. Refraction plays a much less significant role indoors. Consider a multiuse gymnasium that also serves as an auditorium. With a normal heating and air-conditioning system, efforts are made to avoid large horizontal or vertical temperature gradients. If there is temperature uniformity, and no troublesome drafts, sound refraction effects should be reduced to inconsequential levels.

Consider the same gymnasium also used as an auditorium but with less sophisticated air conditioning. Assume that a large heater is ceiling-mounted. The unit would produce hot air near the ceiling and rely on slow convection currents to move some of the heat down to the audience level.

This reservoir of hot air near the ceiling and cooler air below could have a minor effect on the transmission of sound from the sound system and on the acoustics of the space. The feedback point of the sound system might shift. The standing waves of the room might change slightly as longitudinal, and transverse sound paths are increased in length because of their curvature due to refraction. Flutter echo paths may also shift. With a sound system mounted high at one end of the room, sound paths could be curved downward. Such downward curvature might actually improve audience coverage, depending somewhat on the directivity of the radiating system.

Refraction in the Ocean

In 1960, a team of oceanographers headed by Heaney performed an experiment to monitor the propagation of underwater sound. Charges of 600 lb were discharged at various depths in the ocean off Perth, Australia. Sounds from these discharges were detected near Bermuda, a distance of over 12,000 miles. Sound in seawater travels 4.3 times faster than in air, but it still took 3.71 hours for the sound to make the trip.

Oceanic refraction played a significant role in this experiment. The depth of the ocean may be over 5,000 fathoms (30,000 ft). At about 700 fathoms (4,200 ft) an interesting effect occurs. The sound-speed profile shown in Fig. 8-6A very approximately illustrates the principle. In the upper reaches of the ocean, the speed of sound decreases with depth because temperature decreases. At greater depths the pressure effect prevails, causing speed of sound to increase with depth because of the increase in density. The V-shaped profile changeover from one effect to the other occurs near the 700-fathom (4,200-ft) depth.

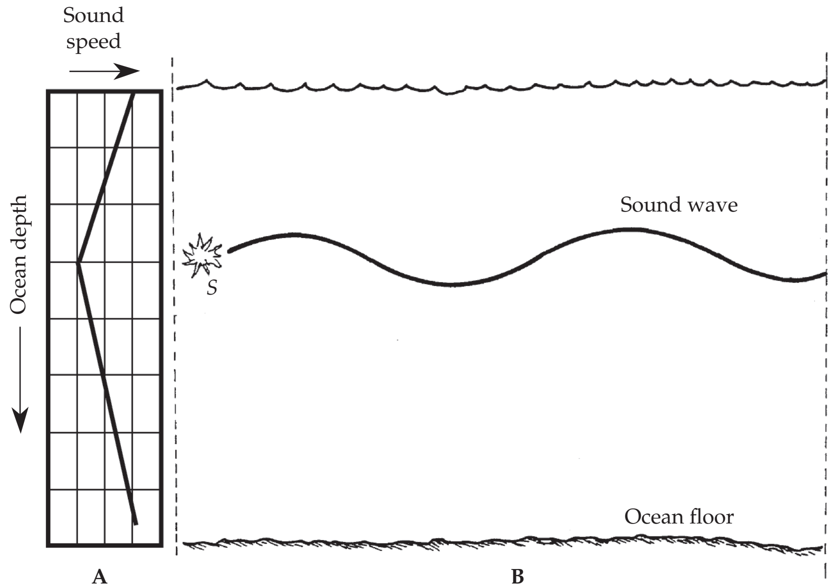


Figure 8-6 Schematic showing how oceanic refraction can affect sound propagation underwater. (A) Sound speed decreases with depth in the upper reaches of the ocean (temperature effect) and increases at greater depths (pressure effect), creating a sound channel at the inversion depth (about 700 fathoms). (B) A ray of sound is kept in this sound channel by refraction. Sound travels great distances in this channel because of the decreased losses.

A sound channel is created by this V-shaped sound-speed profile as shown in Fig. 8-6B. A sound emitted in this channel tends to spread out in all directions. Any ray traveling upward will be refracted downward, while any ray traveling downward will be refracted upward. Thus, sound energy in this channel is propagated great distances with modest losses.

Refraction in the vertical plane is prominent because of the vertical temperature/pressure gradient. There is relatively little horizontal sound-speed gradient and therefore very little horizontal refraction. Sound tends to spread out in a thin sheet at this 700-fathom depth. Spherical divergence in three dimensions is changed to two-dimensional propagation at this unique depth.

These long-distance sound channel experiments have suggested that such measurements can be used to monitor global climate change by detecting changes in the average temperature of the oceans. The speed of sound is a function of the temperature of the ocean. Accurate measurements of time of transit over a given course yield information on the temperature of that ocean.

Key Points

- Refraction is the change in the direction of travel of sound because of differences in the velocity of propagation caused by changes in the transmission medium.

- Sound travels faster in warm air than in cool air. When there is cool air near the surface of the earth and the warmer air above, the tops of wavefronts go faster than the lower parts. The tilting of the wavefronts directs the sound rays downward. Sound would be bent down toward the surface and may follow the earth's curvature and be heard at relatively great distances.
- When air near the surface of the earth is warmer than the air higher up, the lower parts of the wavefronts travel faster than the tops, resulting in an upward refraction. Sound energy would be dissipated upward, reducing the chances of it being heard at any great distance at the surface.
- Although not a case of refraction, wind gradients may affect the propagation of sound. Usually, plane waves of sound traveling with the wind will bend downward; plane waves traveling against the wind will bend upward.
- Refraction has an important effect on sound outdoors and sound traveling great distances, but plays a much less significant role indoors.

CHAPTER 9
Diffusion

To make their calculations easier, theorists often assume a completely diffuse sound field, one that is isotropic and homogeneous. That is, a sound field where at any point, sound can arrive from any direction, and a field that is the same throughout the room. In practice, that rarely occurs, especially in small rooms. Instead, as we plainly hear every day, the characteristics of sound are markedly different throughout most rooms. In some cases, directionality is welcome, because it may, for example, help a listener localize the source of the sound. In most room designs, diffusion is used to more effectively distribute sound and to provide a more equal response throughout a room that immerses the listener in the sound, while preserving a level of directionality that is appropriate for the application. It is often difficult to provide sufficient diffusion, particularly at low frequencies and in small rooms, because of the room's modal response. The goal of most room designs is to obtain sound energy across the audible frequency range that is uniformly distributed throughout the room. This is unattainable, but diffusion greatly assists in the effort.

The Perfectly Diffuse Sound Field

Even though unattainable, it is instructive to consider the characteristics of a perfectly diffuse sound field. Randall and Ward have suggested the following ideals:

- The frequency and spatial irregularities obtained from steady-state measurements must be negligible.
- Beats in the decay characteristic must be negligible.
- Decays must be perfectly exponential (they will appear as straight lines on a logarithmic scale).
- Reverberation time will be the same at all positions in the room.
- The character of the decay will be essentially the same for all frequencies.
- The character of the decay will be independent of the directional characteristics of the measuring microphone.

These six factors are observation-oriented. More theoretically, a diffuse sound field would be defined in terms such as energy density, energy flow, and superposition of an infinite number of plane progressive waves. However, these six characteristics point us to practical ways of judging the diffuseness of the sound field in a given room.

Evaluating Diffusion in a Room

To obtain the frequency response of an amplifier, a variable-frequency signal is input and the output is observed to measure the response. The same approach can be applied to a room's playback system by reproducing a variable-frequency signal through a loudspeaker and measuring the acoustical signal picked up by a microphone located in the room. However, the acoustical response of playback systems in rooms is never as flat as the response of electrical devices. These deviations are partly due to nondiffusive conditions in the room. As noted, diffusion is welcome because it helps envelop the listener in the sound field. On the other hand, too much diffusion can make it difficult to localize the sound source.

Steady-State Measurements

Figure 9-1 shows the steady-state response of a small studio with a volume of 12,000 ft^3. In this example, the loudspeaker was placed in one lower tri-corner of the room, and the microphone was at the upper diagonal tri-corner about 1 ft from each of the three surfaces. These positions were chosen because all room modes terminate in the corners and all modes should be represented in the response. The fluctuations in this response cover a range of about 35 dB over the linear 30- to 250-Hz sweep. The nulls are very narrow, and the narrow peaks show evidence of being single modes because the mode bandwidth of this room is close to 4 Hz. The wider peaks are the combined effect of several adjacent modes. The rise from 30 to 50 Hz is due primarily to loudspeaker response, and the 9-dB peak between 50 and 150 Hz is due to the loudspeaker radiating into a one-quarter space; therefore, these are testing artifacts that are not part of the room response.

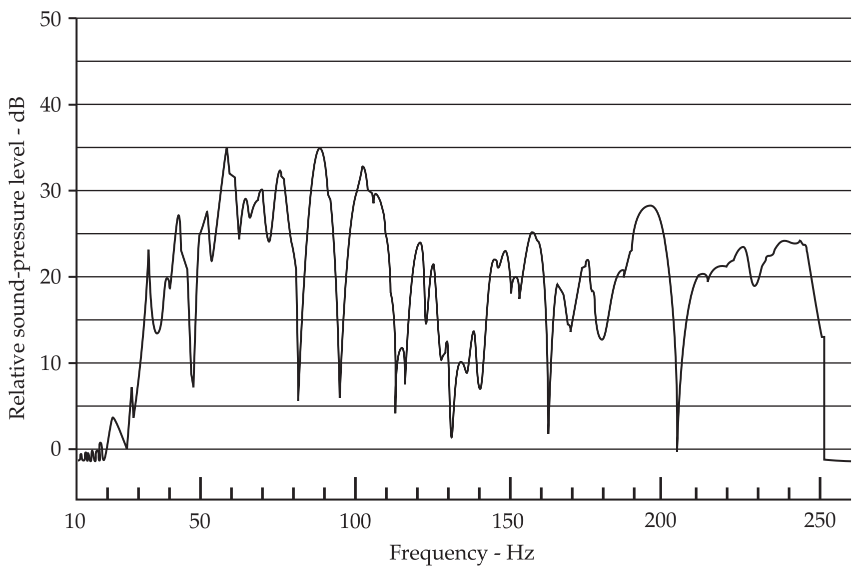


FIGURE 9-1 The recorder tracing of a slowly swept sine-wave sound-transmission response of a 12,000-ft^3 studio. Fluctuations of this magnitude, which characterize many studios, are evidence of nondiffusive conditions.

The response of Fig. 9-1 is typical of even the best studios. Such variations in response are evidence of a sound field that is not perfectly diffuse. A steady-state response such as this taken in an anechoic room would still show variations, but of lower amplitude. A very live room, such as a reverberation chamber, would show even greater variations.

Fixed measurements are one way to obtain the steady-state response of a room. Another method of appraising room diffusion is to rotate a highly directional microphone in various planes while holding the loudspeaker frequency response constant and recording the microphone's output to the constant excitation of the room. This method is used with some success in large spaces, but the method is ill adapted to smaller rooms, where diffusion problems are greatest. In principle, however, in a totally homogeneous sound field, a highly directional microphone pointed in any direction should pick up a constant signal. Both fixed- and rotating-measurement methods reveal the same deviations from a truly homogeneous sound field. We see that the criteria of negligible frequency and spatial irregularities are not met in any practical room.

Decay Beats

By referring ahead to Fig. 11-9, we can compare the smoothness of the reverberation decay for the eight octaves from 63 Hz to 8 kHz. In general, the smoothness of the decay increases as frequency is increased. The reason for this, as explained in Chap. 13, is that the number of modes within an octave span increases greatly with frequency, and the greater the mode density, the smoother their average effect. Conversely, in this example, beats in the decay are greatest at 63 and 125 Hz. If all decays have the same character at all frequencies and that character is smooth decay, complete diffusion prevails. In practice, decays (such as those of Fig. 11-9) with significant changes in character are more common, especially for the 63- and 125-Hz decays.

The beat information on the low-frequency reverberation decay makes possible a judgment on the degree of diffusion. The decays of Fig. 11-9 indicate that the diffusion of sound in this particular studio is about as good as can be achieved by traditional means. Reverberation-time measuring devices that yield information only on the average slope and not the shape of the decay omit information that most consultants consider important in evaluating the diffuseness of a space.

Exponential Decay

A truly exponential decay can be viewed as a straight line on a level (logarithmic scale) versus time plot, and the slope of the line can be described either as a decay rate in decibels per second or as a reverberation time in seconds. The decay of the 250-Hz octave band of noise pictured in Fig. 9-2 has two exponential slopes. The initial slope gives a reverberation time of 0.35 sec and the final slope a reverberation time of 1.22 sec. The latter slow decay when the level is low is probably due to a specific mode or group of modes encountering low absorption either by striking the absorbent at grazing angles or striking boundaries where there is little absorption. This is typical of one type of nonexponential decay, or stated more precisely, of a dual exponential decay.

Another type of nonexponential decay is illustrated in Fig. 9-3; the response deviates from a straight-line slope. This is a decay of an octave band of noise centered on 250 Hz in a 400-seat chapel, poorly isolated from an adjoining room. Decays taken in the

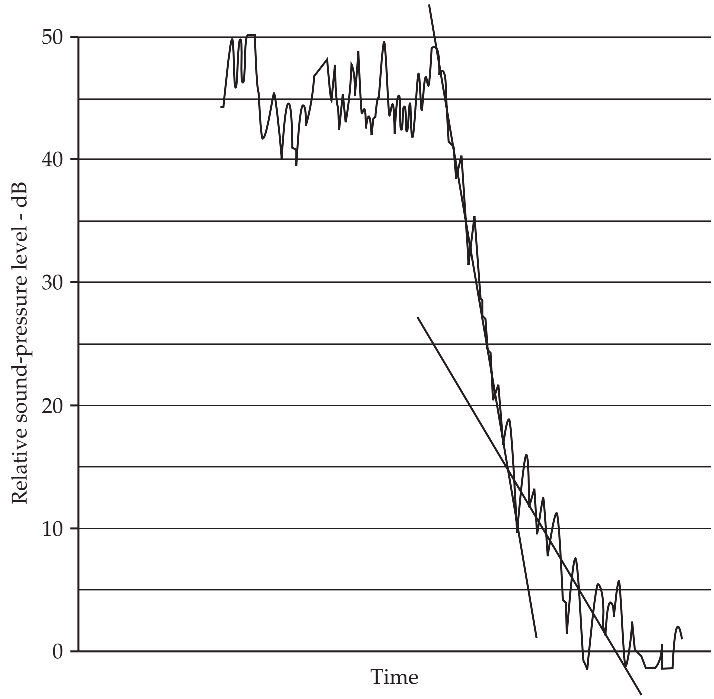


Figure 9-2 A typical double-slope decay, showing a lack of diffuse sound conditions. The slower decaying final slope is probably due to modes that encounter lower absorption.

presence of acoustically coupled spaces are characteristically concave upward (such as in Fig. 9-3) and often the deviations from the straight line are even greater. When the decay traces are nonexponential, that is, when they depart from a straight line in a level versus time plot, we conclude that true diffuse conditions do not prevail.

Spatial Uniformity of Reverberation Time

When reverberation time for a given frequency is reported, it is most accurately stated as the average of multiple observations at each of several positions in the room. This accounts for the fact that reverberatory conditions differ from place to place in the room. Figure 9-4 shows the results of measurements in a video studio of 22,000-ft^3 volume. The multiple uses of the space require variable reverberation time, which is accomplished with hinged wall panels that can be closed, revealing absorbent sides, or opened, revealing reflecting sides. Multiple reverberation decays were recorded at the same three microphone positions for both “panels reflective” and “panels absorptive” conditions. The open and filled circles show the average values for the reflective and absorptive conditions, respectively, and the solid, dashed, and dotted lines represent the average reverberation time at each of the three microphone positions. It is evident that there is considerable variation, which means that the sound field of the room is not completely homogeneous during this transient decay period. Inhomogeneities of the sound field

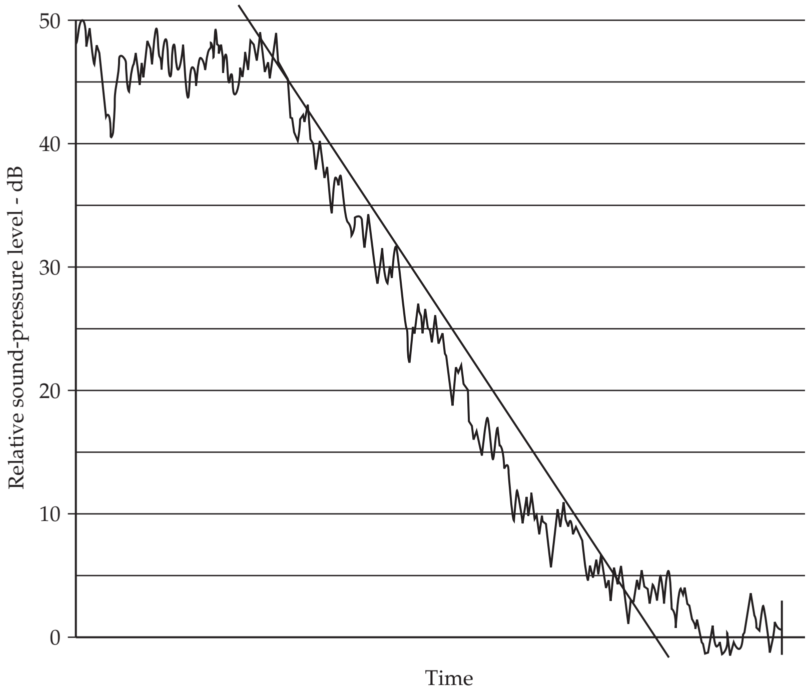


FIGURE 9-3 The nonexponential form of this decay is attributed to the acoustically coupled spaces in this example. The absence of a diffuse sound field is indicated.

are one reason why reverberation times vary from point to point in the room, but there are other factors as well. Uncertainties in fitting a straight line to the decay also contribute to the spread of the data, but this effect should be relatively constant from one position to another. It is reasonable to conclude that spatial variations in reverberation time are related, at least partially, to the degree of diffusion in the space.

Standard deviations of the reverberation times give us an insight of the spread of the data as measured at different positions in a room. When we calculate an average value, we lose all evidence of the spread of the data going into the average. The standard deviation can be used to quantify the data spread. Plus or minus one standard deviation from the mean value embraces 68% of the data points if the distribution is normal (Gaussian), and reverberation data should qualify reasonably well. Table 9-1 shows an analysis of the reverberation times for the video studio plotted in Fig. 9-4. For the "panels reflective" condition at 500 Hz, the mean reverberation time is 0.56 sec with a standard deviation of 0.06 sec. For a normal distribution, 68% of the data points would fall between 0.50 and 0.62 sec. That 0.06 standard deviation is 11% of the 0.56 mean. The percentages listed in Table 9-1 give us a rough appraisal of the precision of the mean.

The columns of percentage in Table 9-1 are plotted in Fig. 9-5. Variability of reverberation time values at the higher frequencies settles to reasonably constant values of around 3% to 6%. Because we know that each octave at high frequencies contains a large number of modes that results in smooth decays, we conclude that at higher audible

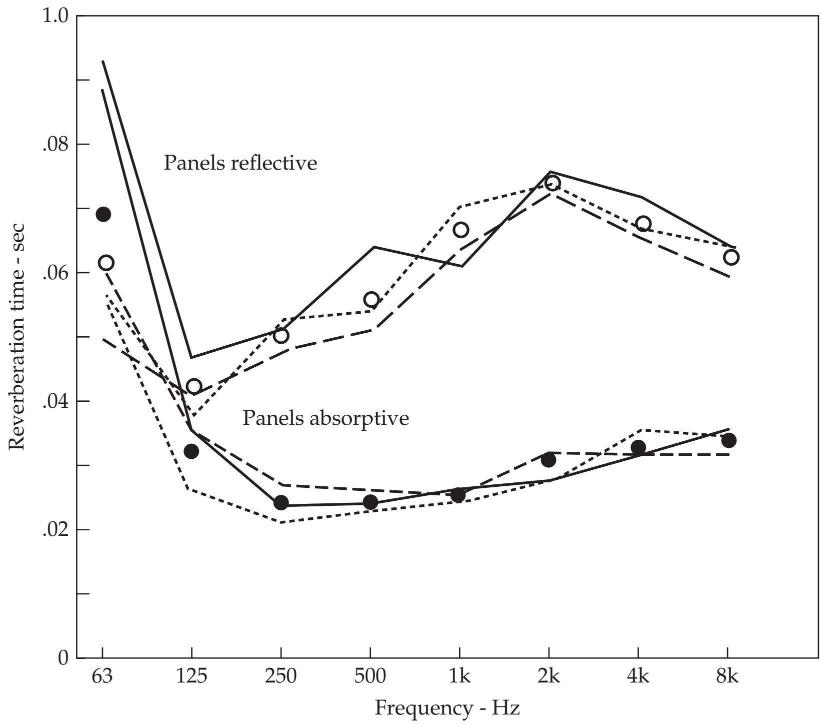


Figure 9-4 Reverberation time characteristics of a 22,000-ft^3 video studio with acoustics adjustable by hinged panels, absorbent on one side and reflective on the other. At each frequency, the variation of the average reverberation time at each of the three positions indicates nondiffuse conditions. This is particularly evident at low frequencies.

Octave-Band Center Frequency (Hz)	Panels Reflective			Panels Absorptive		
	RT_{60}	Standard Deviation	Standard Deviation % of Mean	RT_{60}	Standard Deviation	Standard Deviation % of Mean
63	0.61	0.19	31	0.69	0.18	26
125	0.42	0.05	12	0.32	0.06	19
250	0.50	0.05	10	0.24	0.02	8
500	0.56	0.06	11	0.24	0.01	4
1,000	0.67	0.03	5	0.26	0.01	4
2,000	0.75	0.04	5	0.31	0.02	7
4,000	0.68	0.03	4	0.33	0.02	6
8,000	0.63	0.02	3	0.34	0.02	6

Table 9-1 Analysis of the Reverberation Time of a Video Studio with a Volume of 22,000 ft^3

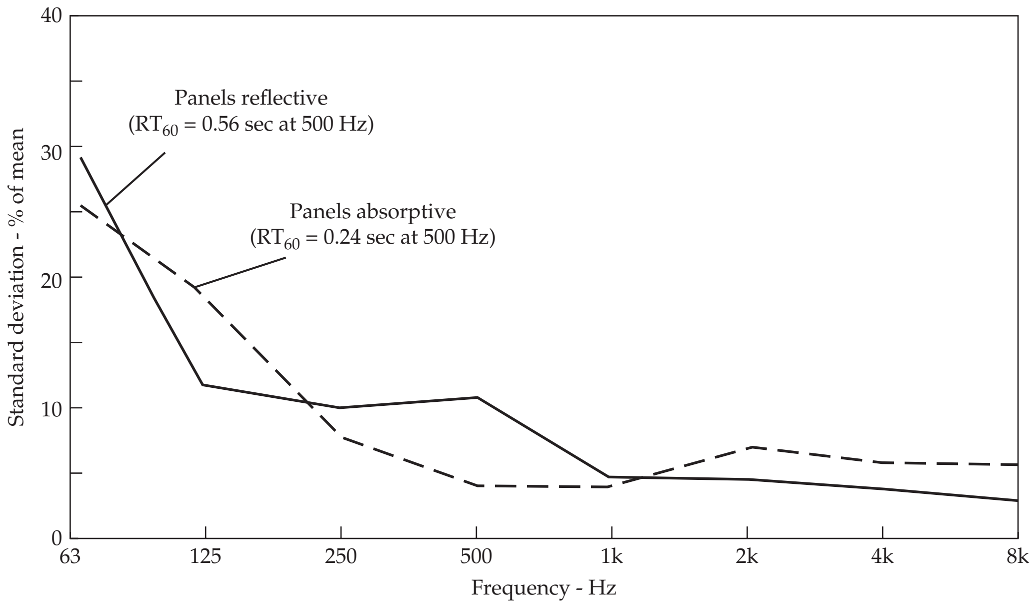


FIGURE 9-5 Graphical presentation of the reverberation time variations of the data of Table 9-1. The standard deviation, expressed as a percentage of the mean value, shows a lack of diffusion, especially below 250 Hz.

frequencies essentially diffuse conditions exist, and that the 3% to 6% variability is normal experimental measuring variation. At low frequencies, however, the high percentages (high variabilities) are the result of greater mode spacing producing considerable variation in reverberation time from one position to another. These high percentages include the uncertainty in fitting a straight line to the uneven decay characteristic of low frequencies. However, as Fig. 9-4 showed, there are great differences in reverberation time between the three measuring positions. Therefore, for this studio for two different conditions of absorbance (panels open/closed), diffusion is poor at 63 Hz, somewhat better at 125 Hz, and reasonably good at 250 Hz and above.

Geometrical Irregularities

Researchers have studied what types of wall protuberances provide the best diffusing effect. Somerville and Ward reported that geometrical diffusing elements reduced fluctuations in a swept-sine steady-state transmission test. They reported that the depth of such geometrical diffusers must be at least one-seventh of a wavelength before their effect is felt. They studied cylindrical, triangular, and rectangular elements and found that the straight sides of the rectangular-shaped diffuser provided the greatest effect for both steady-state and transient phenomena. Other experience indicates superior subjective acoustical properties in studios and concert halls in which rectangular ornamentation in the form of coffering is used extensively.

Absorbent in Patches

Applying all the absorbent in a room on one or two surfaces does not result in a more diffuse condition, and the absorbent is not used most effectively. Consider the results of an experiment showing the effect of distributing the absorbent. The experimental room is approximately a 10-ft cube and the room was tiled (certainly not ideal for a recording or listening room, but acceptable for this experiment). For test 1, reverberation time for the bare room was measured and found to be 1.65 sec at 2 kHz. For test 2, a common commercial absorber was applied to 65% of one wall (65 ft^2), and the reverberation time at the same frequency was found to be about 1.02 sec. For test 3, the same area of absorber was divided into four sections, one piece mounted on each of four of the room's six surfaces (one on each of three walls and one piece on the floor). This decreased the reverberation time to about 0.55 sec.

The area of the absorber was identical between tests 2 and 3; the only difference was that in test 3 it was in four pieces. By the simple expedient of dividing the absorbent and distributing it, the reverberation time was cut almost in half. Inserting the values of the reverberation time of 1.02 and 0.55 sec and the volume and area of the room into the Sabine equation (discussed in Chap. 11), we find that the average absorption coefficient of the room increased from 0.08 to 0.15 and the number of absorption units from 48 to 89 sabins. This extra absorption is due to an edge effect related to the diffraction of sound that makes a given sample appear to be much larger acoustically. Stated another way, in this example, the sound-absorbing efficiency of 65 ft^2 of absorbing material is only about half that of four 16-ft^2 pieces distributed about the room, and the edges of the four pieces total about twice that of the single 65-ft^2 piece. Therefore, one advantage of distributing the absorbent in a room is that its sound-absorbing efficiency is greatly increased, at least at certain frequencies. The preceding statements are true for 2 kHz, but at 700 Hz and 8 kHz, the difference between one large piece and four distributed pieces is small.

Another significant result of distributing the absorbent is that it contributes to the diffusion of sound. Patches of absorbent with reflective walls showing between the patches have the effect of altering wavefronts, which improves diffusion. For example, sound-absorbing modules placed along a wall distribute the absorbing material and improve their effect, and simultaneously contribute to the diffusion of sound.

Concave Surfaces

A concave surface such as that in Fig. 9-6A tends to focus sound energy and consequently should be avoided because focusing is the opposite of the diffusion we normally seek. The radius of curvature determines the focal distance; the flatter the concave surface, the greater the distance at which sound is concentrated. Such surfaces often cause problems in microphone placement. Concave surfaces might produce some awe-inspiring effects in a whispering gallery, but they are to be avoided in listening rooms and small studios.

Convex Surfaces: The Polycylindrical Diffuser

The polycylindrical diffuser (poly) is an effective diffusing element, and one relatively easy to construct; it presents a convex section of a cylinder. Three things can happen to sound falling on such a cylindrical surface made of plywood or hardboard: The sound

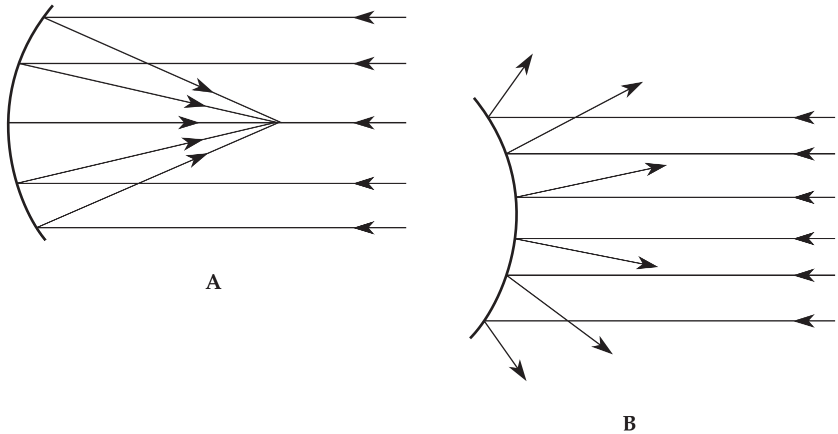


FIGURE 9-6 Generally, concave surfaces are undesirable, whereas convex surfaces are very desirable. (A) Concave surfaces tend to focus sound. Concave surfaces should be avoided if the goal is to achieve well-diffused sound. (B) Convex surfaces tend to diffuse sound.

can be reflected and thereby dispersed as in Fig. 9-6B; the sound can be absorbed; or the sound can be reradiated. Such cylindrical elements serve as absorbers in the low-frequency range where absorption and diffusion are often needed in small rooms. The reradiated portion, because of the diaphragm action, is radiated almost equally throughout an angle of roughly 120°, as shown in Fig. 9-7A. A similar flat element reradiates sound at a much narrower angle, about 20°, as shown in Fig. 9-7B. Therefore, reflection, absorption, and reradiation characteristics favor the use of the cylindrical surface.

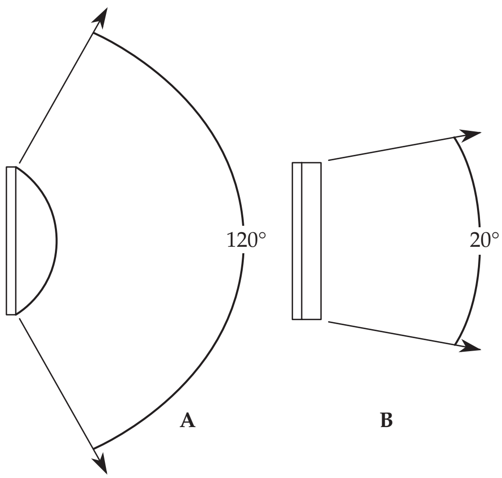


FIGURE 9-7 Polycylindrical diffusers, when properly designed, are very effective at providing wideband diffusion. (A) A polycylindrical diffuser reradiates sound energy not absorbed through an angle of about 120°. (B) A similar flat element reradiates sound at a much smaller angle of about 20°.

Some practical polys and their absorption characteristics are presented in Chap. 12. The dimensions of such diffusers are not critical, although to be effective their size must be comparable to the wavelength of the sound being considered. The wavelength of sound at 1,000 Hz is a bit over 1 ft and at 100 Hz it is about 11 ft. A poly element 3 or 4 ft across would be effective at 1,000 Hz, much less so at 100 Hz. In general, a poly base or chord length of 2 to 6 ft with depths of 6 to 18 in meet most needs. Axes of symmetry of the polys on different room surfaces should be mutually perpendicular.

It is important that diffusing elements be characterized by randomness. A wall lined with polys, all of a 2-ft chord and of the same depth, might be beautiful to behold but is not very effective for diffusion. The regularity of the structure would cause it to act as a diffraction grating, affecting one particular frequency in a much different way than other frequencies, which is counter to the ideal of wide-frequency diffusion.

Plane Surfaces

Geometrical sound diffusing elements made up of two flat surfaces to give a triangular cross section, or of three or four flat surfaces to give a polygonal cross section, may also be used. In general, their diffusing qualities are inferior to the cylindrical section.

Key Points

- In most room designs, diffusion is used to more effectively distribute sound and to provide a more equal response throughout a room.
- However, too much diffusion can make it difficult to localize the sound source.
- In practical rooms, sound fields are far from perfectly diffuse; the steady-state response can show significant frequency and spatial variations.
- The degree of diffusion in a room can be evaluated from the beat information in the low-frequency reverberation decay response.
- Similarly, decay traces that are nonexponential indicate that true diffuse conditions do not prevail.
- Spatial variations in reverberation time are related, at least partially, to the degree of diffusion in the space.
- According to one study, the depth of geometrical diffusers must be at least one-seventh of a wavelength before their effect is felt.
- Patches of absorbent with reflective walls showing between the patches have the effect of altering wavefronts, which improves diffusion.
- The polycylindrical diffuser is an effective diffusing element, presenting a convex section of a cylinder, and providing the beneficial characteristics of reflection, absorption, and reradiation.
- For improved effectiveness, diffusing elements should be physically characterized by randomness.

CHAPTER 10

Comb-Filter Effects

When a signal is combined with a delayed version of itself, constructive and destructive interference results in a frequency response with a series of regularly spaced nulls. Because of its appearance, this filtering effect is called a comb filter. In acoustics, comb filtering can occur when a signal combines with its delayed reflection. Comb filtering is a steady-state phenomenon. It has limited application to music and speech, which are highly transient phenomena. With transient sounds, the audibility of a delayed replica is more the result of successive sound events. A case might be made for comb-filter effects during brief periods of speech and music that approach steady state. However, the study of the audible effects of delayed reflections is better handled with the generalized threshold approach of sound reflection. Still, it is important to understand the nature of comb filtering, and know when it will, and will not, pose an acoustical problem.

Comb Filters

A filter changes the frequency response or transfer function of a signal. For example, an electronic filter might use active circuitry to attenuate low frequencies of a signal to reduce unwanted noise. A mechanical filter could use a system of ports and cavities to alter an acoustical signal, such as is used in some microphones to adjust the pickup pattern.

Electronic devices or algorithms can generate delayed replicas of sounds that are mixed with the original sound to create a comb-filter response in the output signal. Rather than using a fixed delay, the devices can continuously vary the delay to produce unique phasing and flanging effects. Whatever the means and application, these audible effects are the result of comb filters. In acoustics, comb filters are not purpose-built devices; instead, a comb-filter response is almost always an unwanted condition resulting from poorly controlled room reflections.

Superposition of Sound

Imagine a laboratory with a large tank of shallow water. Two stones are dropped in the tank simultaneously. Each stone causes circular ripples to flow out from the drop points. Each set of ripples expands through the other ripple pattern. We note that at any point in the water, the net effect is the combination of both ripple patterns at that point. As we will see later, both constructive and destructive interference will result. This is an example of superposition.

The principle of superposition states that every infinitesimal volume of a medium is capable of transmitting many discrete disturbances in many different directions, all simultaneously and with no detrimental effect on other disturbances. If you were able to observe and analyze the motion of a single air particle at a given instant under the influence of several disturbances, you would find that its motion is the vector sum of the various particle motions required by each of the disturbances passing by. At that instant, the air particle moves with an amplitude and a direction of vibration to satisfy the requirements of each disturbance just as a water molecule responds to each of several disturbances in the ripple tank.

For example, at a point in space, assume an air particle responds to a passing disturbance with a given amplitude and 0° direction. At the same instant another disturbance requires the same amplitude, but with a 180° direction. This air particle satisfies both disturbances at that instant by not moving at all.

Tonal Signals and Comb Filters

A microphone is a passive instrument. Its diaphragm responds to fluctuations in air pressure that occur at its surface. If the rate of such fluctuations (frequency) falls within its operating range, it generates an output voltage proportional to the magnitude of the pressure. For example, if a 100-Hz acoustical tone actuates the diaphragm of a microphone in free space, a 100-Hz voltage appears at the microphone terminals. If a second 100-Hz tone, identical in pressure but 180° out of phase with the first signal, strikes the microphone diaphragm, one acoustically cancels the other and the microphone voltage falls to zero. If an adjustment is made so that the two 100-Hz acoustical signals of identical amplitude are in phase, the signals reinforce each other, and the microphone delivers a 100-Hz signal that twice the original output voltage, an increase of 6 dB. The microphone responds to the pressures acting on its diaphragm. That is, the microphone responds to the vector sum of air pressure fluctuations impinging upon it. This characteristic of the microphone can help us understand acoustical comb-filter effects.

A 500-Hz sine tone is shown as a frequency component in Fig. 10-1A. All of the energy in this pure tone is located at this frequency. Figure 10-1B shows an identical signal except it is delayed by 0.5 msec with respect to the signal of A. The signal has the same frequency and amplitude, but the timing is different. Consider both A and B as acoustical signals combining at the diaphragm of a microphone. Further, consider signal A to be a direct signal and B a reflection of A from a nearby side wall. What is the nature of the combined signal output by the microphone?

Because signals A and B are 500-Hz sine tones, both vary from a positive peak to a negative peak 500 times per second. Because of the 0.5-msec delay, these two tonal signals will not reach their positive or negative peaks at the same instant. Often along the time axis, both are positive or both are negative, and at times one is positive while the other is negative. When the sine wave of sound pressure representing signal A and the sine wave of sound pressure representing signal B combine, they produce another sine wave of the same frequency, but of different amplitude; this is shown in Fig. 10-1C.

Figure 10-1 shows the two 500-Hz tones as lines in the frequency domain. Figure 10-2 shows the same 500-Hz direct tone and delayed tones in the time domain. The delay is accomplished by applying the second tone to a delay device with different delay settings, and combining the original and the delayed tones.

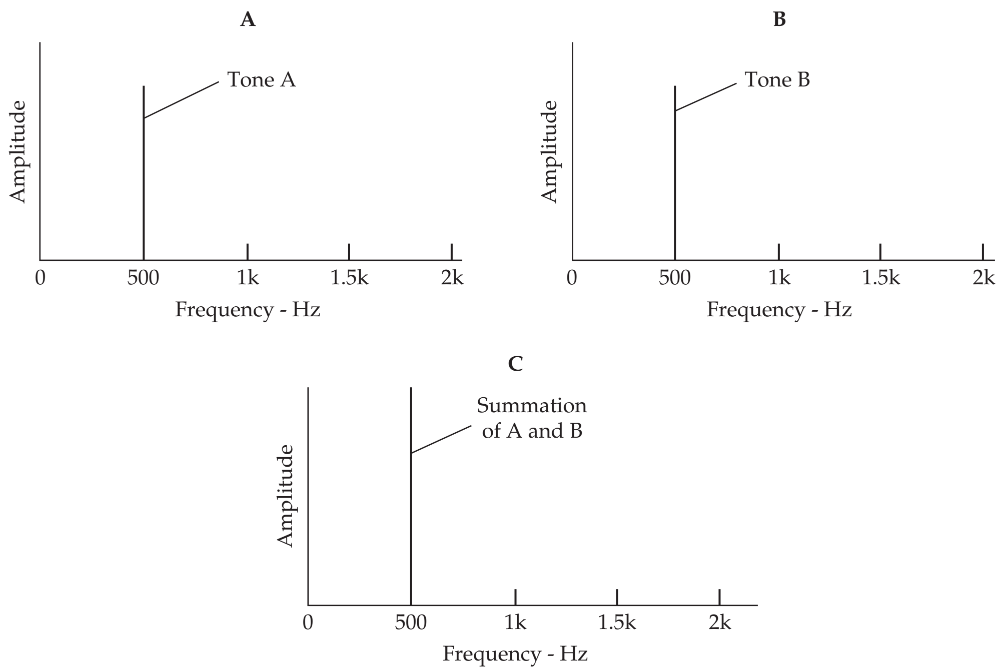


FIGURE 10-1 Tonal signals and time delay. (A) A sine wave having a frequency of 500 Hz. (B) Another sine wave of 500 Hz that is delayed 0.5 msec from A. (C) The summation of A and B. The 500-Hz signal and its delaye d counterpart reach their peaks at slightly different times, but adding them together simply yields another sine wave; there is no comb filtering. A linear frequency scale is used.

In Fig. 10-2A, the direct 500-Hz tone is shown originating at zero time. One cycle of a 500-Hz tone (1/500 = 0.002 sec) takes 2 msec. One cycle is also equivalent to 360°. The direct 500-Hz signal, e, is plotted according to the time and degree scales at the bottom of this figure.

At 500 Hz, a delay of 0.1 msec is equivalent to 18°; a delay of 0.5 msec is equivalent to 90°; a delay of 1 msec is equivalent to 180°. These three delayed signals (e_1, e_2, and e_3) are also shown in Fig. 10-2A.

Figure 10-2B shows the result when the direct signal is combined with each of the delayed signals. The combination of e and e_1 reaches a peak of approximately twice (+6 dB) that of e. A shift of 18° is a small shift, and e and e_1 are practically in phase. The combination of e and e_2, at 90° phase difference, has a lower amplitude, but is still a sine form. Adding e to e_3, with a shift of 180°, yields zero amplitude as adding two waves of identical amplitude and frequency but with a phase shift of 180° results in cancellation of one by the other.

Adding direct and delayed sine waves of the same frequency results in other sine waves of the same frequency. Adding direct and delayed sine waves of different frequencies gives periodic waves of irregular wave shape. Adding direct and delayed periodic waves does not create comb filtering. Comb filtering requires signals having distributed energy such as music, speech, and pink noise.

Signal	Delay-msec	Angle-degree
e	0	0
e_1	0.1	18
e_2	0.5	90
e_3	1.0	180

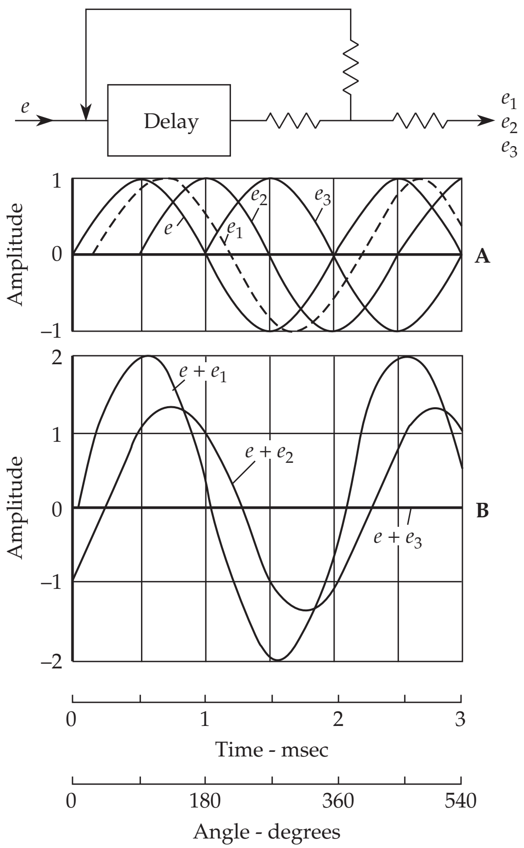


FIGURE 10-2 A demonstration showing the result of combining sine waves. (A) 500-Hz sine waves are displayed with delays of 0.1, 0.5, and 1.0 msec (conforming to the distributed spectrum cases in Fig. 10-4). (B) Combining sine waves such as these does not yield comb filtering, but simply other sine waves. A distributed spectrum is required for the formation of comb filtering. A linear frequency scale is used.

Comb Filtering of Music and Speech Signals

The spectrum of Fig. 10-3A can be considered an instantaneous segment of music, speech, or any other signal having a distributed spectrum. Figure 10-3B is essentially the same spectrum but delayed 0.1 msec from Fig. 10-3A. Considered separately, the delay difference is inconsequential. However, their summation produces a new result. Figure 10-3C is the acoustical combination of the A and B sound-pressure spectra at the

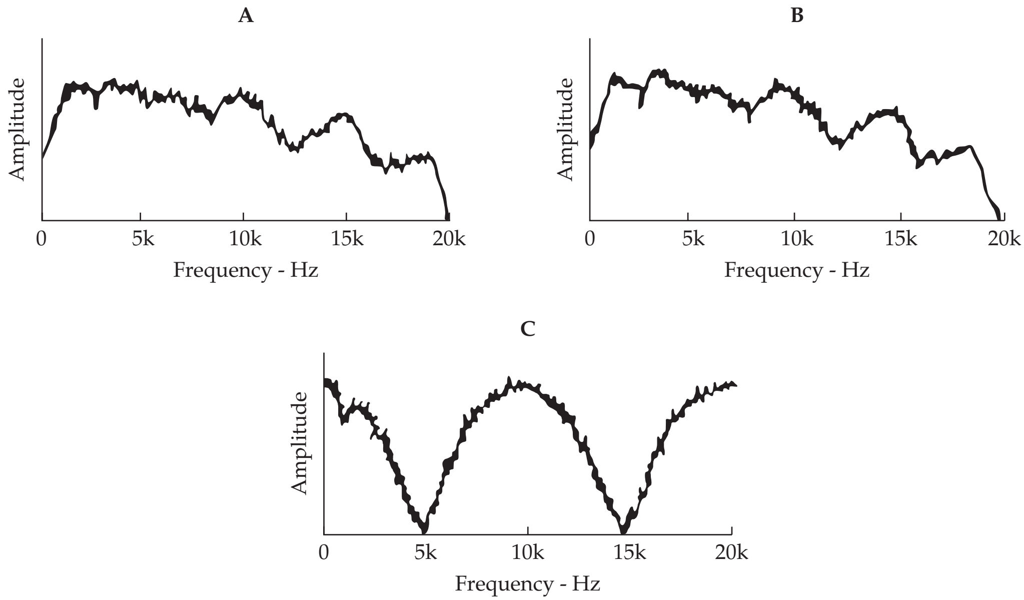


Figure 10-3 Comb filtering of signals having distributed spectra. (A) Instantaneous spectrum of music signal. (B) Replica of (A) which is delayed 0.1 msec. (C) The summation of (A) and (B) showing typical comb filtering. A linear frequency scale is used.

diaphragm of a microphone. The resulting response of Fig. 10-3C is different from the result of combining tonal signals. This response shows comb filtering with characteristic peaks (constructive interference) and nulls (destructive interference) in the frequency response. Plotted on a linear frequency scale, the pattern looks like a comb, hence the term, comb filter.

If a loudspeaker had a free-field frequency response like the one in Fig. 10-3C, it would certainly be rejected by most listeners. Yet many rooms are improperly designed, resulting in comb-filter interference. Fortunately, our brains are adept at interpreting a direct sound and multiple reflections so the perceived effect may be less severe than it looks. Even so, comb-filter distortion is a serious problem.

Comb Filtering of Direct and Reflected Sound

The 0.1-msec delay in Fig. 10-3 could have been from a digital-delay device, or because of the path-length difference of a reflection from a wall or other object. In the case of reflection, the spectral shape of a signal will be changed somewhat upon reflection, depending on the angle of incidence, the acoustical characteristics of the reflecting surface, and so on. When a direct sound is combined with its reflection, a comb-filter pattern is produced, with characteristic nulls (also called notches) in the frequency response. Nulls result when two signals are out of phase; they are one-half wavelength apart in time. Peaks are produced when signals are in phase. The frequency of the peaks and nulls is determined by the delay between the direct and reflected signals. The frequency of the first null occurs where the period is twice the delay time, that is, $f = 1/(2t)$ Hz, where t is the delay in seconds. The frequency of the first peak is twice the frequency of the first null. Furthermore, the spacing between successive nulls, and the spacing between successive peaks, is $1/t$ Hz. Examples of these calculations are given later in this chapter.

A reflection delayed by 0.1 msec will have traveled (1,130 ft/sec) (0.0001 sec) = 0.113 ft farther than the direct signal. This difference in path length, only about 1.35 in, could result from a grazing angle with both source and listener, or microphone, close to the reflecting surface. Greater delays are expected in more normal situations such as those of Fig. 10-4. The spectrum of Fig. 10-4A is from a random noise generator driving a loudspeaker and received by an omnidirectional microphone in free space. Noise of this type is widely used in acoustical measurements because it is a continuous signal; its energy is distributed throughout the audible frequency range, and it is more similar to speech and music signals than sine or other periodic waves.

In Fig. 10-4B, the loudspeaker faces a reflective surface; the microphone diaphragm is placed about 0.7 in from the reflective surface. Interference occurs between the direct sound the microphone picks up from the loudspeaker and the sound reflected from the surface. The output of the microphone shows the comb-filter pattern characteristic of a 0.1-msec delay. The combination of the direct and the reflected rays shows cancellations at 5 and 15 kHz (and every 10 kHz).

Placing the microphone diaphragm about 3.4 in from the reflective barrier, as shown in Fig. 10-4C, yields a 0.5-msec delay, which results in the comb-filter pattern shown. Increasing the delay to 0.5 msec has increased the number of peaks and the number of nulls fivefold. In Fig. 10-4D, the microphone is 6.75 in from the reflective barrier, giving a delay of 1.0 msec. Doubling the delay has doubled the number of peaks and nulls. As noted, if t is the delay in seconds, the frequency of the first null is $1/(2t)$ Hz and the spacing of successive nulls is $1/t$ Hz.

Increasing the delay between the direct and reflected components increases the number of constructive and destructive interference events proportionally. Starting with the flat spectrum of Fig. 10-4A, the spectrum of B is distorted by the presence of a reflection delayed by 0.1 msec. An audible response change would be expected. One might suspect that the distorted spectrum of D might be less noticeable because the multiple, closely spaced peaks and narrow nulls tend to average out the overall response aberrations.

Reflections following closely after the arrival of the direct component are expected in small rooms because of the short delays created by the room. Conversely, reflections in large spaces would have longer delays, which generate more closely spaced comb-filter peaks and nulls. Thus, comb-filter effects resulting from reflections are more commonly associated with small-room acoustics. The size of various concert halls and auditoriums renders them relatively immune to audible comb-filter distortions; the peaks and nulls are so numerous and packed so closely together that they merge into an essentially uniform response. Figure 10-5 illustrates the effect of passing a music signal through a 2-msec comb filter. The relationship between the nulls and peaks of response is related to several musical notes as indicated. Middle C, (C_4), has a frequency of 261.63 Hz and is close to the first null of 250 Hz. The next higher C, (C_5), has a frequency twice that of C_4 and is treated favorably with a +6-dB peak. Other Cs up the keyboard will be either discriminated against with a null or favored with a peak in response, or something in between. Whether viewed as fundamental frequencies or a series of harmonics, the timbre of the sound suffers.

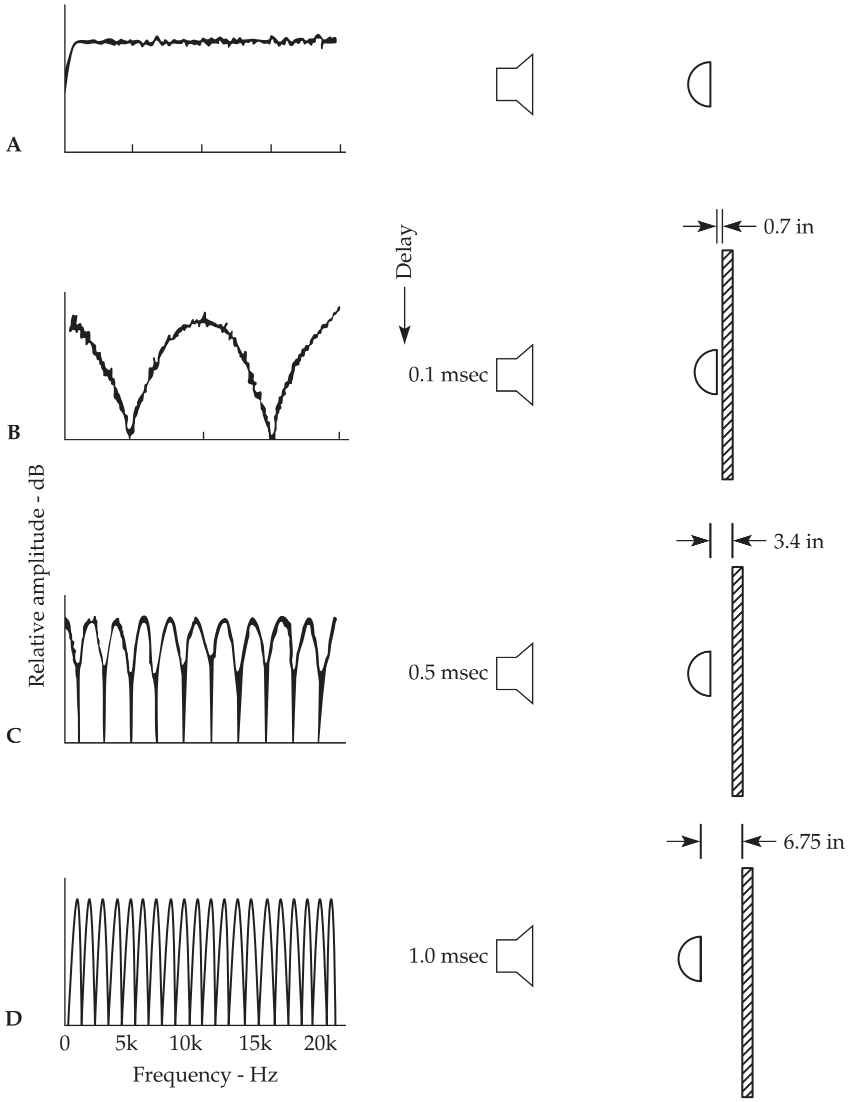


Figure 10-4 A demonstration of comb filtering in which direct sound from a loudspeaker is acoustically combined with a reflection from a surface at the diaphragm of an omnidirectional microphone. (A) No surface, no reflection. (B) Placing the microphone 0.7 in from the surface creates a delay of 0.1 msec. This short delay creates broadly spaced cancellations. (C) A delay of 0.5 msec creates cancellations that are much closer together. (D) A delay of 1 msec results in cancellations that are even closer together. A linear frequency scale is used.

The comb filters illustrated in Figs. 10-3 to 10-5 are plotted on a linear frequency scale. In this form, the comb appearance and visualization of the delayed effects are most graphic. A logarithmic-frequency scale, however, is more common in the electronics and audio industry, and is more representative of what we hear. A comb filter resulting from a delay of 1 msec plotted to a logarithmic frequency scale is shown in Fig. 10-6.

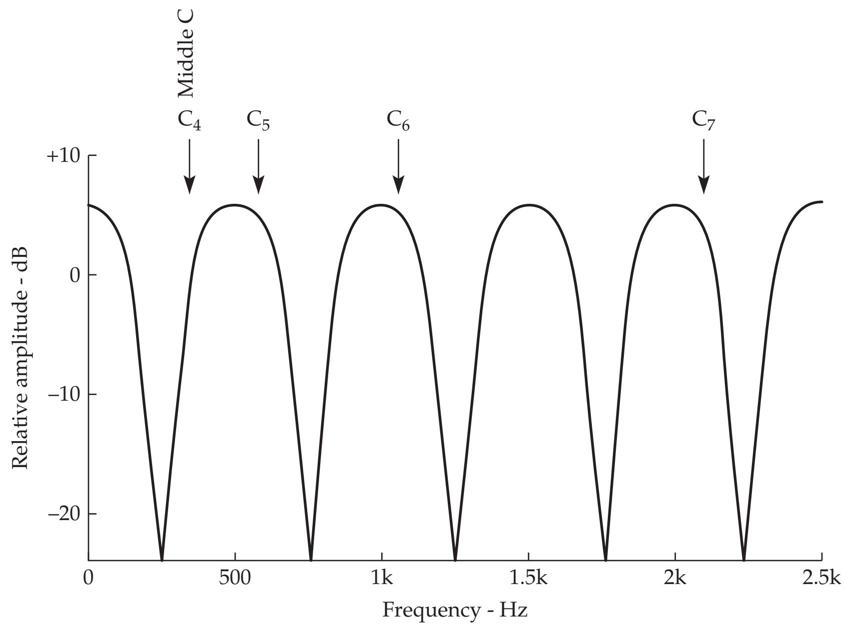


FIGURE 10-5 Passing a music signal through a 2-msec comb filter affects the components of that signal. Components spaced one octave can be boosted 6 dB at a peak or essentially eliminated at a null, or can be given values between these extremes. A linear frequency scale is used.

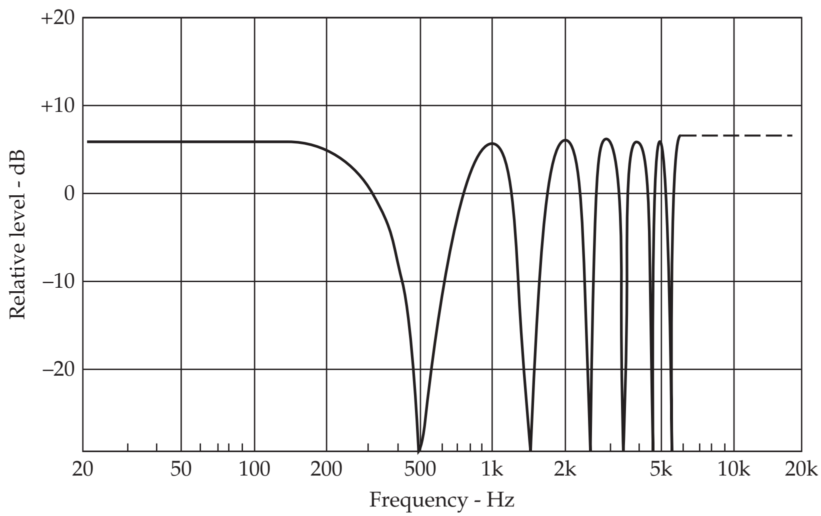


FIGURE 10-6 Plotting to the more familiar logarithmic scale helps estimate the effects of comb filtering on a signal.

Comb Filters and Critical Bands

One way to evaluate the relative audibility of comb-filter effects is to consider the critical bands of the human ear. The critical bandwidths at representative frequencies are shown in Table 10-1. The bandwidths of the critical bands vary with frequency. For example, the critical bandwidth of the human ear at 1 kHz is about 128 Hz. A peak-to-peak comb-filter frequency of 125 Hz corresponds to a reflection delay of about 8 msec, which corresponds to a difference in path length between the direct and reflected components of about 9 ft (1,130 ft/sec × 0.008 sec = 9.0 ft). This case for an 8-msec delay is plotted in Fig. 10-7B; two comb peaks fall within a critical band. Figure 10-7A shows a shorter delay of 0.5 msec; the width of the auditory critical band is comparable to one comb peak. Figure 10-7C shows a longer delay of 40 msec; the width of the critical band is large, relatively, so that no analysis of the comb filter is possible.

These examples tend to confirm the observation that in large spaces (long delays) comb filters are inaudible, whereas they often are very troublesome in small spaces (short delays). Also, critical bands are much narrower at low frequencies, suggesting that comb-filter effects are more audible at low frequencies.

The relative coarseness of the critical bands suggests that the ear is relatively insensitive to the peaks and nulls resulting from a 40-msec delay (see Fig. 10-7C). Therefore, the human ear may not interpret response aberrations such as timbral changes resulting from 40-msec combing, or combing from longer delays. On the other hand, the combing resulting from the 0.5-msec delay (see Fig. 10-7A) could be delineated by the ear's critical band at 1,000 Hz, resulting in a perceived timbre of the signal. Figure 10-7B illustrates an intermediate example in which the ear may be marginally able to analyze the 8-msec combed signal. The width of the critical bands of the auditory system increases rapidly with frequency. It is difficult to imagine the complexity of the interaction between a set of critical bands and a constantly changing music signal, with diverse combing patterns from a host of reflections. Only carefully controlled psychoacoustics experiments can determine whether the resulting differences are audible.

For similar reasons we note that a 1/3-octave frequency response is widely used because it is a good indicator of subjective frequency balance; this is because the 1/3-octave response approximates the critical bandwidth accuracy of our hearing.

Center Frequency (Hz)	Width of Critical Band* (Hz)
100	38
200	47
500	77
1,000	128
2,000	240
5,000	650

*Calculated equivalent rectangular band.

Table 10-1 Auditory Critical Bands (*Moore and Glasberg*)

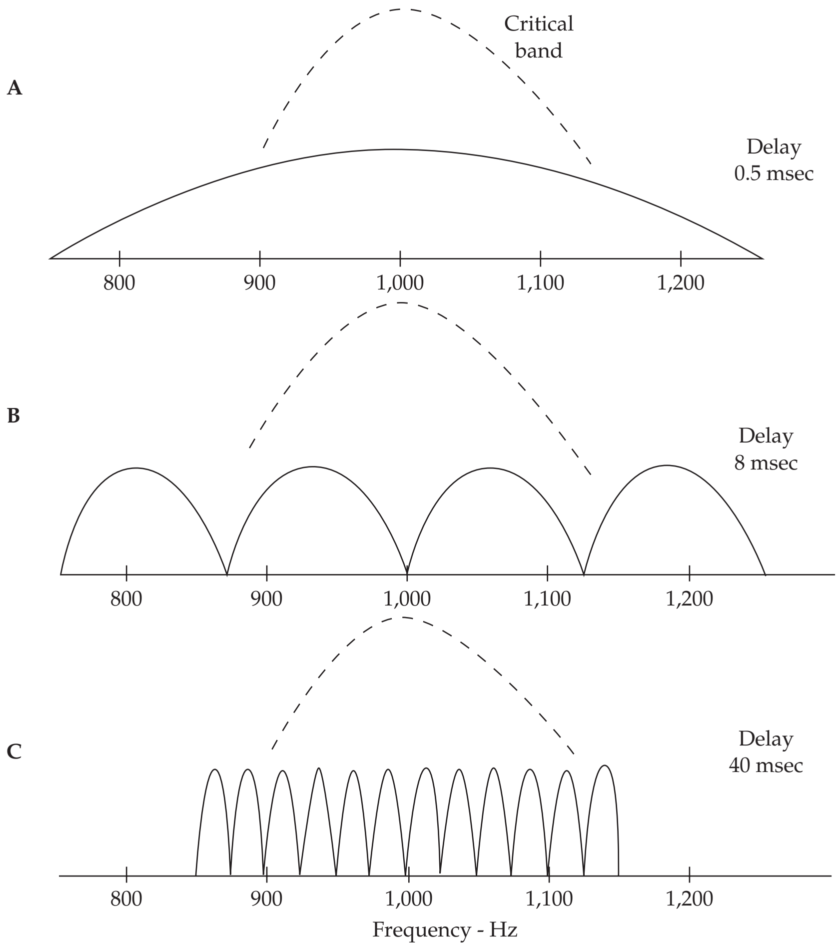


Figure 10-7 To estimate the perceptual significance of comb filters, three delay examples are compared to the auditory critical band effective at a frequency of 1,000 Hz. (A) Delay of 0.5 msec. (B) Delay of 8 msec. (C) Delay of 40 msec. A linear frequency scale is used.

When a high-resolution frequency-response measurement is converted into 1/3 octave, the sharp peaks and dips previously found in the frequency response are diminished, leaving the major variations in frequency response. This response more closely corresponds to the way we perceive sounds. For example, at high frequencies, the effects of reflections are often not perceived as important frequency-response problems, but rather as spatial imaging problems.

Comb Filters in Multichannel Playback

In multichannel playback, for example, in standard stereo playback, the input signals to each ear come from two loudspeakers. These signals are displaced in time with respect to each other because of the loudspeaker spacing; the result is the generation of comb filters. Blauert indicated that comb-filter distortion is not generally audible. As the perception of timbre is formed, the auditory system disregards these distortions; this is called binaural suppression of differences in timbre. However, no generally accepted theory exists to explain how the auditory system accomplishes this. Distortion can be heard by plugging one ear; however, this destroys the stereo effect. Comparing the timbre of signals from two loudspeakers (producing comb-filter distortion) to one loudspeaker (that does not) will demonstrate that stereo comb-filter distortion is barely audible. The timbre of the two is essentially the same. Furthermore, the timbre of the stereo signal changes little as the head is turned.

Controlling Comb Filtering

In most rooms, the effects of comb filtering are controlled by attenuating and diffusing room reflections. Absorption attenuates sound energy in reflections from room boundaries, whereas diffusion distributes reflections over time. Both approaches are commonly used and produce somewhat different perceptions of the sound field. Because absorption reduces the effect of specular reflections, the resulting sound field may have more distinct spatial phantom images. In contrast, diffusion may produce images with more perceived spaciousness. This sense of envelopment can be controlled by using different types of diffusers. With either approach, reflection or diffusion control must be broadband so that it does not alter the frequency response of the reflected or diffused sound. A good room design will combine both broad bandwidth absorption and diffusion.

When a loudspeaker is the sound source, comb filtering can be controlled through a room design that takes into account the loudspeaker's placement and directivity, and minimizes boundary reflections. As before, any reflection or diffusion control must be broadband. Since the directivity of most loudspeakers decreases as frequency decreases, special attention must be given to low-frequency reflection and diffusion control. These techniques are discussed in more detail in subsequent chapters.

Reflections and Spaciousness

A reflected sound reaching the ear of a listener is always somewhat different from the direct sound. There are numerous causes for this. The reflecting sound will be changed by the frequency characteristics of the reflecting wall. By traveling through the air, both the direct and reflected components of a sound wave are altered slightly, due to the air's absorption of sound, which varies with frequency. The amplitude and timing of the direct and reflected components differ. The human ear responds to the frontal, direct component somewhat differently than to the lateral reflection from the side. The perception of the reflected component is always different from the direct component. The amplitudes and timing will be related, but with an interaural correlation less than the maximum.

Weakly correlated input signals to the ears contribute to the impression of spaciousness in terms of width, depth, and height. If no reflections occur, such as when listening outdoors, there is no feeling of spaciousness. If a room supplies "correct" input signals to the ears, the perception of the listener is that of being completely enveloped and immersed in the sound. The lack of strong correlation is a prerequisite for the impression of spaciousness.

Comb Filters in Microphone Placement

When two microphones separated in space pick up a single sound at slightly different times, their combined output will be similar to the single microphone with delayed reflections. Therefore, spaced microphone stereo-pickup arrangements are susceptible to comb-filter problems. Under certain conditions the combing is audible, imparting phasiness to the overall sound reproduction, interpreted by some as room ambience. It is not ambience, however, but distortion of the time and intensity cues presented to the microphones. It is evident that some people find this distortion pleasing, so spaced microphone pickups are favored by many producers and engineers.

Comb-Filter Effects in Practice: Six Examples

Example 1 Figure 10-8 shows three microphone placements that produce comb filters of varying degrees. A hard, reflective floor is assumed in each case; other room reflections are not considered. The numerical results of these placements are given in Table 10-2. In a close source-to-microphone setup, the direct component travels 1 ft and the floor-reflected component travels 10.1 ft. The difference between these (9.1 ft) means that the floor reflection is delayed 8.05 msec (9.1/1,130 = 0.00805 sec). The first null is therefore at 62 Hz with subsequent null and peak spacing of 124 Hz. The level of the reflection is –20 dB referred to the direct component (20 log 1.0/10.1 = 20 dB). The direct component is thus 10 times stronger than the floor reflection. The effect of the comb filter would be negligible in this case.

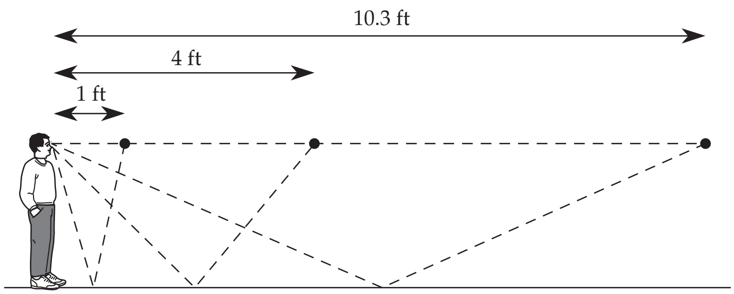


Figure 10-8 Common microphone placements can produce comb-filter effects (see Table 10-2). At a distance of 1 ft, the reflection level of –20 dB yields minimum comb-filter problems. At a distance of 4 ft, the reflection level of –8 dB may yield some comb-filter problems. (C) At a distance of 10.3 ft, the reflection level almost equals the direct level, and thus comb-filter problems are certain.

Path Length (ft)		Difference				
Direct	Reflected	Distance (ft)	Time (msec)	First Null 1/(2*t*) (Hz)	Peak/Null Spacing 1/*t* (Hz)	Reflection Level (dB)
1.0	10.1	9.1	8.05	62	124	–20
4.0	10.0	6.0	5.31	94	189	–8
10.3	11.5	1.2	1.06	471	942	–1

Table 10-2 Comb-Filter Effects from Microphone Placement (Refer to Fig. 10-8)

The two other microphone placements (see Fig. 10-8) yield lower reflection levels (see Table 10-2). A source-to-microphone distance of 4 ft presents an intermediate case, where the reflection level is 8 dB below the direct-signal level; the comb-filter effect would be marginal. With a source-to-microphone distance of 10.3 ft, the reflection-level difference is only 1 dB; the reflection is almost as strong as the direct signal; the comb-filter effect would be considerable. In contrast to these microphone placements, consider what would happen if the microphone was placed directly on the floor some distance from the source. This technique would essentially eliminate the difference between the direct and reflected path length.

Example 2 Figure 10-9 shows two microphones, for example, on a podium. Stereo reproduction systems are relatively rare in auditoriums; the chances are very good that the two microphones are fed into a monaural system and thus become an excellent producer of comb-filter effects. The common excuse for two microphones is to give the speaker greater freedom of movement or to provide a spare microphone in case of the failure of one. Assuming the microphones are properly polarized and the talker is dead center, there would be a helpful 6-dB boost in level. Assume also that the microphones are 24 in apart and the talker's lips are 18 in from a line drawn through the two microphones and on a level with the microphones. If the talker moves laterally 3 in, a 0.2-msec delay is introduced, attenuating important speech frequencies. If the talker does not move, the speech quality would probably not be good, but it would be stable. Normal talker movements shift nulls and peaks up and down the frequency scale with quite noticeable shifts in quality.

Example 3 Figure 10-10 shows comb-filter possibilities in a singing group with each singer holding a microphone. Each microphone is fed to a separate channel but ultimately mixed together. Each singer's voice is picked up by all microphones but only adjacent singers may create noticeable comb filters. For example, the voice of singer A is picked up by both microphones and mixed, and may produce a comb-filter response resulting from the path difference. However, if singer A's mouth is at least three times farther from singer B's microphone than from A's own microphone, the comb-filter effects are minimized. This "3:1 rule" works because maintaining this distance means that delayed replicas are at least 9 dB below the main signal. Comb-filter peaks and nulls are 1 dB or less in amplitude and thus essentially imperceptible.

Example 4 Figure 10-11 shows dual monaural loudspeakers, for example, one on stage left and the other on stage right comprising an auditorium sound system. Two sources radiating identical signals create a comb-filter response over the audience area. On the line of symmetry (often down the center aisle), both signals arrive at the same time and no comb-filter response is produced there. Equi-delay contours range out from stage center over the audience area, the 1-msec-delay contour nearest the center line of symmetry, and greater delays as the sides of the auditorium are approached.

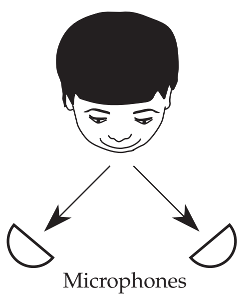


Figure 10-9 Comb filtering is produced with two microphones feeding into the same monaural amplifier with a sound source that moves about.

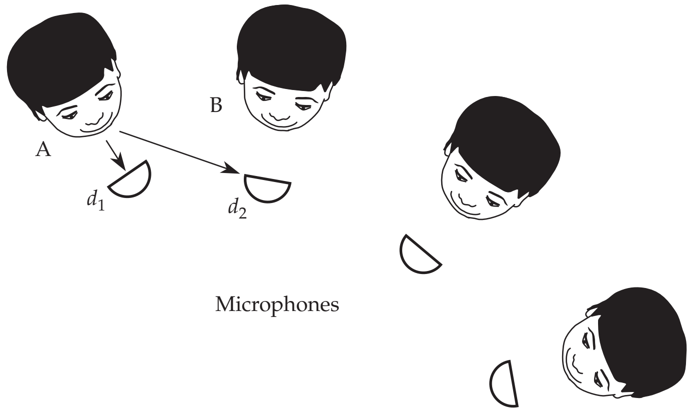


Figure 10-10 For a group singing with multiple microphones, if d_2 is at least three times greater than d_1, the comb-filter effect is minimized.

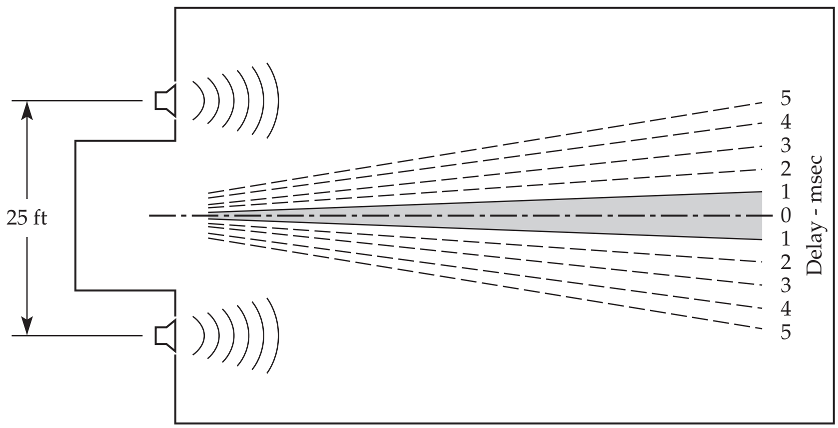


Figure 10-11 In the common split system in which two loudspeakers radiate identical signals, zones of constructive and destructive interference result which degrade sound quality in the audience area.

Example 5 Figure 10-12 shows the frequency-response bands in a three-way loudspeaker. Frequency f_1 is radiated by both low-frequency and midrange drivers, and both outputs may be essentially equal in magnitude; moreover, the two radiators are physically displaced. These are the ingredients for comb filtering. The same process is at work at f_2 between the midrange and high-frequency drivers. Only a narrow band of frequencies is affected, the width of which is determined by the relative amplitudes of the two radiations. The steeper the crossover curves, the narrower the frequency range affected.

Example 6 Figure 10-13 shows a microphone that is flush mounted on a table surface. One advantage is an approximate 6-dB gain in sensitivity due to the pressure rise at the surface. Another advantage is the minimization of comb-filter effects. A direct signal from the source strikes the microphone diaphragm; since the diaphragm is flush with the surface, there can be no reflections from the surface. Hence comb filtering of the frequency response is avoided.

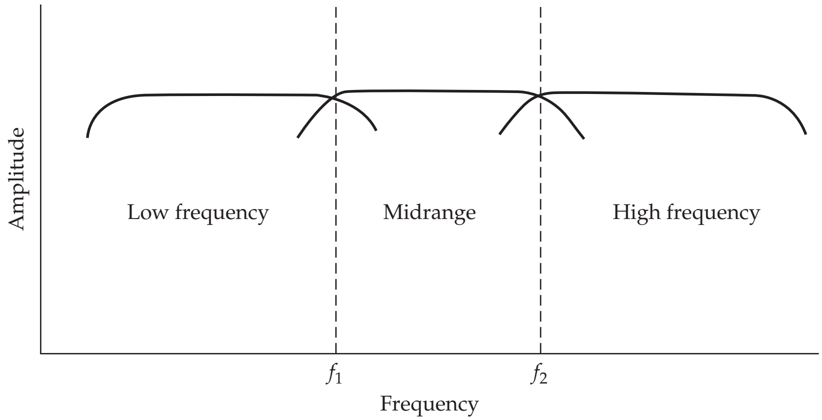


FIGURE 10-12 Comb-filter distortion can occur in the crossover regions of a three-way loudspeaker because the same signal is radiated from two physically separated drivers.

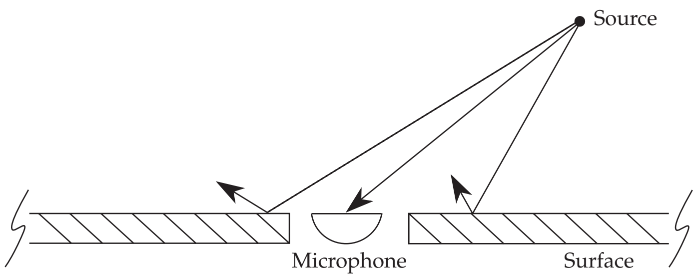


FIGURE 10-13 With a flush-mounted microphone, sound from the source *S* that strikes the surface does not reach the microphone, thus avoiding comb-filter effects. Another advantage of this mounting is a level increase due to the pressure buildup at the reflecting surface.

Estimating Comb-Filter Response

A few simple relationships can be used to estimate the effect of comb filters on the response of a signal. As noted, if the delay is t seconds, the frequency of the first null is $f = 1/(2t)$ Hz. We also observe that when $f = n/(2t)$, nulls recur at $n = 1, 3, 5$, and so on. Also, if the delay is t seconds, the spacing between nulls is $1/t$ Hz. For example, for a delay of 1 msec (0.001 sec) the frequency of the first null is $1/(2 \times 0.001) = 500$ Hz. The nulls are spaced 1,000 Hz apart ($1/0.001 = 1{,}000$ Hz) and thus recur at 500, 1,500, 2,500 Hz, and so on. Of course, between each adjacent pair of nulls, there is a peak at which the two signals are in phase. The frequency of the first peak is twice the frequency of the first null. Again noting that $f = n/(2t)$, peaks recur at $n = 2, 4, 6$, and so on. Also, as with null spacing, the spacing between peaks is $1/t$ Hz. For the same 1-msec delay, the frequency of the first peak is 1,000 Hz. The peaks are spaced 1,000 Hz apart and thus recur at 1,000, 2,000, 3,000 Hz, and so on. This and similar examples are shown in Table 10-3.

Adding two sine waves with the same frequency and the same amplitude, in phase, doubles the amplitude; yielding a peak 6 dB higher than either component by itself ($20 \log 2 = 6.02$ dB). The nulls will be at a theoretical minimum as they cancel at phase

Delay (msec)	Frequency of Lowest Null (Hz)	Spacing between Nulls and Spacing between Peaks (Hz)
0.1	5,000	10,000
0.5	1,000	2,000
1.0	500	1,000
5.0	100	200
10.0	50	100
50.0	10	20

Table 10-3 Comb-Filter Peaks and Nulls

opposition. In this way, the entire response curve can be sketched as the phase of the two waves alternates between the in-phase and the phase-opposition condition through the spectrum.

To summarize, in this example the $1/(2t)$ expression above gives a null at 500 Hz, which robs energy from any distributed signal subject to that delay. A music or speech signal passing through a system having a 1-msec delay will have components removed or reduced. This is another case of comb-filter distortion.

Figures 10-14 and 10-15 are included as graphical solutions for estimating comb-filter response.

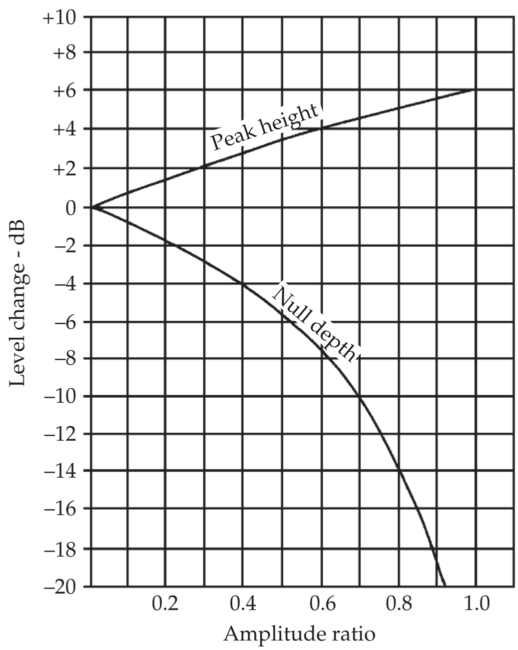


Figure 10-14 The effect of amplitude ratios on comb-filter peak height and null depth.

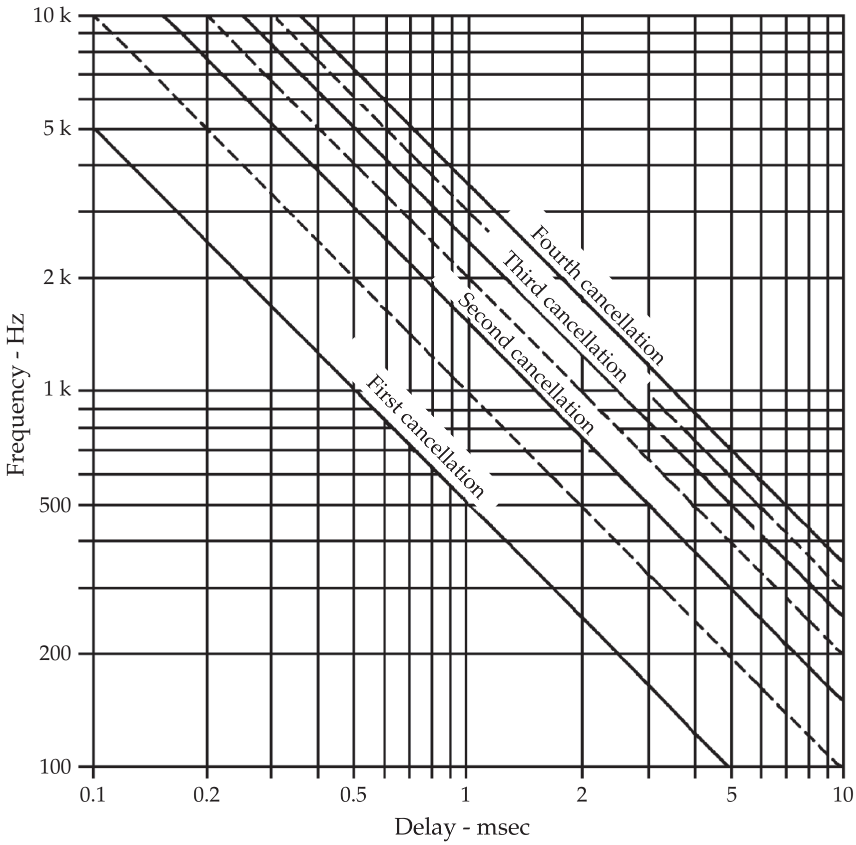


Figure 10-15 The magnitude of the delay determines the frequencies at which destructive interference (cancellations) and constructive interference (peaks) occur. The broken lines indicate the peaks between adjacent cancellations.

Key Points

- When a signal is combined with a delayed version of itself, the result is a comb-filter frequency response with a series of regularly spaced peaks and nulls.
- Comb filtering is a steady-state phenomenon and has limited application to music and speech.
- In acoustics, a comb-filter response is almost always an unwanted condition resulting from poorly controlled room reflections.
- Adding direct and delayed periodic waves does not create comb filtering. Comb filtering requires signals having distributed energy such as music, speech, and pink noise.
- The frequency of the peaks and nulls of a comb-filter response is determined by the delay between the direct and reflected sound; the frequency of the first null occurs where the period is twice the delay time.

- Comb-filter effects resulting from reflections are more commonly associated with small-room acoustics because of short delay times.
- The size of various concert halls and auditoriums renders them relatively immune to audible comb-filter distortions because of long delay times.
- Because human-hearing critical bands are much narrower at low frequencies, comb-filter effects may be more audible at low frequencies.
- Weakly correlated input signals to the ears contribute to the impression of spaciousness in a room.
- Spaced-stereo microphone arrangements are susceptible to comb-filter problems. Under certain conditions the combing can be audible, imparting phasiness to the overall sound reproduction.

CHAPTER 11
Reverberation

If you press the accelerator pedal of an automobile, the vehicle accelerates to a certain speed. If the road is smooth and level, this speed will remain constant. With a constant force on the accelerator, the engine produces enough horsepower to overcome frictional and aerodynamic losses, and a balanced (steady-state) condition results. If you take your foot off the accelerator, the car will gradually slow, and come to a stop.

Sound in a room behaves similarly. When a loudspeaker is turned on, it emits noise in a room that quickly grows to a certain level. This level is the steady-state or equilibrium point at which the sound energy radiated from the loudspeaker is enough to overcome losses in the air and at the room boundaries. A greater sound energy radiated from the loudspeaker will result in a higher equilibrium level, whereas less energy to the loudspeaker will result in a lower equilibrium level.

When the loudspeaker is turned off, it takes a finite length of time for the sound level in the room to decay to inaudibility. This decaying aftereffect of the sound in a room, after the excitation signal has been removed, is defined as reverberation. Reverberation time is measured from the moment when a signal is turned off, but clearly the effects of reverberation are always present when a noncontinuous signal is sounding in a room.

Reverberation has an important bearing on the acoustical quality of sound in a room. For example, a symphony orchestra recorded in a large anechoic chamber, with almost no room reverberation, would yield a recording of very poor quality for normal listening. This recording would be even thinner, weaker, and less resonant than most outdoor recordings of music, which are noted for the emptiness of their sound. Clearly, symphonic and other music requires reverberation to achieve an acceptable sound quality. Similarly, many music and speech sounds require a room's reverberant assistance to sound natural because we are accustomed to often hearing those sounds in reverberant environments.

Formerly, reverberation was considered the single most important characteristic of an enclosed space for speech or music. Today, although it is still very important, reverberation is considered one of several important and measurable parameters that define the sound quality of an acoustical space.

Growth of Sound in a Room

When a sound is initiated in a room, the room will contain the energy as it builds to a steady-state value. The time required to reach this steady-state value is determined by the rate of growth of sound in the room. In turn, this growth rate is determined by the level of energy at the source, and the room's acoustics.

Let us consider a source S and a listener L in a room, as shown in Fig. 11-1A. As source S is instantaneously energized, sound travels outward from S in all directions. Sound travels a direct path to the listener L and we shall consider zero time (Fig. 11-1B) as that time at which the direct sound reaches the ears of the listener. The sound pressure at L instantly jumps to a value D less than that which left S due to spherical divergence and small losses in the air. The sound pressure at L stays at this value until reflection R_1 arrives and then the sound pressure jumps to the $D + R_1$ value. Shortly thereafter R_2 arrives, causing the sound pressure to increase a bit more. In this analysis, the arrival of each successive reflected component causes the level of sound to increase stepwise. In practice, because of the great number of components, the energy growth occurs smoothly. Also, these additions are, in reality, vector additions involving both magnitude and phase, but we are keeping things simple for the purpose of illustration.

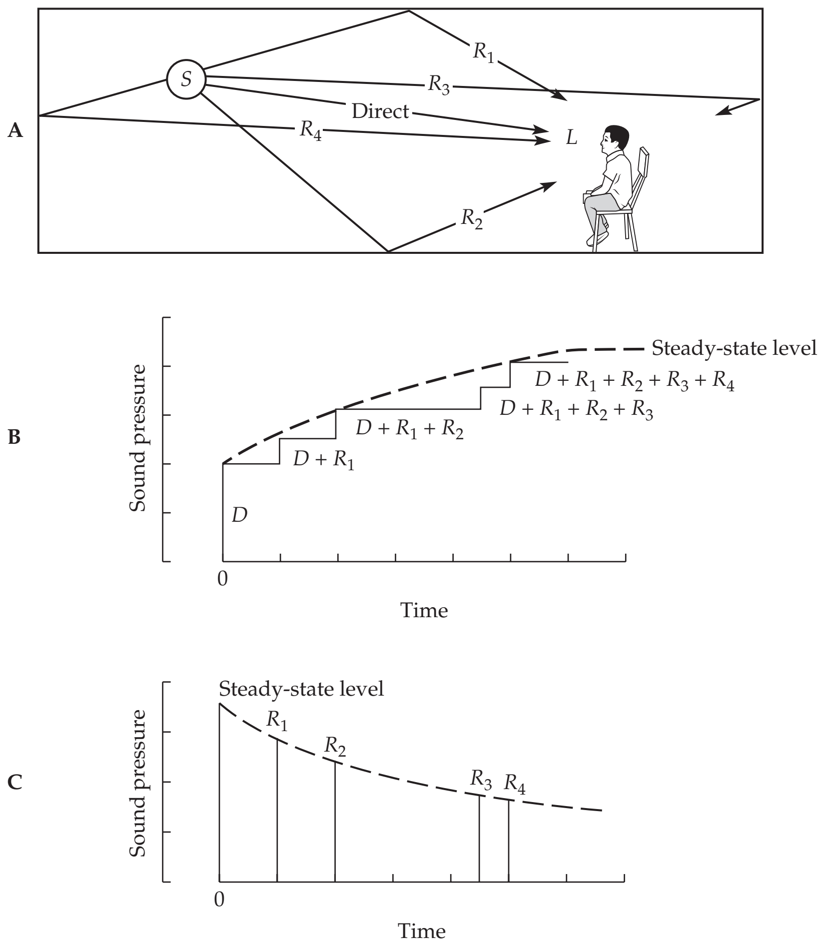


FIGURE 11-1 The growth and decay of sound in a room. (A) The direct sound arrives first at time $t = 0$, with reflected components arriving later. (B) In this analysis, the sound pressure at L grows stepwise. In practice, the growth curve appears as a smooth (dashed) line. (C) The sound decays exponentially after the source ceases.

Sound pressure at the listener grows as one reflected component after another adds to the direct sound and previous components. The sound pressure at *L* does not instantly go to its final value because sound travels by paths of varying lengths. Given the speed of sound, reflected components are delayed by an amount proportional to the difference in distance between the reflected paths and the direct path. The growth of sound in a room is thus relatively slow due to finite transit time, but in practice, sound growth is so fast as to be perceived as being instantaneous by a listener. On the other hand, in comparison, sound decay is very slow and is usually readily heard (as reverberation) by a listener. Therefore, the characteristics of sound decay are more important in practical room acoustical design.

The ultimate level of sound in the room is determined by the energy from the source *S*. The energy it radiates is dissipated as heat in wall reflections and other boundary losses, along with a small loss in the air itself. With a constant input to *S*, the sound-pressure level grows, as shown in Fig. 11-1B, to a steady-state equilibrium. Increasing the input to the source *S* yields a new equilibrium of room-sound-pressure level versus room losses.

Decay of Sound in a Room

After turning off source *S*, the room is momentarily still filled with sound energy, but equilibrium is destroyed because the losses are not balanced by energy from *S*. Support is cut off to the rays of sound moving through the room. Sound energy in the room begins to decay.

What is the fate, for example, of the ceiling reflected component R_1 (see Fig. 11-1A)? As *S* is cut off, R_1 is on its way to the ceiling. It loses energy at the ceiling reflection and heads toward *L*. After passing *L* it strikes the rear wall, then the floor, the ceiling, the front wall, the floor again, and so on, losing energy at each reflection. Soon it is so weak that it can be considered dead. The same thing happens to R_2, R_3, R_4, and a multitude of other sound rays not shown. Figure 11-1C shows the exponential decrease of the first reflection components, which would also apply to the wall reflections not shown and to the many multiple reflection components. The sound in the room thus dies away because of losses at reflections, the damping effect of the air, and divergence. However, it takes a finite time to do so because of the speed of sound and the path lengths dictated by the room's dimensions.

Idealized Growth and Decay of Sound

Purely from the view of geometrical (ray) acoustics, the decay of sound in a room, as well as its growth, is a stepwise phenomenon. However, in the practical world, the great number of small steps involved results in smooth growth and decay of sound. Idealized forms of growth and decay of sound in a room are shown in Fig. 11-2A. Here the sound pressure is shown on a linear scale and is plotted against time. Figure 11-2B shows the same growth and decay, except that sound-pressure level is plotted in decibels, that is, on a logarithmic scale.

During the growth of sound in a room, power is being applied to the sound source. During decay, the power to the source has been stopped, hence the difference in the shapes of the growth and decay curves. The decay of Fig. 11-2B is a straight line in this idealized form, and this becomes the basis for measuring the reverberation time of an enclosed space.

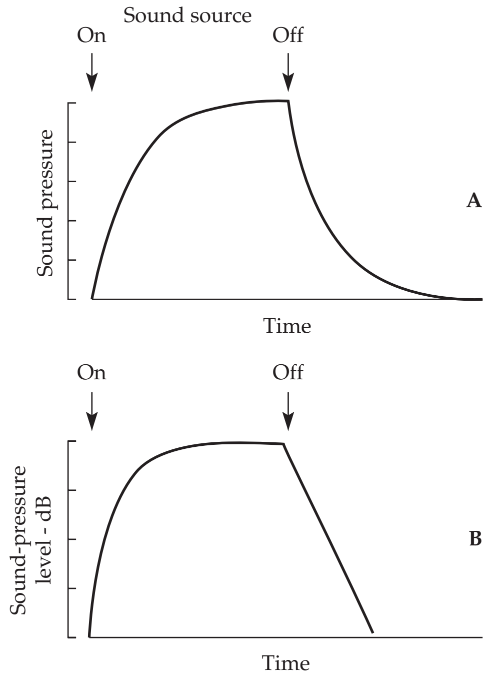


Figure 11-2 The growth and decay of sound in a room. (A) Sound pressure on a vertical scale is measured in linear units. (B) Sound-pressure level on a vertical scale is measured in logarithmic units (decibels).

Calculating Reverberation Time

Reverberation time (RT) is defined as the time in seconds required for sound intensity in a room to drop 60 dB from its original level. This represents a change in sound intensity or sound power of 1 million (10 log 1,000,000 = 60 dB), or a change of sound pressure or sound-pressure level of 1 thousand (20 log 1,000 = 60 dB). This reverberation time measurement is referred to as RT_{60}. The 60-dB figure was chosen arbitrarily, but it roughly corresponds to the time required for a loud sound to decay to inaudibility. The equation used to calculate reverberation time, often called the Sabine equation, is presented in the next section.

One approach to measuring reverberation time is illustrated in Fig. 11-3A. Using a recording device that gives a trace of the decay, the time required for the 60-dB decay can be measured. While simple to measure in theory, in practice, problems can be encountered. For example, obtaining a straight decay spanning 60 dB or more as in Fig. 11-3A is a difficult practical problem. Background noise suggests that a higher source level might be needed. For example, this would occur if the background noise level is 30 dB (as in Fig. 11-3A). Source levels greater than 90 dB are quite attainable. However, if the noise level is near 60 dB, as shown in Fig. 11-3B, a source level greater than 120 dB is required. If a 100-W (watt) amplifier driving a loudspeaker gives a sound-pressure level of 100 dB at the required distance, doubling the power of the source

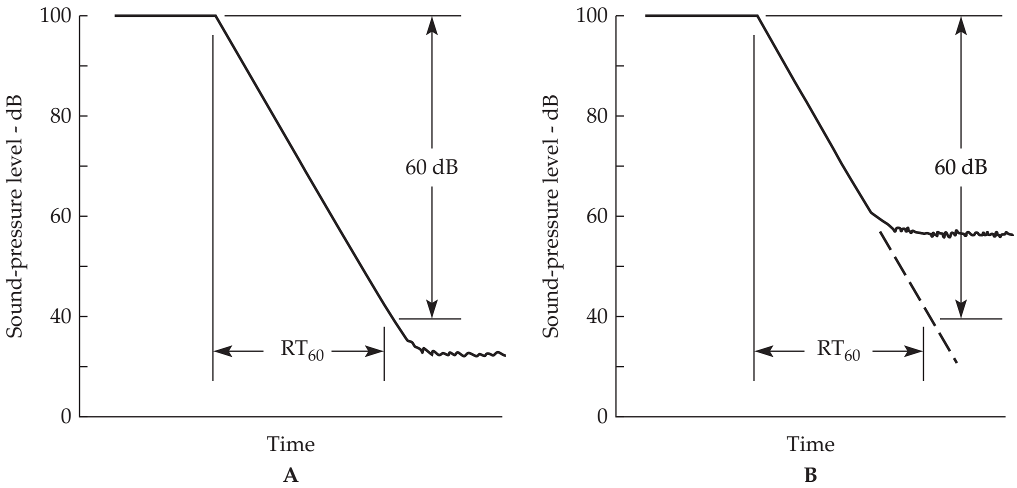


FIGURE 11-3 The length of the decay trace depends on the strength of the source and the noise level. (A) An example of a full 60-dB decay. Practical circumstances rarely allow this. (B) The slope of the limited decay is extrapolated to determine the reverberation time.

increases the sound-pressure level by only 3 dB; hence 200 W gives 103 dB, 400 W gives 106 dB, 800 W gives 109 dB, and so on. The limitations of size and cost can set a limit on the maximum levels in a practical case.

The situation of Fig. 11-3B is commonly encountered, where circumstances yield a trace showing less than the desired 60 dB range. The solution is to extrapolate the usable initial portion of the decay. For example, the time required for sound pressure to fall 30 dB is extended to yield an estimate for the time required to fall 60 dB.

Actually, it is important to strive for the greatest decay range possible because we are interested in the entire extent of the decay. For example, it has been demonstrated that in evaluating the quality of speech or music, the human ear is most sensitive to the first 20 or 30 dB of decay. On the other hand, the significance of double-slope phenomena (see Fig. 9-2) is revealed only near the end of the decay. In practice, the highest amplitude of sound source reasonably attainable is used, and filters can be incorporated to improve the signal-to-noise ratio of the measurement.

Sabine Equation

In the 1890s, Wallace Clement Sabine, a Harvard University physics professor, was asked to remedy the poor acoustics of Fogg Lecture Hall on the campus. At the time, the study of room acoustics was not well developed. Sabine studied the interplay of absorption and reverberation. He used a portable wind chest and organ pipes as a sound source, a stopwatch, and keen ears to measure the time from the interruption of the source to inaudibility with different amounts of absorption. For the latter, he and his assistants borrowed hundreds of seat cushions from the nearby Sanders Theater. After years of work, in 1900 he subsequently published the first reverberation-time equation—an equation we still use today. In many respects, Sabine founded the field of modern architectural acoustics.

Sabine observed that reverberation time depends on room volume and absorption. The greater the absorption, the shorter the reverberation time. Likewise, the greater the room volume, the longer the reverberation time because sound will strike the absorbing room boundaries less often.

The quantity $4V/S$ describes the average distance a sound travels between two successive reflections; it is sometimes called the mean free path. In the equation, V is the room volume, and S is the room surface area. For example, consider a room measuring $23.3 \times 16 \times 10$ ft. The volume is 3,728 ft^3 and the surface area is 1,533 ft^2. The mean free path for this room is $4V/S$ or (4)(3,728)/1,533 = 9.7 ft. Since the speed of sound is 1,130 ft/sec, on average a sound ray will travel 8.5 msec before striking another room surface. It might take between four to six reflections before the sound energy in this sound ray is mainly lost, a decay process that would take 42.5 msec. But the number of reflections actually required to deplete the sound energy will depend on how absorptive the room is. For example, the decay process will require more reflections in a live room with low absorption, and thus take longer. In addition, because the mean free path is longer in large rooms, the decay process will take longer in large rooms.

Using this kind of statistical theory and geometrical ray acoustics, Sabine devised his equation for room reverberation. In particular, he devised the following relationship:

$$\mathrm{RT}_{60} = \frac{0.049V}{A} \tag{11-1}$$

where RT_{60} = reverberation time, sec
V = volume of room, ft^3
A = total absorption of room, sabins

The Sabine equation can also be expressed in metric units:

$$\mathrm{RT}_{60} = \frac{0.161V}{A} \tag{11-2}$$

where RT_{60} = reverberation time, sec
V = volume of room, m^3
A = total absorption of room, metric sabins

Using the Sabine equation is a straightforward process, but a few notes are worth considering. The quantity A is the total absorption in the room. For example, this accounts for the absorption of room surfaces. (In many cases, audience absorption must be considered as well, and if desired, a value for air absorption may be included, as described in the following text.) It would be easy to obtain the total absorption if all the room surfaces were uniformly absorbing, but this condition rarely exists. Walls, floor, and ceiling may be covered with quite different materials, and the doors, and windows must be considered separately as well. The unit of sound absorption used in the formula, the sabin, is named after Professor Sabine, and is very close to the absorption provided by one Sanders Theater seat cushion. Note that 1 sabin = 0.093 metric sabin.

The total absorption A in a room can be found by considering the amount of absorption contributed by each type of surface. This can be calculated using the absorption coefficient α which characterizes the absorptivity of a material. The absorption coefficient α ranges from 0 to 1.0 where 1.0 represents complete absorption. To obtain the total absorption A of the room, it is necessary to combine the respective absorptions of the various materials in the room by multiplying the square-foot area S_i of each type of material by its respective absorption coefficient α_i, and summing the result to obtain total absorption. In particular, $A = \Sigma S_i\alpha_i$, where i represents each of the surface areas and its respective absorption coefficient. The quantity $\Sigma S_i\alpha_i/\Sigma S_i$ represents the average absorption coefficient α_{average}.

For example, let us say that an area S_i (expressed in square feet or square meters) is covered with a material having an absorption coefficient α_1, as obtained from the table in App. C. This area then contributes $(S_1)(\alpha_1)$ absorption units, in sabins, to the room. Likewise, another area S_2 is covered with another kind of material with absorption coefficient α_2, and it contributes $(S_2)(\alpha_2)$ sabins of absorption to the room. The total absorption in the room is $A = S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \ldots$, etc. With a figure for A in hand, it is a simple matter to use Eq. (11-1) or (11-2) and calculate the reverberation time. This is further demonstrated in the examples at the end of this chapter.

The absorption coefficients of practically all materials vary with frequency. For this reason, it is necessary to calculate total absorption at different frequencies. A typical reference frequency for reverberation time is 500 Hz, and 125 Hz and 2 kHz are used as well. To be precise, any reverberation time calculation should be accompanied by an indication of frequency. For example, a reverberation time at 125 Hz might be quoted as $\text{RT}_{60/125}$. When there is no frequency designation, the reference frequency is assumed to be 500 Hz.

A word about the limitations of the Sabine equation is in order. For live rooms, the assumed statistical conditions prevail and the Sabine equation gives accurate results. However, in very absorbent rooms, the equation produces erroneous results. For example, consider again the room measuring $23.3 \times 16 \times 10$ ft. This room has a volume of 3,728 ft^3 and a total surface area of 1,532 ft^2. If we further assume that all of its surfaces are perfectly absorbent ($\alpha = 1.0$), we see that total absorption is 1,532 sabins. Substituting these values:

$$\text{RT}_{60} = \frac{(0.049)(3{,}728)}{1{,}532} = 0.119 \text{ sec}$$

Clearly, since the correct RT_{60} value in a perfectly absorbing room should be zero, the equation is in error. Perfectly absorbing walls would allow no reflections. This paradox results from assumptions upon which the Sabine equation is derived; in particular, it assumes that sound in a room is diffuse, as in a live room. As a result, the Sabine equation is most accurate in live rooms where the average absorption coefficient is less than 0.25.

Eyring-Norris Equation

Eyring and Norris, and others, have presented reverberation equations intended to overcome the limitations of the Sabine equation, and can be used accurately in more absorptive rooms. They are used for rooms in which the average absorption coefficient is greater than 0.25. For rooms with an average absorption coefficient of 0.25 or less, these equations are basically equivalent to the Sabine equation.

In particular, Eyring and Norris proposed an alternative equation for absorptive rooms:

$$\text{RT}_{60} = \frac{0.049V}{-S\ln(1-\alpha_{\text{average}})} \tag{11-3}$$

where RT_{60} = reverberation time, sec
V = volume of room, ft^3
S = total surface area of room, ft^2
ln = natural logarithm (to base "e")
$\alpha_{average}$ = average absorption coefficient ($\Sigma S_i \alpha_i / \Sigma S_i$)

Young points out that the absorption coefficients published by materials manufacturers (such as listed in App. C) are Sabine coefficients and can be applied directly in the Sabine equation. Young recommends that Eq. (11-1) or (11-2) be used for engineering computations rather than the Eyring-Norris equation or its several derivatives. Two unassailable reasons for this are simplicity and consistency. Many technical writings offer the Eyring-Norris or other equations for studio use. There is authoritative backing for using Eyring-Norris for more absorbent spaces, but commonly available coefficients apply only to Sabine. For this reason, we use the Sabine equation in this volume. Other researchers have suggested alternative reverberation equations; these include Hopkins-Striker, Millington, and Fitzroy.

Air Absorption

In large rooms, where sound travels over long path lengths, the passage through air can effectively add absorption to the room, lowering reverberation times. Air absorption is only significant at higher frequencies—above 2 kHz. Air absorption is not a significant factor in small rooms, and can be neglected. When accounted for, a term $4mV$ is added to the denominator of reverberation time equations, where m is the air attenuation coefficient in sabins per foot (or sabins per meter) and V is room volume in cubic feet (or cubic meters). For example, the Sabine and Eyring-Norris equations, respectively, are modified as:

$$RT_{60} = \frac{0.049V}{A + 4mV} \tag{11-4}$$

$$RT_{60} = \frac{0.049V}{-S\ln(1-\alpha_{average}) + 4mV} \tag{11-5}$$

Some values of m in sabins per foot are: 0.003 at 2 kHz, 0.008 at 4 kHz, and 0.025 at 8 kHz; and in sabins per meter, 0.009, 0.025, and 0.080, respectively. The value of m depends on relative humidity of air, and the values given apply to RH between 40% and 60%. Air absorption increases at low humidity.

Measuring Reverberation Time

There are many approaches to measuring the reverberation time of a room, and many instruments are available to serve those who have an interest in the effects of reverberation. For example, sound contractors need to know the approximate reverberation time of the spaces in which they are to install a sound-reinforcement system, and measuring it may be more conclusive than calculating it. Measurements can be more accurate than calculations because of inherent uncertainty in absorption coefficients. Acoustical consultants called upon to correct a problem space or verify a carefully designed and newly constructed space, generally lean toward the method of recording multiple sound decays. These sound decays give a wealth of detail meaningful to the practiced eye.

Impulse Sources

Both impulse sources and steady-state sources can be used to examine a room's response. Any sound source used to excite an enclosure must have enough energy throughout the spectrum to ensure that decays are sufficiently raised above the noise floor to give the required accuracy. Examples of impulse sound sources are powerful electrical spark discharges, pistols firing blanks, and balloon bursts. For large spaces, even small cannons have been used as impulse sources to provide adequate energy, especially at lower frequencies. Whatever the actual method of producing an impulse, the impulse decay, the room's response measured over time, can be used to examine the temporal characteristics of sound in a room. For example, the more diffuse the room's sound field, the smoother the decay response. Conversely, for example, a room with echoes will yield a nonuniform decay response as energy is concentrated at the times when echoes appear.

The impulse decays for a small studio room are shown in Fig. 11-4. The sound source was an air pistol that ruptures paper discs. The peak sound-pressure level at 1-m distance is 144 dB, and the duration of the major pulse is less than 1 msec. Such a pulse is ideal for recording echograms.

In Fig. 11-4, the straight, upward traveling segment on the left side is the same slope for all decays because it is a result of instrumentation limitation. The useful measure of reverberation is the downward traveling, more irregular slope on the right. This slope yields a reverberation time after the manner of Fig. 11-3. Note that the octave-band noise level is higher for the lower frequency bands. The impulse is barely above the noise floor for the 250 Hz and lower octaves. This is often a major limitation of the method.

Steady-State Sources

As noted, steady-state sources can be used to examine a room's response; for example, as noted, Sabine used a wind chest and organ pipes for his early measurements. However, care must be taken to choose a steady-state source that provides accurate response data.

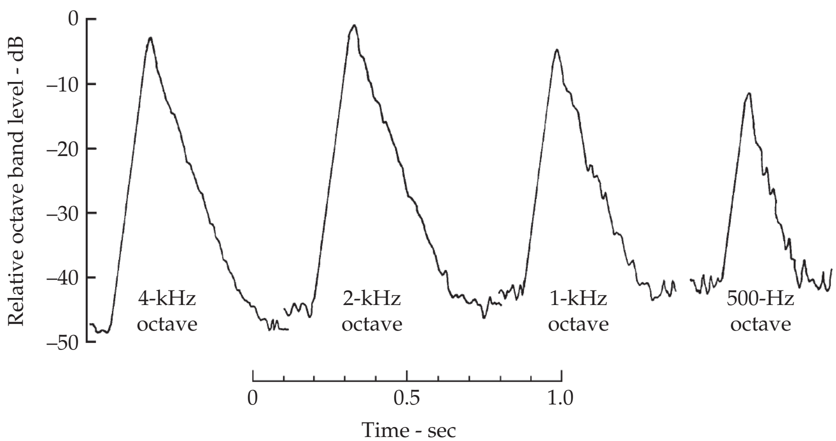


FIGURE 11-4 Reverberatory decays produced by impulse excitation of a small studio. The upgoing left side of each trace is instrumentation-limited; the downgoing right side is the reverberatory decay.

Sine-wave sources providing energy at a single frequency give highly irregular decays that are difficult to analyze. Warbling a tone, which spreads its energy over a narrow band, is an improvement over the fixed tone, but random noise sources are preferred. Bands of random (white or pink) noise give a steady and dependable indication of the average acoustical effects taking place within that particular part of the spectrum. Octave and 1/3-octave bands of random noise are commonly used. Steady-state sources are useful for measuring the growth of sound in a room, as well as the decay response.

Measuring Equipment

The equipment layout shown in Fig. 11-5 can be used to obtain reverberation decays. In this example, a wideband pink-noise signal is amplified and applied to a loudspeaker. A switch for interrupting the noise excitation is provided. By aiming the loudspeaker into a corner of the room (especially in smaller rooms), all resonant modes are excited; this is because all modes terminate in the corners.

An omnidirectional microphone is positioned on a tripod, usually at ear height for a listening room, or microphone height for a room used for recording. The smaller the microphone capsule, generally the less its directional effects. Some larger-capsule microphones (e.g., 1-in-diameter diaphragms) can be fitted with random incidence correctors, but using a smaller microphone (e.g., 1/2-in-diameter diaphragm) is preferred for more uniform sensitivity to sound arriving from all angles. Dedicated sound-level meter/analyzers use a high-quality, small capsule condenser microphone mounted on a stalk that spaces it away from the main body of the meter. The Brüel & Kjaer (B&K) 2245 sound-level meter/analyzer is an example of such an instrument. Devices such as

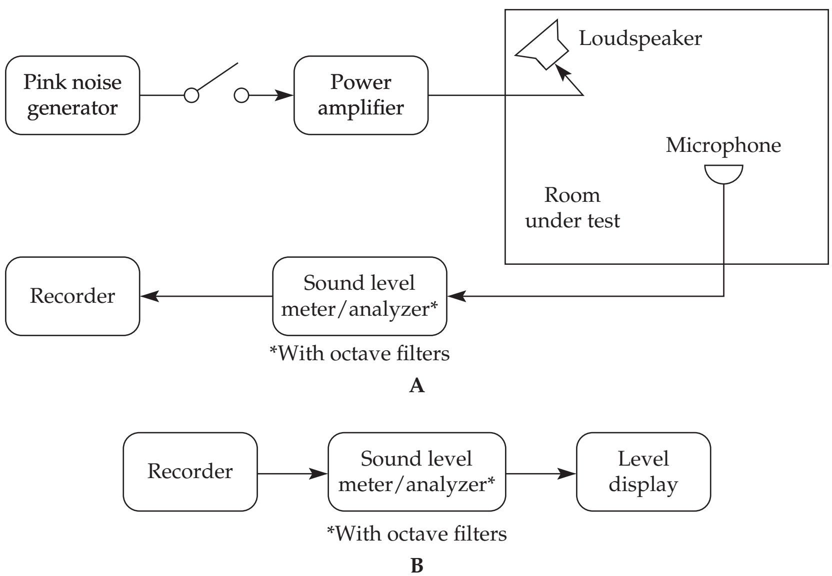


Figure 11-5 Equipment configuration for measuring the reverberation time of a room. (A) Recording decays on location. (B) Later decay playback for analysis.

these can be used to measure and analyze reverberation and are also used for broadband measurements, general-purpose sound-pressure level measurements, and occupational, product, and environmental noise measurements.

Measurement Procedure

The measurement procedure begins when wide-band pink noise is at a steady-state condition in the room. The level is usually loud enough to require the use of ear protectors for everybody in the room. When the noise is stopped, the sound in the room decays into silence. The microphone, at its selected position, picks up this decay, which is stored for later analysis.

The signal-to-noise ratio determines the length of the reverberatory decay available for study. As noted, it is often not possible to realize the entire 60-dB decay used in the definition of RT_{60}. It is quite possible, however, to obtain 45- to 50-dB decays with the equipment shown in Fig. 11-5 by the simple expedient of double filtering. For example, to obtain an RT_{60} measurement at 500 Hz, the octave filter centered on 500 Hz in the sound-level meter is used both in recording and in later playback.

The analysis procedure outlined in Fig. 11-5B uses recorder playback a sound-level meter/analyzer, and a graphic display. The data from the recorder is applied to the sound-level meter/analyzer and its output is applied to the graphic display, completing the equipment interconnection. In practice, these devices can be combined into one instrument. The appropriate octave filters are switched in as the reverberatory decay data is analyzed.

Reverberation and Normal Modes

The natural resonances of a room are revealed in its normal modes, as described in Chap. 13. It is necessary to somewhat anticipate this topic to understand the relationship of these natural resonances with the reverberation of the room. For now, simply stated, most rooms have preferred resonance frequencies where sound energy at those frequencies is accentuated.

The Sabine equation or alternatives are widely used to characterize a room's reverberation time. However, a single-number calculation or measurement cannot completely describe the behavior of room reverberation, particularly when considering the effects of normal modes of rooms. Room modes pose particular difficulties when characterizing the reverberation time of small rooms.

Consider an untreated, small studio room. Starting with a sine-wave oscillator set to about 20 Hz below the first axial mode, the acoustics of the room do not load the loudspeaker and a relatively weak sound is produced with the amplifier gain turned up full (even assuming the use of a good subwoofer). As the oscillator frequency is adjusted upward, however, the sound becomes very loud as the (1, 0, 0) mode (24.18 Hz in this example) is energized, as shown in Fig. 11-6. Adjusting the oscillator upward, we go through a weak valley, but at the frequency of the (0, 1, 0) mode (35.27 Hz), there is high-level sound once more. Similar peaks are found at the (1, 1, 0) tangential mode (42.76 Hz), the (2, 0, 0) axial mode (48.37 Hz), and the (0, 0, 1) axial mode (56.43 Hz). These peaks and valleys are due to room modes.

Now that the loudness of peaks and valleys of the room's modes have been noted, let's examine the decay of sound. After exciting the (1, 0, 0) mode at 24.18 Hz to steady-state conditions, the sound source is switched off; the decay time of the resulting reverberation

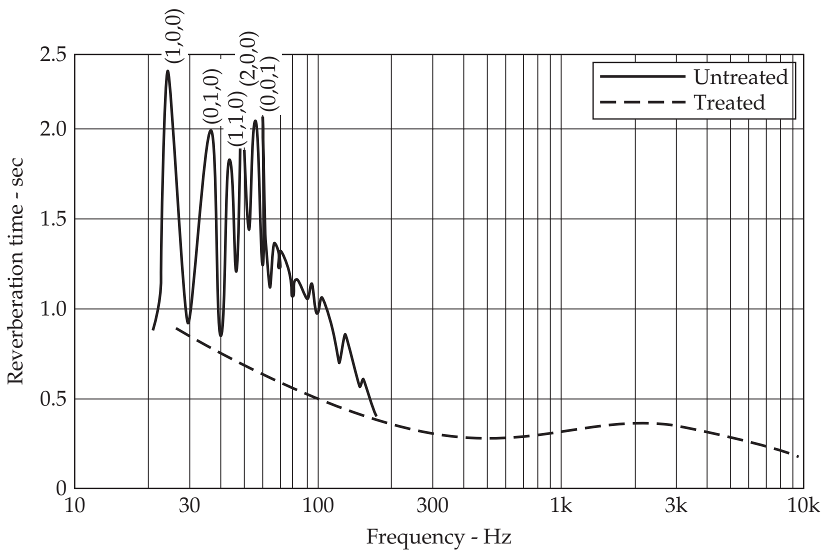


Figure 11-6 Reverberation time measured with pure sine signals at low frequencies reveals slow sound decay (long reverberation time) at the modal frequencies. These peaks apply only to specific modes and are not representative of the room as a whole. High modal density, resulting in uniformity of distribution of sound energy and randomizing of directions of propagation, is necessary for reverberation equations to apply. (*Beranek, Schultz.*)

is measured as 2.3 sec. Similar decays are observed at 35.27, 42.76, 48.37, and 56.43 Hz with faster decays (shorter reverberation times) at frequencies between those modes. The long decay times at the modal frequencies are decay rates characteristic of individual modes, not of the room as a whole.

Long reverberation time implies low absorbance, and short reverberation time implies high absorbance. It is interesting that the sound absorbing qualities of the walls, floor, and ceiling can vary considerably within a frequency range of a few hertz. For the (1, 0, 0) mode, only the absorbance of the two ends of the room comes into play; the four other surfaces are not involved. For the (0, 0, 1) mode, only the floor and ceiling are involved. In this low-frequency range, we have measured the decay rate of individual modes, but not the average condition of the room.

We see now why it is difficult to apply the concept of reverberation time to small rooms having dimensions comparable to the wavelength of sound. Schultz states that reverberation time is a statistical concept "in which much of the mathematically awkward details are averaged out." In small rooms these details are not averaged out.

The reverberation-time equations of Sabine, Eyring-Norris, and others are based on the assumption of an enclosed space with a highly uniform distribution of sound energy and random direction of propagation of sound. At low frequencies in the room presented in Fig. 11-6, energy is distributed very unevenly and direction of propagation is far from random. After the room is treated, reverberation time measurements would follow the broken line, but statistical randomness still does not prevail below 200 Hz even though modal frequencies are brought under some measure of control.

Analysis of Decay Traces

An octave band of pink noise viewed on an oscilloscope shows a trace that looks very much like a sine wave, except that it is constantly shifting in amplitude and phase, because of the random property of noise. This characteristic of random noise has its effect on the shape of the reverberatory decay trace. Consider what this constantly shifting random noise signal does to the normal modes of a room. In this room, when the axial, tangential, and oblique resonant modes are considered, they are quite close together in frequency. For example, the number of modes included within an octave band centered on 63 Hz are: 4 axial, 6 tangential, and 2 oblique modes between the –3-dB points. These are shown in Fig. 11-7 in which the taller lines represent the more dominant axial modes, the intermediate height the tangential modes, and the shorter lines the oblique modes.

When the excitation noise from the loudspeaker energizes the room, it excites one mode, and an instant later another mode. While the response shifts to the second mode, the first mode begins to decay. Before it decays very far, however, the random-noise signal may once more instantaneously return to the frequency of the first mode, giving it another boost. All the modes of the room are in constant agitation, alternating between high and somewhat lower levels, as they begin to decay between stimuli. It is strictly random, but it can be said with confidence that each time the excitation noise is stopped, the modal excitation pattern will be somewhat different. For example, the 12 modes in the 63-Hz octave will all be highly energized, but each to a somewhat different level the instant the noise is stopped. This helps address, but does not completely explain, the effect that room normal modes have on the measured reverberation time in small rooms.

Mode Decay Variations

To illustrate this discussion, let's consider measurements in a real room. The room is a rectangular studio for voice recording having the dimensions 20 ft, 6 in × 15 ft × 9 ft, 6 in, with a volume of 2,921 ft^3. The measuring equipment is that described in Fig. 11-5, and the technique is that described previously. Four successive 63-Hz octave decays from the graphic recorder are shown in Fig. 11-8A. These decays are not identical, and differences

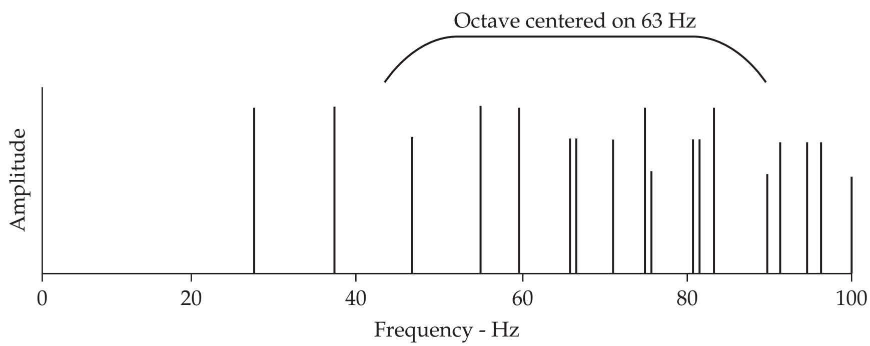


FIGURE 11-7 An example of normal modes included in an octave centered on 63 Hz (–3-dB points). The tallest lines indicate axial modes, the intermediate lines indicate tangential modes, and the shortest lines indicate oblique modes.

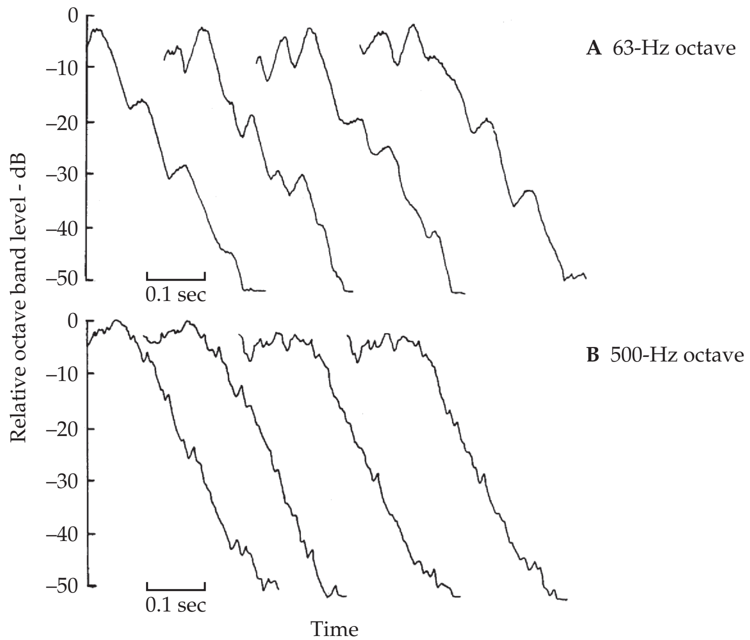


FIGURE 11-8 Decays of random noise recorded in a small studio having a volume of 2,921 ft^3. (A) Four successive 63-Hz octave decays recorded under identical conditions. (B) Four successive 500-Hz octave decays also recorded under identical conditions. The differences noted result from the differences in random-noise excitation the instant the source is turned off to start the decay.

can be attributed to the random nature of the noise signal. In particular, the fluctuations in the decays result from beats between closely spaced modes. Because the excitation level of the modes is constantly shifting, the form and degree of the beat pattern shift from one decay to another depending on where the random excitation happens to be the instant the noise is stopped.

Even though the four decays are similar, fitting a straight line to evaluate the reverberation time of each can be affected by the beat pattern. For this reason, it is good practice to record perhaps five decays for each octave for each microphone position in a room. With eight octaves (63 Hz to 8 kHz), five decays per octave, and three microphone positions, this means 120 separate decay readings for each room, which is laborious. However, this approach yields a statistically significant view of the variation with frequency. A handheld reverberation-time measuring device could accomplish this with less work, but it may not give detail of the shape of each decay. There is much information in each decay, and acoustical flaws can often be identified from aberrant decay shapes.

Four decays at 500 Hz are also shown in Fig. 11-8B for the same room and the same microphone position. The 500-Hz octave (354 to 707 Hz) embraces about 2,500 room modes. With such a high modal density, the 500-Hz octave decay is much smoother than the 63-Hz octave with only a dozen modes. Even so, the irregularities for the 500-Hz decay of Fig. 11-8B result from the same cause. Remembering that some modes die away faster than others, the decays in Fig. 11-8 for both octaves are composites of all modal decays included.

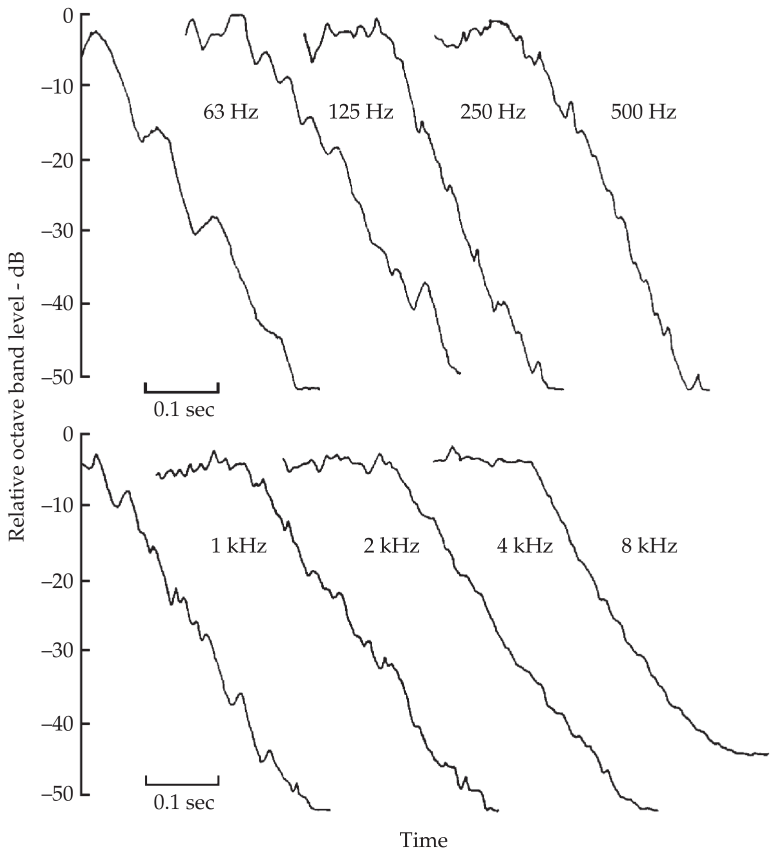


FIGURE 11-9 Decay of octave bands of noise in a voice studio. Fluctuations due to modal interference are greatest for low-frequency octaves containing fewer modes.

Frequency Effect

Figure 11-9 shows decays for octave bands of noise from 63 Hz to 8 kHz as measured in the 2,921-ft^3 voice studio. The greatest fluctuations are in the two lowest bands, the least in the two highest. This is what we would expect, given that the higher the octave band, the greater the number of normal modes included, and the greater the statistical smoothing. However, we should not necessarily expect the same decay rate because reverberation time is different for different frequencies. In this particular voice studio, a uniform reverberation time with frequency was the primary design goal, which was approximated in practice.

Reverberation Characteristic

Rooms absorb sound at different rates at different frequencies, producing different reverberation times, and this can greatly influence the room's sonic characteristic. For example, a room with longer reverberation times at higher frequencies, but shorter reverberation times at low frequencies may be described as sounding "thin" or "harsh." Because reverberation time will vary at different frequencies, it is common practice to measure reverberation times at different octaves. Rather than plot amplitude versus time as in a reverberation time graph, this analysis plots reverberation time versus frequency; this is sometimes called the reverberation characteristic.

This reverberation-time analysis is quite different from a real-time analysis that shows level versus frequency. For example, consider a room that exhibits excessive reverberation time at low frequencies, causing poor bass clarity. If the problem is mistakenly addressed through equalization, by rolling off low frequencies, a real-time analysis might show a flat low-frequency response. But reverberation time measurements would show that reverberation time is still too long at low frequencies and bass clarity is still poor; the problem has not been addressed. That is because equalization will not change the reverberation characteristic. To solve the problem of frequency balance, acoustical treatment must be applied to change the reverberation characteristic.

To measure the reverberation characteristic, a pink-noise source is band-pass filtered to produce octave-band noise at various center frequencies. For example, these center frequencies may be used: 63, 125, 250, 500, 1,000, 2,000, 4,000, and 8,000 Hz. The source is activated to produce a steady-state level, then turned off. The RT_{60} is the time required for the level to drop 60 dB. As noted, because of dynamic-range constraints, the first portion of the decay can be extrapolated to produce the full measurement. The entire measurement is conveniently performed by recording the steady-state noise and the decay. Because of fluctuations in level at low frequencies, measurements must be repeated. Because only the time decay of a narrow band is relevant, the frequency responses of the source loudspeaker and microphone are relatively unimportant. However, because of room modes, measurements at low frequencies are affected by the position of the source and the microphone.

As noted, the results can be plotted as reverberation time versus frequency. Figure 11-10 shows an example of a room's reverberation characteristic before and after room treatment. As shown, the room exhibited a significant rise in reverberation time in the upper bass and lower midrange. After treatment, reverberation time was much flatter, with a desirable moderate increase in reverberation time at low frequencies. Generally, in rooms used for music, it is preferred to have a smooth reverberation-time characteristic, with somewhat higher RT_{60} at low frequencies. Undue bumps in the reverberation-time characteristic can be reduced by adding absorption treatment in that frequency range. A practical example of room treatment to obtain desirable reverberation times at different frequencies is given at the end of this chapter.

Reverberation Time Variation with Position

In most rooms, there is enough variation of reverberation time from one physical position to another to justify taking measurements at several positions. The average then gives a better statistical picture of the behavior of the sound field in the room. If the room is symmetrical, it may be efficient to spot all measuring points on one side of the room to increase the effective coverage for a given effort.

Decay Rate and the Reverberant Field

The definition of reverberation time is based on uniform distribution of energy and random directions of propagation. These idealized conditions do not exist in small, absorptive rooms because of room mode effects, so what we measure, technically,

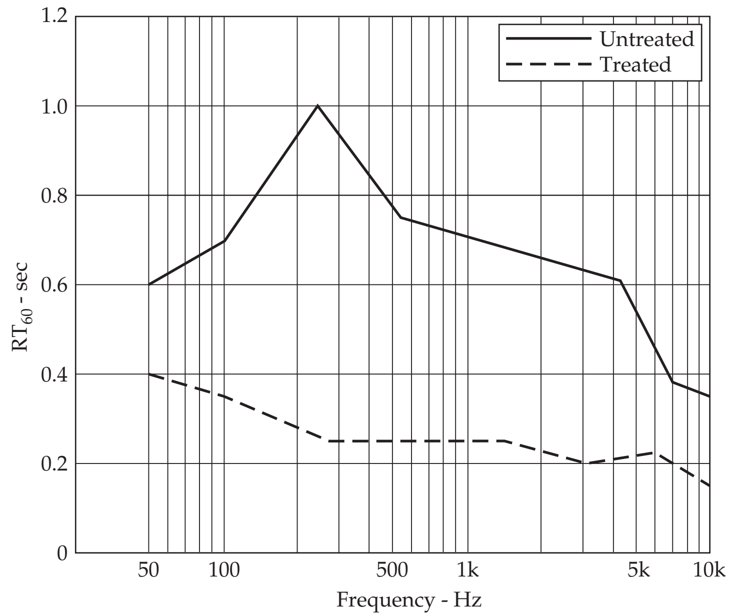


Figure 11-10 An example of a room's reverberation characteristic before and after room treatment. A significant rise in reverberation time in the upper bass and lower midrange is changed to a flatter characteristic with a moderate increase in reverberation time at low frequencies.

should not be called reverberation time. It is more properly termed a decay rate. For example, a reverberation time of 0.3 sec is equivalent to a decay rate of 60 dB/0.3 sec = 200 dB/sec. Speech and music in small, absorptive rooms certainly decay, even though the modal density is too low to meet the formal requirements of the definition of reverberation time.

In small and relatively dead rooms such as many studios, control rooms, and home listening rooms, the direct sound from the source usually dominates. A true reverberant field may be below the ambient noise level. However, the reverberation time equations have been derived for conditions that exist only in the reverberant field. In this sense, then, the concept of reverberation time is inapplicable to small, relatively dead rooms. We measure something very much like the reverberation time in large, more live spaces. But in small, dead rooms, more accurately, what we measure is the decay rate of the normal modes of the room.

Each axial mode decays at its own rate determined by the absorbance of a pair of walls and their spacing. Each tangential and oblique mode has its own decay rate determined by distance traveled, the number of surfaces involved, the variation of the absorption coefficient of the surfaces with the angle of incidence, and so on. Whatever average decay rate is measured for an octave of random noise will be representative of the average decay rate at which that octave of speech or music signals would die away. Although the applicability of computing reverberation time from the equations based on reverberant field conditions might be questioned because of the lack of reverberant field, the measured decay rates most certainly apply to this space and to these signals.

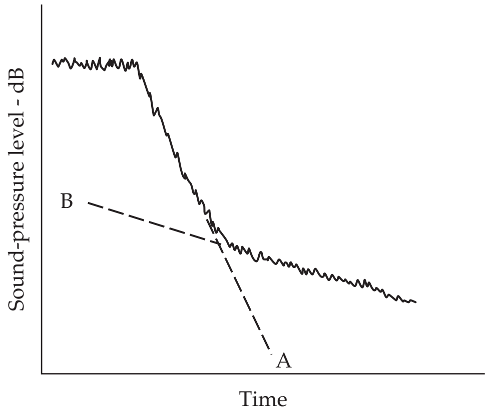


FIGURE 11-11 Reverberatory decay with a double slope due to acoustically coupled spaces. The shorter reverberation time represented by slope A is that of the main room. A second, highly reflective space is coupled through an open doorway; its longer reverberation time is represented by slope B. Those seated near the doorway are subjected first to the main-room response and then to the decay of the coupled space.

Acoustically Coupled Spaces

The shape of a reverberation decay can reveal acoustical problems in the space. One common effect that alters the shape of the decay is due to acoustically coupled spaces. This is quite common in large public gathering spaces but is also found in offices, homes, and other small spaces. The principle involved is illustrated in Fig. 11-11. In this example, the main space, perhaps an auditorium, is acoustically quite dead and has a reverberation time corresponding to slope A. An adjoining hall with hard surfaces opens into the main room and has a reverberation time corresponding to slope B. A person seated in the main hall near the connecting opening could very well experience a double-slope reverberation decay. After the sound level in the main room falls to a fairly low level, the main room reverberation would be dominated by sound fed into it from the slowly decaying sound in the adjoining hall. Assuming the reverberation time described by slope A is acoustically correct for the main room, persons subjected to slope B would hear degraded sound.

Electroacoustically Coupled Spaces

What is the overall reverberant effect when music recorded in a studio with one reverberation time is reproduced in a listening room with a different reverberation time? The sound in the listening room is indeed affected by the reverberation of both studio and listening room. This is demonstrated as follows:

- The combined reverberation time is greater than either alone.
- The combined reverberation time is nearer the longer reverberation time of the two rooms.
- The combined decay departs somewhat from a straight line.

- If one room has a very short reverberation time, the combined reverberation time will be very close to the longer one.
- If the reverberation time of each of the two rooms alone is the same, the combined reverberation time is theoretically 20.8% longer than one of them.
- The character and quality of the sound field transmitted by a stereo system conform more closely to the mathematical assumptions of the above than does a monaural system.
- The first five items can be applied to the case of a studio linked to an echo chamber as well as a studio linked to a listening room.

Eliminating Decay Fluctuations

The measurement of reverberation time by the classic method described above requires the recording of many decays for each condition. Schroeder published an alternative method by which the equivalent of the average of a great number of decays can be obtained in a single decay. One practical, but clumsy, method of accomplishing the required mathematical steps is to:

1. Record the decay of an impulse (noise burst or pistol shot) by the normal method.
2. Play back that decay reversed.
3. Square the voltage of the reversed decay as it grows.
4. Integrate the squared signal with a resistance-capacitance circuit.
5. Record this integrated signal as it grows during the reversed decay. Reverse it and this trace will be mathematically identical to averaging an infinite number of traditional decays.

Influence of Reverberation on Speech

Consider what happens to the spoken word "back" in a reverberant space. The word starts abruptly with a "ba . . ." sound and ends with the consonant ". . . ck" which is much lower in level. As measured on a graphic recorder, the "ck" sound is nominally about 25 dB below the peak level of the "ba" sound and reaches a peak nominally about 0.32 sec after the "ba" peak.

Both the "ba" and "ck" sounds are transients that grow and decay after the manner of Fig. 11-2. Sketching these to scale yields Fig. 11-12. The "ba" sound grows to a peak at an arbitrary level of 0 dB at time $t = 0$, after which it decays according to the RT_{60} reverberation time of the room, which in this example is assumed to be 0.5 sec. The "ck" consonant sound, peaking 0.32 sec later, is 25 dB below the "ba" sound peak. It too decays at the same rate as the "ba" sound according to the assumed 0.5-sec RT_{60}. Under the influence of the 0.5-sec reverberation time, the "ck" consonant sound is not masked by the reverberation of "ba." However, if the reverberation time is increased to 1.5 sec, as shown by the broken lines, the consonant "ck" is completely masked by reverberation. Likewise, a syllable that ends one word might mask the opening syllable of the next word.

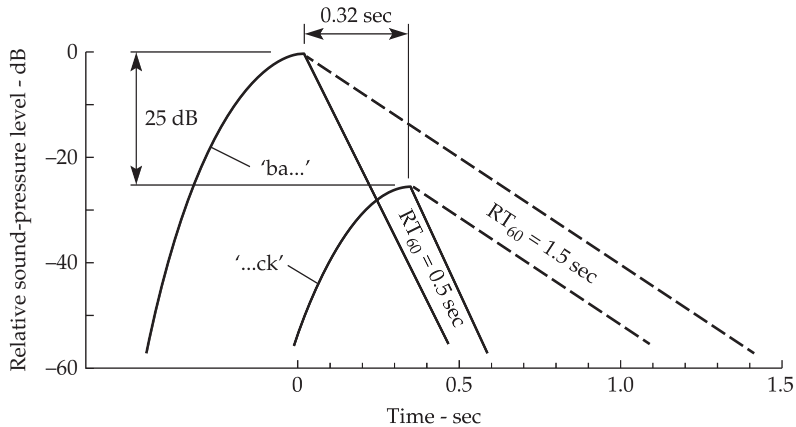


FIGURE 11-12 An illustration of the effects of reverberation on the intelligibility of speech. Understanding the word "back" depends on apprehending the later, lower-level consonant ". . . ck," which can be masked by reverberation if the reverberation time is too long.

Excessive reverberation can impair the intelligibility of speech principally by masking the lower-level consonants. In the word "back," the word is unintelligible without clear audibility of the "ck" part. Understanding the "ck" ending is the only way to distinguish "back" from bat, bad, bass, ban, or bath. In this simplified way, we observe the effect of reverberation on the intelligibility of speech and the reason why speech is more intelligible in rooms having lower reverberation times. Similarly, in a highly reverberant room, a speaker may have to speak slowly to be understood. It is a straightforward procedure to predict with reasonable accuracy the intelligibility of speech in a space, from geometrical factors and knowledge of reverberation time.

On the other hand, a completely dead room, or an outdoor environment, is not optimal for speech intelligibility; at even a moderate distance away, the sound level from a speaker may be too low to clearly hear speech. The reverberation in properly designed live rooms adds acoustical power to original sound sources, reinforcing them and making them more audible, as well as more pleasant sounding. A dead room adds less power to the source, and the overall sound is softer. In a room principally used for speech, such as an auditorium, a good design must therefore balance reverberation time and acoustical gain with intelligibility. In part, this means that different reverberation times determine whether a room is best suited for speech or music.

Influence of Reverberation on Music

The effect of reverberation on speech can be easily gauged according to the intelligibility of speech. The effect of hall resonance or reverberation on music is intuitively grasped but is more difficult to quantify. For example, a reverberation time that seems suitable for one type of music may be unsuitable for another type of music. In any case, the optimal reverberation time for music is both variable and subjective. This topic has received much attention from scientists as well as musicians. Beranek attempted to summarize knowledge and to pinpoint essential characteristics of concert halls and opera houses around the world, but our understanding of the problem is still incomplete. Suffice it to say that the reverberation decay of a music hall is an important factor among many, another being the echo pattern of the early sound of a hall. It is beyond the scope of this book to treat this subject in great detail, but some commonly overlooked points are discussed briefly.

Normal modes have great basic importance in any room's acoustical response, and they are also influential in concert halls and listening rooms. An interesting phenomenon is pitch change during reverberant decay. For example, in reverberant churches, organ tones have been observed to change pitch as much as a semitone during decay. In searching for an explanation for this phenomenon, two factors have been suggested: shift of energy between normal modes and the perceptual dependence of pitch on sound intensity. Balachandran demonstrated the physical (as opposed to psychophysical) reality of the effect using the fast Fourier transform (FFT) technique on a reverberant field created by 2-kHz pulses. In his case study, he showed the existence of a primary 1,992-Hz spectral peak, and curiously, another peak at 3,945 Hz. Because a 6-Hz change would be just perceptible at 2 kHz, and a 12-Hz change perceptible at 4 kHz, we see that the 39-Hz shift from the octave of 1,992 Hz would give an impression of pitch change. The reverberation time of the hall in which this effect was recorded was approximately 2 sec.

Optimum Reverberation Time

Considering the full range of possible reverberation times, seemingly there must be an optimum time between the too-dry condition of the outdoors and anechoic chambers, and the obvious problems associated with excessively long reverberation times as in a masonry cathedral. There is great disagreement as to what the optimal value is in any given room because it is a subjective problem, with diverse cultural and aesthetic expectations; thus differences in opinions must be expected. The optimal value depends not only on the person making the judgment but also on the type of sound sources being considered.

In general, long reverberation times yield a loss of intelligibility in speech and a lack of definition in music. Spaces for speech particularly require shorter reverberation times than for music because of the importance of the clarity provided by direct sound. In dead spaces in which reverberation time is very short, loudness and tonal balance in music may suffer. It is not possible to precisely specify optimum reverberation times for different venues, but Figs. 11-13 to 11-15 show approximate recommendations given by experts who do not always agree with each other.

Reverberation chambers, which are used for measuring absorption coefficients, are carefully designed for the longest practical RT_{60} to achieve the maximum accuracy. In this unique application, the optimum reverberation time is the maximum attainable.

The best reverberation time for a space in which music is played depends on the size of the space and the type of music. No single optimum universally fits all types of music; the best that can be done is to establish a range based on subjective judgment. For example, slow, solemn, melodic music, such as some organ music, is best served by a long reverberation time. Quick, rhythmic music, such as some chamber music, requires a shorter reverberation time.

The reverberation times chosen for places of worship such as temples, churches, and mosques generally present a compromise between music and speech. The reverberation times in Fig. 11-13 range from the highly reverberant conditions favored in

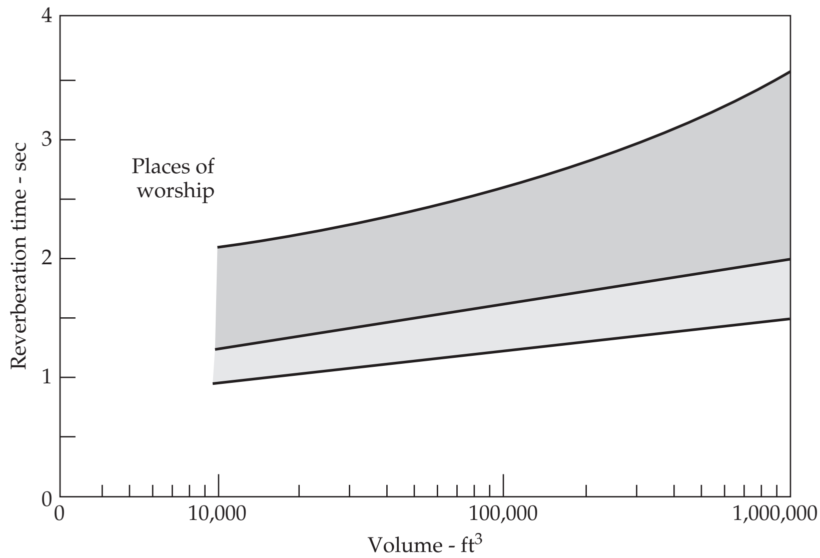


Figure 11-13 Range of optimum reverberation times for places of worship. Liturgical churches and cathedrals employ longer reverberation times shown in the upper area of the graph. Places of worship having services more oriented to speech are designed for shorter reverberation times shown in the lower area.

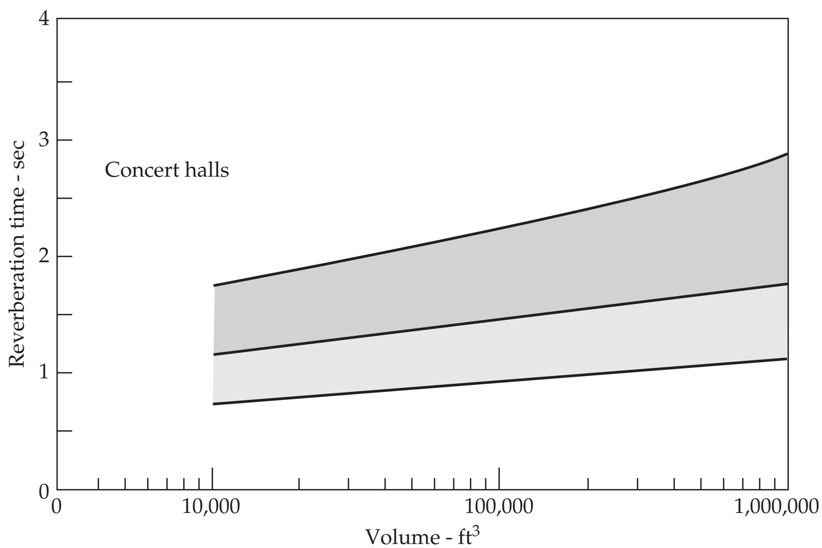


Figure 11-14 Range of optimum reverberation times for concert halls. Symphonic music calls for longer reverberation times shown in the upper area of the graph. Opera and chamber music benefit from relatively shorter reverberation times shown in the lower area.

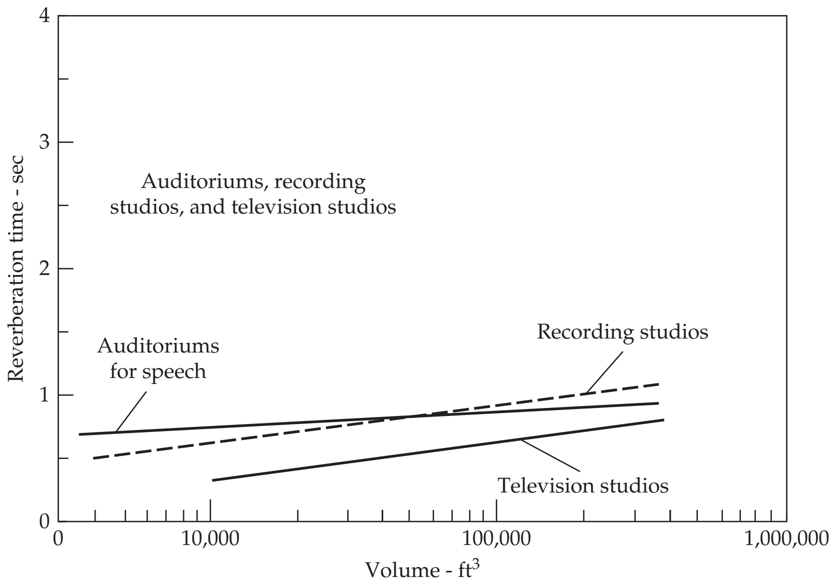


Figure 11-15 Range of optimum reverberation times for auditoriums, recording studios, and television studios. These rooms feature short reverberation times. Auditoriums prioritize speech intelligibility. Recording studios have short reverberation times with some diversity in localized reverberation. Television studios require speech clarity while minimizing other studio noise.

liturgical churches and cathedrals in the upper area, to the shorter reverberation times of more sermon-oriented places of worship in the lower area. It is important to note that when a sound reinforcement system is used, its performance criteria, and the room's natural acoustics, must be carefully integrated.

Figure 11-14 represents the range of recommended reverberation times for different-sized concert halls. Symphonic music calls for longer reverberation times in the upper area while lighter music requires shorter reverberation times. The lower area of the graph applies to opera and chamber music.

Generally, spaces used primarily for music recording (recording studios) or live speech (auditoriums and television studios) require similar reverberation times, as shown in Fig. 11-15. Recording studios present particular problems that do not conform to simple rules. Multitrack recording in which musical instruments are recorded on separate tracks for later mixdown generally requires quite dead spaces to realize adequate acoustical separation between tracks. Music producers often require different reverberation times for different instruments; hence both reflective areas and absorptive areas may be found in the same studio. The range of reverberation time realized in this manner is limited, but proximity to reflective surfaces does affect local conditions. However, it is generally preferable to record instruments in an absorptive studio space, and add reverberation artificially during mixdown. For this reason, most recording studios have acoustics that are much more absorptive than found in live music performance spaces.

Television broadcast studios as represented in Fig. 11-15 have somewhat shorter reverberation times to deaden the sounds associated with some video equipment, dragging cables, and other production noises. It should also be remembered that acoustics in television are dominated by the setting and local furnishings. In auditoriums, reverberation times are also generally short, and sound reinforcement systems are typically employed with an emphasis on providing speech intelligibility. Movie theaters are also usually quite absorptive; adequate reverberation is already recorded in the soundtrack, and in particular added room reverberation would reduce the intelligibility of movie dialogue. For hearing-impaired listeners, reverberation times should be decreased for better intelligibility. Many spaces that are used for multiple functions can benefit from adjustable acoustical methods.

Bass Rise of Reverberation Time

The goal in most recording studios, as well as many other rooms, is to achieve a reverberation time that is consistent throughout the audible spectrum. This can be difficult to realize, especially at low frequencies. Adjustment of reverberation time at high frequencies is easily accomplished by adding or removing relatively inexpensive absorbers. At low frequencies, the situation is quite different as absorbers that are effective at low frequencies are bulky, more costly, difficult to install, and their performance is less predictable.

Researchers at the British Broadcasting Corporation (BBC) observed that subjective judgments indicated a tolerance for a certain amount of bass rise of reverberation time. Investigating this in controlled tests, Spring and Randall found that bass rise to the extent indicated in Fig. 11-16 was tolerated by the test subjects for voice signals. Taking the 1-kHz value as reference, rises of 80% at 63 Hz and 20% at 125 Hz were found to be acceptable. These tests were made in a studio 22 × 16 × 11 ft (volume about 3,900 ft^3) for which the midband reverberation time was 0.4 sec, which agrees fairly well with Fig. 11-15.

Bass rise in reverberation time for music performance has traditionally been accepted to add sonority and warmth to the music in concert halls. A bass rise may help overcome the ear's relative insensitivity at low frequencies, or may simply be a cultural preference. Presumably, a somewhat greater bass rise than that for speech would be desirable in listening rooms designed for music. One metric used to define a bass boost is the bass ratio (BR), where $BR = (RT_{60/125} + RT_{60/250})/(RT_{60/500} + RT_{60/1,000})$. In other words, the sum of RT_{60} at 125 and 250 Hz is divided by the sum of RT_{60} at 500 and 1,000 Hz; a value greater than unity would show that reverberation time is longer at low frequencies.

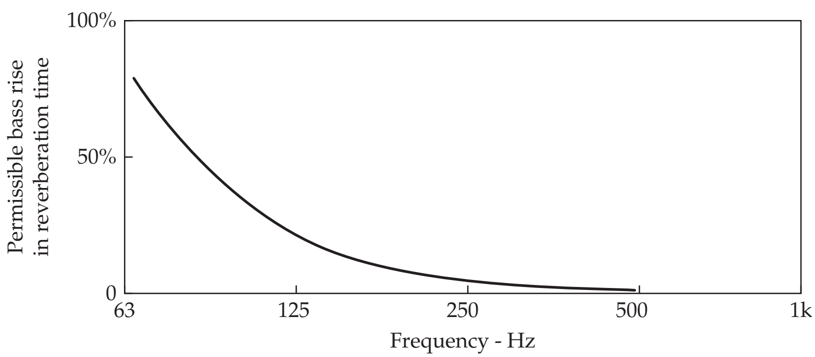


FIGURE 11-16 Permissible bass rise of reverberation time for voice studios derived by subjective evaluation in controlled tests by BBC researchers. (*Spring and Randall.*)

Some designers recommend a bass ratio of 1.1 to 1.45 for halls with a reverberation time less than 1.8 sec, and a bass ratio of 1.1 to 1.25 for halls with a longer reverberation time. A bass ratio of less than 1.0 is not recommended. This is explored more fully in Chap. 28.

Initial Time-Delay Gap

One important characteristic of natural reverberation in concert halls was revealed by Beranek's study of halls around the world. At any given seat, the direct sound arrives first because it follows the shortest path. Shortly after the direct sound, the reverberant sound begins to arrive. The initial time-delay gap (ITDG) is the time between the arrival of the direct sound, and the onset of reverberant sound, as shown in Fig. 11-17. If this gap is less than about 40 msec, the ear integrates the direct and the reverberant sound successfully. In addition to all of the reflections responsible for achieving reverberation density, the ITDG is another important delay that must be considered. In particular, this gap is important in concert-hall design (and in artificial reverberation algorithms) because it is the cue that gives the ear information on the size of the hall.

Listening Room Reverberation Time

The reverberation characteristic of a typical home listening room is of interest to the audiophile, the broadcaster, and the recording engineer. The monitoring room of the broadcast or recording studio should have a reverberation time similar to that of the listening room in which the final product will be heard. However, generally, such professional rooms should be deader than the home listening room, which will add its own reverberation to that of recording or broadcast studio.

Figure 11-18 shows the average reverberation time of 50 British living rooms measured by Jackson and Leventhall using octave bands of noise. The average reverberation time decreases from 0.69 sec at 125 Hz to 0.4 sec at 8 kHz. This is considerably higher than earlier measurements of 16 living rooms made by BBC engineers in which average reverberation times ranged from 0.35 to 0.45 sec. Apparently, the living rooms measured by the BBC engineers were better furnished than those measured by Jackson and Leventhall.

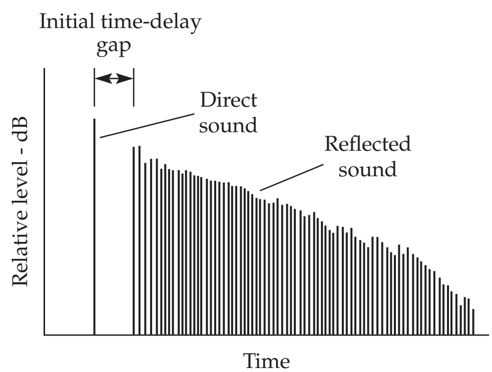


FIGURE 11-17 The introduction of an initial time-delay (ITDG) gap plays an important role in room reverberation. That time gap between the arrival of the direct sound, and the first reflected sounds, helps identify the size of the room.

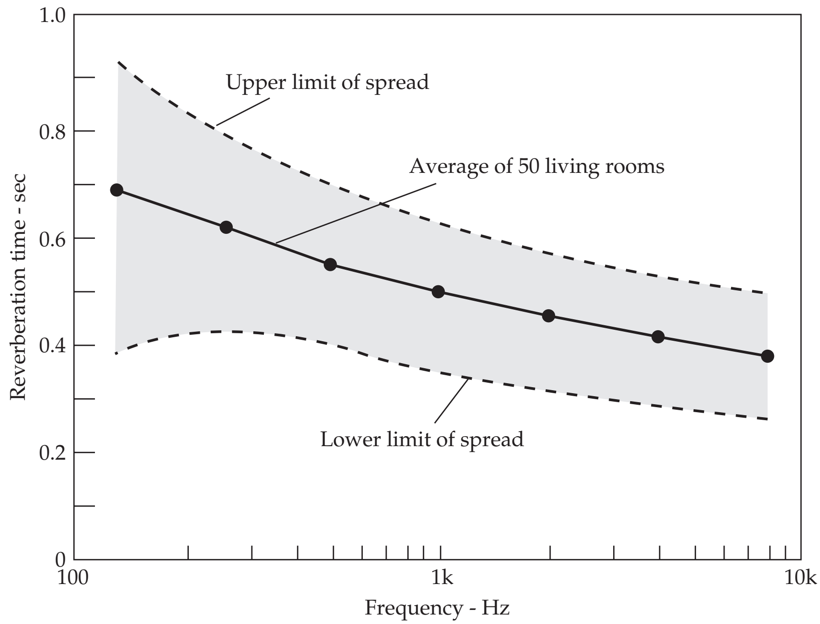


FIGURE 11-18 Average reverberation time for 50 British living rooms. (*Jackson and Leventhall.*)

The 50 rooms of the Jackson-Leventhall study were of varying sizes, shapes, and degrees of furnishing. The sizes varied from 880 ft^3 to 2,680 ft^3, averaging 1,550 ft^3. An optimum reverberation time for speech for rooms of this size is about 0.3 sec (see Fig. 11-15). Only those living rooms near the lower limit approach this, and in them, we would expect to find much heavy carpet and overstuffed furniture. These reverberation measurements tell us little or nothing about the possible presence of timbral defects. The BBC engineers checked for timbral changes and reported problems in a number of the living rooms studied.

Artificial Reverberation

Artificial reverberation is considered a necessity for many kinds of music recording projects. Music tracks recorded in dry (nonreverberant) studios lack the richness of the room effect contributed by most acoustical spaces. The addition of artificial reverberation to such recordings is standard practice, and there is great demand for processors that will provide natural-sounding artificial reverberation.

There are many ways to generate artificial reverberation, but the challenge is finding that method that mimics actual acoustical spaces and does not introduce frequency-response aberrations into the signal. Historically, a dedicated reverberation chamber was employed. The program was played over a loudspeaker in the chamber, picked up by microphones, and the reverberated signal mixed back into the original in the amount to achieve the desired effect. Small reverberation chambers were afflicted with serious timbral problems because of widely spaced modes. Large chambers were expensive.

Even though reverberation chambers had certain desirable qualities, the limitations outweighed the advantages and they are now a thing of the past.

Most artificial reverberation is created digitally using software programs that conceptually mimic the characteristics of a reverberant room. The principle of digital reverberation is illustrated in the signal flow schematic of Fig. 11-19. The input signal is delayed, and a portion of the delayed signal is fed back and mixed with the incoming signal, the mixture being delayed again, and so on. This replicates the action of sound rays reflecting from many room surfaces with an attenuation at each reflection.

Schroeder found that approximately 1,000 echoes per second are required to avoid a flutter effect and to sound natural to the ear. With a 40-msec delay, only $1/0.04 = 25$ echoes are produced each second, a far cry from the 1,000 per second desired. One solution is to arrange many simple reverberators in parallel. Four such simple reverberators, arranged in parallel, might produce $4 \times 25 = 100$ echoes per second. It would require 40 such reverberators in parallel to achieve the required echo density.

One approach to produce the necessary echo density, and also a flat frequency response, is illustrated in Fig. 11-20. Numerous delays feed back on themselves, combining to feed other delays, which in turn recirculate back to the first delay. The + signs in Fig. 11-20 indicate mixing (addition), and the × signs indicate gain (multiplication).

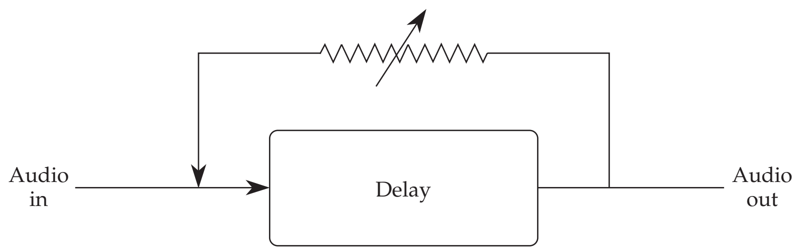


FIGURE 11-19 A simple digital reverberation algorithm using a delay line and feedback.

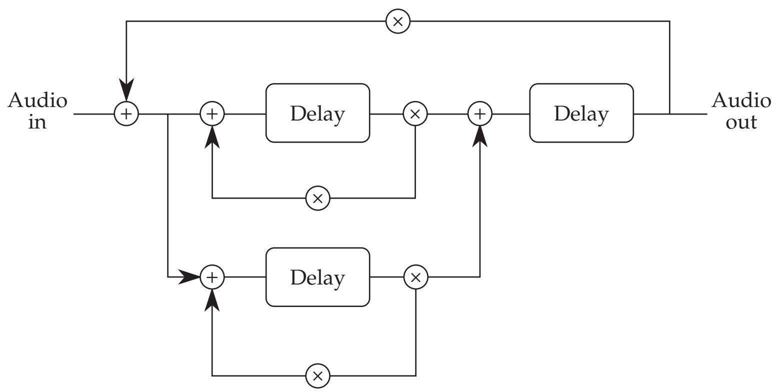


FIGURE 11-20 The required echo density can be achieved in a reverberation algorithm that incorporates numerous delays and recirculation of the signal. Practical digital reverberation algorithms are more complicated than this example.

Multiplying by a fraction less than unity gives a gain of less than unity—in other words, attenuation. The digital reverberator of Fig. 11-20 only suggests how greater echo density along with good frequency response might be achieved. Practical digital-reverberation algorithms are far more complicated than this, and many yield reverberation with a high echo density, flat frequency response, and natural sound.

Examples of Reverberation Time Calculations

As noted, reverberation time is defined as the number of seconds required for sound intensity in a room to drop 60 dB from its original level. The Sabine equation, presented earlier in this chapter as Eq. (11-1), can be used to calculate reverberation time. Two examples of reverberation time calculations are presented, for an untreated, and treated room.

Example 1: Untreated Room This example illustrates the implementation of Sabine equation. The dimensions of an untreated room are 23.3 × 16 × 10 ft. The room has a concrete floor and the walls and ceiling are of frame construction with 1/2-in gypsum board (drywall) covering. As a simplification, the door and a window will be neglected as having a minor effect. Figure 11-21 shows the untreated condition. The concrete floor area of 373 ft^2 and the gypsum board area of 1,159 ft^2 are

Size	23.3 × 16 × 10 ft
Treatment	None
Floor	Concrete
Walls/ceiling	Gypsum board, 1/2-in, on frame construction
Volume	(23.3)(16)(10) = 3,728 ft^3

Material	S ft^2	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$
Concrete	373	0.01	3.7	0.01	3.7	0.015	5.6	0.02	7.5	0.02	7.5	0.02	7.5
Gypsum board	1,159	0.29	336.1	0.10	115.9	0.05	58.0	0.04	46.4	0.07	81.1	0.09	104.3
Total absorption, sabins		339.8		119.6		63.6		53.9		88.6		111.8	
Reverberation time, sec		0.54		1.53		2.87		3.39		2.06		1.63	

S = area of material
α = absorption coefficient for that material and for that frequency
$A = S\alpha$ absorption units, sabins

$$RT_{60} = \frac{(0.049)(3{,}728)}{A} = \frac{182.7}{A}$$

$$\text{Example: For 125 Hz, } RT_{60} = \frac{182.7}{339.8} = 0.54 \text{ sec}$$

Figure 11-21 Room conditions and calculations for reverberation example 1 (Untreated Room).

entered in the table. The appropriate absorption coefficients α are entered from the table in App. C for each material and for the six frequencies. Multiplying the concrete floor area of $S = 373$ ft^2 by the coefficient $\alpha = 0.01$ gives a floor absorption of 3.7 sabins. This is entered under $S\alpha$ for 125 and 250 Hz. The absorption units (sabins) are then figured for both materials and for each frequency. The total number of sabins at each frequency is obtained by adding that of the concrete floor to that of the gypsum board. The reverberation time for each frequency is obtained by dividing $0.049V = 182.7$ by the total absorption for each frequency.

To visualize the variation of reverberation time with frequency, the values are plotted as the untreated curve of Fig. 11-22. The peak reverberation time of 3.39 sec at 1 kHz is excessive and would yield poor sound conditions. Two persons separated by 10 ft would have difficulty understanding each other as the reverberation of one word masks the next word.

Example 2: Treated Room The goal now is to correct the untreated reverberation condition. It is evident that much absorption is needed at midband frequencies, a modest amount at higher frequencies, and very little at lower frequencies. The need is for a material having an absorption characteristic shaped more or less like the untreated reverberation curve. An acoustical tile with a thickness of 3/4 in provides a suitable absorption distribution with respect to frequency. Giving no thought at this point to how it is to be placed in the room, what area of this tile is correct?

Figure 11-23 organizes the calculations. Everything is identical to Fig. 11-21, except that the 3/4-in acoustical tile has been added with coefficients from App. C. In Fig. 11-21, there are a total of 53.9 sabins at the peak reverberation time at 1 kHz and 339.8 sabins at 125 Hz at which the reverberation time is a reasonable 0.54 sec. How much 3/4-in acoustical tiling would be required to add 286 sabins at 1 kHz? The absorption coefficient of this material is 0.84 at 1 kHz. To get 286 sabins at 1 kHz with this material would require $286/0.84 = 340$ ft^2 of the material. This is entered in Fig. 11-23 and the calculations extended. Plotting these reverberation time points gives the treated curve of Fig. 11-22, showing reverberation time that is uniform across the band. The overall precision of coefficients and measurements is limited, so the deviation of the treated curve from flatness is insignificant.

An average reverberation time of 0.54 sec in a room of 3,728 ft^3 volume would be quite acceptable for a music listening room. If it were decided that the 0.85-sec point at 250 Hz should be brought down closer to 0.54 sec, it would require about 100 ft^2 of an absorber tuned to 250 Hz.

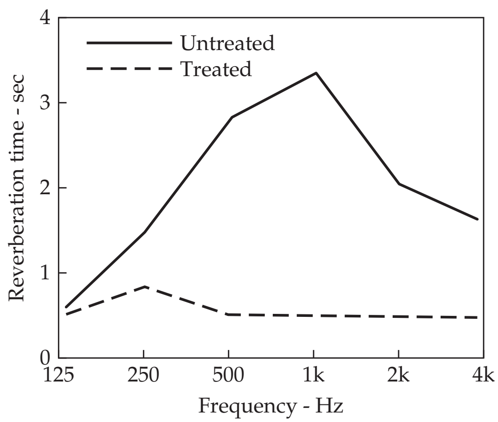


FIGURE 11-22 The calculated reverberation characteristics of a room measuring 23.3 × 16 × 10 ft showing the untreated condition of example 1, and the treated condition of example 2.

Size	23.3 × 16 × 10 ft
Treatment	Acoustical tile
Floor	Concrete
Walls/ceiling	Gypsum board, 1/2-in, on frame construction
Volume	(23.3)(16)(10) = 3,728 ft^3

Material	S ft^2	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$
Concrete	373	0.01	3.7	0.01	3.7	0.015	5.6	0.02	7.5	0.02	7.5	0.02	7.5
Gypsum board	1,159	0.29	336.1	0.10	115.9	0.05	58.0	0.04	46.4	0.07	81.1	0.09	104.3
Acoustical tile	340	0.09	30.6	0.28	95.2	0.78	265.2	0.84	285.6	0.73	248.2	0.64	217.6
Total absorption, sabins		370.4		214.8		328.8		339.5		336.8		329.4	
Reverberation time, sec		0.49		0.85		0.56		0.54		0.54		0.55	

S = area of material
α = absorption coefficient for that material and for that frequency
A = $S\alpha$ absorption units, sabins

$$RT_{60} = \frac{(0.049)(3{,}728)}{A} = \frac{182.7}{A}$$

FIGURE 11-23 Room conditions and calculations for reverberation example 2 (Treated Room).

How should the 340 ft^2 of 3/4-in acoustical tile be distributed? For maximum diffusion it should be applied in irregular patches and distributed to the three axes of the room. As a first approach, the 340 ft^2 could be distributed in proportion to areas on the three axes of the room as follows:

Area of north and south walls = (2)(10)(16) = 320 ft^2 (21%)

Area of east and west walls = (2)(10)(23.3) = 466 ft^2 (30%)

Area of floor and ceiling = (2)(16)(23.3) = 746 ft^2 (49%)

This would put (0.49)(340) = 167 ft^2 on the ceiling, (0.3)(340) = 102 ft^2 distributed between the east and west walls, and (0.21)(340) = 71 ft^2 distributed between the north and south walls. This recommendation should provide the desired results, but some experimentation, and critical listening, should be included in any such project to account for other variables. For example, different placements of absorptive materials in a room will affect the amount of absorption they provide.

Key Points

- After an excitation signal in a room is removed, reverberation is the decaying aftereffect of the sound and it has an important bearing on the acoustical quality of the room.
- The time required to reach a steady-state value in a room is determined by the rate of growth of sound; this growth rate is determined by the source's level of energy and the room's acoustics.

- The characteristics of sound decay (reverberation) in rooms are more audible than sound growth and thus more important in acoustical design.
- Reverberation time (RT_{60}.) is a measure of the rate of decay of sound; it is defined as the time in seconds required for sound intensity in a room to decrease 60 dB from its original level.
- The Sabine equation shows that the greater the room absorption, the shorter the reverberation time. Also, the greater the room volume, the longer the reverberation time.
- The mean free path ($4V/S$) is the average distance a sound travels between two successive reflections.
- The total absorption of a room is the summation of the respective absorptions of the various materials in the room, calculated from the surface area and absorptivity of each material.
- The absorption coefficient α characterizes the absorptivity of a material; it ranges from 0 to 1.0 where 1.0 represents complete absorption.
- The Sabine equation is most accurate in live rooms where the average absorption coefficient is less than 0.25.
- In large rooms, the long path lengths through air can effectively add absorption to the room thus lowering reverberation times; air absorption is only significant at frequencies above 2 kHz.
- Sound-level meter/analyzers can be used to measure and analyze room reverberation as well as many other acoustical characteristics.
- Room modes greatly influence the reverberation time in small rooms having dimensions comparable to the wavelength of sound at low mode frequencies.
- Rooms absorb sound at different rates at different frequencies, producing different reverberation times; this influences the room's acoustical quality. For example, a room with excessive reverberation time at low frequencies may exhibit poor bass clarity.
- Because of room modes, reverberation-time measurements at low frequencies are affected by the position of the source and the microphone.
- Excessive reverberation can impair the intelligibility of speech principally by masking the lower-level consonants.
- When a live room is properly designed, the reverberation adds acoustical power to original sound sources, reinforcing them and making them more audible.
- In a room principally used for speech, such as an auditorium, a good design must balance reverberation time and acoustical gain with intelligibility.
- In a concert hall or opera house, the effect of hall resonance or reverberation on music is intuitively grasped but is more difficult to quantify.
- The optimal reverberation time for a space subjectively depends on the size of the room, and its function. Spaces used for speech require shorter reverberation times than for music because of the importance of the clarity provided by direct sound.

- Bass rise in reverberation time for music performance has traditionally been accepted to add sonority and warmth to the music in concert halls.
- The initial time-delay gap (ITDG) is the time between the arrival of the direct sound, and the onset of reverberant sound.
- Most artificial reverberation is created digitally using software programs that conceptually mimic the characteristics of a reverberant room.

CHAPTER 12

Absorption

The law of conservation of energy states that energy can neither be created nor destroyed. However, energy can be changed from one form to another. If there is excessive sound energy in a room, the energy itself cannot be eliminated, but it can be transformed into an innocuous form. This is the function of sound-absorbing materials. Generally, sound absorbers can be considered as belonging to one of these three types: porous absorbers, panel absorbers, and volume absorbers. Generally, porous absorbers are most effective at higher frequencies, whereas panel and volume absorbers are most effective at lower frequencies.

All of these types of absorbers operate in fundamentally the same way. Sound exists as the vibratory energy of air particles, and by using absorbers the vibratory energy can be dissipated in the form of heat. Thus, sound energy is reduced. The amount of heat generated from sound absorption is minuscule. It would take the sound energy of millions of people talking to brew a cup of tea, so we must abandon any ecological hopes of warming our homes with sound—even the sound of heated arguments.

Dissipation of Sound Energy

A sound wave *S* traveling in air strikes a concrete block wall covered with an acoustical material, as shown in Fig. 12-1. What happens to the energy contained in the sound wave? As a sound wave travels through air, there is first a small heat loss *E* from air absorption that is appreciable only at higher audio frequencies. When the sound wave hits a wall, there is a reflected component *A* returned to the air from the surface of the acoustical material.

More interestingly, some of the sound penetrates the acoustical material represented by the shaded layer in Fig. 12-1. In this illustration, the direction of travel of the sound is refracted downward because the acoustical material is denser than air. There is heat lost *F* by the frictional resistance that the acoustical material offers to the vibration of air particles. As the sound ray strikes the surface of the concrete blocks, two things happen: a component *B* is reflected, and the ray is also bent strongly downward as it enters the much denser concrete blocks. There is further heat loss *G* within the concrete blocks. As the ray travels on, getting weaker all the time, it strikes the concrete-air boundary and undergoes another reflection *C*, and emerges with refraction *D* with additional heat lost (*I*, *J*, and *K*) in all three media.

The sound ray *S* experiences many complex events during its travel through this barrier, and every reflection and passage through air or acoustical material dissipates some of its original energy. The refractions bend the ray but do not necessarily dissipate heat. Fortunately, this minutia is not involved in practical absorption problems. We usually consider only the aggregate of these individual actions.

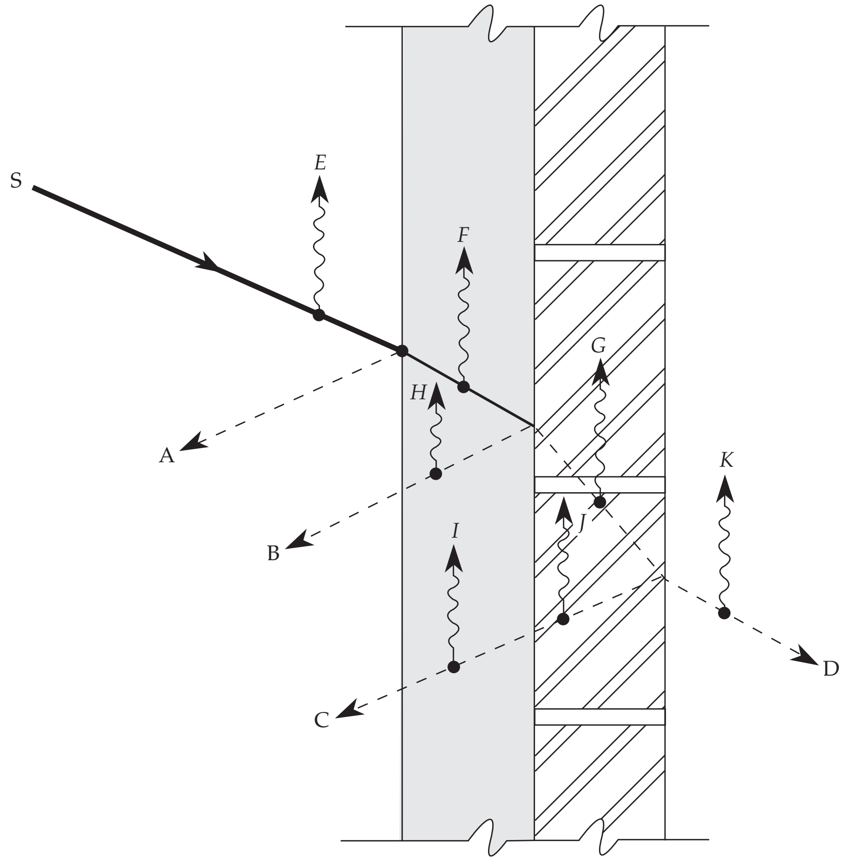


FIGURE 12-1 A sound ray impinging on an acoustical material on a masonry wall undergoes reflection from three different surfaces, and absorption in the air and two different materials, with different degrees of refraction at each interface. In this chapter, the cumulative absorbed component is of chief interest.

Absorption Coefficients

Absorption coefficients are used to rate a material's effectiveness in absorbing sound. Absorption coefficients vary with the angle at which sound impinges upon the material. In an established diffuse sound field in a room, sound is traveling in every imaginable direction. In many calculations, we need sound absorption coefficients that are averaged over all possible angles of incidence. The random incidence absorption coefficient is a coefficient that is averaged over all incidence angles. This is usually referred to as the absorption coefficient of a material, designated as α. The absorption coefficient is a measure of the efficiency of a surface or material in absorbing sound. For example, if 55% of the incident sound energy is absorbed at some frequency, the absorption coefficient α is said to be 0.55 at that frequency. A perfect sound absorber will absorb 100% of incident sound; thus α is 1.0. A perfectly reflecting surface would have α of 0.0.

Different references may use different symbols for the absorption coefficient; for example, a is sometimes used instead of α. Partly, this is because there are several different types of absorption coefficients. As noted, absorption varies according to the incident angle of sound striking the surface (absorption also varies according to frequency). One type of absorption coefficient measures absorption at a specific angle of incidence. Another type of absorption coefficient measures absorption from a diffuse sound field, that is, sound from a random distribution of angles. In this book, α refers to the absorption coefficient from a diffuse sound field (averaging all angles of incidence) at a given frequency. When an absorption coefficient at a particular angle is cited, it will be referred to as α_θ where θ is the angle of incidence.

The sound absorption A provided by a particular area of material is obtained by multiplying its absorption coefficient by the surface area of the material exposed to sound. Therefore:

$$A = S\alpha \tag{12-1}$$

where A = absorption units, sabins or metric sabins
S = surface area, ft^2 or m^2
α = absorption coefficient

Sound absorption A is measured in sabins, in honor of Wallace Sabine. An open window is considered a perfect absorber because sound passing through it never returns to the room. An open window would have an absorption coefficient of 1.0 and by definition; a 1-ft^2 open window provides 1 sabin of sound absorption. Ten square feet of open window would thus give 10 sabins of absorption. As another example, suppose that carpet has an absorption coefficient of 0.55; 20 ft^2 of carpet would therefore provide 11 sabins of absorption.

Either sabins or metric sabins can be used. A metric sabin is the absorption from a 1-m^2 open window. Since 1 m^2 equals 10.76 ft^2, 1 metric sabin equals 10.76 ft sabins. Or, 1 sabin = 0.093 metric sabin.

When calculating a room's total absorption, all the materials in the room, according to their area, will contribute to the total absorption:

$$\Sigma A = S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \ldots . \tag{12-2}$$

where $S_1, S_2, S_3, \ldots$ = surface areas, ft^2 or m^2
$\alpha_1, \alpha_2, \alpha_3, \ldots$ = respective absorption coefficients

Moreover, the average absorption coefficient can be calculated by dividing the total absorption by the total surface area:

$$\alpha_{\text{average}} = \frac{\sum A}{\sum S} \tag{12-3}$$

When absorptive material is placed over a surface, one must take into account the loss of absorption provided by the original surface. The net increase in absorption over an area is the absorption coefficient of the new material minus the coefficient of the original material.

The absorption coefficient of a material varies with frequency. Coefficients are typically published at the six standard frequencies of 125, 250, 500, 1,000, 2,000, and 4,000 Hz. In some cases, a material's absorption is given as a single-number rating known as the noise reduction coefficient (NRC). The NRC is the average of the coefficients for 250, 500, 1,000, and 2,000 Hz (125 and 4,000 Hz are not used). It is important to remember that the NRC is an average value, and also only accounts for absorption at middle frequencies. NRC is therefore most useful for speech applications. When considering wider-band music, individual coefficients, at a wider range of frequencies, should be used.

In some cases, the sound absorption average (SAA) is used to specify absorption. Like the NRC, the SAA is an arithmetic average, but the SAA uses absorption coefficients from twelve 1/3-octave bands from 200 Hz to 2.5 kHz. These coefficients are averaged to obtain the SAA value. Finally, the ISO 11654 standard specifies a single-number weighted sound absorption coefficient for materials using the ISO 354 testing standard.

Reverberation Chamber Method

The reverberation chamber method can be used to determine the absorption coefficients of absorbing materials; it measures the average value. This chamber is a large room (perhaps 9,000 ft^3) with highly reflective walls, floor, and ceiling. The reverberation time of the room is very long, and the longer it is, the more accurate the measurement. A standard sample of the material to be tested, usually measuring 8×9 ft, is mounted on the floor and the reverberation time is measured. Comparing this time with the known reverberation time of the empty room yields the number of absorption units the sample adds to the room. From this, the absorption attributed to 1 ft^2 of material is determined, giving the equivalent of the absorption coefficient in sabins (1 m^2 of material gives absorption units in metric sabins).

The construction of the chamber is important to ensure a large number of modal frequencies and to equalize mode spacing as much as possible. The position of the sound source and the number and position of the measuring microphones must be considered. Large rotating vanes are used to ensure adequate diffusion of sound. Absorption coefficients supplied by manufacturers for use in architectural acoustical calculations may be measured by the reverberation chamber method.

A square-foot open window is the perfect absorber with coefficient equal to 1.0, but some chamber measurements can show absorption coefficients greater than 1. This is because the diffraction of sound from the edges of the standard sample makes the sample appear, acoustically, of greater area than it really is. There is no standard method of making adjustments for this artifact. Some manufacturers publish the actual measured values if greater than unity; others arbitrarily adjust the values down to unity or to 0.99.

Impedance Tube Method

The impedance tube (also called a standing-wave tube or Kundt tube) has been applied to the measurement of the absorption coefficient of materials. For this application, it can quickly and accurately determine coefficient values. It also has the advantage of small size, modest demands in terms of supporting equipment, and it requires only a small sample. This method is primarily used for porous absorbers because it is not suited to those absorbers that depend on area for their effect such as vibrating panels and large slat absorbers.

The construction and operation of the impedance tube are illustrated in Fig. 12-2. The tube usually has a circular cross section with rigid walls. The sample to be tested is cut to fit snugly into the tube. If the sample is intended to be used while mounted on a solid surface, it is placed in contact with the heavy backing plate. If the material is to be used with a space behind it, it is mounted an appropriate distance from the backing plate.

At the other end of the tube is a small loudspeaker with a hole drilled through its magnet to accommodate a long, slender probe tube coupled to a microphone. Energizing the loudspeaker at a given frequency sets up a standing wave due to the interaction of the outgoing wave with the wave reflected from the sample. The form of this standing wave gives important information on the absorption of the material under test.

The sound pressure is maximal at the surface of the sample. As the microphone probe tube is moved away from the sample, the sound pressure falls to the first minimum. Successive, alternating maxima and minima will be detected as the probe tube is further withdrawn. If n is the ratio of the maximum sound pressure to its adjacent minimum, the normal (perpendicular) absorption coefficient α_n is equal to:

$$\alpha_n = \frac{4}{n + (1/n) + 2} \tag{12-4}$$

This is plotted in Fig. 12-3.

The advantages of the standing-wave tube method are offset by the disadvantage that the absorption coefficient so determined is accurate only for normal incidence. In practice, sound impinges on the surface of a material at all angles. Figure 12-4 is a graph for approximately obtaining the random-incidence absorption coefficient from the normal-incidence absorption coefficient as measured by the standing-wave method. The random-incidence coefficients are always higher than normal-incidence coefficients.

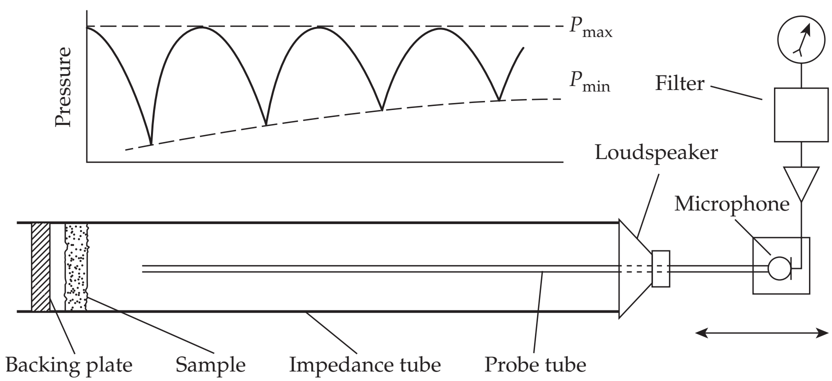


Figure 12-2 The impedance tube method can be used to measure the absorption coefficient of absorbing materials at normal incidence.

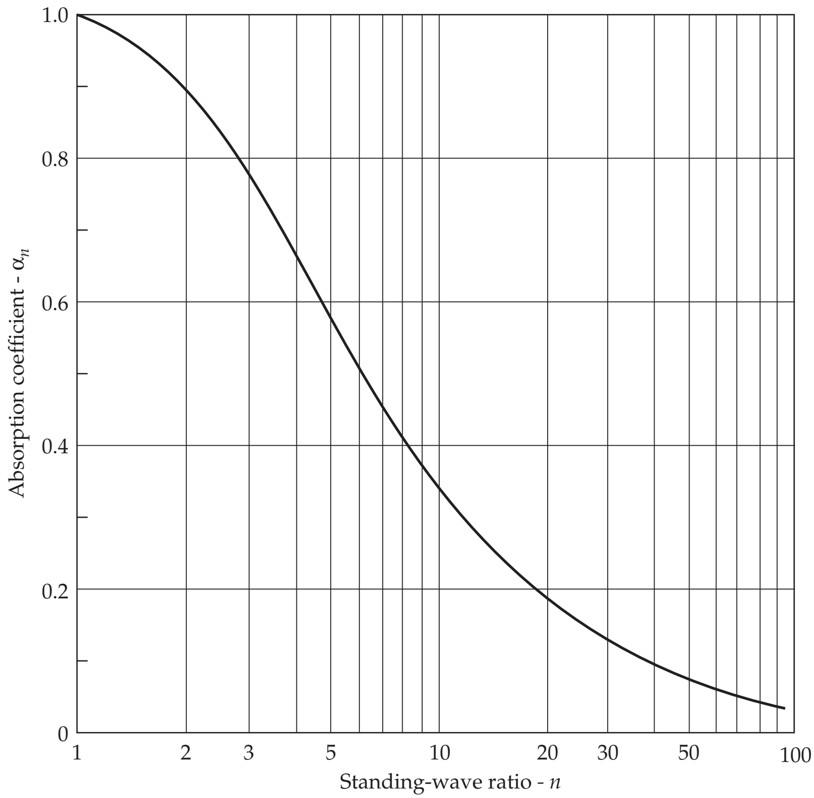


Figure 12-3 Graph for interpreting the standing-wave ratio in terms of the absorption coefficient. The standing-wave ratio can be found by dividing any pressure maximum by its adjacent pressure minimum (see Fig. 12-2).

Individual reflections affect the timbre of sound. In particular, specific normal reflections, called early sound, are of special interest. Although random-incidence coefficients are of interest in room reverberation calculations, for image control problems, normal-incident reflection coefficients are generally required.

Tone-Burst Method

Short pulses of sound can be used to perform anechoic acoustical measurements in ordinary rooms. It takes time for reflections from walls and other surfaces to arrive at a measuring position. With a short-duration pulse, the time gate can be opened only for the desired sound pulse, shutting out the subsequent interfering pulses. This tone-burst method can be used to measure the sound-absorption coefficient of a material at any desired angle of incidence.

Such an arrangement is illustrated in principle in Fig. 12-5. The source-microphone system is calibrated at distance x, as shown in Fig. 12-5A. The geometry of Fig. 12-5B is then arranged so that the total path of the pulse reflected from the material to be tested is equal to this same distance x. The strength of the reflected pulse is then compared to that of the unreflected pulse at distance x to determine the absorption coefficient of the sample.

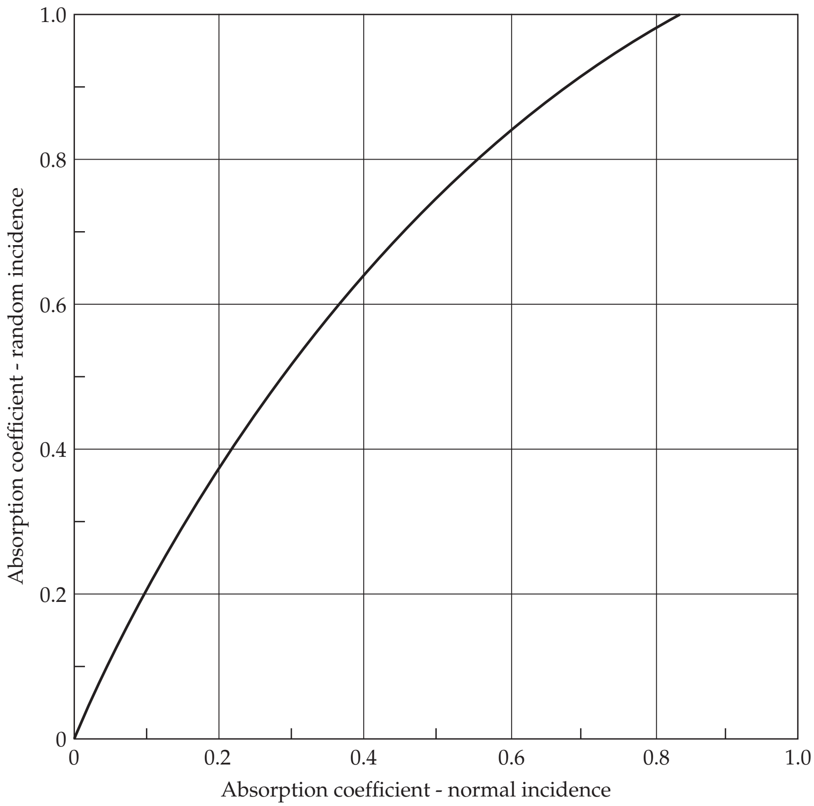


FIGURE 12-4 Graph showing the approximate relationship between absorption coefficients at normal incidence and those with random incidence.

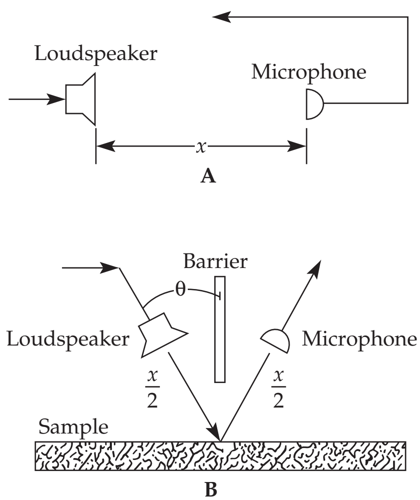


FIGURE 12-5 The absorption coefficients of materials can be determined by a tone-burst method. (A) The source-microphone system is calibrated at distance *x*. (B) The total path length of the pulse reflected from the material under test is equal to distance *x*.

ASTM Mounting Designation*		ABPMA Mounting Designation†
A	Material directly on hard surface	#4
B	Material cemented to plasterboard	#1
C-20	Material with perforated, expanded, or other open facing furred out 20 mm (3/4 in)	#5
C-40	Same type as C-20, furred out 40 mm (1-1/2 in)	#8
D-20	Material furred out 20 mm (3/4 in)	#2
E-400	Material spaced 400 mm (16 in) from hard surface	#7

*American Society for Testing and Materials (ASTM) designation: E795-16.
†Mountings formerly listed by Acoustical and Board Products Manufacturers Association (ABPMA).

Table 12-1 Mountings Commonly Used in Sound-Absorption Measurements

Mounting of Absorbents

The method of mounting the test sample on the reverberation chamber floor is intended to mimic the way the material is used in practice. Table 12-1 lists the standard mountings, both in the ASTM form and in the older ABPMA form.

The mounting method has a major effect on the absorption characteristics of a material, both in a measuring chamber and in practical rooms. For example, the absorption of porous materials is much greater with an airspace between the material and the mounting surface such as a wall. The quarter-wavelength ($\lambda/4$) rule dictates that a porous absorber for normal incidence must be at least a quarter-wavelength thick at the frequency of interest. For example, for a frequency of 1 kHz, the minimum absorber thickness should be about 3.4 in. Tables of absorption coefficients should always identify the standard mounting or include a description of the way the material was mounted during the measurements. Otherwise, the coefficients are of little value. Mounting type A with no airspace between the sound absorbing material and the wall surface is widely used. Another method commonly used is mounting type E-400, which is an approximation to the varying spaces encountered in suspended ceilings, as shown in Fig. 12-6.

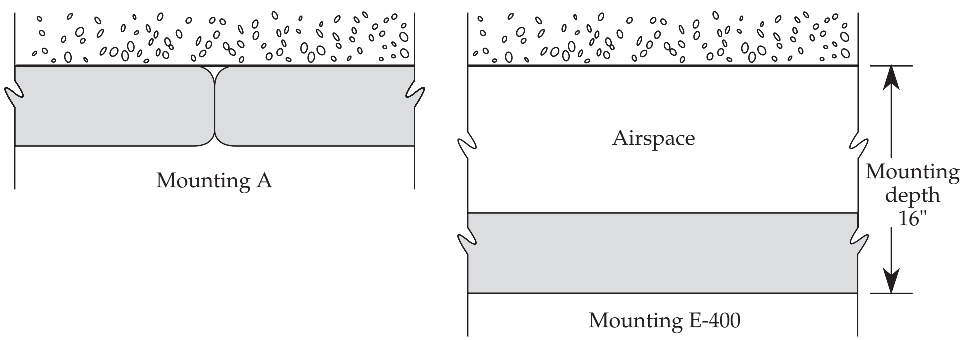


Figure 12-6 Two commonly used standard mountings associated with listings of absorption coefficients. With mounting A the material is flat against the backing. Mounting E-400 applies to suspended ceilings with lay-in panels. (See Table 12-1.)

Mid/High-Frequency Absorption by Porosity

Materials that have a porous composition, that is, with interstices between their matters, can operate as porous sound absorbers. The key word in this discussion of porous absorbers is interstices, the small cracks or spaces in porous materials. If a sound wave strikes a wad of cotton batting, the sound energy causes the cotton fibers to vibrate. The fiber amplitude will never be as great as the air particle amplitude of the sound wave because of frictional resistance. Restricted as this motion is, some sound energy is changed to frictional heat as fibers are set in motion. The sound penetrates into the interstices of the cotton, losing energy as fibers are vibrated. Cotton and many open-cell foams (such as polyurethane and polyester) are excellent sound absorbers because of their open-cell porosity that allows sound waves to penetrate the material. On the other hand, closed-cell materials (such as polystyrene), such as some used for thermal insulation, do not allow sound to penetrate the material and are relatively poor absorbers. The better the airflow through a porous material, the better its ability to absorb sound.

Porous absorptive materials most commonly used as sound absorbers are usually fuzzy, fibrous materials in forms such as boards, foams, fabrics, carpets, and cushions. If the fibers are too loosely packed, there will be little energy lost as heat. On the other hand, if fibers are packed too densely, penetration suffers and the air motion cannot generate enough friction to be effective. Between these extremes are many materials that are very good absorbers of sound. These are commonly composed of cellulose or mineral fiber. Their effectiveness depends on the thickness and density of the material, and depth of the airspace.

The absorption efficiency of materials that depend on the trapping and dissipating of sound energy in tiny pores can be seriously impaired if the surface pores are filled so that penetration is limited. Coarse concrete block, for example, has many such pores and is a fair absorber of sound. Painting may fill the surface pores and greatly reduce sound penetration, and thus absorption. However, if spray-painted with nonbridging paint, the absorption may be reduced only modestly. Acoustical tile painted at the factory minimizes the problem of reduced absorption. Under certain conditions, a painted surface can reduce porosity but act as a diaphragm that might actually become a fair absorber on a different principle, that of a damped vibrating diaphragm.

In some rooms, the acoustical treatment may overly prescribe carpeted floors and drapes, which emphasizes a shortcoming of most porous absorbers—that of poor low-frequency absorption. Tiles of cellulose fiber with perforated faces are also deficient in low-frequency absorption. Porous absorbers are thus proficient at reducing high-frequency sound energy but do not address a major problem of room acoustics—low-frequency standing waves.

To show the general similarity of the absorption characteristics of sound absorbers depending on porosity for their effectiveness, a comparison is made in Fig. 12-7. The acoustical tile, drapes, and carpet show the highest absorption above 500 Hz and relatively low absorption in the low-frequency region dominated by room modes. Coarse concrete blocks show a typical high-frequency absorption peak, but also a more unusual absorption peak around 200 Hz.

Glass-Fiber Low-Density Materials

Great quantities of glass-fiber materials are used in the acoustical treatment of recording studios, control rooms, and public spaces. This glass fiber can consist of both special, high-density materials, and ordinary, low-density building insulation. There is

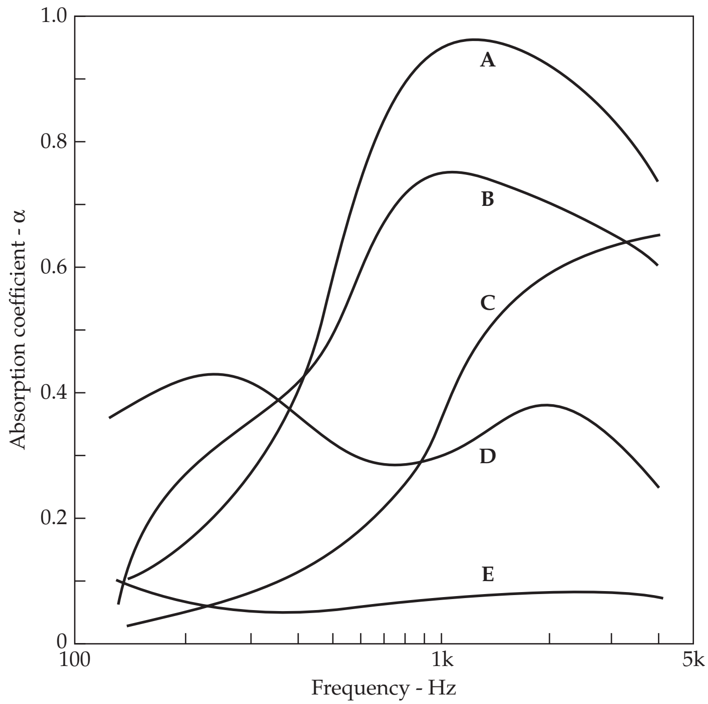


Figure 12-7 Sound absorption coefficients of typical porous materials A, B, and C show a similarity in general shape. Good high-frequency absorption and poor low-frequency absorption characterize porous absorbers. (A) High-grade acoustical tile. (B) Medium-weight (14 oz/yd^2) velour draped to half area. (C) Heavy carpet on concrete without padding. (D) Coarse concrete blocks, unpainted. (E) Coarse concrete blocks, painted.

evidence that thicker panels with a rear air cavity are preferred over thin panels, and low density is also preferred over high density. In wood or steel stud single-frame walls, staggered stud walls, and double walls, thermal insulating glass-fiber batts are commonly used.

This material usually has a density of about 1 lb/ft^3. Such material is often identified as R-11, R-19, or other such numbers. These R-prefix designations describe thermal insulating qualities, but are related to thickness. The thickness of R-8 is 2.5 in, R-11 is 3.5 in, and R-19 is 6 in.

Glass-fiber insulation commonly comes with a Kraft paper backing. Between walls, this paper has no significant acoustical effect, but if building insulation is used as a sound absorber on walls, perhaps behind a fabric facing, the paper becomes significant. Figure 12-8 compares the sound absorption efficiency of R-19 (6-in) and R-11 (3.5-in) glass fiber with the Kraft paper backing exposed, and with the glass fiber exposed to the incident sound. When the paper is exposed, it shields the glass fiber from sound above 500 Hz but has little effect below 500 Hz. The net effect is an absorption peak at 250 Hz (R-19) and 500 Hz (R-11), which may be important in room treatment. With insulation exposed, there is essentially perfect absorption above 250 Hz (R-19) or 500 Hz (R-11).

Glass-fiber insulation is an excellent and inexpensive absorbing material. When used in panels, a cosmetic and protective cover is usually required, but this is true of

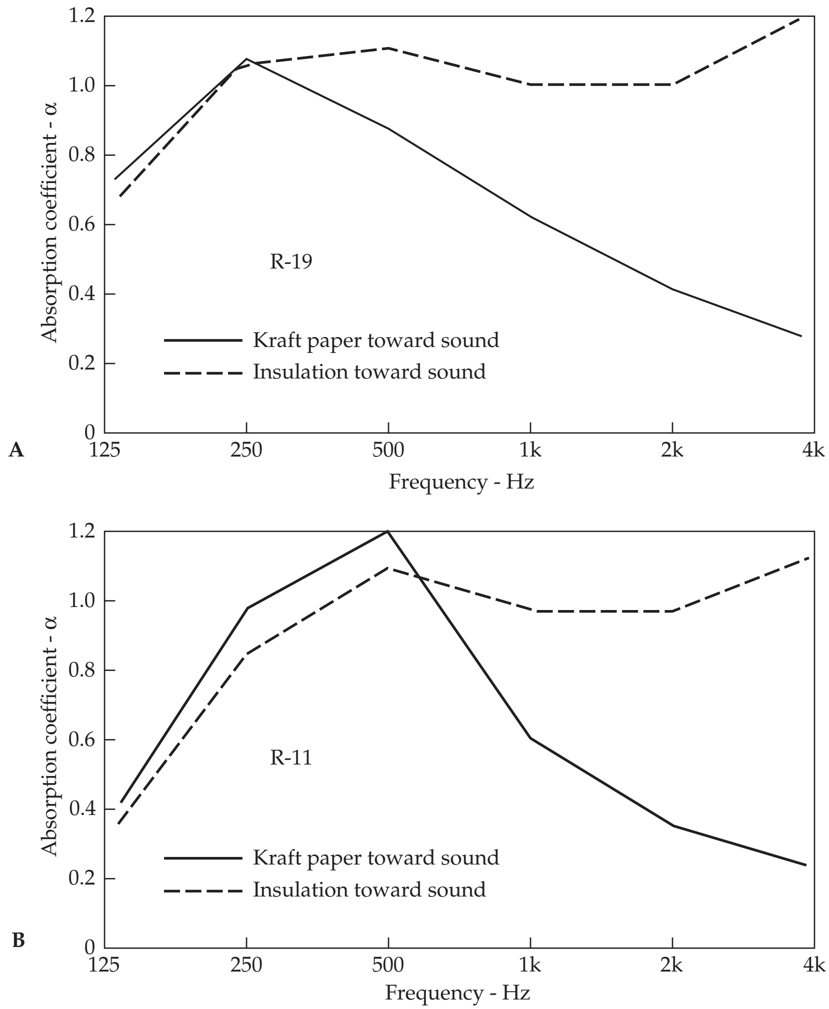


Figure 12-8 When ordinary building insulation is used as a wall treatment (perhaps with fabric facing), the position of the Kraft paper backing becomes important. (A) R-19 glass-fiber building insulation. (B) R-11 glass-fiber building insulation with mounting A (material directly on a hard surface).

denser materials as well. Fabric, expanded metal, metal lath, hardware cloth, or perforated vinyl wall covering can be used as a cover. Absorption coefficients greater than 1.0 can be obtained by spaced-apart glass-fiber panels.

Glass-Fiber High-Density Boards

Semirigid boards of glass fiber can be used in the acoustical treatment of rooms. This type of glass fiber is usually of greater density than glass-fiber building insulation. Typical of such materials are Johns-Manville 1000 Series Spin-Glass and Owens-Corning

Type 703 Fiberglas, both of 3 lb/ft^3 density. These are available in different thicknesses (e.g., 1 to 4 in) yielding different R values (e.g., 4.3 to 17.4). Other densities are available; for example, Type 701 has a density of 1.5 lb/ft^3 and Type 705 has a density of 6 lb/ft^3. These semirigid boards of glass fiber do not excel cosmetically; hence, they are usually covered with fabric. However, they do excel in sound absorption and are widely used for room treatment.

Glass-Fiber Acoustical Tile

Manufacturers of acoustical materials offer competitive lines of 12 × 12 in acoustical tiles. Surface treatments of the tiles include even-spaced holes, random holes, slots, fissured, or other special textures. These tiles are typically available from local building material suppliers. Such tiles are reputable products for noise and reverberation control provided they are used with full knowledge of their limitations. One of the problems of using acoustical tile in critical situations is that absorption coefficients are not always available for a specific tile. The average of the coefficients for eight cellulose and mineral fiber tiles of 3/4 in thickness is shown in Fig. 12-9. The range of the coefficients is indicated by the vertical lines. The average points could be used for 3/4-in tile for which no coefficients are available. Coefficients 20% lower would be a fair estimate for 1/2-in tiles. When acoustical tiles are used in a drop ceiling, they function primarily as porous absorbers at mid and high frequencies. However, the plenum airspace above the tiles causes them to also perform somewhat as panel absorbers at lower frequencies.

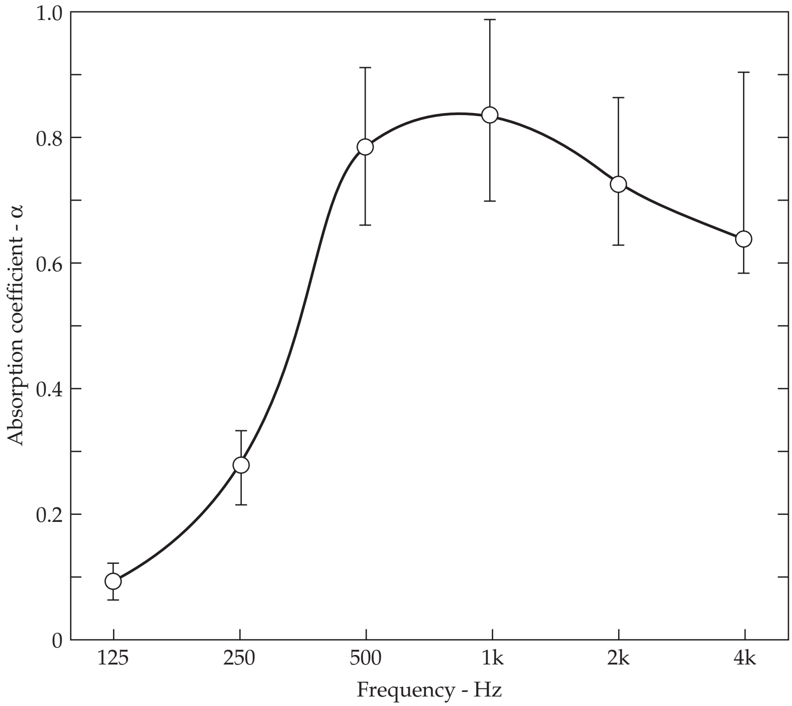


Figure 12-9 The average absorption characteristics of eight acoustical tile brands of 3/4-in thickness. The vertical lines show the spread of the data.

Effect of Thickness of Absorbent

It is logical to expect greater sound absorption from thicker porous materials, but this logic holds primarily for lower frequencies. Absorption is greatest when the porous material is placed at a distance of a quarter wavelength ($\lambda/4$) from a hard reflective surface (or odd multiples of this dimension); this is the point where particle velocity is greatest; in practice, this can be difficult to do. Figure 12-10 shows the effect of varying absorbent thickness where the absorbent is mounted directly on a solid surface (mounting type A). There is little difference above 500 Hz in increasing the absorbent thickness from 2 to 4 in, but there is considerable improvement below 500 Hz as thickness is increased. There is also a proportionally greater gain in overall absorption in a 1-in increase of thickness in going from 1 to 2 in than going from 2 to 3 in or 3 to 4 in. A 4-in thickness of glass-fiber material of 3-lb/ft^3 density has essentially perfect absorption over the 125-Hz to 4-kHz region.

Effect of Airspace behind Absorbent

Effective low-frequency absorption can also be achieved by spacing a porous absorbent out from the wall. A spaced porous absorber can be as effective as a non-spaced absorber of the same thickness. This is an inexpensive way to get improved performance—within limits. Figure 12-11 shows the effect on the absorption coefficient of furring 1-in glass-fiber wallboard out from a solid wall. Spacing 1-in material out 3 in makes its absorption approach that of the 2-in material of Fig. 12-10 mounted directly on the wall.

Effect of Density of Absorbent

Glass fiber and other materials come in various densities ranging from soft thermal insulation batts to semirigid and rigid boards. All of these have their proper place in the acoustical treatment of spaces. Generally sound is able to penetrate the interstices of a

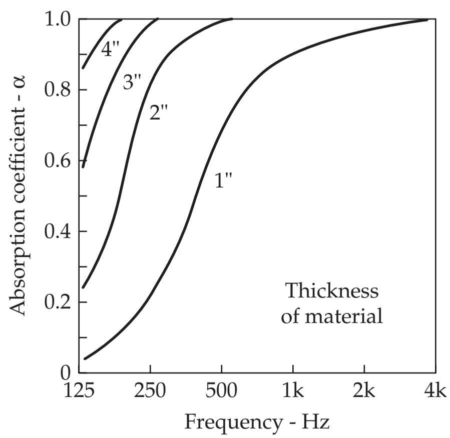


Figure 12-10 The thickness of glass-fiber sound-absorbing material determines the low-frequency absorption. In this example, the material (such as Type 703 Fiberglas) has a density of 3 lb/ft^3 and is mounted directly on a hard surface.

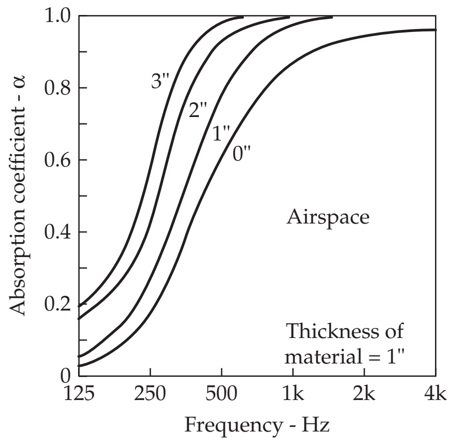


Figure 12-11 The low-frequency absorption of 1-in glass-fiber board is improved materially by spacing it out from the solid wall.

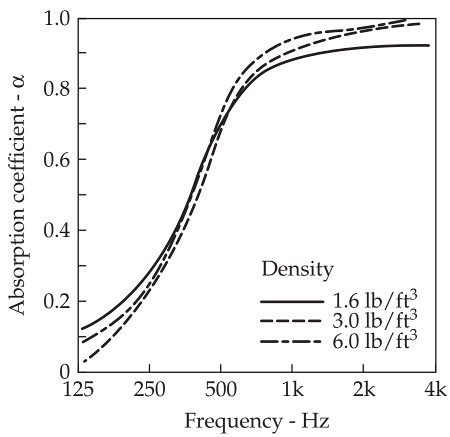


Figure 12-12 The density of glass-fiber absorbing material has relatively little effect on absorption in the range of 1.5 to 6 lb/ft^3. The material is mounted directly on a solid wall.

high-density, harder surface material as well as a soft type. Figure 12-12 shows that there is relatively little difference in the absorption coefficient as the density is varied over a range of almost 4 to 1. However, for very low densities, the fibers are so widely spaced that absorption suffers. For extremely dense boards, the surface reflection is high and sound penetration low.

Open-Cell Foams

Flexible polyurethane and polyester foams are widely used in noise quieting of automobiles, aircraft, machinery, and in various industrial applications. Foams also find application as sound absorbers in architectural applications, including recording studios

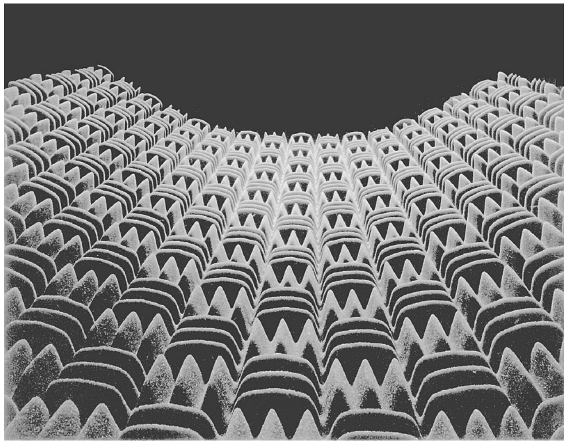


Figure 12-13 Sonex contoured acoustical foam simulating anechoic wedges. This is an open-cell type of foam. (*Pinta Acoustic, Inc.*)

and home listening rooms. Figure 12-13 is a photograph of one form of Sonex, a foam product contoured to simulate the wedges used in anechoic rooms. These are shaped in male and female molds and come in meshed pairs. This material can be glued or stapled to the surface to be treated. Sonex products are manufactured by Pinta Acoustic, Inc.

The sound absorption coefficients of Sonex for thicknesses of 2, 3, and 4 in are shown in Fig. 12-14 for mounting A (material directly on a hard surface). The 2-in glass fiber of Fig. 12-10 (such as Type 703 Fiberglas) is considerably superior acoustically to the 2-in Sonex, but some factors should be considered in this comparison: (1) Type 703 has a density of 3 lb/ft^3 while Sonex is 2 lb/ft^3. (2) In 2-in Sonex, the wedge height and

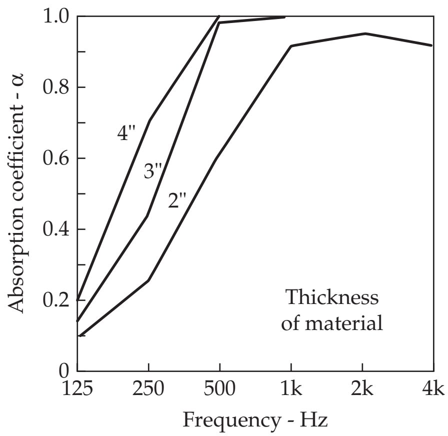


Figure 12-14 Absorption coefficients of Sonex contoured foam of various thicknesses for mounting A. (*Pinta Acoustic, Inc.*)

the average thickness is far less, while the 703 thickness prevails throughout. (3) Comparing the two products is, in a sense, specious because the higher cost of Sonex is justified in the minds of many by its appearance and ease of mounting rather than only acoustical considerations.

Drapes as Sound Absorbers

Drapes are a porous type of sound absorber because air can flow through the fabric. This penetration allows absorption to occur. Variables affecting absorbency include type and weight of material, degree of fold, and distance from the wall. The heavier the fabric, the greater the sound absorption. Heavy velour drapes can provide good absorption, whereas lightweight drapes provide practically no absorption. Figure 12-15 compares the absorption of 10, 14, and 18 oz/yd^2 velour hung straight and presumably at some distance from the wall. In this particular evaluation, the heavy drape using 18-oz/yd^2 velour provides greater absorption centered in the 1,000-Hz region.

The higher the degree of folding, the greater the absorption. This is mainly because folds increase the amount of surface area presented to sound. The effect is shown in Fig. 12-16. The "draped to 7/8 area" means that the entire 8/8 area is drawn in only slightly (1/8) from the flat condition. As can be seen, the deeper the drape fold, the greater the absorption.

The distance a drape is hung from a reflecting surface can have a great effect on its absorption efficiency. In Fig. 12-17A, a drape or other porous material is hung parallel to a solid wall, and the distance *d* between the two is varied. The frequency of the sound impinging on the porous material is held constant at 1 kHz. If the sound absorption provided by the porous material is measured, we find that it varies greatly as the distance *d* from the wall is changed. Examination reveals that the maxima and minima of absorption are related to the wavelength of the impinging sound. The wavelength of sound λ is the speed of sound divided by frequency, which in the case of 1,000 Hz, is 1,130/1,000 = 1.13 ft or about 13.6 in. A quarter wavelength is 3.4 in, and a half wavelength

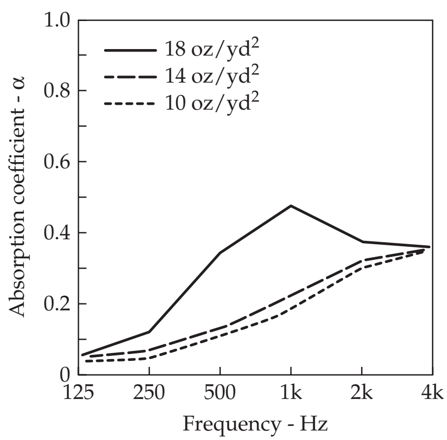


Figure 12-15 The sound absorption of velour hung straight for three different weights of fabric. (*Beranek.*)

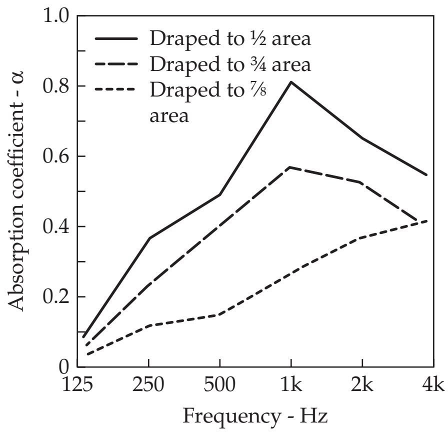


FIGURE 12-16 The effect of draping on the sound absorption of drapes. "Draped to 1/2 area" means that folds are introduced until the resulting drape area is half that of the straight fabric. (*Mankovsky.*)

is 6.8 in. There are absorption peaks at one-quarter wavelength, and if we carry it further than indicated in Fig. 12-17A, peaks are present at each odd multiple of quarter wavelengths. Absorption minima occur at even multiples of quarter wavelengths.

This effect is explained by reflections of the sound from the solid wall. At the wall surface, pressure will be highest, but air particle velocity will be zero because the sound waves cannot move the wall. However, at a quarter wavelength from the wall, pressure is zero, and air particle velocity is maximum. By placing the porous material, such as a drape, a quarter wavelength from the wall, it will have maximum absorbing effect at that corresponding frequency because the particle velocity is maximum at the drape and the greatest frictional losses occur. The same effect occurs at odd multiples of $\lambda/4$, such as $3\lambda/4$, $5\lambda/4$, $7\lambda/4$, and so on. At a half-wavelength distance, particle velocity is at a minimum; hence, absorption is minimum. In practice, because drapes usually have some degree of fold, the quarter-wavelength point occurs at different distances from the wall. Thus, the frequency peaks and nulls tend to be broadened, with less specific effect on response.

In Fig. 12-17B, the spacing of the drape from the wall is held constant at 12 in as the absorption is measured at different frequencies. The same variation of absorption is observed: maximum when spacing from the wall is at odd quarter wavelengths and minimum at even quarter wavelengths. At this particular spacing of 12 in (1 ft), a wavelength of spacing occurs at 1,130/1 = 1,130 Hz, a quarter wavelength at 276 Hz, a half wavelength at 565 Hz, and so on. When referring to quarter wavelengths, sine waves are inferred. Absorption measurements are invariably made with bands of random noise. Hence, we must expect the variations of Fig. 12-17B to be averaged by the use of such bands.

Figure 12-18 shows reverberation chamber measurements of the absorption of 19 oz/yd^2 velour. The solid plot is presumably for a drape well removed from any walls. The other plots, very close together, are for the same material spaced about 4 and 8 in from the wall. The 4-in distance is one wavelength at 3,444 Hz; the 8-in distance is

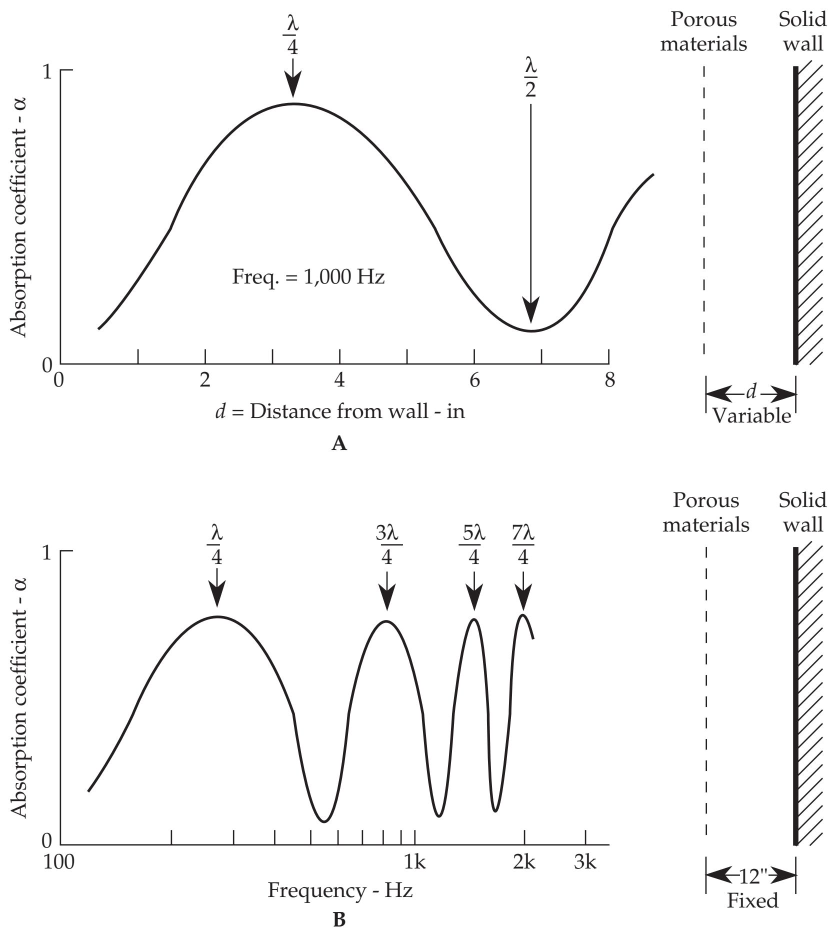


FIGURE 12-17 The sound absorption of a porous material such as a drape varies with the distance from a hard wall. (A) The maximum absorption is achieved when the drape is one-quarter wavelength from the wall; the minimum absorption occurs at a half wavelength. (B) The sound absorption of a porous material hung at a fixed distance from a wall will show maxima at spacings of a quarter wavelength and odd multiples of a quarter wavelength.

at 1,722 Hz. The odd multiples of both the 4- and 8-in quarter wavelengths are seen on the upper part of Fig. 12-18.

The absorption of the velour is greater when spaced from the wall, and the effect is greatest in the 250- to 1,000-Hz region. At 125 Hz, the 10- and 20-cm spacing adds practically nothing to the drape absorption because, at 125 Hz, the quarter wavelength spacing is 2.26 ft.

Carpet as Sound Absorber

Carpet commonly dominates the acoustical picture in many types of spaces. It is the one amenity the owner often specifies in advance and the reason is more often comfort and appearance than acoustic. Carpet and its underlay can provide significant absorption

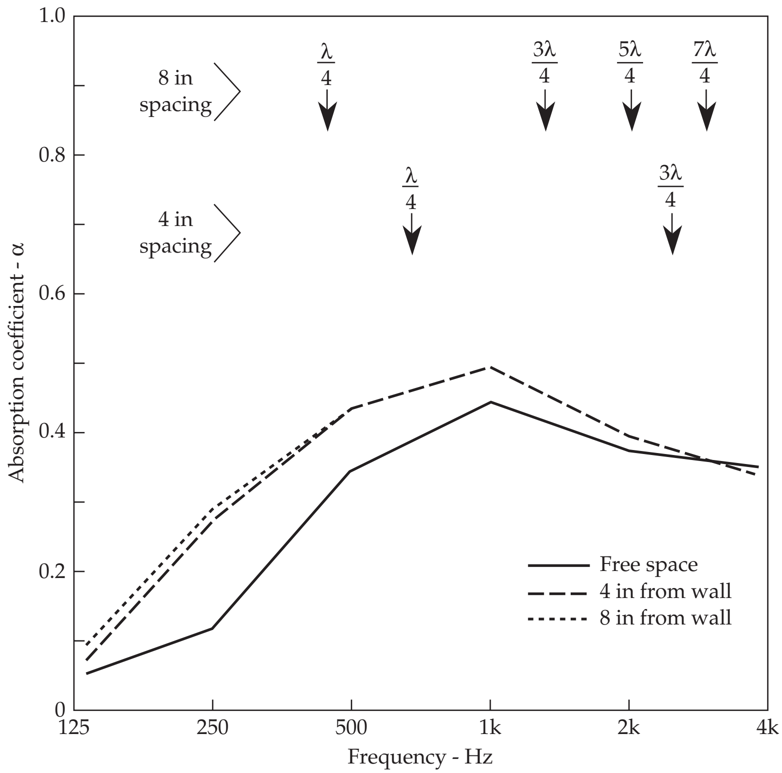


FIGURE 12-18 Measurements of sound-absorption coefficients of a velvet material (19 oz/yd^2) in free space, and about 4 in and 8 in from a solid wall. Also indicated are the points at which absorption will increase due to wall reflection. (*Mankovsky.*)

at mid and high frequencies. Suppose that carpet is placed in a recording studio with a floor area of 1,000 ft^2. Further, suppose that reverberation time is specified to be 0.5 sec, which requires 1,060 sabins of absorption in this room. At higher audio frequencies, a heavy carpet and pad with an absorption coefficient of around 0.6 gives 600 sabins of absorption at 4 kHz. This is about 57% of the required absorption for the entire room before the absorption of walls and ceiling are even considered. The acoustical design is quite limited before it is started.

There is another, more serious problem. This high absorbance of carpet only occurs at higher audio frequencies. Carpet having an absorption coefficient of 0.60 at 4 kHz may offer a coefficient of only 0.05 at 125 Hz. In other words, the 1,000 ft^2 of carpet introduces 600 sabins at 4 kHz but only 50 sabins at 125 Hz. This is a major problem encountered in many acoustical treatments. The frequency-unbalanced absorption of carpet often must be compensated for in other ways, principally with resonant-type, low-frequency absorbers.

To compound the problem of unbalanced absorption of carpet, dependable absorption coefficients are hard to come by. An assortment of types of carpet and variables in underlay add to the uncertainty. Experience and judgment must be used when deciding which absorption coefficients should be employed, particularly in rooms with wall-to-wall carpeting.

Effect of Carpet Type on Absorbance

There are considerable variations in sound absorption between types of carpet. Figure 12-19 shows the difference between a heavy Wilton carpet and a velvet carpet with and without a latex backing. The latex backing seems to increase absorption materially above 500 Hz and to decrease it a modest amount below 500 Hz.

Effect of Carpet Underlay on Absorbance

Foam rubber, sponge rubber, felts, polyurethane, or combinations of these are often used as padding under a carpet. Foam rubber is made by whipping a latex water dispersion, adding a gelling agent, and pouring into molds. The result is always open-celled. Sponge rubber, on the other hand, formed by chemically generated gas bubbles, can yield either open or closed cells. Open cells provide the interstices required for good sound absorption while closed cells do not.

The influence of underlay on carpet absorption is significant. Figure 12-20 shows reverberation-chamber measurements of absorption coefficients for a single Axminster type of carpet with different underlay conditions. Contours A and C show the effect of

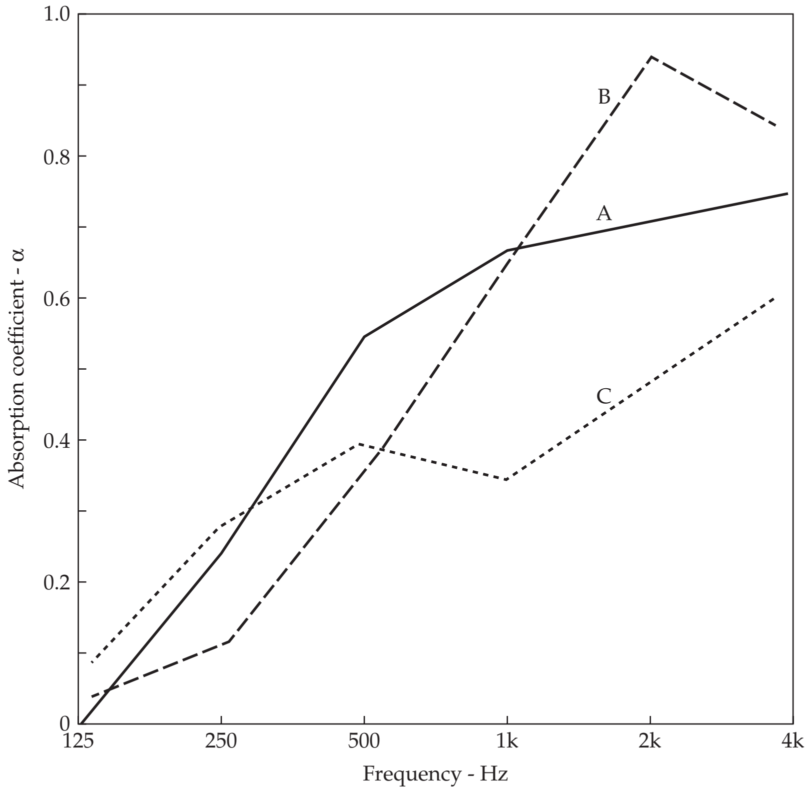


FIGURE 12-19 A comparison of the sound-absorption characteristics of three different types of carpet. (A) Wilton, pile height 0.29 in, 92.6 oz/yd^2. (B) Velvet, latex backed, pile height 0.25 in, 76.2 oz/yd^2. (C) The same velvet without latex backing, 37.3 oz/yd^2, all with 40-oz hair felt underlay. (*Harris*.)

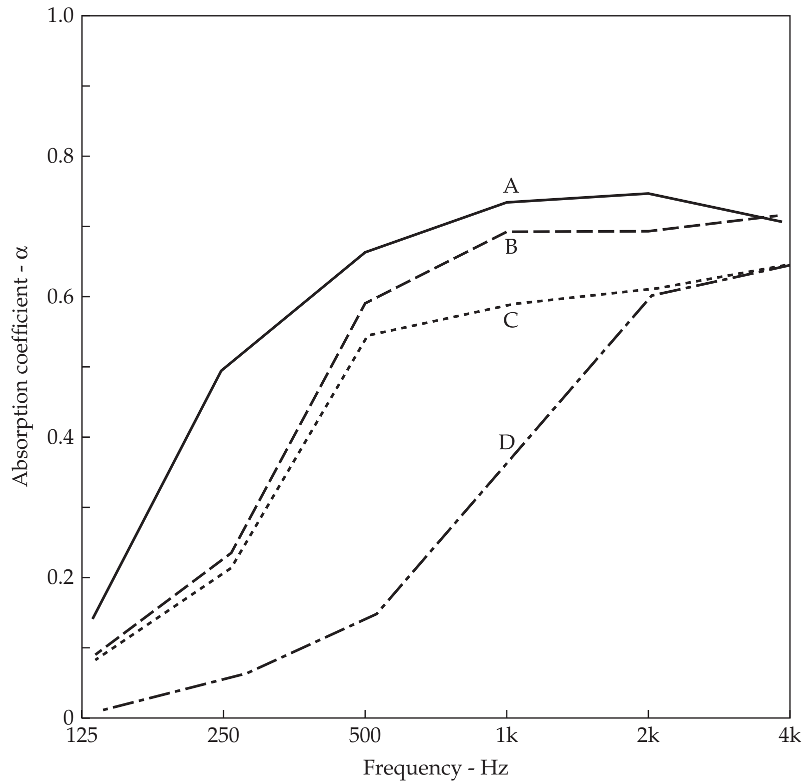


FIGURE 12-20 Sound-absorption characteristics of the same Axminster carpet with different underlay. (A) 80-oz hair felt. (B) Hair felt and foam. (C) 40-oz hair felt. (D) No underlay, on bare concrete. (*Harris.*)

hair felt of 80 and 40 oz/yd^2 weight. Contour B shows an intermediate combination of hair felt and foam. While these three contours differ considerably, they all stand in contrast to contour D for the carpet laid directly on bare concrete. We conclude that the padding underneath the carpet contributes markedly to overall carpet absorption.

Carpet Absorption Coefficients

The absorption coefficients plotted in Figs. 12-19 and 12-20 are taken from a study of carpet characteristics. The coefficients in Fig. 12-21 and included in App. C are plotted for comparison with Figs. 12-19 and 12-20. The absorptivity of carpets varies widely, which accounts for some of the great variability that confronts any designer of acoustical systems.

Sound Absorption by People

The people comprising a concert-hall audience account for a significant portion of the sound absorption of the room—perhaps as much as 75% for a full house. It can also

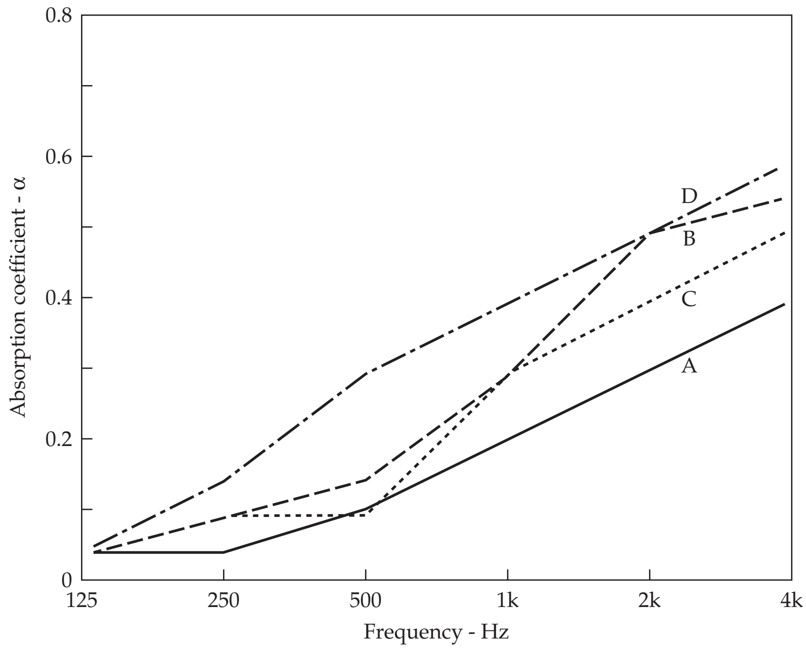


FIGURE 12-21 Carpet absorption coefficients. (A) 1/8-in pile height. (B) 1/4-in pile height. (C) 3/16-in combined pile and foam. (D) 5/16-in combined pile and foam. Compare these contours with those of Figs. 12-19 and 12-20.

make an acoustical difference whether one or several people are in a small monitoring room. The problem is how to rate human absorption and how to use it in calculations. One method uses the surface area over which an audience is seated. Or, we may simply consider the number of people present. In any case, the absorption units (sabins) due to people must be considered at each frequency and then added to the sabins of the carpet, drapes, and other absorbers in the room at each frequency. Table 12-2 lists the absorption per person of informally dressed college students in a classroom along with a range of absorption for more formally dressed people in an auditorium environment.

	Frequency (Hz)					
	125	**250**	**500**	**1,000**	**2,000**	**4,000**
College students informally dressed, seated in tablet arm chairs	NA	2.5	2.9	5.0	5.2	5.0
Audience seated, depending on spacing and upholstery of seats	2.5–4.0	3.5–5.0	4.0–5.5	4.5–6.5	5.0–7.0	4.5–7.0

TABLE 12-2 Sound Absorption by People (Sabins per Person)

For 1 kHz and higher, the absorption offered by college students in informal attire in the furnishings of a classroom falls at the lower edge of the range of a more average audience. The low-frequency absorption of the students, however, is considerably lower than that of the more formally dressed people. The rule of thumb employed by some acousticians simply attributes 5 sabins at 500 Hz, per seated person.

Sound propagated across rows of people, as in an auditorium or concert hall, is subjected to an unusual type of attenuation. In addition to the normal decrease in sound with distance from the stage, there is an additional dip of as much as 15 to 20 dB around 150 Hz and spreading over the 100- to 400-Hz region. In fact, this is not strictly an audience effect because it prevails even when the seats are empty. A similar dip in sound-pressure level affects important first reflections from the side walls. This apparently results from interference. The angle of incidence also plays a role. When an audience is seated on a relatively flat floor, the angle of sound incidence is low and there is greater absorption. With a higher angle of incidence (e.g., in stadium seating), there is less absorption.

Sound Absorption in Air

For large rooms, the absorption of sound by the air in the space is significant for frequencies of about 2,000 Hz and above. Depending on the acoustical design, air absorption could account for 20% to 25% of the total absorption in a large room. Air absorption can be estimated from:

$$A_{\mathrm{air}} = mV \tag{12-5}$$

where m = air attenuation coefficient, sabins/ft^3 or sabins/m^3
V = volume of room, ft^3 or m^3

The value of the air attenuation coefficient m varies with humidity. With humidity between 40% and 60%, the values of m at 2,000, 4,000, and 8,000 Hz are: 0.003, 0.008, and 0.025 sabins/ft^3 and 0.009, 0.025, and 0.080 sabins/m^3, respectively.

For example, a church seating 2,000 people has a volume of 500,000 ft^3. At 2,000 Hz and 50% relative humidity, the air absorption is 0.003 sabins/ft^3. This yields 1,500 sabins of absorption at 2,000 Hz.

Panel (Diaphragmatic) Absorbers

The absorption of sound at lower audible frequencies can be effectively achieved by resonant (also called reactive) absorbers such as panel (diaphragmatic) absorbers and Helmholtz (volume) resonators. Glass fiber and acoustical tiles are common forms of porous absorbers in which the sound energy is dissipated as heat in the interstices of the fibers. However, the absorption of glass fiber and other fibrous absorbers at low audio frequencies is quite poor. To absorb well, the thickness of the porous material must be comparable to the wavelength of the sound. At 100 Hz, the wavelength is 11.3 ft, and using any porous absorber approaching this thickness would be impractical. For this reason, resonant absorbers are often used to obtain absorption at low frequencies.

A mass suspended from a spring will vibrate at its natural frequency. Panels designed with an air cavity behind them act similarly. The mass of the panel and the springiness of the air in the cavity are together resonant at some particular frequency.

Sound is absorbed as the panel is flexed because of the damping caused by frictional heat losses of the material within the panel. (Similarly, a mass on a spring will stop oscillating because of damping.) Absorption provided by panel absorbers is usually relatively modest because the resonant motion also radiates some sound energy. Panels made of limp materials with high damping provide greater absorption.

Damping increases as the velocity of the panel increases, and velocity is highest at the resonant frequency. There the absorption of sound is maximal at the frequency at which the structure is resonant. As noted, the enclosed and sealed air cavity behind the panel acts as a spring; the greater the depth of the airspace, the less stiff the spring. Likewise, a smaller airspace acts as a stiff spring. The frequency of resonance for a flat, unperforated panel can be estimated from:

$$f_0 = \frac{170}{\sqrt{(m)(d)}} \tag{12-6}$$

where f_0 = frequency of resonance, Hz
m = surface density of panel, lb/ft^2 or kg/m^2
d = depth of airspace, in or m

Note: In metric units, change 170 to 60.

For example, consider a piece of 1/4-in plywood spaced out from the wall on 2×4 studs on edge, which yields an airspace of about 3-3/4 in behind. The surface density of 1/4-in plywood, 0.74 lb/ft^2, can be measured or found in references. Substituting, we obtain a resonant frequency of about 102 Hz.

Figure 12-22 shows panel resonant frequencies as a function of cavity depth in inches, and surface density in ounces per square foot. Knowing the thickness of the plywood and the depth of the space behind the plywood, the frequency of resonance can be read off the diagonal lines. Equation (12-6) applies to membranes and diaphragms of materials other than plywood such as Masonite, fiberboard, or even Kraft paper. For other than plywood, the surface density must be determined. The surface density is easily found by weighing a piece of the material of a known area. The surface area of a panel absorber should be at least 5 ft^2.

How accurate are Eq. (12-6) and Fig. 12-22? Actual measurements on three plywood membrane absorbers are shown in Fig. 12-23. Contour A shows the simple case of 3/16-in plywood panels on 2-in battens. This structure is resonant at about 175 Hz. The peak coefficient is about 0.3 which is about as high as one can expect for such structures. Contour B is for 1/16-in plywood with a 1-in glass-fiber blanket and 1/4-in airspace behind it. Contour C is the same except for a 1/8-in panel. Note that the glass-fiber filler has about doubled the peak absorption. The glass fiber has also shifted the peaks to a point about 50 Hz lower. Such calculations of the frequency of peak absorption at resonance are not perfect, but they are a good first approximation of sufficient accuracy for most purposes.

Absorption is increased when the cavity is filled with a porous material such as glass-fiber insulation. This is because the insulation increases damping. The material may be stuffed loosely inside the cavity or attached to the back of the panel. The panel is most effective when placed at a pressure maximum for the desired absorption frequency; this might be on an end wall, a midpoint, or a corner of a room. Panels are relatively ineffective when placed at pressure minima.

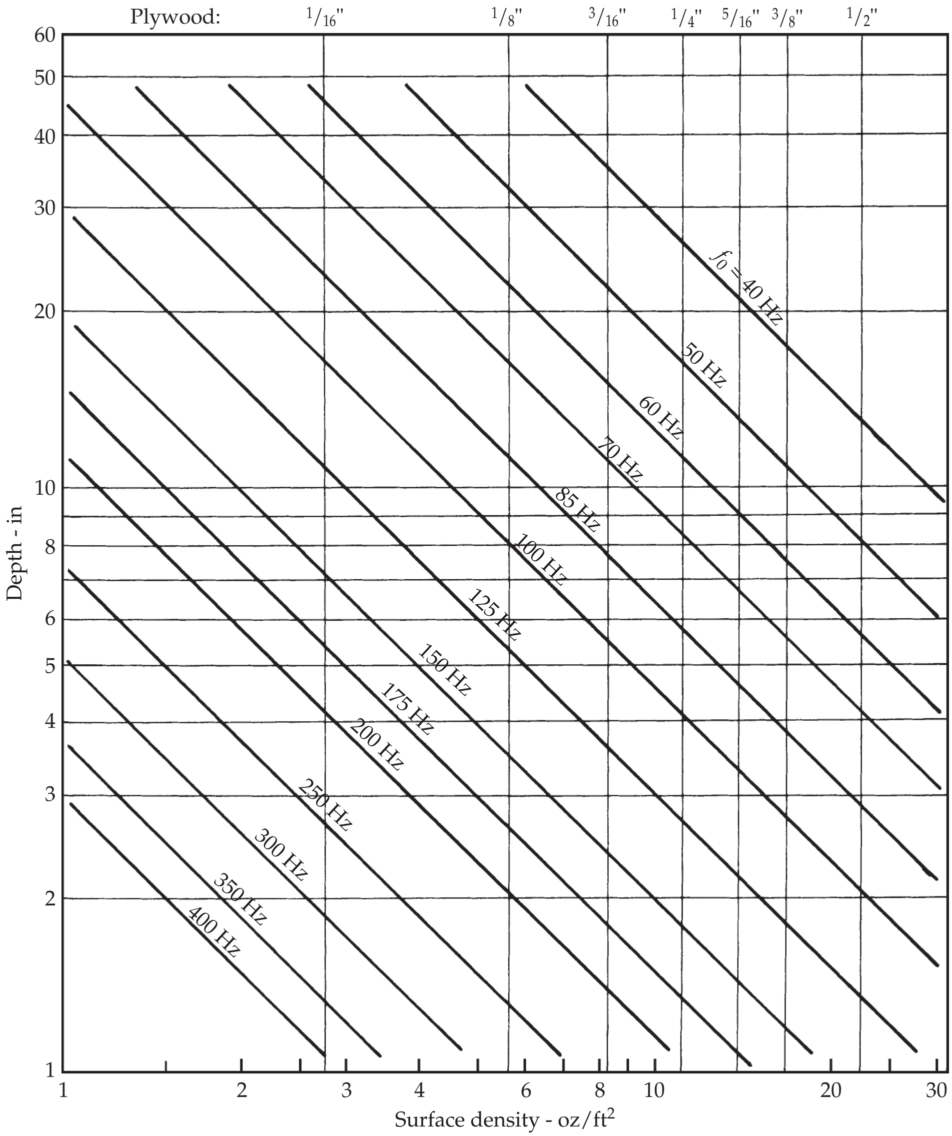


Figure 12-22 Design chart for resonant panel absorbers. (See also Fig. 12-34.)

Some music rooms owe their acoustical excellence to the low-frequency absorption offered by extensive paneled walls. Plywood or tongue-and-groove flooring or subflooring vibrates as a diaphragm and contributes to low-frequency absorption. Drywall construction on walls and the ceiling does the same thing. All such components of absorption must be included in the acoustical design of a room, large or small.

Drywall or gypsum board plays an important role in the construction of homes, studios, control rooms, and other spaces. Drywall absorbs sound by a flexural, diaphragmatic

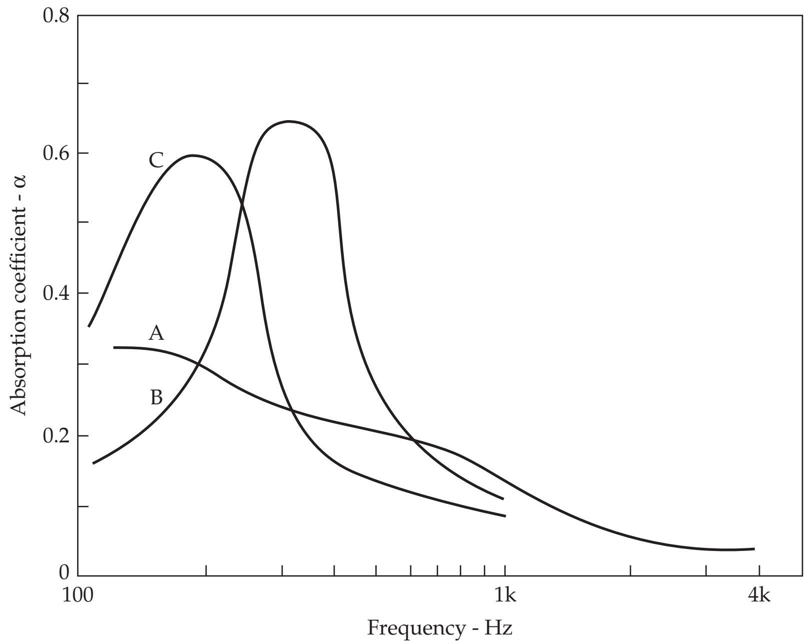


Figure 12-23 Absorption measurements of three panel absorbers. (A) 3/16-in plywood with 2-in airspace. (B) 1/16-in plywood with 1-in mineral wool and 1/4-in airspace. (C) The same as (B) but for 1/8-in panel.

action, working as a resonant system. Drywall is particularly important in the absorption of low-frequency sound. Usually, such low-frequency absorption is welcome, but in larger spaces designed for music, drywall surfaces can absorb so much low-frequency sound as to prevent the achievement of the desired reverberate conditions. Drywall of 1/2-in thickness on studs spaced at 16 in offers an absorption coefficient of 0.29 at 125 Hz and even higher at 63 Hz (which would be of interest in music recording studios). Drywall absorption in small audio rooms must be recognized and its low-frequency absorption included in calculations. This is sometimes difficult because different thicknesses of drywall are used, and the frequency of peak absorption varies according to thickness and airspace. A 1/2-in drywall has a surface mass of 2.1 lb/ft^2 and double 5/8-in drywall has a surface mass of 5.3 lb/ft^2. With an airspace of 3-3/4-in, the 1/2-in panel resonates at 60.6 Hz, and the double 5/8-in panel resonates at 38.1 Hz.

We have noted that porous materials commonly show their greatest absorption in the high-frequency region. Vibrating panel arrangements show their best absorption in the low frequencies. In treating small listening rooms and studios, we find that structures giving good low-frequency absorption are invaluable in controlling room modes.

Panel absorbers are quite simple to build. An example of a panel absorber to be mounted on a flat wall or ceiling surface is shown in Fig. 12-24. A 1/4- or 1/16-in plywood panel is fastened to a wooden framework to give the desired spacing from the wall. A glass or mineral fiber blanket of 1 to 1-1/2 in is glued to the wall surface. An airspace of 1/4 or 1/2 in should be maintained between the absorbent and the rear surface of the plywood panel.

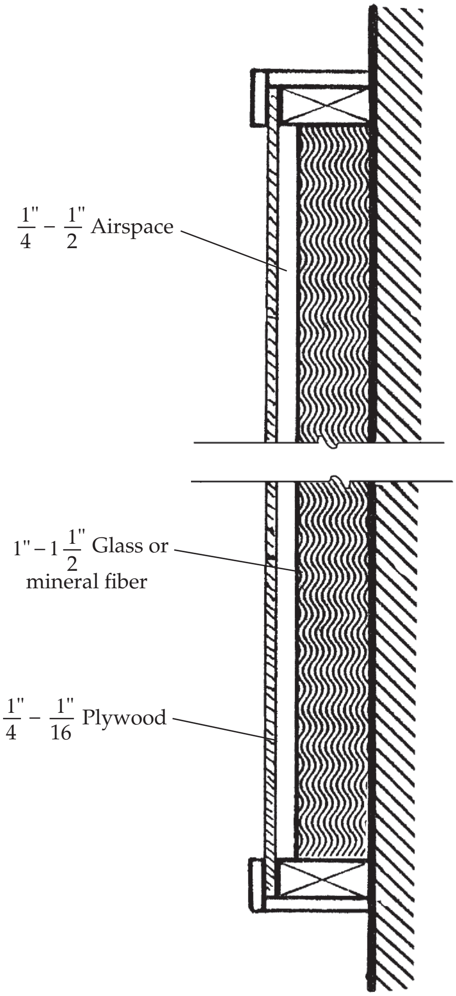


Figure 12-24 Typical resonant panel absorber with wall mounting.

A corner panel absorber is shown in Fig. 12-25. For computations, an average depth is used. Depths greater and smaller than the average simply mean that the peak of absorbance is broader than that of an absorber with uniform depth. Spacing the absorber 1/4 to 1/2 in from the rear of the plywood panel is simple if a mineral fiberboard such as Tectum is used. Using a flexible blanket of glass fiber requires support by hardware cloth, open-weave fabric, or expanded metal. For applications in which reflected mid/high-frequency sound from the panel absorber might create problems, a facing of glass-fiber board would not interfere with the low-frequency absorbing action if it was spaced to avoid damping of the vibration of the plywood panel. All room modes terminate in the corners of a room. A corner panel absorber could be used to control such modes.

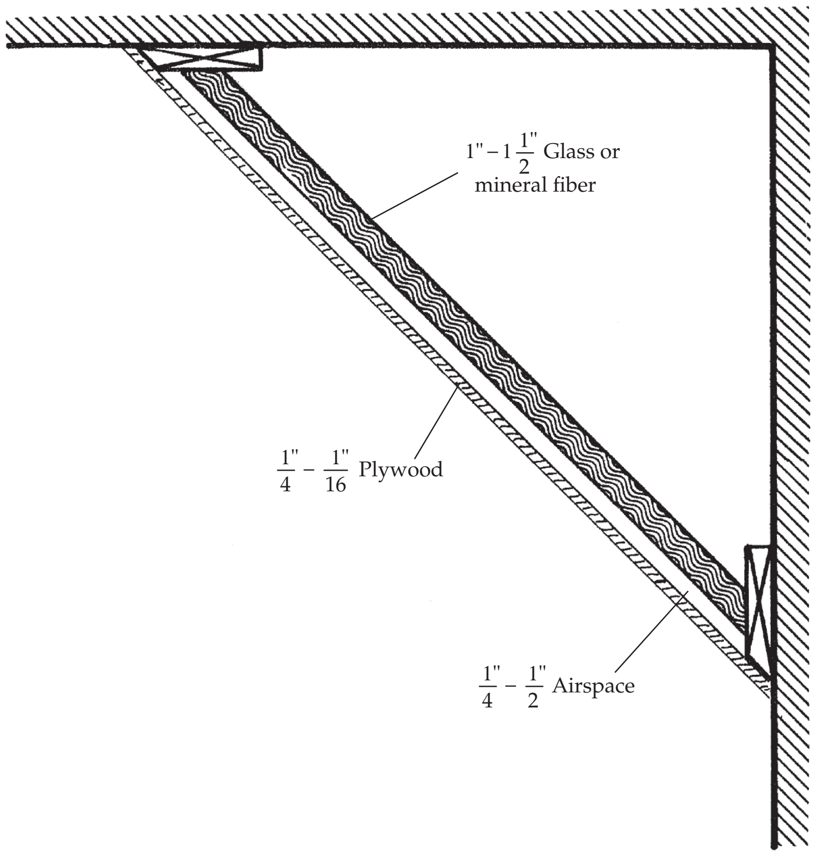


Figure 12-25 Typical resonant panel absorber with either vertical or horizontal corner mounting.

Polycylindrical Absorbers

Flat-panel absorbers using a plywood or other hardboard skin can provide a reasonable amount of absorption. But wrapping a panel around semicylindrical bulkheads can yield additional benefits. With polycylindrical elements (polys), it is acoustically possible to achieve a good diffuse field along with liveness and brilliance, factors that tend to oppose each other in rooms with flat surfaces. The larger the chord dimension, that is, the larger the width of the semicircular bulkhead, the better the bass absorption. Above 500 Hz there is little significant difference between the polys of different sizes.

Polys perform differently, depending on whether they are empty or filled with absorbent material. Figures 12-26C and D show the increase in bass absorption resulting from filling the cavities with absorbent. If needed, this increased bass absorption can be easily achieved by simply filling the polys with glass fiber. If the bass absorption is not needed, the polys can be used empty. This adjustability can be an important asset in the acoustical design of listening rooms and studios.

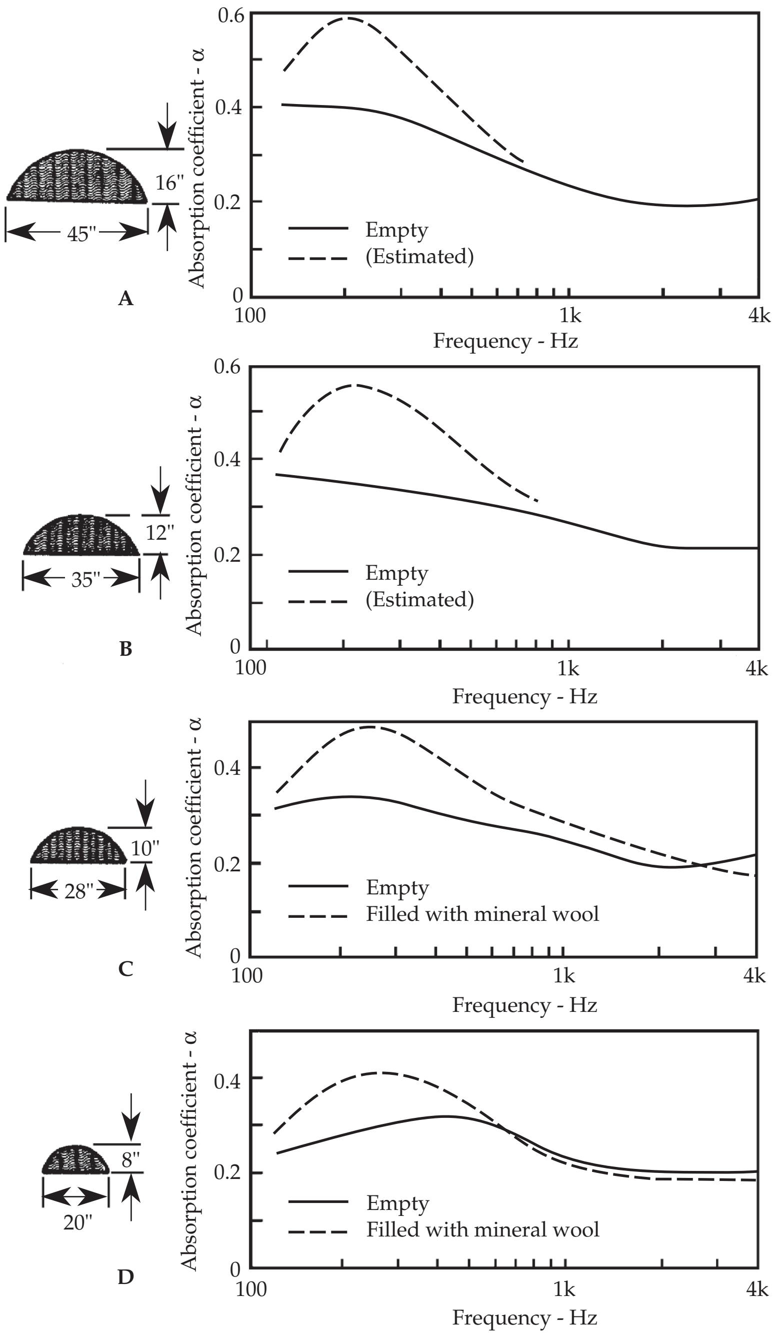


FIGURE 12-26 Measured absorption of polycylindrical absorbers of various chord and height dimensions. (A and B) Only empty poly data are available; the broken lines are estimated absorption when filled with mineral wool. (C and D) Both empty polys and polys filled with mineral wool. (*Mankovsky.*)

Polycylindrical Absorber Construction

The construction of polycylindrical absorbers is fairly simple. A framework for vertical polys is shown in Fig. 12-27, mounted above a structure intended for a low-frequency slat absorber. The variable chord dimensions are apparent, as are the random placement of bulkheads so that cavities will be of various volumes, resulting in different natural cavity frequencies. It is desirable that each cavity be essentially airtight, isolated from adjoining cavities by well-fitted bulkheads and framework. Irregularities in the rear wall can be sealed with a nonhardening acoustical sealant. The bulkheads of each poly are cut to the same radius on a bandsaw. Sponge rubber weather stripping with an adhesive on one side is stuck to the edge of each bulkhead to ensure a tight seal against the plywood or other hardboard cover. If such precautions are not taken, rattles and coupling between cavities can result.

The polys of Fig. 12-28 use 1/8-in tempered Masonite as the poly skin. A few hints can simplify the job of stretching this skin. In Fig. 12-28, slots of a width to snugly fit the

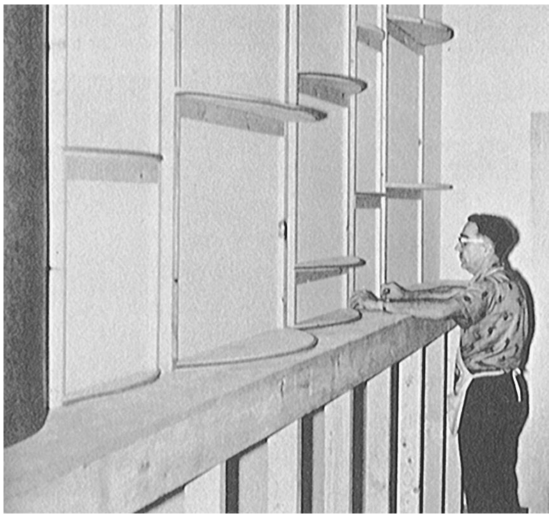


FIGURE 12-27 The construction of polys in a motion picture sound mixing studio. Foam rubber anti-rattle strips are applied on the edge of each bulkhead. Also note the random spacing of the bulkheads. (*Moody Institute of Science.*)

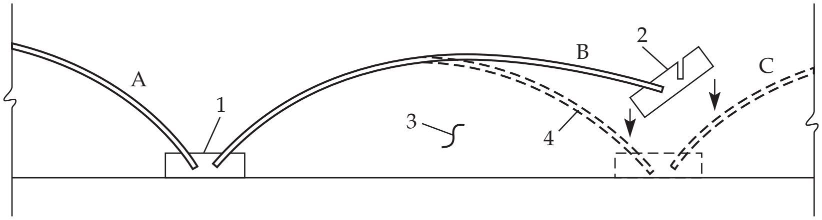


FIGURE 12-28 The method of stretching the plywood or hardboard skin over the poly bulkheads shown in Fig. 12-27.

Masonite are cut along the entire length of strips 1 and 2 with a radial saw. Assume that poly A is already mounted and held in place by strip 1, which is nailed or screwed to the wall. Working from left to right, the next job is to mount poly B. First, the left edge of Masonite sheet B is inserted in the remaining slot of strip 1. The right edge of Masonite sheet B is then inserted in the left slot of strip 2. If all measurements and cuts have been accurately made, swinging strip 2 against the wall should make a tight seal over the bulkheads 3 and weather stripping 4. Securing strip 2 to the wall completes poly B. Poly C is mounted in a similar fashion and so on to the end of the series of polys. In some designs, the axes of symmetry of the polys on the side wall are made perpendicular to those on the rear wall. If polys were used on the ceiling, their axes should be perpendicular to both the others.

It is practical and acceptable to construct each poly as an entirely independent structure rather than build them on the wall. Such independent polys can be spaced at will.

Bass Traps: Low-Frequency Absorption by Resonance

The term "bass trap" describes many kinds of low-frequency sound absorbers, including panel absorbers, but perhaps this term should be reserved for a particular type of reactive cavity absorber. Sound absorption at the lowest octave or two of the audible spectrum is often difficult to achieve. The bass trap is commonly used in recording studio control rooms to reduce standing waves at these bass frequencies. A true bass trap is shown in Fig. 12-29. It is a box or cavity of critical depth and with a mouth opening of size to suit a particular purpose. In particular, a bass trap is a tuned cavity with a depth

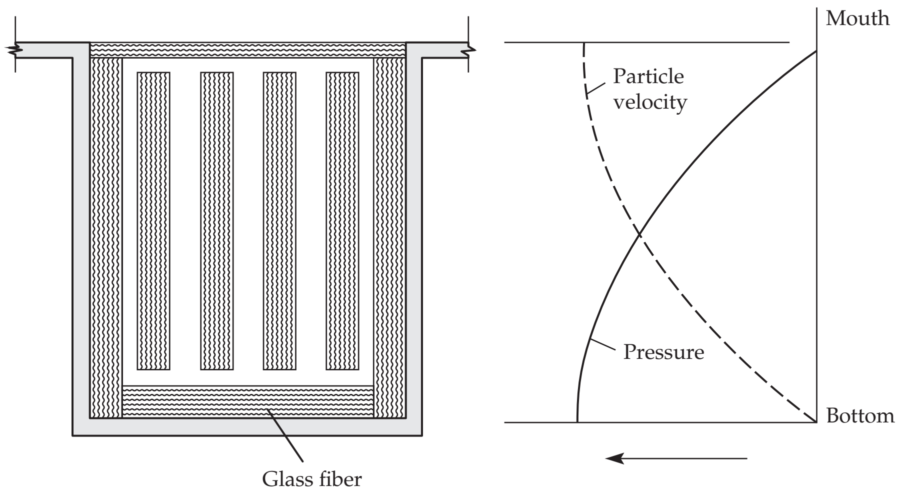


FIGURE 12-29 The operation of a bass trap depends on reflections of sound from the bottom of the trap. The pressure for the frequency at which the depth is a quarter wavelength is maximal at the bottom and the particle velocity is zero at the bottom. At the mouth, the pressure is zero (or very low) and the particle velocity is maximal. Absorbent placed where the particle velocity is maximal will absorb sound very effectively. The same action occurs at odd multiples of the quarter wavelength.

of a quarter wavelength (point of maximum particle velocity) at the design frequency at which maximum absorption is desired.

The concept of wall reflection and quarter-wavelength depths (graphically portrayed for drapes in Fig. 12-17) also applies to bass traps. The sound pressure at the bottom of the cavity is maximal at the quarter-wavelength design frequency. The air particle velocity is zero at the bottom. At the mouth, the pressure is zero and the particle velocity is maximal, which results in two phenomena. First, a glass-fiber semirigid board across the opening offers great friction to the rapidly vibrating air particles, resulting in maximum absorption at this frequency. Second, the zero pressure at the opening constitutes a vacuum that acts like a sound sink. The bass trap's effect, then, is greater than its opening area would suggest.

The bass-trap effect, like the drape spaced from a reflective wall, occurs not only at a quarter-wavelength depth but also at odd multiples of a quarter wavelength. Large trap depths are required for very low bass frequencies. For example, a quarter wavelength for 40 Hz is 7 ft. Unused spaces above control room ceilings and between inner walls and outer shells are often used for trap space. The famous Hidley bass trap design is an example of this type of trap.

Helmholtz (Volume) Resonators

The Helmholtz type of resonator is widely used to achieve absorption at lower audio frequencies. The operation of resonator absorbers can be easily demonstrated. Blowing across the mouth of any bottle produces a tone at its natural frequency of resonance. The air in the cavity acts like a spring, and the mass of the air in the neck of the bottle reacts with this springiness to form a resonating system, much as a weight on a spring vibrating at its natural period. Absorption is maximal at the frequency of resonance and decreases at nearby frequencies. The resonant frequency of a Helmholtz resonator is given by:

$$f_0 = \left(\frac{c}{2\pi}\right)\sqrt{\frac{S}{(V)(l_{\text{eff}})}} \tag{12-7}$$

where c = speed of sound in air, 1,130 ft/sec or 344 m/sec
S = cross-sectional area of the neck, ft^2 or m^2
V = volume of the cavity, ft^3 or m^3
l_{eff} = effective length of the neck
= physical length of the neck + correction term, ft or m (see below)

In this equation, the effective length of the neck (l_{eff}) is the physical length of the neck plus a correction term. This term is needed because the mass of the air in the neck effectively extends beyond the neck as that air interfaces with the air both inside and outside the resonator. For a Helmholtz resonator with a circular neck, the correction term of 1.6R is added to the physical length of the neck, where R is the radius of the neck. (Both 1.6R and 1.7R are cited in different sources.) The resonant frequency of a Helmholtz resonator with a circular neck opening is thus:

$$f_0 = \left(\frac{c}{2\pi}\right)\sqrt{\frac{S}{(V)(l+1.6R)}} \tag{12-8}$$

where c = speed of sound in air, 1,130 ft/sec or 344 m/sec
S = cross-sectional area of the neck, ft^2 or m^2
V = volume of the cavity, ft^3 or m^3
l = physical length of the neck, ft or m
R = radius of the neck, ft or m

For a Helmholtz resonator with a square neck opening, the correction term is $0.9L$, where L is the length of the side of the square opening, in ft or m. The denominator inside the square root is thus $(V)(l + 0.9L)$. In some Helmholtz resonators, there is no neck; the dimension of the wall thickness of the cavity itself serves as the neck length dimension. These equations are generally valid when the dimensions of the neck are small compared to the dimensions of the cavity, and the dimensions of both are small compared to the wavelength of the resonant frequency.

If the volume of the air cavity or the length or diameter of the neck changes, the frequency of resonance changes. The width of this absorption band depends on the friction of the system. A glass bottle offers little friction to the vibrating air and would have a very narrow absorption band. Adding a bit of gauze across the mouth of the bottle or stuffing a wisp of cotton into the neck, the amplitude of vibration is reduced and the width of the absorption band is increased. For maximum effectiveness, Helmholtz absorbers should be placed in areas of high modal sound pressure for the tuned frequency.

To demonstrate the effectiveness of a continuously swept narrow-band technique of measuring absorption coefficients, Riverbank Acoustical Laboratories measured the absorption of Coca-Cola bottles. A tight array of 1,152 empty 10-oz bottles was arranged in a standard 8- × 9-ft space on the concrete floor of a reverberation chamber. A single, well-isolated bottle had an absorption of 5.9 sabins at its resonance frequency of 185 Hz, but with a bandwidth (between −3dB points) of only 0.67 Hz. Absorption of 5.9 sabins is about what a person would absorb at 1 kHz, or what 5.9 ft^2 of glass fiber (2 in thick, 3 lb/ft^3 density) would absorb at midband. The sharpness of this absorption characteristic corresponds to a very high Q (quality factor) of 185/0.67 = 276.

The sound impinging on a Helmholtz resonator that is not absorbed is reradiated. As the sound is reradiated from the resonator opening, it tends to be radiated in a hemisphere. This means that unabsorbed energy is diffused, and diffusion of sound is very desirable in a studio or listening room.

In Helmholtz resonators, we have acoustical artifacts that far antedate Helmholtz himself. Bronze jars have been found in ancient Greek and Roman open-air theaters. Large jars may have been used to absorb sound at lower frequencies. Groupings of smaller jars may have supplied sound absorption at higher frequencies. In medieval times, resonators were used in a number of churches in Sweden and Denmark. Pots like those of Fig. 12-30 were embedded in the walls, presumably to reduce low-frequency modes. Ashes have been found in some of the pots, perhaps introduced to lower the Q of the ceramic pot and thus broaden the frequency of its effectiveness.

Helmholtz resonators are often used in the form of acoustical blocks. These blocks are formed of concrete with an open slot facing the closed cavity. A two-cell unit would have two slots. In some cases, a metal divider is placed inside each cavity, or a porous absorber is placed in the slot inside the cavity. Figure 12-31 shows idealized square boxes with tubular necks forming a perforated face resonator. Stacking these boxes enhances the resonator action. Consider a box of length L, width W, and depth H with a lid of thickness equal to the length of necks of the boxes. In this lid, holes are drilled

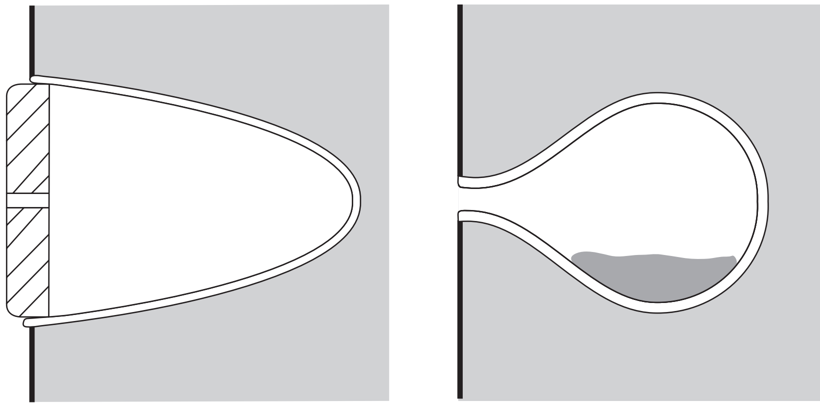


Figure 12-30 Pots embedded in the walls of medieval churches in Sweden and Denmark served as Helmholtz resonators, absorbing sound. Ashes, found in some of the pots, may have served as a dissipative agent. (*Brüel.*)

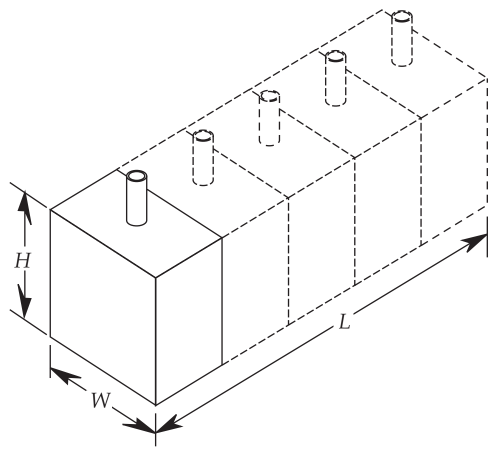


Figure 12-31 Development of a perforated-face Helmholtz resonator from a single rectangular-box resonator.

having the same diameter as the holes in the neck. The partitions between each segment can be removed without greatly affecting the Helmholtz action.

Figure 12-32 illustrates a square box with an elongated slit neck. These can also be stacked in multiple rows. This design is similar to a slot-type resonator. The separating walls in the air cavity can also be eliminated without greatly degrading the resonator action. A word of caution is in order, however. Subdividing the airspace can improve the action of perforated face or slit resonators because this reduces spurious, unwanted modes of vibration being set up within the air cavity.

Various types of acoustical concrete masonry units (ACMU) are available; many are based on designs pioneered by The Proudfoot Company, now owned by Sound Seal. For example, SoundBlox and SoundCell concrete masonry units supply load-bearing

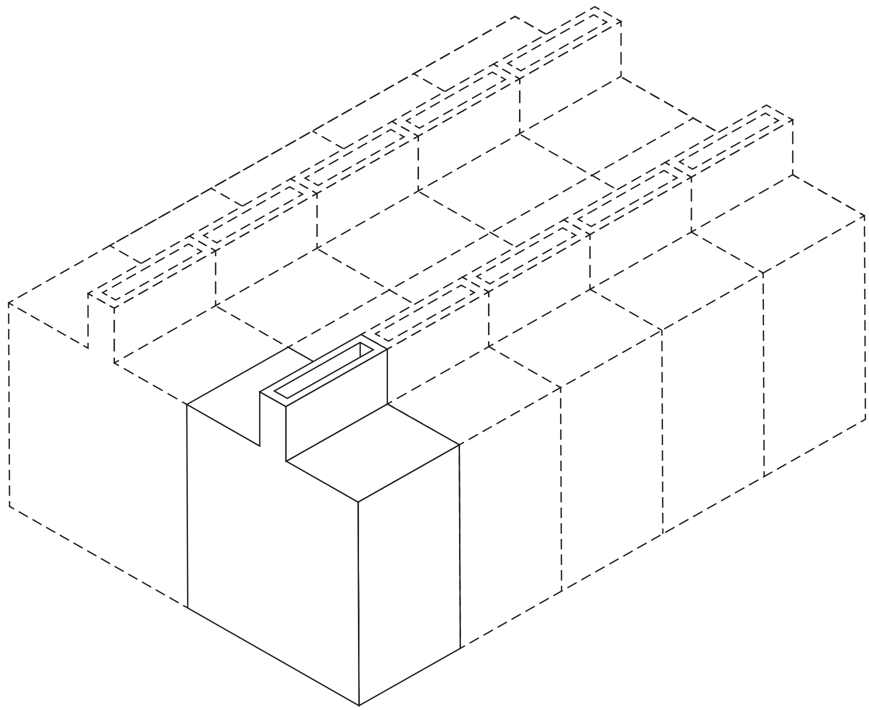


Figure 12-32 Design of a slot-type Helmholtz resonator from a single rectangular-box resonator.

ability and the mass required for sound isolation, and also enhanced low-frequency absorption through Helmholtz resonators formed by slots and cavities in the blocks. The masonry units are available in numerous configurations. SoundBlox and Sound-Cell units are available through Sound Seal and regional manufacturers.

Perforated Panel Absorbers

Perforated panels, for example, using hardboard, plywood, aluminum, or steel, spaced from the wall constitute a resonant type of sound absorber. Each hole acts as the neck of a Helmholtz resonator, and the share of the cavity behind belonging to that hole is comparable to the cavity of a Helmholtz resonator. In fact, we can view this structure as a host of coupled resonators. If sound arrives perpendicular to the face of the perforated panel, all the tiny resonators are in phase. For sound waves striking the perforated board at an angle, the absorption efficiency is somewhat decreased. This loss can be minimized by sectionalizing the cavity behind the perforated face with an egg crate type of divider of wood or corrugated paper.

The frequency of resonance of perforated panel absorbers with circular holes backed by a subdivided airspace is given approximately by:

$$f_0 = 200\sqrt{\frac{p}{(d)(t)}} \tag{12-9}$$

where f_0 = frequency of resonance, Hz
p = perforation percentage
= hole area divided by panel area × 100 (see Fig. 12-33A and B)
t = effective hole length, in, with correction factor applied
= (panel thickness) + (0.8)(hole diameter), in
d = depth of airspace, in

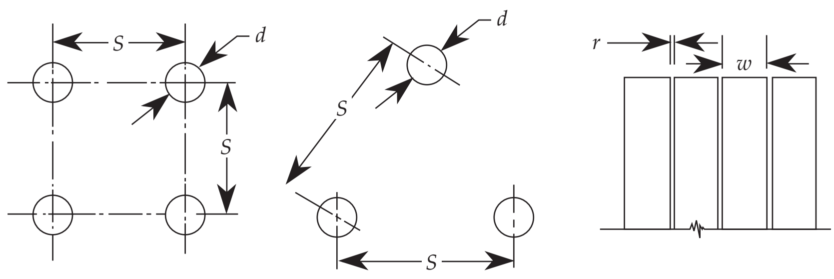
There is some confusion in the literature concerning p, the perforation percentage. Some writers use the decimal ratio of hole area to panel area, rather than the percentage of hole area to panel area, introducing an uncertainty factor of 100. This perforation percentage is easily calculated by reference to Fig. 12-33. The perforation percentage for two types of circular-hole configurations is shown in Fig. 12-33A and B, and the percentage for slat absorbers (described later) is shown in Fig. 12-33C.

The frequency of resonance for perforated panels with circular holes presented in Eq. (12-9) is presented in graphical form in Fig. 12-34 for a panel thickness of 3/16 in. Common pegboard with holes with diameter 3/16 in, spaced on 1-in centers with the square configuration of Fig. 12-33 has 2.75% of the area in holes. If this pegboard is spaced out from the wall by 2 × 4 studs on edge, the system resonates at about 420 Hz and the peak absorption appears near this frequency.

In commonly available perforated materials, such as pegboard, the holes are so numerous that with practical airspaces, only resonances at higher frequencies can be obtained. To obtain low-frequency absorption, the holes can be drilled by hand. Drilling 7/32-in holes on 6-in centers gives a perforation percentage of about 1.0%. Of course, with zero perforation percentage, the panel behaves as a solid panel absorber.

Figure 12-35 shows the effect of varying hole area from 0.18% to 8.7% in a structure of otherwise fixed dimensions. The plywood is 5/32-in thick perforated with 3/16-in holes, except for the 8.7% case in which the hole diameter is about 3/4 in. The perforated plywood sheet is spaced 4 in from the wall and the cavity is half-filled with glass fiber; the other half is airspace.

Figure 12-36 is identical to Fig. 12-35, except that the perforated plywood is spaced 8 in and glass fiber of 4-in thickness is mounted in the cavity. The effect of these changes is a substantial broadening of the absorption curve.



Perforation % = $78.5\left(\frac{d}{S}\right)^2$ Perforation % = $90.6\left(\frac{d}{S}\right)^2$ Perforation % = $100\left(\frac{r}{w+r}\right)$

A B C

FIGURE 12-33 Formulas for calculating perforation percentage for perforated panel resonators, including slat absorbers. (A and B) The perforation percentage for two types of circular-hole configurations. (C) The perforation percentage for slat absorbers.

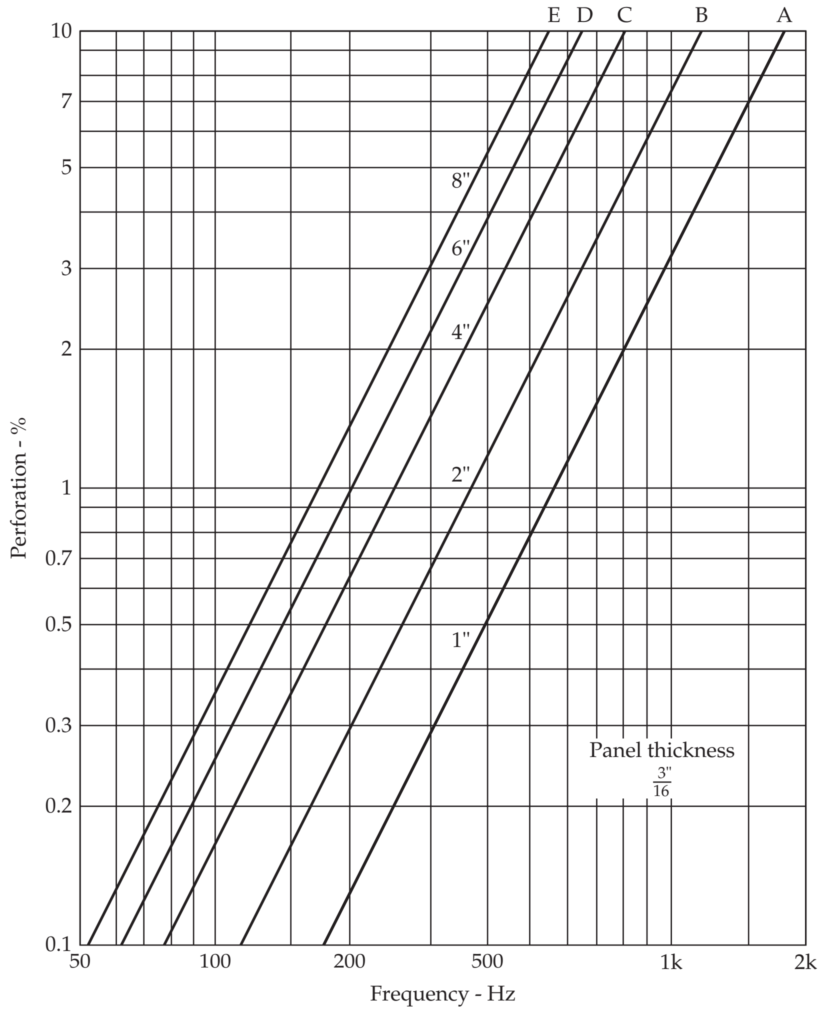


Figure 12-34 A graphic presentation of Eq. (12-9) relating percent perforation of perforated panels, the depth of airspace, and the frequency of resonance. The graphs are for a panel of 3/16-in thickness. (See also Fig. 12-22.) The lines are drawn to correspond to furring lumber which is furnished; the line for 8 in is actually a 7-3/4-in airspace. (A) 1-in furring lumber. (B) 2-in furring lumber. (C) 4-in furring lumber. (D) 6-in furring lumber. (E) 8-in furring lumber.

It would be unusual to employ such perforated panel absorbers without acoustical resistance in the cavity in the form of glass-fiber batts or boards. Without such resistance, the absorption bandwidth is very sharp. However, one possible use of such sharply tuned absorbers would be to control specific troublesome room modes or isolated groups of modes with otherwise minimum effect on the signal and overall room acoustics. When a perforated panel is placed over a porous absorber, the panel adds low-frequency absorption compared to the porous material alone. However, the panel may also reduce the high-frequency absorption of the porous material.

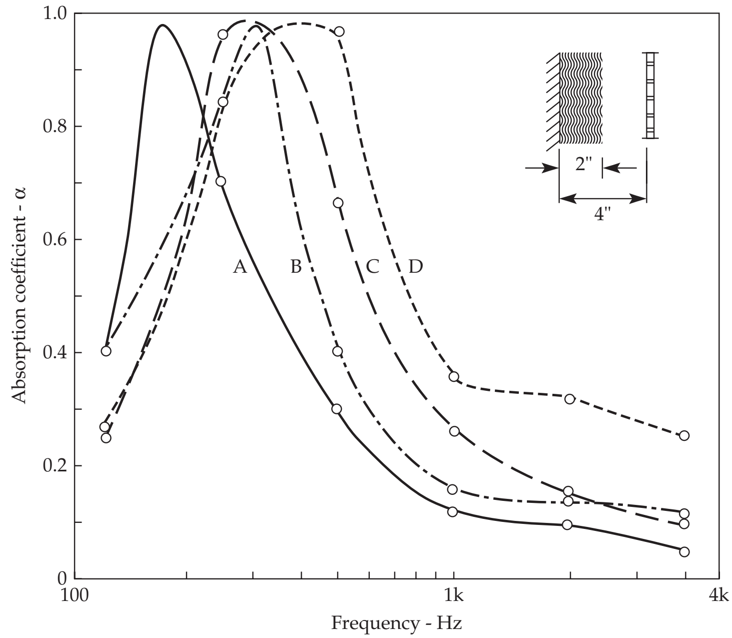


Figure 12-35 Absorption measurements of perforated panel absorbers with 4-in airspace, half-filled with mineral wool and for panel thickness of 5/32 in. (A) 0.18% perforation. (B) 0.79% perforation. (C) 1.4% perforation. (D) 8.7% perforation. The presence of the mineral wool shifts the frequency of resonance considerably from the theoretical values of Eq. (12-9) and Fig. 12-34. (*Data from Mankovsky.*)

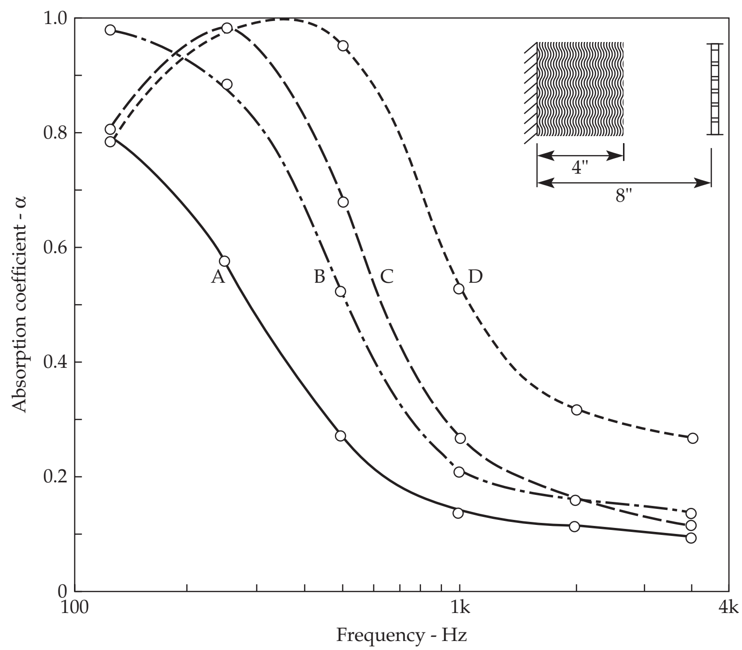


Figure 12-36 Absorption measurements on the same perforated panel absorbers of Fig. 12-35, but the airspace is increased to 8 in, half of which is taken up with mineral fiber. Panel thickness is 5/32 in. (A) 0.18% perforation. (B) 0.79% perforation. (C) 1.4% perforation. (D) 8.7% perforation. (*Data from Mankovsky.*)

Table 12-3 includes the calculated frequency of resonance of 48 different combinations of airspace depth, hole diameter, panel thickness, and hole spacing. This listing should assist in approximating the desired condition.

Slat Absorbers

Another type of resonant absorber, called a slat absorber, utilizes closely spaced slats over an air cavity. The mass of the air in the slots between the slats reacts with the springiness of air in the cavity to form a resonant system, comparable to a Helmholtz resonator. A glass-fiber board usually introduced behind the slots acts as a resistance, broadening the peak of absorption. The narrower the slots and the deeper the cavity, the lower the frequency of maximum absorption. The percent perforation of a slat absorber is shown in Fig. 12-33C.

The resonance frequency of the slat absorber can be estimated:

$$f_0 = 216\sqrt{\frac{p}{(d)(D)}} \tag{12-10}$$

where f_0 = frequency of resonance, Hz
p = perforation percentage (see Fig. 12-33C)
D = depth of airspace, in
d = thickness of slat, in

Placement of Materials

Placing sound-absorbing materials in distributed patches, as opposed to a continuous area, effectively contributes to greater absorption. (Such placement of absorbers can also improve diffusion.) If several types of absorbers are used, it is desirable to place some of each type on ends, sides, and ceiling so that all three axial modes (longitudinal, transverse, and vertical) will come under their influence. In rectangular rooms, it is most effective to place absorbing material near corners and along edges of room surfaces. In speech studios, some absorbent that is effective at higher audio frequencies should be applied at head height on the walls. In fact, material applied to the lower portions of high walls can be as much as twice as effective as the same material placed elsewhere. Untreated surfaces should never face each other; otherwise, flutter echoes may occur.

Reverberation Time of Helmholtz Resonators

There exists the possibility that acoustically resonant devices, such as Helmholtz absorbers, can “ring” with a “reverberation time” of their very own, thus causing timbral changes to audio signals. It is true that any resonant system, electronic or acoustical, has a certain time constant associated with it. The Q-factor (quality factor) describes the sharpness of tuning, as shown in Fig. 12-37. Once the tuning curve has been obtained experimentally, the width of the tuning curve at the −3 dB points gives Δf. The Q of the system is then $Q = f_0/\Delta f$, where f_0 is the frequency to which the system is tuned. In one study, measurements on a number of perforated and slat Helmholtz absorbers gave Q factors around 1 or 2 but some as high as 5. Table 12-4 shows how the decay rate of resonant absorbers with different Q factors relates to reverberation time (f_0 = 100 Hz).

Depth of Airspace	Hole Diameter	Panel Thickness	Percent Perforation	Hole Spacing	Frequency of Resonance (Hz)
3-5/8	1/8	1/8	0.25	2.22	110
			0.50	1.57	157
			0.75	1.28	192
			1.00	1.11	221
			1.25	0.991	248
			1.50	0.905	271
			2.00	0.783	313
			3.00	0.640	384
3-5/8	1/8	1/4	0.25	2.22	89
			0.50	1.57	126
			0.75	1.28	154
			1.00	1.11	178
			1.25	0.991	199
			1.50	0.905	217
			2.00	0.783	251
			3.00	0.640	308
3-5/8	1/4	1/4	0.25	4.43	89
			0.50	3.13	126
			0.75	2.56	154
			1.00	2.22	178
			1.25	1.98	199
			1.50	1.81	217
			2.00	1.57	251
			3.00	1.28	308
5-5/8	1/8	1/8	0.25	2.22	89
			0.50	1.57	126
			0.75	1.28	154
			1.00	1.11	178
			1.25	0.991	199
			1.50	0.905	218
			2.00	0.783	251
			3.00	0.640	308
5-5/8	1/8	1/4	0.25	2.22	74
			0.50	1.57	105
			0.75	1.28	128
			1.00	1.11	148

Table 12-3 Helmholtz Low-Frequency Absorber Perforated-Face Type (All Dimensions in Inches)

Depth of Airspace	Hole Diameter	Panel Thickness	Percent Perforation	Hole Spacing	Frequency of Resonance (Hz)
			1.25	0.991	165
			1.50	0.905	181
			2.00	0.783	209
			3.00	0.640	256
5-5/8	1/4	1/4	0.25	4.43	63
			0.50	3.13	89
			0.75	2.56	109
			1.00	2.22	126
			1.25	1.98	141
			1.50	1.81	154
			2.00	1.57	178
			3.00	1.28	218

Table 12-3 Helmholtz Low-Frequency Absorber Perforated-Face Type (All Dimensions in Inches) (*Continued*)

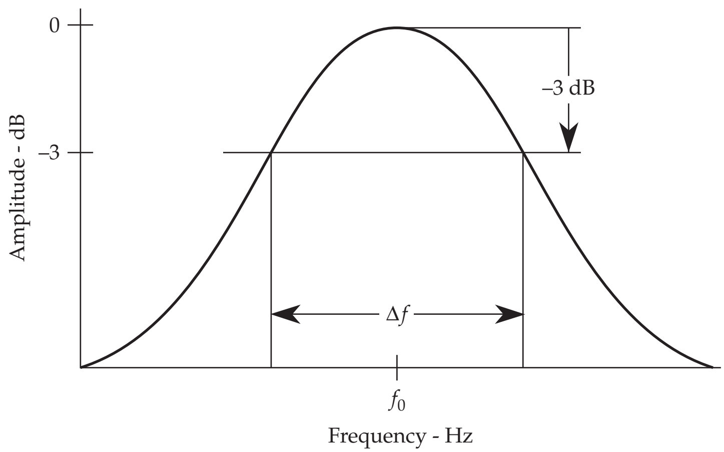


Figure 12-37 After the tuning curve of a Helmholtz type resonant absorber has been determined, its Q factor can be found from the expression, $f_0/\Delta f$. The "reverberation time" of such absorbers is very short for Q factors normally encountered.

Q	Reverberation Time (sec)
100	2.2
5	0.11
1	0.022

Table 12-4 Sound Decay in Resonant Absorbers, f_0 = 100 Hz

With resonant absorber Q factors of 100, problems could be encountered in a room having a reverberation time of perhaps 0.5 sec as the absorbers regenerate sound for several seconds. However, Helmholtz absorbers with such Q factors would be very special devices, perhaps made of ceramic. Absorbers made of wood with glass fiber to broaden the absorption curve have low Q factors so their sound dies away much faster than the studio or listening room itself.

Reducing Room Modes with Absorbers

Because of the relatively narrow absorption bandwidth possible with resonator absorbers, they are ideal for reducing room modes. For example, using time-energy-frequency analysis the low-frequency modal structure of a room can be revealed, as shown in Figs. 12-38 and 12-39. In this example, a mode that could cause audible distortion produces a reverberant tail at 47 Hz, as seen on the extreme left of Fig. 12-38. Steps can be taken to reduce its energy so that it will behave as the other modes of the room.

The solution is a highly tuned absorber placed in the room at a point of high pressure of the 47-Hz mode. High-pressure points of the 47-Hz mode are located by energizing the room with a 47-Hz sine wave from a loudspeaker and measuring the response with a sound-level meter. A location that is both convenient and effective will probably be found in a corner. Figure 12-39 shows the low-frequency modal structure of the same room after the introduction of a tuned Helmholtz resonator absorber.

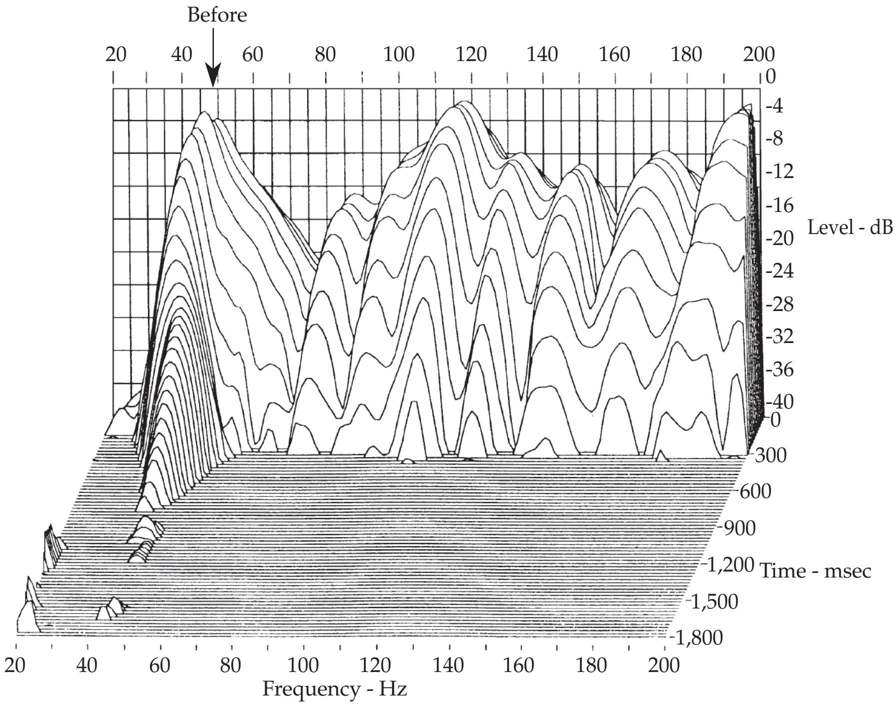


Figure 12-38 Low-frequency modal structure of the sound field of a small room before the introduction of the tuned Helmholtz resonator absorber.

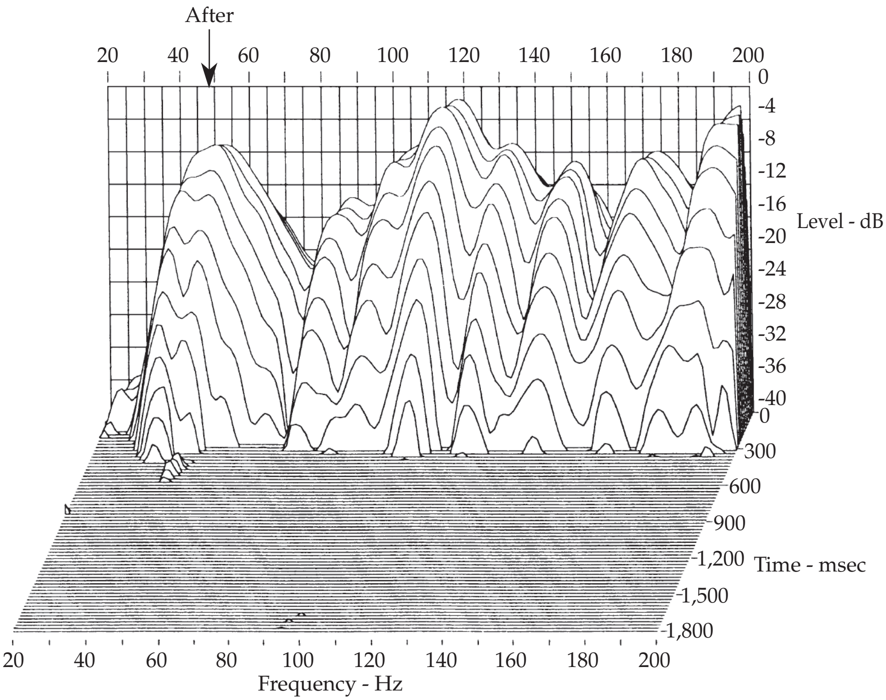


FIGURE 12-39 Low-frequency modal structure of the sound field of the same small room after the introduction of the tuned Helmholtz resonator absorber.

The resonator, shown in Fig. 12-40, can be inexpensively constructed from a concrete-forming tube available in building-supply stores. Laminated wood covers are tightly fitted into both ends of the tube. The length of a PVC pipe forming the duct opening can be varied to tune the resonator to a specific frequency. An absorbent such as glass-fiber batt should be used to partially fill the resonator.

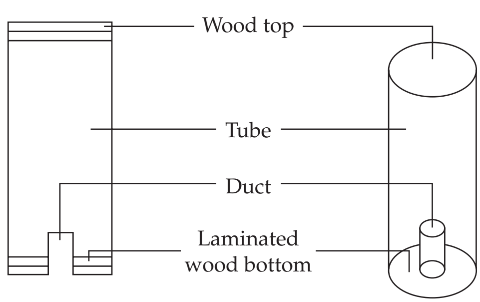


FIGURE 12-40 Details of a tuned Helmholtz resonator design.

Increasing Reverberation Time

Low-Q Helmholtz resonators are capable of shortening reverberation times by increasing absorption. High-Q resonators can increase reverberation time through the storage of energy as described by Gilford. To achieve the high Q factors necessary, plywood, particleboard, Masonite, and other such materials must be abandoned and materials such as ceramics, plaster, and concrete used in resonator construction. By properly tuning the resonators, the increase in reverberation time can be placed where needed in regard to frequency.

Absorption Module Design

The British Broadcasting Corporation (BBC) developed a series of modules for the acoustical treatment of their small voice studios. These kinds of modules have been applied in hundreds of studios with satisfactory acoustical results. In this design, the walls are covered with standard-sized modules, say 2 × 3 ft, having a maximum depth of perhaps 8 in. These can be framed on the walls to give a flush surface appearing very much like an ordinary room, or they can be made into boxes with grille cloth covers mounted on the walls in regular patterns. All modules can be made to appear identical, but the similarity is only skin deep.

There are commonly three, or perhaps four different types of modules, each making its own distinctive acoustical contribution. Figure 12-41 shows the radically different absorption characteristics obtained by merely changing the covers of the standard module.

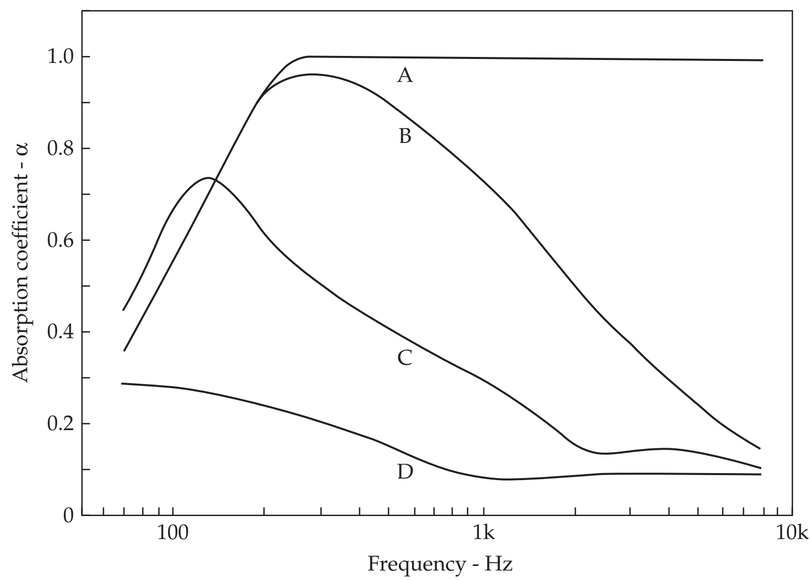


Figure 12-41 Modular absorber with a 7-in airspace and a 1-in semirigid glass-fiber board of 9 to 10 lb/ft^3 density behind the perforated cover. (A) No perforated cover, or one with at least 25% perforation or more. (B) 5% perforated cover. (C) 0.5% perforated cover. (D) 3/4-in plywood cover, essentially neutralizing the module. (*Data from Brown.*)

This plot is for a module measuring 2 × 3 ft having a 7-in airspace and a 1-in semirigid glass-fiber board of 3 lb/ft^3 density inside. The wideband absorber has a highly perforated cover (25% or more perforation percentage) or no cover at all, yielding essentially complete absorption down to about 200 Hz. Even better low-frequency absorption can be achieved by breaking up the airspace with egg-crate-type dividers of corrugated paper to discourage unwanted resonance modes. A cover 1/4 in thick with a 5% perforation percentage will peak in the 300- to 400-Hz range. A true bass absorber is obtained with a low-perforation cover (0.5% perforation). If essentially neutral modules are desired, they can be covered with plywood with a thickness of 3/8 or 1/4 in, which would give relatively low absorption with a peak around 70 Hz. Using these three or four modules as acoustical building blocks, the desired absorption characteristic can be designed into a studio by specifying the number and distribution of each of the basic modules.

Figure 12-42 shows an adaptation of the BBC design where the wall is used as the bottom of the module box. In this case, the module size is 2 × 4 ft. The modules are fastened to 2- × 2-in mounting strips that in turn are fastened to the wall. A studio wall 10 ft high and 23 or 24 ft long might use 20 modules of distributed types, four modules

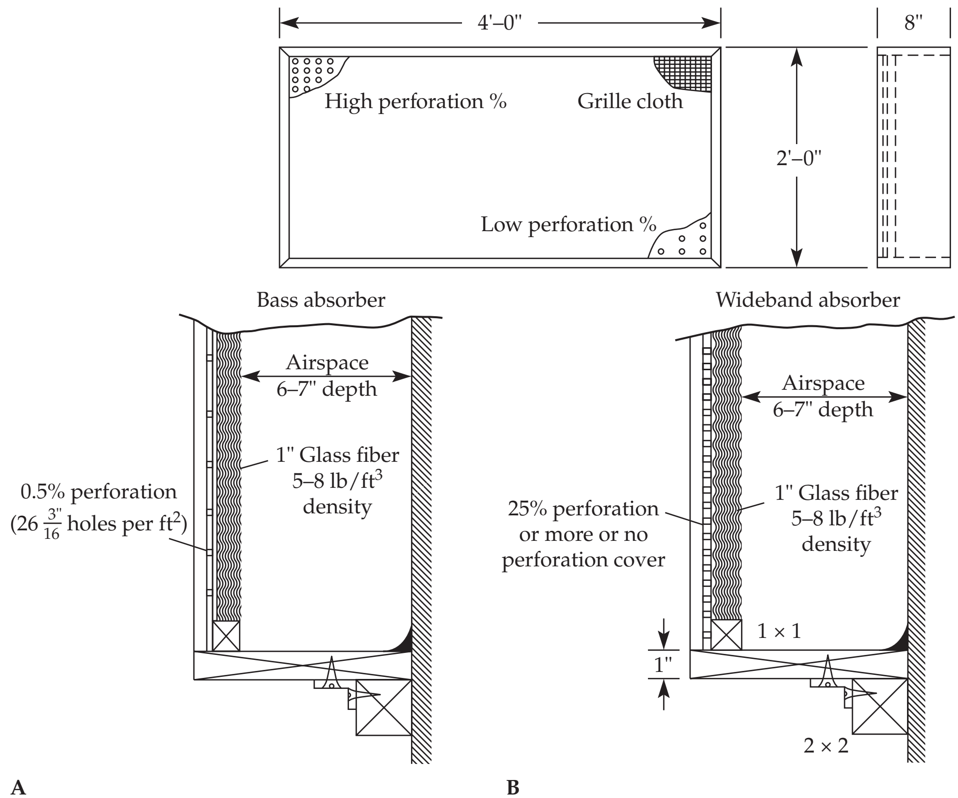


Figure 12-42 Plan for a practical module absorber utilizing the wall as the bottom of the module. (A) Bass absorber. (B) Wideband absorber.

high and five long. It is good practice to have acoustically dissimilar modules opposing each other on opposite walls. BBC experience has shown that careful distribution of the different types of modules results in adequate diffusion.

Key Points

- There are three general types of absorbers: porous, panel, and volume or resonance absorbers.
- The absorption coefficient α is a measure of the efficiency of a surface or material in absorbing sound. A perfect sound absorber will absorb 100% of incident sound; thus α is 1.0. A perfectly reflecting surface would have α of 0.0.
- Sound absorption A provided by a particular area of material is obtained by multiplying its absorption coefficient by the surface area of the material exposed to sound.
- The noise reduction coefficient (NRC) is the average of the absorption coefficients for 250, 500, 1,000, and 2,000 Hz.
- Materials that are porous, with internal interstices, can absorb sound at mid and high frequencies; sound energy is reduced because of frictional resistance in the material.
- The mounting method has a major effect on the absorption characteristics of a material; for example, the absorption of porous materials is much greater with an airspace between the material and the mounting surface.
- Glass fiber is an excellent and inexpensive absorbing material. Both glass-fiber batts and panels are widely used in a variety of acoustical applications.
- Sound absorption is improved in thicker porous materials, primarily at lower frequencies. Absorption is greatest when the porous material is placed at a distance of a quarter wavelength ($\lambda/4$) from a hard reflective surface.
- Drapes are a type of porous sound absorber; variables affecting absorbency include type and weight of material, degree of fold, and spacing from the wall.
- Carpet and its underlay can provide significant absorption at mid and high frequencies; to prevent an unbalanced room response, the absorption of carpet can be compensated for with low-frequency absorbers.
- The people comprising a concert-hall audience account for a significant portion of the sound absorption of the room. One rule of thumb attributes 5 sabins at 500 Hz, per seated person.
- For large rooms, the absorption of sound by the air is significant for frequencies of about 2,000 Hz and above.
- The absorption of sound at lower audible frequencies can be effectively achieved by resonant (or reactive) absorbers.
- In a panel absorber, absorption is increased when the cavity is filled with a porous material such as glass-fiber insulation.
- Polycylindrical absorbers comprising a panel wrapped around semicylindrical bulkheads can provide good low-frequency absorption as well as diffusion.

- A bass trap is a type of low-frequency sound absorber using a large reactive cavity to absorb sound at the lowest octaves.
- A Helmholtz resonator uses a volume or cavity to achieve absorption usually at lower audio frequencies. For maximum effectiveness, they should be placed in areas of high modal sound pressure for the tuned frequency.
- A perforated panel absorber is a type of resonant absorber. Each hole acts as the neck of a Helmholtz resonator, and the portion of cavity behind it acts as the cavity of a Helmholtz resonator.
- A slat absorber utilizes closely spaced slats over an air cavity. The mass of the air in the slots reacts with the springiness of air in the cavity to form a resonant system comparable to a Helmholtz resonator.
- Because of the relatively narrow absorption bandwidth possible with resonator absorbers, they are ideal for reducing particular room modes.

CHAPTER 13
Modal Resonances

Sound in enclosed spaces behaves fundamentally differently from sound in a free field. In a free field, sound leaves the source and travels outward unimpeded. It is easy, for example, to determine the sound-pressure level at any distance from the source. In most enclosed spaces, sound from the source is reflected from boundary surfaces such as walls, floor, and ceiling. The resulting sound-pressure level at any point in the enclosure is a combination of the direct sound and the reflected sound. In particular, modal resonances are established; the sound-pressure level will be different at different places in the enclosed space and will also vary according to frequency. These resonant frequencies and the variations they create are a function of the dimensions of the enclosed space.

Most rooms can be considered as enclosed spaces; they are essentially containers of air. As such, they exhibit modal resonances, preferred frequencies where energy is concentrated. Likewise, these resonances create frequencies where energy is decreased. This complex energy distribution exists in three-dimensional space throughout the room. Many musical instruments use pipes or tubes in their construction; these also operate as enclosed spaces and exhibit modal resonance. Changing the parameters of the modal resonances changes the musical pitch.

We live most of the hours of our lives in enclosed spaces; thus our acoustical lives are governed by the effects of these large resonators. In any case, room resonances affect the sounds we hear whenever we are indoors.

Early Experiments and Examples

Hermann von Helmholtz performed early acoustical experiments with resonators. His resonators were a series of metal spheres of graded sizes, each fitted with a neck, appearing somewhat like the round-bottom flask found in a chemistry laboratory. In addition to the neck there was another small opening to which he applied his ear. The resonators of different sizes resonated at different frequencies, and by pointing the resonator necks toward the sound under investigation he could use the loudness of the sound of the different resonators to estimate the energy at each frequency.

There were numerous applications of this principle long before the time of Helmholtz. A thousand years ago, resonators were embedded in church walls in Sweden and Denmark with the mouths flush with the wall surface, apparently for sound absorption. The interior walls of some modern buildings use concrete blocks with slits, leading to enclosed resonant cavities inside. Energy absorbed from sound in the room causes the resonators to vibrate at a characteristic frequency. Part of the energy is absorbed, part reradiated. The energy reradiated is sent in every direction, contributing to the diffusion of sound in the room. The resonator principle, old as it is, continually appears in new applications.

Resonance in a Pipe

The two ends of the closed pipe of Fig. 13-1A can be likened to two opposing walls of a room. The pipe gives us a simple, one-dimensional example; what happens between opposite walls of a rectangular room can be examined without being bothered by the reflections from the other four surfaces. The pipe is a resonator capable of vibrating at its characteristic frequencies when excited in some way. We will consider its behavior

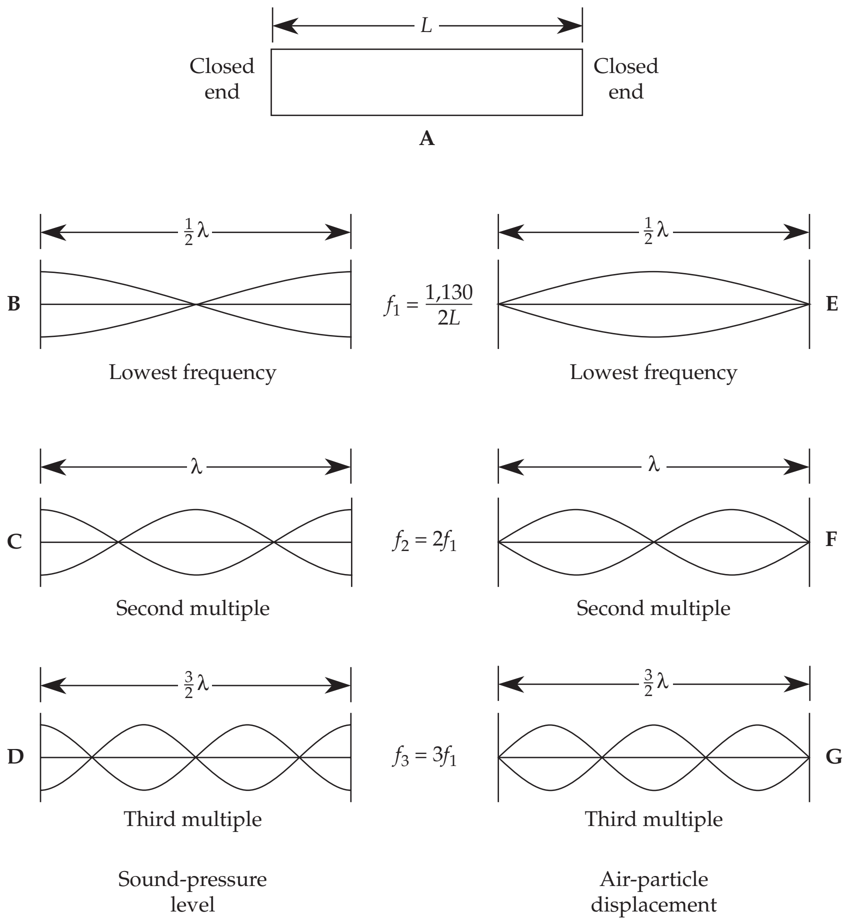


FIGURE 13-1 A pipe closed at both ends demonstrates how resonance occurs between two opposing walls of a room. The length of the pipe (distance between the walls in a room) determines the characteristic frequency of resonance and its harmonics. (A) Closed pipe. (B) Plot of sound-pressure level of f_1. (C) Plot of sound-pressure level of f_2. (D) Plot of sound-pressure level of f_3. (E) Plot of air-particle displacement of f_1. (F) Plot of air-particle displacement of f_2. (G) Plot of air-particle displacement of f_3.

with sound with wavelength much larger than the diameter of the pipe, so that sound travels the length of the pipe. When a stream of air is blown across a lip at the edge of an organ pipe, the air column inside the pipe will begin to vibrate. More precisely, for our experiment, a small loudspeaker can be placed inside a pipe closed at both ends. A sine wave signal is fed to the loudspeaker and varied in frequency. A small listening hole drilled in the pipe at the end that is opposite the loudspeaker makes it possible to hear the tones radiated by the loudspeaker. As the frequency is increased, nothing unusual is noted until the frequency radiated from the loudspeaker coincides with the natural frequency of the pipe. At this frequency f_1, modest energy from the loudspeaker is strongly reinforced, and a relatively loud sound is heard at the listening hole. We note that the wavelength of this frequency is twice the length of the pipe. As the frequency is increased, the loudness is again low until a frequency of $2f_1$ is reached, at which point another strong reinforcement is noted. Such resonant peaks can also be detected at $3f_1$, $4f_1$ . . . , and at all multiples of f_1.

Now let us assume that we can measure and record the sound pressure along the length of the pipe. Figures 13-1B, C, and D show how the sound pressure varies along the pipe for three different excitation frequencies. A sound wave traveling to the right is reflected from the right end, returning as a sound wave out of polarity with itself (delayed by a half period), traveling to the left. The left-going waves react with the right-going waves to create, by superposition, longitudinal standing waves at the natural modal frequency of the pipe as well as multiples of that frequency. These standing waves have areas of cancellation (nodes) and reinforcement (antinodes) between the reflecting surfaces. We note that the resonant frequencies f_n of a closed pipe are determined by the length of the pipe. In particular,

$$f_n = \frac{nc}{2L} \tag{13-1}$$

where n = an integer ≥ 1
L = length of the pipe, ft or m
c = speed of sound = 1,130 ft/sec or 344 m/sec

For example, if the closed pipe is 5-ft long, the fundamental resonant frequency f_1 is 1,130/10 = 113 Hz. The second resonant frequency f_2 is (2)(1,130)/10 = 226 Hz, and so on.

When sounding the f_1 frequency, as shown in Fig. 13-1B, measuring probes inserted through tiny holes along the pipe would detect zero pressure at the center of the pipe, high pressure near the closed ends of the pipe, and so on. Similar nodes (minima) and antinodes (maxima) can be observed at $2f_1$ and $3f_1$, as shown in Figs. 13-1C and D. Likewise, other resonant frequencies would exist at higher-order multiples of f_1.

In the same way, the dimensions of a room determine its characteristic frequencies much as though there were a north-south pipe, an east-west pipe, and a vertical pipe, the pipes corresponding to the length, width, and height of the room, respectively. In other words, rooms behave the same as closed pipes, but in three dimensions. In addition, rooms can exhibit other kinds of modes that reflect from more than two surfaces.

We also note that the air-particle displacement is exactly defined in any standing wave, as shown in Figs. 13-1E, F, and G. In particular, at any point, the particle displacement is exactly out of phase with the sound-pressure level; there is a displacement node at every pressure antinode and a displacement antinode at every pressure node. For example, particle displacement is always zero at both ends of a closed pipe. For the lowest mode (see Fig. 13-1E) particle displacement is maximum at the center point; this center position from the end of the pipe corresponds to $\lambda/4$ for this mode. Similarly, for any mode, although the physical distance differs, the first position of maximum particle displacement (and minimum particle velocity) corresponds to $\lambda/4$. This explains why in a closed pipe, and in a room, porous absorbers are most effective when placed at a distance of $\lambda/4$ from a boundary.

Any room with reflective walls will illustrate the effect of resonance and the resulting reinforcement of sound at certain frequencies related to the dimensions of the room. Exciting the air at frequencies far removed from these characteristic frequencies results in weaker sounds.

Imagine yourself inside the pipe shown in Fig. 13-1D. At that resonant frequency, you would hear the sound pressure rise and fall as you traversed its length. A person in a room is, in a sense, inside a large pipe closed at both ends. However, there is one important difference; it is now a three-dimensional rather than an essentially one-dimensional system like the pipe. When the walls of a room are reflective, there is a characteristic modal frequency of resonance associated with the length, another with the width, and still another with the height of the room. In the case of a cubical room, all three modal frequencies coincide to give a mighty reinforcement at the basic characteristic modal frequency and multiples of it. This results in a very uneven response in the room, and for this reason, cubical rooms, or rooms with any equal dimensions, should be avoided.

Indoor Reflections

Anyone can appreciate the difference between sound conditions indoors and outdoors. Outdoors in an open space, the only reflecting plane may be the ground. If that happens to be covered with a foot of snow, which is an excellent absorber of sound, it may be difficult to carry on a conversation with someone 20 ft away. Indoors, sound energy is contained, resulting in a louder sound for a given effort. In an auditorium, a speaker can be heard and understood by hundreds of people with no reinforcement but that of reflecting surfaces.

Consider sound reflections from a single wall. In Fig. 13-2, a point source of sound, *S*, is a given distance from a rigid wall. The spherical wave fronts (solid lines, traveling to the right) are reflected from this surface (broken lines). The reflections from the surface traveling to the left act exactly as though they were radiating from another identical point source, an image source *I*, an equal distance from the reflecting surface but on the opposite side. This is the simplest case of one source, one image, and a reflecting surface, all in free space.

Now let's consider the isolated reflecting surface of Fig. 13-2 as the east wall of a rectangular room, as shown in Fig. 13-3. Source *S* still has its image in the east wall of the room, now labeled I_E. The source also has other images. I_N is the image of *S* in the north wall reflecting surface, I_W is the image of *S* in the west wall, and I_S is the image of *S* in the south wall. Use your imagination to visualize I_F, the image in the floor, and I_C, the image in the ceiling. All of these six images, like *S*, are sending sound energy back into the room. Because of absorption at the reflecting surfaces, their contributions will be somewhat weaker at a given point *P* in the room, but they all make their contribution.

There are also images from sound that reflects from more than one surface. For example, the I_{NS} image is a result of sound that leaves *S*, reflects from the north wall,

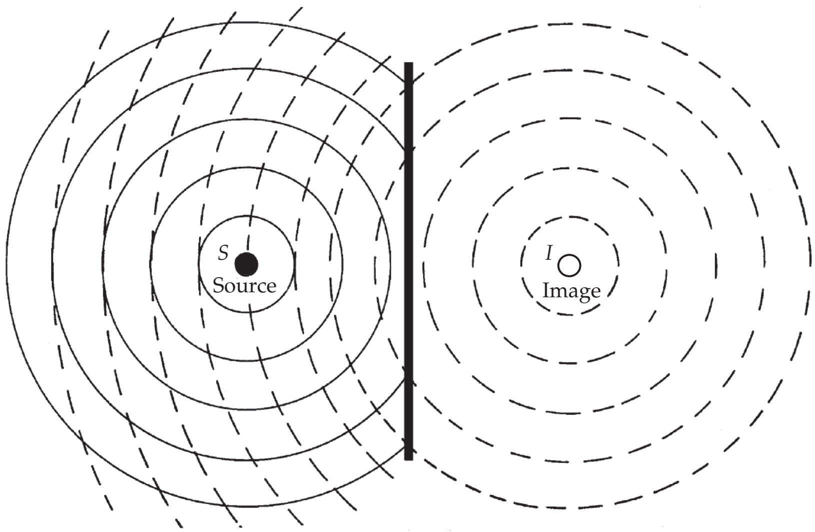


Figure 13-2 Sound radiated by a point *S* is reflected by the rigid wall. The reflected wave can be considered as coming from *I*, an image of *S*.

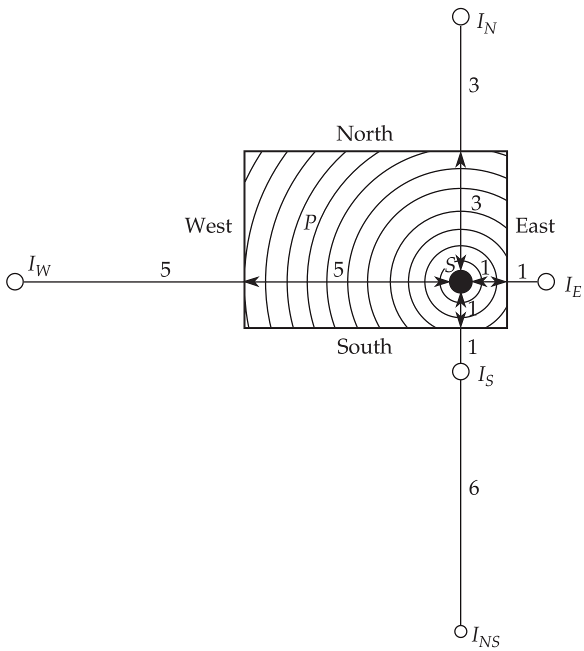


Figure 13-3 In an enclosure, the source *S* has six primary images, one in each of the six surfaces of the enclosure. An infinite number of other images are generated by multiple reflections, and reflections from multiple surfaces. Sound intensity at point *P* is made up of the direct sound from *S* plus the contributions from all images.

then reflects from the south wall. Similarly, the other surfaces produce the same multiple reflections. And, the process continues, for example, reflecting from north to south to north again, ad infinitum. Finally, other sounds may reflect at many other angles, for example, bouncing from each of the north, west, south, and east walls.

After many surface reflections, images are so weak that they can be neglected for the sake of simplicity. In any case, the sound field at a point *P* in a room comprises the direct sound from the source *S* plus the vector sum of the contributions of all the images of *S*. That is, the sound at *P* comprises the direct sound from *S* plus single or multiple reflections from all six surfaces.

Two-Wall Resonance

Figure 13-4 shows two parallel, reflective walls of infinite extent. When a loudspeaker radiating noise excites the space between the walls, this wall-air-wall system exhibits a resonance at a frequency of $f_1 = 1{,}130/2L$ or $565/L$, where L = the distance in feet between the two walls and 1,130 the speed of sound in ft/sec. The fundamental frequency f_1 is considered a natural frequency of the space between the reflective walls, and it is accompanied by a series of modes each of which also exhibits resonance. Thus, similar resonances occur at $2f_1$, $3f_1$, $4f_1$ . . . through the spectrum. Various names have been applied to such resonances, including standing waves, room resonances, eigentones, permissible frequencies, natural frequencies, or simply modes. In adding two more mutually perpendicular pairs of walls, to form a rectangular enclosure, we also add two more resonance systems, each with its own fundamental and modal series.

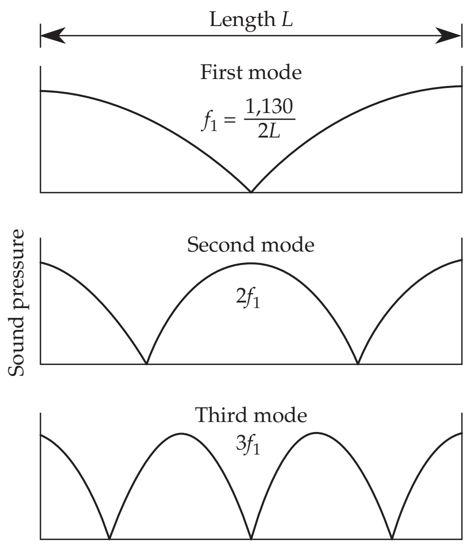


Figure 13-4 The space between two parallel, reflective walls can be considered a resonant system with a frequency of resonance of $f_1 = 1{,}130/2L$. This system is also resonant at integral multiples of f_1.

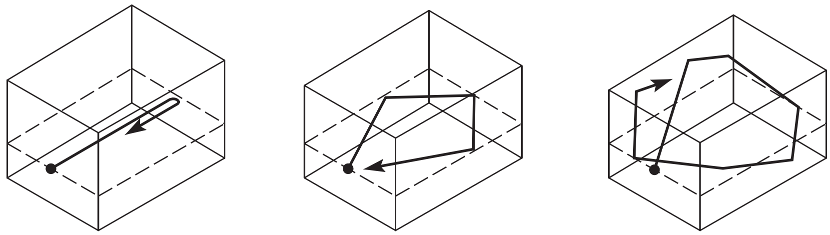


Figure 13-5 A visualization of axial, tangential, and oblique room modes using the ray concept.

Actually, the situation is far more complicated. Thus far, only axial modes have been discussed, of which each rectangular room has three, plus a modal series for each. Axial modes reflect from two opposite and parallel wall surfaces, tangential modes reflect from four wall surfaces, and oblique modes reflect from all six surfaces. If axial modes are referenced at a 0-dB level, tangential modes are plotted at –3 dB, and oblique modes at –6 dB. In practice, wall surfaces will greatly influence the actual energy in any particular mode. These three types of modes are shown in Fig. 13-5.

These mode diagrams offer clarity, but they lack rigor. In these diagrams, rays of sound are pictured as obeying the law stating that the angle of incidence equals the angle of reflection. For higher audio frequencies, this ray concept is quite useful. However, when the size of the enclosure becomes comparable to the wavelength of the sound, problems arise and the ray approach collapses. For example, a studio 30-ft long is only 1.3 wavelengths long at 50 Hz. Rays lose all meaning in such a case. A wave approach is employed to study the behavior of sound of longer wavelengths.

Frequency Regions

The audible spectrum is very wide when viewed in terms of wavelength. At 16 Hz, usually considered the low-frequency limit of the human ear, the wavelength is 1,130/16 = 70.6 ft. At the upper limit of hearing, usually quoted to be 20 kHz, the wavelength is only 1,130/20,000 = 0.056 ft or about 0.7 in. The behavior of sound is greatly affected by the wavelength of the sound in comparison to the size of objects encountered. In a room, sound of 0.7-in wavelength is scattered (diffused) significantly by a wall irregularity of a few inches. The effect of the same irregularity on sound of 70-ft wavelength would be insignificantly small. The heart of the acoustical problem is that no single analytical approach can cover sound of such a wide range of wavelengths.

In considering the acoustics of small rooms, the audible spectrum can be arbitrarily divided into four regions: A, B, C, and D, as shown in Fig. 13-6. Room size determines how the audible spectrum must be divided for acoustical analysis. In very small rooms, with too few modal resonances spaced too far apart, a great portion of the audible spectrum is dominated by modal resonances.

Region A is the very-low-frequency region below a frequency of 1,130/2L or 565/L, where L is the longest dimension of the room. Below the frequency of this lowest axial mode, there is no resonant support for sound in the room. This does not mean that such very-low-frequency sound cannot exist in the room, only that it is not boosted by room resonances because there are none in that region.

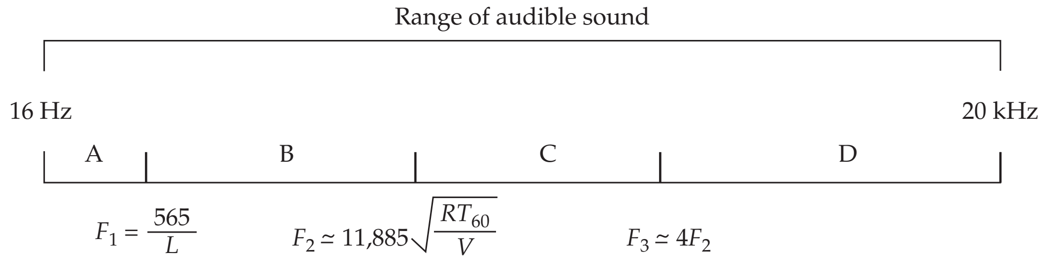


Figure 13-6 When dealing with the acoustics of enclosed spaces, it is convenient to consider the audible frequency range as comprising four regions: A, B, C, and D delineated by frequencies F_1, F_2, and F_3. There is no modal boost for sound in region A. In region B, room modes dominate. Region C is a transition zone in which diffraction and diffusion dominate. In region D, specular reflections and ray acoustics prevail.

Region B is that region in which the dimensions of the room are comparable to the wavelength of sound being considered. It is bounded on the low-frequency end by the lowest axial mode, 565/L. The upper boundary is not definite but an approximation of what has been called the cutoff or crossover frequency can be obtained from this equation:

$$F_2 = 11{,}885\sqrt{\frac{\mathrm{RT}_{60}}{V}} \tag{13-2}$$

where F_2 = cutoff or crossover frequency, Hz
RT_{60} = reverberation time of the room, sec
V = volume of the room, ft^3

Note: In some literature, the constant is given as 11,250.

Region C is a transition region between region B, in which wave acoustics must be used, and region D in which ray acoustics are valid. It is bounded on the low-frequency end approximately by the cutoff frequency F_2 and on the high end approximately by $F_3 = 4F_2$. This region is more difficult to analyze, dominated by wavelengths often too long for ray acoustics and too short for wave acoustics.

Region D describes the spectral area above F_3 that covers higher audible frequencies with short wavelengths; geometric ray acoustics apply. Specular reflections (angle of incidence equals angle of reflection) and the sound ray approach to acoustics prevail. In this region statistical approaches are generally possible.

In summary, as an example, consider a room measuring 23.3 × 16 × 10 ft. Volume is 3,728 ft^3, and reverberation time is 0.5 sec. Region A is below 565/23.3 = 24.2 Hz. There is no resonant boost for sound. Region B is between 24.2 and 130 Hz. The wave acoustical approach of modal resonances is used to predict response. Region C is between 130 Hz and (4)(130) = 520 Hz. This is a transitional region. Region D is above 520 Hz. The modal density is very high, statistical conditions generally prevail, and geometric ray acoustics can be used.

Room-Mode Equation

A solution of the wave equation can be used to calculate the room mode frequencies for a rectangular enclosure. The geometry used is shown in Fig. 13-7, which fits the familiar, mutually perpendicular x, y, z coordinates of three-dimensional space to a room. For convenience, the longest dimension L (length of the room) is placed on the x axis, the next longest dimension W (width) is placed on the y axis, and the smallest dimension H (height) on the z axis. The goal is to calculate the permissible frequencies corresponding to the modes of a rectangular enclosure. We employ the room-mode equation first stated by Rayleigh:

$$\text{Frequency} = \frac{c}{2}\sqrt{\frac{p^2}{L^2} + \frac{q^2}{W^2} + \frac{r^2}{H^2}} \tag{13-3}$$

where L, W, H = room length, width, and height, ft or m
p, q, r = integers 0, 1, 2, 3, . . .
c = speed of sound, 1,130 ft/sec or 344 m/sec

This equation gives the frequency of every axial, tangential, and oblique mode of a rectangular room. The integers p, q, and r are the only variables once L, W, and H are set for a given room. Room modes are possible only when p, q, and r are whole numbers (or zero) because this is the condition that creates a standing wave. There are many combinations of integers when fundamentals (associated with 1), second modes (associated with 2), and third modes (associated with 3), and so on, are introduced.

These integers serve to identify the mode as axial, tangential, or oblique, and also identify the frequency of a given mode. An axial mode has two zero terms such as (1, 0, 0) or (0, 0, 3); a tangential mode has one zero term such as (1, 1, 0) or (0, 3, 3); an oblique mode has no zero terms such as (1, 1, 1) or (3, 3, 3). Furthermore, the mode number indicates the frequency multiple. For example, the axial modes (0, 2, 0) and (0, 0, 2) are a multiple of the (0, 1, 0) and (0, 0, 1) modes, respectively; their frequency is twice as high. Similarly, higher integer values describe higher multiples of the fundamental mode. Tangential and oblique modes are described in the same way.

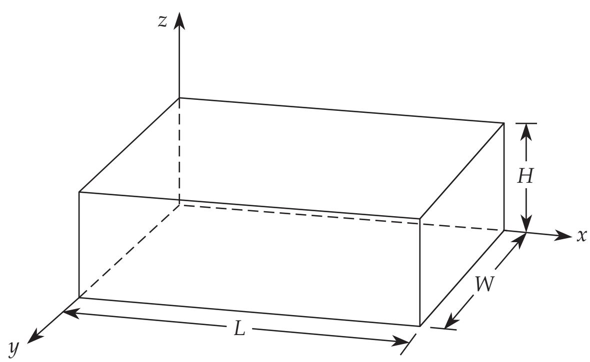


Figure 13-7 Orientation of rectangular room of length L, width W, and height H with respect to the x, y, and z coordinates for calculating room-mode frequencies.

If $p = 1$, $q = 0$, and $r = 0$ (the 1, 0, 0 mode), the width and height terms drop out and the equation becomes:

$$\text{Frequency} = \frac{c}{2}\sqrt{\frac{p^2}{L^2}} = \frac{c}{2L} = \frac{1{,}130}{2L} = \frac{565}{L} \tag{13-4}$$

This is the axial mode corresponding to the length of the room. We observe that it is identical to the mode equation for a closed pipe. The width axial mode (0, 1, 0) and the height axial mode (0, 0, 1) are calculated similarly by substituting the appropriate dimensions.

The relative dimensions of a room determine the modal response and the suitability of the dimensions. If a rectangular room has two or three dimensions that are equal or a multiple of each other, the modal frequencies will coincide, resulting in peaks at those frequencies and dips at other frequencies. For example, if a room has the dimensions of $10 \times 20 \times 30$ ft (height, width, length), all three axes of the room will have modal frequencies at 56.5, 113, 169.5, 226, 282.5, 339 Hz, and other frequencies. These coincident frequencies, called degenerate frequencies, will result in poor room frequency response and unequal energy distribution. As we will see, modal response can be greatly improved through careful selection of room dimension ratios. In this case, we see that the room dimension ratio of 1:2:3 is a poor choice.

Mode Calculations—An Example

The utility of the room-mode equation, Eq. (13-3), is best demonstrated by an example. The dimensions of a room are: length $L = 12.46$ ft, width $W = 11.42$ ft, and average height $H = 7.90$ ft. (The ceiling actually slopes along the length of the room with a height of 7.13 ft on one end and 8.67 ft on the other.) The values of L, W, and H are inserted into the mode equation along with combinations of integers p, q, and r. Note that the room is not a rectangular parallelepiped, which is the basis of the room-mode equation. For the following computation, the average ceiling height is used; in practice, this would introduce some error in the results.

Table 13-1 lists some modes for this example room in ascending frequency. It shows p, q, and r and the resulting modal frequency for each combination. Furthermore, each frequency is identified as axial, tangential, or oblique by the number of zeros in that particular p, q, and r combination. The lowest natural room frequency is 45.3 Hz, which is the (1, 0, 0) axial mode, associated with the longest dimension, the length L, of the room below which there is no modal boost of the sound. Mode 7, the (2, 0, 0) mode, which is the second axial mode associated with the length L yields a frequency of 90.7 Hz. In the same manner, mode 17, the (3, 0, 0) mode, is the third axial mode associated with the length L, mode 36 is the fourth mode, and so forth.

Axial modes and their spacings have been scrutinized in studio design. Large gaps between axial modes can lead to an uneven low-frequency response. However, there is more to room acoustics than axial modes. As Table 13-1 shows, there are many tangential and oblique modes between the axial modal frequencies that have an effect, though weaker.

Room modes are certainly more than theoretical curiosities. In fact, they play a dominant role in the low-frequency response of a room where modes are more isolated in frequency and thus more audible. It is easy to demonstrate their profound effect. If a modal frequency (e.g., a 136-Hz sine wave in this example room) is played through a

Mode Number	Mode Frequency (Hz)	Integers *p, q, r*	Axial	Tangential	Oblique
1	45.3	1, 0, 0	x		
2	49.5	0, 1, 0	x		
3	67.1	1, 1, 0		x	
4	71.5	0, 0, 1	x		
5	84.7	1, 0, 1		x	
6	87.0	0, 1, 1		x	
7	90.7	2, 0, 0	x		
8	98.1	1, 1, 1			x
9	98.9	0, 2, 0	x		
10	103.3	2, 1, 0		x	
11	108.8	1, 2, 0		x	
12	115.5	2, 0, 1		x	
13	122.1	0, 2, 1		x	
14	125.6	2, 1, 1			x
15	130.2	1, 2, 1			x
16	134.2	2, 2, 0		x	
17	136.0	3, 0, 0	x		
18	143.0	0, 0, 2	x		
19	144.8	3, 1, 0		x	
20	148.4	0, 3, 0	x		
21	150.1	1, 0, 2		x	
22	151.4	0, 1, 2		x	
23	152.1	2, 2, 1			x
24	153.7	3, 0, 1		x	
25	155.2	1, 3, 0		x	
26	158.0	1, 1, 2			x
27	161.5	3, 1, 1			x
28	164.8	0, 3, 1		x	
29	168.2	3, 2, 0		x	
30	169.4	2, 0, 2		x	
31	170.9	1, 3, 1			x
32	173.9	0, 2, 2		x	
33	173.9	2, 3, 0		x	
34	176.4	2, 1, 2			x
35	179.7	1, 2, 2			x

Table 13-1 Mode Calculations (Room Dimensions: 12.46 × 11.42 × 7.90 ft)

Mode Number	Mode Frequency (Hz)	Integers *p, q, r*	Axial	Tangential	Oblique
36	181.4	4, 0, 0	x		
37	182.8	3, 2, 1			x
38	188.0	4, 1, 0		x	
39	188.1	2, 3, 1			x
40	195.0	4, 0, 1		x	
41	196.2	2, 2, 2			x
42	197.4	3, 0, 2		x	
43	197.9	0, 4, 0	x		
44	201.2	4, 1, 1			x
45	201.3	3, 3, 0		x	
46	203.0	1, 4, 0		x	
47	203.5	3, 1, 2			x
48	206.1	0, 3, 2		x	
49	206.6	4, 2, 0		x	
50	210.4	0, 4, 1		x	
51	211.1	1, 3, 2			x
52	213.7	3, 3, 1			x
53	214.6	0, 0, 3	x		
54	215.3	1, 4, 1			x
55	217.7	2, 4, 0		x	
56	218.6	4, 2, 1			x
57	219.3	1, 0, 3		x	
58	220.2	0, 1, 3		x	
59	220.8	3, 2, 2			x
60	224.8	1, 1, 3			x
61	225.2	2, 3, 2			x
62	226.7	5, 0, 0	x		
63	229.1	2, 4, 1			x
64	231.0	4, 0, 2		x	
65	232.1	5, 1, 0		x	
66	232.9	2, 0, 3		x	

Table 13-1 Mode Calculations (Room Dimensions: 12.46 × 11.42 × 7.90 ft) *(Continued)*

loudspeaker and the listener walks through the room, the loud antinodes and soft nodes will be clearly audible at different locations along the length of the room. Depending on the mode, the rise and fall of the tone's loudness may be heard across the room's length, width, or height, or a combination of those. Alternatively, if a listener is seated in one location, and a sine-wave generator is used to sweep across a low-frequency range, even if the loudspeaker's response is flat, great variations in level will be heard at different frequencies because of room modes. Moreover, a listener seated in another location will hear a different response.

Consider what would happen if a sound source was placed at a node; it would only weakly sound that frequency. Or, consider a mixing engineer seated at an antinode; the loudness of certain frequencies would be accentuated. Room modes can greatly affect a room's low-frequency response, depending on the physical positioning of sound sources and receivers in the room.

Experimental Verification

The modal frequencies listed in Table 13-1 largely determine the response of this particular room for the frequency range specified. To evaluate their relative effects, a swept sine-wave transmission experiment was performed. In effect, this measures the frequency response of the room. Knowing that all room modes terminate in the corners of a room, a loudspeaker (JBL 2135) was placed in one low tri-corner of the room and a measuring microphone was placed in the diagonal high tri-corner. The loudspeaker was then energized by a slowly swept sine-wave signal, sweeping from 10 to 250 Hz. The room response to this signal at the microphone was recorded. The resulting trace is shown in Fig. 13-8.

Attempts have been made to identify the effects of each mode in reverberation chambers with all six surfaces hard and reflective. In such cases, the prominent modes stand out as sharp spikes. The practical room in which the recording of Fig. 13-8 was made is not a reverberation chamber. Instead of concrete, the walls are of frame construction covered with gypsum board (drywall); carpet over plywood comprises the floor, and doors almost cover one wall. There is a large window, pictures on the walls, and some furniture. It is evident that this is a fairly absorbent room. The reverberation time is 0.33 sec at 125 Hz. This room is much closer acoustically to studios and control rooms than to reverberation chambers, and this is why it was chosen.

There are 12 axial modes, 32 tangential modes, and 22 oblique modes in the 45.3- to 232.9-Hz response, and the room response in this range is the composite effect of all 66 modes. Using only Fig. 13-8, an attempt to correlate the peaks and valleys of the experimentally obtained room response to specific axial, tangential, and oblique modes would be disappointing. The response can be attributed to modes and the interaction of modes; for example, modes close together would be expected to reinforce room response if in phase, and cancel if out of phase. In practice, as we will see, a more extensive analysis is needed to completely understand modal room response.

The three major dips are so narrow that they would take little energy from a distributed speech or music spectrum. If they are neglected, the remaining fluctuations are more modest. Fluctuations of this magnitude in such steady-state swept-sine transmission tests are characteristic of even carefully designed studios, control rooms, and listening rooms. The ear commonly accepts such deviations from a flat response. The modal structure of a space always gives rise to these

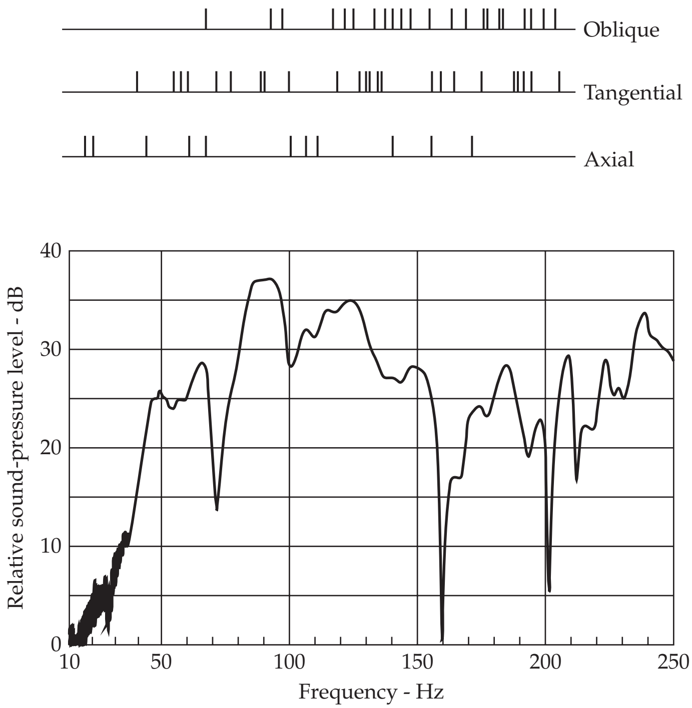


Figure 13-8 Swept sine-wave transmission run in the test room of Table 13-1. The location of each axial, tangential, and oblique modal frequency is indicated.

kinds of fluctuations. However, there is more to acoustical performance than the steady-state response considered so far.

Mode Decay

The steady-state response of Fig. 13-8 tells only part of the story. The ear is sensitive to transient effects, and speech and music are made up almost entirely of transients. Reverberation decay is one transient phenomenon that is easily measured. When broadband sound such as speech or music excites the modes of a room, our interest centers on the decay of the modes. The 66 modes in the 45.3- to 232.9-Hz band form the microstructure of the reverberation of the room in that range. Reverberation is commonly measured in octave bands. Octave bandwidths of interest are shown in Table 13-2.

Each reverberatory decay by octave bands thus involves an average of the decay of many modes, but it takes an understanding of the decay of the individual modes to explain the decay of the octave band of sound. The higher the octave center frequency, the greater the number of modes included.

All modes do not decay at the same rate. Mode decay depends, among other things, on the way absorbing material is distributed in the room. Carpet on the floor of the test room has no effect on the (1, 0, 0) or (0, 1, 0) axial modes which involve only walls.

Limits		Modes		
Octave (Hz)	(–3-dB Points) (Hz)	Axial	Tangential	Oblique
63	45–89	3	3	0
125	89–177	5	15	8
250	177–233	4*	14*	14*

*Partial octave.

Table 13-2 Modes in Octave Bands

Tangential and oblique modes, which involve more surfaces, would be expected to die away faster than axial modes that involve only two surfaces. On the other hand, absorption is greater for the axial modes in which the sound impinges on the surfaces at right angles than for the low angle of incidence common for tangential and oblique modes.

The reverberatory decays in the test room using sine-wave excitation are shown in Fig. 13-9. Single prominent modes decaying alone yield smooth, logarithmic traces. The dual-slope decay at 240 Hz is interesting because the low value of the early slope (0.31 sec) is probably dominated by a single mode involving much absorption, later giving way to other modes that encounter much less absorption.

Identifying these modes from Table 13-1 is difficult, although you might expect the axial mode at 214.6 Hz to die away slowly and the group of modes near 220 Hz to be highly damped. It is common to force nearby modes into oscillation, which then decay at their natural frequency.

Figure 13-10 shows measurements using octave bands of pink noise. The decay of the 125-Hz octave band of noise (0.33 sec) and the 250-Hz octave band (0.37 sec) averages many modes and should be considered more or less the "true" values for this frequency region. However, normally, many decays for each band would be taken to provide statistical significance. The measured reverberation times for this room are summarized in Table 13-3.

Mode Bandwidth

Normal modes determine room resonances. Taken separately, each normal mode exhibits a resonance curve such as shown in Fig. 13-11. Bandwidth is defined as the half-power (–3-dB) points on either side of a resonant peak. Each mode has a bandwidth determined by the expression:

$$\text{Bandwidth} = f_2 - f_1 = \frac{2.2}{RT_{60}} \tag{13-5}$$

where f_2 and f_1 are frequencies at –3 dB, Hz
RT_{60} = reverberation time, sec

Bandwidth is inversely proportional to reverberation time. The deader the room, the broader the bandwidth of the resonance curve. In electric circuits, the sharpness of the tuning curve depends on the amount of resistance in the circuit; the greater the

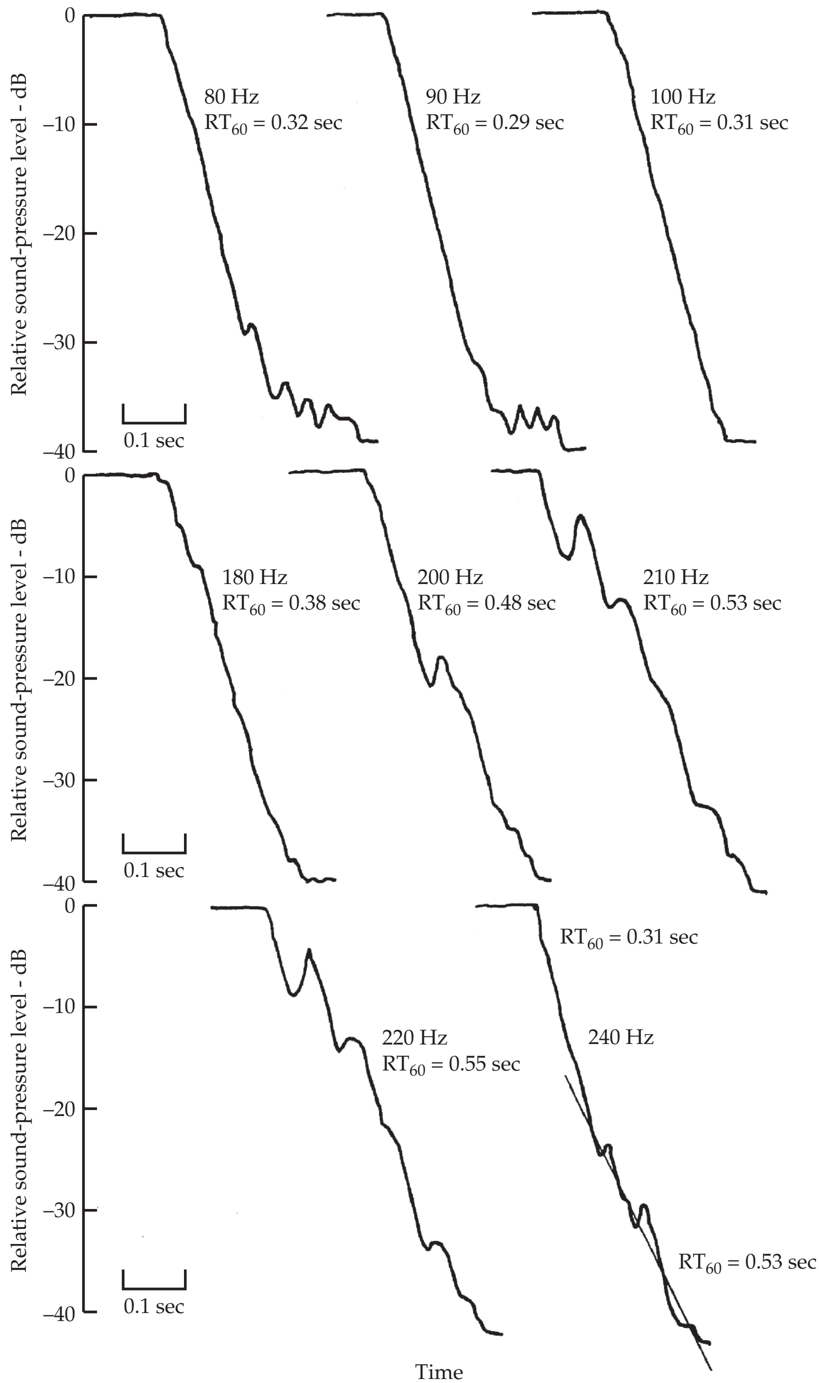


FIGURE 13-9 Pure-tone reverberation-decay recordings made in a test room. Single modes decaying alone yield smooth, logarithmic traces. Beats between neighboring modes cause irregular decays. The two-slope pattern at 240 Hz reveals the smooth decay of a single prominent mode for the first 20 dB, after which one or more lightly absorbed modes dominate.

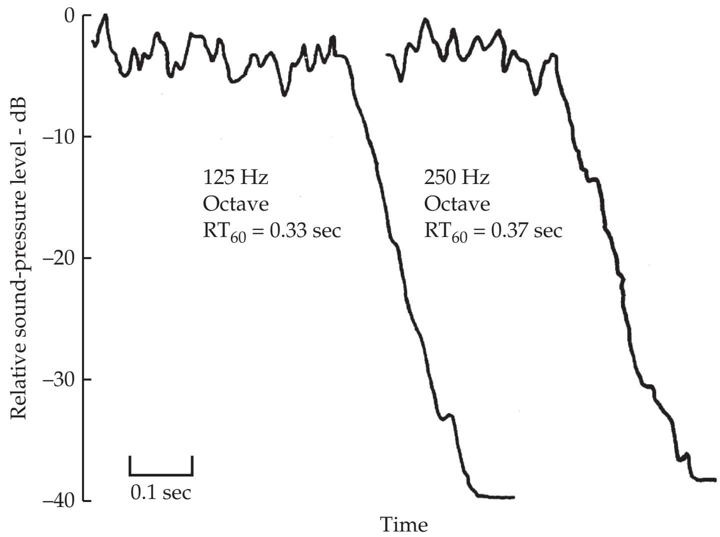


Figure 13-10 Octave-band pink-noise reverberation-decay recordings made in a test room. Decay of the 125- and 250-Hz octave bands indicates the average decay of all modes in those octaves.

resistance, the broader the tuning curve. In room acoustics, the reverberation time depends on absorption (resistance). The analogy is fitting (the more absorption, the shorter the reverberation time, and the wider the mode resonance). For convenient reference, Table 13-4 lists a few values of bandwidth in relation to reverberation time.

Figure 13-12 shows an expanded version of the 40- to 100-Hz portion of Fig. 13-8. The axial modes from Table 13-1 falling within this range are plotted with a bandwidth of 6 Hz at the –3-dB points. The axial modes are peaked at zero reference level. The tangential modes have only one-half the energy of the axial modes, so their peaks are plotted 3 dB (10 log 0.5) below the axial modes. The oblique modes have only one-fourth the

Frequency (Hz)	Average Reverberation Time (sec)
80	0.32
90	0.29
100	0.31
180	0.38
200	0.48
210	0.53
220	0.55
240	0.31 and 0.53 (double slope)
125-Hz octave noise	0.33
250-Hz octave noise	0.37

Table 13-3 Measured Reverberation Time of Test Room

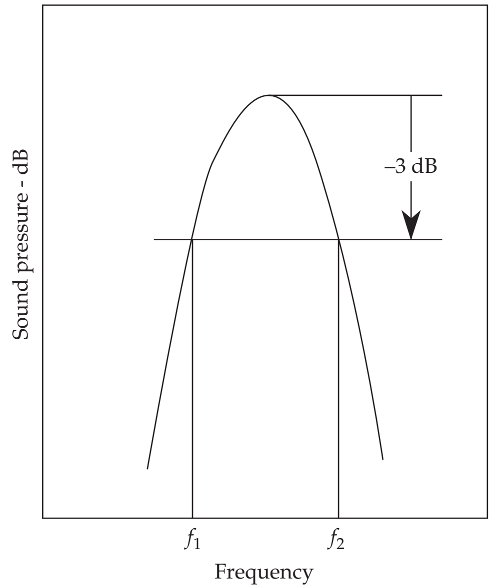


FIGURE 13-11 Each room mode has a finite bandwidth. Bandwidth is customarily measured at –3-dB points. The more absorbent the room, the greater the bandwidth.

energy of the axial modes, so the single oblique mode at 98.1 Hz that falls within this range is plotted 6 dB (10 log 0.25) below the axial mode peaks.

The mode bandwidth of a typical room is about 5 Hz. This means that adjacent modes tend to overlap for rooms with short reverberation time, which is desirable. As the skirts of the resonance curves of adjacent modes overlap (e.g., as indicated by the axial and tangential modes labeled in Fig. 13-12), energizing one mode by driving the room at its resonant frequency will also tend to force the other mode into excitation. When the first excitation frequency is removed, the energy stored in the other mode decays at its own frequency. The two will beat with each other during the decay. Very uniform decays, such as the 80, 90, and 100-Hz modes in Fig. 13-9, are probably single

Reverberation Time (sec)	Mode Bandwidth (Hz)
0.2	11
0.3	7
0.4	5.5
0.5	4.4
0.8	2.7
1.0	2.2

TABLE 13-4 Mode Bandwidth

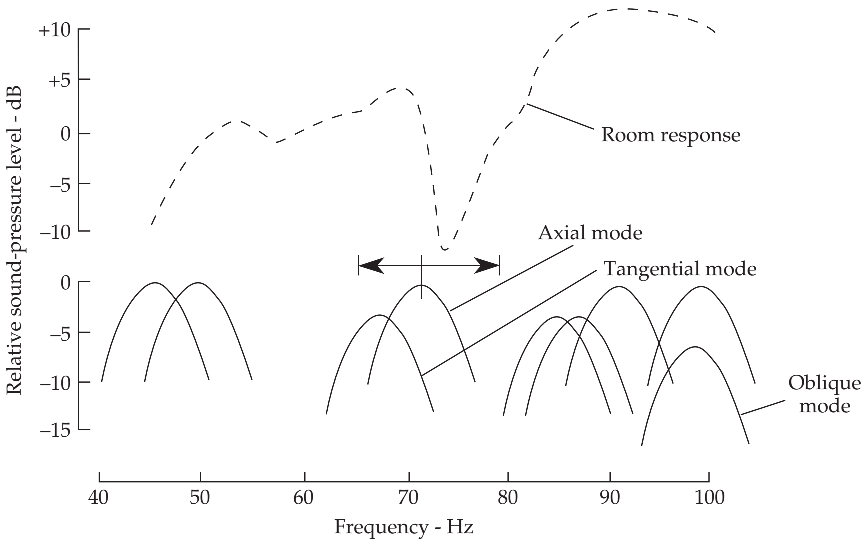


FIGURE 13-12 Correlation of measured swept-sine response and calculated modes of the test room. The portion of the room response from 40 to 100 Hz from Fig. 13-8 is shown. Axial, tangential, and oblique modes are plotted.

modes sufficiently removed from neighbors so as to act independently without irregularities caused by beats.

The overall response of the test room (labeled in Fig. 13-12) is most certainly made up of the collective contributions of the various modes tabulated in Table 13-1. Can the response be accounted for by the collective contributions of the axial and tangential modes, and the single oblique mode in this frequency region? It seems reasonable to account for the 12-dB peak in the room response between 80 and 100 Hz by the combined effect of the two axial, two tangential, and one oblique mode in that frequency range. The falloff below 50 Hz is undoubtedly due to loudspeaker response. This leaves the 12-dB dip at 74 Hz yet to be accounted for.

The axial mode at 71.5 Hz is the vertical mode of the test room with a sloping ceiling. The frequency corresponding to the average height is 71.5 Hz, but the one corresponding to the height at the low end of the ceiling is 79.3 Hz and for the high end it is 65.2 Hz. The uncertainty of the frequency of this mode is indicated by the double arrow above the 71.5-Hz mode (marked in Fig. 13-12). If this uncertain axial mode were shifted to a slightly lower frequency, the 12-dB dip in response could be better explained. It would seem that a dip in response should appear near 60 Hz, but none was found.

Although this test room experimentally verifies many aspects of room-mode theory, more importantly, it demonstrates several practical lessons. The test room is not a rectangular parallelepiped, which is the assumed basis of the mode equation. In addition, when combining the effects of individual modes to obtain an overall response, phase must be taken into account. These components must be combined vectorially with both magnitude and phase fully considered. Finally, and importantly, each mode is not a fixed entity, but one whose magnitude varies from zero to maximum, depending on the location in the room. Individual pressure plots vary across the room, and it is not possible to determine the overall magnitude at a point in the room by simply summing all the modes. For example, at any point, a mode could be zero or maximum. Room response should be considered as combined modal responses distributed over the physical dimensions of the room, discussed as follows.

Mode Pressure Plots

The modal pattern of a room creates a very complex sound field. To illustrate this, several sketches of sound-pressure distributions are shown. The one-dimension organ pipe of Fig. 13-1 can be compared to the (1, 0, 0) axial mode of Fig. 13-13 for a three-dimensional room. The sound pressure is higher (1.0) near the ends and zero along the center of the room. Figure 13-14 shows sound-pressure distribution when only the (3, 0, 0) axial mode is energized. The sound-pressure nodes and antinodes in this case are straight lines as shown in Fig. 13-15.

Three-dimensional sketches of sound-pressure distribution throughout a room become difficult at higher modes, but Fig. 13-16 is an attempt to show the (2, 1, 0) tangential mode. We see sound-pressure maxima in each corner of the room with two more maxima at the center edges. This is topographically portrayed in Fig. 13-17 that shows the pressure contour lines. The broken lines crisscrossing the room between the maxima of sound mark the zero-pressure regions.

Imagine how complicated the sound-pressure pattern would be if all the modes were concurrently or sequentially excited by voice or music energy moving up and down the spectrum while constantly shifting in intensity. The plot of Fig. 13-17 shows sound-pressure maxima in the corners of the room. These maxima always appear in room corners for all modes. To excite all modes, place the sound source in a corner. Conversely, if you wish to measure all modes, the microphone should be located in a corner.

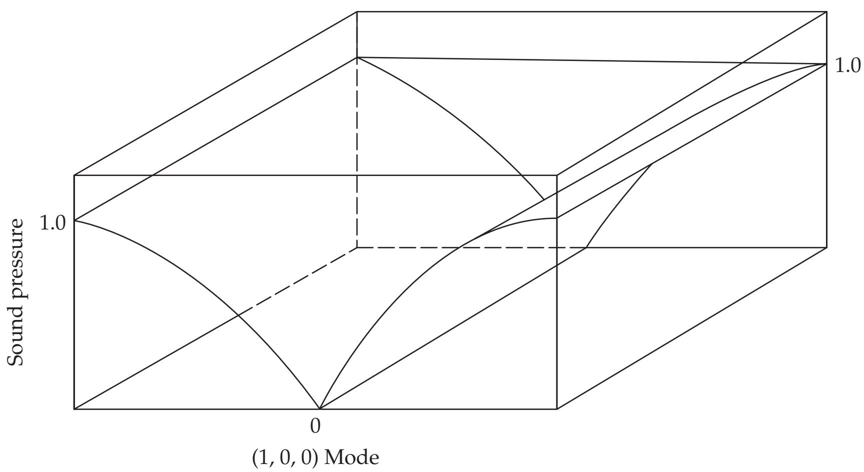


Figure 13-13 A representation of the sound-pressure distribution of the (1, 0, 0) axial mode of a room. The sound pressure is zero in the vertical plane at the center of the room and maximum at the ends of the room. This is comparable to f_1 of the closed pipe of Fig. 13-1.

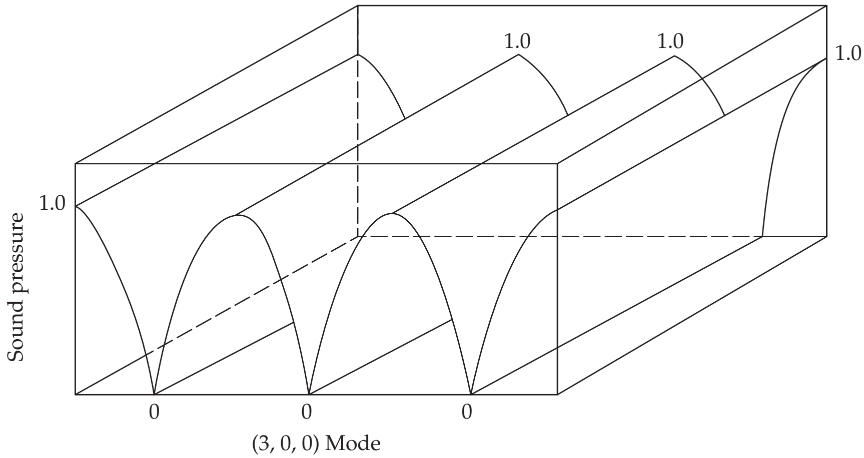


FIGURE 13-14 A representation of the sound-pressure distribution of the (3, 0, 0) axial mode of a room.

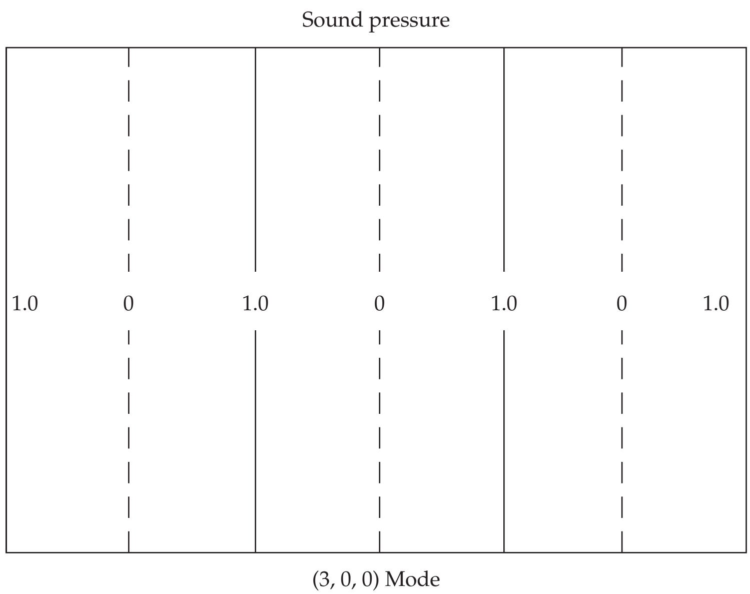


FIGURE 13-15 Sound-pressure contours on a section through a rectangular room for the (3, 0, 0) axial mode.

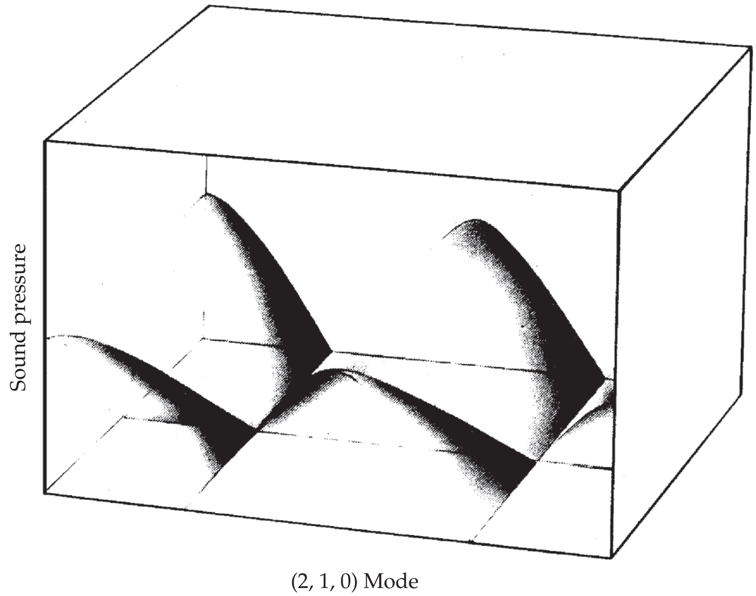


FIGURE 13-16 Three-dimensional representation of the sound-pressure distribution in a rectangular room for a (2, 1, 0) tangential mode. (*Brüel & Kjaer Instruments, Inc.*)

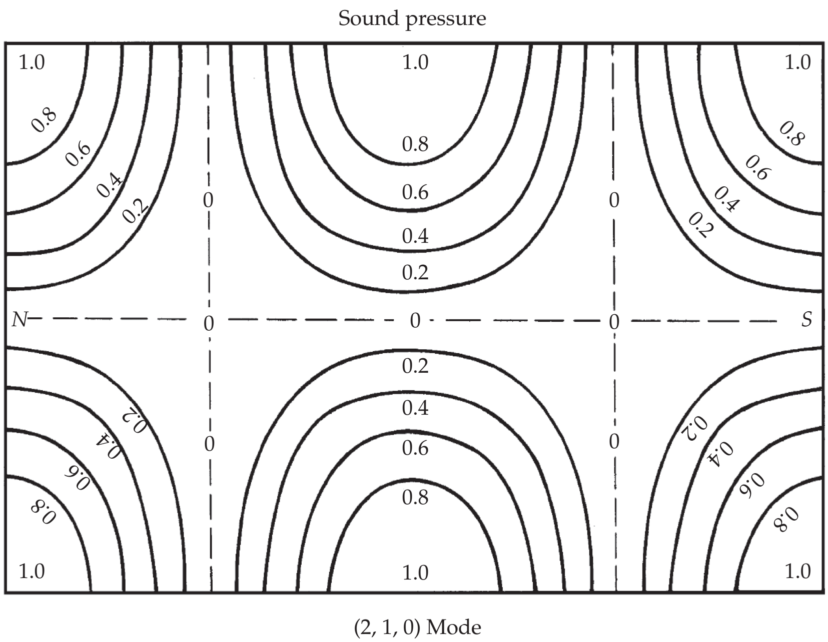


FIGURE 13-17 Sound-pressure contours of the rectangular room of Fig. 13-16 for a (2, 1, 0) tangential mode.

Mode Density

Mode density increases with frequency. For example, in Table 13-1, in the one-octave spread between 45 and 89 Hz, only 6 modal frequencies are counted. In the next highest octave there are 28 modes. Even in this very limited low-frequency range below 200 Hz, we see modal density increasing with frequency. Figure 13-18 shows that at higher frequencies the rate of increase rises dramatically. Above about 300 Hz or so, the mode spacing is so small that the room response smooths markedly with frequency. The following equation can be used to determine the approximate number of modes in a given bandwidth at a given center frequency:

$$\Delta N = \left[\frac{4\pi V f^2}{c^3} + \frac{\pi S f}{2c^2} + \frac{L}{8c}\right]\Delta f \tag{13-6}$$

where ΔN = number of modes
Δf = bandwidth, Hz
f = center frequency, Hz
$V = (l_x l_y l_z)$ = room volume, ft^3
$S = 2(l_x l_y + l_y l_z + l_x l_z)$ = room surface area, ft^2
$L = 4(l_x + l_y + l_z)$ = lengths of all edges in room, ft
c = speed of sound = 1,130 ft/sec

This equation shows the modal density at a given bandwidth increases with frequency. Similarly, modal density increases as room size increases.

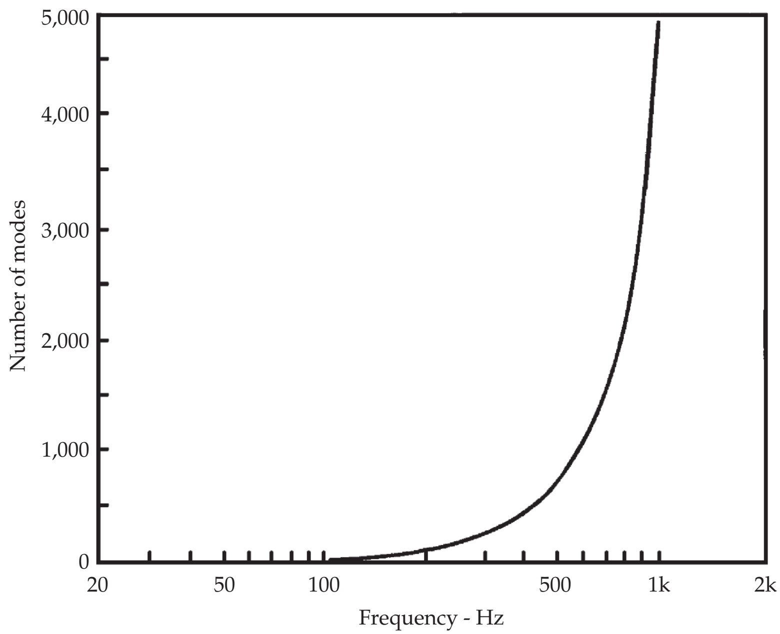


FIGURE 13-18 The number of modal frequencies increases with frequency.

Room modes are most important in smaller rooms where modal density is low; that particularly occurs at low frequencies. In large rooms, the modal density is relatively high except at extremely low frequencies, and room modes play a less important role. The cutoff frequency between a small room and a large room can be estimated with the Schroeder equation:

$$f_c = 11{,}885\sqrt{\frac{RT_{60}}{V}} \tag{13-7}$$

where f_c = cutoff frequency, Hz
RT_{60} = reverberation time, sec
V = room volume, ft^3 or m^3

Note: In metric units, change 11,885 to 2,000.

Mode Spacing and Timbral Defects

Timbral defects are frequency-response anomalies in an audio signal and are potentially audible. They can influence the quality of sound in any acoustically sensitive space, ranging from listening rooms to concert halls. The task is to determine which, if any, of the hundreds of modal frequencies in a room are likely to cause audible defects.

Timbral defects can contribute to poor characteristics for recording or other critical work. A musical pitch falling between widely separated modes may be abnormally weak and may decay faster than other notes. In a sense, it is as though that particular note was sounded outdoors while the other notes were simultaneously sounded indoors.

The spacing of the modal frequencies is a critically important factor. In the D region of Fig. 13-6, the modal frequencies of a small room are so close together that they tend to merge helpfully and harmlessly. In the B and C regions, below about 300 Hz, their separation is greater and it is in this region that problems can arise. To avoid timbral defects, modes should not be isolated from one another, but neither should they coincide with each other.

How much separation is allowed between modal frequencies before timbral defects arise? Gilford states his opinion that an axial mode separated more than 20 Hz from the next axial mode will tend to be isolated acoustically. It will tend not to be excited through coupling due to overlapping skirts but will tend to act independently. In this isolated state, it can respond to a component of the signal near its own frequency and give this component its proportional resonant boost, thus potentially creating timbral defects.

Gilford's main concern was that wide axial-mode spacing can cause frequency-response deviations resulting from independent and uncoupled modal action. Another criterion for mode spacing was suggested by Bonello. He analyzed modal separations to avoid degeneracy (coincident) effects. In this type of analysis, some separation is preferred. Also, he considered all three types of modes, not axial modes alone. He stated that it is desirable to have all modal frequencies in a critical band at least 5% of their frequency apart. For example, one modal frequency at 20 Hz and another at 21 Hz would be barely acceptable. However, a similar 1-Hz spacing would not be acceptable at 40 Hz (5% of 40 Hz is 2 Hz).

Zero spacings between modal frequencies are a common source of frequency-response deviations. Zero spacing means that two modal frequencies are coincident and such degeneracies tend to overemphasize signal components at that frequency. Modes should not be widely separated from one another, but neither should they coincide with each other.

Audibility of Timbral Defects

Any ear can be offended by timbral defects caused by isolated or boosted modes, but even a trained ear needs some instrumental assistance to identify and evaluate such response deviations. In a BBC Research Department study, observers listened to persons speaking into a microphone in the studio under investigation, the voices being reproduced in another room over a high-quality playback system. Observers' judgments were aided by selectively amplifying a narrow frequency band (10 Hz) to a level about 25 dB above the rest of the spectrum. The output was mixed in small proportions with the original signal to the loudspeaker, the proportions being adjusted until they were barely perceptible as a contribution to the whole output. Any frequency-response deviations were then made clearly audible when the selective amplifier was tuned to the appropriate frequency.

In most studios tested this way, only one or two obvious timbral defects were found in each. Figure 13-19 is a plot of audible timbral defects in 61 male voices observed over a period of 2 years. Most fall between 100 and 175 Hz. Female voice response variations occur most frequently between 200 and 300 Hz.

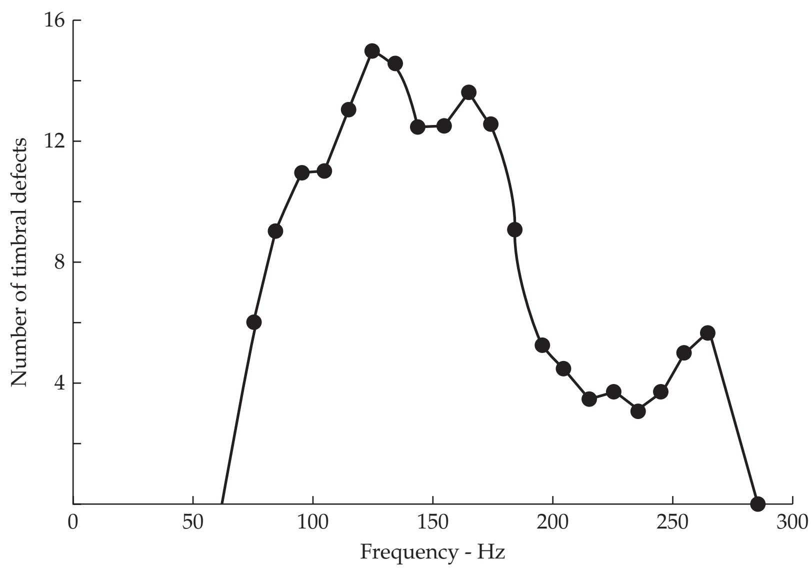


FIGURE 13-19 A plot of timbral defects in 61 male voices observed over a period of 2 years in BBC studios. Most occur in the 100- to 175-Hz region. Most response variations in female voices occur in the 200- to 300-Hz region. (*Gilford.*)

Optimal Room Proportions

Can a room be proportioned to achieve optimal modal distribution throughout the room? This question is answered by many strong opinions—some of them supported by quite convincing experiments—and some just as strong but without much support.

Room geometry greatly affects acoustical performance. The ubiquity of rectangular rooms is largely due to the economy of construction, but the geometry has acoustical advantages. The axial, tangential, and oblique modes can be calculated with little effort and their distribution studied. For a first approximation, a good approach is to consider only the more dominant axial modes. Degenerate modes can be identified and other room faults revealed.

The relative proportioning of length, width, and height of an acoustically sensitive room is an important consideration. When plans are being drawn for such a room, one should start with basic room proportions. Cubical rooms are anathema because their modal distribution is poor.

The literature contains early quasi-scientific guesses, and later statistical analyses of room proportions that give good modal distribution. Bolt gave a range of room proportions producing smooth room characteristics at low frequencies in small rectangular rooms. Volkmann's 2:3:5 proportion is sometimes used. Boner suggested the $1:\sqrt[3]{2}:\sqrt[3]{4}$ (or 1:1.26:1.59) ratio as optimum. Sepmeyer suggested several favorable ratios. Louden listed 125 dimension ratios based on the 1:1.4:1.9 ratio and arranged them in descending order of room acoustical quality.

Table 13-5 summarizes some of the favorable rectangular-room proportions suggested by these researchers. Figure 13.20 plots these proportions in relation to the favorable area suggested by Bolt. Most of the ratios fall on or very close to the "Bolt area." This would suggest that a ratio falling in the Bolt area will yield acceptable low-frequency room quality as far as distribution of axial-mode frequencies is concerned. However, particularly with small rooms, where satisfactory modal response is most difficult, any proposed ratio must be tested. The frequency range of validity for the Bolt specification varies with room volume; for example, in an 8,000-ft^3 room, the validity range is about 20 to 80 Hz.

Author		Height	Width	Length	In Bolt's Range?
Sepmeyer	*A*	1.00	1.14	1.39	No
	B	1.00	1.28	1.54	Yes
	C	1.00	1.60	2.33	Yes
Louden	*D*	1.00	1.4	1.9	Yes
	E	1.00	1.3	1.9	No
	F	1.00	1.5	2.1	Yes
Volkmann (2:3:5)	*G*	1.00	1.5	2.5	Yes
Boner $1:\sqrt[3]{2}:\sqrt[3]{4}$	*H*	1.00	1.26	1.59	Yes

Table 13-5 Rectangular Room Dimension Ratios for Favorable Mode Distribution

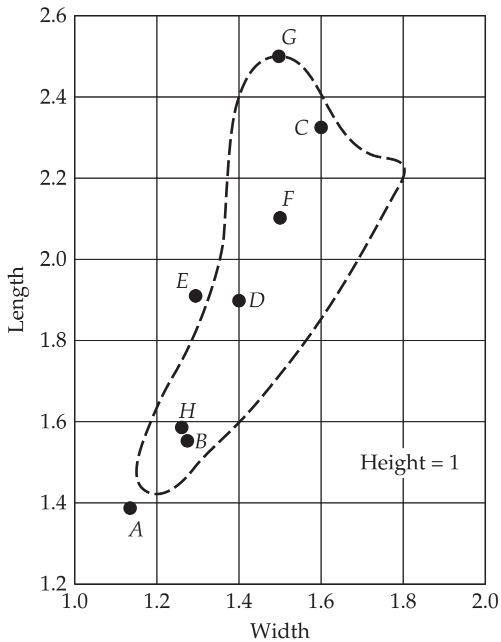


Figure 13-20 A chart of favorable room dimension ratios to achieve uniform distribution of modal frequencies in a room. The broken line encloses the Bolt area. The frequency range of validity for the Bolt specification varies with room volume. The letters refer to Table 13-5.

One cannot tell by looking at a room's dimensional ratio whether it is desirable or not; specific analysis is required. Assuming a room height of 10 ft, and choosing the other two dimensions as favorable ratios, an axial mode analysis can be made. The resulting modes for room dimension ratios *A* through *H* (from Table 13-5) are plotted in Fig. 13-21.

With a ceiling height of 10 ft, the rooms in Fig. 13-21 are all relatively small. All small rooms suffer the same problem of having axial-mode separation that is greater than desired. The more uniform the spacing, the better. Mode coincidences are a potential problem and they are identified in Fig. 13-21 by the "2" or "3" superscripts above them to indicate the number of coincident resonances. Modes very close together, even though not actually coincident, can also present problems. With these rules to follow, which of the distributions of Fig. 13-21 are preferred?

In this example, we might reject *G* because of two triple coincidences spaced apart from neighboring modes. We might also eliminate *F* because of three double coincidences and some spacings. We can neglect the effect of the double coincidences near 280 Hz in *C* and *D* because frequency-response deviations are rarely experienced above 200 Hz. Aside from these reservations, there is little to differentiate between the remainder. Each has flaws, but each would probably serve well, alerted as we are to potential problems. This simple analysis of mode distribution only considers axial modes, knowing that the weaker tangential and oblique modes can only help by filling in between the more widely spaced axial modes. This may alter our opinion of a particular preferred ratio.

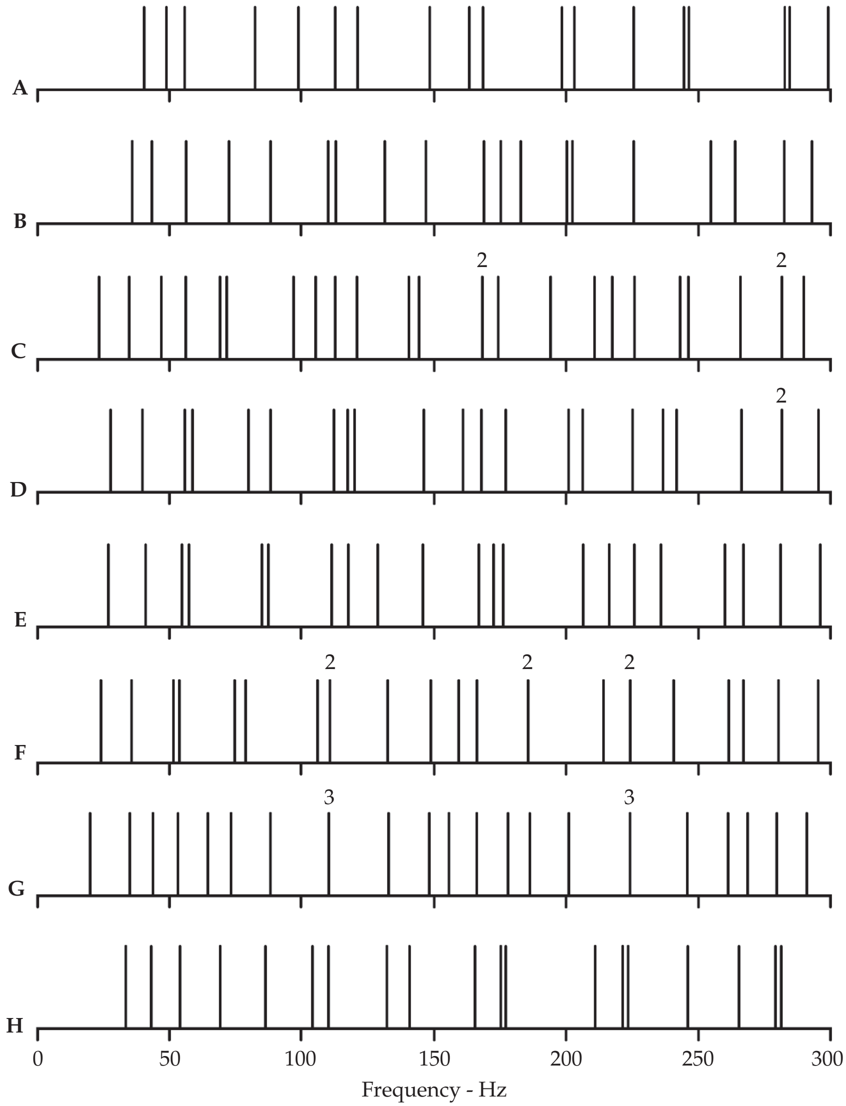


FIGURE 13-21 Plots of axial mode distribution for the eight favorable room proportions of Table 13-5. The numbers indicate the number of modes coincident at those particular frequencies. A room height of 10 ft is assumed.

If a new room is being constructed, you may have the freedom to move a wall or raise or lower the ceiling to improve modal distribution. When choosing room dimensions, a primary goal is to avoid coincidences of axial modes. For example, if a cubical space were analyzed, the three fundamentals and all harmonics would coincide. This produces a triple coincidence at each modal frequency, and also gaps between modes. Unquestionably, sound in such a cubical space would be acoustically very poor. Rather, room modes should be as evenly spaced in frequency as possible.

For example, a room measuring 19.42-ft long, 14.17-ft wide, and 8-ft high will have 21 axial modes between 29.1 and 290.9 Hz. If evenly spaced, the spacing would be about 13 Hz, but spacings vary from 3.2 to 29.1 Hz. However, there are no coincidences; the closest axial modes are 3.2 Hz apart. While this cannot be represented as the best proportioning possible, this room, properly treated, will yield good sound quality from a modal standpoint. The proper starting point is proper room proportions.

In adapting an existing space, you may lack the freedom to vary room proportions. However, a study of the axial modes can still be very helpful. For example, if analysis reveals a coincidence that is well separated from neighboring modes, one is alerted to potential problems with an understanding of the cause. As one solution, a Helmholtz resonator may be tuned to the offending coincidence frequency to control its effect. Alternatively, if space permits, a new wall placed within an existing wall might improve the modal distribution.

Bonello Criterion

In reviewing various preferred room ratios, we have observed that choosing the right ratio of dimensions is crucial in obtaining good modal response. For example, the ratio of any two dimensions should not be a whole number or close to a whole number. Bonello suggests a method for determining the acoustical desirability of the proportions of rectangular rooms. He divides the low end of the audible spectrum into bands 1/3-octave wide and considers the number of modes in each band below 200 Hz. The 1/3-octave bands are chosen because they approximate the critical bands of the human ear.

To meet Bonello's criterion, each 1/3 octave should have more modes than the preceding one, or at least the same number. Modal coincidences are not tolerated unless there are at least 5 modes in that band. Does a room measuring 15.4 × 12.8 × 10 ft meet this criterion? Figure 13-22 shows that it passes the test. The plot climbs steadily upward with no downward anomalies. The horizontal section at 40 Hz is allowed. This suggests good modal response.

Although many authors have suggested various room dimension ratios for optimal modal response, it is important to remember that there is no ideal ratio of dimensions. Furthermore, the quest for a perfect ratio is an impossible one. In real rooms, the structural integrity of the room is not uniform at low frequencies; with a given sound source location, the various modes are not excited equally; and a seated listener can only hear a few of the modes. Modal response is a genuine problem, but predicting the response with certainty using generalized assumptions is very difficult. In other words, working from general guidelines and recommendations, the modal response of each room must be considered on a case-by-case basis.

Splaying Room Surfaces

Splaying (canting) one or two walls of an acoustically sensitive room does not eliminate modal problems, although it might shift them slightly in the room and produce nominally better diffusion. Splayed walls may not add cost during new construction, but they can be quite expensive in adapting an existing space. Flutter echoes can be controlled by splaying one of two opposing walls. The amount of splaying is usually between 1:20 and 1:10, for example, 1 ft in 20 ft and 1 ft in 10 ft. In some recording studio control-room designs, the front surfaces of the room are splayed so that early reflections from the

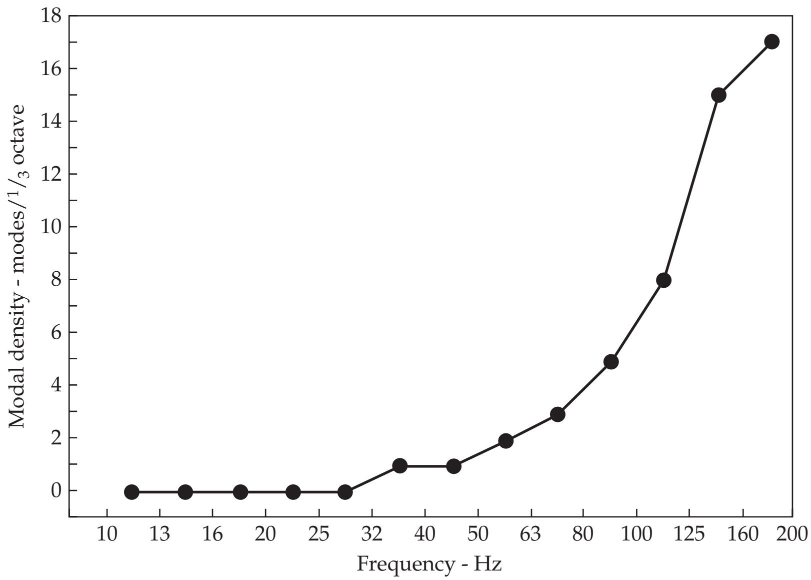


Figure 13-22 A plot showing the number of modes in 1/3-octave bands for a room measuring 15.4 × 12.8 × 10 ft. The plot climbs steadily upward with no downward anomalies; hence the room meets the Bonello criterion.

monitor loudspeakers are directed away from the mixing position; this type of design is sometimes called a reflection-free zone. This is discussed in more detail in Chap. 24.

Nonrectangular Rooms

The acoustical benefit derived from the use of nonrectangular geometries in audio rooms is controversial. As Gilford noted, slanting the walls to avoid parallel surfaces does not eliminate timbral defects, but it does make them more difficult to predict. Trapezoidal-shaped spaces, commonly used as the outer shell of recording studio control rooms, guarantee asymmetrical low-frequency sound fields even though it is generally agreed that bilateral symmetry in a control room is highly desirable.

Computer studies based on the finite element approach reveal what happens to a low-frequency sound field in a nonrectangular room. The results of a study conducted by van Nieuwland and Weber are shown in Figs. 13-23 through 13-26, comparing a rectangular geometry to a nonrectangular one. Highly contorted sound fields are shown for the four modes in the nonrectangular geometry. A shift in frequency of the standing wave from that of the rectangular geometry of the same area is indicated: –8.6%, –5.4%, –2.8%, and +1% in the four cases illustrated. This would support the common statement that splaying of walls helps slightly in breaking up degeneracies, but shifts of 5% or more are needed to avoid the effects of degeneracies. The proportions of a rectangular room can be selected to eliminate, or at least greatly reduce degeneracies, while in the case of the nonrectangular room, a prior examination of degeneracies is difficult. Making the sound field asymmetrical by splaying walls introduces unpredictability in the design. If the decision is made to splay walls in a room, say 5%, a reasonable approximation would be to analyze the equivalent rectangular room having the same volume.

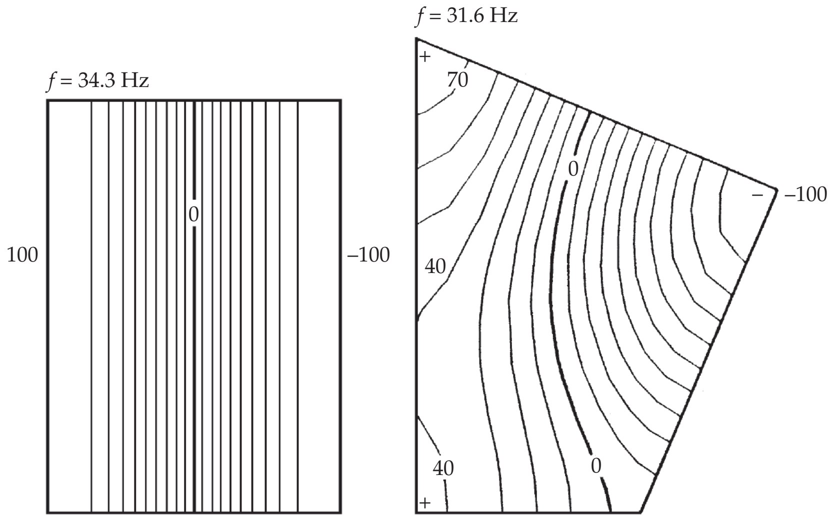


Figure 13-23 Comparison of the modal patterns for a two-dimensional rectangular room measuring 16.4 × 23 ft and a nonrectangular room of the same area. The sound field of the (1, 0, 0) mode is distorted in the nonrectangular room and the frequency of the mode is shifted slightly.

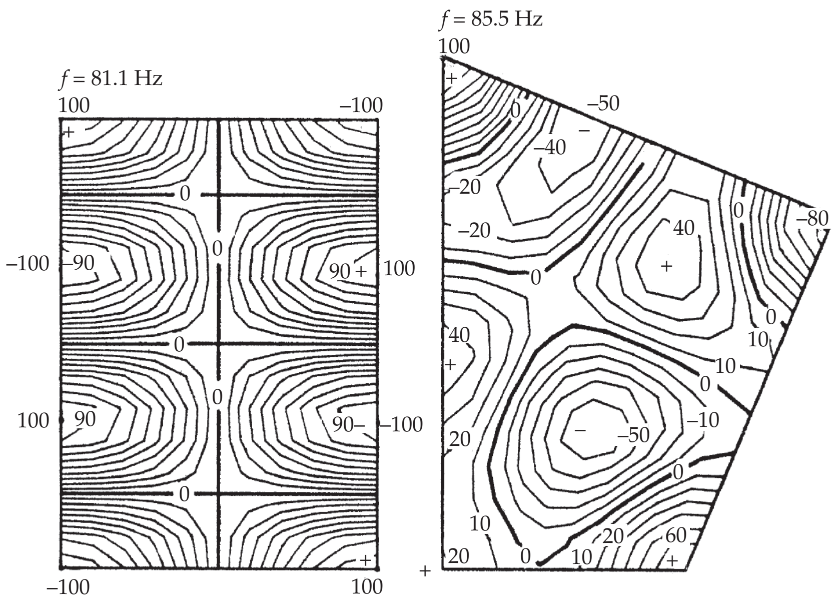


Figure 13-24 The same two-dimensional rectangular room and nonrectangular room of Fig. 13-23 comparing the (1, 3, 0) mode. The sound field is distorted and the frequency is shifted.

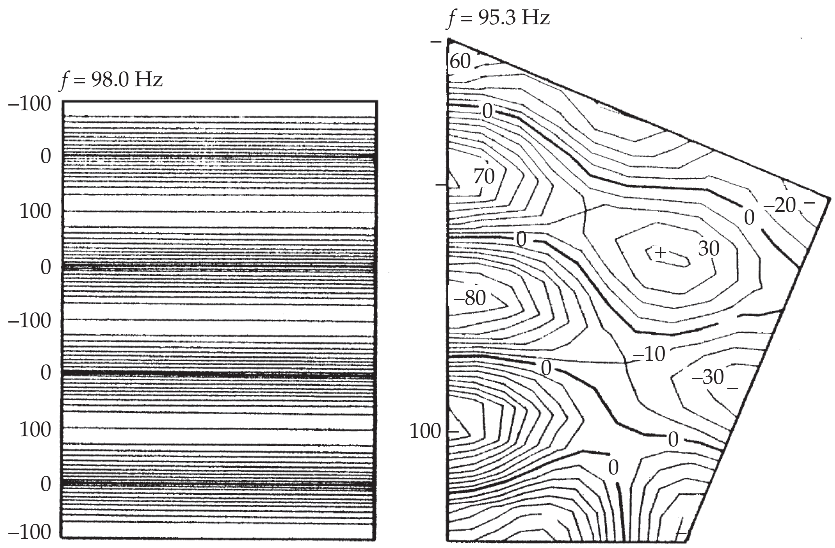


Figure 13-25 The same two-dimensional rectangular room and nonrectangular room of Fig. 13-23 comparing the (0, 4, 0) mode. The sound field is distorted and the frequency is shifted.

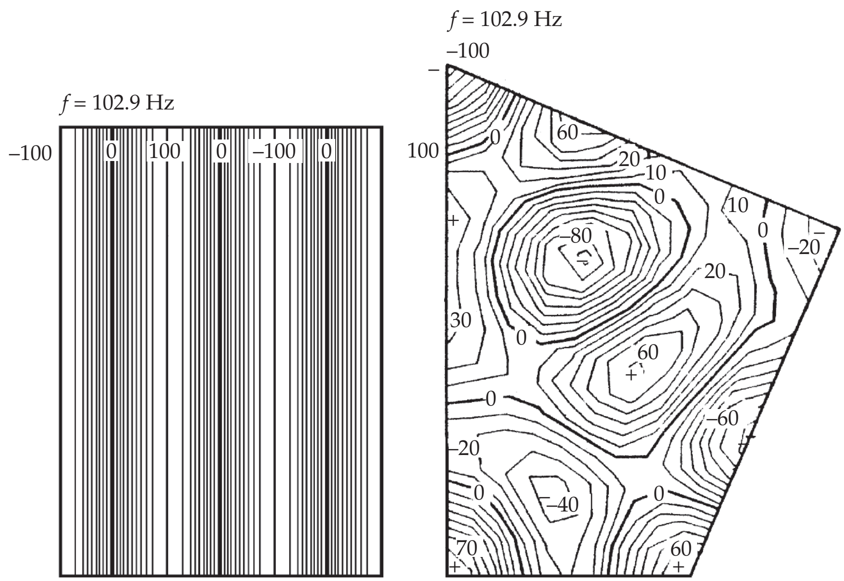


Figure 13-26 The same two-dimensional rectangular room and nonrectangular room of Fig. 13-23 comparing the (3, 0, 0) mode. The sound field is distorted and the frequency is shifted.

Controlling Problem Modes

As noted, a Helmholtz resonator is one possible solution for controlling room modes, achieved by adding absorption at a narrow bandwidth. If the goal is to bring a mode or closely spaced group of modes under control, the placement of the resonator is important. Suppose that the (2, 1, 0) mode of Fig. 13-17 is causing timbral changes in voice quality and it is necessary to introduce narrow-band absorption at the (2, 1, 0) frequency. For maximum effectiveness, Helmholtz resonators should be placed in areas of high modal sound pressure for the tuned frequency. If the Helmholtz resonator were placed at a pressure node (zero pressure), it would have little effect. Placed at one of the antinodes (pressure peaks), it would have high interaction with the (2, 1, 0) mode. Therefore, any corner would be acceptable, as would the pressure-peak locations on the walls.

Building a resonator with very sharp tuning (high Q) can be demanding. The flexing of wooden boxes introduces losses that lower the Q. To attain a truly high-Q resonator with sharp tuning, the cavity must be made of concrete, ceramic, or other hard, nonyielding material, but fitted with some means of varying the resonance frequency.

Sound absorption at the lowest octave or two of the audible spectrum is often difficult to achieve. Bass traps are commonly used in recording studio control rooms to reduce room modes at bass frequencies. Large trap depths are required for the absorption of very low bass frequencies. Unused spaces above control room ceilings and between inner walls and outer shells are often used for trap space.

Simplified Axial-Mode Analysis

To summarize the axial-mode analysis, the method is applied to an example room. The dimensions of the room are 28 × 16 × 10 ft. The 28-ft length resonates at 565/28 = 20.2 Hz, the two side walls 16 ft apart resonate at 565/16 = 35.3 Hz, and the floor-ceiling combination resonates at 565/10 = 56.5 Hz. These three axial resonances and the series of multiples for each are plotted in Fig. 13-27. There are 27 axial resonance frequencies below 300 Hz. For this exercise, the weaker tangential and oblique modes are neglected.

Because many timbral defects are traceable to axial modes, their spacings will be examined in detail. Table 13-6 presents a convenient form for this simplified analysis of

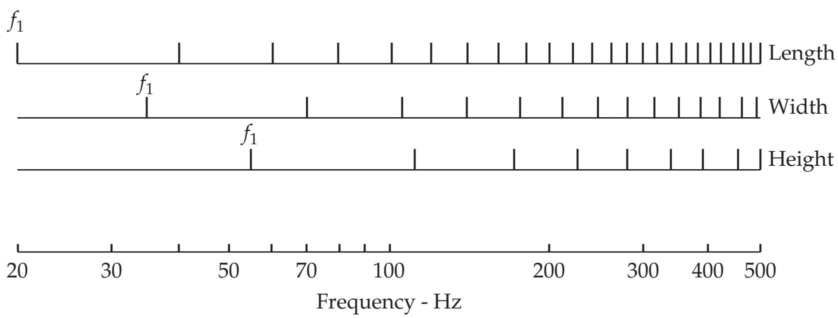


Figure 13-27 The axial mode frequencies and multiples of a room measuring 28 × 16 × 10 ft.

Room Dimensions = 28.0 × 16.0 × 10.0 ft					
	Axial Mode Resonances			Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
	Length $L = 28.0$ ft $f_1 = 565/L$ (Hz)	Width $W = 16.0$ ft $f_1 = 565/W$ (Hz)	Height $H = 10.0$ ft $f_1 = 565/H$ (Hz)		
f_1	20.2	35.3	56.5	20.2	15.1
f_2	40.4	70.6	113.0	35.3	5.1
f_3	60.5	105.9	169.5	40.4	16.1
f_4	80.7	141.3	226.0	56.5	4.0
f_5	100.9	176.6	282.5	60.5	10.1
f_6	121.1	211.9	339.0	70.6	10.1
f_7	141.3	247.2		80.7	20.2
f_8	161.4	282.5		100.9	5.0
f_9	181.6	317.8		105.9	7.1
f_{10}	201.8			113.0	8.1
f_{11}	222.0			121.1	20.2
f_{12}	242.1			141.3	0
f_{13}	262.3			141.3	20.1
f_{14}	282.5			161.4	8.1
f_{15}	302.7			169.5	7.1
				176.6	5.0
				181.6	20.2
				201.8	10.1
				211.9	10.1
				222.0	4.0
				226.0	16.1
				242.1	5.1
				247.2	15.1
				262.3	20.2
				282.5	0
				282.5	0
				282.5	20.2
				302.7	

Mean axial mode spacing: 10.5 Hz.
Standard deviation: 6.9 Hz.

Table 13-6 Simplified Analysis of Axial Modes axial modes. The resonance frequencies from the *L*, *W*, and *H* columns are arranged in ascending order in the fourth column. This makes it easy to examine the critical factor of axial mode spacing.

The L-f_7 resonance at 141.3 Hz coincides with the W-f_4 resonance also at 141.3 Hz. This means that these two axial modes can act together to create a potential response deviation at that frequency. This coincidence is also separated by 20 Hz from neighboring modes. At 282.5 Hz we see a triple degeneracy of L-f_{14}, W-f_8, and H-f_5 modes which, together, would seem to be a troublesome source of timbral defects. They are also separated from neighbors by 20 Hz. If this was a proposed room design, the room dimensions should be re-evaluated for a better modal response. In an existing room, a Helmholtz resonator, sharply tuned and properly located, offers a possible solution.

Key Points

- Acoustical resonances, generally known as normal modes or standing waves, exist naturally in enclosures. They yield unequal energy distribution throughout the enclosure space, particularly at low frequencies in most rooms.
- A pipe closed at both ends models the resonant condition between opposing surfaces in a room. Longitudinal standing waves are created at the natural modal frequency and its multiples. The resonant frequencies of the pipe are determined by the length of the pipe.
- At any point, the particle displacement is exactly out of phase with the sound-pressure level; there is a displacement node at every pressure antinode, and a displacement antinode at every pressure node. For example, particle displacement is always zero at both ends of a closed pipe.
- Small rooms whose dimensions are comparable to the wavelength of audible sound may have the problem of excessive separation between modes.
- As frequency is increased, the number of modes greatly increases. In most rooms, above 300 Hz, average mode spacing becomes so small that room response tends to become smoother.
- The room-mode equation uses the relative dimensions of a room to determine the modal response and hence the suitability of the dimensions. If a rectangular room has two or three dimensions that are equal or a multiple of each other, coincident modal frequencies will result.
- Coincident modal frequencies, called degenerate frequencies, will result in poor room frequency response and unequal energy distribution with a large separation between modes. Modal response can be greatly improved through careful selection of room dimension ratios.
- Axial modes comprise two waves traveling in opposite directions, moving parallel to one axis, and striking only two walls. Axial modes make the most prominent contribution to the acoustical characteristics of a space. Because there are three axes to a rectangular room, there are three fundamental axial frequencies, each with its own series of modes.
- Tangential modes comprise four waves that reflect from four walls and move parallel to two walls. Tangential modes have only half the energy of axial

modes, yet their effect on room acoustics can be significant. Each tangential mode has its series of modes.

- Oblique modes comprise eight waves reflecting from all six walls of an enclosure. Oblique modes, having only one-fourth the energy of axial modes, are less prominent than the other two.
- Because of room modes, a room's low-frequency response can vary greatly at different positions in the room; this is true for both sound sources and receivers placed in the room.
- To be effective in absorbing a given mode, absorbing material must be located on surfaces where the modal pressure is high. For example, carpet on the floor has no effect on horizontal axial modes. Tangential and oblique modes are associated with more surface reflections than axial modes; thus more treatment locations are possible.
- Axial, tangential, and oblique modes decay at different rates. Each reverberatory decay by octave bands thus involves an average of the decay of many modes.
- The mode bandwidth of a typical room is about 5 Hz. This means that adjacent modes tend to overlap for rooms with a short reverberation time, which is desirable.
- The modal density at a given bandwidth increases with frequency. Similarly, modal density increases as room size increases.
- To avoid timbral defects, modes should not be widely separated from one another, but neither should they coincide with each other.
- A room's dimension ratio that falls in the Bolt area will generally yield an acceptable low-frequency response as far as distribution of axial-mode frequencies is concerned.
- With Bonello's criterion, each 1/3 octave should have more modes than the preceding one, or at least the same number. Modal coincidences are not tolerated unless there are at least 5 modes in that band.
- Splaying one or two walls of a room does not eliminate modal problems, although it might shift them slightly in the room and produce nominally better diffusion.
- Predicting modal response with certainty is very difficult. Working from general guidelines and recommendations, the modal response of each room must be considered on a case-by-case basis.

CHAPTER 14
Schroeder Diffusers

After remarkable insight and much experimentation, Manfred R. Schroeder developed a particularly efficient type of diffuser. This quadratic residue diffuser (QRD) is a construction that uses a series of wells of different depths and constant width, separated by thin dividers. The relative depth of each well is calculated using one of several number sequences that optimize diffusion in a plane transverse to the alignment of the wells. The sequence can be repeated through multiple periods to extend the size of the diffuser. As with any reflection phase-grating diffuser, the maximum frequency for diffusion is determined by the well width, and the minimum frequency for diffusion is determined by the well depth.

Experimentation

Using number theory, Schroeder hypothesized that a surface with grooves arranged in a certain way would efficiently diffuse sound to a degree unattainable by many other methods. In particular, he discovered that maximum-length sequences can be used to create pseudo-random noise by application of certain sequences of +1 and −1. The power spectrum (from the Fourier transform) of such noise is essentially flat. A wide and flat power spectrum is related to reflection coefficients and angles, and this indicates that diffusion can be obtained by applying +1 and −1 in a maximum-length sequence. The −1 value suggests a reflection from the bottom of a groove in a wall with a depth of a quarter wavelength. The +1 value suggests a reflection from the wall itself without any groove.

Using 3-cm microwaves, Schroeder tested a piece of sheet metal bent into the shape of Fig. 14-1. This shape followed the binary maximum-length sequence with period length 15:

$$- + + - + - + + + + - - - + -$$

Notably, well depths are one-quarter wavelength.

The resulting reflection pattern, shown in Fig. 14-2A, indicated excellent diffusion. Reflected sound is diffused over a wide angle. In contrast, a metal sheet with well depth of one-half wavelength yielded strong specular reflection, but little diffusion of energy, as shown in Fig. 14-2B.

When a narrow strip of metal covered just one of the (−) grooves, the reflection pattern showed essentially specular reflection of most of the energy back toward the source. In other words, covering one of the slots greatly degraded the favorable diffusion properties. The particular sequence of well depths, as predicted by theory, is crucial in providing efficient diffusion.

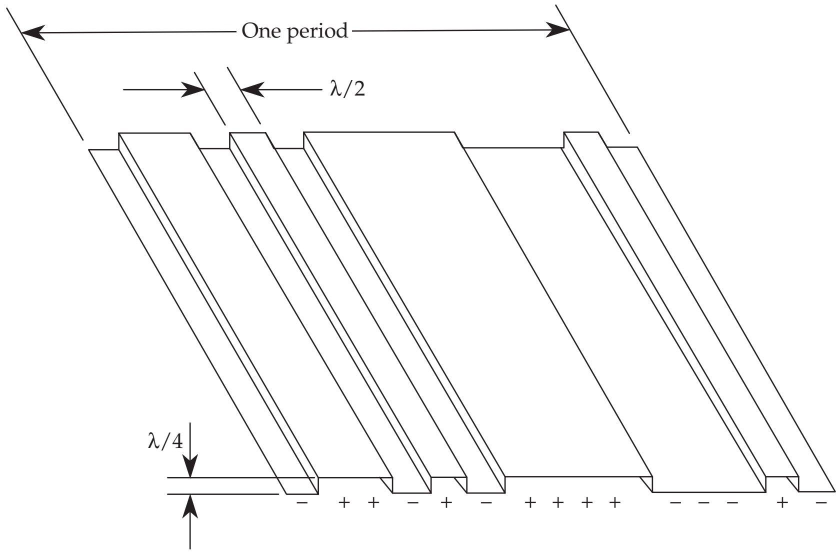


FIGURE 14-1 An experimental metal sheet folded to conform to a maximum-length sequence, used by Schroeder to check the diffusion of 3-cm radio waves. The well depths are all one-quarter wavelength.

Reflection Phase-Grating Diffusers

Schroeder diffusers, otherwise known as reflection phase-grating diffusers, provide excellent performance. The diffuse reflection pattern of Fig. 14-2A is superior to many other sound diffusers. Adjustment of room proportions, splaying of walls, the use of semispherical, polycylindrical, triangular, cubical, and rectangular geometrical protru-

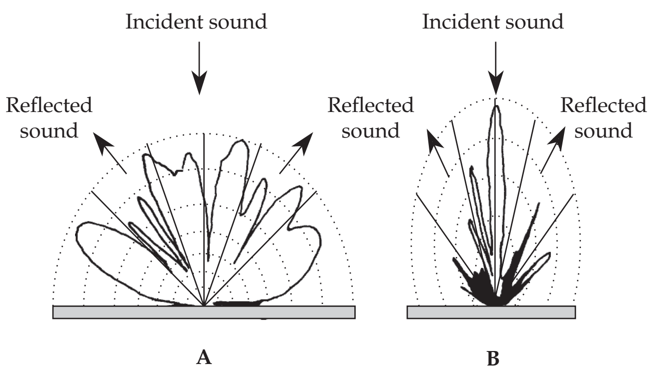


FIGURE 14-2 Diffusion is profoundly affected by well depth. (A) When the diffuser well depths of Fig. 14-1 are one-quarter wavelength, a very favorable diffusion pattern is obtained. (B) When the wells are one-half wavelength, almost pure specular reflection results, the same as for a flat sheet of metal. (*Schroeder.*)

sions, and the distribution of absorbing material are helpful in promoting diffusion but typically are not as efficient as the Schroeder diffuser.

Diffraction phase-grating diffusers do have limitations. Because of the one-quarter wavelength groove depth requirement of the binary maximum-length diffuser, the sound-diffusing properties of the surface depend on the wavelength of the incident sound. Experience has indicated that reasonable diffusion results over a band of plus or minus one-half octave of the frequency around which the diffuser is designed. For example, consider a maximum-length sequence diffuser with a sequence length of 15. A design frequency of 1 kHz gives a one-half wavelength groove depth of 7.8 in, and a one-quarter wavelength groove depth of 3.9 in. A single period of this diffuser would be about 5-ft wide and effective from about 700 to 1,400 Hz. Many such units would be required to provide diffusion over a reasonable portion of the audible band. Even so, diffraction phase-grating diffusers provide good results.

Quadratic Residue Diffusers

Schroeder reasoned that an incident sound wave falling on a reflection phase grating would diffuse sound almost uniformly in all directions. The phase shifts (or time shifts) can be obtained from an array of wells of depths determined by a quadratic residue sequence. This is the theoretical basis for the quadratic residue diffuser (QRD). The maximum well depth is determined by the longest wavelength to be diffused. The well width is about a half wavelength at the shortest wavelength to be scattered. The depths of the sequence of wells are determined by the statement:

$$\text{Well depth proportionality factor} = n^2 \text{ modulo } p \qquad (14\text{-}1)$$

where n = integer ≥ 0
p = prime number

A prime number is a positive integer that is divisible without a remainder by only 1 and the prime number itself. Examples of prime numbers are 5, 7, 11, 13, and so on. The modulo refers to the residue or remainder. For example, inserting $n = 5$ and $p = 11$ into the above equation gives 25 modulo 11. The modulo 11 means that 11 is subtracted from 25 until the significant residue is left. In other words, 11 is subtracted from 25 twice and the residue 3 is the well depth factor. As another example, for $n = 8$, $p = 11$, the result is: (64 modulo 11) = 9. In a similar fashion, the well depth factors for all the wells in a QRD panel can be calculated.

A value is selected for prime number p, and the well depth factor for each corresponding integer n value is obtained. In Fig. 14-3, quadratic residue sequences are listed for the prime numbers 5, 7, 11, 13, 17, 19, and 23, a separate column for each. To check the above example for $n = 5$ and $p = 11$, enter the p column marked 11, run down to $n = 5$, and find 3, which checks the previous computation. The numbers in each column of Fig. 14-3 are proportional to well depths of different quadratic residue diffusers. At the bottom of each column of Fig. 14-3 is a sketch of a quadratic residue diffuser profile with well depths proportional to the numbers in the sequence. The broken lines indicate thin dividers between the wells. As an example, Fig. 14-4 shows a model for a quadratic residue reflection phase-grating diffuser for $p = 17$. In this example, the sequence period is repeated twice. Also, note the symmetry of the sequence.

Quadratic residue sequences

n	*p* = 5	7	11	13	17	19	23
0	0	0	0	0	0	0	0
1	1	1	1	1	1	1	1
2	4	4	4	4	4	4	4
3	4	2	9	9	9	9	9
4	1	2	5	3	16	16	16
5	0	4	3	12	8	6	2
6		1	3	10	2	17	13
7		0	5	10	15	11	3
8			9	12	13	7	18
9			4	3	13	5	12
10			1	9	15	5	8
11			0	4	2	7	6
12				1	8	11	6
13				0	16	17	8
14					9	6	12
15					4	16	18
16					1	9	3
17					0	4	13
18						1	2
19						0	16
20							9
21							4
22							1
23							0

Well depth proportionality factor = n^2 modulo p

where n = integer ≥ 0

p = prime number

Figure 14-3 Quadratic residue sequences for prime numbers from 5 to 23. The diffuser profiles at the foot of each column have well depths that are proportional to the sequence of numbers above.

Thin and rigid separators between the wells, usually metallic, are commonly used to maintain the acoustical integrity of each well. Without separators the effectiveness of the diffuser is decreased. In the absence of dividers, the stepped phase shifts for sound arriving at angles other than the perpendicular tend to be confused.

Primitive Root Diffusers

Primitive root diffusers are also based on number theory sequences. However, they use a sequence that is different from the quadratic residue sequence:

$$\text{Well depth proportionality factor} = g^n \text{ modulo } p \qquad (14\text{-}2)$$

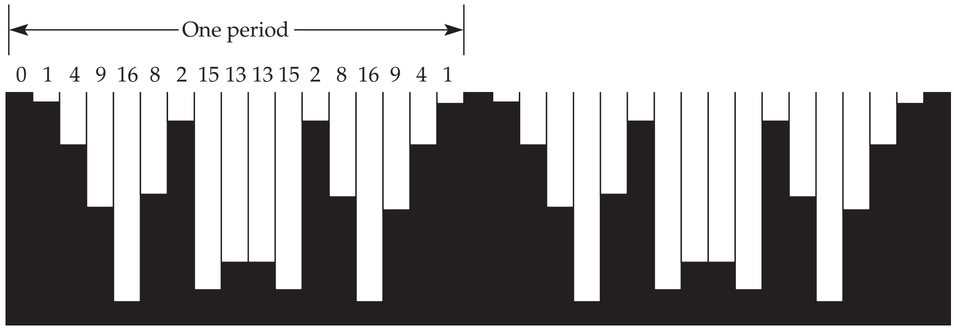


Figure 14-4 A typical quadratic residue diffuser based on the prime number 17 column of Fig. 14-3. The depths of the wells are proportional to the sequence of numbers in the prime 17 column. Two periods are shown illustrating how adjacent periods are fitted together.

where g = least primitive root of p
n = integer ≥ 0
p = prime number

Figure 14-5 shows primitive root sequences for six different combinations of g and p. The sketches at the bottom of each column show that primitive root diffusers are not symmetrical like those of the quadratic residue diffusers. In most cases this is a disadvantage, but in some cases it is an advantage. There is an acoustical limitation with primitive root diffusers in that the specular mode is not suppressed as well as it is in quadratic residue diffusers. Commercial development has largely utilized quadratic residue sequences.

Performance of Diffraction-Grating Diffusers

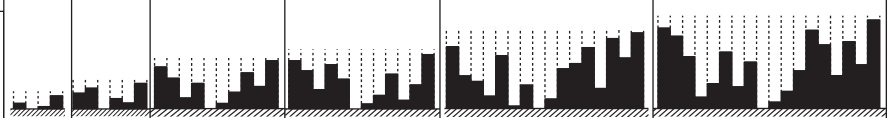
In designing an audio space, an acoustician has three building blocks: absorption, reflection, and diffusion. The effects on incident sound of these three physical principles are compared in Fig. 14-6. Figure 14-6A shows a sound impinging on the surface of a sound absorber. Energy is largely absorbed, but a small fraction is reflected. The temporal response shows a greatly attenuated reflection from the surface of the absorber.

The same sound falling on a hard, reflective surface, as shown in Fig. 14-6B, yields a reflection of almost the same intensity as the sound falling on the surface itself, just reduced slightly by losses at the reflective surface. The polar plot shows the energy concentrated around the angle of reflection. The width of the response in the polar plot is a function of the wavelength of the sound and the size of the reflecting surface.

A sound falling on a diffuser, such as a quadratic residue diffuser, as shown in Fig. 14-6C, is diffracted throughout the hemidisc. The diffused energy falls off exponentially. The polar plot shows energy spread more or less equally throughout 180° but somewhat reduced at grazing angles. The specular reflection mode is suppressed.

Figure 14-7 shows the uniformity of the angular distribution of scattered energy from a diffraction-grating diffuser through a wide frequency range. The left column shows polar distribution of octave bands of energy centered from 250 to 8,000 Hz, Primitive root sequences

n	$p = 5$, $g = 2$	$p = 7$, $g = 3$	$p = 11$, $g = 2$	$p = 13$, $g = 2$	$p = 17$, $g = 3$	$p = 19$, $g = 2$
1	2	3	2	2	3	2
2	4	2	4	4	9	4
3	3	6	8	3	10	8
4	1	4	5	3	13	16
5		5	10	6	5	13
6		1	9	12	15	7
7			7	10	11	14
8			3	9	16	9
9			6	5	14	18
10			1	10	8	17
11				7	7	15
12				1	4	11
13					12	3
14					2	6
15					6	12
16					1	5
17						10
18						1



Well depth proportionality factor = g^n modulo p

where g = least primitive root of p

n = integer ≥ 0

p = prime number

FIGURE 14-5 Primitive root sequences for six combinations of prime number and least primitive roots. The diffuser profiles at the foot of each column have well depths that are proportional to the sequence of numbers above. Note that these diffusers are not symmetrical as with quadratic residue diffusers.

a span of five octaves. The right column shows the effect for sound incident at 45° for the same frequencies. The results for all angles of incident sound for the lowest frequency are dependent on the diffuser's well depth. The upper frequency is directly proportional to the number of wells per period and inversely proportional to the well width. For comparison, the diffusion of a flat panel is shown by a light line. The uniformity of spatial diffusion is determined by the length of the diffuser period. Good broad-bandwidth, wide-angle diffusion requires a large period with a large number of deep,

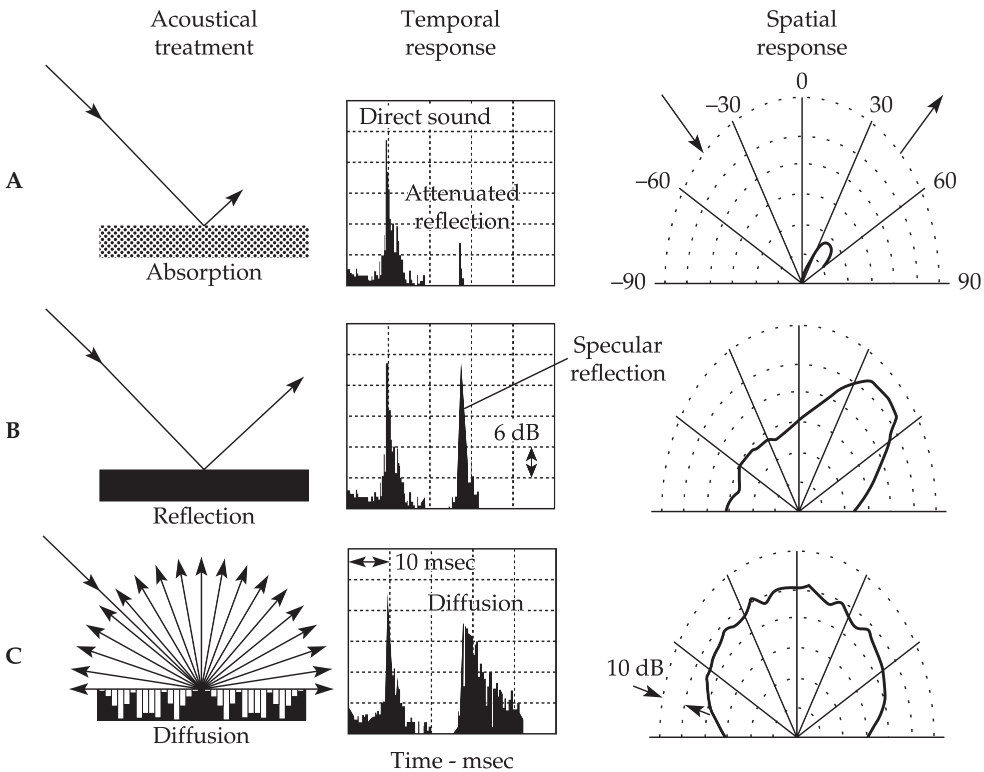


Figure 14-6 A comparison of the physical characteristics of three treatments, showing temporal response and spatial response. (A) Absorption. (B) Reflection. (C) Diffusion. (*D'Antonio.*)

narrow wells. For example, a diffuser might have 43 wells with a width of only 1.1 in, and a maximum well depth of 16 in.

These polar plots are smoothed by averaging over octave bands. Comparable polar plots, for single frequencies based on a far-field diffraction theory, show a host of tightly packed lobes that have little practical significance. Near-field Kirchhoff diffraction theory shows less lobing.

Figure 14-8 compares the reflective return from a flat panel with that from a quadratic residue diffuser. The large peak to the left is the direct sound. The second large peak is the specular reflection from a flat panel. Note that the energy of this sharp specular panel reflection is only a few dB below that of the incident sound. The diffused energy from the quadratic residue diffuser is significantly spread out in time. Most importantly, and revealed in the polar diagrams of Fig. 14-7, the grating diffracts sound throughout 180°, not only in the specular direction like a flat panel.

Reflection Phase-Grating Diffuser Applications

The acoustical treatment of large and small spaces is well served by phase-grating diffusers. A large space is one whose normal modal frequencies are so closely spaced as to avoid low-frequency resonance problems. This includes concert halls, auditoriums,

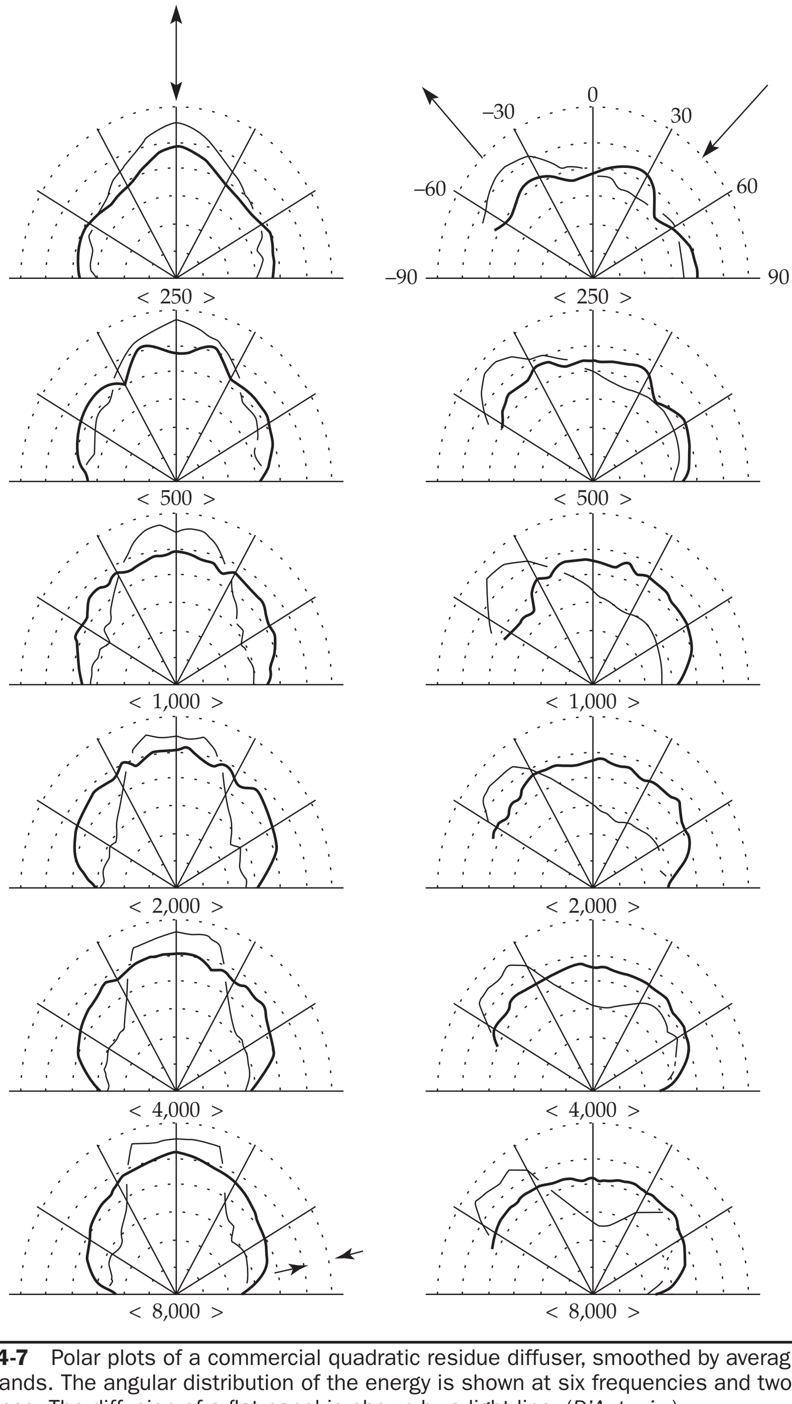


Figure 14-7 Polar plots of a commercial quadratic residue diffuser, smoothed by averaging over octave bands. The angular distribution of the energy is shown at six frequencies and two angles of incidence. The diffusion of a flat panel is shown by a light line. (*D'Antonio.*)

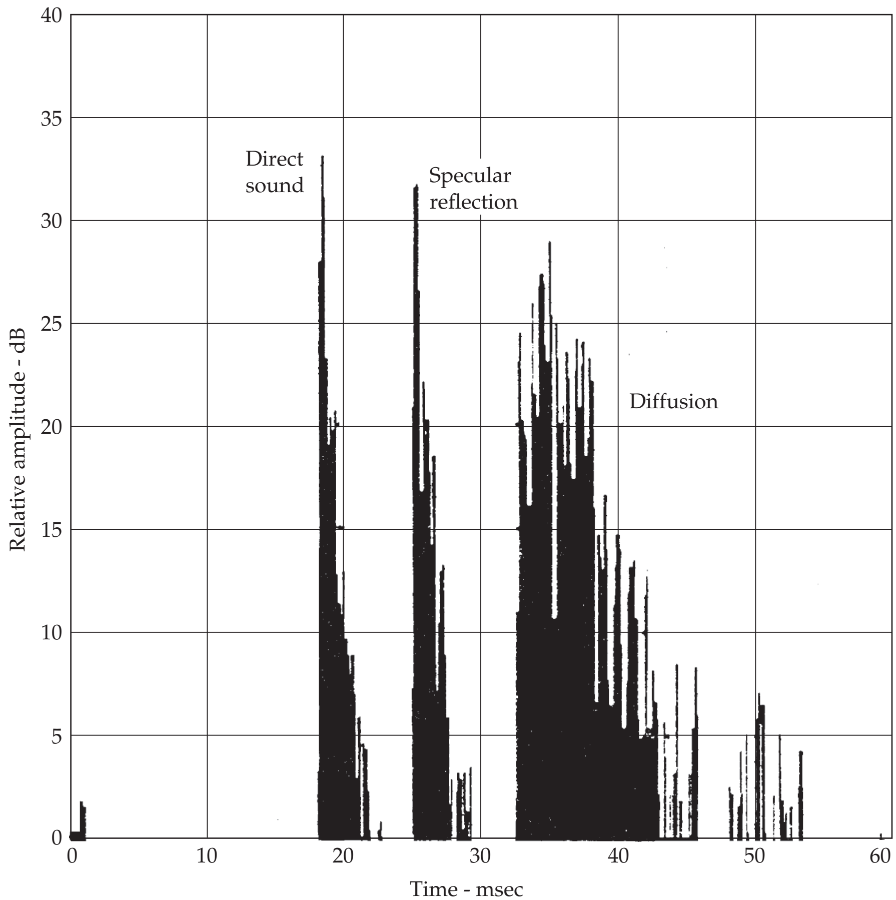


FIGURE 14-8 An energy-time plot comparing a specular reflection from a flat panel to the energy diffused from a quadratic residue diffuser. The peak energy from a diffusing surface is somewhat lower than that of the flat panel but it is spread out in time. (*D'Antonio.*)

and many places of worship. The sound quality of a concert hall is greatly influenced by the reflections from side walls. Side walls provide necessary lateral reflections, and diffraction-grating diffusers placed along the center of a hall at ceiling level can diffuse sound from the stage laterally to people in the seats. Troublesome specular reflections can be controlled by strategic placement of diffusers.

In places of worship, there is often an acoustical design conflict between the intelligibility of the spoken word and conditions for enjoyment of music. The rear wall is often the source of reflections that create disturbing echoes. To make this wall absorbent is often detrimental to music conditions. Making the rear wall diffusive, however, minimizes the echo problem while at the same time conserving useful music and speech energy. Music directors are often faced with the problem of singers or instrumental players not hearing each other well. Surrounding the music group with reflection phase-grating diffusers both conserves music energy and disperses it to achieve ensemble between musicians.

Difficult small-room acoustics are also helped by diffraction-grating diffusing elements. The need to cant walls and distribute absorption material to achieve sound diffusion is relaxed by the proper use of grating diffusers. For example, through proper design, it is possible to obtain acceptable voice recordings from small announce booths because diffusing elements create the sound of a larger room.

Numerous diffuser products incorporating quadratic residue theory are commercially available. Figure 14-9 shows a QRD-1911 diffuser above and two QRD-4311 diffusers below. In the model numbers, the "19" indicates that it is built on prime 19 and the "11" specifies well widths of 1.1 in. (The sequence of numbers in the prime 19 column of Fig. 14-3 specify the proportionality factors for well depths of the diffuser.) The QRD-4311 in the lower portion of Fig. 14-9 is based on prime 43 with well widths also of 1.1 in. (For practical reasons, the columns of Fig. 14-3 stop at prime 23; primes between 23 and 43 are 29, 31, 37, and 41.)

This particular cluster of quadratic residue diffusers offers excellent diffusion in the horizontal hemidisc, as shown in Fig. 14-10A. The specular reflection from the face of the diffuser is shown in Fig. 14-10B. The vertical wells of the QRD-4311 scatter sound horizontally and the horizontal wells of the QRD-1911 scatter sound vertically. Together they produce a virtual hemisphere of diffusion. These two commercial products are manufactured by RPG Acoustical Systems.

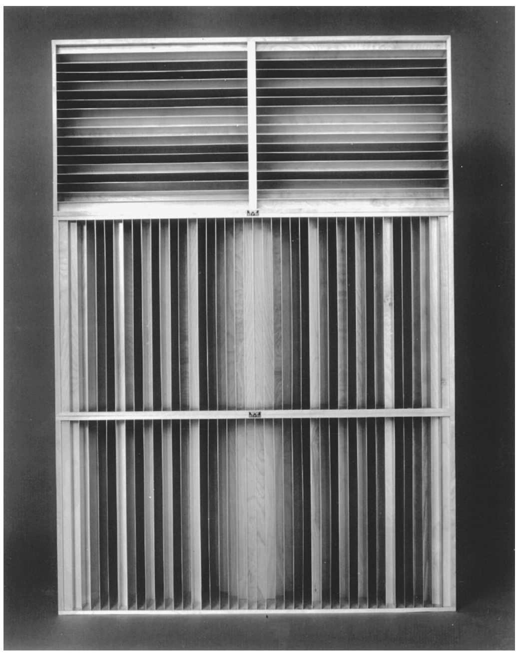


FIGURE 14-9 A cluster of commercial quadratic residue diffusers with a single QRD-1911 unit mounted above, and two QRD-4311 units mounted below. The hemidisc of diffusion for the lower unit is horizontal, that of the upper unit is vertical. (*RPG Acoustical Systems.*)

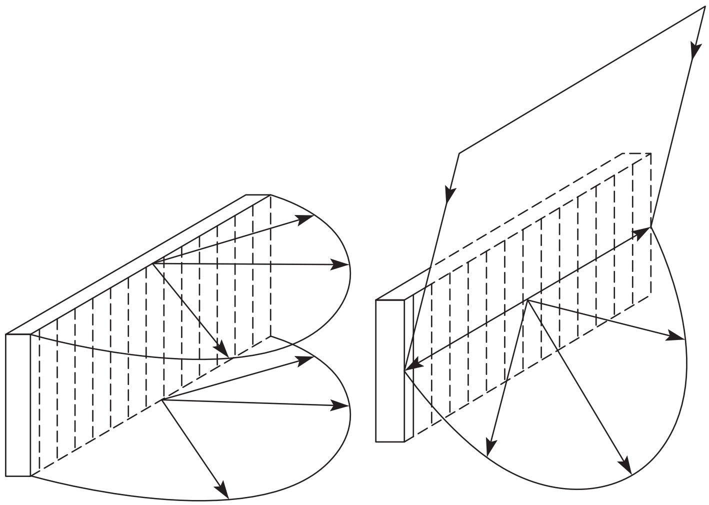


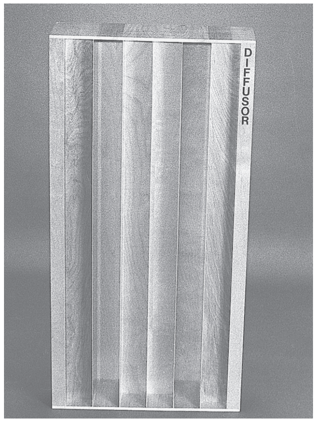
Figure 14-10 Sound is diffused over a hemidisc geometry. (A) A one-dimensional quadratic residue diffuser scatters sound in a hemidisc. (B) This hemidisc may be one specularly directed by the orientation of the source with respect to the diffuser.

Figure 14-11A shows a QRD-734 diffuser; this is a 2- × 4-ft model suitable for use in suspended ceiling T-frames as well as other applications. Figure 14-11B shows a Triffusor which incorporates reflective, absorptive, and diffusive faces. A group of these, set into a wall, offers options in the acoustics of a space by rotating the individual units. Figure 14-11C shows an Abffusor, which combines broadband absorption and diffusion in the same unit. These three commercial products are manufactured by RPG Acoustical Systems.

Flutter Echo

If two opposing reflective surfaces of a room are parallel, there exists the possibility of periodic or near-periodic echoes called flutter echoes. This applies to either horizontal or vertical modes. Such successive, repetitive reflections, equally spaced in time, can be audibly disturbing and can degrade the intelligibility of speech and the quality of music. When the time between echoes is short, they can produce a perception of a pitch or timbral change in music. If the time between echoes is greater than about 30 to 50 msec, the periodicity is audible as a distinct flutter echo. This time period is within the Haas fusion zone, and might otherwise be inaudible, but the periodicity of the echo makes it more audible. The lack of ornamentation in modern architecture, and hence poor diffusion, can result in a greater possibility of flutter echoes. Flutter echoes are often associated with specific relative placement of a sound source and listener.

Two opposing, parallel surfaces of a room should never be highly reflective. Flutter echoes can be reduced with careful placement of sound absorbing material on one or both walls. Flutter echo can also be reduced by splaying walls by 5° to 10°. However, splaying is impractical in many cases, and increasing absorption may degrade the acoustical quality of a space. Diffuser wall treatments can reduce reflections by scattering



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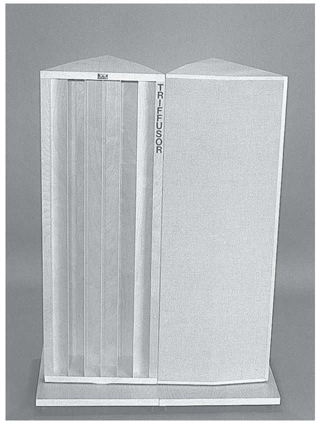

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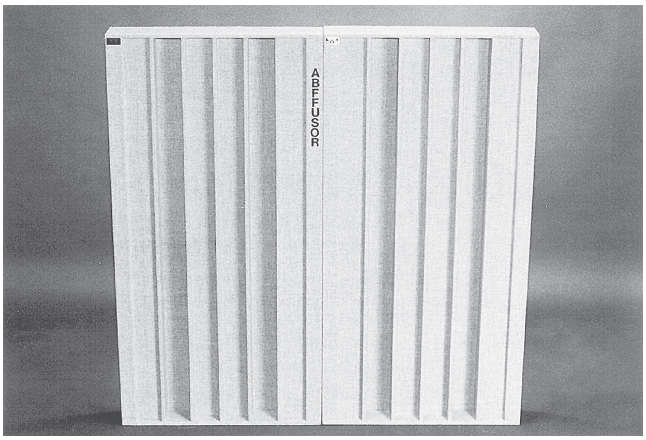

C

Figure 14-11 Three proprietary sound diffusing systems. (A) A broad bandwidth and wide-angle QRD-734 unit. (B) Triffusor having one side absorptive, one diffusive, and one reflective for acoustical variability. (C) Abffusor diffuser/absorber with broad bandwidth. (*RPG Acoustical Systems.*)

sound rather than absorbing it. A Flutterfree, as shown in Fig. 14-12, is a commercial example of an architectural hardwood molding which reduces specular reflection and provides diffusion. The molding, with a width of 4 in and a length of 4 or 8 ft, works as a one-dimensional reflection phase-grating diffuser because of the wells routed into its surface. The depths of the wells follow a prime 7 quadratic residue sequence. These moldings may be affixed to a wall butted together or spaced apart, positioned horizontally or vertically. If they are vertical, spectral reflections are controlled in the horizontal plane and vice versa.

These moldings can also be employed as slats for a Helmholtz slat-type low-frequency absorber. While low-frequency sound is being absorbed by the Helmholtz

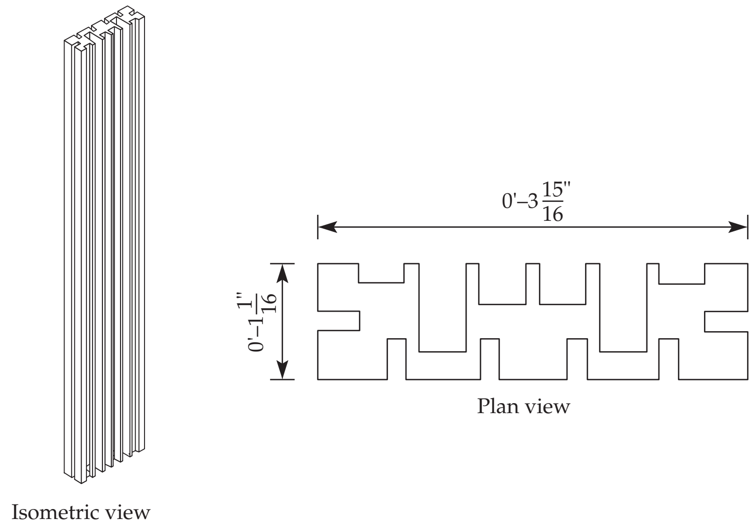

Figure 14-12 The Flutterfree is a nonabsorptive flutter echo control molding. It is a quadratic residue diffuser based on the prime 7. It can also serve as slats in a Helmholtz slat-type low-frequency absorber. (*RPG Acoustical Systems.*)

absorber, the surface of each slat performs as a mid/high-frequency sound diffuser. The Flutterfree is manufactured by RPG Acoustical Systems.

Application of Fractals

Certain production limitations have been encountered in the development of reflection phase-grating diffusers. For example, the low-frequency diffusion limit is determined principally by well depth, and the high-frequency limit is determined principally by well width. Manufacturing constraints may dictate a well-width limit of about 1 in, and a well-depth limit of about 16 in, beyond which the units become diaphragmatic.

To increase the effective bandwidth, the principle of self-similarity has been applied in the form of diffusers employing fractals. For example, a fractal design could create a triple-level unit with a diffuser within a diffuser within a diffuser, as shown in the progressive illustration of Fig. 14-13. A Diffractal is a commercial example of this type of unit. Three sizes of quadratic residue diffusers are required to make up a complete Diffractal. The various diffusers operate analogously to the woofer, midrange, and tweeter in a three-way loudspeaker, operating independently to reproduce a wideband response.

Figure 14-14 shows a DFR-82LM Diffractal with a height of 7 ft, 10 in, a width of 11 ft, and a depth of 3 ft. This unit covers the range of 100 to 5,000 Hz. The low-frequency portion is based on a prime 7 quadratic residue sequence. A mid-frequency Diffractal is embedded at the bottom of each well of the larger unit. The frequency range of each section and the crossover points of these composite units are calculable.

Figure 14-15 shows a larger DFR-83LMH unit with a height of 6 ft, 8 in, a width of 16 ft, and a depth of 3 ft. This is a three-way unit covering a frequency range of 100 Hz to 17 kHz.

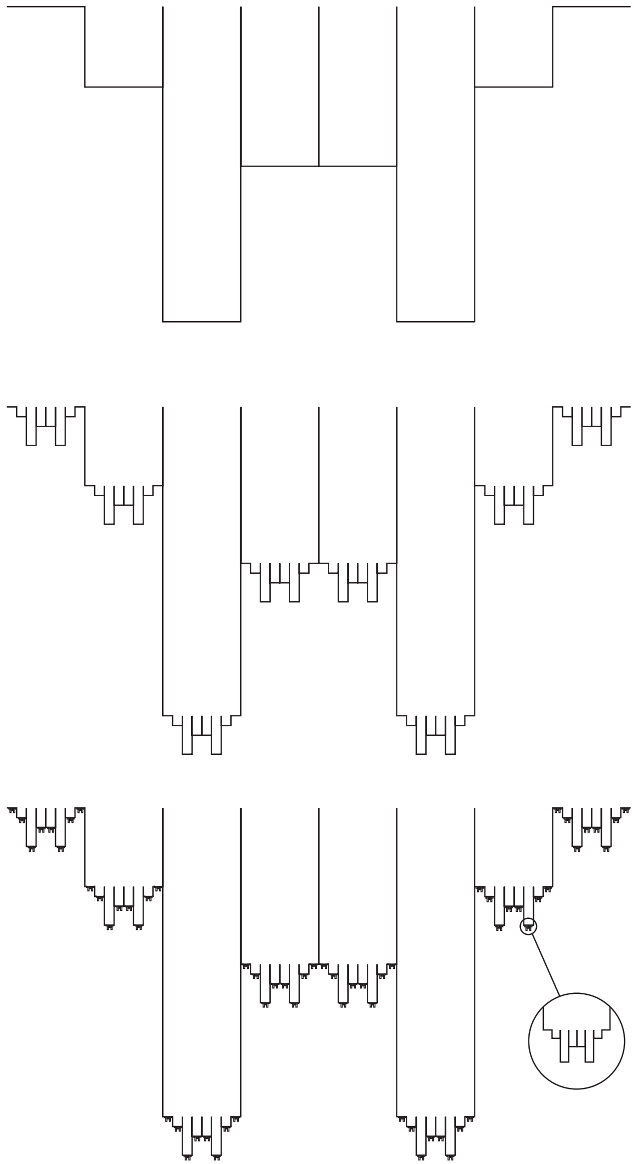

Figure 14-13 Fractal theory can be used to increase the effective bandwidth of diffusers. This essentially allows the design of multiple diffusers within diffusers. In this way, high-frequency fractal diffusers can be nested within low-frequency diffusers.

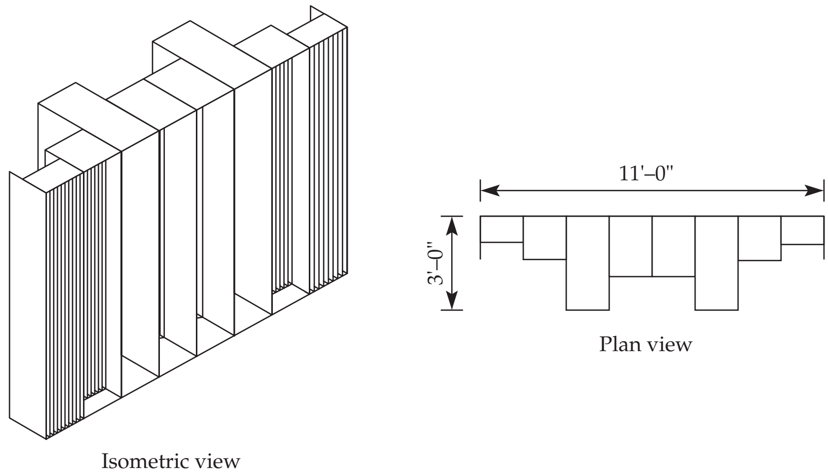

Figure 14-14 The DFR-82LM Diffractal is a two-way wideband diffuser employing a fractal design. It comprises a low-frequency unit with a midrange unit embedded at the bottom of each well, creating a diffuser within a diffuser. (*RPG Acoustical Systems.*)

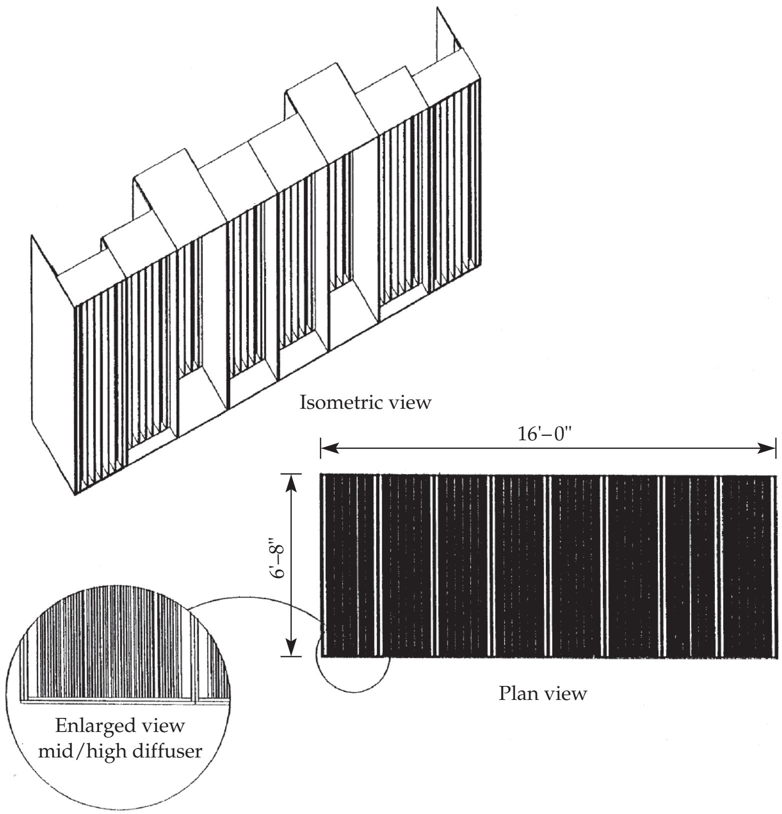

Figure 14-15 The larger DFR-83LMH Diffractal is a three-way wideband diffuser employing a fractal design. Fractals are set in the wells, creating a diffuser, within a diffuser, within a diffuser. (*RPG Acoustical Systems.*)

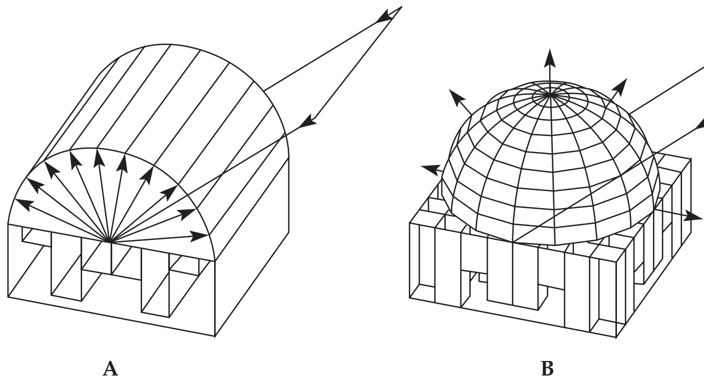

Figure 14-16 Comparison of diffraction patterns. (A) The hemicylindrical form of the one-dimensional quadratic residue diffuser. (B) The hemispherical form of the two-dimensional diffuser. (*RPG Acoustical Systems.*)

The depths of the wells of the low-frequency unit follow a prime 7 quadratic residue sequence. Fractals are set in the wells of fractals that are set in the low-frequency wells. The Diffractal is manufactured by RPG Acoustical Systems.

Diffusion in Three Dimensions

The reflection phase-grating diffusers discussed previously have rows of parallel wells. These can be called one-dimensional units because sound is scattered in a hemidisc, as shown in Fig. 14-16A. There are occasions in which hemispherical coverage is desired, as shown in Fig. 14-16B. The Omniffusor is an example of a commercial unit that provides hemispherical diffusion. It consists of a symmetrical wooden array of 64 square cells, as shown in Fig. 14-17. The depth of these cells is based on the phase-shifted prime 7 quadratic residue number theory sequence.

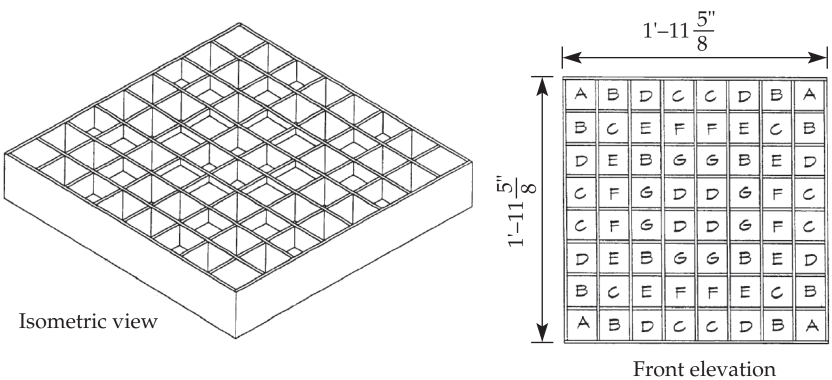

Figure 14-17 The Omniffusor is a two-dimensional unit which diffuses sound in both the horizontal and vertical planes for all angles of incidence. (*RPG Acoustical Systems.*)

Similarly, the FRG Omniffusor consists of an array of 49 square cells based on the two-dimensional phase-shifted quadratic residue number theory. This unit is made of glass-fiber reinforced gypsum. It is light in weight and is therefore useful for application to large surface areas. These units are manufactured by RPG Acoustical Systems.

Diffusing Concrete Blocks

Concrete masonry units (CMU) are widely used in wall construction for their load-bearing ability; also, their mass provides transmission loss for sound isolation. As noted in Chap. 12, acoustical concrete masonry units (ACMU) can also provide low-frequency absorption. The DiffusorBlox system provides load-bearing ability, transmission loss, and low-frequency absorption, as well as sound diffusion. The DiffusorBlox comprises three distinct blocks, all of which nominally measure 8 × 16 × 12 in. A typical block is shown in Fig. 14-18. These concrete blocks are characterized by a surface containing a partial sequence of varying well depths, separated by dividers; an internal five-sided cavity that can accept a glass-fiber insert; an optional rear half-flange for reinforced construction; and an optional low-frequency absorbing slot. Typical walls constructed of DiffusorBlox are illustrated in Fig. 14-19. DiffusorBlox is licensed by RPG Acoustical Systems.

Measuring Diffusion Efficiency

A measure of the effectiveness of a diffuser can be obtained by comparing the sound intensity in the specular direction with the intensity at ±45° of that direction. This can be expressed as:

$$\text{Diffusion coefficient} = \frac{I(\pm 45°)}{I(\text{specular})} \tag{14-3}$$

The diffusion coefficient is 1.0 for a perfect diffuser. This coefficient varies with frequency and is commonly expressed in graphical form. The variation of the diffusion coefficient with frequency for several typical diffuser units is shown in Fig. 14-20. For comparison, diffusion from a flat panel is included as a broken line.

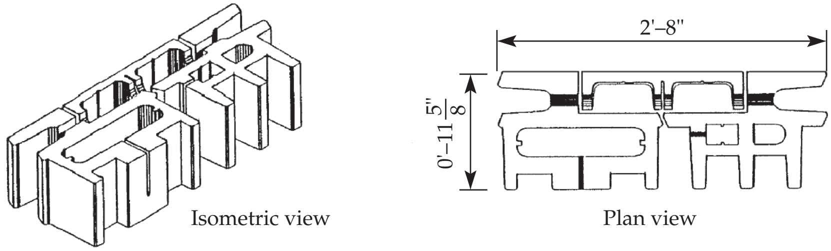

FIGURE 14-18 The DiffusorBlox concrete block offers the transmission loss of a heavy wall, absorption via Helmholtz resonator action, and diffusion through quadratic residue action. The blocks are formed on standard block machines using licensed molds. (*RPG Acoustical Systems.*)

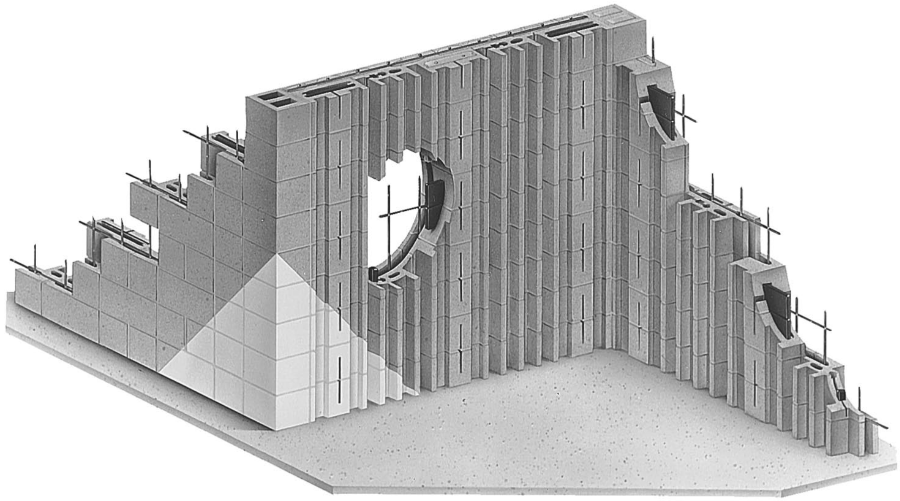

Figure 14-19 Typical wall configurations using DiffusorBlox. (*RPG Acoustical Systems.*)

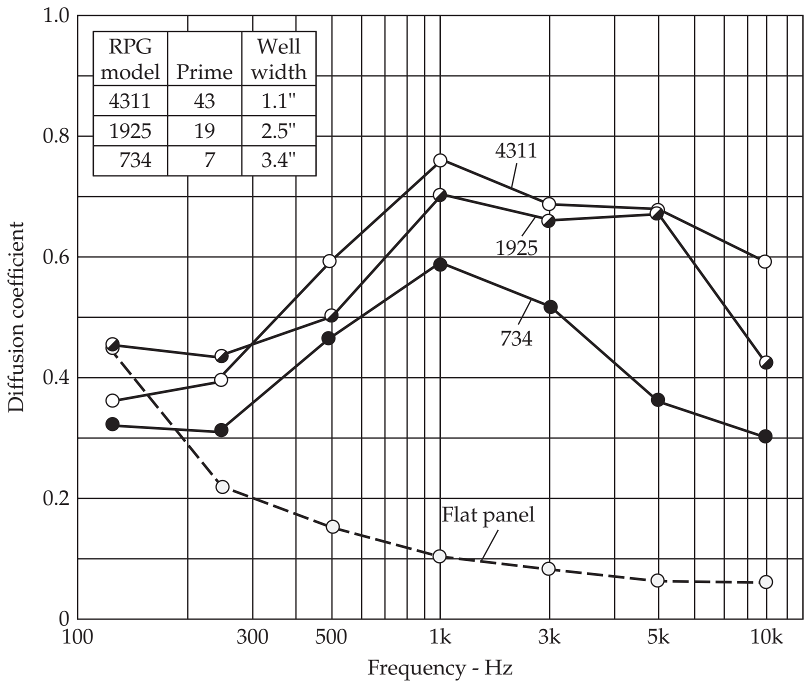

Figure 14-20 Comparison of diffusion coefficients with respect to frequency of three commercial diffusion units (QRD-4311, -1925, -734), and a flat panel. (*RPG Acoustical Systems.*)

These measurements were all made under reflection-free conditions on sample areas of 64 ft^2 and using the time-delay-spectrometry technique.

The number of wells and well widths affect the performance of diffuser units. For example, a QRD-4311 diffuser has relatively deep well depths and narrow well widths that are feasible from a manufacturing standpoint. It yields high diffusion coefficients over a wide frequency range, as shown in Fig. 14-20. For comparison, the QRD-1925 and QRD-734 units, built with the primes 19 and 7 and well widths of 2.5 and 3.4 in, are also shown in Fig. 14-20. The diffusion performance of these, while good, is somewhat inferior to the QRD-4311.

Several methods have been devised to determine how uniformly a surface diffuses sound. One such method is described in *AES-4id-2001* (*s2008*) *AES Information Document for Room Acoustics and Sound Reinforcement Systems—Characterization and Measurement of Surface Scattering Uniformity.* This document describes guidelines for evaluating diffusion surfaces and estimating diffusion coefficients using measurements or predictions of scattered polar responses. The method measures the 1/3-octave polar responses of a diffusing surface and compares them to diffusion of a flat reflective reference surface. Measurements are taken for five angles of incidence: 0°, ±30°, and ±60°. The five directional coefficients are averaged to obtain the incidence diffusion coefficient of the surface. The method described in *ISO 17497-2:2012 Acoustics—Sound-Scattering Properties of Surfaces—Part 2: Measurement of the Directional Diffusion Coefficient in a Free Field* similarly can be used to obtain diffusion scattering coefficients; this standard is derived from the AES document. Some manufacturers employ these methods and publish results that quantify the performance of their diffusers.

Comparison of Gratings with Conventional Approaches

Figure 14-21 compares the diffusing properties of five types of surfaces: (A) flat panel, (B) flat panel with distributed absorption, (C) monocylinder, (D) bicylinder, and (E) quadratic residue diffuser. The left column is for sound at 0° incidence and the right column is for sound at 45° incidence. The "fore-and-aft" scale is diffraction from 90° through 0° to –90°. The horizontal frequency scale ranges from 1 to 10 kHz. These three-dimensional plots provide a comprehensive evaluation of diffusion properties.

Regarding these measurements, Peter D'Antonio has made these notes: The first six energy-frequency curves contain artifacts of the measurement process which should be disregarded because they are not in the anechoic condition. For 0° incidence, the specular properties of the flat panel with distributed absorption are quite evident by the pronounced peak at 0°, the specular angle. The good spatial diffusion of the monocylinder is illustrated by the relatively constant energy response from 90° to –90°. The bicylinder shows two closely spaced peaks in the time response. Although the spatial diffusion appears good, there is appreciable equal-spaced comb filtering and broadband high-frequency attenuation. This accounts, in part, for the relatively poor performance of cylindrical diffusers. Quadratic residue diffusers maintain good spatial diffusion even at 45° incidence. The dense notching is uniformly distributed across the frequency spectrum and energy is relatively constant with scattering angle, indicating good performance.

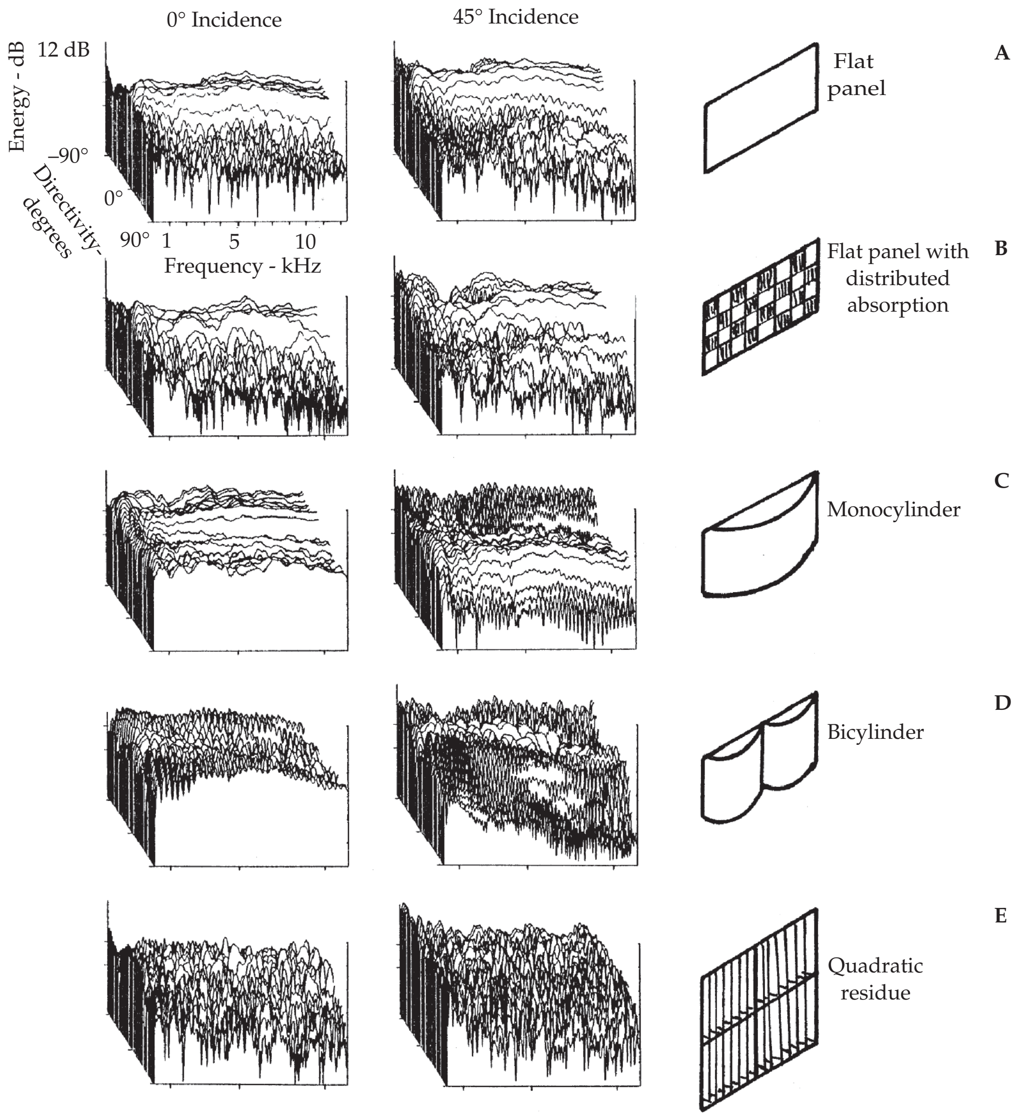

FIGURE 14-21 Energy-frequency-directivity plots comparing diffuser surfaces. (A) Flat panel. (B) Flat panel with distributed absorption. (C) Monocylinder. (D) Bicylinder. (E) Quadratic residue diffuser. (*D'Antonio.*)

Key Points

- Reflection phase-grating diffusers can conserve useful sound energy and disperse it, while controlling troublesome specular reflections.
- A quadratic residue diffuser (QRD) is a type of reflection phase-grating; it uses a series of wells of different depths and constant width calculated to optimize diffusion.
- In a QRD, the particular sequence of well depths, as predicted by theory, is crucial in providing efficient diffusion.
- In a QRD, the maximum well depth is determined by the longest wavelength to be diffused. The well width is about a half wavelength at the shortest wavelength to be diffused.
- Sound falling on a reflection phase-grating diffuser is diffracted throughout a hemidisc. The energy is spread more or less equally throughout 180°, but is somewhat reduced at grazing angles. The specular reflection mode is suppressed.
- In a reflection phase-grating diffuser, the vertical wells scatter sound horizontally and the horizontal wells scatter sound vertically. Together they produce a virtual hemisphere of diffusion.
- Flutter echoes are periodic or near-periodic echoes from two opposing reflective surfaces such as parallel walls in a room. Two opposing, parallel surfaces of a room should never be highly reflective. Diffuser wall treatments can reduce reflections by scattering sound.
- A measure of the effectiveness of a diffuser can be obtained by comparing the sound intensity in the specular direction with the intensity at ±45° of that direction. The diffusion coefficient is 1.0 for a perfect diffuser. This coefficient varies with frequency.

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CHAPTER 15

Adjustable Acoustics

If an audio room is used for only one purpose and one type of music, it can be treated acoustically with some precision. However, economics may dictate that a given room must serve more than one purpose. Such a multipurpose room carries with it some acoustical compromises. In some cases, a room's acoustical character must be variable to accommodate different types of music. For example, a studio may be used to record a string quartet in the afternoon and a rock band that night. In either case, it is necessary to weigh any compromises arising from multiuse or variability against the ultimate sound quality of the design. For this chapter, it is desirable to cast aside the impression that acoustical treatment is an inflexible exercise and instead consider means of acoustical adjustability.

Draperies

As radio broadcasting developed in the 1920s, draperies on the wall and carpets on the floor were often used to "deaden" studios. It became apparent that this radio studio treatment was quite unbalanced, absorbing middle- and high-frequency energy but providing little absorption at lower frequencies. As proprietary acoustical materials became available, hard floors became common and drapes all but disappeared from studio walls.

A decade or two later, acoustical engineers, interested in adjusting the acoustical environment of the studio, turned with renewed interest to draperies. A good example of this return to draperies was illustrated in 1946 in the rebuilding of Studio 3A of the National Broadcasting Company in New York City. This studio was redesigned with the goal of providing optimum conditions for recording music for home playback and for recording transcriptions for broadcast. The reverberation-frequency characteristics for these two applications differ. By using drapes and hinged panels (considered later), the reverberation time was made adjustable over more than a two-to-one range. The heavy drapes were lined and interlined and were hung some distance from the wall to make them more absorbent at lower frequencies (see Figs. 12-15 through 12-18). When the drapes were withdrawn, polycylindrical elements having a plaster surface were exposed.

If due regard is given to the absorption characteristics of draperies, there is no reason, other than cost, why they should not be used. The effect of the fullness of the drape must be considered. The acoustical effect of an adjustable element using drapes can thus be varied from that of the drape itself when closed, to that of the material behind when the drapes are withdrawn into an alcove, as shown in Fig. 15-1. The wall treatment behind the drape could be anything from hard plaster for minimum sound

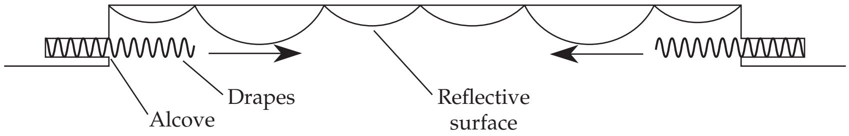

Figure 15-1 The ambience of a room may be varied by pulling absorptive drapes in front of reflective areas into alcoves.

absorption to resonant structures having maximum absorption in the low-frequency region, more or less complementing the effect of the drape itself. Acoustically, there would be little point in retracting a drape to reveal material having similar acoustical properties.

Portable Absorptive Panels

Portable absorbent panels offer a certain amount of flexibility in adjusting listening room or studio acoustics. The simplicity of such an arrangement is illustrated in Fig. 15-2. In this example, a shallow wooden cabinet holds a perforated hardboard facing with acoustically transparent cloth covering, a glass-fiber layer, and an interior air cavity. This type of panel can be easily mounted on a wall or removed as needed. For example, panels may be placed in a room to decrease reverberation for voice recording or removed from the room to obtain a live ambience for instrumental music recording.

Figure 15-3 shows one method for mounting panels on the wall with beveled cleats; the panels can be easily removed by lifting them off the cleats. Hanging such units on the wall adds absorption, and contributes somewhat to sound diffusion. There is some

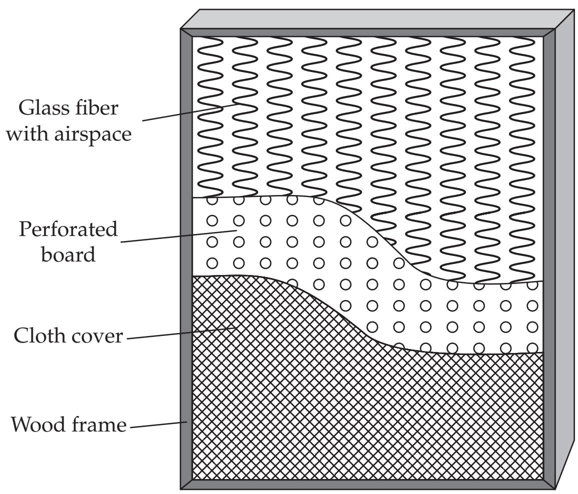

Figure 15-2 Removable wall panels can be used to adjust the reverberation characteristics of a room. For maximum variability, unused panels should be removed from the room entirely.

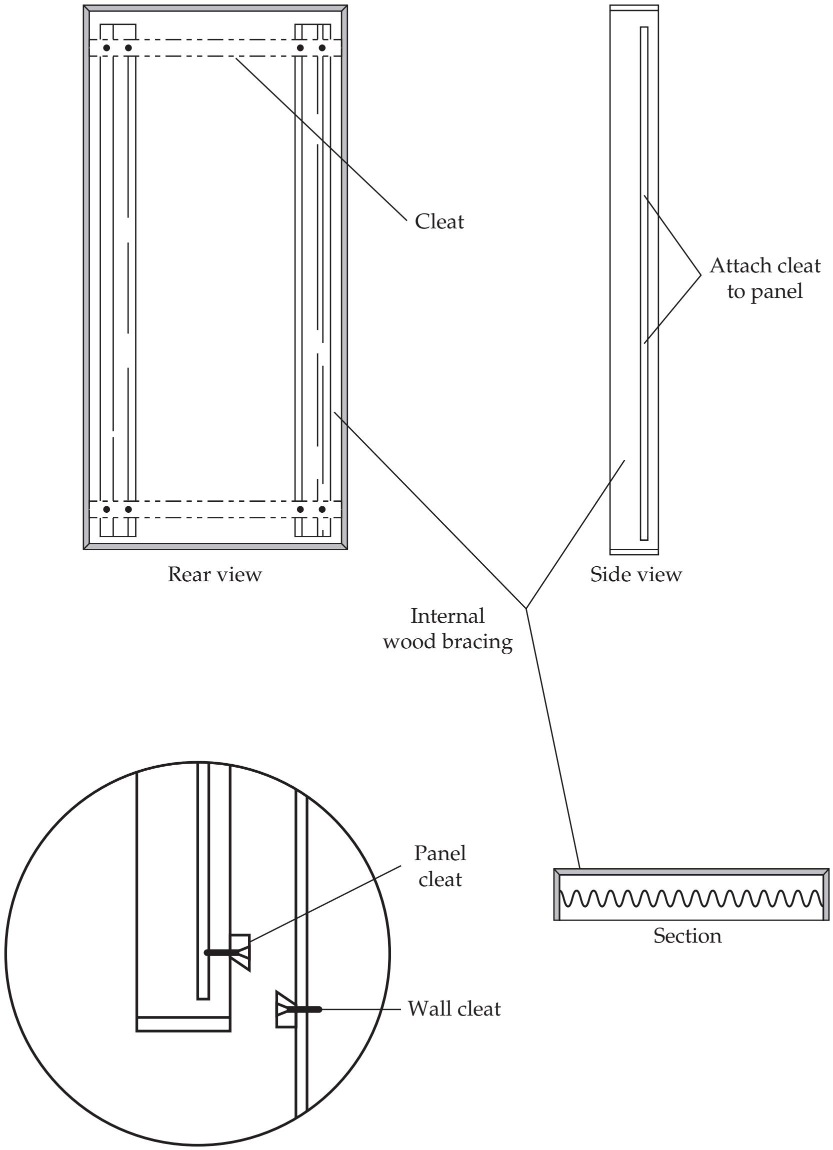

Figure 15-3 One method for mounting panels on walls using beveled cleats so that the panels are easily removable.

compromising of the effectiveness of the panels as low-frequency resonators because the units hang loosely from a mounting strip. Leakage coupling between the cavity and the room tends to degrade the resonant effect.

Freestanding acoustical flats (sometimes called baffles or gobos) are useful studio accessories, often used to improve the acoustical separation between instruments. A flat typically has one reflective side and one absorptive side. A typical flat consists of a frame of 1- × 4-in lumber with plywood back filled with a low density (e.g., 3 lb/ft^3) glass-fiber board faced with a fabric. Strategically arranging a few such flats can give a certain amount of local control of acoustics at mid to high frequencies, and also provide some isolation at higher frequencies.

The baffle suggested in Fig. 15-4 introduces diffusion into baffle technology. Sound returned from the 4- × 4-ft diffusing area is different from that returned from a flat reflective surface. For example, the return is about 8 dB lower in intensity, it is greatly diffused through the half-space, and it is spread over several milliseconds of time.

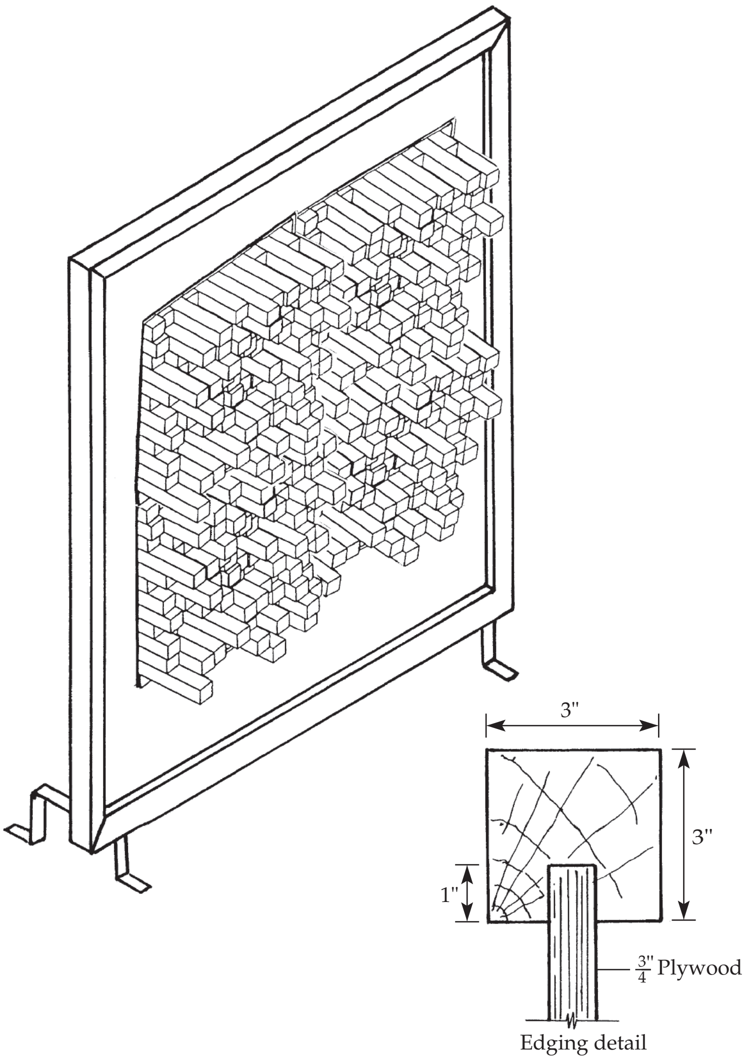

Figure 15-4 A nontraditional freestanding baffle using a diffusing surface.

Musicians may sense a fullness to the sound. To achieve this, the baffle in Fig. 15-4 has four Skyline diffusing modules mounted on one side; these are manufactured by RPG Acoustical Systems. These particular modules are primitive root diffusers; other kinds of diffusers, commercial, or scratch-built, could be employed. Alternatively, the surface could be left as a flat reflector.

Hinged Panels

One of the most effective and least expensive methods of adjusting studio acoustics is the hinged panel arrangement of Fig. 15-5A and B. When the panels are closed, all surfaces are reflective (plaster, plasterboard, or plywood). When opened, the exposed surfaces are absorptive (glass fiber or carpet). For example, 3 lb/ft^3 density glass-fiber boards 2- to 4-in thick could be used to cover the absorptive surfaces. These panels could be covered with acoustically transparent cloth to improve appearance. Spacing the glass fiber from the wall would improve absorption at low frequencies.

Louvered Panels

Multiple louvered panels can be adjusted by the action of a single lever in the frames, as shown in Fig. 15-6A. Behind the louvers is a low-density glass-fiber board or batt. The width of the panels determines whether they form a series of slits (Fig. 15-6B) or seal tightly together (Fig. 15-6C). Opening the louvers of the panels in Fig. 15-6C slightly would acoustically approach the slit arrangement of Fig. 15-6B, but it might be mechanically

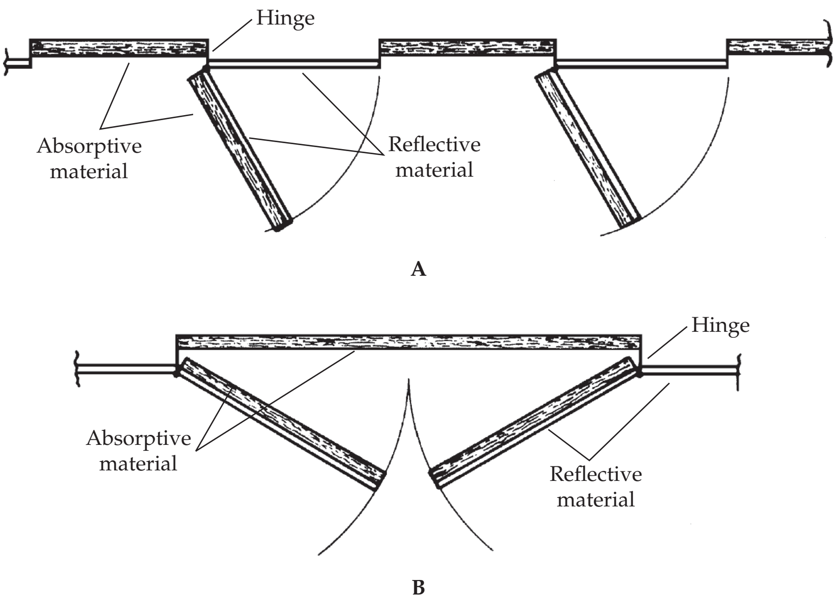

Figure 15-5 Hinged panels, reflective on one side and absorbent on the other, provide an inexpensive and effective method of incorporating variability in room acoustics. (A) Single panel design. (B) Dual panel design.

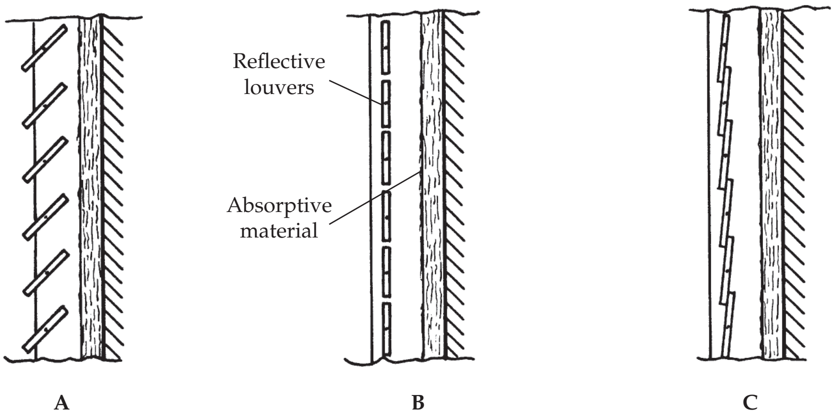

Figure 15-6 Louvered panels allow a wide range of acoustical variability. (A) Louvered panels may be opened to reveal absorbent material within. (B) Panels are closed to present a reflective surface. (C) Short louvers can change from a slat resonator (closed) to reveal absorbent material within (open).

difficult to arrange for a precise slit width. Variations in width of the slits would cause a widening of the resonant curve.

The louvered panel arrangement is very flexible. The glass fiber can be of varying thickness and density and fastened directly to the wall or spaced out by different amounts. The louvered panels can be of reflective material (glass, hardboard) or of more absorptive material (softwood) and they can be solid, perforated, or arranged for slat-resonator operation. In other words, almost any absorption-frequency characteristic can be matched with the louvered structure with the added feature of adjustability.

Absorptive/Diffusive Adjustable Panels

Absorptive/diffusive panels can combine broadband absorption in the far field with horizontal or vertical diffusion in the near field for all angles of incidence. These panels work on the absorption phase grating principle using an array of wells of equal width separated by thin dividers. They are designed to diffuse that portion of sound that is not absorbed. The depth of the wells can be determined by a quadratic residue sequence of numbers. These panels are available commercially or can be custom built.

The Abffusor is an example of a commercial absorptive/diffusive panel. It measures approximately 2×4 ft or 2×2 ft and provides diffusion down to 100 Hz. Panels can be mounted in a suspended ceiling grid or as independent elements. The absorption characteristics of the panel for two mountings are shown in Fig. 15-7. Mounted directly on a wall, the absorption coefficient is about 0.42 at 100 Hz. With 400 mm of airspace between the panel and the surface, the coefficient is doubled. The latter is approximately the performance with the panel mounted in a suspended ceiling grid. Near perfect absorbance is obtained above 250 Hz. The unit thus provides both mid/high-band sound absorption and sound diffusion. The Abffusor is manufactured by RPG Acoustical Systems.

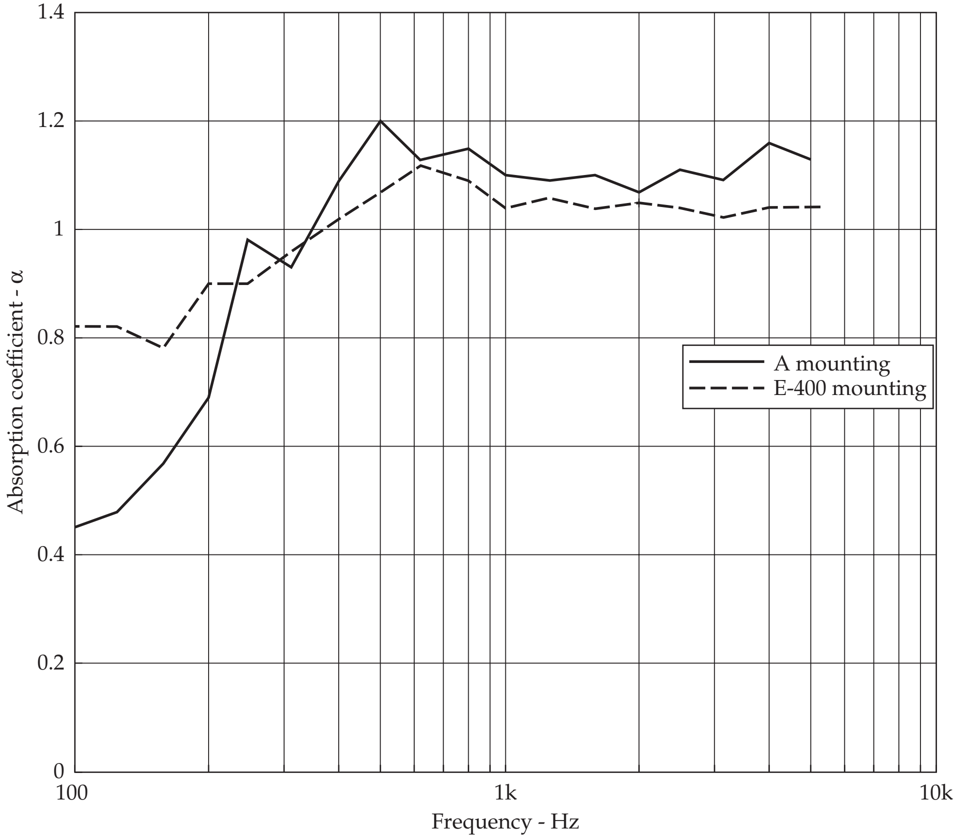

Figure 15-7 The absorption characteristics of an Abffusor panel for two mountings: Directly on wall (A mounting), and with a 400-mm airspace (E-400 mounting).

Variable Resonant Devices

Resonant structures can be used as variable sound absorbing elements. An example using perforated panels is shown in Fig. 15-8A. Varying the panel position shifts the resonant peak of absorption, as shown in Fig. 15-8B. The approximate dimensions in this example are: Panel width 2 ft, panel thickness 3/8 in, hole diameter 3/8 in, hole spacing 1-3/8 in on centers.

An important element of a variable resonant absorber is a porous cloth with the proper flow resistance covering either the inside or outside surface of the perforated panel. When the panel is in the open position, the mass of the air in the holes and the compliance or "springiness" of the air in the cavity behind act as a resonant system. The cloth offers a resistance to the vibrating air molecules, thereby absorbing energy. When the panel in this example is closed, the cavity virtually disappears and the resonant peak is shifted from about 300 to 1,700 Hz (see Fig. 15-8B). In the open condition, the absorption for frequencies higher than the peak remains remarkably constant out to 5 kHz.

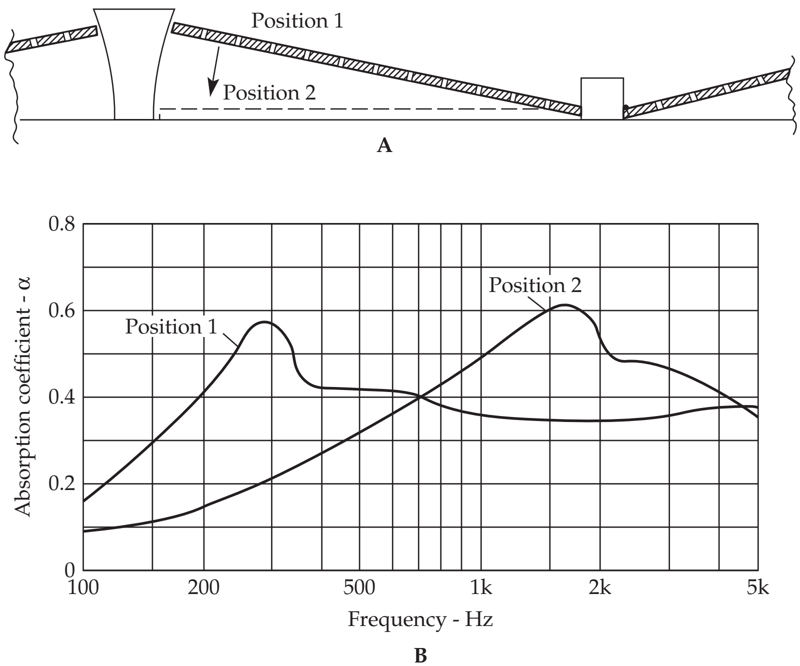

Figure 15-8 Hinged, perforated panels can be used to vary resonant absorption. A porous cloth with the proper flow resistance covers one side of the perforated panel. (A) The panels can be placed in two positions (1, 2) on a surface. (B) Changes in the panel position shift the response of the absorption characteristic.

One Hollywood dubbing studio uses another interesting resonant absorber design. Movie dialogue looping requires variable voice recording conditions to simulate the many acoustical situations of motion picture scenes portrayed. In this studio, reverberation time had to be adjustable over a two-to-one range for the 80,000-ft^3 stage.

Both side walls of the stage used the variable arrangement of Fig. 15-9, which shows a cross section of a typical element extending from floor to ceiling with all panels hinged on vertical axes. The upper and lower hinged panels of 12 ft length are reflective on one side (two layers of 3/8-in plasterboard) and absorbent on the other (4-in glass-fiber board). When open, they present their absorbent sides and reveal slat resonators (1- × 3-in slats spaced 3/8 in to 3/4 in with glass-fiber board behind), which utilize the space behind the canted panels. In some areas, glass fiber is fastened directly to the wall. Diffusion is less of a problem when only highly absorbent surfaces are exposed but when the reflective surfaces are exposed, the hinged panels meet, forming good geometric diffusing surfaces. This design by William Snow illustrates the flexibility offered in combining different types of absorbers in an effective yet inexpensive arrangement.

Rotating Elements

Rotating elements of the type shown in Fig. 15-10 provide unique adjustability; because of size constraints, they are most often used in larger rooms. In this particular configuration, the flat side is absorbent and the cylindrical diffusing element is reflective.

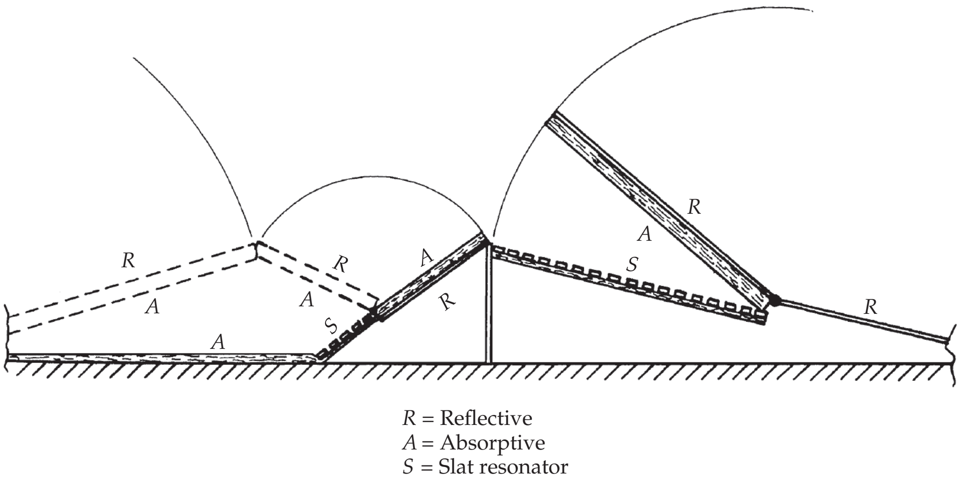

Figure 15-9 Variable acoustical elements can be used in a mixing-looping stage. Reflective areas are presented when the doors are closed. Absorbent areas and slat resonators are presented when opened.

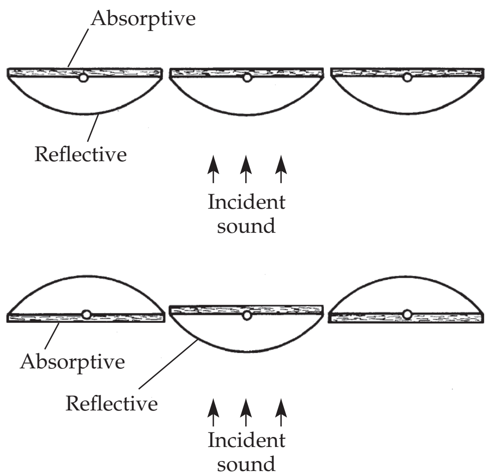

Figure 15-10 Rotating elements can vary the reverberation characteristics of a room. They have the disadvantage of requiring considerable space to accommodate the rotating elements.

A disadvantage of this type of system is the space required for rotation. The edges of the rotating element fit tightly to minimize coupling between the studio and the space behind the elements.

A music room may be designed with a series of rotating cylinders partially protruding through the ceiling. The cylinder shafts are ganged and rotated with a rack-and-pinion drive in such a way that sectionalized areas of the exposed cylinder gives moderate low-frequency absorption increasing at high frequencies, good low-frequency

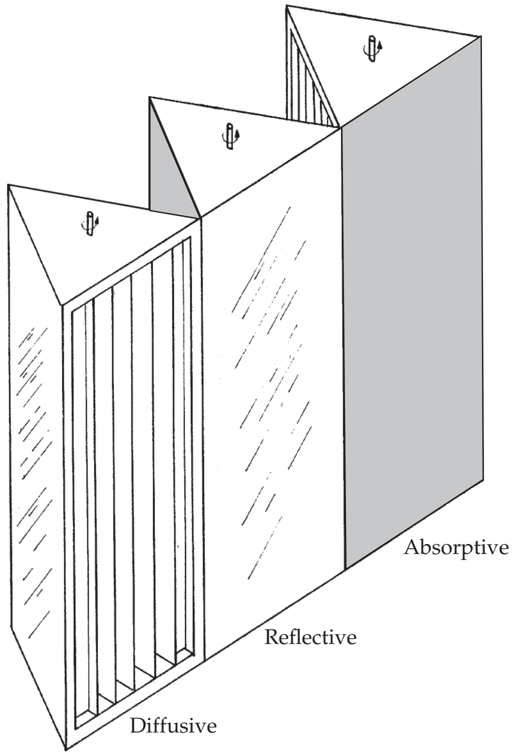

Figure 15-11 An example of a design that uses rotatable equilateral-triangular prisms with absorptive, reflective, and diffusive sides. Rotation of the individual units can provide a high degree of acoustical variability.

absorption decreasing at high frequencies, and high reflection absorbing little energy at low or high frequencies. However, such arrangements are expensive and mechanically complex.

Figure 15-11 shows an example of a design that provides a high degree of adjustability. It comprises rotatable equilateral-triangular prisms with absorptive, reflective, and diffusive sides. Nominally each prism could measure 4-ft in height and 2-ft in face width. Alternatively, a nonrotating design with two absorptive sides and one diffusive side could be placed in room corners. In a normal mounting, the edges would be butted and each unit supplied with bearings for rotation. In this way, an array of these units could provide all absorptive, all diffusive, all reflective, or any desired combination of the three surfaces. As with other rotating absorbers, this unit would require a large mounting space.

Modular Low-Frequency Absorptive Devices

It is relatively difficult to control bass frequencies, particularly in small rooms. In many cases, low-frequency absorbers, often called bass traps, are incorporated into the ceiling and walls of a room. However, smaller, more portable bass traps offer an alternative approach. An example of a modular, low-frequency absorber is shown in Fig. 15-12. This bass trap, based on a design first proposed by Harry Olson, is marketed as the Tube Trap. It is a cylindrical unit available in 9-, 11-, and 16-in diameters and 2- and 3-ft lengths. Smaller-diameter units may be stacked on top of larger-diameter units. Performance is typically optimized by placing them in the corner of a room. A quarter-round adaptation may also be used. The trap is a simple cylinder of 1-in glass fiber, given structural strength by an exoskeleton of wire mesh. A plastic sheet designated as a "limp mass" covers half of the cylindrical surface. For protection and appearance, a fabric cover is added.

As with any absorber, the total absorption is given by: (area)(coefficient) = sabins of absorption. With absorption modules such as this, it is expedient to rate the sabins of absorption contributed by each module. The absorption characteristics of the 3-ft-long Tube Traps and of the 9-, 11-, and 16-in-diameter models are shown in Fig. 15-13. Appreciable absorption, especially with the 16-in trap, is achieved below 125 Hz.

When tubes are stacked in each corner behind loudspeakers, the limp mass, which covers only half of the area of the cylinder, provides reflection for midrange and higher frequencies. Moreover, low-frequency energy passes through this limp mass and is absorbed. Through reflection of mid/high frequencies, it is possible to control the brightness of the sound at the listening position. Figure 15-14 shows the two positions of the tubes. If the reflector faces the room (Fig. 15-14B), the tube fully absorbs the lower frequency energy while the listener receives brighter sound. Mid/high-frequency sound is diffused by its cylindrical shape. If less bright sound is preferred, the reflective side is placed to face the wall. This may introduce timbral changes resulting from the cavity formed by the intersecting wall surfaces and the cylindrical reflective panel. By placing absorptive panels on the wall surfaces, as indicated in Fig. 15-14A, this timbral change can be controlled.

Modular low-frequency absorbers such as Tube Traps can also be placed in the rear two corners of the room if experimentation indicates it is desirable. Two tubes may be

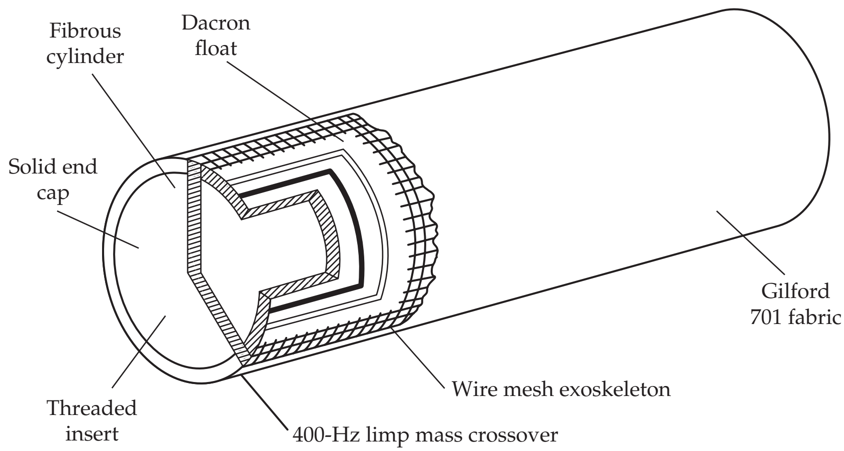

FIGURE 15-12 The construction of the Tube Trap. It is a cylinder of 1-in glass fiber with structural support. A plastic limp-mass covers half of the cylindrical surface, which reflects and diffuses sound energy above 400 Hz. (*Acoustic Sciences Corporation.*)

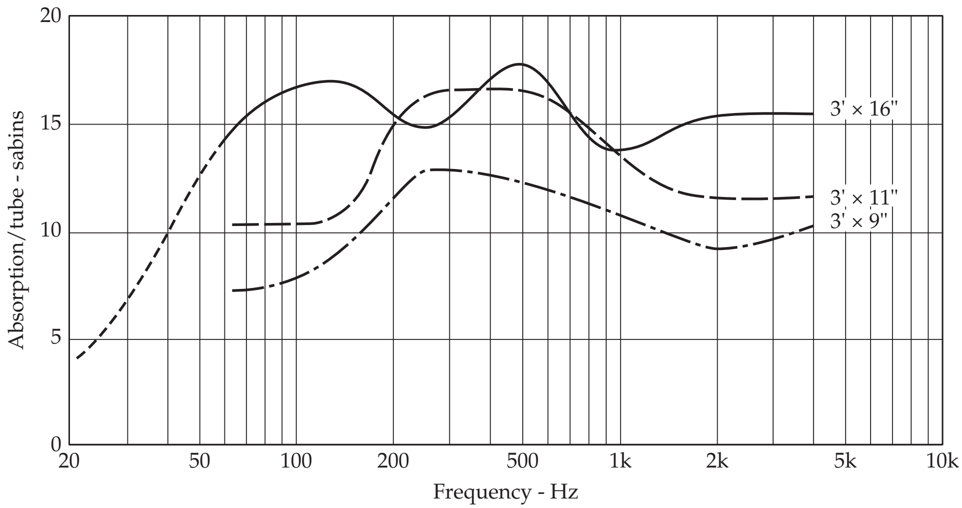

FIGURE 15-13 Absorption characteristics of three sizes of Tube Traps. The 16-in unit provides good absorption down to about 50 Hz.

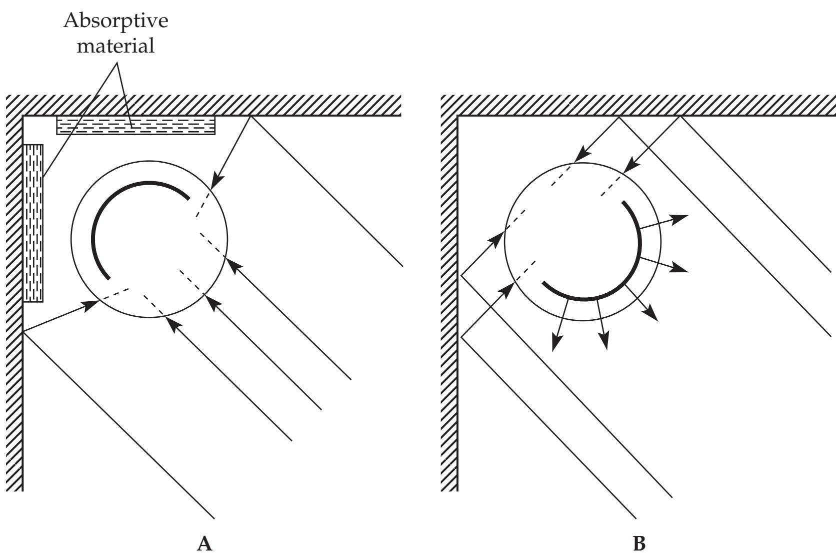

FIGURE 15-14 The positioning of the limp mass reflector in a Tube Trap gives some control over the brightness of sound in the room. (A) With the limp mass facing the corner, the absorbent side of the cylinder absorbs sound over a wide range. (B) If the limp mass faces the room, frequencies above 400 Hz are reflected.

stacked in a corner, the lower, larger one absorbing lower frequencies and the upper, smaller one absorbing moderate low and midrange energy. Half-round units can be used to control sidewall reflections or provide general absorption elsewhere. Whether these types of adjustable modules plus carpet, furnishings, structural absorption (wall, floor, ceiling) combine to provide the proper overall decay rate (liveness, deadness) must be determined by listening, calculation, or measurement. The Tube Trap is manufactured by Acoustic Sciences Corporation.

An alternative to a tube design is a prism with two absorptive sides and one diffusive side. As an example, this type of design could be used in a stereo listening room, placed behind the loudspeakers with the absorptive sides facing the corner and the diffusive sides facing outward to the room. This placement would help control normal modes while adding diffusion to the room. Nominally, diffused reflections might be reduced 8 to 10 dB, minimizing their possible contribution to perceptual confusion of the stereo image. This is in contrast to the higher-level limp mass reflections of the trap in Fig. 15-14B, which, as early reflections, may tend to confuse the stereo image.

Key Points

- Draperies can provide useful acoustical variability. The effect of the fullness of the drape must be considered; absorption can be varied from that of the drape itself when closed to that of the surface behind when the drapes are withdrawn into an alcove.
- Absorptive portable panels can be mounted on a wall or removed as needed. For example, panels may be used to decrease reverberation or removed from the room to obtain a live ambience.
- Hinged panels provide an effective and low cost method to adjust acoustics. When the panels are closed, surfaces are reflective (plaster, plasterboard, or plywood). When opened, the exposed surfaces are absorptive (glass fiber or carpet).
- Resonant structures can be used as variable sound absorbing elements. Varying the panel position shifts the resonant peak of absorption.
- Different types of absorbers and diffusers can be combined in unique ways to provide acoustical adjustability suited for any particular room or application.
- Modular, low-frequency absorbers can provide a portable cost-effective method to control bass response. Further, some models can combine acoustical design features to provide greater flexibility.

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CHAPTER 16

Sound Isolation and Site Selection

In rooms that are not acoustically sensitive, sounds external to these rooms, such as from aircraft, trains, road traffic, barking dogs, and lawn equipment, as well as internal sounds such as from plumbing and air-handling might be a natural part of everyday soundscapes, and so might not be objectionable or even noticed at all. But in acoustically sensitive rooms, during a pause or a quiet musical or speech passage, such sounds can be incongruous. Thus, the background noise levels in recording studios, listening rooms, concert halls, and other acoustically sensitive spaces must be minimized if these rooms are to be used for their intended purpose.

From an acoustical standpoint, before construction planning can be specified, two major questions must be answered. First, what is the noise level of the external environment where the room will be located? Second, what background noise level goal will be set for inside the room? This will determine the magnitude of the task, and the degree to which isolation must be designed into the structure. Specifically, the difference between these two noise levels defines the transmission loss that must be achieved through the room design.

The task of controlling interfering noise is perhaps the most challenging problem in architectural acoustics. Interior room treatment is critically important, but even the most expertly designed room treatment is worthless if noise intrudes into its space. Likewise, in many cases, sound originating in the space and interfering with neighboring spaces can be a serious issue. It is a difficult fact of room design that low ambient noise levels are an important prerequisite for most applications. To be useful, a design must suppress any internally generated noise, such as from equipment and adjoining rooms, and also provide isolation from any external noise, such as from road traffic. Equally vexing is the fact that quiet air is expensive; there are almost no economical shortcuts in obtaining the low ambient noise levels required for high-quality audio recording, playback, or listening. Still, an understanding of the behavior of noise sources and noise transmission will enable the designer to reach even the most demanding noise criteria, and minimize the effects of unwanted sound.

The questions of sound isolation in general as well as site selection are considered in this chapter. The more specific demands of walls, floors, ceilings, doors, and windows, as well as heating, ventilating, and air-conditioning (HVAC) systems are considered in subsequent chapters. To avoid confusion, it is worth noting that the terms "sound isolation" and "sound insulation" are used interchangeably. The term "sound proofing" should not be used, as it promises much more than it can usually deliver.

Propagation through Barriers

Sound can propagate through any medium; for example, it can pass through air and through solids. The former is airborne transmission and the latter can occur as structureborne transmission. For example, sound from a distant room may travel down a corridor and into your room via an air path, but it may also pass through the concrete floor that is common to both rooms and reradiate from the floor as airborne sound in your room. In fact, sound travels more efficiently in more dense mediums. Generally, airborne sound is higher in frequency (above 100 Hz) and structureborne sound is lower in frequency (below 100 Hz). In some cases, structureborne sound is present only as a vibration that is felt. Any barrier or partition must be designed to minimize both airborne and structureborne transmission. Airborne transmission is minimized by sealing any air leaks in a partition that would otherwise provide an air path through the partition. Air leaks can seriously degrade the acoustical performance of even the most formidable wall. Structureborne transmission can be reduced by decoupling elements of a partition, and thus interrupting the transmission path; for example, two partition leaves separated by resilient mounts will reduce structureborne transmission. Also, structureborne sound is reduced by eliminating any resonant conditions in the transmission frequency range.

Approaches to Noise Control

There are five basic approaches to reducing noise in an acoustically sensitive space:

- Locating the receiving room in a quiet place
- Reducing the noise output of the offending source
- Interposing a sound-isolating barrier between the noise and the room
- Reducing the noise energy within the source room and/or receiving room
- Minimizing both airborne and structureborne noise

Locating an acoustically sensitive room away from outside interfering sounds is an elegant solution but a rare luxury because of the many factors (other than acoustical) involved in site selection. Clearly, sites near airports, railroads, highways, or other noise sources are always problematic. It is useful to remember that doubling the distance from a noisy sound source reduces the level of airborne noise approximately 6 dB (3 dB when considering a line of traffic). Whenever possible, floor plans should place noise-sensitive rooms away from exterior noise sources such as roads; for example, a non-sensitive room should be positioned between the sensitive room and the road. Likewise, the sensitive room should be placed away from interior noise sources such as machinery rooms. If the room in question is a listening room or home studio which is part of a residence, due consideration must be given to serving the other needs of the occupants. If the room is a professional recording or broadcast studio, it may be part of a multipurpose complex and the noises originating from office machines, air-handling equipment, foot traffic within the building, or even sounds from other studios, must be considered.

When optimal site selection is not available, the next best solution is to reduce the noise output of the offending source. This is usually the most logical and efficient approach. Sometimes this is possible but sometimes it is not. For example, a noisy machine may be placed successfully in a sound-isolating box, but it is probably impossible to make traffic engine and tire noise softer. Many techniques are available to reduce the noise output of some sources. For example, the noise output of a ventilating fan might be reduced 20 dB by installing a pliant mounting or by decoupling a metal air duct with a special collar. Installing a carpet in a hall might solve a foot traffic noise problem, or a rubber pad might reduce an equipment vibration problem. In most cases, working on the offending source and thus reducing its noise output is far more productive than corrective measures between the source and the receiving room or within the receiving room in question.

Although it is often difficult and expensive, a common noise-reduction solution is to establish a noise-isolating barrier between the noise source and an acoustically sensitive space. A barrier such as a wall offers a transmission loss to sound transmitted through it, as discussed later. An impinging noise level would be reduced by the transmission loss of the wall. The walls, floor, and ceiling of an acoustically sensitive room must all give at least the required transmission loss to outside noises, reducing them to acceptable levels inside the room. However, a wall with some transmission loss can only reduce the noise level by that amount if the wall is not flanked or bypassed by other acoustical paths.

Protecting a site from street traffic noise can be difficult. For example, a masonry barrier may be built along a highway to shield nearby residences from traffic noise. Shrubbery and trees can modestly help in shielding a site from street sounds; for example, a cypress hedge 2-ft thick gives about a 4-dB reduction.

It is sometimes useful to apply treatment to the interior of a noise-source room or to a noise-receiving room, to reduce overall ambient noise levels in the rooms. The level of noise in any room can be modestly reduced by introducing sound-absorbing material into the room. For example, if a sound-level meter registers an ambient noise level of 45 dB inside a studio, this level might be reduced to 40 dB by covering the walls with quantities of absorbing materials. Going far enough to reduce the noise significantly, however, would probably make the reverberation time too short. The control of reverberation must take priority. The amount of absorbent installed for the control of reverberation will reduce the noise level only slightly, and beyond this we must look to other methods for further noise reduction.

Noise can permeate a studio or other room when it is transmitted through air pathways (airborne) or transmitted through solid structures (structureborne). Noise can also be radiated by diaphragm action of large surfaces or a combination of these three mechanisms. The sound of a truck driving by is mainly heard as an airborne noise, but some vibration from the tires may pass through the ground. Likewise, the sound of a plane taking off, a band's loud jam session, and a crying baby are all airborne sounds. Quieter jet engines, reduced volume controls, and pacifiers would all be examples of noise source attenuation. Many structureborne noises are caused by vibration or an impact from a piece of machinery; the energy can be conveyed through a building's structure and reradiated as audible airborne sound. The building structure efficiently converts the vibration into sound, in much the same way that a guitar string by itself produces very little sound, but when attached to a guitar body, the sound can be quite loud.

Airborne Noise

If an airway exists, sound will easily travel through the air. An air leak that creates an airway through an otherwise effective sound barrier will seriously compromise its effectiveness. In fact, the most serious weak link in any partition is an air leak. For example, a hole measuring 1 in^2 will transit as much sound as an entire 100-ft^2 gypsum-board partition. As another example, suppose that an opening 10-in wide allows noise to enter a room to yield a 60-dB noise level. If the opening is reduced tenfold to 1 in, ignoring diffraction, the noise level would still be 50 dB. If the opening is reduced to one-hundredth of its original size to 0.1 in, the noise level would still be 40 dB. In other words, even a small air leak is highly detrimental. Similarly, any flanking path will allow sound to travel around a barrier, severely compromising its sound isolation. For example, sound can easily travel from room to room through a common plenum or air-handling ducts.

A crack under a door or a wall penetration created by a loosely fitting electrical outlet box can compromise the isolating properties of an otherwise excellent partition. Air tightness is absolutely necessary to isolate against airborne noises. For this reason, construction elements such as louvered doors and windows must be avoided, masonry walls should be painted when possible to seal them, and it is critically important to seal any holes or leaks in partitions with a flexible, nonhardening sealant. Similarly, rubber gaskets must be used around doors or other openings. Even a careful design can be compromised by sloppy construction that will lead to degraded transmission loss below the design intent.

Transmission Loss

The purpose of a sound-isolating wall or barrier is to attenuate impinging sound, and thus isolate the interior from the outside noise. A barrier's ability to attenuate noise passing through it can be specified by its transmission loss (TL). TL is the loss as sound passes through a barrier. In particular, TL can be defined as the difference between the sound-pressure level (SPL) on the source side of the barrier, and the SPL on the receiver side:

$$\mathrm{TL} = \mathrm{SPL}_{\text{source side}} - \mathrm{SPL}_{\text{receiver side}} \qquad (16\text{-}1)$$

For example, as shown in Fig. 16-1, if a wall has a transmission loss of 45 dB, an outside noise level of 80 dB would be reduced to 35 dB inside, that is, (80 dB – 45 dB = 35 dB). A wall with a 60-dB TL would reduce the same 80 dB noise level to 20 dB. The higher the TL value, the greater the attenuation provided by a barrier. However, a barrier's TL rating is only valid if it is not flanked or bypassed or otherwise defeated by

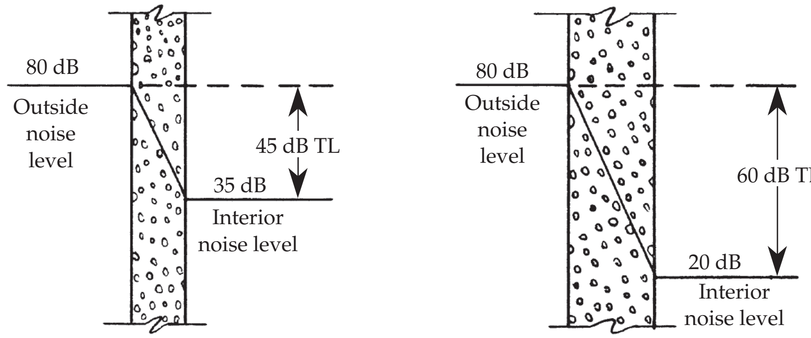


Figure 16-1 The difference between the outside noise level and the interior noise level determines the transmission loss (TL) of a barrier.

other acoustical paths. As noted, a good deal of sound can pass through even a small crack or aperture in an otherwise solid wall because the air leak has a transmission loss of zero.

It is important to note that absorption coefficients (α) are based on a linear scale, and TL values are based on a logarithmic scale. So, comparing them can be misleading. We define τ as the transmission coefficient, the amount of sound that passes through a material, where $\tau = 1 - \alpha$. We relate τ to TL as:

$$TL = 10 \log \frac{1}{\tau} \tag{16-2}$$

So, for example, a glass-fiber material might have a high absorption coefficient of 0.9 at 500 Hz which would yield a τ of 0.1, that is, $(1 - 0.9 = 0.1)$. And TL of the glass fiber would be 10, that is, 10 log (1/0.1), which is quite poor. This explains, among other things, why porous absorbers are poor sound barriers, especially at low frequencies. Recalling that porous absorbers allow airflow, this result is not surprising. In fact, as we shall see, solid massive barriers provide the best sound isolation. The total amount of sound power that passes through a partition is proportional to $S\tau$, where S is its area and τ is the transmission coefficient.

Effect of Mass and Frequency

For isolating against outside airborne sounds, the general rule is the heavier the wall the better. The more massive the wall, the more difficult it is for sound waves in air to move it. Figure 16-2 shows how the transmission loss of a rigid, solid wall is related to

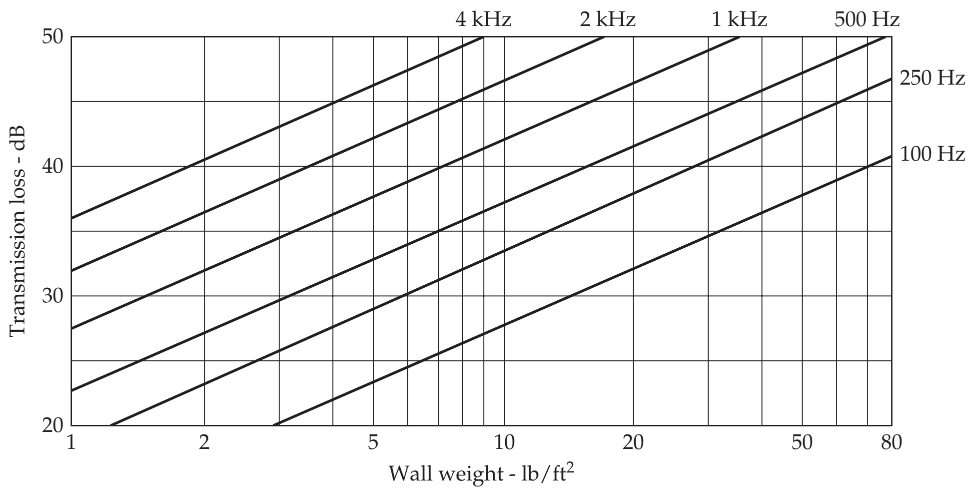


FIGURE 16-2 The mass of the material in a barrier rather than the type of material determines the transmission loss of sound passing through the barrier. The mass law predicts that a doubling of wall mass will increase TL by 6 dB (in practice, 5 dB is usually achieved). The transmission loss also depends on frequency, although values at 500 Hz are commonly used in casual estimates. The wall weight is expressed in pounds per square foot of wall surface.

the density of the wall. The wall weight in Fig. 16-2 is expressed in pounds per square foot of surface, sometimes called the surface density. For example, if a 10- × 10-ft concrete block wall weighs 2,000 lb, the wall weight would be 2,000 lb/100 ft^2, or 20 lb/ft^2. The thickness of the wall is not directly considered.

The transmission loss increases as the mass of the barrier increases. Also, transmission loss increases as the frequency increases. In a single-layer panel (such as concrete or brick wall), these effects can be approximated as follows:

$$\text{TL} = 20 \log (fm) - 33 \tag{16-3}$$

where f = frequency of sound, Hz
m = surface mass of barrier, lb/ft^2 or kg/m^2

Note: In metric units, change 33 to 47.

From these equations, since 20 log (2) equals 6 dB, we see that when mass is doubled, it theoretically results in a 6-dB increase in TL. This is sometimes called the theoretical mass law; it predicts that every doubling of mass will increase TL by about 6 dB. Moreover, we also see that any doubling of frequency (an octave increase) will also result in a 6-dB increase in TL. Because it assumes that the barrier has zero stiffness, this is more accurately known as the limp mass law. But in addition to mass, stiffness and damping also affect transmission loss. For example, real-world barriers have some stiffness; the stiffer the panel, the lower the TL. Moreover, the thicker the panel (which increases mass), the greater the stiffness (which reduces TL). So, the 6-dB prediction is not completely accurate. In practice, a doubling of mass usually yields about a 5-dB increase in TL.

Suppose that a barrier with a 4-in thickness has a TL of 40 dB at 500 Hz. If the thickness (and mass) of the barrier is doubled to 8 in, the new TL would theoretically be about 45 dB. However, the law of diminishing returns is at work: To achieve another 5 dB increase in TL, the barrier's thickness would have to be increased to 16 in, and another 5 dB would demand 32 in, and so on. Mass is extremely useful for sound isolation but is not always the best approach. Also, as we will see, other factors such as the coincidence effect will affect TL.

The transmission losses indicated in Fig. 16-2 are based on the mass of the material rather than the type of material. We see that materials with greater mass are more effective at sound isolation. It is mainly the mass of the material that matters, not the material itself. For example, the transmission loss through a layer of lead of certain thickness can be matched by a plywood layer about 95 times thicker.

Coincidence Effect

The coincidence effect creates an "acoustical hole" in a barrier; that is a phenomenological name, but it describes a factor that must be seriously considered when analyzing the transmission loss of a barrier. At lower frequencies, the TL of a barrier is mainly determined by mass; TL increases by about 5-dB/octave. But at some frequency region above this, the stiffness of the barrier will yield a resonance; the barrier will flex and bend at the same wavelength of the incidence sound, and as a result, it will more easily transmit sound. At and around the resonance frequency, the TL will drop by 10 to 15 dB below the theoretical level; this is called the coincidence-frequency dip. Above this frequency region, the TL will again rise with frequency according to the mass law, or even exceed the 5-dB/octave slope.

For a given material, the coincidence frequency is inversely proportional to the panel's thickness. Therefore, the coincidence frequency can be raised by decreasing thickness. This can be advantageous because it may move the coincidence-frequency dip to a frequency that is high enough to be above our frequency range of interest; for example, in a voice studio, this would be above speech frequencies. However, when thickness is decreased, overall TL is decreased as well.

Different materials exhibit very different coincidence frequencies: An 8-in-thick concrete wall at about 100 Hz; 1/2-in plywood panel at about 1.7 kHz; 1/8-in glass at about 5 kHz; 1/8-in lead at about 17 kHz. Materials with greater damping have a smaller coincidence-frequency dip. In some applications, a damping layer can be added to a material with little inherent damping. For example, laminated glass (but not thermal glass) has more damping than an ordinary glass pane. A discontinuous structure such as bricks set in mortar conducts sound less efficiently (providing higher TL) than a more homogeneous material like steel. The coincidence effect and other acoustical holes in glass windows are described in more detail later.

From Fig. 16-2 we can also see that the higher the frequency, the greater the transmission loss, or in other words, the better the barrier isolates against outside noise. In this figure, the line for 500 Hz is made heavier than the lines for other frequencies because it is common to use this frequency for comparisons of barriers of different materials. However, below 500 Hz the barrier is less effective at sound isolation, and for frequencies greater than 500 Hz it is more effective.

Separation of Mass

Ideally, transmission loss could be greatly improved by positioning two barriers between the noise source and the receiving room. This is because masses that are separated by an unbridged air cavity can provide very effective sound isolation. For example, the combined TL of two identical walls would theoretically be twice the individual TLs. This would be much more effective than simply doubling the surface weight of one wall. For example, an 8-in concrete wall may have a TL of 50 dB, and a 16-in concrete wall may have a TL of 55 dB. Two separated 8-in concrete walls will theoretically provide a TL of 80 dB (40 + 40); this clearly outperforms one 16-in wall. However, completely unbridged cavities are unattainable; thus in practice, the combined TL will be much less than the theoretical value.

Only in the case of two separate structures, each on its own foundation, is an unbridged condition approached. Walls are usually at least connected at the footer and header. The depth of an air cavity between panels affects the system's stiffness; the larger the cavity, the lower the coincidence-effect resonant frequency. Generally, the cavity depth between masses should be as great as allowable; very narrow cavities yield decreased performance. The coincidence-frequency dip of two separated panels can be reduced by using two panels of different surface weight. For example, performance of a double-pane window can be improved by using panes of different thickness.

Porous Materials

Porous materials such as glass fiber (rock wool, mineral fiber) are excellent sound absorbers and good heat insulators. However, as noted, they are of limited value in isolating against sound when used as standalone absorbers or when placed on wall surfaces.

Using glass fiber to reduce sound transmission will help, but only moderately. The transmission loss for porous materials is directly proportional to the thickness traversed by the sound. This loss is about 1 dB (100 Hz) to 4 dB (3 kHz) per inch of thickness for a dense, porous material (e.g., rock wool, density 5 lb/ft^3) and less for lighter porous materials. This direct dependence of transmission loss on thickness for porous materials is in contrast to the relatively high transmission loss for solid, rigid walls. As described later, porous absorbers can improve sound isolation when placed inside wall cavities.

Building insulation installed within a wall increases its transmission loss a modest amount, primarily by reducing cavity resonance that would tend to couple the two wall faces at the resonance frequency of the cavity. A certain increase in the transmission loss of the wall can also be attributed to attenuation of sound in passing through the glass-fiber material, but this loss is small because of the low density of the material. Considering all mechanisms, the transmission loss of a staggered-stud wall with a layer of gypsum board on each side can be increased about 7 dB by adding 3-1/2 in of glass-fiber insulation. A double wall might show as much as a 12 dB increase by adding 3-1/2 in, and 15 dB with 9 in of insulation.

Porous absorbers are useful for reducing reflected or ambient sound in a noise-source room. Additional absorption is effective at reducing sound levels when a room is relatively reflective; in some cases, a 10-dB reduction may be achieved. Clearly, when a worker is near loud machinery or some other loud noise source, absorption placed on the walls will not reduce direct sound reaching the worker. When possible, noise sources should be placed closer to the center of the room, and away from reflective walls. Absorption is also somewhat helpful in a noise-receiving room, to lower ambient sound levels. The amount of sound transmitted through a barrier also depends on the surface area of the barrier. In particular, the noise reduction between two adjacent rooms is given by:

$$\mathrm{NR} = \mathrm{TL} + 10 \log\left(\frac{A_{\mathrm{receiving}}}{S}\right) \tag{16-4}$$

where NR = noise reduction, dB
TL = transmission loss of barrier
$A_{\mathrm{receiving}}$ = absorption in receiving room, sabins
S = surface area of common barrier, ft^2

Sound Transmission Class

Sound transmission class (STC) is a single-number integer used to rate a barrier's sound isolation performance. It describes a series of transmission-loss measurements made at frequency intervals (usually 1/3 octave) throughout a frequency range. In other words, STC is obtained by a best fit of a standard STC contour to the actual TL measurements.

The solid line of Fig. 16-3 is a replotting of data from the mass law graph of Fig. 16-2 for a wall weight of 10 lb/ft^2. If the mass law were perfectly followed, we would expect the transmission loss of a practical wall of this density to vary with frequency as shown by the solid line. However, actual measurements of transmission loss of this wall might be more like the broken line of Fig. 16-3. These deviations reflect resonance (such as the

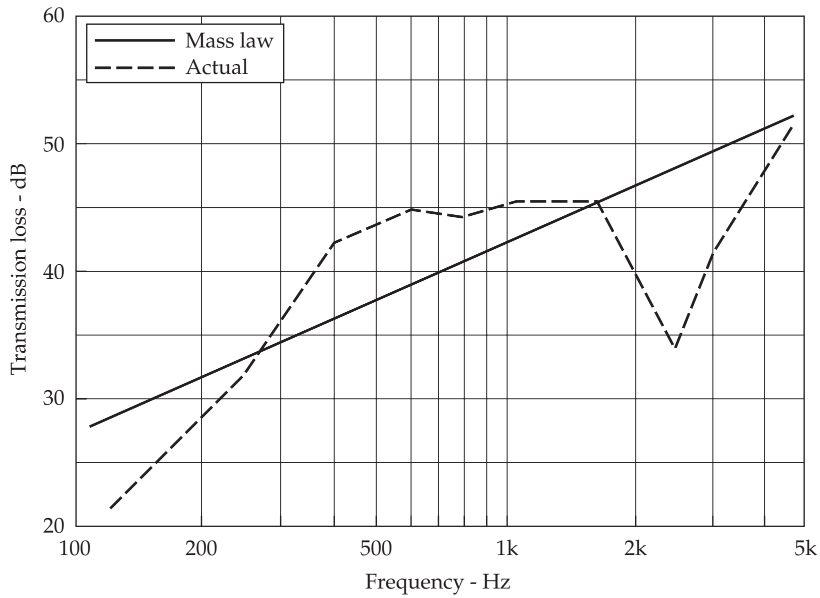


Figure 16-3 Actual measurements of transmission loss in walls often deviate considerably from the mass law (see Fig. 16-2) because of resonances and other effects.

coincidence effect) and other factors of the wall panel which are not included in the simple mass law concept.

Because of such commonly occurring irregularities, it is of practical value to use a single number such as STC that gives a reasonably accurate indication of the sound transmission loss characteristics of a wall. This is done with a procedure specified by the American Society for Testing and Materials (ASTM) to determine the STC of a sound-isolating wall. The ASTM E-413 standard states that STC is intended to correlate with subjective impressions of sound isolation from normal sources in homes and offices. Although STC values are very helpful, they are not intended to describe more "industrial" applications such as recording studios, in particular when loud music is the noise source. This is because STC is intended mainly for use in the speech-frequency region.

The STC value of a wall can be determined graphically. The measured TL contour of the wall is compared to a reference STC contour using a special procedure. This standard contour covers the frequency range from 125 Hz to 4 kHz. The contour comprises three line segments with different slopes: 3 dB per 1/3 octave from 125 to 400 Hz; 1 dB per 1/3 octave from 400 Hz to 1.25 kHz; flat from 1.25 to 4 kHz. The first segment has a rise of 15 dB, and the second segment has a rise of 5 dB. After comparing the measured TL contour with the standard contour, the STC assigned to the panel is the TL value of the standard contour at 500 Hz. The higher the STC, the higher the TL.

The results of such classification have been applied to walls of various types for ready comparison. An STC rating of 50 (dB is omitted in the rating) for a wall would mean that it is better in isolating against sound than a wall of STC 40. In practice, given a quiet receiving room, a wall with an STC of 30 would allow speech to be heard on the other side. An STC of 50 would block most loud speech, but music could be heard. An STC of 70 would block all speech, but some music, particularly bass frequencies, may be heard.

It is not proper to consider STC ratings as averages, but the procedure does escape the pitfalls of averaging dB transmission losses at various frequencies. As noted, the STC value only covers the speech frequency range, so it has limited applications when music is the sound source. In particular, a high STC rating will not ensure good low-frequency attenuation. In fact, the STC rating is not intended for external partitions where, for example, traffic noise may be an issue. Also, STC ratings do not account for specific dips in transmission loss. In some cases, the ceiling sound transmission class (CSTC) or the outdoor-indoor transmission class (OITC) is cited. These are similar to STC but are determined by somewhat different procedures. In both cases, the higher the value, the greater the transmission loss. In some countries, the Sound Reduction Index (SRI) is used instead of the STC.

Structureborne Noise

Sound travels very efficiently (with little attenuation) through dense materials, including such building materials as concrete and steel; this is structureborne noise. Structureborne noise such as vibration from outside traffic or HVAC units, or even the impact of footsteps in a distant part of a building can easily reach an acoustically sensitive room. Structureborne vibrations can cause walls and floors to vibrate, reradiating energy into a room as airborne noise. Unwanted sounds can invade a room by mechanical transmission through solid structural members such as wood, steel, concrete, or masonry. Air-handling noises can be transmitted to a room by the sheet metal of the ducts (as well as by the air in the ducts), or both. Water pipes and plumbing fixtures, unfortunately, have excellent sound-carrying capabilities.

Structureborne noise is thus most efficiently controlled at the source of the noise. Massive, rigid partitions such as concrete walls are most useful for attenuating airborne noise but offer little resistance to structureborne noise. On the other hand, lightweight materials offer little protection against airborne noise, but can be used to decouple elements of structures, and are thus effective against structureborne noise.

It is difficult to make a solid structure vibrate through airborne noise falling upon it because of the inefficient transfer of energy from tenuous air to a dense solid. On the other hand, a motor bolted to a floor, a slammed door, or a machine on a table with legs on a bare floor can cause the structure to vibrate significantly. These vibrations can travel great distances through solid structures with little loss. With wood, concrete, or brick beams, longitudinal vibrations are attenuated only about 2 dB over 100 ft. Sound travels easily in solids; for example, sound travels in steel about 20 times farther than in air for the same loss. Although joints and cross-bracing members increase the transmission loss, TL is still very low in common structural configurations. An efficient way to minimize structureborne transmission is through decoupling. For example, vibration from a machine placed directly on a concrete floor will pass easily through the floor, but if the machine is mounted on springs on an isolation pad, which in turn is also decoupled from the floor, the vibration pass-through can be greatly reduced.

Structureborne noise is often generated by an impact on a structural surface. Even a brief impact can impart tremendous energy into a structure. For example, footsteps on a wooden floor can transmit sound loudly to the room below. The impact insulation class (IIC) is a single value rating that can be used to quantify impact noise in a floor/ceiling. Measurements are taken at sixteen 1/3-octave bands from 100 to 3,150 Hz and are overlaid on a standard contour. The higher the IIC value, the better the isolation, and the lower the received noise level. Because the standard contour does not fully account for poor performance at low frequencies, the IIC can sometimes overrate lightweight floors.

Perhaps the most efficient way to minimize impact noise in a floor/ceiling is to install a soft floor covering. For example, a carpet and underlay could be placed on a concrete floor. This greatly reduces impact energy. The IIC value might improve by 50 points; however, the STC value would remain about the same. Carpet is somewhat less effective on most wooden floors but still provides significant improvement. As described in Chap. 17, a floating floor can also be used to improve IIC as well as STC.

Noise Transmitted by Diaphragmatic Action

Although very little airborne sound energy is transmitted directly to a rigid structure, airborne sound can cause a wall to vibrate as a diaphragm and the wall, in turn, can transmit the sound through the interconnected solid structure. Such structureborne sound might then cause another wall at some distance to vibrate, reradiating noise into the space we are interested in protecting. Thus, two walls interconnected by a solid structure can serve as a coupling agent between exterior airborne noise and the interior of the listening room or studio itself.

Noise and Room Resonances

Room resonances can affect the problem of outside noise in a studio. Any prominent modes persisting in spite of acoustical treatment make a room very susceptible to interfering noises having appreciable energy at these frequencies. In such a case, a feeble interfering sound could be augmented to a disturbing level by the resonance effect. Increased isolation or increased absorption around that modal frequency would be required.

Site Selection

Site selection is an important consideration in the construction of any acoustically sensitive space. Sophisticated acoustical treatment that optimizes frequency response and reverberation would be wasted on a space that has a high ambient noise level or is prone to occasional intrusive sounds. Noise levels can be satisfactorily reduced by appropriate floor, wall, and ceiling construction, but such sound isolating construction is expensive. It is far better to situate the building on a site that has low ambient noise levels to begin with. However, problematically, quiet sites are often difficult to find and may be unsuitable for other reasons such as accessibility or convenience. But whatever site is selected, even if the space is to be built in an existing location, it is important to understand the ambient noise levels surrounding the building.

In some locations, background noise levels are extremely low. For example, in wilderness areas of some national parks, the $L_{90(1)}$ reading (discussed later) might be below 10 dBA in octave bands from 250 to 2,000 Hz. A quiet suburban neighborhood at night, in the winter, might have noise levels measuring from 30 to 40 dBA. Unfortunately, it is rarely possible to find a practical building site with ambient noise levels so low. Noise surveys (discussed later) are often used to evaluate ambient noise levels at a site.

Site selection begins with an understanding of the intended purpose of the acoustically sensitive space. Not all spaces demand the same stringency in noise levels. For example, in a venue such as a concert hall, a fleeting sound may cause a minor annoyance. For example, in some excellent concert halls, the faint sound of a subway train might be occasionally audible. A recording studio has perhaps the most critical requirements in terms of sound isolation; in a recording studio, a fleeting sound could become part of a recorded track. Either the track would have to be rerecorded, or else the sound would be a permanent fixture that would leave a legacy of annoyances. In practical terms, a high-quality studio demands an ambient noise level that is as quiet as possible.

Recording studios and other acoustically sensitive spaces are rarely located in isolated buildings with no exterior noise problems. In many cases, the building is uncomfortably near a road, flight path, or other noise source. For example, vehicular traffic on an urban street can generate unacceptably high noise levels as sound reflects from one building surface to another and balconies reflect sound back onto the street level. In addition, it may be necessary to deal with other building tenants, some of whom might be serious noisemakers in their own right. For example, an offset printing shop, machine shop, carpenter shop with power tools, or an automobile repair facility can all pose serious problems. Such circumstances require special solutions and can significantly increase the cost of design and construction. The cost of locating a building away from noise sources must be compared to the cost of providing the building with sufficient isolation to allow operation in a noisy environment. In particular, it can be very expensive to isolate a building in an environment with low-frequency noise. In that case, seeking a quieter site may be more cost-effective than providing acoustical isolation.

The best way to achieve a quiet space is to locate the building away from any external noise sources. Roadways and highways, airports and flight paths, railroad tracks, heavy-industry factories, and similar noise sources will necessitate rigorous isolation treatment. Choosing a site away from these sources will lower costs. For example, a recording studio located at least several miles from a commercial airport can be designed with a much lower isolation requirement. In some cases, a site in an otherwise noisy area might be suitable because other buildings block sound from specific noise sources such as a highway. When choosing a quiet site, as much as possible, anticipate future building projects that may bring unwanted noise to the area. Also consider that in the future, otherwise favorable zoning assignments could be changed unfavorably.

Many noise sources are temporary, and are thus more difficult to anticipate. For example, the heavy machinery that is part of any city soundscape can appear outside any street address, and possibly stay for days, weeks, or months. For example, noise from a road improvement project could essentially shut down the operation of a recording studio during the duration of the project. Ideally, an acoustically sensitive facility, particularly one that is a commercial business, should be located away from any potential temporary noise sources, or designed to isolate the interior from unexpected temporary noise.

In some unique cases, people can create very loud unwanted noise levels. In particular, the spectators in sporting events venues can generate levels in excess of 100 dBA. For example, a crowd's noise level in an outdoor football stadium might reach 111 dBA and the interior of a hockey arena might reach 113 dBA. Sound levels at music concerts can reach similar or higher levels. Even the noise levels at a school playground during recess can create problems for an acoustically sensitive space.

In the home, although most noise levels are relatively low, there are numerous interior noise sources that can interfere with home recording and listening. Outdoor light machinery such as lawn mowers and leaf blowers can generate relatively high noise levels. A barking dog might create an impulsive noise level of 90 dBA measured at 5 ft. Also, clearly, internal noise sources within the home must be considered.

The best way to deal with a noise problem is at the source. As one example, a recording studio in a remote area was troubled by infrequent low-frequency sounds. The noise was traced to cars passing over an old wooden bridge near the building. The most expedient solution was to pay to build a new and quieter bridge. More commonly, for example, noise sources such as HVAC units can be placed on isolation pads or moved to another part of the building.

In many cases, for example, with traffic and airplane noise, source treatment is not possible. Exterior HVAC systems require ventilation; thus isolation materials cannot be directly applied. In some cases, when the source cannot be treated, barriers can be used to mitigate noise intrusion. Barriers can successfully reduce high-frequency noise (e.g., tire noise) but are much less effective with low-frequency noise (e.g., combustion-engine noise). This is because the longer wavelengths of low frequencies will diffract, thus bending around obstacles such as walls. Earthen berms and masonry walls provide some isolation. The taller the barrier, the better. If possible, absorption should be placed on the barrier surface. Trees and vegetation are not efficient as isolators. When using a barrier, it should be placed close to either the noise source or the desired quiet area. In addition, the barrier must extend horizontally well beyond the noise source or quiet area to avoid flanking.

Roadway vehicle traffic poses a great challenge to acoustical designers. In an urban area, many commercial sites will have traffic noise levels that exceed residential noise level standards. Noise from highways is particularly difficult to control. Costly remedies such as concrete barriers and earthen berms are often constructed. According to the Federal Highway Administration, very generally, a barrier will provide only 5 to 10 dB of attenuation. Because of diffraction of sound over and around barriers, their practical maximum attenuation is about 15 dB. Attenuation of at least 3 to 5 dB can be expected when the barrier interrupts the line of sight to the source. Each additional 2 ft of height adds about 1 dB of attenuation as the barrier's shadow zone increases. A barrier's length should be about four times the distance between the source and the listener. If the sound source can be seen over, through, or around a barrier, the barrier provides no significant sound attenuation from that source.

The Noise Survey

Factors such as land value, building cost, and rental cost must be considered when locating a business. Clearly, more desirable properties will demand higher prices. It is also expensive to provide sound isolation for a building. To identify a suitable location, while minimizing acoustically mandated construction or remodeling costs, it is important to compare potential locations on the basis of a noise-level survey. A noise-level survey conducted over a period of time is a simple procedure, but one that can yield valuable information.

In the methodology described here, a noise survey is taken "by hand" with a sound-level meter that measures only sound levels. This might seem elementary and time-wasting, but it will provide a learning experience to those conducting the survey, giving them a fundamental understanding of the process. Moreover, although it is laborious, this kind of survey is inexpensive and yields good results.

First, a single measuring point outdoors is selected that is on the proposed building site, and which appears to sample the known noisemakers of the neighborhood. A 24-hour period of measurements is then scheduled for this point. A sampling interval could be hourly, or every 15 sec. It is suggested that a measurement be taken every minute on the minute. This yields a set of 1,440 samples, which should give a reasonable picture of the environmental noise. Readings should be taken with the sound-level meter set for A-weighting, resulting in dBA measurements.

A suggested data sheet template for recording noise measurements is shown in Fig. 16-4. Sound-level readings will not be logged as numbers, but only as slashes in the appropriate column in which the levels fall, marked in the upper part of the template.

Noise Survey Worksheet											
	40–42	42–44	44–46	46–48	48–50		70–72	72–74	74–76	76–78	78–80
24 Hour Noise Measurement Tally - dBA											
Total											
Cumulative											
Percentage											

FIGURE 16-4 A template for collecting noise survey field data.

The fifth slash in any given column should be a diagonal to help in the counting later. Each column contains 3-dBA readings, such as 48–50, 72–74, and so on. Little final accuracy will be lost through this simplified 3-dBA coarseness. Once every minute, a slash will be added to this form in the appropriate column. This is the only duty until the 24-hour measurement period is completed. Of course, this data collection process could easily be automated.

At the end of the 24-hour period, the measured data is used to build a statistical distribution curve. In particular, the lower part of the template in Fig. 16-4 can now be completed. The Total row is filled in numerically, with the total number of slash marks in each column. When the Total row is full, entries are made in the Cumulative row. The total number of slashes in the 78–80 column is added to the total of the 76–78 column, and recorded in the associated box in the Cumulative row. The total of the 74–76 column is then added to the sum of the 76–78 and the 78–80 totals, and so on. The last cumulative entry will be the total of all slash marks on the data sheet.

Next, the Percentage row is completed. The cumulative value in the first box is divided by the total number of slashes. This is multiplied by 100 to express the value as a percentage, and the result is recorded in the first box in the Percentage row. The second cumulative amount is then divided by the total in the Cumulative row, expressed as percentage, and recorded in the second box in the Percentage row, and so on. As with data collection, this tallying procedure could be automated.

This data is used to draw a distribution curve such as the example shown in Fig. 16-5. The first point on the distribution curve is found by plotting the first percentage value

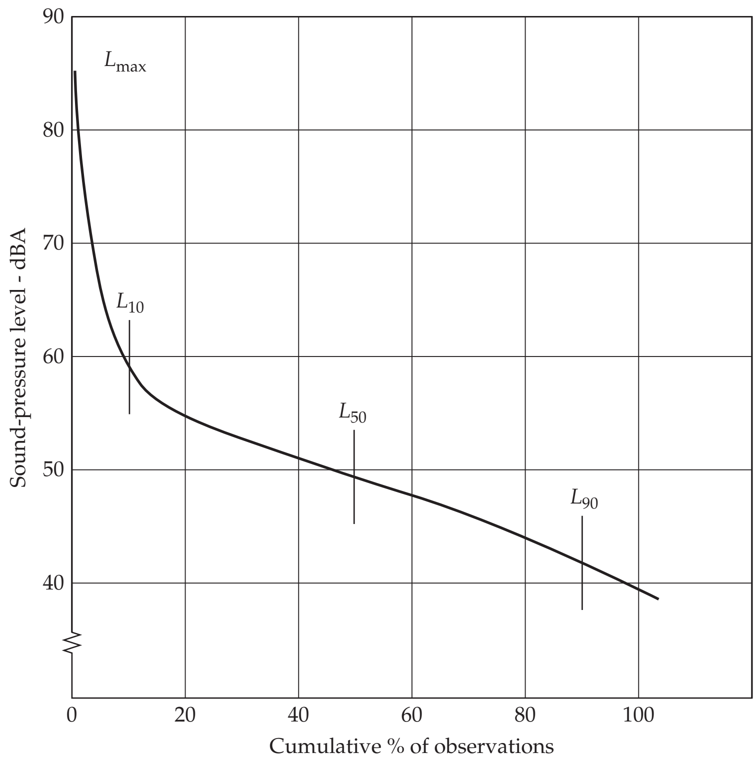


FIGURE 16-5 An example of the statistical distribution of sound-level measurements.

in the percentage box against the 79-dBA level. The second plotted point is the second percentage value against 77 dBA, and so on. This distribution curve gives a statistical picture of the environmental noise at the measurement location, for that 24-hour period. A final report should include all pertinent numerical information, as well as a drawing of the site showing building footprint, data measurement position, and locations and distances of known sound sources.

Assessment of Environmental Noise

Exterior environmental noise sources such as traffic, aircraft, and industrial noise are often difficult to quantify because they typically vary over time. A variety of descriptors have been devised to measure and evaluate environmental noise. These include L_{eq}, L_{dn}, L_{max}, L_n, and SEL; they are measured in dBA. L_{eq} is defined as the continuous equivalent sound level. It is the sound-pressure level, if it was constant, that would contain the same energy as the fluctuating sound (such as traffic levels) that is being monitored over the evaluation time period. The evaluation period is indicated in parenthesis; for example, $L_{eq(1)}$ refers to a 1-hour period and $L_{eq(24)}$ refers to a 24-hour period.

L_{dn} (also known as DNL) is a day-night equivalent sound level. It is a continuous $L_{eq(24)}$ measurement with a value of 10 dBA added to levels measured between 10 PM and 7 AM because noise is more annoying during nighttime sleeping hours. Some acousticians argue that L_{dn} does not fully account for isolated loud events such as aircraft flyovers.

The L_{dn} measure is intended to assess noise level, for example, near airports. The Federal Aviation Administration may use L_{dn} to specify aircraft noise levels. In some cases, a community noise equivalent level (CNEL) is employed; it differs from L_{dn} by adding weighting factors for evening hours. The L_{eq} is increased by 5 dBA for evening hours (7 PM to 10 PM), and is increased by 10 dBA for nighttime hours (10 PM to 7 AM). A variety of other metrics are used to quantify aircraft noise, including perceived noisiness (PN), judged or calculated perceived noise level (PNL), tone-corrected perceived noise level (TPNL), effective perceived noise level (EPNL), and others.

When recording data points in a template (such as shown in Fig. 16-4), there will be times when airplanes fly overhead, trains pass by, and sirens wail, giving levels far above the usual background values. The maximum levels must be recorded separately, as the noise assessment is not complete without them. From all of these extra high values, one is the highest and it becomes L_{max} for the survey. It is possible that this is the most important reading taken because noise such as this could penetrate the walls of a space, albeit infrequently. All of the maximum values together can be analyzed separately. If the sound of an airplane passing overhead is heard several times during a 24-hour period, this may be the one noise that dominates building design decisions. On the other hand, if an airplane passes overhead only once a week, it should have a lesser influence on the design.

Noise levels over time at a site can be quantified using the L_n designation. L_n defines a percentile sound level, where *n* (0 to 100) is the percentage of the measurement time period when a certain sound level measured in dBA is exceeded. The descriptors L_{10}, L_{50}, and L_{90} are often used to indicate noise levels with respect to thresholds. For example, $L_{10} = 70$ dBA describes an SPL measurement where a value of 70 dBA was exceeded 10% of the time. L_{10} indicates the loudest noises over time. The relationship of L_{10}, L_{50}, and L_{90} is shown in Fig. 16-6 with an example of a noise plot. In addition, the measuring time period is quoted; for example, $L_{10(1)}$ denotes a 1-hour time period. L_n provides a way to document fluctuations

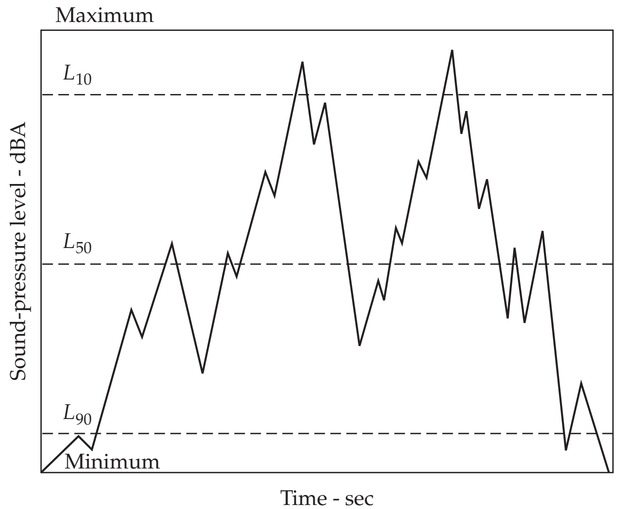


FIGURE 16-6 The descriptors L_{10}, L_{50}, and L_{90} are used for noise assessment with level thresholds. An example of a noise plot is shown.

in sound-level measurements; for example, a large difference between L_{10} and L_{90} readings over a 1-hour period would indicate significant changes in a noise environment.

The statistical distribution of sound levels can be easily plotted. A distribution graph (such as shown in Fig. 16-5) is indefinite on the L_{max} value but is very definite on L_{10} (the noise level exceeded 10% of the time), L_{50} (the noise level exceeded 50% of the time, i.e., the median sound level), and L_{90} (the noise level exceeded 90% of the time). These data are very important in deciding how much isolation a structure must provide.

The question will arise, how will L_{10}, L_{50}, and L_{90} vary throughout the day? This is where integrating instruments excel. Figure 16-7 illustrates one way of presenting such information. Note that with these SPL measurements, the frequency response of the noise is not taken into consideration.

The sound exposure level (SEL) defines and measures the total energy of a single noise event such as a train passing by or an aircraft flyover. For easier comparison, the SEL compresses noise events into a 1-sec period, even though the actual event time may be greater than 1 sec. Because of this compression, SEL values lasting more than 1 sec are higher than other measurement values. The noise exposure forecast (NEF) can be used to estimate single-noise events.

Measurement and Testing Standards

Numerous standards have been developed for assessing environmental noise. For example, the American National Standards Institute (ANSI) and the American Society for Testing and Materials (ASTM) publish documents that define and describe standard methods. When these standards are followed, the results of noise measurements can be considered to be credible. Some commonly used standards include *ANSI/ASA S1.13-2005 (R2010) Measurement of Sound Pressure Levels in Air*; *ANSI/ASA S12.9-2013 (R2018) Quantities and Procedures for Description and Measurement of Environmental Sound*; *ANSI/ASA S1.26-2014 (R2019) Methods for Calculation of the Absorption of Sound by the Atmosphere*; and *ASTM E1014-12 (2021) Standard Guide for Measurement of Outdoor A-Weighted Sound Levels*.

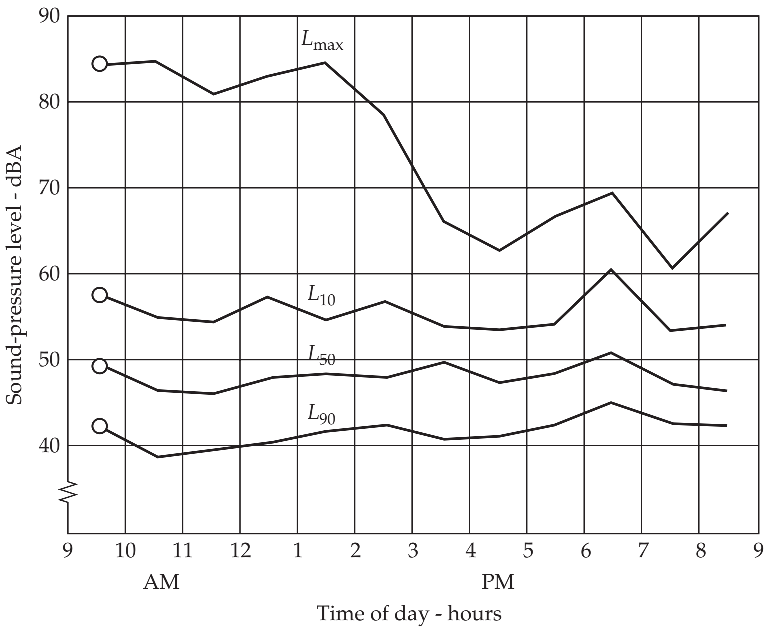


Figure 16-7 An example of sound-level measurements taken over 12 hours.

ANSI/ASA S1.13-2005 (R2010) Measurement of Sound Pressure Levels in Air defines survey, field, and laboratory methods to measure sound levels. The survey method uses a handheld SPL meter to measure sound levels in the environment. To obtain source-only readings, level readings are adjusted when large differences (4 to 15 dB) are noted when a sound source is on or off; this is described in more detail later. The field method uses an octave or narrowband analyzer to perform spectrum analysis of the noise level; the measuring microphone must be mounted on a tripod or suspended, and observers should be a distance away. The laboratory method is used in controlled environments such as anechoic or reverberation chambers. The standard also describes fluctuating or impulsive sounds, distances from obstacles, sizes of obstacles, room interior measuring placement, atmospheric and terrain conditions, and other factors that influence measurements.

ASTM E1014-12 (2021) Standard Guide for Measurement of Outdoor A-Weighted Sound Levels describes simple guidelines for measuring outdoor noise levels. An SPL meter is used; a headphone output is recommended so the operator can detect unwanted sounds such as wind noise. Measurements should be taken during noisy and quiet periods of the day, when a sound source is operating and when it is not. The number of instantaneous measurements taken must be at least 10 times the range of the reading levels. For example, if the range is 5 dB, 50 measurements must be taken.

Recommended Practices

A number of recommended practices should be followed when taking noise measurements. A calibrator should be used to check the accuracy of any sound-level meter. A calibrator fits over the microphone and generates a tone (e.g., a 1-kHz sine wave) or a series of tones at a certain sound-pressure level. A meter should accurately read the SPL of the tone emitted by the calibrator, or be adjusted to an accurate reading. A meter should be checked before it is used, and afterward as well to ensure that its calibration has not drifted. Adjustments to calibration may be necessary if a weighting network filter is used. When a 1-kHz calibration tone is used, no adjustment is needed because both A- and C-weighting networks are equal (0 dB) at that frequency.

For outdoor measurements, a properly fitting windscreen should be used to reduce wind noise at a measuring microphone. Even then, measurements should not be taken when wind speeds are greater than 15 mph. Measurements should not be taken while it is raining or even if the ground is wet after a rain; the added sound of rain or wet surfaces may affect the readings; for example, traffic sounds are different on wet pavement. Measurements should not be taken when snow is on the ground. Other meteorological conditions such as relative humidity, temperature, and barometric pressure can affect sound-level readings. They should be noted in the report, and corrections applied if necessary. The measuring microphone should be placed at least 3 to 4 ft away from any reflecting surface including the ground, as well as the observer, to avoid reflections; sound-pressure level increases significantly (up to 6 dB) near a reflective surface.

When measuring the noise level of a sound source, if the SPL with the source operating is between 4 and 15 dBA higher than the level when the source is not operating (background noise level), the measured level must be adjusted. In particular, the background noise level is significant and must be subtracted from the combined noise level to accurately determine the noise level of the source alone. The amount subtracted depends on the difference between the two measured levels (source on or off). The corrections are listed in Table 16-1. If the difference is 3 dB or less, one can conclude that the sound level of the source is equal to or less than the background sound level; therefore, the sound level of the source cannot be accurately determined from the reading.

Difference in dBA between SPL Measured with Sound Source Operating, and Background SPL	Correction in dBA to be Subtracted from SPL Measured with Sound Source Operating to Obtain SPL from Source Alone
4	2.2
5	1.7
6	1.3
7	1.0
8	0.8
9	0.6
10	0.4
11	0.3
12	0.3
13	0.2
14	0.2
15	0.1

Table 16-1 Standard Corrections for Measured Ambient Sound-Pressure Levels [*ANSI/ASA S1.13-2005 (R2010).*]

Noise Measurements and Construction

Balanced noise criteria (NCB) contours are often used to define the noise level within a structure. These curves plot octave SPL with respect to frequency and show different levels of maximum permissible noise. The curves slope downward with frequency because the ear is less sensitive to low frequencies and because many noise sources are dominated by low-frequency content. The NCB contours set limits on the background noise in a room. For example, an NCB-15 contour is very stringent and would be used in a critical application such as a professional recording studio. Noise criteria curves are discussed in detail in Chap. 19 in the context of HVAC system specification.

When a noise survey is performed, the ambient sound-pressure level can be measured in each octave band centered at 16 Hz to 8 kHz. These sound-pressure levels are plotted on the family of noise criteria curves. The noise level can then be described by the highest criteria curve reached by measured values. For example, the sound-pressure level in the 1-kHz octave band might reach the NCB-20 curve, and can thus be described in simplified terms as an NCB-20 noise level. In this way, different noise levels can readily be compared on a single-number basis.

The noise criteria contours can be used when designing the partitions of rooms, and in particular, can be used to determine what transmission loss is required. For example, Fig. 16-8A shows an example of a plot of noise levels taken on site. If the design criterion is the NCB-20 contour, the intervening area between the curves shows the transmission loss required of the room partitions across the frequency spectrum. To obtain the required TL curve, the difference between the curves can be plotted at each standard frequency. In this example, at 500 Hz, the difference is 44 dB, and at 1 kHz the difference is 40 dB, as shown in Fig. 16-8B. With this curve, different types of partitions can be examined to find the one that is most suitable for meeting the isolation requirement. Ideally, the TL curve of the selected partition should exceed the difference curve at all frequencies. It is still possible that occasional exterior noises at the site may exceed the designed TL, and those noises will be audible inside.

As noted, the highest environmental noise levels are usually produced by trains, airplanes, and vehicular traffic. To prevent such noises from interfering with studio activities, partitions must be designed to reduce such noises to a tolerable level. While vehicular traffic noise may be fairly continuous, train and airplane noise may be occasional. Figure 16-9 shows the spectrum of noise as a jet passes over a studio. The lower curve is the NCB-15 contour, which may be either the measured background noise of the studio or the studio noise design goal, depending on the intent of the noise survey. The difference between the two curves, in dB, is the attenuation that the studio walls must provide.

Figure 16-10 plots the difference between the jet noise curve and the studio noise curve of Fig. 16-9. This is the attenuation that the studio walls must deliver to bring the outside jet noise down to the desired background noise inside the studio.

What kind of construction is required to give the attenuation of Fig. 16-10? This is the subject of Chaps. 17 and 18. The light, broken line in Fig. 16-10 is the sound transmission class STC-60 contour. The class of walls rated as STC-60 (see Chap. 17) offers approximately 60 dB of attenuation at 500 Hz, which is quite effective. Without belaboring the subject, this is the general method of identifying and specifying the internal background noise requirements of studios and other acoustically sensitive spaces. When designing specific structures, attention must be paid to additional factors such as building codes, materials, costs, quality of work, and so on.

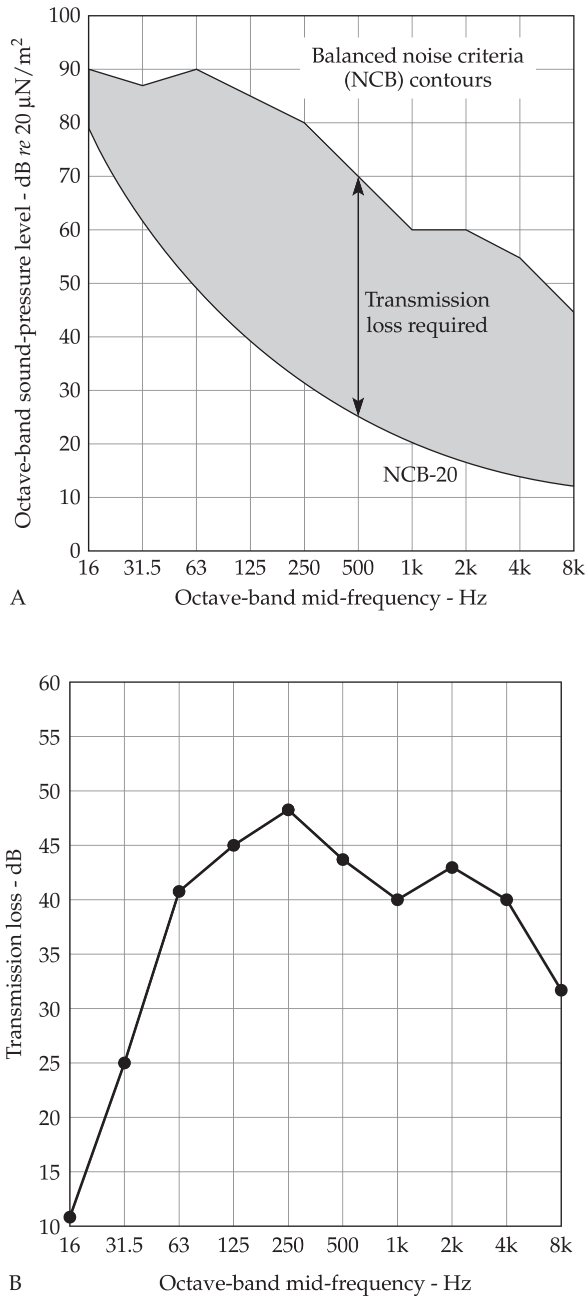


FIGURE 16-8 Use of site measurements and an NCB contour to determine required transmission loss. (A) The TL can be determined by plotting measured site noise levels (top curve) and the desired NCB-20 noise contour (bottom curve). (B) The difference between the curves shows the required transmission loss.

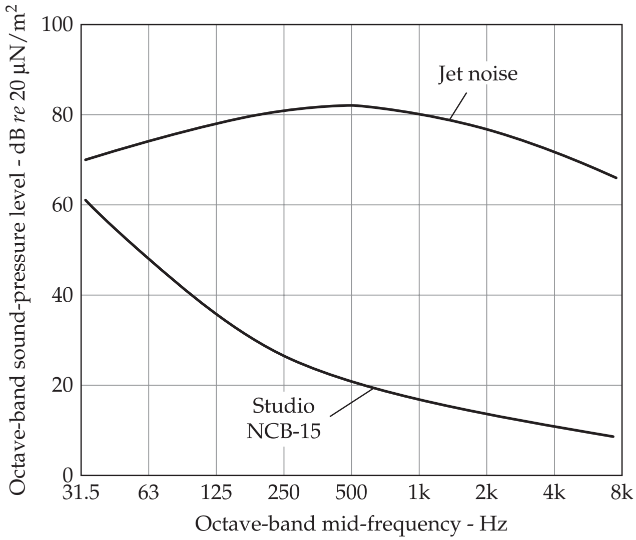


Figure 16-9 A comparison of measured airplane noise versus a studio noise contour of NCB-15.

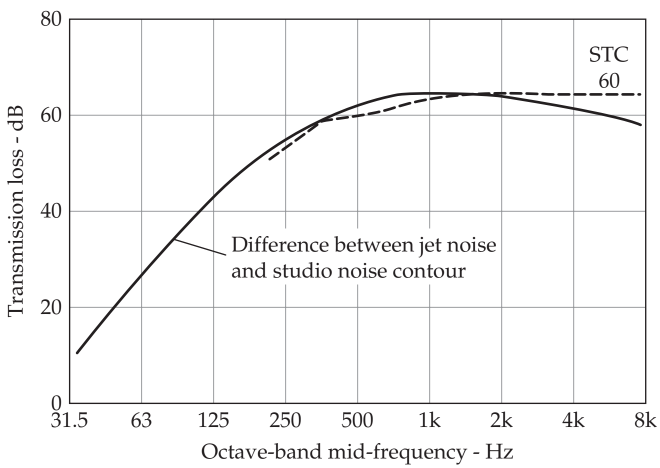


Figure 16-10 Difference between exterior and interior noise showing an STC-60 requirement.

Floor Plan Considerations

As much as the building plans allow, acoustically sensitive rooms should be located away from noise sources. For example, a studio should be located on the side of the building facing away from a busy street. Similarly, within a building, maintenance shops, bathrooms, and HVAC rooms should be located away from a studio. This can be accomplished by placing acoustically neutral rooms such as storage closets in between to act as buffers. Of course, the same considerations apply to vertical locations; for example, an HVAC room should not be placed over a studio. Clearly, windows and doors should not face a noise source such as a busy highway.

Designing within a Frame Structure

A recording studio or other acoustically sensitive space located within a frame structure may have to contend with the noise problem of footfalls. These low-frequency components are often inevitable because they are very difficult to control in lightweight frame structures. If the natural period of vibration of the floor-ceiling structure is about the same frequency as the peak energy of the thuds and thumps of footfalls, the footfall noise will be present and might actually be amplified. Installing carpet on stairs and in hallways will reduce the higher-frequency impact components of footfalls, but the thud and thump low-frequency components will tend to penetrate floating floors, heavy carpet, or other precautionary layers.

If the recording or playback activities are not critical, an extended low-frequency response might not be required. For example, a voice-over studio is concerned only with higher frequencies. In such cases, a high-pass filter can be inserted in the signal path, which would reject both signal components and noise below some frequency, for example, 63 Hz. Extreme isolation measures meant to eliminate these thud and thump problems in frame structures can be an exercise in futility.

Designing within a Concrete Structure

Locating a studio or other acoustically sensitive space within a concrete structure offers some prospect for effective isolation against the low-frequency components of footfall noise. Although expensive, it is possible to build a studio-within-a-studio that offers excellent isolation from airborne and structureborne noise using construction techniques such as massive partitions separated by decoupling.

In most cases, it is not cost-effective to attempt to build an ultra quiet studio. Consider that much audio and visual work is accomplished on a daily basis in environments that are less than perfect. It can be very expensive to get the last few decibels of quiet.

Key Points

- The difference between the noise level external to a room and the desired interior noise level defines the transmission loss that must be achieved through the room design.
- Airborne transmission can be minimized by sealing any air leaks in a partition that would provide an air path through the partition. Structureborne transmission can be reduced by decoupling elements of a partition thus interrupting the transmission path.
- There are five basic approaches to reducing noise: Locate the receiving room in a quiet place, reduce the noise output of the offending source, interpose a sound-isolating barrier between the noise and the room, reduce noise energy within the source room and/or receiving room, minimize both airborne and structureborne noise.

- Sound can pass through even a small crack or aperture in an otherwise solid wall because the air leak has a transmission loss of zero. Airtightness is absolutely necessary to isolate against airborne noises. Similarly, any flanking path will allow sound to travel around a barrier, compromising its transmission loss.
- Transmission loss (TL) is the difference between the sound-pressure level (SPL) on the source side of the barrier, and the SPL on the receiver side. The higher the TL value, the greater the attenuation provided by a barrier.
- The transmission coefficient, notated as τ, is the amount of sound that passes through a material where $\tau = 1 - \alpha$.
- The theoretical mass law predicts that every doubling of mass will increase TL by about 6 dB (in practice, 5 dB is usually achieved). Moreover, any doubling of frequency will also result in a 6-dB increase in TL.
- Around the resonance frequency of a barrier, the TL will drop by 10 to 15 dB below the theoretical level; this is called the coincidence-frequency dip.
- Masses that are separated by an unbridged air cavity are relatively more effective at sound isolation. For example, the combined TL of two identical walls would theoretically be twice the individual TLs.
- Porous absorbers provide little transmission loss, but are useful for reducing reflected or ambient sound levels in a noise-source room.
- Sound transmission class (STC) is a single-number figure used to rate isolation; it describes a series of transmission-loss measurements made at frequency intervals throughout a frequency range.
- Structureborne noise is most efficiently controlled at the source of the noise. Massive partitions offer little resistance to structureborne noise. However, lightweight materials can be used to decouple elements of structures, and are thus effective against structureborne noise.
- The impact insulation class (IIC) is a single value rating that can be used to quantify impact noise in a floor/ceiling.
- Site selection begins with an understanding of the intended purpose of the acoustically sensitive space. Not all spaces demand the same stringency in noise levels.
- A noise survey can yield a distribution curve, giving a statistical picture of the environmental noise at a measurement location for a 24-hour period.
- A variety of descriptors have been devised to measure and evaluate environmental noise. These include L_{eq}, L_{dn}, L_{max}, L_n, and SEL.
- The American National Standards Institute (ANSI) and the American Society for Testing and Materials (ASTM) publish standard methods for assessing environmental noise.
- Balanced noise criteria (NCB) contours are often used to define the noise level within a structure. The NCB can be used to determine what transmission loss is required.
- In most cases, it is not cost-effective to attempt to build an ultra quiet studio. It can be very expensive to get the last few decibels of quiet.

CHAPTER 17

Sound Isolation: Walls, Floors, and Ceilings

The construction of walls, floors, and ceilings is fundamental to any architectural endeavor. When the space within will be acoustically sensitive, those elements take on additional importance. In addition to structural integrity, these partitions must work as sound barriers to isolate the interior space from exterior noise, and to isolate the exterior from the interior sound. To satisfy these acoustical requirements, these structural barriers must be designed and constructed in ways that are considerably different from typical building specifications.

The requirement of sound isolation calls for unique considerations in the ways in which materials are used. In fact, the acoustical demands of room isolation are different from those of interior acoustical treatment. For example, as noted in earlier chapters, porous materials excel as sound absorbers; however, they are quite poor at sound isolation. As another example, the mass of a barrier does not particularly affect its reflective properties, but it greatly affects its transmission loss (TL); the more massive the partition, the better the sound isolation. Overall, sound isolation is harder to achieve at low frequencies and easier at high frequencies. Other factors influence the success of a barrier's sound isolation. For example, two independent barriers will be more efficient than a single barrier. For effective isolation, a barrier must not have any air gaps or leaks; these would allow sound to travel through it with little attenuation. These various factors must all be considered when designing a barrier; for example, an effective wall might comprise two massive partitions that are decoupled, and thoroughly caulked to provide air-tight seals. Doors and windows pose special kinds of problems; these are considered in Chap. 18.

Walls as Effective Noise Barriers

A wall that is effective as a noise barrier is essentially characterized by a high transmission loss, that is, sound energy passing through the wall is significantly decreased. Frequency-response plots show that transmission loss varies with frequency. The graphs of Figs. 17-1 to 17-4 show the measured transmission-loss curves of four common wall structures. For example, the wall in Fig. 17-1 is constructed with 3-5/8-in steel or 4-in wood studs faced with gypsum board ("drywall" or "hardboard") of 1/2-in and 5/8-in thickness.

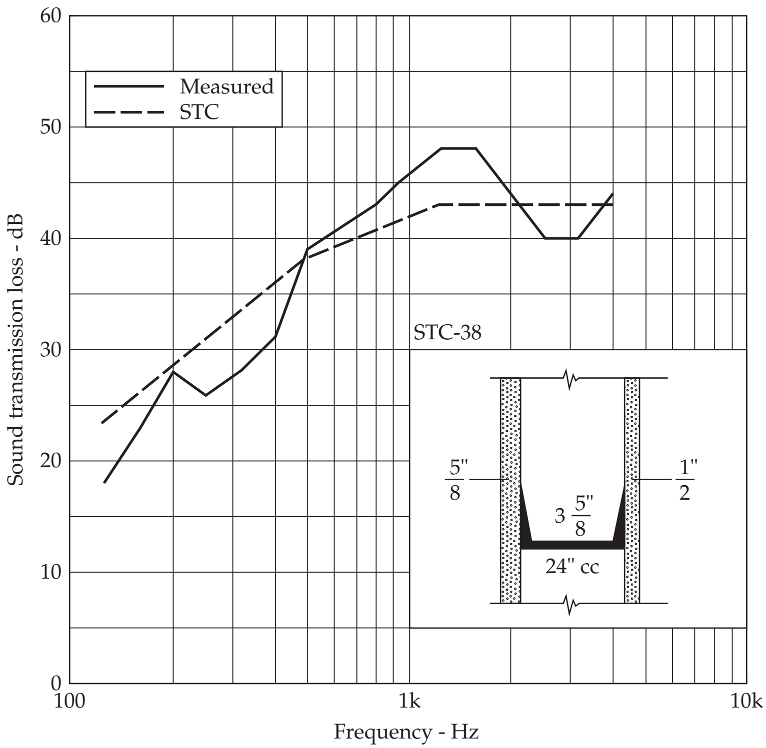


FIGURE 17-1 Transmission loss of a wall constructed of studs and drywall. (*Northwood.*)

The shapes of these transmission-loss curves are certainly not linear. The task of characterizing TL curves has been simplified by the concept of the sound transmission class (STC) rating, a single-number rating of airborne transmission loss derived from a contour curve. The STC contour and rating for each partition are shown in each figure. When evaluating walls with the STC rating alone, it is important to remember the limitations of this single-number rating. STC is described in more detail in Chap. 16.

The Role of Porous Absorbers

Porous sound absorbing materials such as glass fiber are not effective as barriers against sound. They do not increase the transmission loss directly; they are far too light in weight for that. This is true when normal transmission loss is considered, but in some structures, porous materials helpfully absorb sound energy in the cavity, and the added damping decreases the coincidence-frequency dip. Without the glass fiber within the cavity, sounds at and near the resonant frequency of the panels would pass through the wall structure with relatively little attenuation. The glass fiber adds damping to the structure which reduces resonance.

In some wall structures, cavity absorption can improve the transmission loss by as much as 15 dB, principally by reducing resonances in the space between the walls; however, in other structures the effect is negligible. The low-density mineral-fiber batts

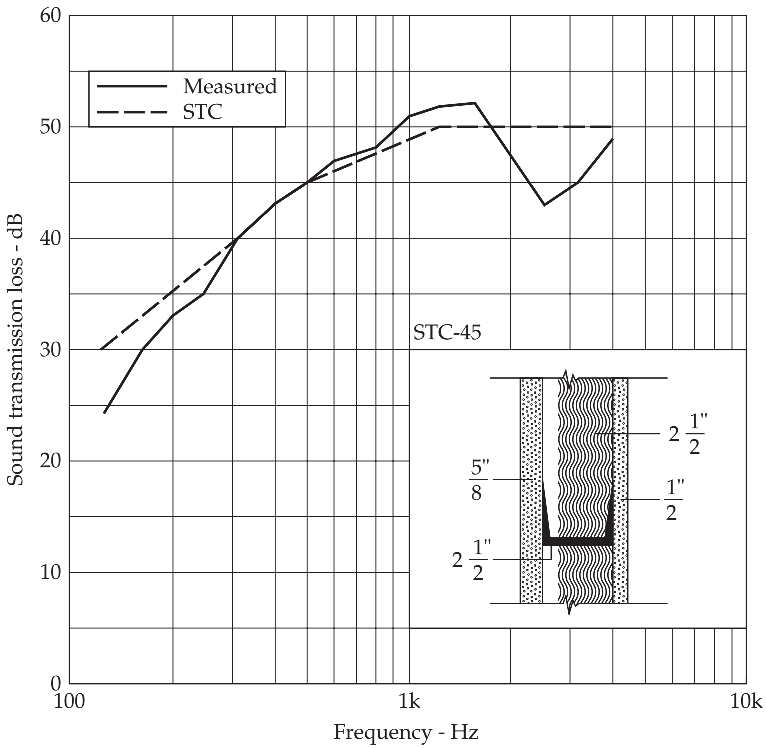


FIGURE 17-2 Transmission loss of a wall constructed of studs and drywall with glass fiber in the interior airspace. (*Northwood.*)

commonly used in building construction are as effective as high-density boards, and they are much cheaper. Mineral-fiber batts within a wall may also meet certain fire-blocking requirements in building codes. Care must be taken to add the glass fiber loosely; if tightly packed, it would tend to couple and thus bridge the panels, decreasing the transmission loss.

The Mass Law and Wall Design

The more massive the partition, the better the sound isolation; this is particularly true at low frequencies. As described in Chap. 16, in practice, sound transmission loss increases about 5 dB for each doubling of surface mass. The surface density is the density (or weight in pounds) of one square foot of partition surface. Transmission loss increases with density and frequency. Figure 17-5 illustrates the 500-Hz mass prediction law as it relates to the four wall structures shown in Figs. 17-1 to 17-4, and three additional wall structures of a general nature for comparison. The black dots represent the sound transmission class ratings of these partitions. Table 17-1 summarizes the data applicable to each of these seven wall structures.

Point 1 shows the performance of the partition of Fig. 17-1 comprising two leaves (one of 1/2-in and the other of 5/8-in drywall plasterboard), with a 3-5/8-in channel

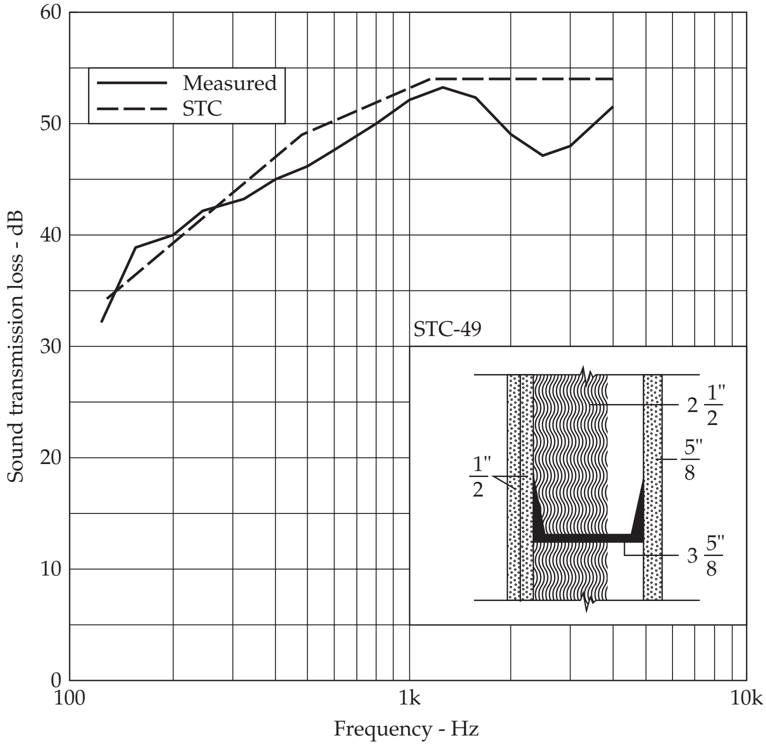


Figure 17-3 Transmission loss of a wall constructed of studs and multiple layers of drywall with glass fiber in the interior airspace. (*Northwood.*)

and with no glass fiber in the interior space. Its surface density (4.8 lb/ft^2) alone would give it a 500-Hz transmission loss of only STC 33. The actual measured value of STC 38 indicates that the two leaves separated by 3-5/8 in perform better than the mass law prediction with the two leaves placed together thus consolidating the mass effect. In other words, the structural form has increased the transmission loss by about 5 dB over that of the mass law prediction.

Point 2 shows the performance of the partition of Fig. 17-2. Even though the surface densities of the partitions of Figs. 17-1 and 17-2 are the same (4.8 lb/ft^2), the latter is rated at STC 45 while the former is only rated at STC 38. This is in spite of the spacing of the two leaves being decreased from 3-5/8 to 2-1/2 in. The reason for this improved performance is that 2 in of glass fiber has been introduced in the interior space. It helps by subduing the resonances in the space, which can tend to decrease partition performance. The glass fiber must be placed loosely in the cavity; tightly packed material can decrease transmission loss.

Point 3 shows the performance of the partition in Fig. 17-3; it indicates still more increase in the STC rating. Surface density has been increased to 7.0 lb/ft^2 by adding gypsum board. Spacing is 3-5/8 in, and glass fiber has been placed in the interior space. The result is an increase to STC 49, which is an increase of 15 dB over the mass law prediction.

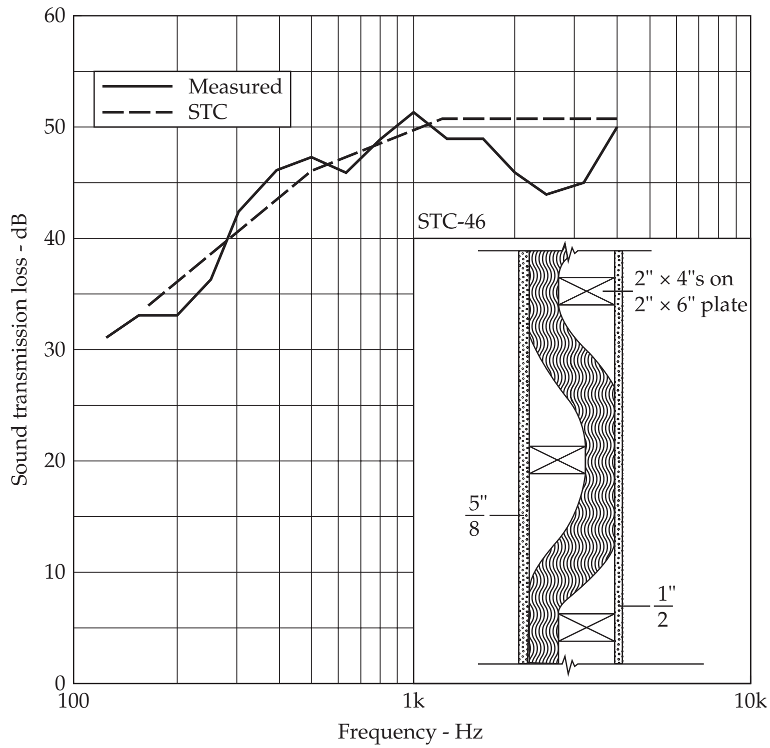


Figure 17-4 Transmission loss of a staggered-stud wall with glass fiber in the interior airspace. (*Northwood.*)

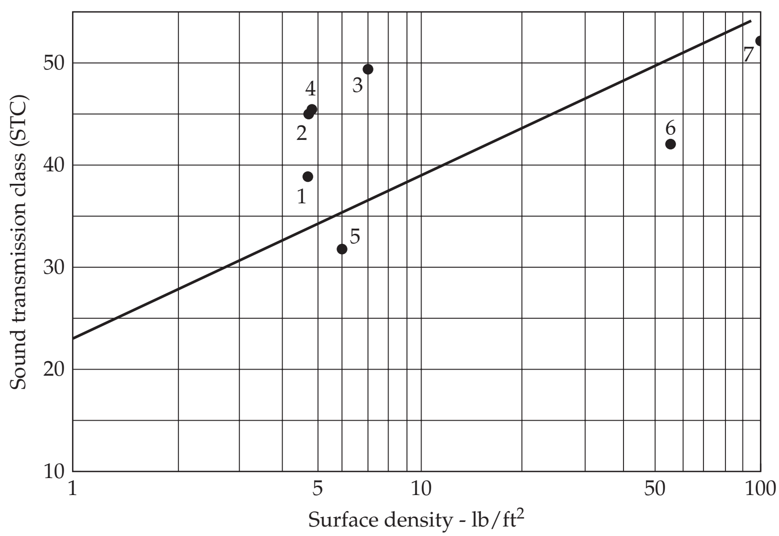


Figure 17-5 A comparison of the STC ratings of seven wall structures relative to the 500-Hz mass-law prediction curve. Structures 1–4 are shown in Figs. 17-1 to 17-4. Structures 5–7 are additional wall structures of a general nature for comparison. Specifications are given in Table 17-1.

From Figure	Point on Fig. 17-5	Leaf A	Leaf B	Surface Density (lb/ft^2)	Leaf Spacing	Glass Fiber	STC	Source
17-1	1	1/2"	5/8"	4.8	3-5/8"	—	38	N*
17-2	2	1/2"	5/8"	4.8	2-1/2"	2"	45	N
17-3	3	1/2" + 1/2"	5/8"	7.0	3-5/8"	2-1/2"	49	N
17-4	4	1/2"	5/8"	4.8	Staggered stud	Yes	46	N
—	5	1/2"	1/2"	4.2	2 × 4" Wood studs	—	32	—
—	6	4-1/2" Brick with 1/2" plaster	—	55.0	—	—	42	—
—	7	9" Brick with 1/2" plaster	—	100.0	—	—	52	—

*Northwood.

Table 17-1 Sound Transmission Loss of Partitions (Data Summary for Fig. 17-5)

Point 4 shows the performance of the staggered-stud partition of Fig. 17-4. This yields STC 46, somewhat short of the STC 49 of Fig. 17-3 in which the surface density was greatly increased.

Point 5 shows the performance of a simple wall of 2-in × 4-in wood studs with 1/2-in drywall attached to each side. The performance is below that of the mass law prediction of STC 32 for a surface density of 4.2 lb/ft^2. Point 2 shows that a partition close to the same surface density can give STC 45 with the addition of a heavier gypsum board sheet and some glass fiber in the space.

Points 6 and 7 represent data for brick walls. They have been added to show examples of heavy structures that underperform the mass-law prediction. Relative to these heavy walls, the lightweight structures considered above can be designed to provide high STC ratings with much less mass. This demonstrates the relative efficiency of more sophisticated, lightweight partition designs as measured by their STC ratings. However, heavy walls usually provide greater isolation at low frequencies. It is important to note, as discussed in Chap. 16, that the STC rating is intended primarily for use in the speech-frequency region. A high STC rating will not ensure good low-frequency noise attenuation. For this reason the STC rating is not intended for use with external partitions where, for example, traffic noise may be an issue. Thus the question of whether a lightweight partition or a heavy partition with similar STC ratings is better depends entirely on the type of acoustical application.

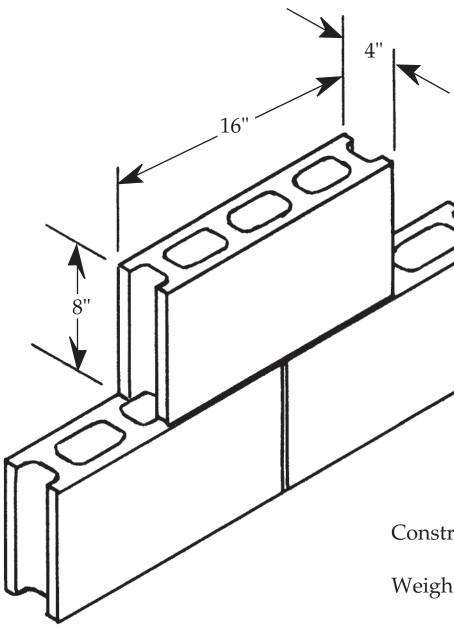
Separation of Mass in Wall Design

When designing barriers it is important to remember that how materials are arranged can be more important than the quantity of materials used. In particular, the technique of separating masses is effective at promoting isolation. The following examples of wall construction demonstrate that multiple partitions designed for separation of mass will outperform single partitions particularly if construction permits each partition to behave independently of the other. The STC rating is used to evaluate the performance of each wall.

Figure 17-6 shows the measured performance of a 4-in concrete-block wall as a sound barrier. A concrete-block wall provides good sound isolation with a mild coincidence-frequency dip. It is interesting to note that plastering both sides increases the transmission loss of the wall from STC 40 to 48. Figure 17-7 shows a considerable improvement in doubling the thickness of the concrete-block wall. In this example, the STC 45 rating is improved to STC 56 by plastering both sides. By adding gypsum-board partitions to both sides of a concrete-block wall, STC can be increased to about 70.

Figure 17-8 shows the very common 2-in × 4-in frame construction with 5/8-in gypsum-board covering. This kind of partition can also offer good isolation, albeit with a coincidence-frequency dip; the two sides of the partition provide separation of mass. In this construction, the STC of 34 without glass fiber in the wall cavity is improved only to STC 36 by filling the cavity with glass-fiber material, a minor improvement that would probably not justify the added cost. Of course, glass fiber may be warranted for thermal insulation or other reasons.

Figure 17-9 shows an acoustically efficient and inexpensive staggered-stud wall construction. The studs are alternately connected to the panels, but tied to the same



Construction - 4" concrete block

Weight - 18 lb/ft^2 (plain)
30 lb/ft^2 (plastered both sides)

Sound transmission class -
STC 40 plain
STC 48 plastered both sides

Figure 17-6 Concrete-block (4-in) wall construction. (*Solite Corporation.*)

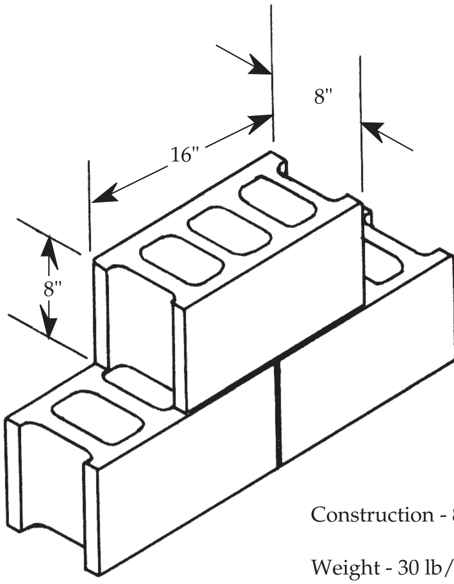
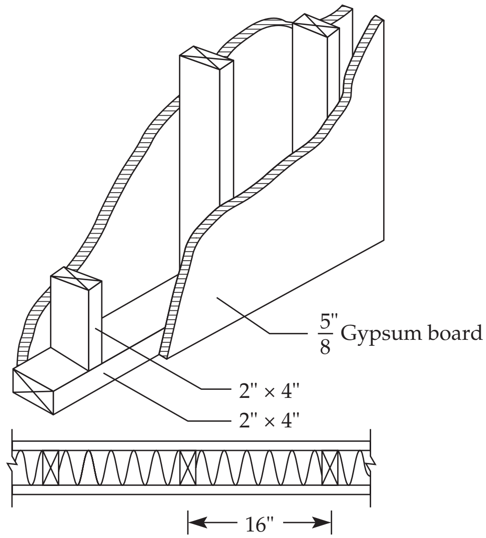


FIGURE 17-7 Concrete-block (8-in) wall construction. (*Solite Corporation, and LECA.*)

headers and footers. Here the isolation provided by the inherently low coupling between the two independent wall diaphragms is further improved by filling the space with glass fiber. Ideally, the top and bottom of the partition should be decoupled from the ceiling and floor. Attaining the full STC 52 rating would require careful construction to ensure that the two wall surfaces are truly independent and not acoustically compromised by back-to-back electric conduits, outlet boxes, or other devices placed in the cavity.

Figure 17-10 shows a double-wall construction. The two walls are entirely separate, each having its own 2-in × 4-in plate. Without glass fiber this wall is only marginally better (STC 43 vs. STC 42) than the staggered-stud wall of Fig. 17-9, but by filling the inner space with glass-fiber insulation, STC ratings as high as 58 are possible. In some designs, QuietRock is specified instead of gypsum board. QuietRock is an internally damped wallboard; it is more expensive than gypsum board, but is superior in terms of STC rating.

To further demonstrate the principle of separation of masses, consider the four gypsum-board partitions in Fig. 17-11. Each wall uses double-plate construction with wood studs, insulation, and no cross bracing. Wall (A) is probably the most common construction, yielding an STC of 56 in this particular test. Wall (B) adds an internal gypsum-board layer, but this increases coupling, and the STC drops to 53. Wall (C) adds another internal layer, and STC drops further to 48. Mass is increased, but the separating air cavity is degraded by the additional layers. Wall (D) provides the best



Construction - Standard-stud partition

Weight - 7.3 lb/ft^2

Sound transmission class -
STC 34 without glass fiber
STC 36 with $3\frac{1"}{2}$ glass fiber

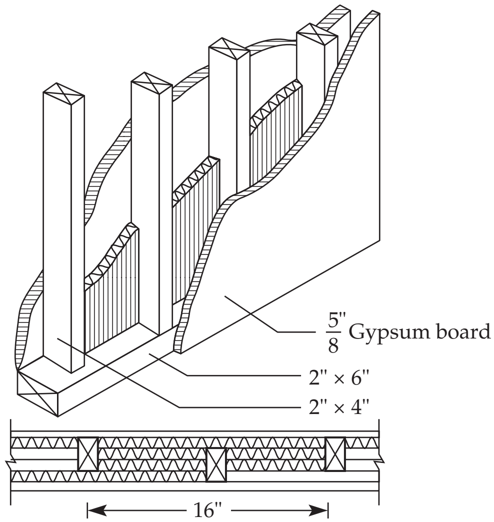
Figure 17-8 Standard-stud partition. (*Owens Corning Corporation.*)

sound-isolating performance of this group. In this design the additional layers are moved to the outside surfaces of the partition; STC is much improved at 63. In all cases, for best performance, gypsum-board joints should be offset and carefully taped.

Wall Design Summary

Figure 17-12 summarizes seven simple ways to achieve efficient isolation in a basic wall structure: (A) increase weight; (B) increase spacing of leaves; (C) staggered studs; (D) leaves of different weight; (E) resilient strips; (F) glass-fiber blankets; (G) perimeter caulking. Figure 17-12A demonstrates that mass is needed for a wall to be an effective barrier. This mass can easily be added to a wall in the form of sheets of gypsum board using screws or adhesive. Table 17-2 lists the surface density of common sound barrier walls, which vary between 3 and 11 lb/ft^2.

Mass is needed for isolation. In addition, the positioning of the mass and in particular the spacing between masses is important. If the spacing between the two leaves of a wall is eliminated or made very small, the transmission loss will be reduced to mass functioning as though it were a single-leaf wall of combined weight. When the two leaves are spaced widely apart and decoupled (Fig. 17-12B), the transmission loss of this double-wall construction approaches the sum of the two as though two separate



Construction - Staggered-stud partition

Weight - 7.2 lb/ft^2

Sound transmission class -
STC 42 without glass fiber
STC 46 to 52 with glass fiber

FIGURE 17-9 Staggered-stud partition. (*Owens Corning Corporation.*)

walls existed. A practical wall spacing of about 4 in is somewhere between the two extremes. Transmission loss is increased only a small amount by increasing the spacing from 4 to 6 in, but this is one factor in the significant performance increase of staggered-stud walls.

The two sides of staggered-stud walls (Fig. 17-12C) are only coupled at the perimeters. They are normally built on 6-in plates instead of the usual 4-in plate. Though nominal in itself, the added spacing gives some increase in transmission loss. Making the mass of the two leaves different (Fig. 17-12D) makes the resonant frequencies of the two leaves different. If these frequencies coincided, the dip in the transmission-loss curve would be more pronounced at that frequency. When the leaf masses are different, the resonant frequencies are offset and this tends to smooth the transmission-loss curve. For example, one gypsum-board face could use 5/8-in board while the other uses a double layer of 5/8-in boards.

Figure 17-12E illustrates the use of resilient strips for the purpose of providing some decoupling and isolation of a layer of gypsum board from the wall itself. This also helps make the resonance points of the two leaves appear at different frequencies, avoiding deep dips in the transmission-loss curve. Some resilient channels are S-shaped, with one end secured to the wall and the other secured to the gypsum board. When using resilient strips, care must be taken to ensure that isolation is preserved; for example, the proper screws should be used. Also, care must be taken when mounting cabinets or

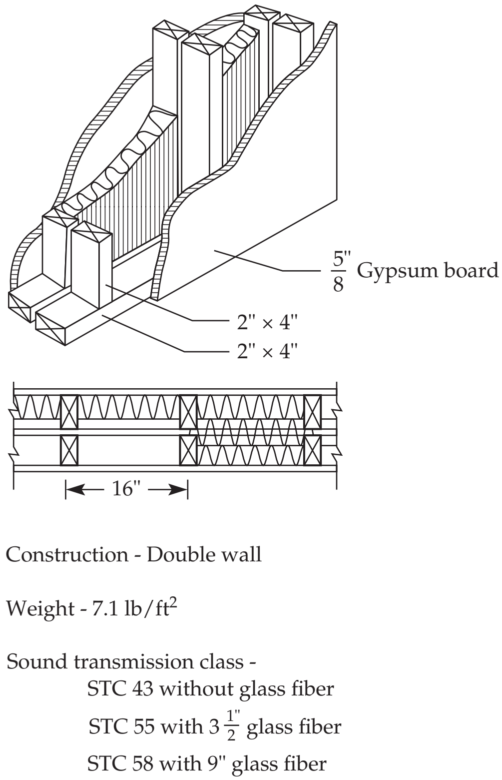


FIGURE 17-10 Double-wall partition. (*Owens Corning Corporation.*)

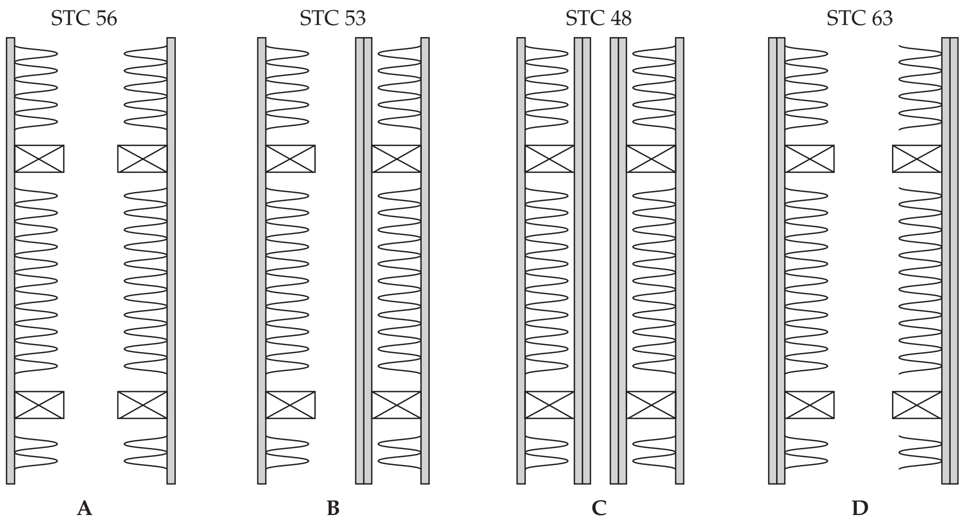
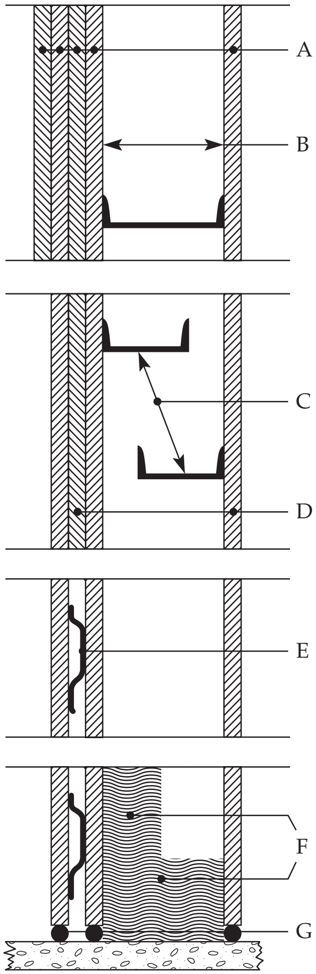


FIGURE 17-11 Four designs for gypsum-board partitions with glass-fiber fill. (A) STC 56. (B) STC 53. (C) STC 48. (D) STC 63. (*Berger and Rose.*)



Ways to improve wall isolation:
A. Increase weight
B. Wider spacing of leaves
C. Staggered studs
D. Leaves of different weight
E. Resilient strips
F. Glass-fiber blankets
G. Perimeter caulking

FIGURE 17-12 Seven ways to improve sound isolation of a wall.

other attachments to the wall; for example, screws that are too long will bridge the isolating strip by solidly attaching the gypsum board to the underlying stud.

Thermafiber, a type of building insulation, can be placed in the interior cavity between the two leaves of a wall (Fig. 17-12F). This is an economical way to increase the transmission loss of many wall designs. When low-density glass-fiber insulation is used, the cavity should be completely filled; for every doubling of insulation thickness STC increases by about 2 dB. Glass fiber should not be packed too tightly because it could conduct sound energy through the partition. For the same reason, mineral fiber (which is stiffer) should not completely fill the cavity. Placing glass fiber in the interior cavity is particularly helpful in staggered-stud walls in which the faces are acoustically decoupled from each other. Thermafiber is a product of Owens Corning Corporation.

Figure 17-12G recalls the importance of sealing all edges of a partition. The importance of liberal use of sealant cannot be overstressed. It is an inexpensive way to achieve

Leaf A	Leaf B	Surface Density (lb/ft^2)
Unfilled Steel Stud Partition		
3/8"	3/8"	3.0
1/2"	1/2"	4.0
5/8"	5/8"	5.0
3/8" + 3/8"	3/8" + 3/8"	6.0
1/2" + 1/2"	1/2" + 1/2"	7.5
5/8" + 5/8"	5/8" + 5/8"	10.0
Unfilled Wood Stud Partition		
3/8"	3/8"	4.0
1/2"	1/2"	5.0
5/8"	5/8"	6.0
3/8" + 3/8"	3/8" + 3/8"	7.0
1/2" + 1/2"	1/2" + 1/2"	8.5
5/8" + 5/8"	5/8" + 5/8"	11.0

TABLE 17-2 Surface Densities of Common Partitions (Densities Listed Include Weight of Studs)

added transmission loss in a wall. Normal framing always results in cracks; they might not be important in nonacoustical walls, but they are very important in walls in acoustically sensitive spaces. Sound penetrating cracks under, over, and around a wall can easily mean the loss of 10 points from an STC rating. Figure 17-13 illustrates both the problem and the solution.

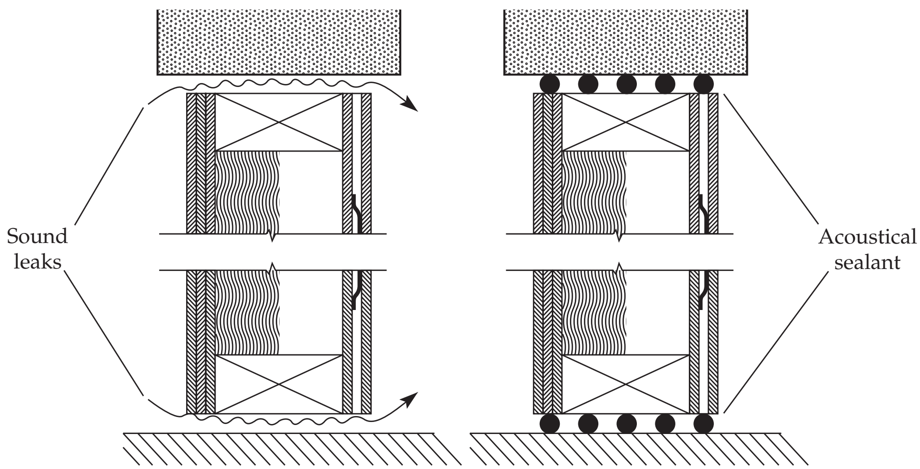


FIGURE 17-13 The importance of sealing wall elements. (A) Sound penetrates through air leaks in the partition. (B) Sound leaks are stopped by acoustical sealant.

For example, before the 2-in × 4-in or 2-in × 6-in plate is bedded on the floor, several strips of acoustical sealant should be applied to the floor and/or the plate to seal the ever-present crack between the concrete and the plate. Every edge-framing member should be so sealed. Additional sealant should be applied to the periphery of the gypsum board wall. The sealant used should be of the nonhardening variety, often referred to as "acoustical sealant."

Improving an Existing Wall

It is often necessary to adapt an existing space for a studio or other acoustically sensitive room. Existing walls are rarely built for high transmission loss. One way to improve an acoustically weak existing wall is to construct a new partition on either side of it. The design and construction of that additional wall will determine whether it succeeds as a sound barrier. Figure 17-14 details a practical approach to such a wall improvement.

The airspace gap between the partitions should be as wide as possible; 1 in is a minimum, but the gap should be maximized as floor area allows. The additional wall embodies the features that have been described, such as Thermafiber insulation in the airspace and multiple layers of gypsum board. The outer layer of gypsum board could be mounted on metal resilient strips, and more than two layers of gypsum board could be included to yield an additional improvement in STC. Such steps may be unwarranted because of flanking sound traveling around the barrier. As another measure, an existing wall's transmission loss can be increased by simply adding a 1/2-in gypsum-board layer to each side of the wall. When possible, seams in the new layer should not coincide with seams in the original layer; all seams should be caulked to prevent sound leaks. This technique preserves the airspace gap between the original leaves of the existing wall, and adds weight to the leaves.

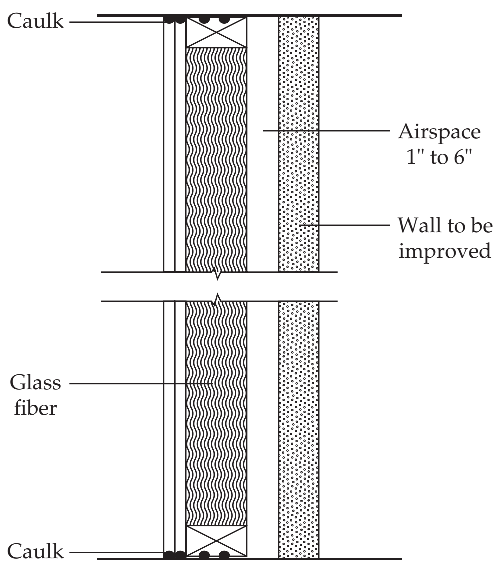


Figure 17-14 Improving isolation of an existing wall by adding an additional wall.

Material	Attenuation (dB/100 ft)
Iron	0.3 to 1
Brickwork	0.5 to 4
Concrete	1.0 to 6
Wood	1.5 to 10

Table 17-3 Attenuation of Longitudinal Waves (*Harris.*)

Flanking Sound

Solid materials are efficient carriers of sound. You can place an ear on the rail of a railroad track and hear an oncoming train that is several miles away, but still not audible through the air. Similarly, in a high-rise concrete structure, elevator sounds and other noises can be carried throughout the structure with very little loss. Wooden structures are also good conductors of sound. Table 17-3 compares the attenuation of sound in iron, brickwork, concrete, and wood. In concrete, for example, sound can travel 100 ft with an attenuation of only 1 to 6 dB.

Structure carries sound from one point to another, but that sound can radiate into an otherwise quiet space largely because walls and floors act as diaphragms. Instead of the noise staying in the beams and columns of the structure, it vibrates the walls and floors, reradiating sound into the air within the rooms. Because of this, protection from both airborne and structureborne sound must be considered. Figure 17-15 illustrates the travel of noise through structure to walls which radiate into the space of a protected area.

The specific case of a frame structure is shown in Fig. 17-16. Flanking sound can travel by a combination of airborne and structureborne paths from the noisy space to the protected area through an attic space or a subfloor space. Such flanking sound can easily nullify the cost spent on improving a shared wall.

Gypsum Board Walls as Sound Barriers

As noted, common wall construction with studs and gypsum board can provide relatively good sound isolation. Isolation can be greatly improved by using double layers of gypsum board; layers should be offset so that seams are staggered. When using wood studs, multiple layers should preferably be secured with viscoelastic adhesives instead of rigid adhesives or screws. Isolation of wood-stud walls can be improved by mounting the gypsum board on resilient channels instead of directly on the studs; the gypsum board is screwed to the channels and the screws are not allowed to touch the studs.

Local building codes may dictate metal studs; because of their ability to flex, metal studs provide slightly better sound isolation compared to wood studs. With metal-stud walls, isolation is improved when gypsum board is screwed to the metal studs instead of using adhesive; a line of adhesive stiffens the flanges whereas individual screws do not. Care must be taken so that the drywall will not vibrate against the metal studs causing audible buzzes; this can be accomplished by increasing the number of screws

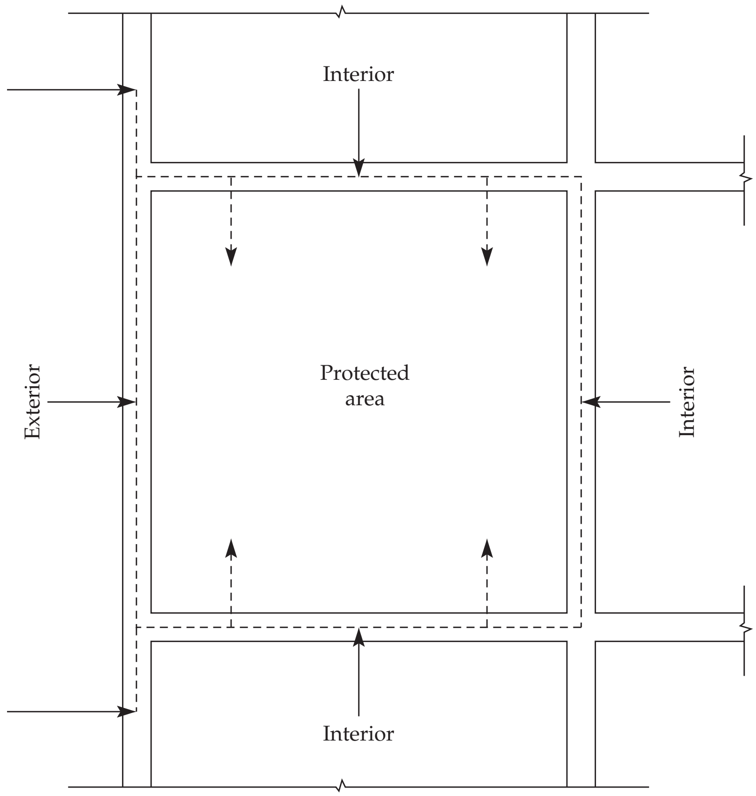


Figure 17-15 Structureborne sound intrusion.

and laying a bead of nonhardening caulk along each stud. As with any acoustical construction, any sound leaks should sealed with caulking and flanking paths must be eliminated.

Masonry Walls as Sound Barriers

Brick and concrete block walls can provide very effective sound isolation. These materials are heavy, and the mass law favors weight. Masonry walls provide relatively more isolation than lightweight walls at low frequencies; therefore, they are superior in most music applications. However, the weight of concrete blocks varies, thus some walls provide better isolation than others. Isolation can be increased by adding well-tamped sand or mortar in concrete block cells. Isolation can also be increased by adding a gypsum board layer or ideally by building a double concrete block partition where the walls are acoustically decoupled. Placing glass fiber or other sound absorption in wall air cavities can also increase isolation. Masonry walls do have a drawback; they are efficient transmitters of impulse noise through structureborne transmission. Therefore, care must be taken to isolate impulsive sound sources from a masonry wall.

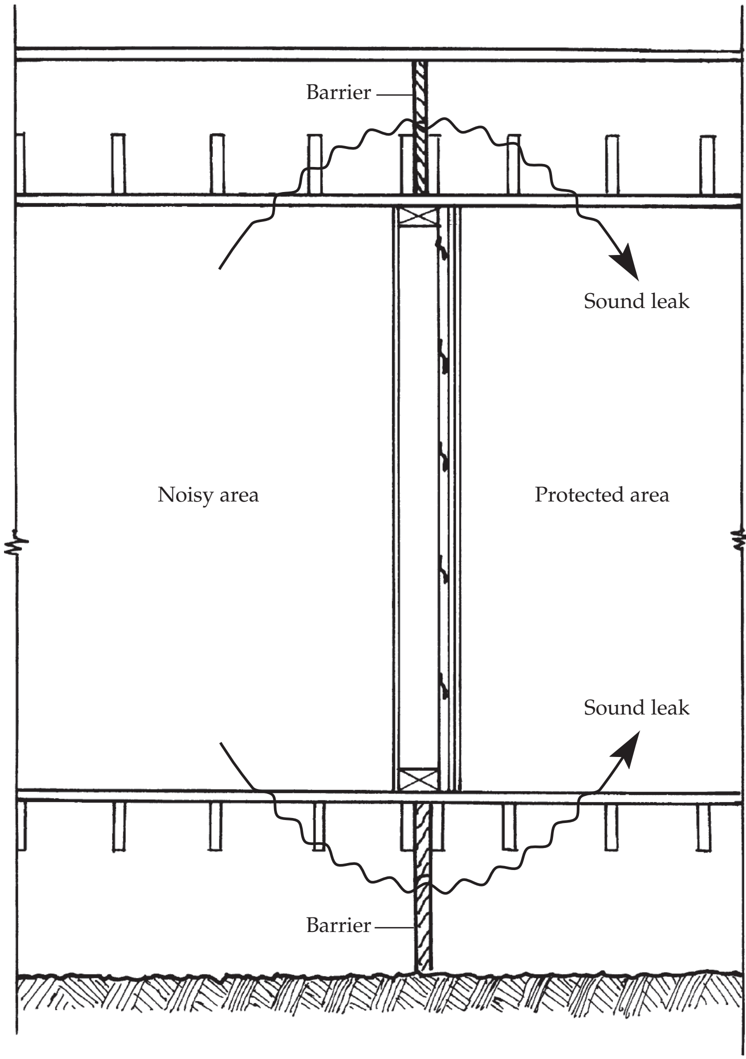


FIGURE 17-16 Flanking paths and barriers.

The performance of brick and concrete block walls is summarized in Table 17-4. The STC of brick walls can be equaled by frame walls, but plastered concrete block walls, with their STC ratings of 57 and 59, are difficult to equal with frame construction. Circumstances often dictate which wall types are most practical, both from acoustical and construction standpoints.

The smooth measured transmission-loss curves of both unplastered single- and double-leaf concrete block walls are shown in Fig. 17-17. A comparison with the concrete block walls of Table 17-4 shows that plastering is more effective on the single-leaf wall (an increase of 6 dB) than the double-leaf wall (no increase).

Wall Description	Surface Density (lb/ft²)	STC	Reference
Brick wall, 4-1/2" plastered both sides	55	42	Table 17-1
Brick wall, 9" plastered both sides	100	52	Table 17-1
Double 6" concrete block walls spaced 6 in	100	59	Figure 17-17
Single 12" concrete block wall	100	51	Figure 17-17

TABLE 17-4 Summary of the Acoustical Performance of Masonry Sound Barriers

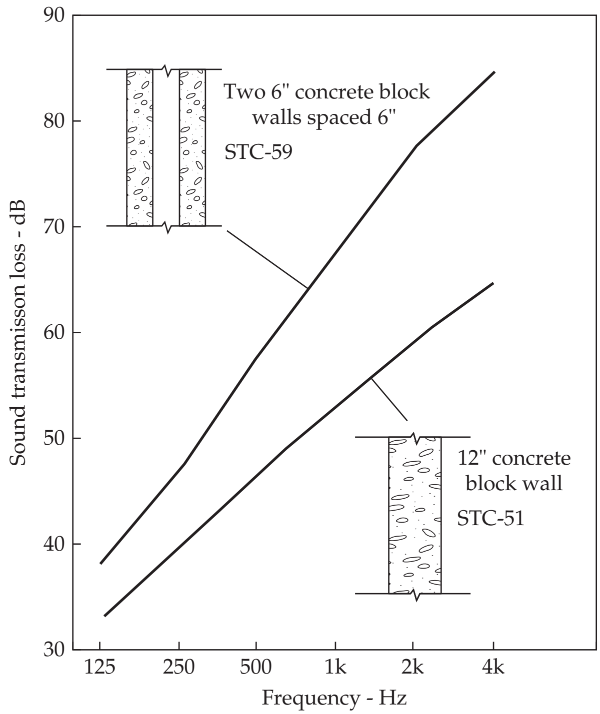


FIGURE 17-17 The transmission loss of concrete walls. (*Egan.*)

Three examples of brick walls are shown in Fig. 17-18. An 8-in thick brick wall provides good sound isolation, but performance can be significantly improved by adding layers to one side. These layers comprise vertical wood furring strips, resilient metal channels, and 1/2-in gypsum board. Voids are filled with 2 in of glass fiber. The added layers can improve transmission loss by 20 dB or more at higher frequencies. However, a design using two 4-in brick walls separated by a 4-in airspace provides isolation that

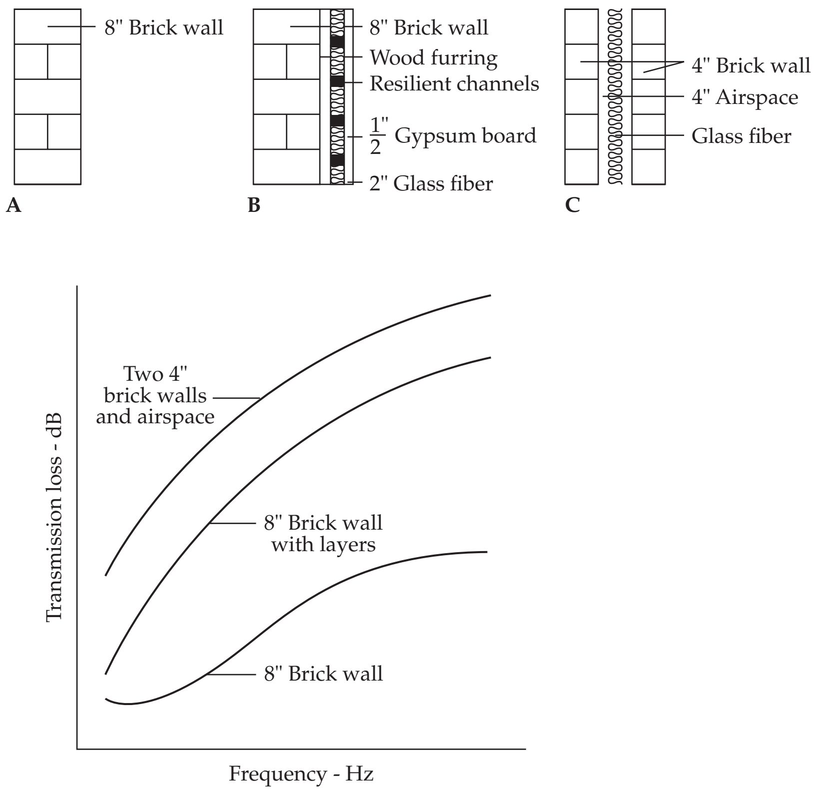


Figure 17-18 The transmission loss of masonry walls can be improved by adding a layer or by constructing two walls. (A) An 8-in brick wall provides good sound isolation. (B) Added layers improve transmission loss at higher frequencies. (C) Two 4-in brick walls separated by a 4-in airspace provide superior isolation.

is superior to the other two designs. Sound absorption such as a glass-fiber blanket should be placed in the cavity between the two walls. This two-wall design is simple, but its total depth of 12 in is a drawback. The double wall design demonstrates the effectiveness of separation of masses; its total weight is the same as one 8-in wall, but it decidedly outperforms it. For example, at high frequencies, its transmission loss may be as much as 35 dB greater.

Weak Links

The performance of sound-isolating partitions is very much affected by the weak-link principle. No matter how high the transmission loss of a barrier might otherwise be, it can be severely compromised by any lower-performance breach, even if that breach has a relatively small surface area. In fact, when a weak link is present, the overall isolation of the barrier will be close to that of the weak link. As noted in Chap. 16, any air leak in a partition can be very problematic. For example, Table 17-5 shows how transmission loss is degraded by air leaks of varying size in a wall with an original TL of 45 dB.

Percent of Wall Area Having Air Opening	Resulting Wall TL (dB)	Resulting Decrease in TL (dB)
0.01	39	6
0.1	30	15
0.5	23	22
1	20	25
5	13	32
10	10	35
20	7	38
50	3	42
75	1	44
100	0	45

Table 17-5 Reductions in Transmission Loss from Openings in a Wall with Initial TL of 45 dB (*Cowan.*)

For example, if the wall area is 100 ft^2, an air opening measuring 14.4 in^2 (0.1% of surface area) would decrease the wall's TL from 45 dB to 30 dB. A 1/4-in space under a door would create an opening of about this size.

The installation of electrical fixtures such as outlet boxes and studio microphone panels must be done properly, otherwise, a wall's isolation can be compromised. As noted, fixtures should not be mounted on exactly opposite positions (back-to-back) on a wall. Sound can leak through openings. Rather, boxes should be staggered apart; also, clearly, the boxes should be tightly caulked. When possible, electric boxes and other services should be surface mounted.

Doors and windows can also easily undermine the acoustical integrity of a partition. For example, a brick wall might have a TL of 50 dB. When a window with a TL of 20 dB and occupying a surface area of about 10% of the wall is incorporated in the wall, the new TL may be just 30 dB. This illustrates the difficulty in designing composite barriers. To avoid weak links, all the components of the barrier such as windows and doors must provide a transmission loss close to that of the wall itself; for example, the difference in TL should be less than 5 dB. This complicates the design and increases the costs of windows and doors. The problem of composite barriers, and the design of windows and doors, is considered in Chap. 18.

Summary of Wall STC Ratings

Figure 17-19 summarizes the points made in this chapter regarding economical ways in which the transmission loss of a wall might be increased. Mass is the most vital element of any noise barrier. In Fig. 17-19A, the mass of a partition is low, but it can be increased easily by adding multiple layers of gypsum-board panels. These can be affixed with screws or adhesive.

In Fig. 17-19B, an increase in spacing of the two leaves of a partition is achieved by staggered studs. This will increase its STC value. In Fig. 17-19C, staggered studs of metal or wood are used to increase the spacing between the two leaves. In addition,

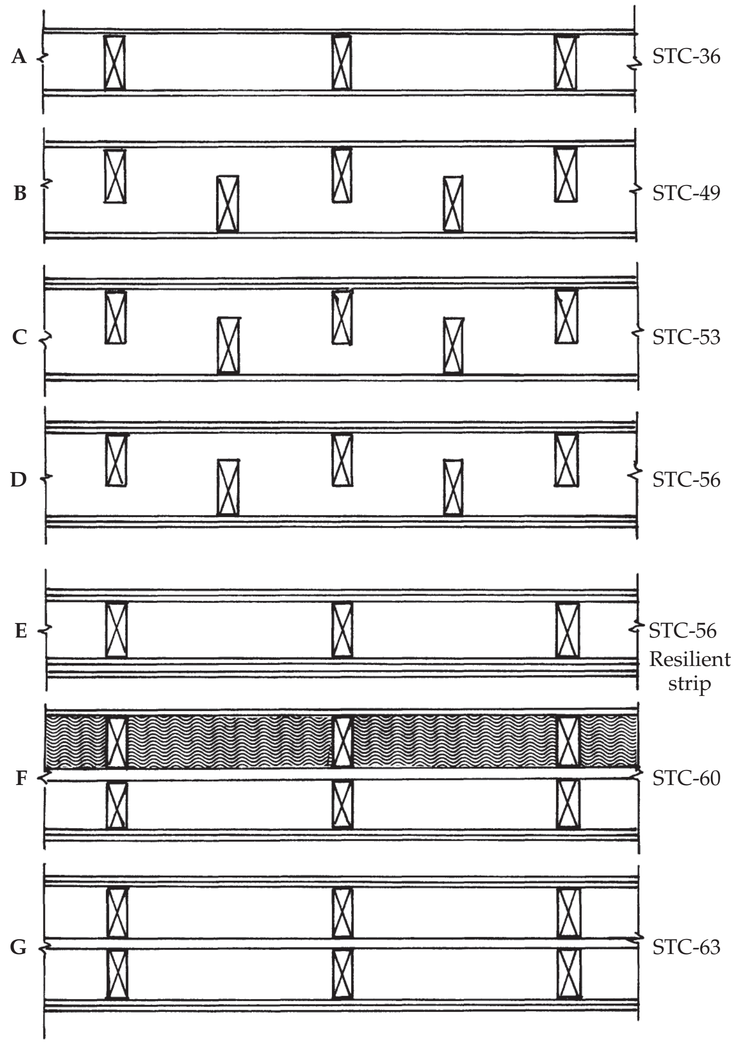


Figure 17-19 The STC ratings for various wall structures.

another layer of gypsum board has been added to one side to make the surface densities of the two sides different. This places their resonances at different frequencies, smoothing the transmission loss of the wall. In Fig. 17-19D, the surface density of the two leaves is made different by using layers of wall board of different thicknesses on each side. In Fig. 17-19E, the use of resilient strips is suggested. If one face of a wall is mounted resiliently to the structure, the STC will be increased significantly. In Fig. 17-19F, we see that a small improvement can be expected by using glass fiber in a cavity space, or by increasing its thickness. The improvement obtained by adding glass fiber is modest, but is inexpensive and worth the effort. In Fig. 17-19G, we see the use of two completely separate walls. In all arrangements, proper sealing is most important. Copious use of caulking material could be the single most important step in the entire process.

The STC rating, which attempts to give a practical and simple shortcut for characterizing barrier performance, has been widely accepted, but its limitations must always be kept in mind. The ultimate test of any noise barrier system, which can be made only when the studio is completed, is noise level measurements within the studio itself.

Floating Floors

As noted, the best way to reduce the transmission of structureborne noise is to decouple one structure from another; any discontinuity will help interrupt the transmission path. A decoupled "room-within-a-room" construction technique is often used to minimize the problem of structureborne noise. The design begins with floating floors, upon which are built floating walls and floating ceilings, all decoupled from the potentially vibrating structural room construction. A floating floor is an additional floor raised and decoupled from the structural floor. The floating floor may be built of wood or concrete. In either case, it is isolated from the structural floor by using springs, isolation blocks, or other elastic elements. It is important to verify that the structural floor can withstand the added weight of the floating floor. A well-constructed floating floor, particularly a floating concrete slab, can provide a high impact insulation class (IIC) value as well as a high STC value. Impact insulation class is discussed later.

A floating floor acts as a mechanical low-pass filter system, attenuating low-frequency noise and vibration above its cutoff frequency. Clearly, the cutoff frequency should be lower than the frequency of any undesired noise or vibration. The cutoff frequency of the system can be estimated as:

$$f_0 = \frac{3.13}{d^{1/2}} \tag{17-1}$$

where f_0 = cutoff frequency, Hz
d = static deflection of the isolator, in or cm

Note: In metric units, change 3.13 to 5.

The static deflection is the decrease in the height of the isolating pads or springs when the floating floor is placed on them. The cutoff frequency should be at least half (preferably one-quarter) of the lowest frequency to be attenuated.

Construction of a poured concrete floating floor begins by placing compressed glass-fiber board (perhaps 1-in thick) around the perimeter of the room to isolate the floating floor from the structure, as shown in Fig. 17-20. A strip of wood (perhaps 1/2-in thick) is set on top of the perimeter board. Compressed glass-fiber cubes, molded neoprene cubes, or other isolation means are distributed across the structural floor so that compression of the mounts and the loading are matched. Sheets of plywood (perhaps 1/2-in thick) are laid on the mounts and the edges are fastened together with metal straps and screws. The plywood is covered by a plastic-sheet vapor barrier, overlapping the edges by at least a foot, and running up the perimeter boards. When preparing a form for a poured floating floor, it is essential to ensure that concrete from the pour cannot leak down to touch the structural floor; this would "short circuit" the intended discontinuity and seriously compromise the performance of the floating floor.

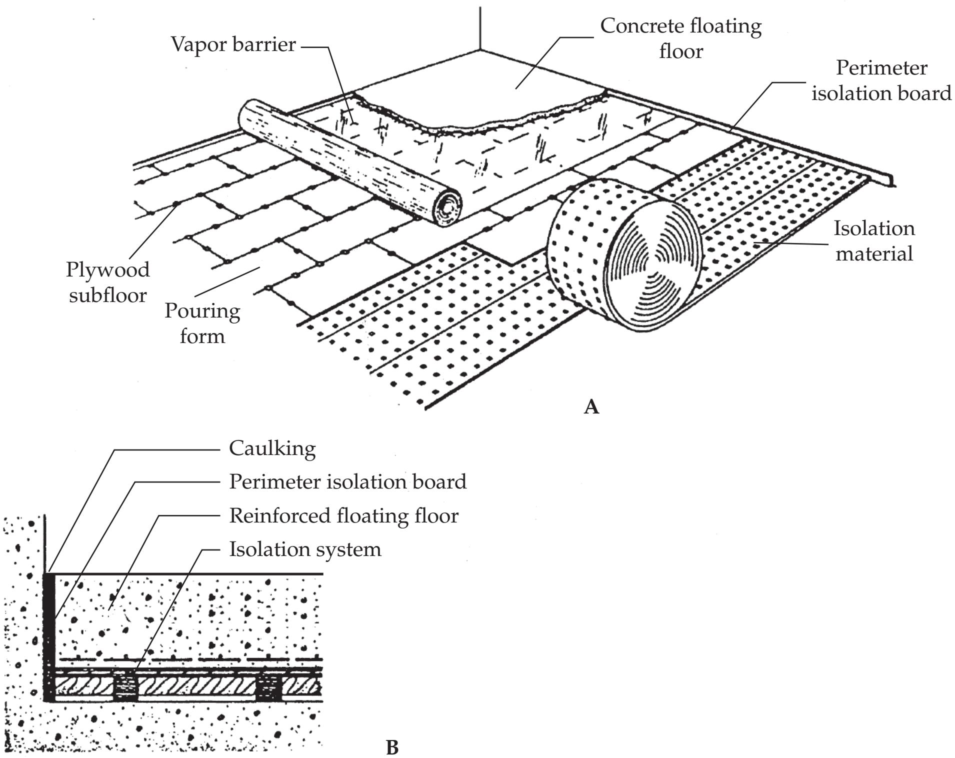


Figure 17-20 Example of a concrete floating floor construction. (A) Overall floor assembly. (B) Detail of floor perimeter. (*Kinetics Noise Control.*)

Welded screen reinforcing mesh is blocked to be positioned in the center of the floating slab. The floating slab is then poured. Care is taken to ensure that the floating slab does not touch the structural elements of the room. After the slab is cured, the plastic sheet can be cut, and the wood perimeter strip removed. The resulting perimeter gap is sealed with a nonhardening sealant. In some designs, the isolation pads have metal housings and an internal screw. The floating slab is poured directly onto the vapor barrier resting on the structural slab, and after it is cured, the floating slab is raised using floor jacks from above. When constructing any floor, particularly a concrete floating floor, it is important to anticipate placement of conduits and other infrastructure that may lie beneath the floor.

A floating floor can also be constructed of plywood, and placed over a concrete or plywood structural floor, as shown in Fig. 17-21. Construction of the floating floor begins by placing compressed glass-fiber blocks or boards, or other isolation means on the structural floor. The perimeter of the floating floor is isolated from the surrounding walls. Wood sleepers are placed over the pads and glass fiber is placed between the sleepers. One or more layers of plywood panels are secured to the sleepers. This type of floor can be constructed in residential housing, and provides good performance. Carpeting will improve the IIC value. However, the construction is not particularly good at isolating low-frequency impact noise.

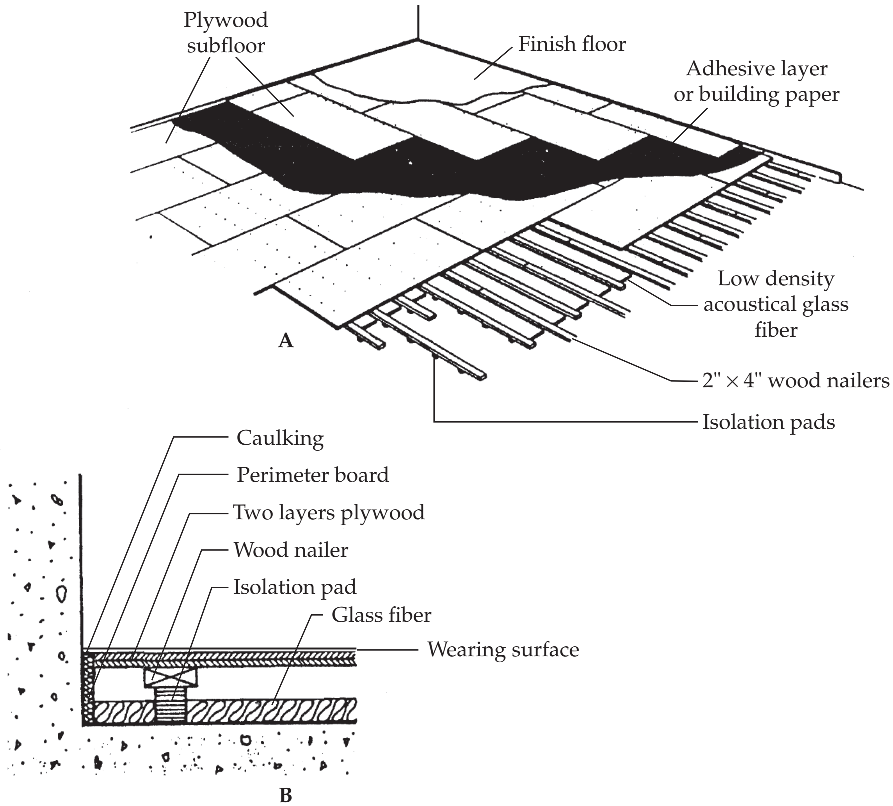


FIGURE 17-21 Example of a wood floating floor construction. (A) Overall floor assembly. (B) Detail of floor perimeter. (*Kinetics Noise Control.*)

Floating floors are normally not used in budget construction. They are more typically found in more expensive structures. However, a budget floating floor can provide reasonably good performance; an example is shown in Fig. 17-22. The weight of the two layers of gypsum board and the finish floor is supported by a layer of glass fiber. The springiness of the partially compressed glass fiber is an essential component of this design. If it is totally compressed, the plywood/finish-floor layers would essentially rest on the basic floor, and little sound isolation would result. As noted, it is important that the gypsum-board/finish-floor mass does not touch the structure, including the edges. The glass fiber turned up at the edges is used to isolate the floor mass edge from the structure. Alternatively, a strip of compressed glass fiber could be used to provide isolation from the wall.

Floating Walls and Ceilings

Floating walls and ceilings are isolated from the structural floor, walls, and ceiling to isolate the inner room from structureborne noise. Floating walls can be constructed with metal studs placed on the floating floor. The top of each wall section is secured with resilient sway braces connected to the structural wall. The walls can be faced with

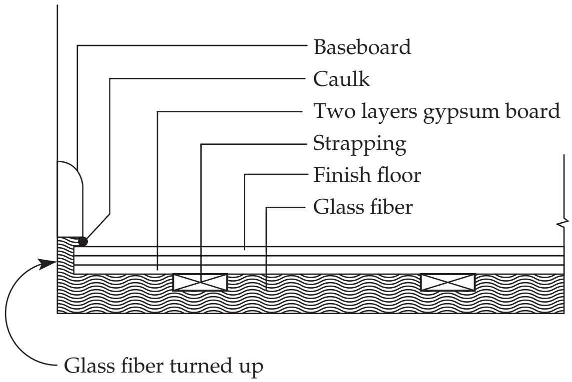


Figure 17-22 A low-cost wood floating floor.

two layers of standard gypsum drywall. Glass-fiber insulation can be used. It is important to ensure that any penetrations in the wall such as pipes or ducts are isolated from the floating wall. Floating (suspended) ceilings are constructed of metal frames that are supported by wires on isolation hangers hung from the structural ceiling. Drywall is used to cover the frame. An acoustical sealant is used around the periphery of the ceiling.

Resilient Hangers

As noted, resilient channels can be used to provide some isolation to gypsum-board sheets. Another system using resilient hangers can replace the resilient channels or be added to them; it is illustrated in principle in Fig. 17-23. If some ceiling height can be

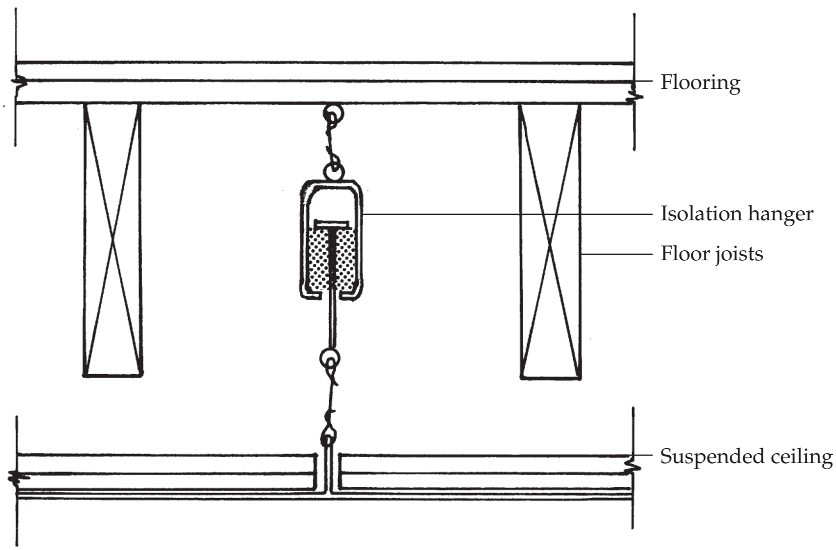


Figure 17-23 An example of a resilient hanger. The hanger is selected so that its deflection falls within its specified range.

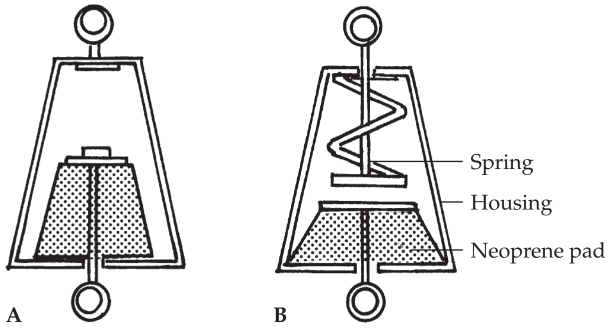


Figure 17-24 Two types of resilient hangers. (A) Neoprene pad. (B) Neoprene pad and steel spring.

sacrificed, a suspended ceiling can be installed, one that is isolated from the structure by resilient hangers. The resilience of the hanger illustrated in Fig. 17-23 is provided by the shaded material, which could be compressed glass fiber of the proper density. The load of the ceiling gypsum boards and supporting frame must be distributed between hangers to keep the deflection of each hanger within its rated deflection range. Underloading or overloading a hanger could "short circuit" its isolation.

Several hanger designs are available. The two hangers shown in Fig. 17-24 utilize neoprene alone or a combination of neoprene and a steel spring. The frequency ranges of these two products differ; the selection of the proper hanger for the specific job is important, and data about each is available from the manufacturer. The weight of the ceiling must be carefully matched to the deflections of the hangers to achieve maximum isolation. The periphery of the suspended ceiling should be sealed with a nonhardening acoustical sealant.

Floor/Ceiling Construction

Floor/ceiling construction deals with many of the same issues as wall construction. In particular, the floor/ceiling must provide sufficient isolation between the rooms above and below. Clearly, noise intrusion can move in either direction, and a good floor/ceiling design can isolate a quiet lower room from a noisy upper room, or vice versa. However, unlike most wall designs, floor/ceilings must take into account the directionality of the noise transfer. In particular, it is somewhat more difficult to isolate a lower room from an upper room; this is because of footfall sounds. Footfall sounds present a challenge to builders, residents, and studio operators alike.

The sounds associated with the activity of people in the space above are a common source of complaints from the people living below. Such noises destroy the concept of privacy in living quarters. Such sounds can also be a source of much trouble when the noises penetrate a sound-sensitive space below such as a recording studio. As with many other noise intrusion problems, the most effective remedy is to stop the noise at its source. This is the case with footfall sounds. For example, the easiest way to diminish footfall sounds (and other impact noises) in a lower apartment is to generously offer to install plush carpets and carpet pads in the apartment above.

Case Study of Footfall Noise

To illustrate the difficulty in designing an acoustically proficient floor/ceiling, consider the following case. Blazier and Dupree give an account of footfall noise problems. The homeowners in a luxury condominium complex brought an $80 million class-action suit against the developers. The major claim was based on annoyance caused by footstep noise generated by the activity of residents above being transmitted through the structure to their neighbors below. Even when cooperative upstairs neighbors agreed to go barefooted or to wear soft-soled shoes, the thuds and thumps were still clearly audible below. The footfall impacts also resulted in "feelable" vibrations of the floor/ceiling structure, vibrations that even affected closet doors and light fixtures. Complaints were general throughout the building.

The average purchase price of the apartments was quite high, with marketing claims of acoustical privacy. Because of the upscale nature of the project, the builders incorporated special design features to provide significantly better impact noise isolation than that required by California construction standards for multifamily housing. The floor/ceiling construction in question is shown in Fig. 17-25. Double joists are used to stiffen the floor plane. The ceiling of double 1/2-in gypsum board is mounted on resilient channels. The floor above starts with the usual 3/4-in plywood subfloor, on which is placed Kraft paper, a resilient mat, reinforcing wire, and 1-1/4-in mortar. On top of this is 3/8-in ceramic tile. To minimizing flanking, a dense glass-fiber perimeter strip isolates the floor from the structure.

To provide data for the defense, it was decided to build an off-site laboratory mockup duplicating a pair of typical stacked rooms. The floor/ceiling structure was

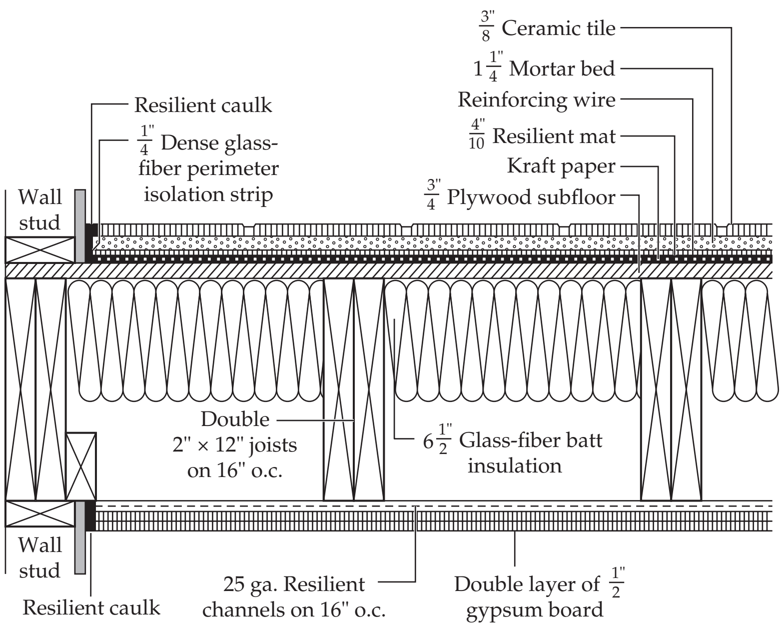


Figure 17-25 Example of floor/ceiling construction. (*Blazier.*)

that described in Fig. 17-25. The lower test room was used to measure impact sound-pressure levels resulting from three types of noise sources: a standard impact tapping machine, a standardized "live-walker," and a calibrated tire drop. The measurement microphone on a 20-in boom was located in the center of the room below, and was rotated slowly in a horizontal plane during the integrating period. Data were obtained in 1/3-octave bands from 2 Hz to 4 kHz.

Among many conclusions, the one concerning footwear is very interesting. In the live-walker tests it was found that for frequencies below about 63 Hz, the impact sound-pressure levels below the walking surface were amazingly close for leather heel/leather sole, rubber heel/leather sole, track shoe, and barefoot cases. Figure 17-26 shows that the peak energy of these live-walker tests falls in the 15- to 30-Hz region. This coincides with the fundamental natural frequency of the floor/ceiling system which, with typical lightweight structural framing, is also between 15 and 30 Hz.

The addition of floating floors or carpeting decreases the transmission of the higher-frequency components of footfall noise, but there is no economically practical method of avoiding the thuds and thumps of footfalls with typical lightweight structural framing. To obtain the stiffness necessary to reduce the thuds and thumps, a concrete structural floor system is required.

In Fig. 17-26, note the measurements made three octaves below the usual low-frequency limit of 125 Hz. Measuring no lower in frequency than 125 Hz misses common impact sounds 40 dB higher than at 125 Hz. Carpet is effective in reducing noises above 125 Hz, but completely ineffective in the 15- to 30-Hz range. Noise in the frequency range below 30 Hz is a concern even when footfall noise is not an issue; for example, many audio playback systems with subwoofers can easily reproduce music in this region.

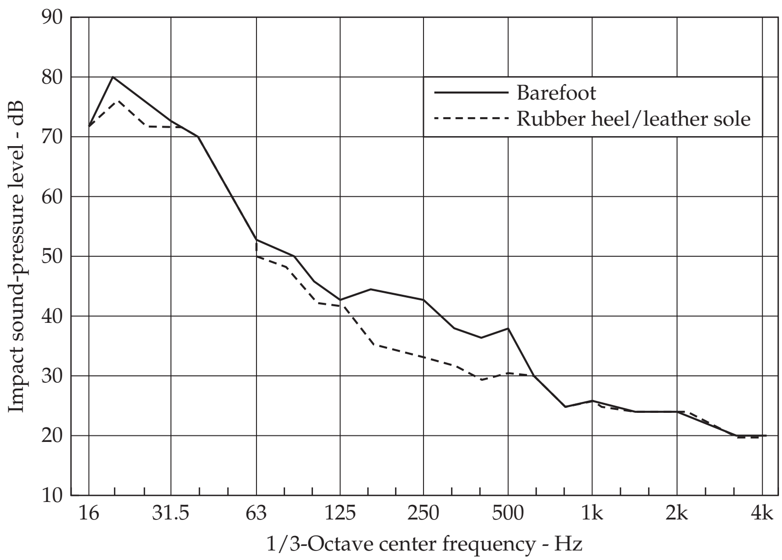


Figure 17-26 Measurements of floor/ceiling footfall noise.

Floor/Ceiling Structures and Their IIC Performance

As noted, floor/ceilings are particularly prone to impact noise. Walking with hard heels, dropping objects, moving chairs, and similar activities can create noise that is radiated downward through the floor to the room below. Impact noise can also radiate outward through structural elements to adjacent rooms. Impact noise is most effectively treated at the source. Hard-surface floors such as concrete and terrazzo perform poorly. Soft floor coverings such as carpet with pad and rubber tile perform much better. Floating floors offer superior performance. Ceilings can be improved to some degree by adding suspended tiles. When very high isolation is required, several or all of these measures may be needed.

Impact insulation class (IIC) is a single-number rating of the impact sound performance of floor/ceiling constructions over a standard frequency range. This IIC single-number rating for impulse sounds is comparable to the STC single-number rating for steady-state sounds. The higher the IIC rating, the more effective the floor/ceiling construction in reducing transmission of impact sounds such as footfalls. Numerically, IIC might vary from 20 (poor performance) to 60 (good performance). IIC data is measured at sixteen 1/3-octave bands from 100 to 3,150 Hz and a reference contour is used to yield the IIC value. The curve-matching method for determining IIC values is similar to the method used for determining STC ratings. Note that an IIC rating does not consider noise below a 100-Hz center frequency; a floor/ceiling could have a high IIC value, but still transmit considerable low-frequency noise. IIC is described in *ASTM E989-18 Standard Classification for Determination of Single-Number Metrics for Impact Noise.*

Floor/Ceilings in Frame Buildings

The first and most distressing factor in floor/ceiling problems is that often there is no access to the floor side—the side that is most amenable to remedies—because it is someone else's property. Instead, the only hope for improving the isolation of a room from the neighbors' noise from above is what can be done to the ceiling. Floor/ceiling constructions in frame structures are quite limited in scope, and most of them would be included in the three designs shown in Fig. 17-27.

In Fig. 17-27A, the 2-in × 10-in joists commonly have a 5/8-in gypsum board ceiling nailed to the lower edges and a 1/2-in plywood subfloor above. In many buildings, the finish flooring above is probably something very much like the 25/32-in tongue-and-groove oak flooring pictured. If this is the "as found" condition, what can be done without access to the floor side? The first step would be to remove the gypsum-board ceiling fastened to the lower edge of the joists and place 3-in glass-fiber insulation between the joists as shown in Fig. 17-27B. This insulation could be 6-in or 10-in thick, but very little improvement would be gained by going beyond the 3-in batts.

The ceiling itself can be improved by mounting resilient strips on the joists and a layer of gypsum board to the strips, and then adding a second layer of gypsum board as shown in Fig. 17-27C. The resilient strips could instead be mounted between the two layers of gypsum board. We are still limited by not having access to the floor side above. What performance can be expected from these improvements made to the ceiling?

Figure 17-28 shows the transmission-loss performance of the three floor/ceiling arrangements of Fig. 17-27. Note that the curves are quite smooth and free from untoward resonance effects. The as-found condition (A) gives STC-37, and adding the glass fiber

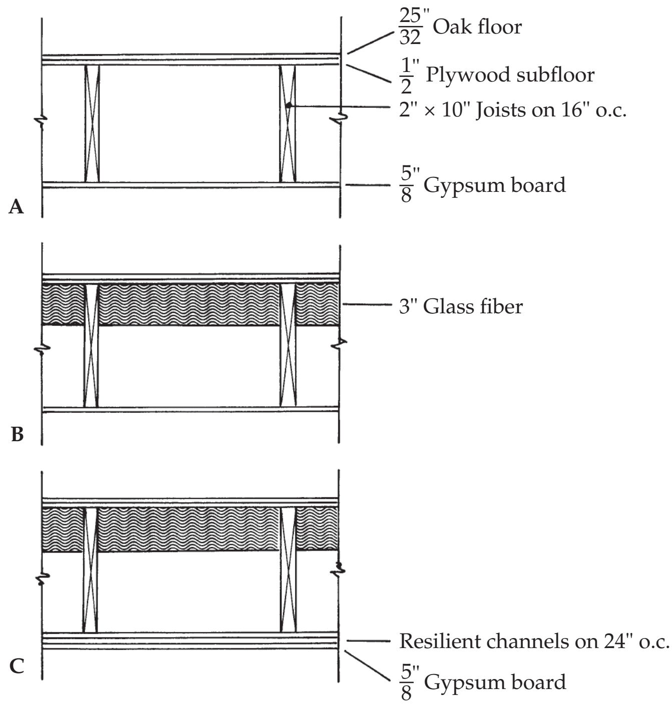


Figure 17-27 Three common floor/ceiling constructions. (A) The "as found" construction. (B) Glass fiber is added. (C) Glass fiber and resiliently mounted gypsum-board layers are added. (*Egan.*)

(B) gains only 3 STC points, yielding STC-40. Improving the ceiling (C) by adding two layers of gypsum board with one of them resiliently mounted increases the rating to STC-47, a gain of 7 points. From the as-found situation of Fig. 17-28A a total of 10 points in STC rating has been gained by the addition of glass fiber and resilient channels. This is close to the maximum improvement that can be expected. Figure 17-28 also includes IIC measurements.

Floor Attenuation with Concrete Layers

The improvement in attenuation that might be expected by adding a 1-1/2-in layer of troweled cellular concrete of density 105 to 120 lb/ft^3 is shown in Fig. 17-29. The two floor/ceiling constructions in the figure are identical, except one has 3-1/2 in of glass fiber and resilient channels and the other does not. An STC-59 rating is attained by the construction having the glass fiber and the resilient channels, a very worthwhile increase of 10 STC points. This 10-point increase in STC compares to the 10-point increase from STC-37 to STC-47 of Fig. 17-28 obtained by adding gypsum board, glass fiber, and resilient channels to the bare-bones "as found" condition of Fig. 17-27A. Adding a concrete topping, more gypsum-board layers, glass fiber, and resilient channels increases a poor floor/ceiling structure with an STC-37 rating to a very respectable STC-59 rating. This is a substantial improvement in STC that will be meaningful in the finished structure.

Figure 17-30 shows a comparison of the best floor/ceiling structure without a concrete layer with the best floor/ceiling structure with a concrete layer. An estimate of the

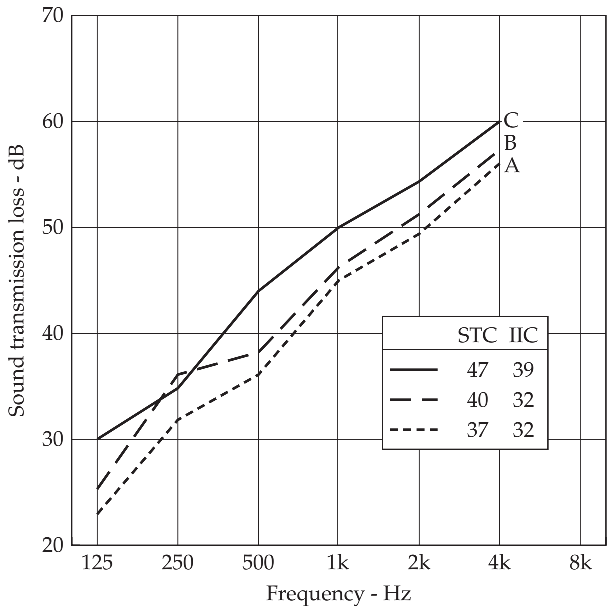


FIGURE 17-28 Transmission-loss comparison of the three floor/ceiling constructions shown in Fig. 17-27. (A) The "as found" construction. (B) Glass fiber is added. (C) Glass fiber and resiliently mounted gypsum-board layers are added. (*Egan*.)

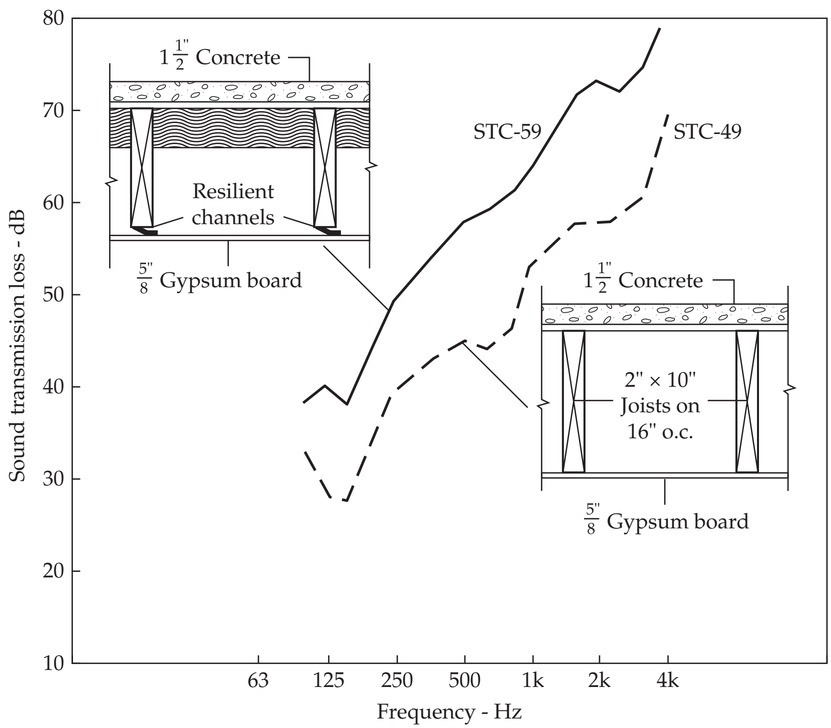


FIGURE 17-29 The transmission-loss effect of concrete topping on a floor. (*Grantham and Heebink*.)

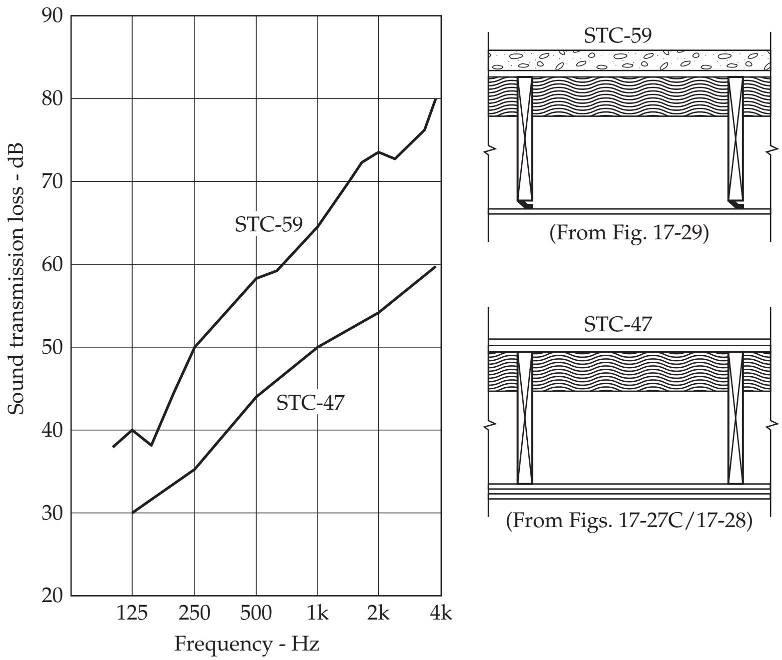


FIGURE 17-30 An example of a concrete layer that significantly improves the transmission loss.

effect of the concrete layer at different frequencies can be made by comparing the STC-59 curve of Fig. 17-30 with the STC-47 curve below it. The concrete contributes about 10 STC points at low frequencies, about 15 points at 500 Hz, and about 20 points above 2 kHz.

Plywood Web versus Solid Wood Joists

Some floor/ceiling systems in frame structures use plywood web beams instead of conventional 2-in × 10-in or 2-in × 12-in wood joists. They are made with 2-in × 3-in flanges and a 3/8-in plywood web. These plywood web beams, illustrated in Fig. 17-31, offer certain advantages. As they are usually fabricated off the building site, they contribute to construction speed and efficiency. From the standpoint of the architect or designer, 12-in plywood web beams allow greater spans. From the standpoint of the acoustician, they contribute to the stiffness of the floor plane and thus provide higher transmission loss to low-frequency noises.

Figure 17-31 presents the transmission-loss characteristics of two good floor/ceiling systems, both offering an STC-58 rating. Both use resilient channels, both have 3-1/2-in glass fiber, and both have a 3/4-in plywood subfloor. The floor/ceilings differ only in that one uses 2-in × 12-in plywood web joists while the other uses 2-in × 10-in solid wood joists. There is little consistent difference between the two curves. The plywood web joists give 5-dB-higher attenuation to sound at 100 Hz than the solid wood joists. The stiffening of the floor plane by the plywood web joists results in greater low-frequency attenuation, which is noticeable in Fig. 17-32. The reverse is true in the 1,000- to 2,000-Hz region, where the solid wood joists offer greater attenuation than the plywood web joists. Carpets help in the 1,000- to 2,000-Hz region, but not in the 15- to 31-Hz region.

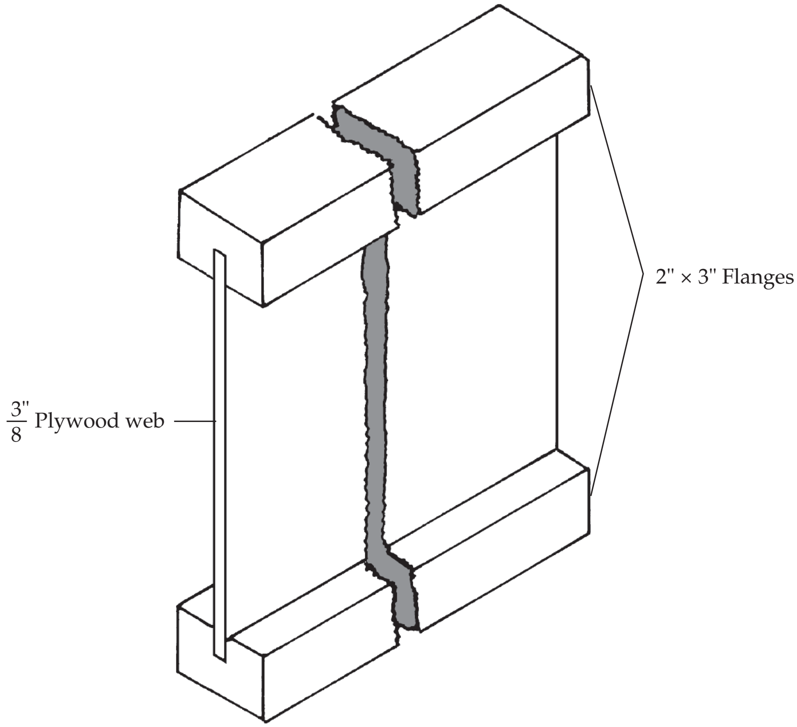


Figure 17-31 An example of a plywood web beam that can be used as a floor/ceiling joist.

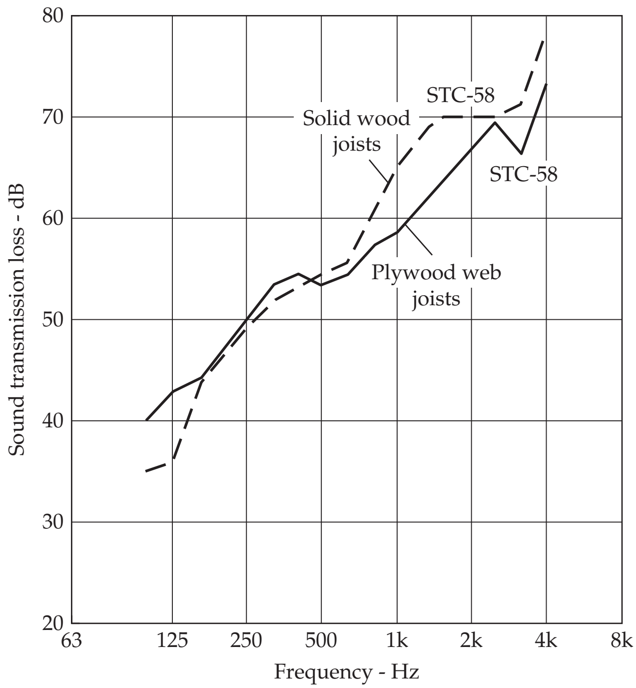


Figure 17-32 A comparison of the transmission-loss response of plywood web joists versus solid wood joists. (*Grantham and Heebink.*)

Key Points

- To satisfy acoustical requirements, structural barriers must be designed and constructed in ways that are considerably different from typical building specifications.
- More sophisticated lightweight partition designs can yield efficient transmission loss. However, heavy walls usually provide greater isolation at low frequencies.
- Multiple partitions designed for separation of mass will outperform single partitions particularly if construction allows each partition to behave independently of the other.
- Efficient wall isolation can be achieved by: increase weight, increase spacing of leaves, staggered studs, leaves of different weight, resilient strips, glass-fiber blankets, and perimeter caulking.
- An acoustically weak existing wall can be improved by adding a new partition on either side of it, preferably spaced apart from the existing wall.
- Masonry walls provide relatively more isolation than lightweight walls at low frequencies; therefore, they are superior in most music applications. However, masonry walls are efficient transmitters of impulse noise through structureborne transmission.
- Even a barrier with an otherwise high transmission loss can be severely compromised by any lower-performance breach, such as an air leak. In fact, the overall isolation of the barrier will be close to that of the weak link.
- Unless care is taken, doors and windows can undermine the acoustical integrity of a partition. The components of the barrier such as windows and doors must provide a transmission loss close to that of the wall itself.
- A floating floor is an additional floor raised and decoupled from the structural floor. This can reduce structureborne noise.
- Floor/ceilings must take into account the directionality of the noise transfer. In particular, because of footfall sounds, it is more difficult to isolate a lower room from an upper room.
- Impact insulation class (IIC) is a single-number rating of the impact sound performance of floor/ceiling constructions over a standard frequency range. The IIC rating for impulse sounds is comparable to the STC rating for steady-state sounds.
- Floor/ceiling noise intrusion from above is problematic because often there is no access to the floor side—the side that is most amenable to remedies. Improvements can only be done to the ceiling, limiting options.
- Where practicable, adding a concrete topping, more gypsum-board layers, glass fiber, and resilient channels will substantial improve an acoustically poor wood floor/ceiling structure.

CHAPTER 18

Sound Isolation: Windows and Doors

Windows are a functional necessity in most rooms, including those in acoustically sensitive facilities. Glass establishes visual communication between two rooms that may be essential for the task at hand. For example, a recording engineer in a control room must be able to immediately see what musicians are doing in the studio. Occasionally, although many acousticians might not be enthusiastic, a studio owner may want a large window in an exterior studio wall to permit a scenic view.

If a window is placed, for example, in the wall between a control room and studio, or in a wall facing loud outdoor ambient sound levels, the window's sound transmission loss (TL) must be comparable to that of the wall itself. A window with insufficient transmission loss would be a weak link that would seriously compromise the acoustical isolation of the wall. For example, a well-built staggered-stud or double-stud wall might have a sound transmission class (STC) rating of 50 and provide sufficient isolation, as would a concrete-block wall. To approach this TL performance with a window requires careful design and installation. Particularly when relatively thick panes, and multiple panes, are used, glass can provide adequate sound isolation. In some designs, laminated glass is used to provide higher levels of isolation. As in any sound-sensitive design, care must be taken during construction to ensure that the window is installed without sound leaks or other detriments.

Although a room can be constructed without windows, it cannot function unless it has doors. Moreover, whereas many windows are designed to be permanently sealed, useful doors are designed to be opened. This basic requirement of doors makes it more difficult to achieve a reliable seal against sound leaks. Also, door thresholds are subject to wear and tear from foot traffic as well as moving equipment; this makes it difficult to provide floor seals that are durable over time. A good door design thus requires both a door panel that is sufficiently massive and solid, and also a closure system that reliably seals the entire perimeter around the door. A good door can be constructed from common building materials, but in many cases, specially engineered acoustical doors are selected. The latter can be quite expensive.

Finally, when a door is open, no matter what its cost, it provides absolutely no acoustical isolation. To overcome this, a sound lock can be incorporated into a room design. The double doors in a sound lock add additional isolation when both doors are closed and at least some isolation when one door is open. Sound locks remove some of the burden from individual door design, but are an option only when sufficient floor space is available.

Single-Pane Windows

Single-pane (also called single-leaf or single-glazed) windows, such as the common household window, provide relatively poor sound isolation. Those living near a highway or an airport, for example, find that windows of this type allow significant noise to pass through. Typical single-pane 1/8-in glass may have an STC rating of 25. The mass of the glass pane attenuates the passage of sound through it according to the expression:

$$TL = 20 \log (fm) - 48 \tag{18-1}$$

where TL = transmission loss, dB
f = frequency, Hz
m = surface density, lb/ft^2

If the density of glass is assumed to be 160 lb/ft^3, the surface density may be found for any thickness of the glass pane. The value is also often provided by the manufacturer. There is a lack of unanimity regarding the value of the constant to be subtracted. Quirt recommends 48 dB, but others prefer 34 dB. This is of oblique importance in this discussion because the transmission loss of various glass arrangements to be presented has been determined by reliable measurements. The mass law as applied to glass panes, expressed graphically in Fig. 18-1, emphasizes that mass is the major component of transmission loss of glass. Transmission loss increases as the thickness increases. Also, clearly, transmission loss increases with frequency. Different glass-pane configurations (multiple panes, differing thickness, and subduing of resonances) will be presented as ways to further enhance the transmission loss of windows.

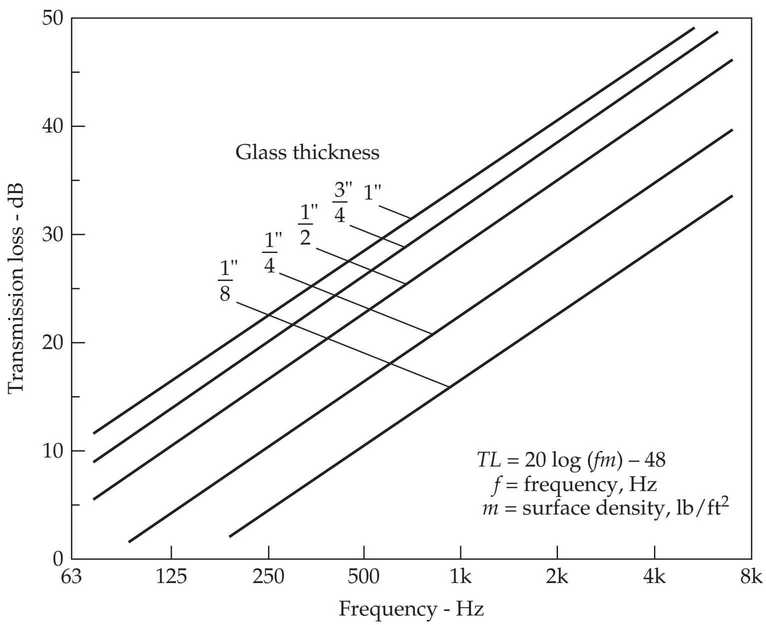


FIGURE 18-1 The mass law applied to glass panes showing panes of different thicknesses.

Window panes should be mounted with neoprene perimeter gaskets; this is preferred over caulk or putty. Generally, sealed windows will provide 3- to 5-dB greater isolation than comparable windows that can be opened, even when the openings are gasketed. When opening windows are required, for example, for ventilation, double-pane, double-sash windows are preferred over single-pane, single-sash windows.

Double-Pane Windows

Single-pane windows offer insufficient transmission loss for most acoustical applications; however, the additional mass provided by additional panes provides greater loss. In many applications, double-pane (also called double-leaf or double-glazed) windows can provide excellent results. There are many qualifications to this statement, but the transmission loss of the double-pane windows shown in Fig. 18-2 shows an improvement in transmission loss over the single-pane mass law shown in Fig. 18-1. Generally, a simple double-pane window (two panes separated by an airspace) can provide about 30 dB of transmission loss. Prefabricated double-pane, sound-isolating windows that are available commercially may provide an STC rating of 50 or more.

In Fig. 18-2, two double-pane windows with approximately 4-in spacing are compared on the basis of the thickness of glass. The solid curve represents measurements made with a 1/8-in pane and a 3/32-in pane. The broken-line curve shows measurements made with a 1/2-in pane and a 1/4-in pane. These windows use panes of glass of different thickness, and thus different mass; this beneficially acts to distribute resonances.

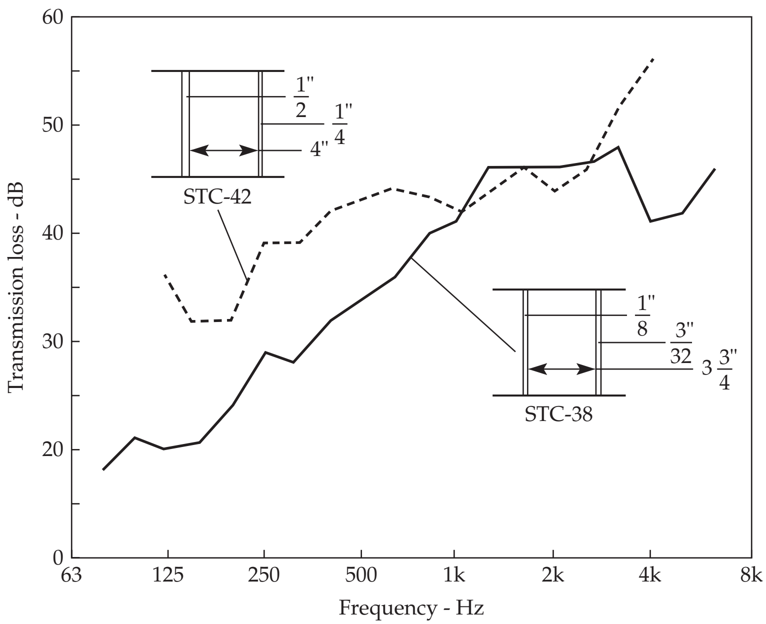


FIGURE 18-2 A comparison of transmission losses in two double-pane windows. Solid curve: 1/8-in pane and 3/32-in pane. (*Libby-Owens-Ford.*) Broken curve: 1/2-in pane and 1/4-in pane. (*Sabine, et al.*)

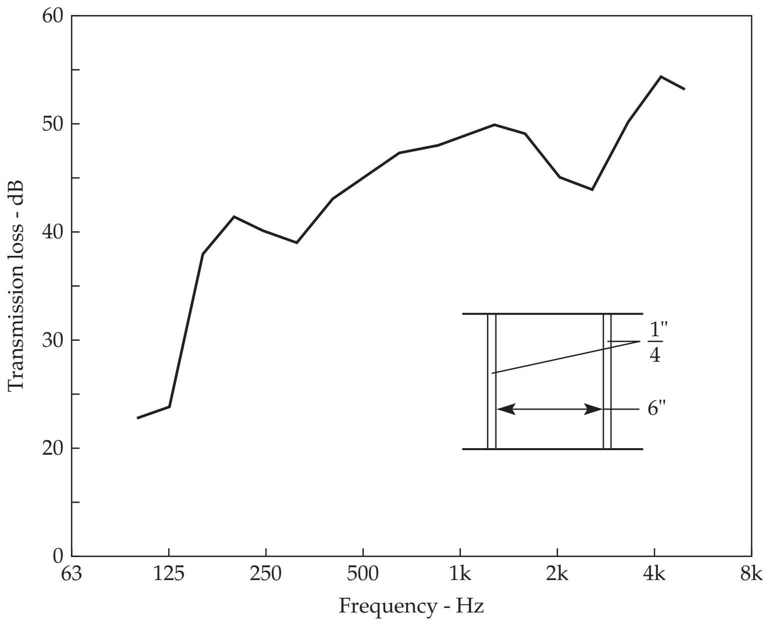


FIGURE 18-3 Transmission loss with heavy glass panes and large spacing between panes. (*Quirt.*)

This is discussed in more detail later. The thinner glass window and the heavier glass window have similar transmission loss in the 1- to 3-kHz region, but the heavier glass is far superior below 1 kHz. The STC ratings, which show an advantage of only 4 points for the heavier glass, are inadequate for describing the performance of these two windows over the audible band.

The measured transmission loss of another double-pane window is shown in Fig. 18-3. In this window two 1/4-in glass panes are separated by 6 in. The overall transmission loss of the 6-in spacing is quite similar to that of the heavier glass window of Fig. 18-2. The only way the modest specific effects of dissimilar panes, glass surface density, and glass spacing can be observed is in direct comparisons; these will be presented later. The irregularities of the usual measured transmission-loss curves, due principally to resonances, tend to hide these other variables of direct interest.

Acoustical Holes in Glass: Mass-Air-Mass Resonance

An acoustical hole is a phenomenological name for a narrow region of the audible spectrum in which sound more readily passes through a barrier such as a wall, window, or door. Sound within a frequency range is attenuated several decibels less than the mass law predicts; the hole appears as a dip or flattening of the transmission-loss curve. Acoustical holes in glass can be the result of multiple causes including the coincidence effect, and are usually traceable to resonances; this is illustrated in the following examples.

Several different resonance effects alter the shape of the transmission-loss curve of any glass window, decreasing the window's attenuation in a particular frequency range. The mass-air-mass resonance of a double window is the result of the mass of one glass pane being coupled to the other glass pane by the air in the cavity between them. Sound impinging on the first glass pane causes it to vibrate, the air in the cavity acts like a spring, and this causes the second glass pane to vibrate. This resonant system can be likened to masses attached to each end of a spring.

Quirt studied the effect of glass spacing on two 1/8-in glass panes. He found that at 250 Hz the sound transmission-loss curve went through a very pronounced null at spacings between 1/2 and 1 in, as shown in Fig. 18-4. At 800 Hz, the sound transmission loss showed no such dip. Mass-air-mass resonance is largely a low-frequency effect; in fact, with certain glass panes at certain spacings, the resonant frequency is so low that it does not appear in the usual 63 Hz to 8 kHz measuring range.

The mass-air-mass resonant frequency can be estimated from the following equation:

$$f = 170\sqrt{\frac{m_1 + m_2}{m_1 m_2 d}} \tag{18-2}$$

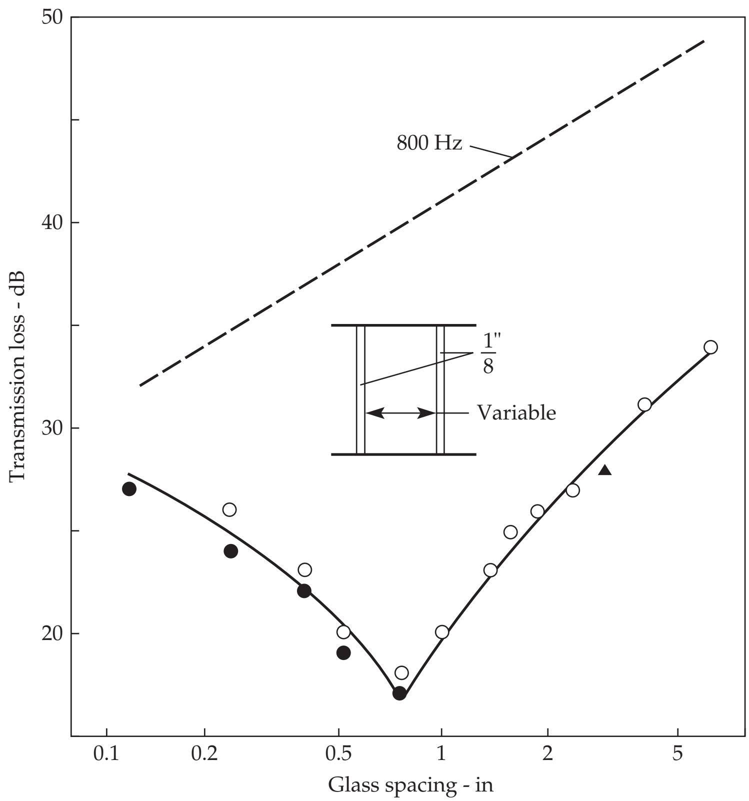


FIGURE 18-4 The mass-air-mass resonance effect. (*Quirt.*)

where f = mass-air-mass resonant frequency, Hz
m_1 = surface density of glass pane A, lb/ft^2
m_2 = surface density of glass pane B, lb/ft^2
d = spacing between glass panes, in

Using this equation, the mass-air-mass resonant frequencies for several glass spacings and weights (thickness) have been calculated and plotted in Fig. 18-5. This figure shows that resonant frequencies are above 100 Hz only for lighter glass panes and smaller spacings; as noted, mass-air-mass resonance is primarily a low-frequency effect.

A prominent mass-air-mass resonance appears in the sound transmission-loss curve of Fig. 18-6. The notch at about 400 Hz is a good example of an acoustical hole in the glass. Sound near 400 Hz is attenuated about 5 dB less than at adjoining frequencies. Both the very small spacing of 1/4 in and the very light glass of 1/8-in thickness have purposely been selected to raise the mass-air-mass resonance to about 400 Hz to demonstrate its presence. For more practical double-pane windows, the resonance might be at too low a frequency to appear on a measured transmission-loss curve.

Acoustical Holes in Glass: Coincidence Resonance

The transmission-loss dip at about 4,000 Hz in Fig. 18-6 is caused by the coincidence effect. It, too, can be classed as an acoustical hole, but its cause is entirely different. The flexural, bending vibrations of the glass panel interact with the impinging sound in such a way that an abnormal amount of sound is transmitted through the glass at the coincidence frequency. When the phase of the pressure crests of the incident sound coincides with the vibrational crests of the panel, this coincidence effect lowers the sound transmission loss. This means that sound at or near this frequency penetrates the window more easily. The frequency at which the coincidence effect occurs in a window may be estimated from:

$$f = 500/t \tag{18-3}$$

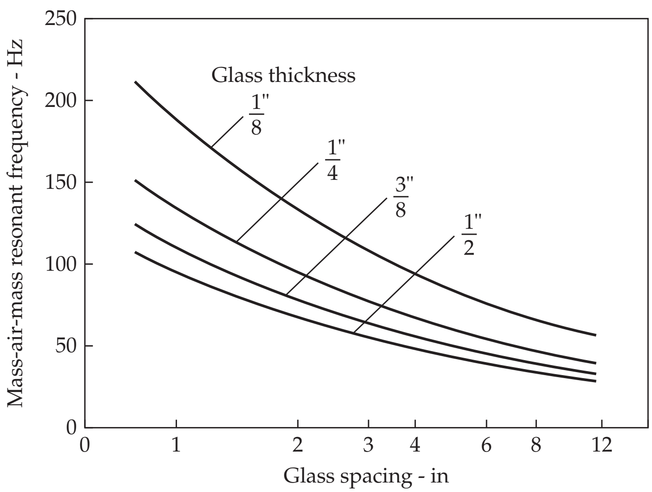


Figure 18-5 The mass-air-mass resonant frequency.

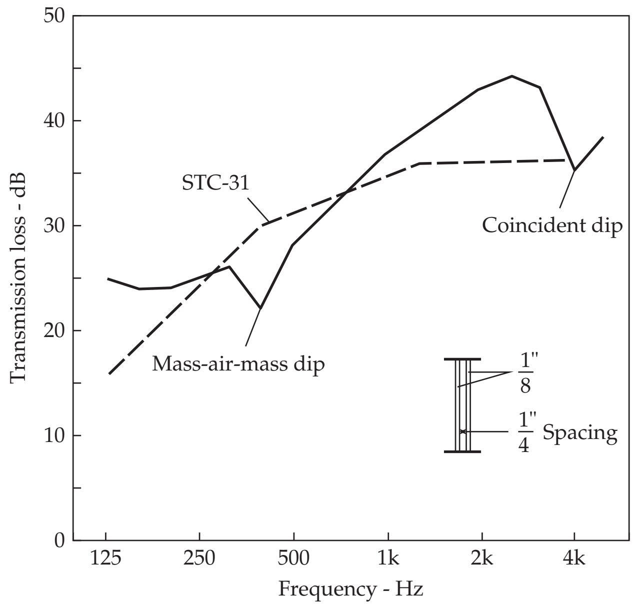


FIGURE 18-6 A comparison between mass-air-mass resonance and coincidence dip. (*Quirt.*)

where f = coincidence frequency, Hz
t = thickness of glass pane, in

The window of Fig. 18-6 employs glass 1/8 in thick. Equation 18-3 then becomes $f = (500)(8) = 4{,}000$ Hz, which is close to the observed frequency of the coincidence dip.

Coincidence-effect dips can be minimized in double-pane windows. If each pane has a different thickness, the resonant point of each pane occurs at a different frequency. At the frequency region where one pane is acoustically weak, the other pane is not. Clearly, a double-pane window in which both panes are of the same thickness would still encounter a coincidence-dip problem.

Acoustical Holes in Glass: Standing Waves in the Cavity

The airspace cavity between the two panes in a double-pane window is capable of supporting standing waves, much as in a room. Modes are associated with the length, height, and depth of the cavity as shown in Fig. 18-7. Axial modes strike two surfaces; tangential modes strike four surfaces; oblique modes strike all six surfaces. Because of the greater number of reflections encountered, the tangential modes and the oblique modes have lower energy levels; Morse and Bolt calculated these to be −3 dB and −6 dB respectively below the axial modes. The frequencies of all three modes may be calculated using eq. 13-3.

Some of the modal frequencies associated with a window cavity having dimensions of $8 \times 4 \times 0.5$ ft are listed in Table 18-1. The lowest axial-mode frequency associated with the 8-ft length, the (1, 0, 0) mode, is 70.6 Hz. The lowest axial-mode frequency associated

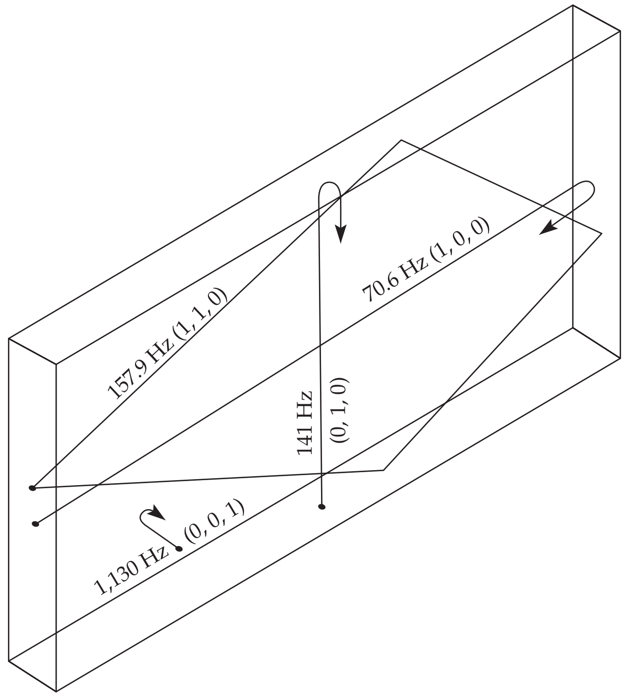


Figure 18-7 Standing wave modes in the space between two glass panes.

with the 4-ft width of the cavity, the (0, 1, 0) mode, is 141.3 Hz. The lowest axial-mode frequency associated with the 6-in depth of the cavity, the (0, 0, 1) mode, is 1,130 Hz.

With knowledge of these modal frequencies, we can determine if the absorbent material on the periphery of the space between the glass panes (the "reveals") is capable of damping these resonances. If not sufficiently damped, it is possible that minor acoustical holes will appear at certain modal frequencies. The thickness of this periphery absorbent is severely limited by the space available. The 70.6-Hz and 141.3-Hz axial modes are the lowest to be absorbed. The 1-in-thick glass fiber commonly used as cavity absorbent does not absorb these frequencies well, and there is no space for a Helmholtz resonator absorber. Thus the cavity resonances at 70 Hz and 141 Hz will probably be significant in the finished window.

Glass Mass and Spacing

A compilation of measurements made by Quirt of double-pane windows for 1/4-in and 1/8-in glass is presented in Fig. 18-8. The effects of glass mass and spacing are illustrated. The upper curve is for two 1/4-in glass plates, and the lower curve is for two 1/8-in glass plates. The two curves indicate that there is a 3-point gain in the STC rating for every doubling of the mass of the glass. The curves also show that doubling the spacing between panes results in approximately a 3-point gain in the STC rating. Absorptive material should be placed in the interior areas between panes; this can add 2 to 5 dB

p, q, r	Axial (Hz)	Tangential (Hz)	Oblique (Hz)
1, 0, 0	70.6		
0, 1, 0	141.3		
1, 1, 0		157.9	
0, 0, 1	1,130.0		
1, 0, 1		1,132.2	
0, 1, 1		1,138.8	
2, 0, 0	141.3		
2, 0, 1		1,138.8	
1, 1, 1			1,141.0
0, 2, 0	282.5		
2, 1, 0		199.8	
1, 2, 0		291.2	
0, 2, 1		1,164.8	
0, 1, 2		2,264.4	
2, 1, 1			1,147.5
1, 2, 1			1,166.9
2, 2, 0		315.8	
3, 0, 0	211.9		
0, 0, 2	2,260.0		
3, 1, 0		254.6	
0, 3, 0	423.8		
2, 2, 1			1,173.3
3, 0, 1		1,149.7	

TABLE 18-1 Partial List of Observation Window Cavity Resonant Frequencies (Cavity Dimensions: 8 × 4 × 0.5 ft)

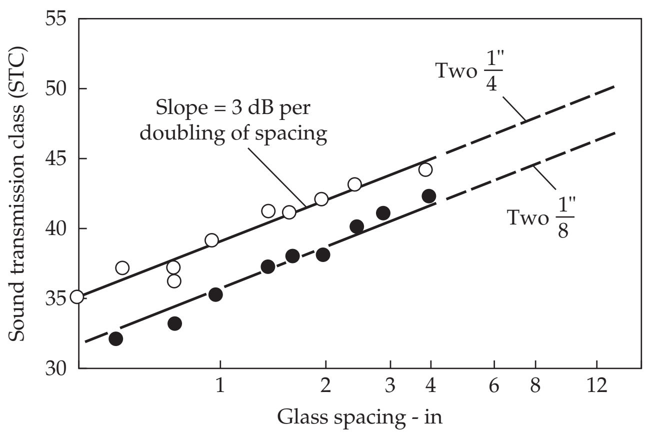


FIGURE 18-8 The effect of glass-pane spacing on STC rating. (*Quirt.*)

of attenuation. To achieve a high transmission loss with double-pane windows, several modest gains must be combined to obtain satisfactory performance.

Dissimilar Panes

Transmission loss can be improved by using two glass panes of different thicknesses. For example, consider two double-pane windows, both with approximately 4-in spacing, one with 1/8-in and 3/32-in glass and the other with 1/2-in and 1/4-in glass (see Fig. 18-2). As both windows use dissimilar glass, specific evaluation of the dissimilarity factor is not possible. However, it is known that windows of different mass on the two sides will distribute the mass-air-mass resonances, resulting in a smoother transmission-loss curve.

The mass-air-mass resonance of the lighter window (the solid-line curve of Fig. 18-2) was found by calculation to be about 90 Hz and that of the heavier window (the broken-line curve) to be about 57 Hz. These points are not discernible on the two curves. The coincidence effect involves only a single pane of glass. By calculation, the coincidence frequencies of the lighter window were found to be 4,000 Hz for the 1/8-in glass and 5,300 Hz for the 3/32-in glass. There is a pronounced dip in the solid-line curve in this region of the spectrum. The coincidence frequency for the 1/4-in glass was found to be 2,000 Hz and that of the 1/2-in glass was 1,000 Hz. Small dips can be seen in the broken-line curve at both of these frequencies. Staggering these dips by using glass of different thickness tends to smooth the transmission-loss curve and minimize the effect of the acoustical hole. If both panes resonated at the same frequency, the dip would have been deeper. Therefore, isolation is improved when panes are of different thicknesses, and widely spaced. In comparison, windows with panes of the same thickness and with narrow spacing perform relatively poorly.

Laminated Glass

Glass may be laminated by placing a layer of polyvinyl butyral (PVB), often of 0.015 in thickness, between two layers of glass. Architectural laminated glass, the most familiar form, consists of two plies of glass bonded together by a plastic interlayer (usually PVB) under a pressure of about 250 psi and a temperature of 250° to 300°F. The laminating layer increases the weight of the glass and modestly increases the transmission loss. This is shown in a comparison of equivalent unlaminated and laminated glass:

Unlaminated	Laminated
1/4-in glass: STC-29	Two plies 1/8-in glass: STC-33
1/2-in glass: STC-33	Two plies 1/4-in glass: STC-36

The use of laminated glass in double-pane windows results in greater sound transmission loss because of increased mass. There is also a small damping effect, which is an added advantage.

Plastic Panes

Plastic has the advantage of flexibility, and it does not shatter like glass. Such characteristics might suggest plastic instead of glass in a window in certain circumstances. The principal difference from the sound transmission-loss point of view is that the mass of plastic is about half that of glass and a corresponding double thickness of it would be required. When the masses are the same, plastic panes perform similarly to glass panes. Plastic sheets can be cold-bent to form convex windows. Light transparency of plastic is good, and optical distortion is minor.

Slanting the Glass

In many double-pane studio/control room observation windows, one of the panes is slanted from vertical. As far as transmission loss of the window is concerned, slanting glass yields the same isolation performance as parallel glass if the average separation of the slanting glass is equal to the parallel glass separation. Based on a number of measurements, Quirt advises that nonparallel glazing does not appear to offer any significant benefits. On the other hand, from a room acoustics standpoint, it can be advantageous to slant one pane of glass so that reflections from the glass are angled downward to floor absorbers such as carpet. For example, this can control a potential reflection on the studio side of the window.

Third Pane

Extensive measurements were made by Quirt on windows with a third pane of glass. Most of the three-pane results were very similar to those of the two-pane results. Small differences between three-pane windows and comparable two-pane windows showed a small advantage of the three-pane over the two-pane; this was attributed to a diminished coincidence effect. For most room window designs, the small advantage of a three-pane window over a two-pane window does not seem to justify the added cost and effort. In some cases, sealed triple-pane windows are used; for example, some airplanes use triple-pane windows.

Cavity Absorbent

The addition of a 1-in thickness of glass fiber around the perimeter of the interglass space was shown to be advantageous by Quirt. The improvement was limited to higher frequencies, as one would expect from the characteristics of the absorbent.

Thermal Glass

Two sheets of glass mounted together with an airspace of the order of 1/8 in to 1/4 in form a very effective structure for reducing heat or cold transmission. The sound isolating properties of such glass are the same as that of a glass of the combined thickness. In other words, small airspaces have a negligible acoustical effect, and such glass performs on the mass law alone. It is thus important to observe the mass of the panes. It is possible, for example, for a double-pane window designed for thermal insulation to have an STC value less than a single-pane window.

Example of an Optimized Double-Pane Window

Figure 18-9 shows the measured transmission loss of a window that has an STC rating of 55 and a remarkably smooth transmission-loss curve. Surely the resonances of various kinds are well controlled to yield this smoothness. This window has dissimilar

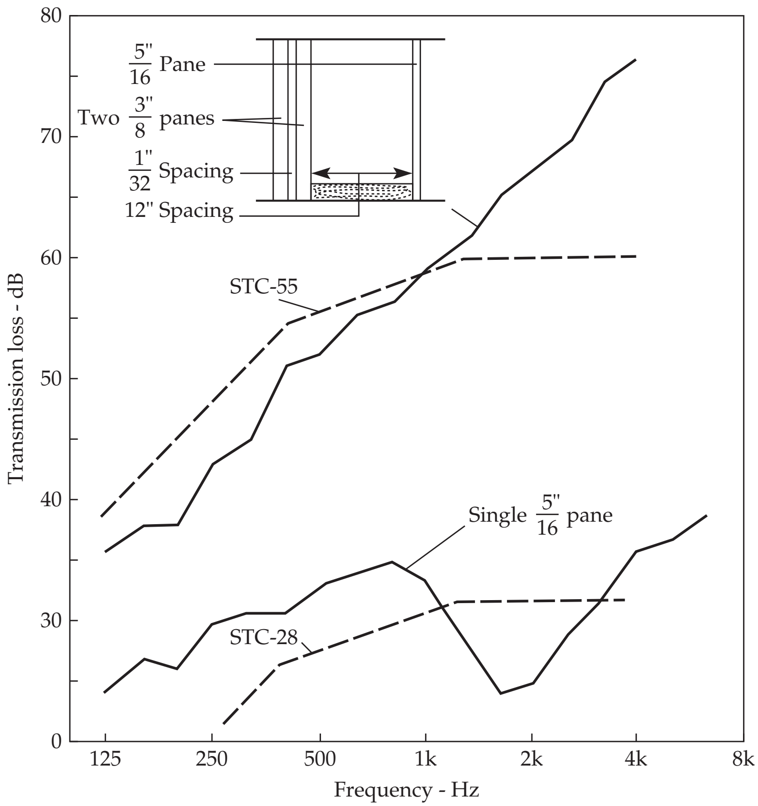


FIGURE 18-9 An example of an optimized double-pane window. (*Cops.*)

glass panes. The double-pane leaf consists of two 3/8-in glass panes with a 1/32-in PVB layer between them. The other pane is 5/16-in glass. The two panes are 12 in apart and an absorbent lines the periphery between the two panes. The important factors in this design are large separation, heavy but dissimilar panes, and absorbency. However, the absorbent produces a good improvement in sound isolation only when the mass of the panes is relatively light, which is the case for most windows. The lower curve in Fig. 18-9 is for a single 5/16-in glass pane to emphasize the great effect of adding the second, heavy double-pane to accomplish the STC-55 rating. A large coincidence dip greatly reduces the STC rating of the single pane.

Construction of an Observation Window

Some recording studios are located in scenic spots and incorporate large viewing windows. Windows such as these can make it difficult to achieve sufficiently low background noise levels, and can create unwanted interior reflections. If the acoustically difficult idea of viewing windows can be successfully rejected in the early planning stages, the only significant glass left in a recording studio is the large observation window between the control room and the studio. Less problematically, occasionally a small observation window is placed between the studio and an exterior hallway, or between the studio and an isolation booth.

An observation window is important in a studio's workflow, but it is also acoustically challenging. The partition between the studio and the control room must provide high transmission loss so that high sound-pressure levels in the studio do not interfere with control-room monitoring.

Similarly, high-level monitoring must not intrude into the microphones in the studio. An observation window may comprise a large percentage of the partition area, and must offer very high transmission loss comparable to that of a massive wall. Many precautions are taken to achieve this. For example, a satisfactory window might require multiple panes, panes that are thick, separation of panes, absorbent in the cavity, and airtight and resilient mounting in the partition. The detailed design and construction of these windows are best left to experienced acousticians and carpenters. However, some elements of the design are common to all critical windows.

In most critical applications, a double-pane window is indicated. Assuming that a double wall or staggered-stud wall is used, the window frame is constructed in two halves. Each half is secured to one wall side with no rigid physical connection to each half; this preserves the wall decoupling. The plan of Fig. 18-10A shows a practical solution to the double-window problem for concrete-block walls. Figure 18-10B is an adaptation to staggered-stud construction. In the latter there are, in effect, two entirely separate frames—one fixed to the inner and the other to the outer staggered-stud walls.

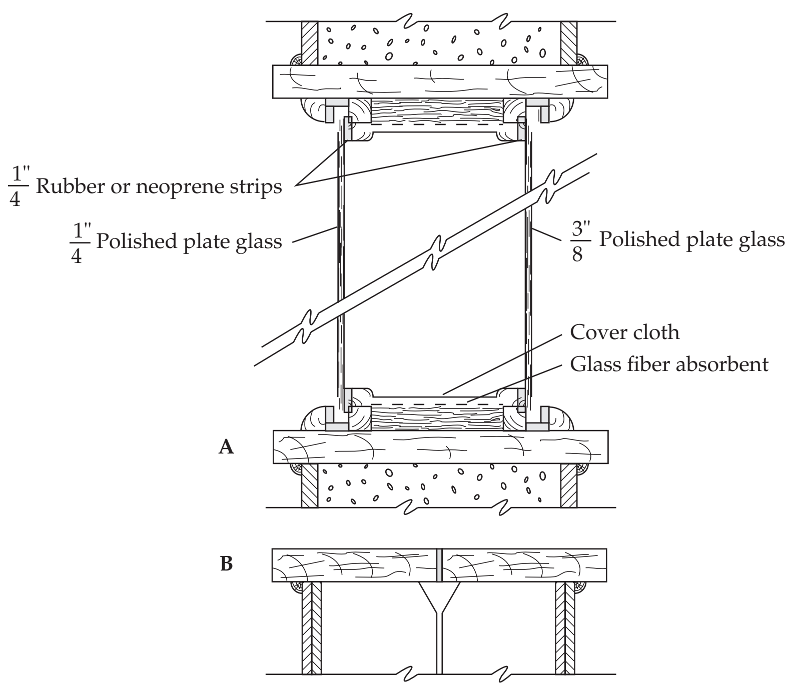


FIGURE 18-10 Wall construction with double-pane window. (A) For concrete-block walls. (B) For staggered-stud walls.

Foam strips or a nonhardening sealant plug the gaps between the window frames to ensure against accidental contact. As in any construction, air leakage must be prevented; details that are common to good wall design, such as caulked seams, are employed.

Heavy plate glass should be used, and the heavier the better. However, even heavy glass will exhibit a coincidence-frequency dip. As noted, there is a slight advantage in having two panes of different thickness. If desired, one glass can be inclined to the other to control light reflections or external sound reflections, but this angle will have a negligible effect on the transmission loss of the window itself. The glass should be isolated from the frame by mounting it on rubber or other pliable strips. The spacing between the two glass panels has an effect. The greater the spacing, the greater the transmission loss. However, there is little advantage in going beyond 8 in, nor serious disadvantage in dropping down to 4 or 5 in. However, a small spacing in a double-pane window, perhaps less than 1 in, can yield a lower STC rating than a single-pane window.

The absorbent material between the pane edges in the design of Fig. 18-10 discourages resonances in the airspace. This adds significantly to the overall sound isolation efficiency of the double window, and it should extend completely around the periphery of the window cavity. If the double window of Fig. 18-10 is carefully constructed, sound isolation should approach that of an STC-50 wall but will probably not quite reach it, particularly if the window area is large. For the staggered-stud wall in which a double window is to be placed, the use of a 2 × 8-in plate instead of the 2 × 6-in plate will simplify mounting of the inner and outer window frames.

Figure 18-11 further illustrates details of a particular observation window design:

- A strong frame of well-seasoned 2 × 10-in or 2 × 12-in wood is necessary for both frame and masonry walls. This frame must be made a part of the wall by packing glass fiber and acoustical sealant at the joints.

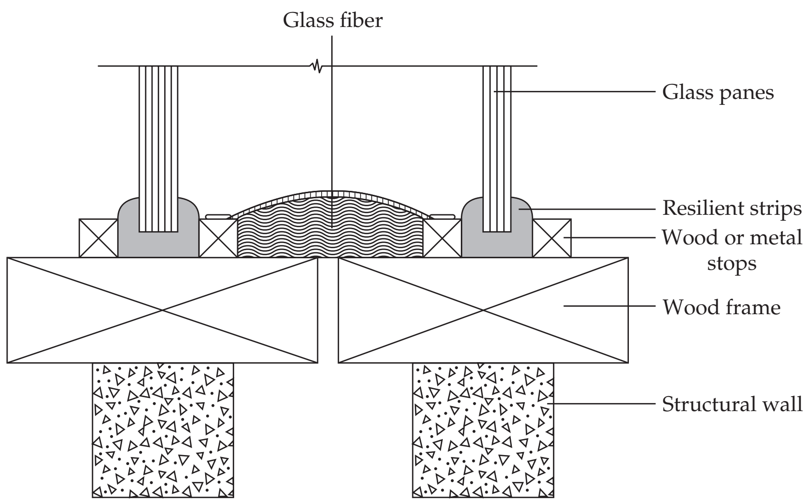


Figure 18-11 Construction details of an observation window.

- The glass panels must be fitted with foam or rubber strips obtained from the glazier. The strip bearing the weight of the glass along the bottom should deflect about 15% under load. The strips on the other three sides can be of lighter foam, as their only function is mechanical isolation and sealing.
- The space between the inner stops should be filled with glass fiber and covered with black cloth or perforated material offering at least 15% in openings.
- The outer stops should be screwed to the frame so that either glass can be removed for cleaning.

Proprietary Observation Windows

A well-constructed studio or other sound-sensitive space deserves the permanently precise treatment of openings that proprietary windows (and doors) provide. These openings are the worst place to economize because deterioration with time is greatest in home-built windows and doors. Figure 18-12 gives cross-sectional views of an STC-38 single-pane window and an STC-55 double-pane window manufactured by Overly Door Company. The transmission-loss curves for these two windows are shown in Fig. 18-13. Aside from a modest coincidence dip in the single-pane window, the curves show good performance.

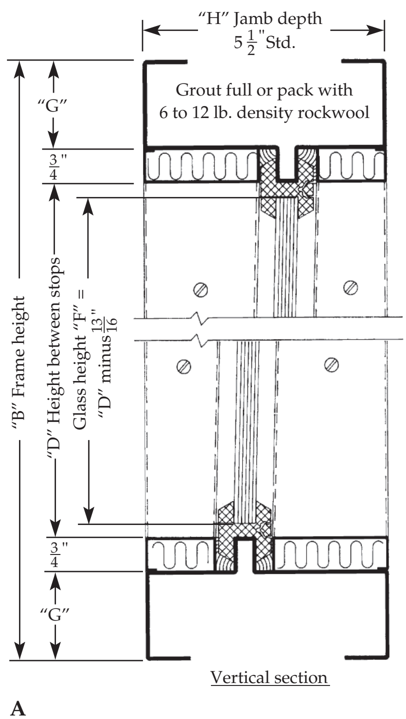


Figure 18-12 Overly proprietary windows. (A) STC-38 single-pane window. (B) STC-55 double-pane window. (*Overly Door Company.*)

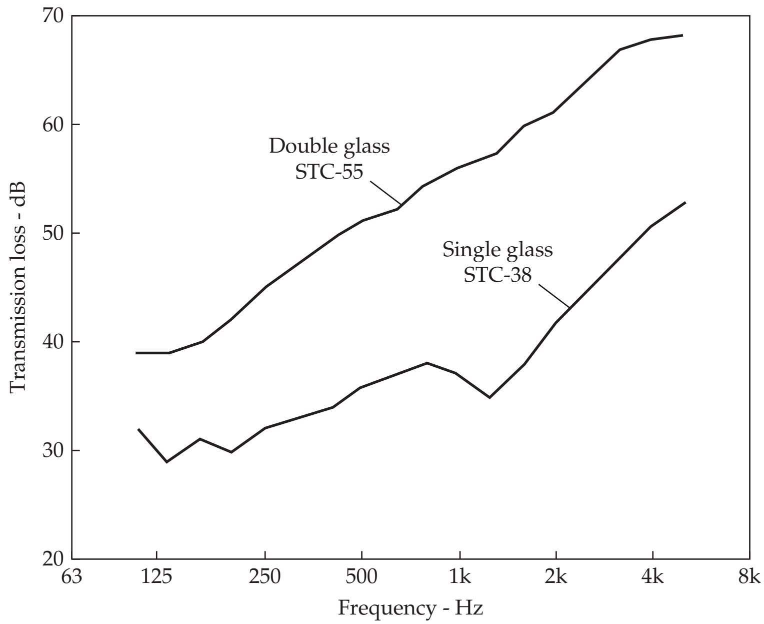


FIGURE 18-13 The sound transmission loss of two Overly windows (see Fig. 18-12). (*Overly Door Company*.)

Sound-Isolating Doors

In the construction of acoustically sensitive spaces, doors are probably the weakest link of all the sound barriers. An acoustically weak door will defeat the isolation provided by an acoustically strong wall. As an extreme example, louvered doors provide little more isolation than an open door. A fixed, sealed window can provide good isolation, but a door, by nature, is not fixed and is more difficult to seal. In fact, whenever possible, doors should not be placed in partitions where high transmission loss is required.

The transmission loss of a door is determined by its mass and stiffness, and by the air-tightness of its seals. Heavy doors provide better isolation than lightweight doors; for example, a 1-3/4-in solid-core door is better than a same-thickness hollow-core door. Hollow-core household doors hung in the usual way have STC ratings in the low 20s, and solid-core household doors have STC ratings in the upper 20s. Even these low values assume good weather stripping. It is possible to obtain a very slight acoustical improvement by padding both sides of a door. A plastic fabric over a 1-in foam rubber sheet can be quilted with upholstery tacks. However, wear and tear may be a drawback. In addition, any air gap below the door will seriously compromise attempts to increase its transmission loss.

For any serious studio design, household doors are too inferior to be considered and must be replaced by heavier self-sealing doors. The hollow-core household door of Fig. 18-14 offers an STC-20 rating but has only about 15 dB of attenuation over much of the spectrum. A good solid-core door might offer an STC-27 rating but is still ineffective as a comprehensive barrier for noise. The sliding-glass door, rated at STC-26, performs similarly to a solid-core door. The sliding-glass door is mentioned here because it is

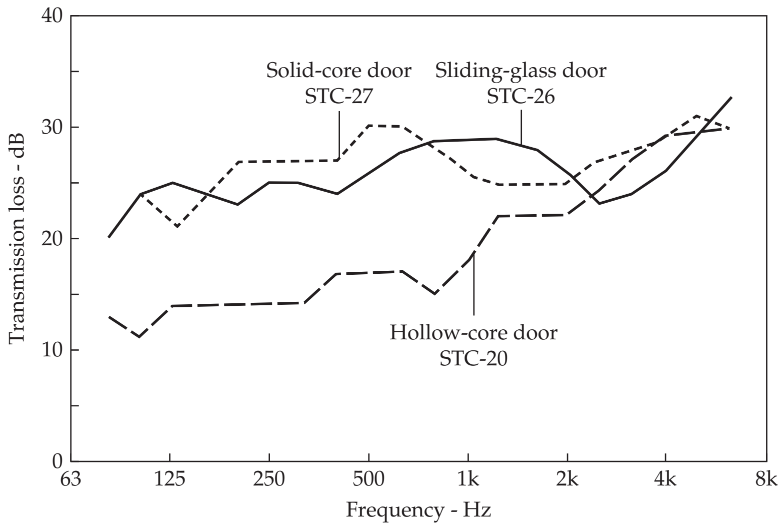


FIGURE 18-14 The transmission loss of three types of doors.

sometimes used for studio drum booths and isolation booths. Although not recommended, a window can be placed in many door types without seriously compromising performance; for example, a 1/4-in plate glass window, 1-ft square and properly sealed, could be used. The only truly satisfactory doors for acoustically sensitive rooms are commercially available metal doors and jambs. They are designed specifically to deliver a certain high transmission loss.

If desired, doors with reasonably good isolating properties can be constructed if the requirements of mass, stiffness, and airtightness are met. Figure 18-15 suggests one

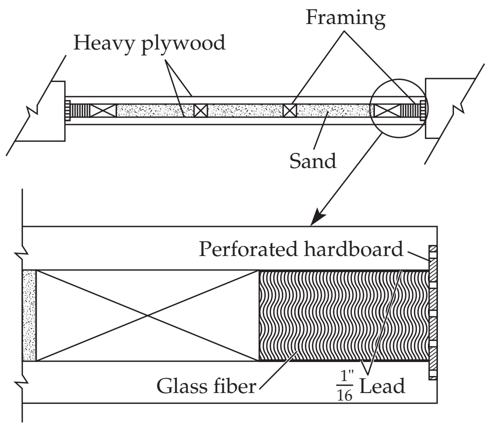


FIGURE 18-15 A reasonably effective and inexpensive acoustical door. Dry sand between the plywood faces adds mass and thus increases transmission loss. Sound traveling between the door and jamb tends to be absorbed by the absorbent door edge.

inexpensive approach to the mass requirement: filling a hollow door with sand. Heavy plywood (3/4 in) is used for the door panels. Mounting hardware and framing must be upscaled as needed to support the added weight.

The isolation of even a good door can be seriously compromised by air leaks. Therefore, the perimeter of any door and frame must be gasketed and airtight. However, achieving a good seal can be difficult. Common weather stripping can be used; several types are available, including strips of felt or neoprene foam with metal backing. These types of seals work well on the tops and sides of doors; however, floor thresholds are subject to wear and tear from foot traffic, rolling dollies, and so on. Most kinds of weather stripping on thresholds can deteriorate rapidly. The detail of Fig. 18-15 shows one approach to the sound leakage problem in which an absorbent edge built around the periphery of the door serves as a trap for sound traversing the gap between door and jamb. This absorbent trap could also be embedded in the door jamb. Such a soft trap could also be used in conjunction with one of several types of seals.

Some door designs use hinged cams that slightly lift the door as it opens, then lowers the door into its frame when it is closed. In this way, the weight of the door itself presses it tightly against its seals, improving isolation. Because the door lifts up when opened, the threshold seal does not wipe against the floor as it opens; this helps preserve the seal. Similarly, in some door designs, a seal automatically drops down to a threshold when the door is closed and lifts when the door is opened; this helps prevent wear and tear on the threshold seal. No matter which type of door seal is used, seals should be inspected periodically to make sure they remain tight.

Figure 18-16 shows a simple door seal that performs reasonably well. This seal comprises a soft rubber or plastic tubing an inch or less in outside diameter with a wall thickness of about 3/32 in. Wooden nailing strips hold the tubing to the door frame by means of a pliant wrapper. A raised sill is required at the floor if the tubing method is to be used all around the door; alternatively, another type of seal such as weather stripping could be used at the bottom of the door. An advantage of a tubing seal is that the degree of compression of the tubing upon which the sealing properties depend is available for inspection.

A complete door plan is shown in Fig. 18-17. It is based on a 2-in-thick solid slab door and utilizes a magnetic seal such as used on refrigerator doors. The magnetic material is barium ferrite in a PVC (polyvinyl chloride) rod. In pulling toward the mild steel strip, a good seal is achieved. The aluminum cover strip decreases sound leakage around the periphery of the door.

A door with transmission loss that equals that of a 50-dB wall requires great care in design, construction, and maintenance. The Overly Model STC488861, shown in Fig. 18-18, is an example of a commercially available acoustical door. It is 1-3/4-in thick with

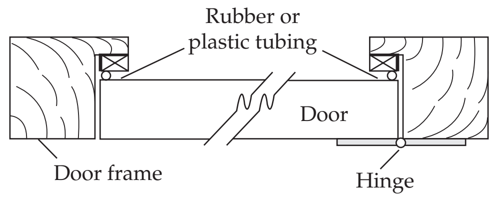
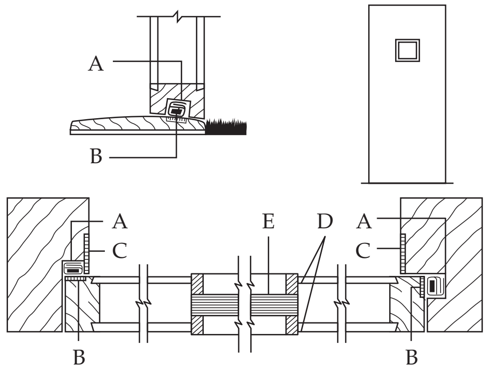


Figure 18-16 A door can be sealed by compressible rubber or plastic tubing held in place by a fabric wrapper.



A - PVC magnetic seal

B - Mild steel $\frac{3''}{4} \times \frac{1''}{8}$ flat

C - Aluminum cover strip $1\frac{1''}{2} \times \frac{1''}{8}$

D - Aluminum glazing beads $\frac{3''}{4} \times \frac{3''}{8}$

E - Polished plate glass $\frac{5''}{4}$

FIGURE 18-17 A door design utilizing magnetic seals of the type used on refrigerator doors.

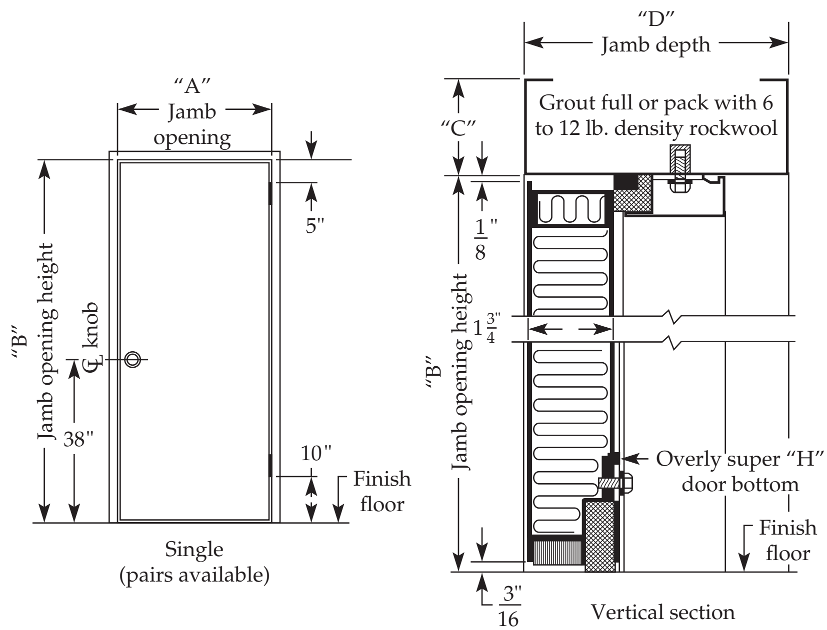


FIGURE 18-18 An Overly rated door. (*Overly Door Company.*)

a surface density of 9.9 lb/ft^2 and is rated at STC-48. Cam-lift hinges are used; a rubber seal is compressed downward against a smooth, raised threshold. This door is an example of an engineered door that will yield superior results both immediately and over time.

Individual door design is critical in achieving high transmission loss, but the way in which doors are placed in a room is also an important factor. For example, a door leading to a corridor should not face an opposite door on the corridor; this would allow sound to more easily pass between the rooms.

Sound Locks

Sound locks are small vestibules with two doors; they are very effective at providing sound isolation. Essentially, because the two doors are placed in series, they can potentially provide a much higher STC rating than a single door. Also, the doors act as independent sound barriers; sound transmission loss is improved because one door can be closed while the other is opened. Sound locks are a common feature in recording studios; the vestibule is placed between the studio and the control room so that loud levels in one room will be less disturbing in the other. Because medium transmission-loss doors can be employed, sound locks are often used to avoid the expense of high transmission-loss doors. However, they require a relatively large floor space.

For optimal and long-lasting performance, the doors of the sound lock should be specially built acoustical doors. For improved performance, all inner surfaces of the sound lock vestibule should be very absorbent. In this way, even if both doors are simultaneously open, the absorption of the vestibule will offer a degree of noise reduction. The ceiling could be suspended and use Tectum ceiling tile. The walls could use the same furred out (C-40 mounting) Tectum wall panel. Tectum is made of wood fibers and is quite resistant to abrasion. Alternatively, the upper parts of the walls could be constructed with glass-fiber batting behind stretched fabric or wire screen; sturdier wainscoting could be used on the lower walls. As another option, Sonex absorbing panels could be used. The floor should be heavily padded and carpeted, but able to withstand the wear and tear of equipment dollies. Tectum products are manufactured by Armstrong World Industries, Inc. Sonex products are manufactured by Pinta Acoustic, Inc.

A short sound-lock vestibule corridor might not be sufficient. For example, anyone hurriedly exiting from either the control room or the studio would go quickly through two doors and a burst of sound could exit or enter. Moreover, a short corridor offers less absorption. Therefore, ideally, although it is never economical to do so, the sound-lock corridor should be as long as possible.

Composite Partitions

A composite partition is one with disparate elements, for example, a wall with a door and a window. The total sound power passing through a composite partition is the sum of the sound powers transmitted by each element at some specified frequency. Each element's sound power is calculated by multiplying its area S_i by its transmission coefficient τ_i. The total transmitted sound power is thus $\Sigma S_i\tau_i$. Alternatively, STC may be used instead of τ, as described later.

Ideally, to avoid a weak link in a composite partition, each element should convey the same transmitted sound power. The TL of different elements is not necessarily equal, but their $S\tau$ products ideally should be equal. For example, if a wall has a large window comprising 20% of the total area, and a door comprising 3.5% of the area, the respective TL values for the wall, window, and door should be 33.6, 27.8, and 20.2 dB. This would yield a composite TL of 30 dB at some specified frequency.

Any weak link in a composite partition is an issue of real concern. For example, suppose that a brick wall provides a TL of 50 dB. When a window with a TL of 20 dB, occupying 12.5% of the total area, is introduced, the composite wall/window TL drops to 29 dB. The damage has been done, and a larger-sized window is not much worse; for example, if the window occupies half of the partition, the composite TL is 23 dB. As noted earlier, any air leak will seriously compromise the isolation of any barrier. For example, a heavy metal plate with holes occupying only 13% of the total area can transmit as much as 97% of the sound impinging on it. Furthermore, if the open area remains, increasing the mass of the metal plate will have little or no effect.

Figure 18-19 illustrates a graphical method to determine how much a high-transmission-loss wall will be compromised by mounting a window or door that has a lower transmission loss. Figure 18-19 applies to a single TL measurement on the TL curve. This enables the building of an accurate point-by-point graph of the new TL curve, with the weak window/door mounted in the strong wall. This assumes that a measured transmission-loss curve is available for both the window/door and the wall.

The following approach uses STC figures from Fig. 18-19; this simplification limits accuracy, but the results are still useful. For example, assume 1,000 ft^2 of wall and 100 ft^2 of weaker window. Also assume that the wall area has an STC-50 rating, and the window an STC-30 rating.

1. Determine the ratio of the glass area to the wall area (100/1,000 = 0.10), and find this value on the horizontal scale of the graph.
2. Determine the difference between the STC of the wall and that of the glass (50 − 30 = 20), and find the 20-dB contour on the graph.
3. The 0.10 vertical line intersects the 20-dB line; follow it to the left scale and read 10 dB.
4. Subtract this 10 dB from the wall STC (50 − 10 = 40).

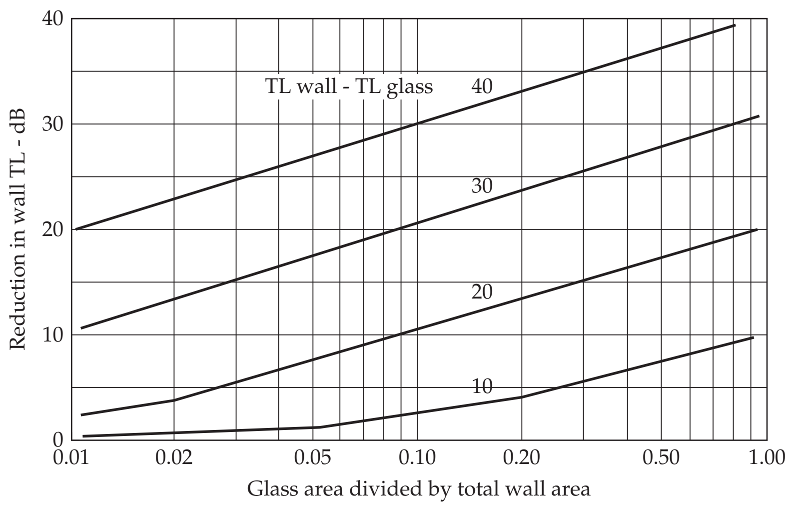


Figure 18-19 Graphical method for estimating the effect of a weak window in a strong wall.

The new STC of the weakened wall is thus shown to be STC-40, which we acknowledge is an approximate (but convenient) result.

Key Points

- When a window or door is placed in a wall, their sound transmission losses must be comparable to that of the wall itself.
- Whereas many windows are designed to be permanently sealed, useful doors are designed to be opened. This makes it more difficult to achieve a reliable seal against sound leaks.
- Single-pane windows, such as the common household window, provide relatively poor sound isolation.
- The additional mass provided by additional panes provides greater loss. In many applications, double-pane windows can provide excellent results.
- Several different resonance effects alter the shape of the transmission-loss curve of any glass window, creating a so-called acoustical hole and lowering its attenuation in a frequency region.
- The airspace cavity between the two panes in a double-pane window is capable of supporting standing waves, much as in a room.
- Isolation is improved when glass panes are of different thicknesses, and widely spaced. Windows with panes of the same thickness and narrow spacing perform relatively poorly.
- For most window designs, the small advantage of a three-pane window over a two-pane window does not justify the added cost.
- A large observation window between a studio and control room requires multiple panes, panes that are thick, separation of panes, absorbent in the cavity, and airtight and resilient mounting in the partition.
- In the construction of acoustically sensitive spaces, doors are probably the weakest link of all sound barriers. An acoustically weak door will defeat the isolation provided by an acoustically strong wall.
- The only truly satisfactory doors for acoustically sensitive rooms are commercially available metal doors and jambs. They are designed specifically to deliver a certain high transmission loss.
- The isolation of even a good door can be seriously compromised by air leaks. Therefore, the perimeter of any door and frame must be gasketed and airtight.
- Sound locks are small vestibules with two doors; they are very effective at providing sound isolation. Because the two doors are placed in series, they can potentially provide a much higher STC rating than a single door.
- A composite partition is one with disparate elements, for example, a wall with a door and a window. The total sound power passing through a composite partition is the sum of the sound powers transmitted by each element at some specified frequency.

CHAPTER 19

Noise Control in Ventilating Systems

In any acoustically sensitive room, interfering sounds can come from other rooms inside the building or from sources outside the building. Some rooms have their own internal noise problems, such as noise generated by equipment cooling fans. But there is one source of noise that is common to almost all acoustically sensitive rooms, and that is the noise coming from the HVAC (heating, ventilating, and air-conditioning) system. This includes noise and vibration from the HVAC system's motors and fans as well as the sound of the airflow through the system's ducts, diffusers, and grilles. The minimization of these noises is the subject of this chapter.

The control of heating and air-conditioning noise and low-frequency vibration can be expensive. A strict noise specification in an air-handling contract for a new structure can escalate the price. Alterations of an existing air-handling system to correct high noise levels can be even more expensive. It is important for studio designers to have an understanding of potential noise problems in air-handling systems so that adequate control and supervision can be exercised during planning stages and installation. This applies equally to the most ambitious professional studio and to a home listening room. Quiet HVAC air can be obtained by considering five factors: careful specification and installation of noise sources such as motors, fans, and grilles; use of sound absorbers in machine rooms and in air ducts to reduce noise; minimization of air turbulence noise in ducts and at duct openings; minimization of acoustical crosstalk between rooms; and reduction of fan speed and air velocity.

Selection of Noise Criteria

One of the first decisions concerning any specification of background noise is the selection of a noise-level goal. The essential question of "How quiet should it be?" is complicated by the fact that the noise level across the entire audio spectrum must be considered. Moreover, because our hearing response is not flat with regard to amplitude or frequency, we must assume that the noise floor criteria will not be flat either. To help quantify the specification of noise criteria, and to more easily communicate it, several noise criteria standards have been devised.

One approach to the question of noise specification is embodied in the family of balanced noise criteria (NCB) curves, as shown in Fig. 19-1. The selection of one of these NCB contours, ranging from NCB-10 to NCB-65, establishes the maximum allowable constant noise-pressure level in each octave band. Expressing the noise goal in this form

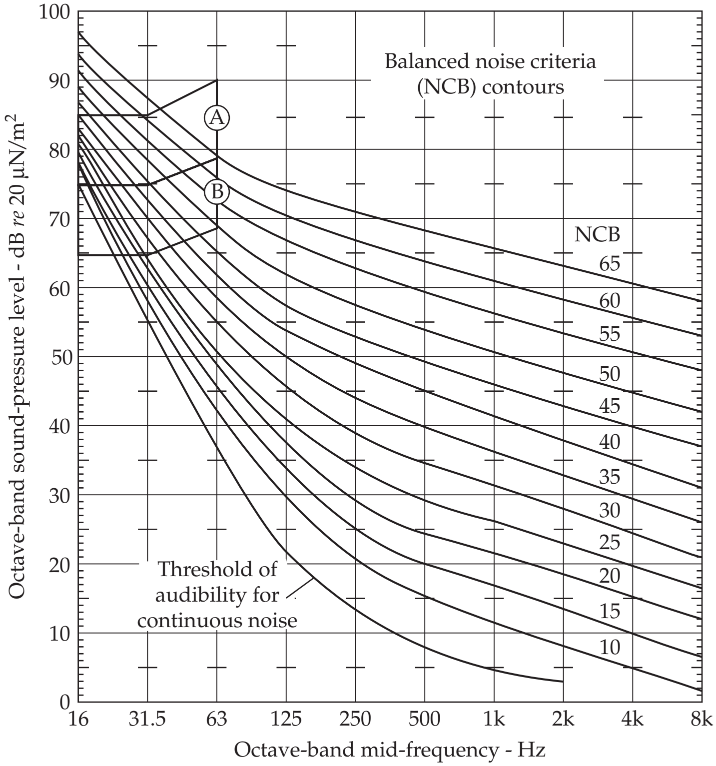


Figure 19-1 Balanced noise criteria (NCB) contours for occupied rooms. (*Beranek.*)

makes it easier to specify, measure, and verify a room's noise levels. The contours are patterned after human hearing equal-loudness contours. The downward slope of these contours reflects both the lower sensitivity of the human ear at low frequencies and the fact that most noises with distributed energy drop off with frequency. The NCB-0 curve represents the threshold of audibility for continuous sound. The use of NCB curves is described in *ANSI/ASA S12.2-2019 Criteria for Evaluating Room Noise*, and in the *American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Handbook*.

To determine whether the noise in a given room meets the contour goal selected, the sound-pressure level is measured in an unoccupied building. Readings are made in each octave at 16-Hz to 8-kHz center frequencies, using a sound-level meter equipped with octave filters. Results are plotted on the standard graph of Fig. 19-1, and a "curve fitting" procedure is used to compare readings to the family of standard curves. The nearest NCB contour that is above the plotted readings is shifted downward, so that the shifted NCB contour is tangent to the plotted readings. The amount of the shift is noted in decibels. The rating of the measured noise level is the shifted NCB contour minus the shift. For example, an NCB-25 contour shifted down by 2 dB yields a measured rating of NCB-23. If the NCB-25 contour had been specified as the highest permissible sound-pressure level in an air-conditioning contract, the HVAC installation in this example would be acceptable.

NCB values are further defined by comparing the perceived balance between low- and high-frequency sounds. The value is then rated as N (neutral), H (hissy), or R (rumbly), depending on the predominant spectral response. Neutral represents a balanced noise spectrum. Hissy defines a noise that contains excessive energy in the 1 to 8-kHz region. Rumbly defines noise that contains excessive energy in the 16 to 500-Hz region. For example, a NCB-25(R) value would indicate a quiet room, but with some rumble below 500 Hz that exceeds the NCB contour by at least 3 dB. The region below 63 Hz (marked by A and B in Fig. 19-1) is further specified according to RV (vibration). Octave-band sound-pressure levels of the magnitudes indicated may induce rattles or vibrations in lightweight partitions and ceiling constructions. As with any single-value audio specification, NCB values can sometimes be misleading because they do not account for specific frequency-response differences. For example, two rooms might have the same NCB but have very different acoustical spectra.

In some cases, room criteria (RC) values are specified. This method is similar to the NCB method. RC contours are measured from 16 Hz to 4 kHz, and range from RC-25 to RC-50, as shown in Fig. 19-2. The contours are a series of straight-line −5 dB/octave slopes. The contours are not associated with equal-loudness contours, but instead represent perceived neutral background noise. Each line is given an RC value that corresponds to the SPL level at 1 kHz. For example, the RC-25 line passes through 25 dB SPL

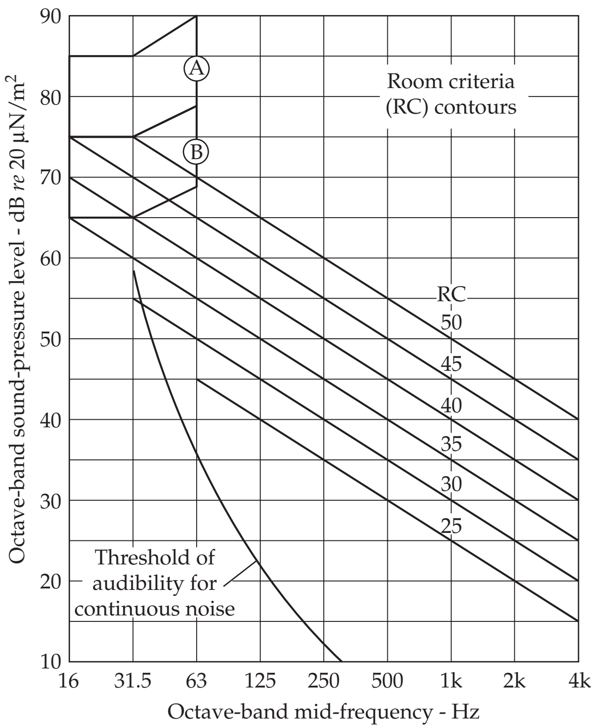


Figure 19-2 Room criteria (RC) contours for occupied rooms.

at 1 kHz. To make an RC determination, octave measurements are taken, an average is computed, and values are compared to the standard RC contours. Similar to NCB, RC values are further defined as N (neutral), H (hissy), or R (rumbly), depending on the predominant spectral response. The region from 16 to 63 Hz is scrutinized for RV (vibration). If vibrations are present, V is added to the R designation, for example, RC-30(RV). RC curves are also standardized in the *ANSI/ASA S12.2-2019* document.

What criteria level should be selected as the allowable limit for background noise in an acoustically sensitive space such as a recording studio? This depends on the general studio quality level to be maintained, the use of the studio, and other factors. The advent of digital audio recording changed our view of which RC (or NCB) contour to select as a goal. Signal-to-noise ratios of more than 100 dB must be accommodated. This means a lower acoustical noise floor. A lower noise floor means stricter construction practices and HVAC contract noise specifications that substantially increase the cost of building or renovating.

When specifying a noise level, the acoustical context of the room must be considered. For example, there is little point in demanding RC-15 from the air-handling system when intrusion of traffic and other noise is higher than this. In general, RC-20 should be the highest contour that should be considered for a recording studio or listening room, and RC-15 is suggested as a practical and attainable design goal for the average studio. RC-10 would be excellent, but it would require additional effort and expense to reduce ambient noise to this level. Table 19-1 lists several types of rooms and suggestions for RC values. The objective in any room design is to achieve a noise level appropriate for the functions of the space—not the lowest possible noise level. Overdesign is as unforgivable as underdesign.

Fan Noise

In noise control, the most efficient solution is to minimize the noise at the source. Machinery and fans are chief contributors to HVAC noise (but not the only culprits) and often require source noise control. Fan noise will travel through a duct, gradually decreasing in intensity as energy is absorbed. Some fan noise will leave the duct openings in a room, whereas other noise will radiate into a room along the length of the duct itself. The sound power output of a fan is largely determined by the air volume and pressure required in the installation, but there are variations between types of fans. Figure 19-3 gives the specific sound power output of two types of fans: the airfoil centrifugal fan and the pressure blower fan. Specific sound power level means that the measurements have been reduced to the standard conditions of 1 ft^3/min and a pressure of 1 in of water. On this basis, noise of various types of fans can be compared equitably. The airfoil centrifugal fan is one of the quietest types of fans available. Large fans are generally quieter than small fans. This is also true of the pressure blower fan, as shown in Fig. 19-3.

All the acoustical power radiated by a piece of machinery must flow out through a hemisphere. This can be measured as a specific sound power level, and is usually evaluated by sound-pressure level readings over the hemisphere. Sound power is proportional to sound pressure squared. Noise ratings of fans in terms of sound power can be converted back to sound-pressure levels applicable to a given room by means of the following formula:

$$L_{\text{pressure}} = L_{\text{power}} - 5 \log V - 3 \log f - 10 \log r + 25 \text{ dB} \tag{19-1}$$

Type of Room	Recommended RC Level (RC Curve)	Equivalent Sound Level (dBA)
Apartments	25–35 (N)*	35–45
Assembly halls	25–30 (N)	35–40
Churches	30–35 (N)	40–45
Concert and recital halls	15–20 (N)	25–30
Courtrooms	30–40 (N)	40–50
Factories	40–65 (N)	50–75
Legitimate theaters	20–25 (N)	30–65
Libraries	35–40 (N)	40–50
Motion picture theaters	30–35 (N)	40–45
Private residences	25–35 (N)	35–45
Recording studios	15–20 (N)	25–30
Restaurants	40–45 (N)	50–55
Sport coliseums	45–55 (N)	55–65
TV broadcast studios	15–25 (N)	25–35
Hospitals/clinics—private rooms	25–30 (N)	35–40
Hospitals/clinics—operating rooms	25–30 (N)	35–40
Hospitals/clinics—wards	30–35 (N)	40–45
Hospitals/clinics—laboratories	35–40 (N)	45–50
Hospitals/clinics—corridors	30–35 (N)	40–45
Hospitals/clinics—public areas	35–40 (N)	45–50
Hotels/motels—individual rooms or suites	30–35 (N)	35–45
Hotels/motels—meeting or banquet rooms	25–35 (N)	35–45
Hotels/motels—service and support areas	40–45 (N)	45–50
Hotels/motels—halls, corridors, lobbies	35–40 (N)	50–55
Offices—conference rooms	25–30 (N)	35–40
Offices—private	30–35 (N)	40–45
Offices—open-plan areas	35–40 (N)	45–50
Offices—business machines/computers	40–45 (N)	50–55
Schools—lecture and classrooms	25–30 (N)	35–40
Schools—open-plan classrooms	30–40 (N)	45–50

*Neutral (N). The levels in the octave bands centered at 500 Hz and below must not exceed the octave-band levels of the reference spectrum by more than 5 dB at any point in the range. The levels in the octave bands centered at 1 kHz and above must not exceed the octave-band levels of the reference spectrum by more than 3 dB at any point in the range.

Table 19-1 Recommended RC Levels for Different Types of Rooms

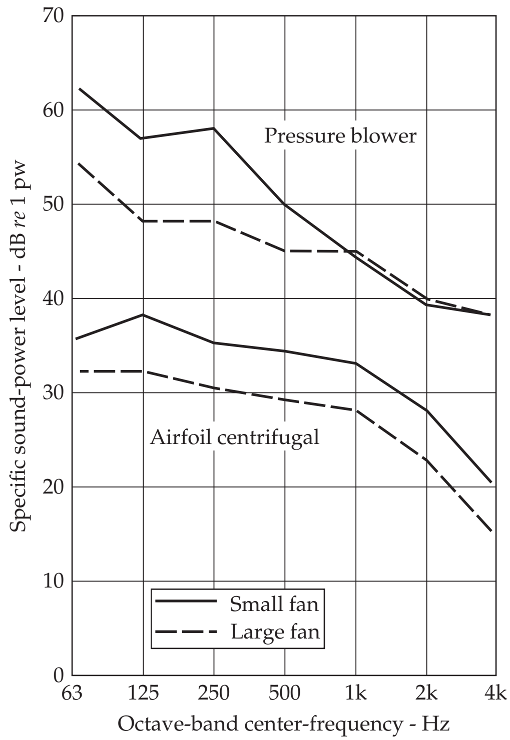


FIGURE 19-3 Noise sound-power output of airfoil centrifugal and pressure blower fans commonly used in HVAC systems. (*ASHRAE Handbook*.)

where $L_{pressure}$ = sound-pressure level, dB
L_{power} = sound-power level, dB
V = room volume, ft^3
f = octave-band center frequency, Hz
r = distance of source to reference point, ft

There is usually a tone generated by fans such that: Frequency = (rev/sec)(number of blades). This tone adds to the sound level of the octave band in which it falls. To account for the contribution of the fan tone, 3 dB for centrifugal fans and 8 dB for pressure fans should be added to the one-octave band level. The manufacturer is the preferred source of noise data for any particular fan.

Machinery Noise and Vibration

A building may have HVAC infrastructure equipment such as pumps, compressors, chillers, and evaporators. In a new room design, the first step in the mitigation of HVAC noise and vibration is to carefully consider the location of the HVAC machinery. HVAC problems can be severe if the HVAC equipment is placed near an acoustically sensitive room. For example, if the equipment room is adjacent to or on the roof directly above a studio, the common wall or roof panel will act as a large diaphragm, vibrating and efficiently radiating noise into the studio.

Therefore, although installation and operating costs may be increased because of longer duct lengths and thermal losses, it is important to locate the equipment as far removed from the acoustically sensitive areas as possible. HVAC equipment should be located on the opposite side of the building, or in a room or structure that is detached from any sensitive rooms. If an air-handling unit must be placed on a roof directly over a sensitive room, it should be positioned over a load-bearing wall, and not in the middle of a roofing span, to minimize diaphragmatic action. Similarly, to minimize vibration, equipment should preferably be placed on the ground, as opposed to a higher floor.

The next step is to consider some form of source isolation against structureborne vibration. In most cases, air-handling equipment is placed in a machinery room. If the equipment is to rest on a concrete slab shared with the building, the machinery-room floor slab should be isolated from the main-floor slab as a floating-floor slab. Compressed and treated glass-fiber strips should be used to separate slabs during pouring. Vibration sources such as motors must be decoupled from the building structure by placing the motor on rubber or glass-fiber pads, coiled springs, or similar devices. In many cases, the motor is mounted on an isolation block, which in turn is isolated with pads or springs from a floating floor. This isolation block, sometimes called an inertia base, is usually made of concrete. It increases the total mass of the system, which reduces vibrations and, in particular, lowers the natural frequency (described later) of the system. This in turn improves the isolation efficiency of the system. Care must be taken to ensure that any isolation method provides complete decoupling; if a solid piece mechanically "short circuits" the decoupling, then isolation can be severely compromised. An example of a mounting method for machinery is shown in Fig. 19-4.

Air-handling units should be placed within noise-control enclosures. Pipes and ducts should be mounted from isolation hangers. Penetrations, where they leave the machinery room, should be flexibly sealed. Interior walls and ceiling of the machinery room should be covered with sound-absorbing materials such as glass-fiber boards to reduce the ambient noise level in the machinery room, and thus reduce the required transmission loss of the enclosure. The noise reduction achieved by adding absorption can be calculated from:

$$\mathrm{NR} = 10 \log \frac{A}{A_0} \ \mathrm{dB} \tag{19-2}$$

where A_0 = absorption before treatment
A = absorption after treatment

For example, if an enclosure initially has 100 sabins of absorption at 500 Hz, and after treatment has 1,000 sabins of absorption, then a 10-dB noise level reduction is achieved.

In practice, regardless of the mounting and installation method, some energy is imparted from the vibration source to the building. This can be viewed in terms of transmissibility (T), which can be expressed as the force transmitted to the building structure divided by the force applied by the sound source. If no isolation is used, the transmissibility may be 1.0. Ideally, no energy from the machinery would be transmitted to the building structure, and transmissibility would be zero. In any case, the lower the

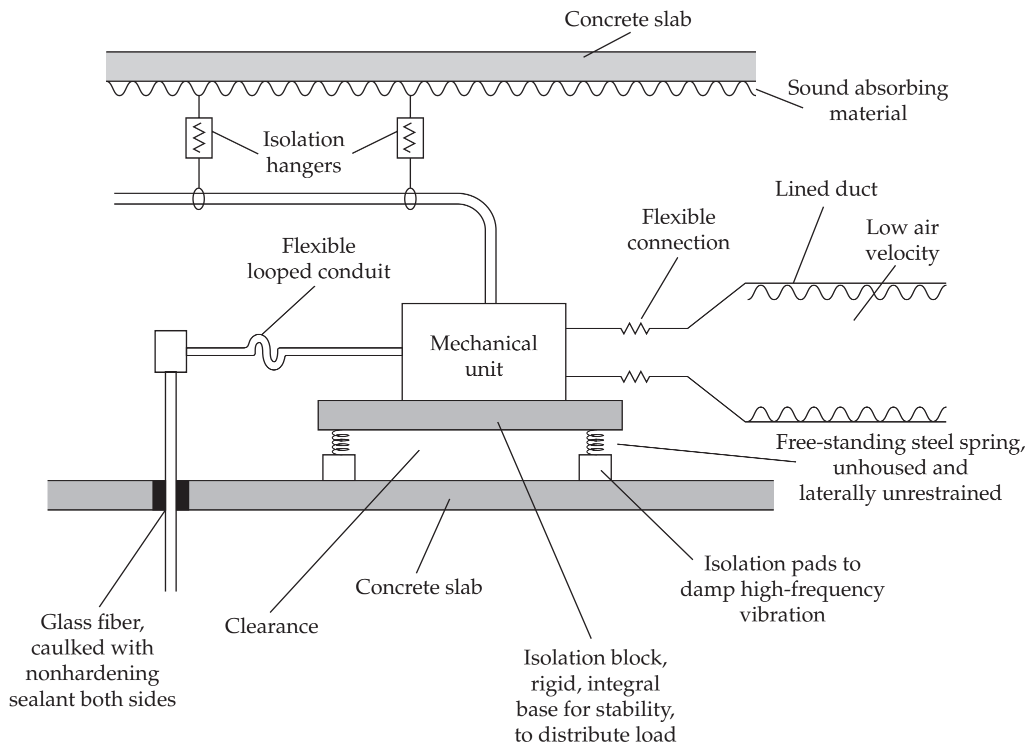


FIGURE 19-4 An example of techniques used to mitigate vibration and noise from equipment in a machine room. Emphasis is placed on isolation mounting and decoupling.

transmissibility, the better. Looked at in another way, transmissibility can be measured by its efficiency; for example, if the transmissibility is 0.35, its efficiency is 65%.

A vibration source such as a motor, mounted on isolation pads or similar means, can be considered as a mass on a spring. As such, it will have a natural frequency of vibration f_n, where it will freely vibrate. The value of f_n is a function of the stiffness of the spring and mass. In addition, f_n is a function of the static deflection of the system. In particular, f_n equals a constant divided by the square root of the static deflection. The static deflection is the decrease in the height of the isolation pad or spring when the equipment is placed on it. The value of deflection is usually provided by the manufacturer. When the deflection is measured in inches, the constant is 3.13. A vibration source will also have its own forcing frequency f. For example, a motor rotating at 1,200 RPM (revolutions per minute) will exhibit a forcing frequency of 20 Hz. When equipment vibrates at several frequencies, the lowest frequency is used in designing the isolation system.

Transmissibility can also be calculated using f and f_n:

$$T = \left| \frac{1}{(f/f_n)^2 - 1} \right| \tag{19-3}$$

where T = transmissibility
f = driving frequency of source, Hz
f_n = natural frequency of system, Hz

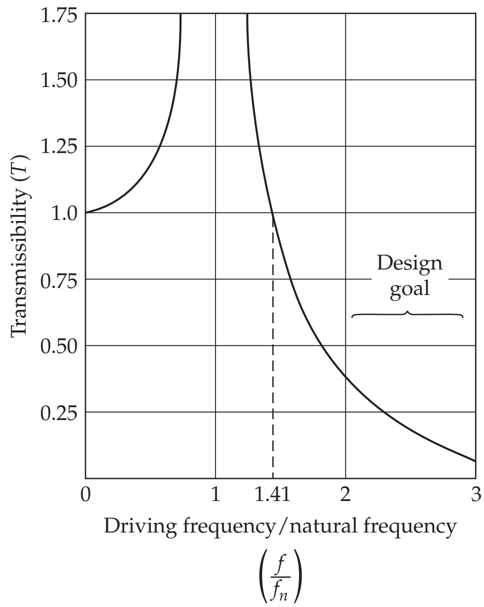


Figure 19-5 Graph of transmissibility versus frequency ratio.

When designing an isolation system, attention must be paid to minimize T. This is done by increasing the value of f/f_n. This, in turn, is usually accomplished in the design by reducing the natural frequency f_n of the system, for example, by specifying a high value for static deflection. Figure 19-5 shows this design goal as a graph of T versus f/f_n. A value of T that is greater than 1.0 would mean that the isolation system is quite unsuccessful; it is amplifying the vibration (and is in resonance when $f/f_n = 1.0$). To avoid that, we see that f/f_n must be greater than 1.41. Many designers aim for a value of f/f_n of about 2 or 3, providing a successful efficiency of about 80%.

Air Velocity

In air distribution systems, the velocity of airflow is a very important factor in keeping HVAC noise at a satisfactorily low level. The higher the airflow velocity, the higher the level of turbulence noise. Other factors can also introduce turbulence noise. Noise generated by airflow varies approximately as the sixth power of the velocity. As air velocity is doubled, the sound level will increase about 16 dB at the room outlet. Some references state that airflow noise varies as the eighth power of the velocity and give 20 dB as the figure associated with doubling or halving the air velocity. In any case, low air velocity is a prerequisite for low noise.

A basic design parameter of an HVAC system is the quantity of air the system is to deliver. There is a direct relationship between the quantity of air, air velocity, and size of the duct. The velocity of the air depends on the cross-sectional area of the duct. For example, if a system delivers 500 ft^3/min and a duct has 1 ft^2 of cross-sectional area, the velocity is 500 ft/min. If the area is 2 ft^2, the velocity is reduced to 250 ft/min; if 0.5 ft^2, velocity is increased to 1,000 ft/min. An air velocity maximum of 500 ft/min is suggested for broadcast studios, recording studios, and other critical spaces. Specifying a large duct size (allowing low air velocity) in initial design will forestall problems later.

High-pressure, high-velocity, small-duct systems are generally less expensive than low-velocity systems. Budget systems commonly create noise problems in studios because of high air velocity and the resulting high noise. Compromise can be made by flaring out the ducts just upstream from the outlet grille and using grilles with large louvered openings, or no grilles at all, in an effort to minimize air turbulence at the grille. The increasing cross-sectional area of the flare results in air velocity at the grille being considerably lower than in the duct feeding it. In some cases, the economy of smaller ducts may be sufficient to pay for silencers to lower the higher noise of higher-velocity flow to acceptable levels.

Even if fan and machinery noise are sufficiently attenuated, by the time the air reaches the acoustically sensitive room, air turbulence associated with nearby 90° bends, dampers, grilles, and diffusers can be serious noise producers, as suggested by Fig. 19-6. Duct transitions and bends should be as rounded or smooth as possible to reduce turbulence; similarly, vanes should be used to direct air smoothly through branch takeoffs. When possible, such fittings should be located 5 to 10 diameters upstream from the outlet to allow turbulence to abate.

Natural Attenuation

When designing an air distribution system, certain helpful natural attenuation effects should be taken into account. Otherwise, if they are neglected, expensive overdesign may result. When a plane wave sound passes from a small space, such as a duct, into a larger space, such as a room, some noise is reflected back toward the source. The effect is greatest for low-frequency sound. Research has also indicated that the effect is significant only when a straight section of ductwork, three to five diameters long, precedes the duct termination. Any terminal device, such as a diffuser or grille, tends to nullify this

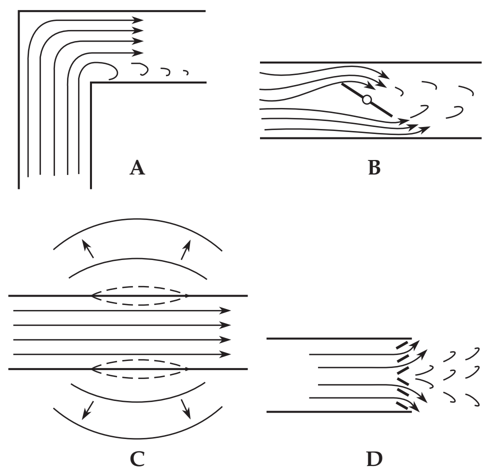


Figure 19-6 Air turbulence caused by discontinuities in the flow path can be a serious source of noise. (A) 90° miter bend. (B) Damper used to control the quantity of air. (C) Sound radiated from duct walls set into vibration by turbulence or noise inside the duct. (D) Grilles and diffusers.

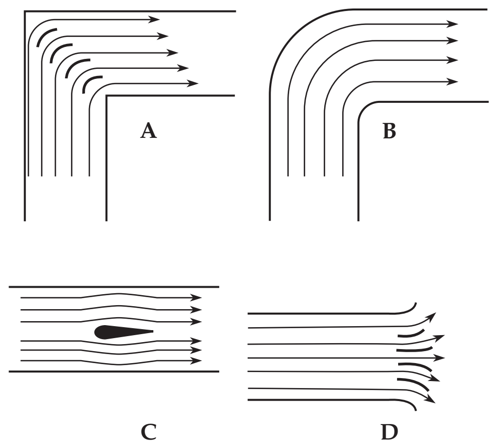


FIGURE 19-7 Air turbulence noise can be materially reduced by using several techniques. (A) Deflectors. (B) Radius bends. (C) Airfoils. (D) Shaped grilles and diffusers.

attenuation effect. A 10-in duct flowing air into a room without a grille can provide a 15-dB reflection loss in the 63-Hz octave. This is about the same attenuation that a 50- to 75-ft run of lined ductwork would give. Figure 19-7 shows several methods of attenuation.

A similar loss occurs at every branch or takeoff. The attenuation depends on the number of branches and their relative area. Two branches of equal area (each one-half the area of the feeding duct) would yield a 3-dB attenuation in each branch. There is also an attenuation of noise in bare, rectangular sheet ducts due to wall flexure amounting to 0.1 to 0.2 dB/ft at low frequencies. Round elbows introduce attenuation, especially at higher frequencies. Right-angle elbows also introduce attenuation, as shown in Fig. 19-8. All of these losses are built into the air-handling system and serve to attenuate fan noise and other noise traveling along the duct.

Duct Lining

In noise control, it is often important to reduce noise along the path the noise travels. The application of sound-absorbing materials to the inside surfaces of ducts is a standard method of reducing noise levels. The attenuation or insertion loss is measured in decibels per length of duct; attenuation is generally poor at low frequencies. Such lining comes in the form of rigid boards and blankets, and in 1/2- to 2-in thicknesses. Typical lining is made of glass fiber coated with a resilient material; in some cases, use of this material is not allowed. Such acoustical lining also serves as thermal insulation when it is required. Generally, the larger the duct size, the lower the insertion loss; this is because there is relatively less material for a given airflow. The approximate insertion loss offered by 1-in duct lining in typical rectangular ducts depends on the duct size, as shown in Fig. 19-9. The approximate insertion loss of round ducts is given in Fig. 19-10. Insertion loss is much lower in round ducts than in lined rectangular ducts with comparable cross-sectional areas.

Care must be taken when ductwork passes through a noisy room on its way to a quiet room. Noise from the noisy room will enter the duct (either through openings or through

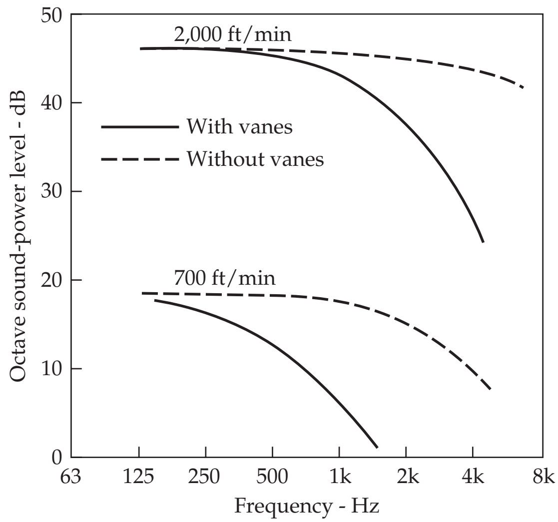


FIGURE 19-8 Noise produced by 12 × 12-in square cross section 90° miter elbow with and without deflection vanes and at 2,000 and 700 ft/min air velocities. Calculations follow ASHRAE procedures.

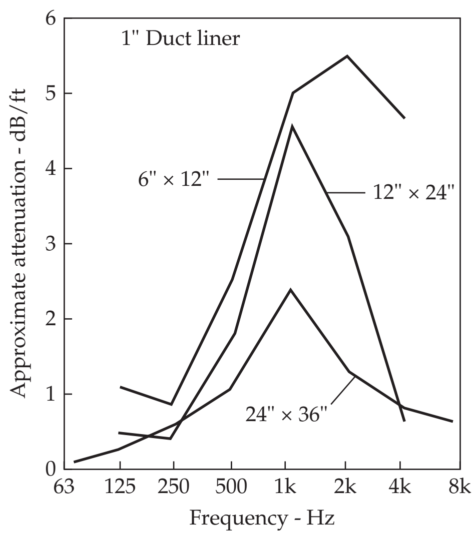


FIGURE 19-9 Insertion loss attenuation (measured in dB/ft) provided by 1-in duct lining on all four sides of rectangular ducts of different cross-section areas. Dimensions shown are the free area inside each duct. Calculations follow ASHRAE procedures.

the duct wall itself) and pass through it to the quiet room, easily violating the transmission loss of a common wall. Noise travels equally upstream or downstream to airflow in the duct. The duct in the noisy room must be isolated to minimize this acoustical crosstalk and attenuation must be provided in the duct between rooms.

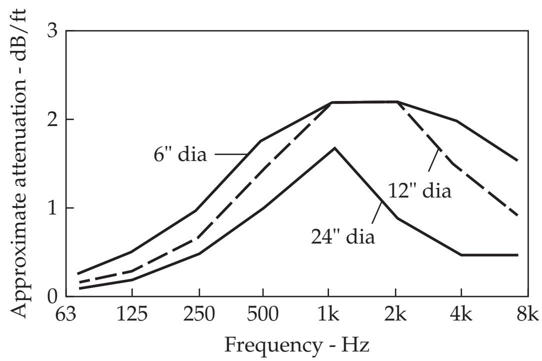


Figure 19-10 Insertion loss attenuation (measured in dB/ft) provided by spiral-wound round ducts with perforated spiral-wound steel liner of different cross-section areas. Dimensions shown are the free area inside each duct. Calculations follow ASHRAE procedures.

Plenum Silencers

A plenum is a space in a building structure between a structural ceiling and some types of dropped ceilings, or a space between a structural floor and some types of raised floors. A plenum can be used as part of an HVAC system to provide a pathway for air handling. A sound-absorbing plenum is an economical way to achieve significant attenuation. Figure 19-11 shows a modest-sized plenum chamber, which if lined with 2-in thickness of 3 lb/ft^3 density glass fiber, will yield a maximum attenuation of about 21 dB. The attenuation characteristics of this plenum are shown in Fig. 19-12 for two thicknesses of lining. With a lining of 4 in of fiberboard of the same density, quite uniform absorption is obtained across the audible spectrum. With glass-fiber board 2-in thick, attenuation falls off below 500 Hz. It is apparent that the attenuation performance of a plenum of a given size is determined primarily by the lining.

Figure 19-13 gives measurements for a practical, lined-plenum muffler approximately the same horizontal dimensions as that of Fig. 19-11, but only half the height and with baffles inside. Attenuation of 20 dB or more above the 250-Hz octave is realized in this case.

Plenum performance can be improved by increasing the ratio of the cross-sectional area of the plenum to the cross-sectional area of the entrance and exit ducts, and by

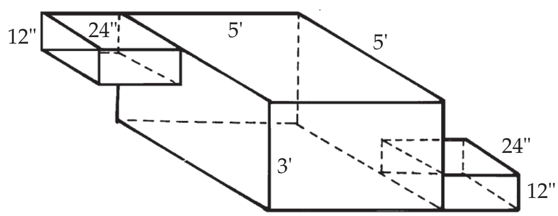


Figure 19-11 Dimensions of a plenum that will yield about 21-dB of attenuation throughout much of the audible spectrum. (*ASHRAE Handbook.*)

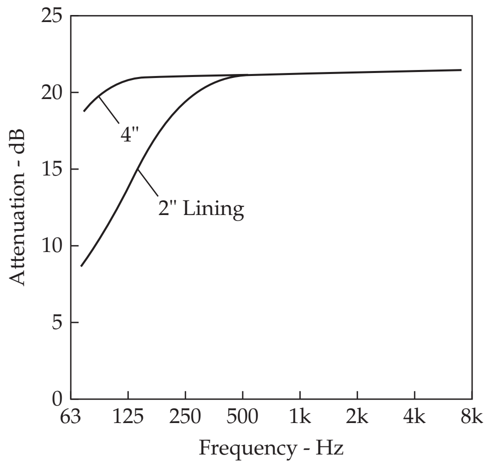


FIGURE 19-12 Calculated attenuation characteristics of the plenum of Fig. 19-11 lined with 2 and 4-in glass fiber of 3 lb/ft^3 density. Calculations follow ASHRAE procedures.

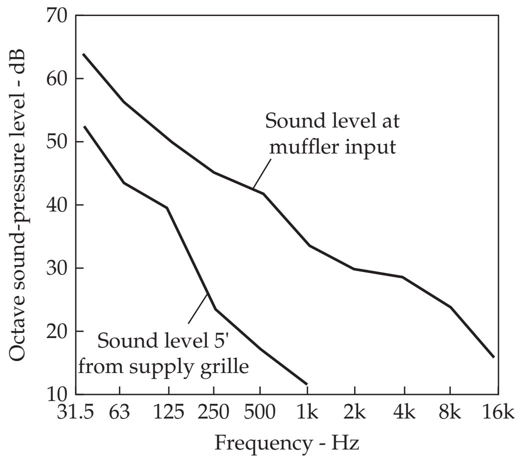


FIGURE 19-13 Noise sound-pressure levels measured at two locations: plenum muffler input, and 5 ft from the supply grille. This plenum muffler is approximately the size of that shown in Fig. 19-11, but only half the height and contains baffles.

increasing the amount or thickness of absorbent lining. A plenum located at the fan discharge can be an effective and economical way to decrease noise entering the duct system. In some cases, unused attic spaces can be used as plenums for noise attenuation.

Proprietary Attenuators

Numerous noise attenuators are available. Cross sections of several types are shown in Fig. 19-14 with their performance plotted below. For comparison, the attenuation of a simple lined duct is given in curve A. Some attenuators have no line-of-sight through

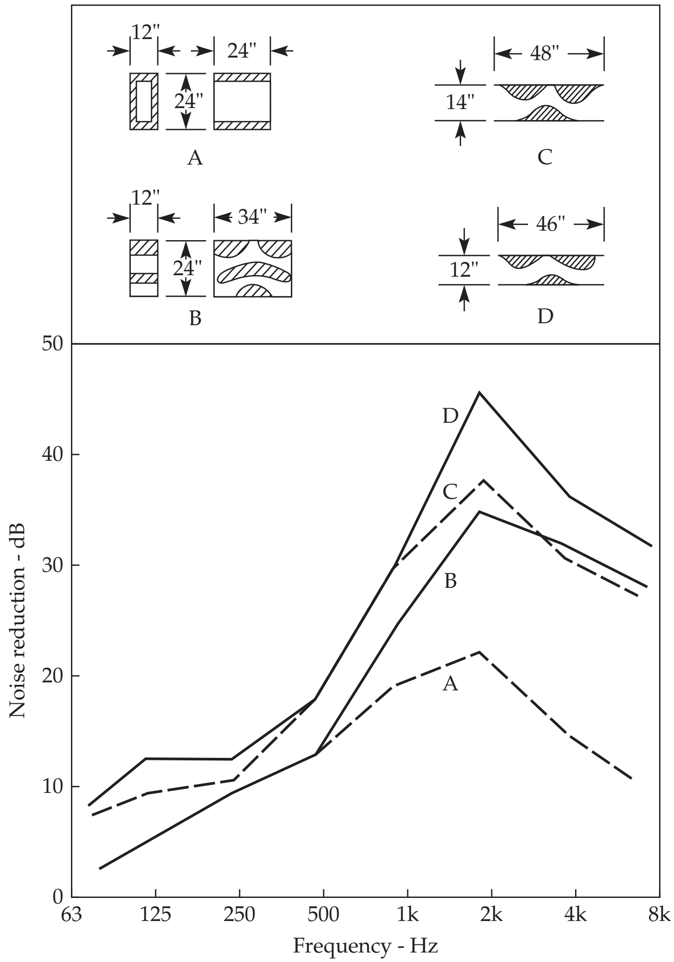


FIGURE 19-14 Attenuation characteristics of three proprietary silencers (B, C, D) compared to the attenuation of a typical lined duct (A). (*Doelling.*)

them; that is, the sound must be reflected from the absorbing material to traverse the unit and hence will have somewhat greater attenuation. In these packaged attenuators, the absorbing material is usually protected by perforated metal sheets. The attenuation of such units is generally very good at midband speech frequencies but not as good at low frequencies.

Reactive Silencers

Some silencers use the principle of an expansion chamber, as shown in Fig. 19-15. These attenuators function by reflecting sound energy back toward the source, thereby canceling some of the sound energy (similar to an automobile exhaust muffler). Because there is both an entrance and exit discontinuity, sound is reflected from two points. The destructive interference (attenuation nulls) alternates with constructive interference

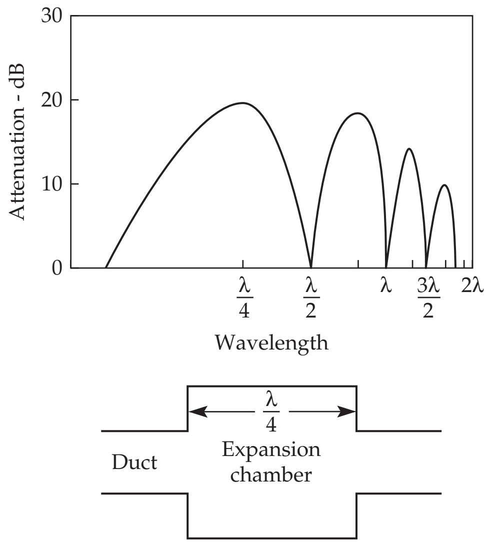


Figure 19-15 Attenuation characteristics of a reactive silencer using an expansion chamber. Sound is attenuated by virtue of the energy reflected back toward the source, canceling some of the oncoming sound. (*Sanders*.)

(attenuation peaks) across the frequency band; the attenuation peaks becoming lower as frequency is increased. These peaks are not harmonically related; therefore, they do not produce high attenuation for a noise fundamental and all of its harmonics, but rather attenuate bands of the spectrum. By tuning, however, the major peak can eliminate the fundamental, while most of the harmonics, of much lower amplitude, would receive some attenuation. Two reactive silencers can be placed in series and tuned so that one covers the nulls of the other. Thus, continuous attenuation can be realized throughout a wide frequency range. No acoustical material, such as an absorber, is required in this type of silencer.

Tuned Silencers

The resonator silencer, illustrated in Fig. 19-16, is a tuned stub that provides high attenuation at a narrow band of frequencies. Even a small unit of this type can produce 40 to 60 dB of attenuation and is effective at reducing tonal HVAC noise, such as fan tones. This type of silencer offers little constriction to airflow, which can be a drawback to other types of silencers.

Duct Location

Careful design is needed when an air-handling system is common between two acoustically sensitive rooms. Care must be taken to minimize duct transmission between them. For example, it would be a design error to specify an STC-60 wall between a studio and control room and then serve both rooms with the same supply and exhaust ducts closely spaced, as in Fig. 19-17. The duct would create a short path speaking tube

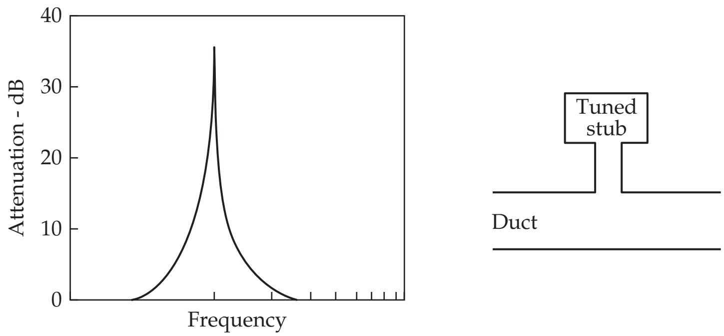


Figure 19-16 Attenuation characteristics of a tuned-stub silencer. *(Sanders.)*

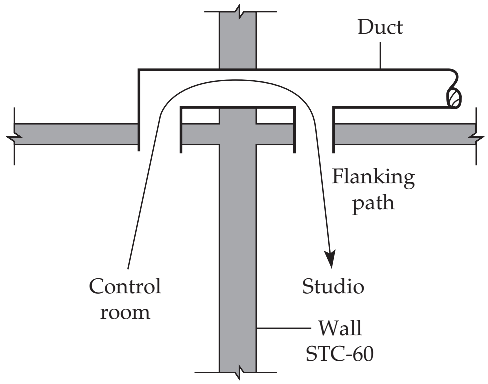


Figure 19-17 A high-transmission-loss wall can be bypassed by sound traveling over a short duct path from grille to grille.

from one room to the other, quite effectively nullifying the transmission loss of the STC-60 wall. With an untrained air-conditioning contractor doing the work, not understanding potential acoustical problems, such errors can easily happen. To obtain sufficient attenuation in the duct system to match the transmission loss of the STC-60 wall would require the application of many of the principles discussed earlier. Figure 19-18 suggests two approaches to the problem; to separate grilles as far as possible if they are fed by the same duct, or better yet, to serve the two rooms with separate supply and exhaust ducts.

ASHRAE

The ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) organization is a prolific resource of HVAC data for the acoustical designer. In this chapter, it is impractical to go deeply into specific design technologies, but the *ASHRAE Handbook* can be a useful guide. It is published in a series of four volumes, one of which is revised each year, ensuring that no volume is older than four years.

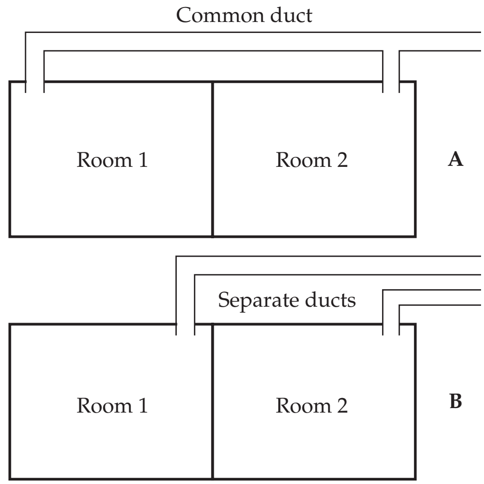


Figure 19-18 Two possible solutions to the problem shown in Fig. 19-17. (A) Separate grilles as far as possible. This approach increases duct path length and thus also increases duct attenuation. (B) Provide each room with a separate duct. This isolates one duct system from the other, providing potentially excellent attenuation.

They introduce fundamental principles, including the source/path/receiver concept, basic definitions and terminology, and acoustical design goals. Other topics include noise control for outdoor equipment installations, system noise control for indoor air-conditioning systems, general design considerations for good noise control, mechanical-equipment-room noise isolation, vibration isolation and control, and troubleshooting techniques for noise and vibration problems. ASHRAE also publishes standard practices, for example, methods to estimate octave noise levels generated by air-handling systems.

Active Noise Control

Devices such as isolation pads, isolation hangers, lined ducts, and silencers all use passive means to control noise. Active noise control devices use more sophisticated methods such as digital signal processing to reduce noise, mainly inside ducts. Noise can be canceled by radiating a replica of the noise signal in inverse phase and combining the original signal with the replica. The canceling noise is equal in frequency and level, but out of phase with the original unwanted noise, resulting in destructive interference and hence noise attenuation. A sensing microphone inputs the original noise to a processor, which analyzes the noise and generates the inverse signal, which is applied to a loudspeaker, as shown in Fig. 19-19. An error microphone can be used to sense residual noise; the error microphone's output is applied to a feedback control system to adaptively optimize the cancellation.

Active noise control is most effective for continuous, low-frequency noise as opposed to impulsive noise. It is applicable to very controlled spaces such as ventilation ducts to reduce HVAC noise; however, because of cost, its use is not widespread.

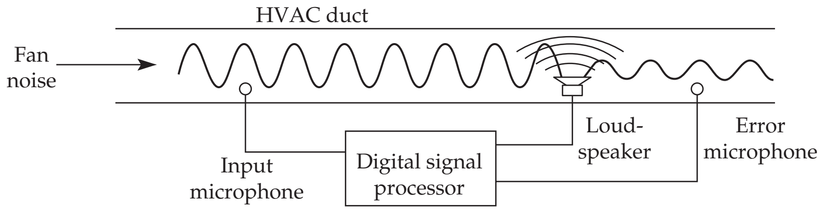


Figure 19-19 Concept of active noise cancellation inside a duct, using digital signal processing.

Ventilation noise or tones can be attenuated by 30 to 40 dB. Active noise control is widely used in headphones. Active noise control can be applied to single points in the immediate vicinity of an industrial area of heavy noise, for example, at a factory workstation. The prospect of cost-effective active noise control in sound-sensitive areas such as home listening rooms, recording studios, or control rooms remains remote.

Key Points

- The goal of the acoustical design of a HVAC system is to reduce noise to an acceptable level. Most HVAC systems do not meet the noise requirements required for acoustically sensitive spaces.
- Quiet HVAC air can be obtained by considering five factors: careful specification and installation of noise sources such as motors, fans, and grilles; use of sound absorbers in machine rooms and in air ducts to reduce noise; minimization of air turbulence noise in ducts and at duct openings; minimization of acoustical crosstalk between rooms; and reduction of fan speed and air velocity.
- NCB contours can be used to establish the maximum allowable constant noise-pressure level in each octave band. In some cases, room criteria (RC) values are specified and are used similarly to NCB contours.
- Some noise energy is concentrated in high frequencies, some in low frequencies. There must be an overall balance in the application of silencers so that the resulting noise follows the proper NCB or RC contour. Otherwise, overdesign can result.
- In noise control, generally the most efficient solution is to minimize the noise at the source. Machinery and fans are chief contributors to HVAC noise often requiring source noise control.
- In a new room design, HVAC equipment should not be placed near an acoustically sensitive room. Also, air-handling equipment in a machinery room must be isolated to mitigate structureborne vibration.
- Transmissibility (*T*) is the force transmitted to the building structure divided by the force applied by the sound source; it ranges from 0 to 1. The lower the transmissibility, the better.
- A vibration source such as a motor, mounted on isolation pads or similar means, can be considered as a mass on a spring. It will have a natural frequency of vibration f_n, where it will freely vibrate.

- The higher the velocity of airflow, the higher the level of turbulence noise. The quantity of air required, velocity of airflow, and duct size must be determined in a way to minimize noise. A maximum air velocity of 500 ft/min is sometimes suggested for critical spaces.
- The most effective way to control airflow noise is to increase duct size which decreases airflow velocity. Alternatively, the economy of smaller ducts may be sufficient to pay for silencers to lower the higher noise of higher-velocity flow to acceptable levels.
- Right-angle bends and dampers create noise due to air turbulence. Such fittings should be located from 5 to 10 diameters upstream from the outlet to allow turbulence to abate.
- Noise and turbulence inside a duct cause the duct walls to vibrate and radiate noise into surrounding areas. Rectangular ducts are inferior to round ducts. Such noise increases with air velocity and duct size but can be controlled with the treatment of isolating material.
- Most acoustical ceilings are not good sound barriers; hence in an acoustically sensitive area, the space above a drop ceiling should not be used for high-velocity terminal units.
- Plenums are an effective way to provide noise attenuation throughout the audible spectrum. They are especially effective at the fan output.
- Proprietary attenuators, reactive silencers, tuned silencers, and active noise control systems all offer specialized means to reduce HVAC noise in rooms.

CHAPTER 20

Acoustics of Listening Rooms and Home Theaters

This chapter considers the area of a home designated for the playback of recorded music. This may be a music listening room, or an audio/video home theater. Some homes have a multipurpose space, typically a living room that also serves as a listening room. Other homes have a room that is dedicated to the enjoyment of music and films. The playback equipment can range from a simple stereo system to an elaborate home cinema. In any case, the room acoustics will play an important role in the sound quality of the audio playback signal.

The acoustical path between the loudspeaker diaphragms and the ear is as important as the quality of audio hardware such as amplifiers and loudspeakers. However, this acoustical path is less tangible and less available for correction and improvement than the hardware. A high-quality listening room or home theater must have excellent sound quality, but that is impossible without sophisticated acoustical treatment of the space housing the loudspeakers and the listener.

For both the critical audiophile and the acoustician, the design of a listening room is as much of a challenge as the design of a control room in a professional recording studio. All the major acoustical problems involved in the design of a listening room or home theater are also present in other small audio rooms. The discussion of the acoustics of listening rooms in this chapter is therefore considered as an introduction to the acoustics of other types of small audio spaces in the following chapters. However, in this chapter, in particular, we will consider the essential problem of how loudspeaker playback is influenced by room acoustics.

Playback Criteria

The acoustics of a space is a vital part of both the recording and reproducing process. In every acoustical event, there is a sound source and a receiving device with an acoustical link between the two. The source/receiver may be a musical instrument/microphone or a loudspeaker/listener. Audio recordings contain the imprint of the acoustics of the recording environment. For example, if the sound source is a symphony orchestra and the recording is made in a concert hall, the reverberation of the hall is very much a part of the orchestral sound. If the reverberation time of the hall is 2 sec, a 2-sec reverberant tail is very evident on every impulsive sound and sudden cessation of music, and it affects the fullness of all the music. In playing this recording in a home listening room, what room characteristics will best complement this type of music?

Another recording may be of popular music. This music was probably recorded in a relatively dead studio with overdubbing. Rhythm sections playing in this dead studio and well separated acoustically are laid down on separate tracks. During subsequent sessions the other instruments and vocals are recorded on additional tracks. Finally, all tracks are combined at appropriate levels in a stereo or multichannel mix down. In the mix down, many effects, including artificial reverberation, are added. What listening room characteristics are best for playback of this recording?

If the taste of the audiophile is highly specialized, the listening-room characteristics can be designed for relatively optimum results for one type of music. If the listener's taste is more universal, the acoustical treatment of the listening room may require compromises to make it suitable for different types of music.

The acoustics of a home theater are designed for optimal playback of motion-picture soundtracks. Although some dialogue in a movie might originate from a shooting location, the great majority of the dialogue is recorded in a studio environment not unlike that used for recording music. In addition, depending on the type of music, the music score might be recorded in a large hall or in a studio. Finally, the many sound effects are added in postproduction and mixed with the other sound elements to create the finished multichannel soundtrack. One particular design criteria of a home theater that differs somewhat from that of a music listening room is the importance of voice intelligibility. If dialogue is not readily intelligible, the movie experience is ruined. This necessitates special care in the design of the acoustics of a home theater.

Perhaps the best guiding principle for listening room or home theater design is one of neutrality. A good room, one suitable for auditioning many kinds of music or movies, should have a neutral sound that adds little of the room's own sonic signature to the reproduced sound. Conversely it should not obscure or degrade the sound that is being replayed.

The dynamic range of reproduced music in a listening room is determined by the softest sounds played, and the loudest. The loudest level depends primarily on the maximum amplifier power, and the power-handling ability and efficiency of the loudspeakers. The softest levels of the dynamic range are limited by environmental or electrical noise. For example, ambient household noise may determine the lower dynamic-range limit. The usable range between these two extremes is usually far less than the range of an orchestra in a concert hall. If peak instantaneous sound levels are considered when determining dynamic range requirements, even greater ranges are required.

Fiedler played back musical performances of high peak levels in a quiet environment to show that a dynamic range of up to 118 dB is necessary for subjectively noise-free reproduction of music. The results are summarized in Fig. 20-1. He considered the peak instantaneous sound level of various sources, as shown at the top of this figure, and the just-audible threshold for white noise added to the program source when the listener is in a normal listening situation, as shown at the bottom of this figure. This suggests that for realistic playback, a listening room requires extensive sound isolation to ensure a low ambient noise level, and a high-power playback system capable of generating high sound-pressure levels with low distortion.

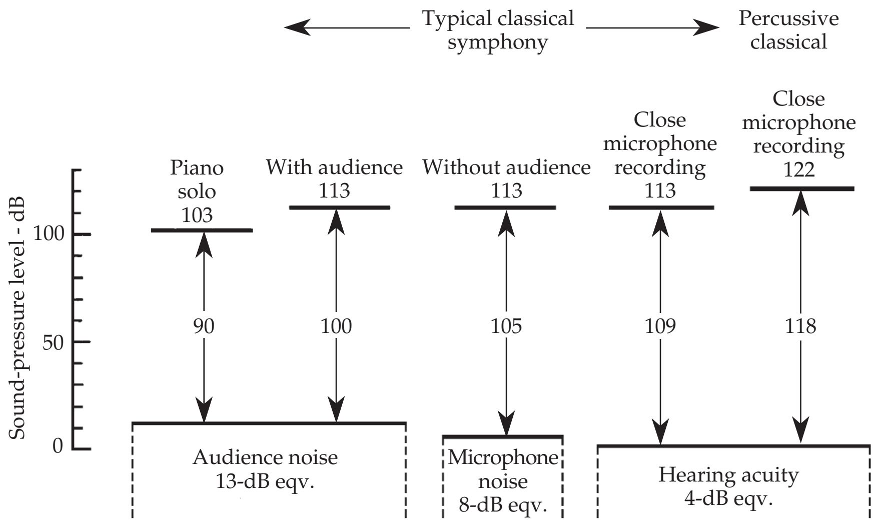


Figure 20-1 A dynamic range of up to 118 dB may be necessary for subjectively noise-free reproduction of music. (*Fielder*.)

Planning the Playback Room

Ideally, a listening room or home theater should be placed in a dedicated room. As with most acoustical spaces, larger room volumes are preferred. Moreover, to truly replicate the acoustical experience of being in a concert hall or a motion-picture theater, a very large room volume is required. A somewhat smaller volume is adequate for modest audio playback. The room can be rectangular; however, care must be taken to eliminate potential flutter echoes from parallel boundary surfaces. In other words, with proper care, the walls do not need to be splayed and the ceiling does not have to be sloping.

When first-class acoustics is the goal, whether the listening room or home-theater space will be new construction or an adaptation of an existing space, room proportions must be considered. The distribution of room resonant frequencies is the logical first consideration. Small rooms are acoustically more difficult than larger ones because modal resonant frequencies too sparsely populate the low-frequency region. Special attention is required to control them and to obtain a satisfactory low-frequency room response. Thus the room selected for the project should be analyzed in terms of modal response, and the measures needed to control room modes should be fundamental to the acoustical design. These issues are discussed in Chap. 13.

In an existing home, most of the typical noise sources are unavoidable. The central HVAC system is probably of the small-duct, high-velocity air type, and will generate air-turbulence noise. Water pipes are probably fastened solidly to the structure so plumbing noise will unfortunately be efficiently radiated into rooms. The noise of daily living is cheerfully generated by the home's occupants. In most cases, very little can be done to reduce background noise in an existing home. However, installation of a solid-core door with gasketed seals is often helpful.

In some cases, it may be possible to choose a room at some distance from the loudest levels of household clatter. This will reduce noise intrusion into the playback room, and conversely reduce noise intrusion to the rest of the house. In a new home design, great care can be taken to isolate the playback room from the rest of the house; moreover, a low-noise HVAC system could be installed. The many aspects of noise control are discussed in Chaps. 16 through 19. It should be noted that noise control can entail relatively high costs.

Acoustical Treatment of Playback Rooms

When treating a playback room, one must consider the time of arrival of reflections; this is the case in any room. In a playback room, we have the advantage of knowing where the sound sources (the loudspeakers) are located. Because of room reflections, an original wavefront will arrive repeatedly and delayed in time because of the varying path lengths of the reflections. This causes irregularities in the signal's frequency response. To overcome this, the early reflections should be absorbed or diffused at the reflecting surfaces. It is also important to control the low-frequency modal resonances of the room. To do this, bass traps or similar absorbers should be placed in the corners of the room. Once the effect of placing low-frequency absorbers in two corners is auditioned, the desirability of adding additional bass traps in the other two corners can be evaluated. Either proprietary or custom-built bass traps can be used.

A playback room should be neither too reverberant nor anechoic. The typical living room reverberation time of about 0.5 sec should be satisfactory for most listeners. If the room is too reverberant, fabric-covered glass-fiber panels could be introduced to achieve the best balance. If there is too much overstuffed furniture, the room might tend toward anechoic; some absorptive furniture may have to be removed. In many playback rooms, diffusers should be mounted on the rear wall. Also, except in overly reverberant rooms, because diffusion does not decrease desirable signal energy, diffusers are preferable to absorbers for controlling early reflections.

Peculiarities of Small-Room Acoustics

The audible spectrum encompasses fully 10 octaves; the extent of this range makes the acoustical analysis of any room problematic. This is even more true for small rooms, which perform quite differently from large rooms. The reason is apparent when room size is considered in terms of the wavelength of sound. The 20-Hz to 20-kHz audio band covers sound wavelengths from 56.5 to 0.0565 ft (0.68 in). Below about 300 Hz (wavelength 3.8 ft), the average listening room must be considered as a resonant cavity. It is not the room structure that resonates; it is the air confined within the room. As frequency increases above 300 Hz, the wavelengths become progressively smaller with the result that sound may be considered as rays.

Reflections of sound from the enclosing surfaces dictate the room response in the low-frequency regions. At low frequencies, reflections result in standing waves and the room acts as a chamber resonating at many different frequencies in three dimensions. These standing waves dominate low-frequency response in a small room. Reflections of sound from the room surfaces are also important at midband and high audible frequencies. There are no problems from cavity resonances at these frequencies, but specular reflections play a major role.

Room Size and Proportion

Acoustical problems are inevitable when sound is recorded or reproduced in small spaces. For example, Gilford states that studio volumes less than approximately 1,500 ft^3 are so prone to acoustical distortion as to make the room impractical. Rooms smaller than this produce sparse modal frequencies with probable wide spacings between modes, which are the source of audible distortions. All other things being equal, it is generally true that the larger the room, the better the sound quality.

As we observed in Chap. 13, it is possible to select room proportions that yield the most favorable distribution of room modes. With new construction, it is strongly advised to use these proportions as a guide, confirming any dimensions by calculation and study of the spacing of axial-mode frequencies.

In most cases, in a home listening room, the room shape and size are already fixed. Nevertheless, the existing room dimensions should be used for axial mode calculations. A study of these modal frequencies will reveal the presence of coincidences (two or more modes at the same frequency) or isolated modes spaced 25 Hz or more from neighbors. Such faults pinpoint frequencies at which acoustical distortion may occur, and steps may be taken to remedy the problem.

There is no such thing as perfect room proportions. It is easy to place undue emphasis on a mechanical factor such as this. In reality, one should be informed on the subject of room resonances and be aware of the consequences. In striving to attain high sound quality, reducing acoustical deficiencies by attention to room modes is only one of many factors to consider.

Reverberation Time

Reverberation time is one of several determinants of acoustical quality of small rooms. The amount of overall absorbance in a listening room establishes the general listening conditions. If the room is excessively absorptive or excessively reverberant, music quality will deteriorate and most listeners will grow fatigued of the unnatural sound field. There is no optimal value for reverberation time in a listening room. A simple conversation test is often all that is needed to ascertain that reverberation time is suitable. If a talker at the location of the center-channel loudspeaker can comfortably be heard and understood by a listener at the listening position, and speech quality sounds natural, then the reverberation time is approximately correct—the room is neither too live nor too dead.

The Sabine equation for reverberation time makes it possible to estimate the amount of absorbing material required for a reasonable reverberant condition. It is expedient to assume a reasonable reverberation time, say about 0.5 sec, for the purposes of these calculations. From this, the total number of sabins of absorption can be estimated, which would result in suitable listening conditions. In many home listening rooms, the structure and the furnishings often supply most of the absorbance required. However, in some cases, absorbing wall panels or carpets may be needed. Careful listening tests must determine the degree of room ambience most suitable for a favorite type of music.

Low-Frequency Considerations

Consider the bare room of Fig. 20-2 as the starting point in analyzing low-frequency response. The room is 21.5-ft long and 16.5-ft wide, with a 10-ft ceiling. These dimensions fix the mode resonances and their multiples. Following discussions in Chap. 13,

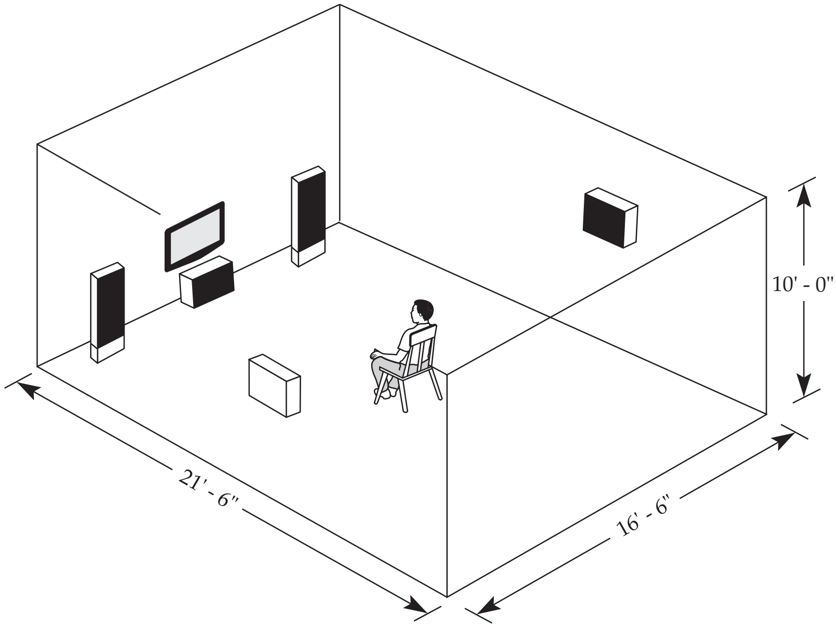


Figure 20-2 Dimensions and layout of a listening room assumed for analysis of axial modes.

axial mode effects will be examined; tangential and oblique modes will be neglected. Axial mode frequencies to 300 Hz are calculated for the length, width, and height dimensions, and tabulated in Table 20-1. These axial modal frequencies are arranged in ascending order of frequency, irrespective of the dimensional source (length, width, or height). Spacings between adjacent modes are entered in the right-hand column. No coincident frequencies are present in this room; only a single pair is as close as 1.7 Hz. This is not surprising because the dimensional ratios 1:1.65:2.15 are well within the Bolt area.

These calculated axial modal frequencies are now applied to the listening room. This is done graphically in Fig. 20-3, following a style used by Toole. The positions of the nulls locate null lines, which are drawn through the listening room. Lines representing the length-mode nulls are drawn through both the elevation view and the plan view because these nulls actually form a null plane that extends from floor to ceiling. The position of the listener's seat can be moved to avoid these particular nulls at 26, 53, and 79 Hz, but observe that there are eight more nulls below 300 Hz.

The three lowest axial-mode nulls associated with the height of the room (56, 113, and 170 Hz) are sketched on the elevation view of Fig. 20-3. These nulls are horizontal planes at various heights. The head of the listener in the elevation view lies between two nulls and at the peak of the 79-Hz resonance.

The three axial modes of lowest frequency are sketched on the plan view. The nulls in this case are vertical planes extending from floor to ceiling. The listener, if situated in dead center of the room, intercepts the nulls of every odd axial mode. Therefore, this would be a poor choice for the primary listening position.

The resonance nulls have been sketched because their location is definite, but between any two nulls of a given axial mode, a peak exists. Although nulls are capable

Room Dimensions = 21.5 × 16.5 × 10.0 ft					
	Axial Mode Resonances			Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
	Length $L = 21.5$ ft $f_1 = 565/L$ (Hz)	Width $W = 16.5$ ft $f_1 = 565/W$ (Hz)	Height $H = 10.0$ ft $f_1 = 565/H$ (Hz)		
f_1	26.3	34.2	56.5	26.3	7.9
f_2	52.6	68.5	113.0	34.2	18.4
f_3	78.8	102.7	169.5	52.6	3.9
f_4	105.1	137.0	226.0	56.5	12.0
f_5	131.4	171.2	282.5	68.5	10.3
f_6	157.7	205.5	339.0	78.8	23.9
f_7	184.0	239.7		102.7	2.4
f_8	210.2	273.9		105.1	7.9
f_9	236.5	308.2		113.0	18.4
f_{10}	262.8			131.4	5.6
f_{11}	289.1			137.0	20.7
f_{12}	315.4			157.7	11.8
				169.5	1.7
				171.2	12.8
				184.0	21.5
				205.5	4.7
				210.2	15.8
				226.0	10.5
				236.5	3.2
				239.7	23.1
				262.8	11.1
				273.9	8.6
				282.5	6.6
				289.1	19.1
				308.2	

Mean axial mode spacing: 11.7 Hz.
Standard deviation: 6.9 Hz.

Table 20-1 Axial Modes of the Example Listening Room

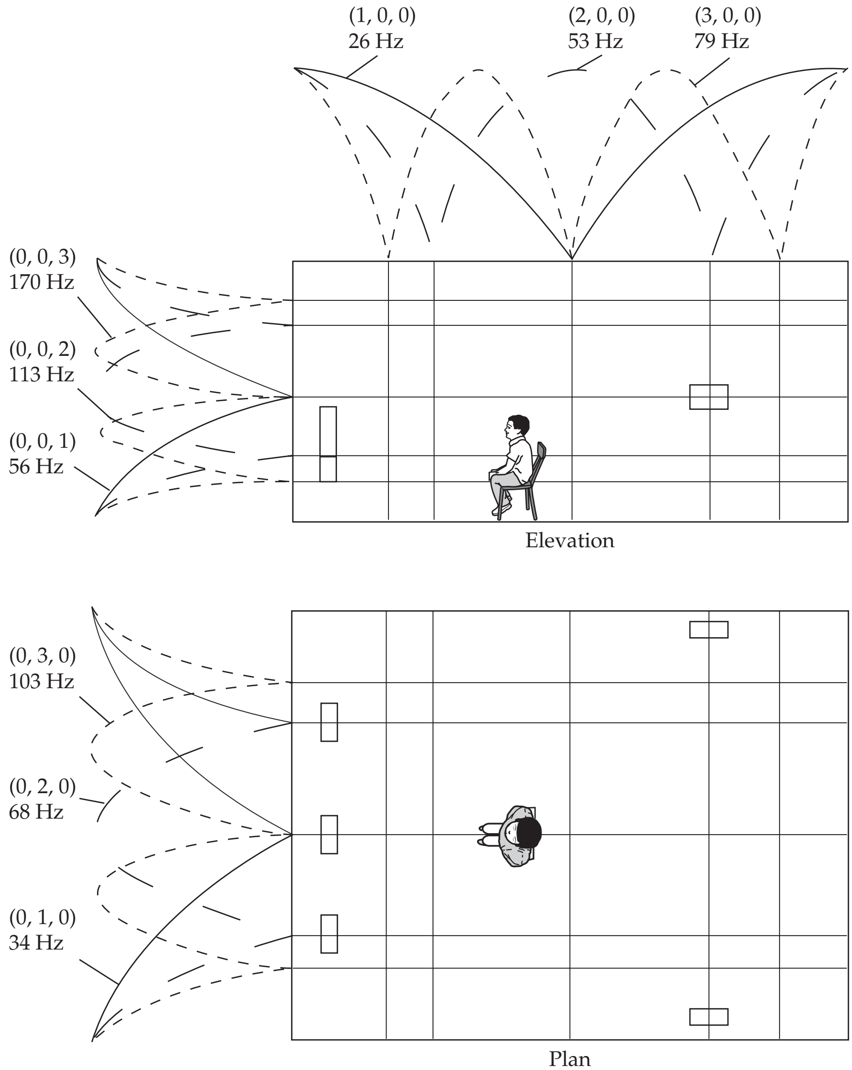


Figure 20-3 Plan and elevation views of the listening room of Fig. 20-2 showing the low-frequency modal pressure distributions of the first three axial modes of Table 20-1.

of removing a sizeable chunk of spectrum, the low-frequency acoustics of the room is dominated by the wide, relatively flat peaks.

The complexity of the modal structure of listening room acoustics is apparent. Only the first three axial modes of length, width, and height have been shown. All of the axial modal frequencies listed in Table 20-1 have an important part to play in the low-frequency acoustics of the space. These axial modes exist only when they are excited by the low-frequency sound of the signal being reproduced in the room. The spectrum of the music is continually shifting; therefore, the excitation of the modes is also continually shifting. For example, the length axial mode at 105.1 Hz (see Table 20-1) is excited only by spectral energy in the music around that frequency.

Modal Anomalies

Momentary deviations from flatness of the room response are an obvious source of low-frequency anomalies. Deviations are caused by concentrations of modes or large spacings between modes. For example, transient bursts of music energy result in unequal, forced excitation of the modes. As the transient excitation is removed, each mode decays at its natural (and often different) frequency. Beats can occur between adjacent decaying modes. Energy at new and different frequencies is injected, which creates audible distortion of the signal.

Control of Modal Resonances

The low-frequency sound field at the listener's ears is made up of the complex vectorial sum of all axial, tangential, and oblique modes at that particular spot in the room. The loudspeakers energize the modal resonances prevailing at their locations. The modes that have nulls at a loudspeaker location cannot be energized, but those having partial or full maxima at this location will be energized proportionally. The interaction of low-frequency resonances in the listening room at the loudspeaker and listening positions is complex and transient, but they can be understood if broken down into the contributions of individual modes.

The position of both the sound sources and the listener should avoid nulls in the modal distribution, axial modes in particular. Loudspeaker positions should be considered tentative, moving them slightly if necessary to improve sound quality. The same is true for listening position. Shifts in position, including height, will affect frequency response and sound quality at the listening position.

Bass Traps for Playback Rooms

In many cases, it is not practical to acoustically treat modes individually. A general low-frequency treatment can more efficiently control "room boom" and other resonance anomalies. In fact, when choosing a listening room, consider that a room of wood-frame and drywall construction could inherently have sufficient structural low-frequency absorption to provide all the general modal control necessary. In contrast, a room constructed of concrete blocks may lack low-frequency absorption and would require additional treatment. Bass traps can provide low-frequency absorption and are particularly efficient when placed in the corners of a room. This is because modal analysis shows that bass energy is concentrated there.

In addition to their beneficial effect on the room response, low-frequency absorbers located in the corners of a room near the loudspeakers can improve stereo or surround imaging that would otherwise be degraded by room resonances. Figure 20-4 suggests four ways that such corner absorption can be obtained. The first unit, shown in Fig. 20-4A, is a Helmholtz resonator trap, built into the corner of the room, floor to ceiling. This could employ either a perforated face or spaced slats. Some design frequency must be assumed, perhaps 100 Hz, and an average depth of the triangular shape must be calculated. The varying trap depth will yield a more broadband absorption. A diaphragmatic absorber could also be used.

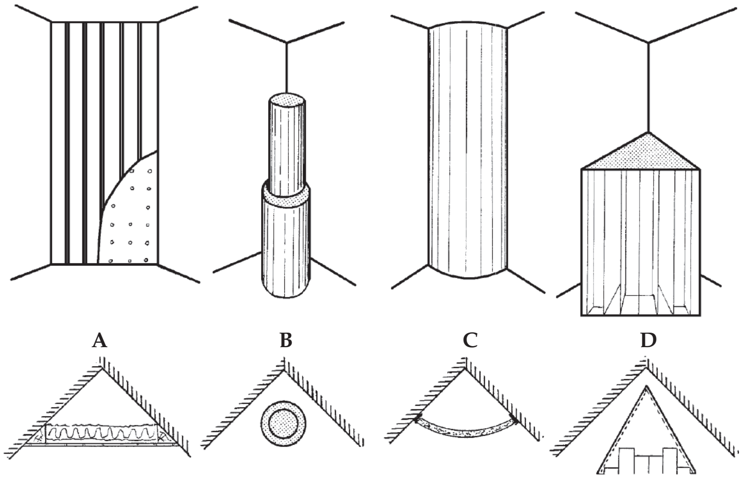


Figure 20-4 Four ways to provide low-frequency absorption. Modules are placed in the corners of a listening room. (A) Built-in corner resonator. (B) Stacked cylindrical resonators. (C) Polycylindrical resonator. (D) Resonator with diffuser.

The cylindrical absorber of Fig. 20-4B provides the necessary absorption by stacking a smaller diameter unit on top of a larger diameter unit, resulting in a total height of perhaps 6 ft. If necessary, for very low-frequency absorption, an even larger-diameter base could be used. These traps are fibrous cylinders with a wire mesh exoskeleton, together acting as resonant cavities. Half of the periphery is a reflective, limp-mass sheet. Low-frequency energy (below 400 Hz) readily penetrates this sheet, whereas high-frequency energy is reflected from it. Used in a corner, the cylinder would be rotated so that the reflective side faces the room. The cylinder thus contributes diffusion to the room as well as to deep-bass absorption.

The corner treatment of Fig. 20-4C uses tracks of 1- × 1/2-in J-metal installed in the corner to hold the edges of a panel. The sheet is bent and then snapped into place. The airspace behind the acoustical panel provides low-frequency absorption. A curved membrane reflector strip within the panel provides wide-angle reflection above 500 Hz.

Another possible corner treatment is shown in Fig. 20-4D. The nominal dimensions of the module are height 4 ft, faces 2 ft. The module has two absorptive sides and one quadratic residue diffuser side. With the absorptive sides facing the corner for modal control, the diffusive side faces the room. The diffuser face diffuses the sound energy falling on it, and also reduces the amplitude of the energy returned to the room.

With any of the devices of Fig. 20-4 in the corners of the listening room near the loudspeakers, the low-frequency room response should be reasonably better than before treatment. In addition, chances are good that sufficient modal control is introduced to minimize stereo image problems resulting from room resonances. Room corners without loudspeakers could be treated similarly if more modal control is required.

Mid/High-Frequency Considerations

The propagation of sound of shorter wavelengths, above about 300 Hz, can be considered as rays that undergo specular reflection. Figure 20-5 shows a listening room and listener for the consideration of the mid- to high-frequency reflections of sound from loudspeakers. Sound from the front right loudspeaker is studied in detail as characteristic of the symmetrical room.

The first sound to reach the listener's ears is the direct sound *D*, traveling the shortest distance. Reflection *F* from the floor arrives next. Reflections from the ceiling *C*, the near side wall *S*1, and the far side wall *S*2 arrive later. Another early reflection *E* from the front wall is from the diffraction of sound from the edges of the loudspeaker cabinet.

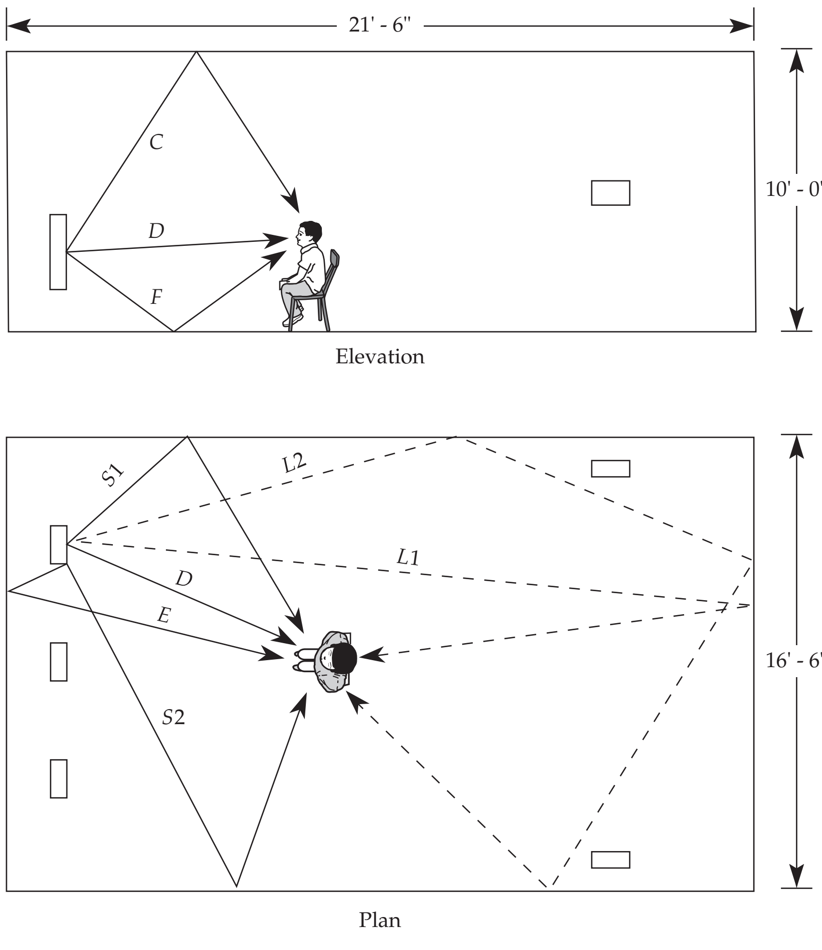


Figure 20-5 Plan and elevation views of a listening room showing the direct sound *D* and early reflections from the floor *F*, the ceiling *C*, the side walls *S*1 and *S*2, and diffraction edges of the loudspeaker cabinet *E*. The later reflections *L*1 and *L*2 are the beginning of the reverberant component.

The direct ray *D* carries important information concerning the signal being radiated. If it is obscured by strong early reflections, for example, the perception of imaging and localization may be diminished.

These rays constitute the direct sound and early reflections, as contrasted to reflections from the rear surfaces of the room and general reverberation, arriving much later. For example, late reflections *L*1 and *L*2 are representative of reflections that help to create the reverberant component.

Olive and Toole investigated the effect of a simulated lateral reflection in an anechoic environment with speech as the test signal. Their work is partially summarized in Fig. 20-6 (a repeat of Fig. 6-11 for convenience). The variables are reflection level and reflection delay. When the reflection level is 0 dB, the reflection is the same level as the direct signal. When the reflection level is –10 dB, it is 10 dB below the direct signal. Curve A is the threshold of audibility of the reflection. Curve B is the image-shift threshold. Curve C is the threshold at which the reflection is heard as a discrete echo.

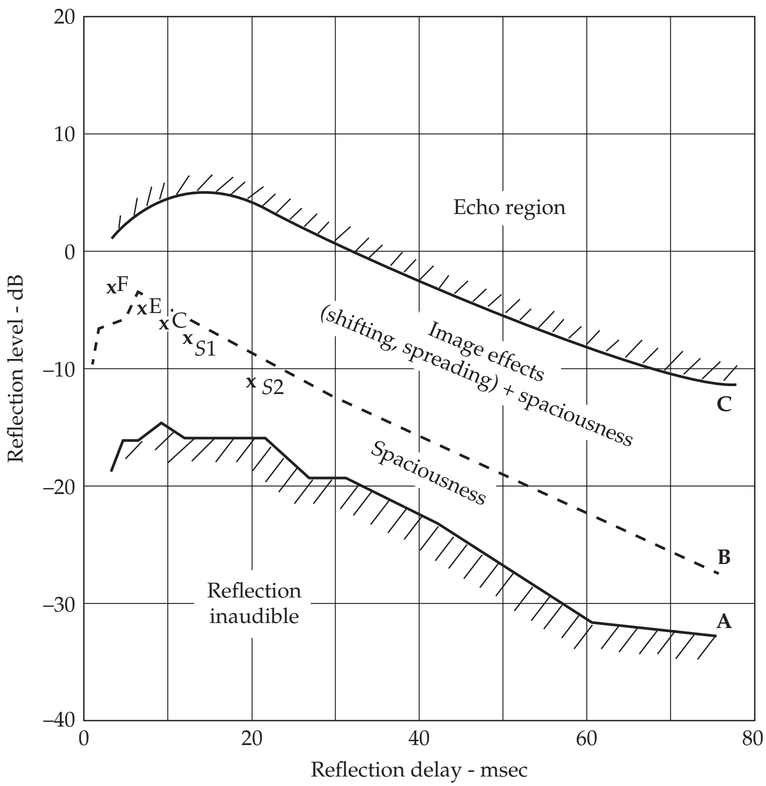


Figure 20-6 Results of investigations into the effect of a simulated lateral reflection in an anechoic environment with speech as the test signal. (A) The absolute threshold of detection of the reflection. (B) The image-shift threshold. (C) The threshold at which the reflection is heard as a discrete echo. In addition, the early room reflections (*F*, *E*, *C*, *S*1, *S*2) calculated in Table 20-2 are plotted. (*Composite of results: Curves A and B from Olive and Toole, Curve C from Meyer and Lochner.*)

Sound Paths	Path Length (ft)	Reflected Path – Direct Path (ft)	Reflection Level* (dB)	Delay† (msec)
D (Direct)	8.0	—	—	—
F (Floor)	10.5	2.5	–2.4	2.2
E (Diffraction)	10.5	2.5	–2.4	2.2
C (Ceiling)	16.0	8.0	–6.0	7.1
*S*1 (Side wall)	14.0	6.0	–4.9	5.3
*S*2 (Side wall)	21.0	13.0	–8.4	11.5
*L*1 (Rear wall)	30.6	22.6	–11.7	20.0
*L*2 (Rear wall)	44.3	36.3	–14.9	32.1

*Reflection level $= 20\log\dfrac{\text{Direct path}}{\text{Reflected path}}$.

†Reflection delay $= \dfrac{(\text{Reflected path}) - (\text{Direct path})}{1{,}130}$.

Table 20-2 Reflection Amplitudes and Delays of the Room Shown in Fig. 20-5

For any delay, the reflection is not heard for reflection levels below curve A. For the first 20 msec, this threshold is essentially constant. At greater delays, progressively lower reflection levels are required for a just-audible reflection. For a home listening room or other small room, delays in the 0- to 20-msec range are significant. In this range, the reflection audibility threshold varies little with delay.

It is instructive to compare the levels and delays of the early reflections of Fig. 20-5 with those of Fig. 20-6. Table 20-2 lists estimates of the level and delay of each of the early reflections (*F*, *E*, *C*, *S*1, *S*2) identified in Fig. 20-5 (assuming perfect reflection and inverse square propagation). These reflections, plotted in Fig. 20-6, all fall within the audible region between the reflection threshold and the echo threshold. The direct signal is immediately followed by competing early reflections of various levels and delays, producing comb-filter distortion.

The level of the competing reflections should be reduced relative to the direct signal, except for the lateral reflections. As noted, Fig. 20-6 is a study of a direct signal and a single lateral reflection. Allowing a single lateral reflection of adjustable level would permit control of the spaciousness and image effects of the room. Therefore, when designing the acoustical conditions of this listening room, the early reflections should be eliminated, except those lateral reflections from the side walls; these subsequently will be adjusted for optimum sound quality.

Identification and Treatment of Reflection Points

One method for reducing the levels of early reflections is to treat the front portion of the room with sound absorbing material. However, this may also absorb the beneficial lateral reflections and may make the room too dead. Instead, the principle recommended here is to add a minimum of absorbing material to treat only the specific surfaces responsible for unwanted reflections.

Locating these reflection points is easy with the help of an assistant equipped with a mirror. For example, with the listener seated at the listening position, the assistant moves a mirror on the floor until the observer can see the tweeter of the front right loudspeaker reflected in it. This is the point where the floor reflection strikes the floor. This point is marked and the procedure repeated for the tweeter of the front left loudspeaker, and the second floor reflection spot is marked. A small throw rug covering these two marks should reduce the floor reflections to inaudibility. The same procedure is carried out for locating the reflection points for the left- and right-side walls (however, see discussion later) and the ceiling reflections. Each of these points should also be covered with enough absorbing material (such as light cloth, heavy cloth, velour, acoustical tile, glass-fiber panel) to ensure sufficient high-frequency absorption at the reflection points.

The point of reflection for the sound energy diffracted from the edges of the loudspeaker cabinet is more difficult to locate. Installing an absorber on the wall behind the loudspeakers should mitigate reflections due to cabinet diffraction. Potential reflections from the tweeters in all the loudspeakers in the playback system are similarly analyzed. When all of the reflection points are covered, as shown in Fig. 20-7, imaging will probably be much clearer and more precise now that the early reflections have been reduced.

Lateral Reflections and Control of Spaciousness

The lateral reflections from side walls can be essentially eliminated by placing absorbing material on the walls. A critical listening test should be performed with the side-wall

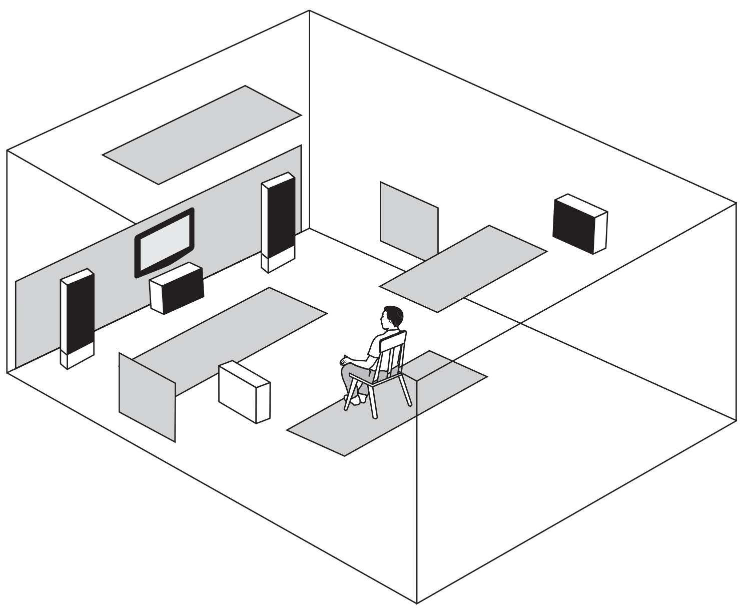


Figure 20-7 The room of Fig. 20-2 with minimum sound absorbing treatment to reduce the level of the early reflections from room surfaces. The lateral reflectivity of the side-wall absorbers may be adjusted to control spaciousness and imaging in the listening room. Additional absorbing material may be needed to adjust the average reverberant character of the room for best listening.

absorbers temporarily removed, but with the floor, ceiling, and diffraction absorbers still in place. In this way, the recommendations of Fig. 20-7 can be tested. Does the strong lateral reflection give the desired amount of spaciousness, or does it cause unwanted image spreading? The magnitude of the lateral reflections can be adjusted by using sound absorbers of varying absorbance (such as light cloth, heavy cloth, velour, acoustical tile, glass-fiber panel) on the side-wall reflection points. For example, the lateral reflections can be reduced somewhat by hanging a light cloth instead of velour.

Techniques such as these provide the ability to adjust the lateral reflections to achieve the desired spaciousness and stereo and surround imaging characteristics to suit the individual listener and to optimize conditions for different types of music or cinema. Further discussion of acoustical design techniques for small rooms is continued in the following chapters.

Loudspeaker Placement

Optimal loudspeaker placement depends greatly on the room geometry. For example, a rectangular room will require different placement from a room with an asymmetrical floor plan. Exact placement also depends on room acoustics, loudspeaker type, and listener preference. Very generally, stereo loudspeakers should be placed in front of the "sweet spot" listening position so there is a ±30° angle between them. A surround-sound system may follow the guidelines provided in the *ITU-R Recommendation BS.775-3, Multichannel Stereophonic Sound System With and Without Accompanying Picture*, (*2012*). In a 5.1-channel system, it recommends placing the center-channel loudspeaker at 0°; front left and right loudspeakers at ±30°; surround loudspeakers at ±110° to ±120°, all relative to the listening position. This is shown in Fig. 20-8. Some listeners prefer the surround loudspeakers placed slightly forward or backward. These positions may vary, for example, with the screen size, or whether the surround loudspeakers are front-firing loudspeakers or bidirectional dipole loudspeakers. In 7.1-channel systems, four surround loudspeakers may be placed at ±60° and ±150°.

The front loudspeakers (particularly the tweeters) should be placed at approximately the same height. The front loudspeakers (particularly the tweeters) should also be placed at approximately ear level when seated and aimed at the listening position. Surround loudspeakers are often placed 2 or 3 ft above ear height when seated.

It is important to remember that sound from a loudspeaker interacts strongly with any reflecting surfaces near it. For example, placing a loudspeaker symmetrically in a tri-corner adds a 9-dB low-frequency peak to its response. When the loudspeaker is placed on the floor close to a wall, the low-frequency peak is 6 dB. When placed close to a single wall, the peak is 3 dB. If a system needs a low-frequency boost, these loudspeaker placements will provide it. If not desired, the peak can be minimized by placing the loudspeaker at different distances from each of the three reflecting surfaces.

True dipole loudspeakers, such as the electrostatic type with a tall, vibrating membrane, have a strong rear radiation component that must be controlled to prevent another source of early reflections. The more common dynamic loudspeakers radiate rearward weakly, chiefly by the diffraction from corners of the cabinet.

Because of low-frequency standing waves, subwoofer placement very much depends on room acoustics and the listening position in the room. A subwoofer placed in a corner will excite the maximum number of room modes, and provide the most bass. Placement along one wall, or away from a wall, will excite relatively fewer modes.

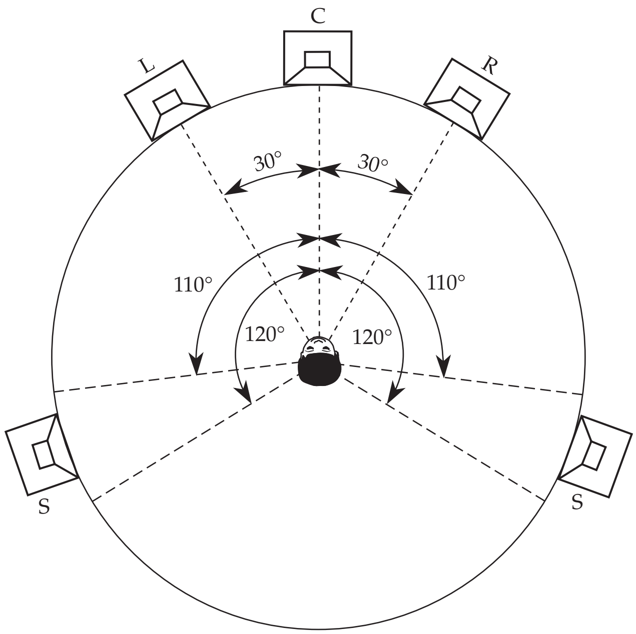


Figure 20-8 For 5.1-channel playback, *ITU-R BS.775-3* recommends placing the center-channel loudspeaker at 0°; front left and right loudspeakers at ±30°; surround loudspeakers at ±110° to ±120°, all relative to the listening position.

Conversely, a listening position, depending on location in the room, will be receptive to greater or fewer bass modes. In some cases, corners will already be occupied by bass-trap absorbers.

One technique for finding optimal subwoofer placement makes clever use of reciprocity. Place the subwoofer in the listening chair, at seated ear level if possible, and play music and movies. Then you move around the room (on hands and knees, placing ear level where the subwoofer will be) and listen for the location that provides the best bass response for a variety of music and movies. Then, place the subwoofer at that location. In some cases, two or more subwoofers in different locations may be needed to provide satisfactory bass response, especially over a larger listening area. Be sure to identify and eliminate any buzzes and rattles caused by the subwoofer or satellite loudspeakers.

Listening Room Plan

Three listening-room designs are presented. The designs describe a single room with progressive additions to optimize the acoustical performance of the room, at progressively higher budget levels. Recommendations for an absorptive panel, bass trap, and diffusive panel are introduced; either scratch-built or commercially manufactured units can be used. The room dimension ratio in this particular example is: 1.0:1.4:1.9.

A simple home listening room is shown in Fig. 20-9; this basic design demonstrates how treatment can be provided on a budget. The side-wall reflections are treated on a

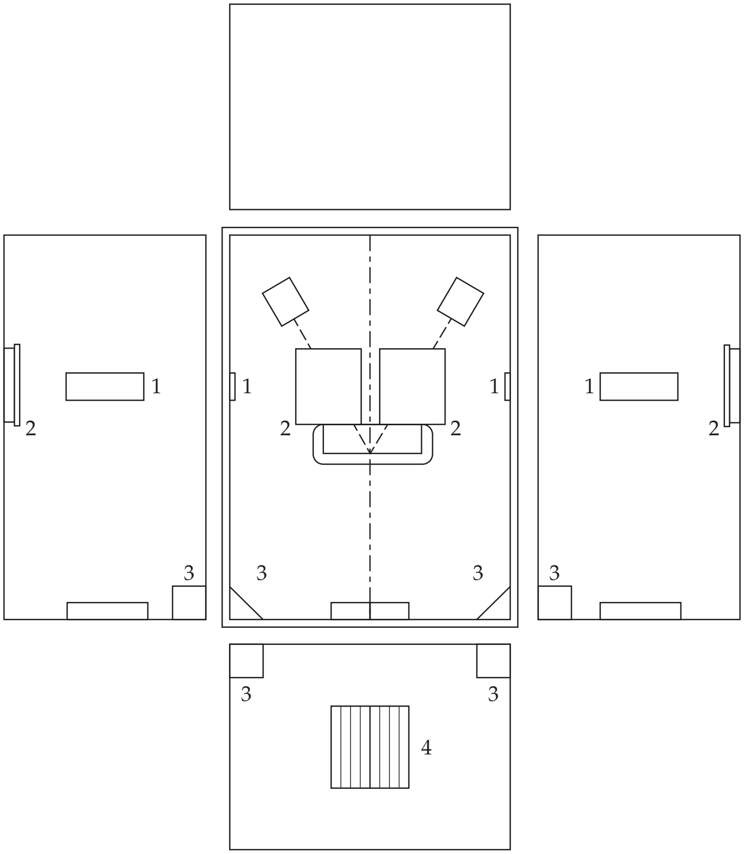


Figure 20-9 Home listening room plan with basic acoustical treatment.

minimum basis with a single absorption panel (1) placed on each side wall to provide broadband absorption and reduce reflected energy. The panel can comprise a glass-fiber sheet surrounded by a wood frame and covered by open-weave fabric. Alternatively, an Abffusor panel can be placed on each side wall. The Abffusor is manufactured by RPG Acoustical Systems.

The ceiling reflection can be controlled with a pair of similar absorption panels (2). Alternatively, a Nimbus ceiling cloud can be used. These panels are thin fabric-covered absorbers spaced out from the ceiling to improve absorption. The panels use 2-in thick, 6-lb/ft^3 glass fiber, and measure 2×4 ft. At this budget level, because it is of secondary importance, there is no treatment of the front wall behind the loudspeakers. The Nimbus is manufactured by Primacoustic.

Low-frequency modes can be controlled by two trapezoidal bass traps (3) located in the room's rear corners. Alternatively, two Modex Corner absorbers can be used. This product is available with a peak absorption at 40, 63, or 80 Hz; the latter two approximately coincide with the frequency of the first ceiling mode (575/8 = 70.6 Hz) for 8-ft ceilings, which is a common ceiling height. The Modex Corner is manufactured by RPG Acoustical Systems.

In this budget design, two diffuser panels (4) are mounted on the rear wall. They return a broad spatial pattern of diffuse reflections to the listening position. Alternatively, QRD-734 diffusing units could be mounted on the rear wall. These wooden units are based on prime 7 quadratic residue theory, with a depth of 9 in. This room should perform well for noncritical listening. The QRD-734 is manufactured by RPG Acoustical Systems.

The home listening room is shown again in Fig. 20-10; this second design is an upgrade of the basic plan. This design adds additional treatment in locations where it most cost-effectively improves the room's playback acoustics. In particular, this design doubles the area of the treatment on the side walls and rear wall. Two broadband absorption panels are now located on each side wall, and two more diffuser panels are added to the rear wall for a total of four. Two more bass traps are also added to the rear corners. The acoustics of this upgraded room will be quite suitable for most listening needs.

The home listening room is shown again in Fig. 20-11; this third design provides acoustically comprehensive, albeit more expensive, treatment. The principal change in

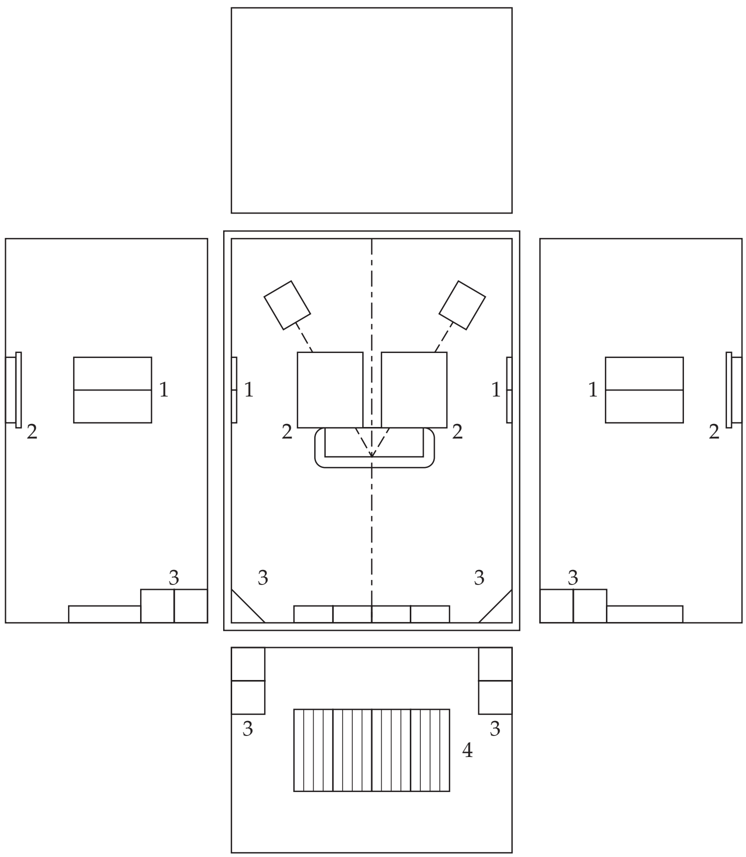


Figure 20-10 Home listening room plan with upgraded acoustical treatment.

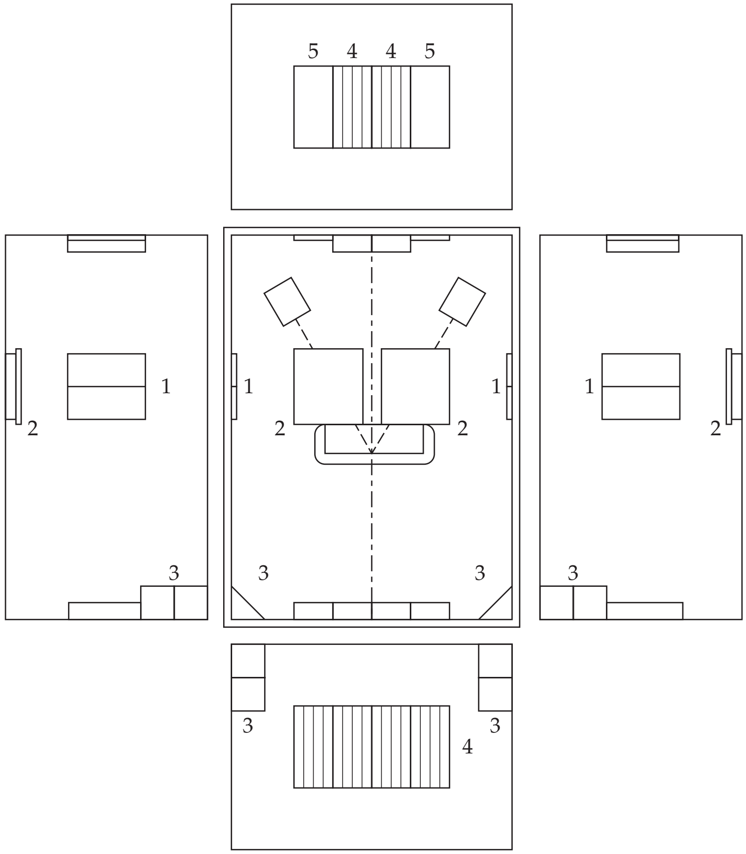


Figure 20-11 Home listening room plan with acoustically comprehensive treatment.

the listening room is treatment of the front wall behind the loudspeakers. The front wall now has two diffuser panels (4) placed on the inside and two absorption panels (5) on the outside. If the floor has a hard surface, it is also suggested (but not shown in this figure) that a rug be placed in front of the sofa to control reflections from the floor. This simple remedy could also be used in the more basic acoustical room designs.

The comprehensively treated listening room in this design should provide excellent sound quality. The imaging should be clear and defined over the entire sweet spot. Moreover, the sweet spot should be much larger than in most rooms; this is an important improvement for anything other than solo listening. Also, listeners should experience a high degree of envelopment in the music and, with controlled early reflections, frequency response at the listening position should be relatively uniform.

Home-Theater Plan

Essentially, a home theater is a listening room with a television. But it could also be argued that a home theater, at least one that aspires to replicate a cinematic experience, is more acoustically sophisticated than most listening rooms because it must satisfy a

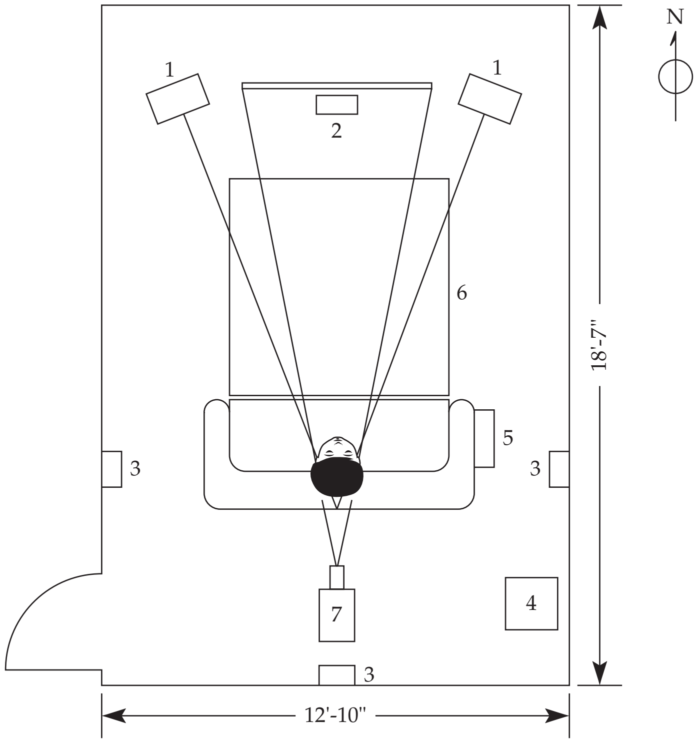


Figure 20-12 A plan view of furniture and playback equipment in a home-theater room.

wider range of acoustical requirements. In that spirit, the acoustical design of a home-theater room is presented.

Figure 20-12 shows the placement of playback equipment and furniture in a home theater. This example room measures 18 ft, 7 in × 12 ft, 10 in × 8 ft. The two main loudspeakers are arranged in a normal configuration; the included angle is typically 60° or slightly more. If possible, in most cases, the loudspeakers should be spaced away from the wall by a foot or so. Otherwise, if the loudspeakers provide bass response, loudspeaker reflections from the wall can create a peak in low-frequency response.

If reflections from the side walls are largely preserved, they will provide a wide front sound stage, extending it over the entire sofa. Conversely, placing absorbers on the side walls will provide a more focused sound stage. Although individual listening preferences may differ, it is generally best to balance these attributes. Diffusion in the back of the room will add spaciousness and envelopment. This too must be balanced with the room's overall acoustics. A good approach is to selectively add treatment a little at a time while listening to the effect of each addition. It is best to use any one treatment moderately; it is possible to overly treat a room.

In this design, the television is about 10 ft from the viewing/listening position. This dimension, typically slightly shorter than the distance between the loudspeakers, requires a relatively large television display (perhaps 80 in) for optimum viewing. Alternatively, a video projector can be used. Placing a video projector in front of the viewers is not recommended because sound reflections from the projector could adversely affect sound quality. More suitably, a projector could be mounted on the ceiling, toward the rear of the room. In either case, fan noise from the projector must be minimized.

The 5.1-channel loudspeaker system comprises a left and right stereo loudspeaker (1), a center loudspeaker (2), surround loudspeakers (3), and a subwoofer (4). The room may also include a control surface (5). If the floor is reflective, a 6- × 6-ft rug (6) might be placed in front of the sofa. The positions of the loudspeakers are predetermined except for the subwoofer. At low frequencies, directional effects are minimal. However, location of the subwoofer in the room will greatly influence its bass response at the listening position. This is discussed later.

Controlling Early Reflections

Another floor plan of the room, shown in Fig. 20-13, shows early reflections that will affect the design of the home theater. To avoid confusion, some of the elements in the floor plan of Fig. 20-12 are omitted. To simplify, four sound rays from the front left loudspeaker are considered. Sound ray *A* from the left loudspeaker is reflected from the left side wall to the listener's ears. Sound ray *B* from the left loudspeaker travels to the rear of the room and will be discussed later. Sound ray *C* from the left loudspeaker strikes

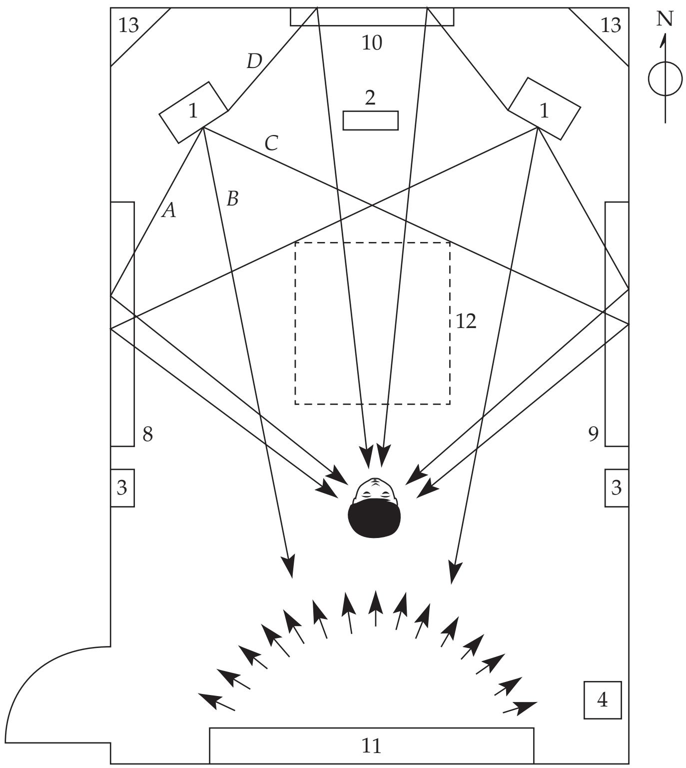


Figure 20-13 Plan view of a home-theater room showing early reflections.

the right side wall and is reflected directly to the listener. Sound ray *D* from the left loudspeaker, generated at the corners of the cabinet by diffraction, travels from the corner of the cabinet to the front wall and is then reflected to the listener.

Sound rays *A*, *C*, and *D* all carry the same program material but reach the listener at slightly different times because of the different path lengths. These rays combine with each other and with the direct ray. When two signals add in phase, a peak is produced; when they add in phase opposition, they cancel, creating a null. A comb-filter response is formed; a nominally flat response is changed to one having alternating peaks and valleys through the spectrum. To minimize audibility of this distortion, the amplitude of the early reflections must be reduced.

Our approach is to produce a reflection-free zone (RFZ) at the listening position, so sound from the loudspeakers is heard without the distortion caused by early reflections. One way this can be accomplished is by absorption. First, panels are placed on the side walls at strategic points where they will intercept rays *A* and *C*. Ray *D* will require another such panel on the front wall behind the loudspeakers.

Broadband absorbing panels using glass-fiber sheets can be placed on the front and side walls (8, 9, and 10). Alternatively, if the room is already highly absorptive, instead of absorptive panels, diffuser panels could be mounted in these locations. As yet another option, Abffusors can be used; they function both as absorbers and quadratic residue diffusers. In either case, panels can be mounted flat on the wall with no airspace behind them. If spaced away from the wall, their absorption would be increased; sound can enter the panel from the sides. For example, a 4-in airspace could be used.

Reflections from the ceiling necessitate another absorber located on the ceiling between the loudspeakers and the listening position. A panel using the same design as the absorptive wall panels can be used. It is shown in this figure as (12). Alternatively, one or two Nimbus ceiling clouds can be used. These panels are thin fabric-covered absorbers designed to be spaced out from the ceiling to improve absorption. To reduce the amplitude of the floor reflections, a 6- × 6-ft rug (6 in Fig. 20-12) is recommended. If properly placed and of the proper material, these absorbing (or diffusing) panels and rug should reduce the amplitudes of the early reflections and minimize the distortion they cause.

Ray *B* (see Fig. 20-13) and its many counterparts from both loudspeakers travel directly to the rear of the room where they encounter an array of diffusers (11). A variety of diffuser designs could be used. For example, a diffusing surface consisting of protrusions can be used. This configuration provides omnidirectional diffusion. Even if number theory is not used in the design, other irregular topographic surfaces will suffice, albeit less efficiently. Alternatively in this design, eight QRD-734 diffusers could be used. When these panels are used, they should be oriented so the four upper diffusers diffuse sound vertically, and the four lower ones diffuse sound horizontally. Many rays sweep past the listener and are reflected from room surfaces on their way to the rear of the room. Some of this energy strikes the diffusers, and some of the energy strikes the wall. Thus energy is returned to the listener either in specular or diffuse form.

Other Treatment Details

The east and west elevations of the home-theater room are shown in Fig. 20-14. These drawings illustrate the early reflections from the floor and ceiling and the absorbers used to control them. As described previously, the side-wall absorbers (8 and 9) and the absorber behind the video display or screen (10) are made of glass-fiber sheet.

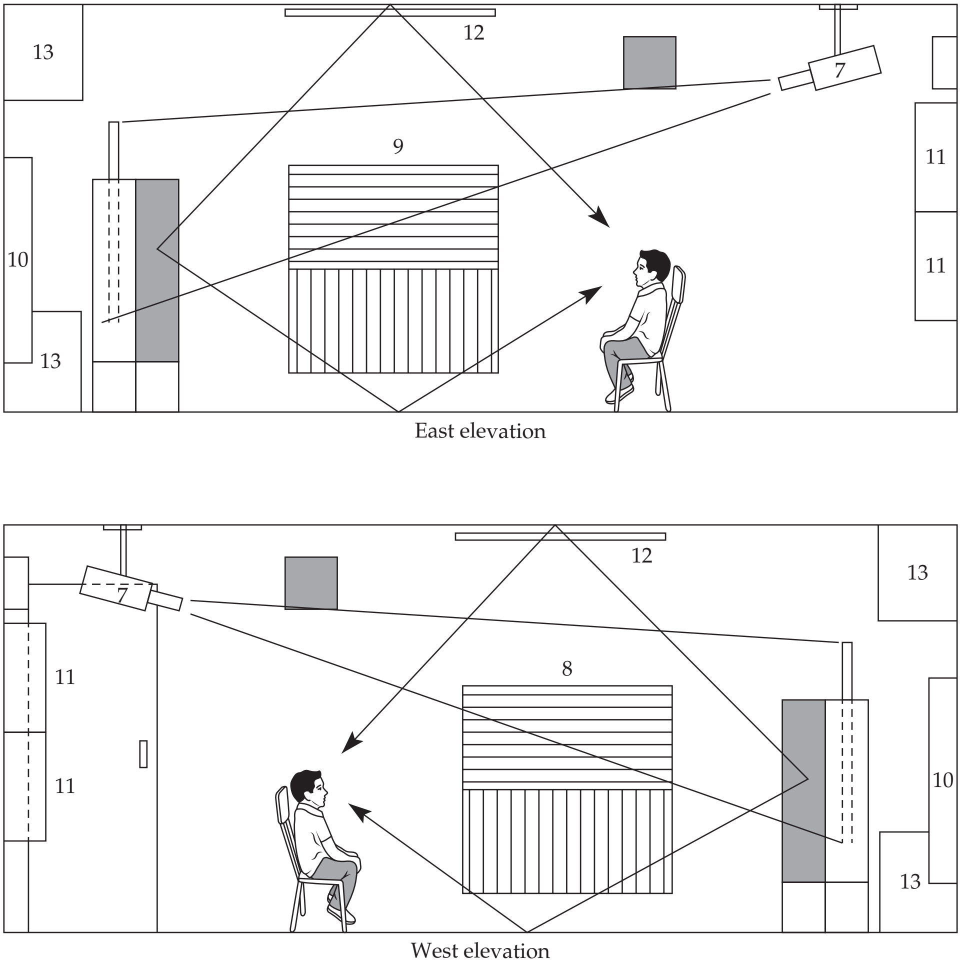


Figure 20-14 East and west elevation views of a home-theater room.

Alternatively, two Abffusors measuring 2 × 2 ft × 4 in can be placed in each location. The upper unit is oriented for horizontal diffusion, and the lower unit is oriented for vertical diffusion. The diffuser behind the video display is identical to the two side-wall diffusers. For clarity, this diffuser application is shown in this figure although other types of treatment could be used.

Bass traps (13) are mounted behind the video display, all at one end of the room, one in each of the four tri-corners. The tri-corner location is the most effective position because this is where all modes terminate. The bass traps suggested for this room are trapezoidal units that are designed for corner mounting. The face of the unit measures 2 × 2 ft. These units use internal glass-fiber sheets; if desired, the cavities between the glass-fiber sheets can be loosely stuffed with glass-fiber batt. Should the room be judged too "boomy," more bass absorbers should be mounted between the existing ones. Alternatively, Modex Corner absorbers can be used for this application. This bass absorber can be designed for peak absorption of 40, 63, or 80 Hz; the latter two approximately coincide with the first-order 70.6-Hz axial mode associated with the height of this room. It is less effective for the modes associated with the length (30.4 Hz) and width (44.1 Hz) of the room, but it still provides absorption at these frequencies.

The north and south elevations are shown in Fig. 20-15. The south elevation shows the location of four absorptive or diffusive panels (11). As noted, alternatively, four Abffusor panels can be used, oriented for vertical and horizontal diffusion. Also in the south elevation sketch, a possible location for the subwoofer (4) is indicated on the floor in the southeast corner of the room. In practice, the subwoofer should be positioned by ear.

As shown in the north elevation, both main front loudspeakers may require bases to bring them up to a desirable height (tweeters placed at approximately ear level when seated). These can be simple 3/4-in plywood boxes, filled with sand to deaden the cavity resonance. There will be numerous pieces of electronic equipment associated with this home theater. None of this equipment should be placed in front of the listener,

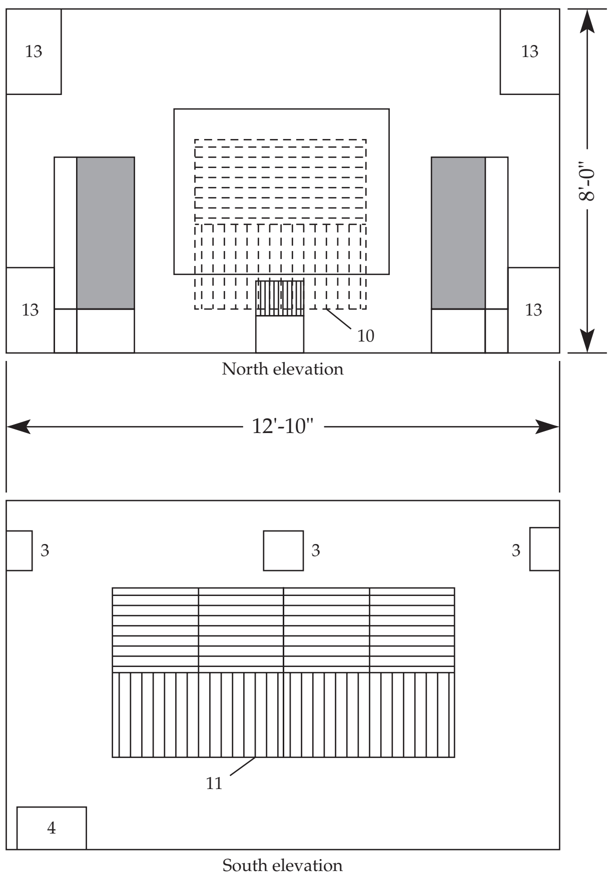


FIGURE 20-15 North and south elevation views of a home-theater room.

where it would produce undesirable early reflections. A short rack located behind the sofa is probably the best location. In that position it can be reached over the back of the sofa, a control surface could be located at the right end of the sofa as shown in Fig. 20-12, or wireless means could be used. However, if any component has a cooling fan, it should be located away from the listener.

Thus far in this design, attention has been given to the front main loudspeakers and subwoofer. Potential reflections from the center-channel loudspeaker or sound bar should fall approximately in the same surface locations as those from the front main loudspeakers; however, this should be checked and the treatment panels moved or enlarged to cover these reflections. When viewing movies in surround sound, the surround loudspeakers are often used to reproduce ambience and surround effects. Dipole loudspeakers excel in this regard. The diffusing panels on the rear wall and other treatment will assist in creating a satisfactory sound field. When front-firing surround loudspeakers (as in surround music playback) are used, more care should be taken to ensure that no unwanted specular reflections are created. If needed, as described previously, absorptive or diffusing panels should be placed as needed.

Key Points

- All the major acoustical problems involved in the design of a listening room or home theater are also present in other small audio rooms. The discussion of the acoustics of the rooms in this chapter is therefore considered as an introduction to the acoustics of other types of small audio spaces in the following chapters.
- A listening room requires extensive sound isolation to ensure a low ambient noise level, and a high-power playback system capable of generating high sound-pressure levels with low distortion.
- Ideally, a listening room or home theater should be placed in a dedicated room. Also ideally, the room should be as large as possible, it should avoid room mode problems, and it should have a quiet HVAC system.
- When treating a playback room, one must consider the time of arrival of reflections, low-frequency absorption, and reverberation time. Except in overly reverberant rooms, diffusers are preferable to absorbers for controlling early reflections.
- If a talker at the location of the center-channel loudspeaker can comfortably be heard and understood by a listener at the listening position, and speech quality sounds natural, then the reverberation time is approximately correct.
- A graphical modal analysis of a room can determine preferred positions for the sound sources and the listener. Shifts in position, including height, will particularly affect low-frequency response and sound quality.
- Bass traps can efficiently provide low-frequency absorption as well as diffusion when placed in the corners of a room near loudspeakers.
- The propagation of sound of shorter wavelengths, above about 300 Hz, can be considered as rays that undergo specular reflection. Early reflections must be carefully controlled. One approach is to add a minimum of absorbing material to treat only the specific surfaces responsible for unwanted reflections.

- The response of a loudspeaker will be influenced by any reflecting surfaces near it. Optimal subwoofer placement very much depends on room acoustics and the listening position in the room.
- In a good listening room or home theater, the imaging should be clear and defined, the sweet spot should be large, listeners should experience a high degree of envelopment, and frequency response at the listening position should be relatively uniform.
- Reflections from the side walls will provide a wide front sound stage, while absorbers on the side walls will provide a more focused sound stage. Diffusion in the back of the room will add spaciousness and envelopment. It is generally best to balance these attributes.
- A good approach is to selectively add treatment a little at a time while listening to the effect of each addition. It is best to use any one treatment moderately; it is possible to overly treat a room.

CHAPTER 21

Acoustics of Home Studios

Home studios are small recording and mixing spaces used by one or a few individuals. Given the sophistication and wide availability of recording hardware and software, a home studio can yield recordings of relatively high quality, provided that a suitable acoustical environment is available. The logical location for a home studio is usually in a bedroom, basement, or garage; occasionally a detached guest house is available. Musicians should not be discouraged by the prospect of recording in modest home studios. Many excellent recordings have come from humble origins. For example, the Foo Fighter's *Wasting Light* album was recorded in lead singer Dave Grohl's garage, and it received four Grammy awards.

With proper acoustical treatment, several good microphones, and accurate monitor loudspeakers, a home studio can be used for both recording and mixing. Other equipment might comprise a small mixing console, peripheral gear, and power amplifiers, or simply a microphone preamplifier and a laptop. One drawback to most home studios is lack of isolation from surrounding rooms or the building exterior. As a result, sounds external to the studio can interfere with recording, and likewise music performance and mixing can be intrusive to others.

The dual-purpose home studio design presented here is somewhat unique. Floor space is at a premium in a home; many times there simply is not enough space to allow for a separate control room and recording room. Dividing a room to allow both purposes would result in separate spaces, but the spaces may be so small as to be useless for either function. In this room design, one room will be used for both recording and mixing. This dual-purpose studio is not ideal but will be fully satisfactory for most music projects. The design is presented in three iterations, evolving from very low cost and progressing toward higher cost. The playback conditions are considered first, and then the suitability of the room for recording is discussed. Also, as a separate concept, a design for a garage studio with separate control room and recording studio is presented.

Home Acoustics: Modes

All small rooms, including small rooms found in homes, have modal resonances at audible frequencies. In particular, at low audio frequencies, the wavelength of sound is of the same order as the dimensions of the room. This means that sound-pressure peaks and nulls will arise as those frequencies are sounded in the music. This can degrade the low-frequency response of the room, including the response of its reverberation, by introducing peaks and nulls throughout the room. The effect of room modes can be decreased somewhat by choosing certain room proportions. However, most home studios are built in existing rooms. With a room of predetermined dimensions, the only approach is treatment such as bass traps to help mitigate the effect of resonant modes. While not a complete solution, treatment can provide satisfactory acoustical results. The theory behind room modes is discussed in Chap. 13.

Home Acoustics: Reverberation

As described in Chap. 11, Jackson and Leventhall reported that the average reverberation time of 50 living rooms was about 0.7 sec at 100 Hz and decreasing to 0.4 sec at higher audio frequencies. This is a reasonable target range for a home studio. The average bedroom is smaller in size than the average living room and would have a somewhat lower reverberation time, but is still quite usable. So, the reverberation times found in most homes are probably acceptable, or at least within a range that can be adjusted to be acceptable, for a home studio. Wall treatment will be used to fine-tune the reverberation time and also to control reflection paths.

Home Acoustics: Noise Control

The ambient noise levels in most homes and neighborhoods are too high for most recording projects. Moreover, most attempts to isolate a home studio will almost certainly be insufficient. The cost of isolation, and the interior volume it occupies, makes good isolation extremely difficult to achieve in most homes. This background noise problem has no easy solution, although working with a close microphone placement will help. Another practical approach is to record late at night when the world is more quiet. However, it is also important to remember that recording and mixing, especially late at night, may be intolerable to others near the studio especially if amplified instruments are played. Many local ordinances limit residential noise levels, particularly at night. For example, the noise-level limit at a property line might be 45 dBA, or 5 dB over ambient noise levels. These low levels can be difficult or impossible to achieve when isolation is poor. Similarly, noise intrusion to others in a house or neighbors in an apartment may be a significant issue.

In many cases, when available, for noise reasons, a basement may be the best location for a home studio. It economically offers a high degree of isolation from the neighboring external environment, if not always from the upstairs house. A freestanding garage separated from a main house would lower noise levels in the house but may not solve noise problems with neighbors.

As in any structure, the windows in a home are an acoustical weak link. External sound can easily penetrate most home windows, and likewise sound from within can emerge to annoy neighbors. Many kinds of windows can be covered with a device that is simple to construct. Much like a hurricane shutter, it shields the window glass, in this case, from noise. Figure 21-1 shows the design. A cover is constructed of several layers of 3/4-in particle board; for example, four layers will yield a transmission loss of about 40 dB at 500 Hz; the transmission loss (TL) curve has an upward slope with respect to frequency. A wood frame is placed around the outer window sill; it is secured with adhesive. The cover is secured to the frame with carriage bolts; this makes the cover

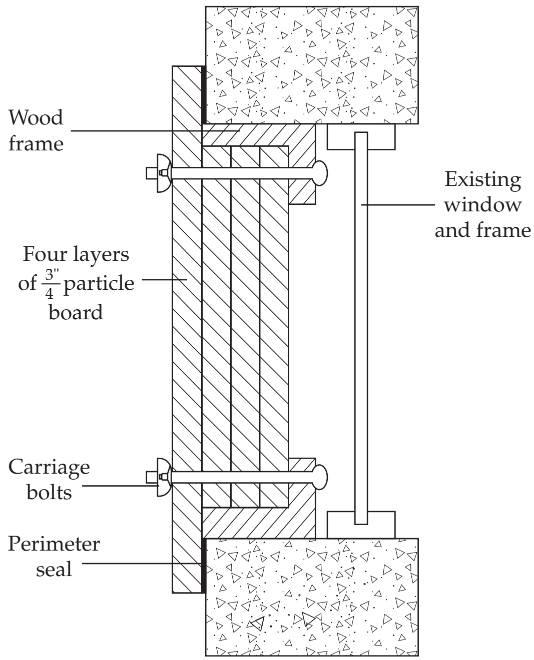


Figure 21-1 Noise intrusion through a window can be reduced by covering the window with a multilayer wood cover, sealed at the periphery.

easily removable when not needed. Alternatively, the cover can be nailed to the frame. Perimeter seals are pressed into place. The cover can be placed on either the exterior or interior side of the window.

Despite the best efforts, a home studio will almost certainly have noise intrusion problems. It should be expected that external noise will sometimes ruin an otherwise satisfactorily recorded take. The cost of absolutely preventing this would far exceed the frustration of occasionally ruined takes. The more serious issue is the potential complaints from neighbors.

Home Studio Budget

Most home studios are built on a modest budget. Although prefabricated acoustical modules greatly simplify construction and produce excellent acoustical results, their cost is beyond the reach of many home-studio builders. Alternatively, modules can be constructed from simple building materials at low cost. These modules take time and effort to build, may lack the finished appearance of commercially manufactured units, and may not be acoustically ideal, but they will certainly be functional and effective. Therefore in this home studio design example, the three types of acoustical treatment (broadband absorption, bass traps, and diffusers) are designed to be built from scratch.

Home Studio Treatment

In an untreated control room or listening room, the desirable direct sound from the loudspeakers to the mixing or listening position is degraded by early reflections from walls, floor, and ceiling. These reflections arrive at the listener at slightly different times, creating acoustical comb filters that cause peaks and notches in the frequency response of the direct sound. This is contrary to the fact that in any critical playback environment, it is essential to have an accurate sound field for mixing and listening. To overcome this problem, this studio will use the concept of a reflection-free zone (RFZ). In an RFZ room design, these early reflections are eliminated or attenuated; this helps preserve the perceived frequency response of the loudspeakers, improves the accuracy of stereo imaging, and enhances the breadth of the sound stage. The RFZ concept is discussed in more detail in Chap. 24.

Figure 21-2 shows an example of an absorption panel that can help create an RFZ in this home studio. The panel is constructed with 1- × 6-in framing with a 1/4-in backing panel. The face of the unit measures 1 ft, 6 in × 4 ft. The interior contains a 4-in glass-fiber sheet or batt that is held in place with open wire mesh; the glass fiber is placed at the front of the frame with a 2-in airspace behind. The front of the panel is covered by an open-weave fabric. The panel can be attached to the structural wall with metal clips. The panel absorbs broadband sound; this is optimized by spacing the glass fiber away from the wall.

Panels can be located on walls and ceiling between the loudspeakers and the mixing/listening position at locations where sound from the loudspeakers would otherwise reflect and then travel to the listening position. These locations can be determined by sitting at the listening position while an assistant moves a mirror across likely reflective surfaces. When the loudspeakers can be seen in the mirror, this identifies positions that should be covered by panels.

As noted previously, small rooms are generally subject to low-frequency reverberation problems resulting from the modal resonances related to the room dimensions. In addition, in some rooms, most available absorption is at mid and high frequencies so low frequencies are relatively unabsorbed and thus more prominent. This problem of "boomy" sound in small rooms is well known. Because of the long wavelength of low-frequency sound, effective absorbers require much space. It is wise to utilize the corners of a room because all modes terminate in corners, making this location most effective

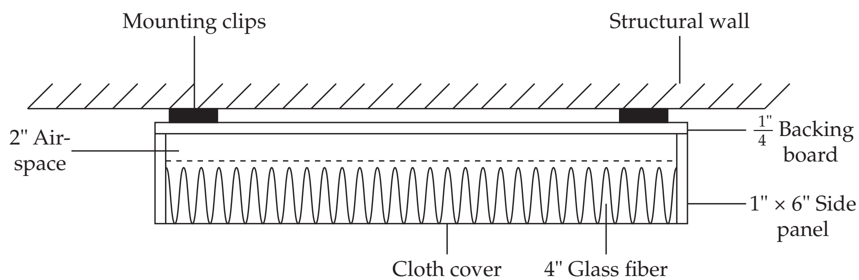


Figure 21-2 A wall-mounted absorptive panel.

for bass absorption. Moreover, the corners of a room are often unused spaces. Bass traps are discussed in Chap. 12.

The problem of boomy sound stems from lack of low-frequency absorption. This is often the case in rooms with walls constructed of masonry such as concrete blocks and brick. On the other hand, rooms constructed with gypsum-board stud walls can be free of this problem because walls of such construction are good natural low-frequency absorbers. This diaphragmatic absorption is quite effective in the 70- to 250-Hz region. Many homes are constructed with drywall partitions; thus bass traps might not be needed. The prudent course is to construct the room without additional low-frequency absorption, then measure and listen to the room. If the reverberation time is consistent across the audio spectrum, the room has adequate low-frequency absorption. If the reverberation time at low frequencies is longer than at high frequencies, bass traps or other low-frequency absorbers may be needed. The question of drywall low-frequency absorption is considered in Chap. 12.

The bass traps suggested for this studio are trapezoidal units that are designed for corner mounting, as shown in Fig. 21-3. The face of the unit measures 2 × 2 ft. These units use internal glass-fiber sheets to provide absorption. Their relatively deep depth in the corner increases absorption at low frequencies. If desired, the cavities between the glass-fiber sheets can be loosely stuffed with glass-fiber batt.

Diffusion can be obtained from large surface irregularities, polycylindricals, and other geometrical shapes. In particular, quadratic residue diffusers are often recommended. These highly efficient diffusing surfaces, based on number theory implemented as a series of parallel wells, can easily be constructed by a carpenter. With proper orientation, both horizontal and vertical diffusion can be obtained. The design of these diffusers is described in Chap. 14.

In this studio design, a different implementation is used. The diffusing surface consists of an array of protrusions as shown in Fig. 21-4. The face of the unit measures 2 × 2 ft. This configuration provides omnidirectional diffusion of first-order and other reflections. Even if number theory is not used in an optimal design, any irregular topographic surface will provide some degree of diffusion. However, if low-frequency

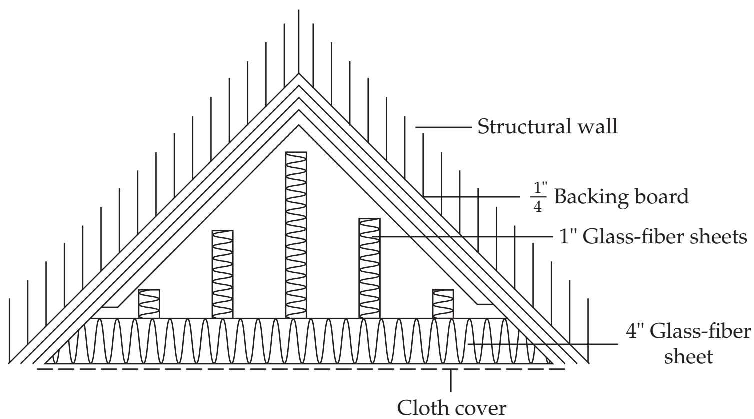


Figure 21-3 A corner-mounted bass trap module.

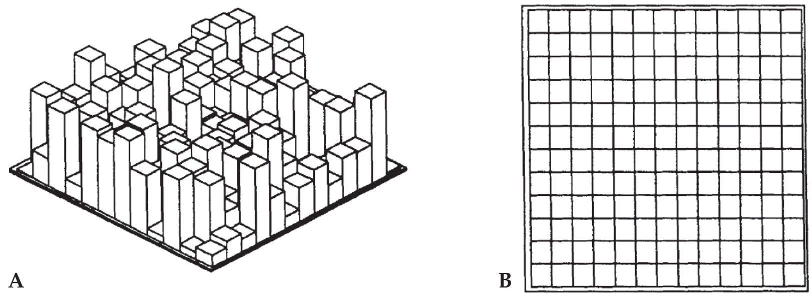


Figure 21-4 A wall-mounted diffusive module. (A) Isometric view. (B) Plan view.

diffusion is desired, the surface projections will have to be relatively large. When these modules are mounted on the rear wall, they will help create a diffuse field in the room. Modules can be mounted to the wall with metal clips. Diffusion is discussed in Chaps. 9 and 14.

Home Studio Plan

In most cases, a home studio is constructed in an existing room; the design must be adapted to whatever the existing room dictates. The rooms in most homes are predictably rectangular, with 8-ft ceilings, and often use drywall construction. In that respect, a generic home studio design can often be successfully employed as a starting point for a customized design. A home studio plan is presented here in three iterations, with increasingly elaborate treatments.

The design for the proposed studio using basic acoustical treatment is shown in Fig. 21-5. To minimize costs, this initial treatment proposes very minimal treatment for early reflections, minimal bass absorption, and minimal rear-wall diffusion. Much potential improvement remains. In many ways, this design is similar to that of a home listening room. We begin by studying the properties of the room for playback; later, we will explore whether this same room can be used successfully for recording.

Two absorptive panels (1) are applied to each side wall. There will be some improvement in sound quality because many of the side-wall reflections are treated. However, the panels cover a limited surface area, so not all of the early reflections will be intercepted. The listening or mixing position will be somewhat troubled by early reflections, because reflections from the front wall between the loudspeakers are left untreated, as are reflections from the floor and ceiling. Two bass traps (2) are mounted in each of the two rear corners. They would help reduce boominess but may not completely eliminate it in some rooms. As discussed, in some drywall construction, minimal bass trapping, or none, may be adequate.

Four diffusing units (3) on the rear wall intercept only a small fraction of the sound energy falling on the rear wall. These four units will not provide a truly diffuse field, but their presence will be noted as a modest improvement in the sound field in terms of imaging and envelopment.

An additional layer of room treatment is shown in Fig. 21-6. It represents the next logical steps and may be considered as an intermediate-quality room design. In this

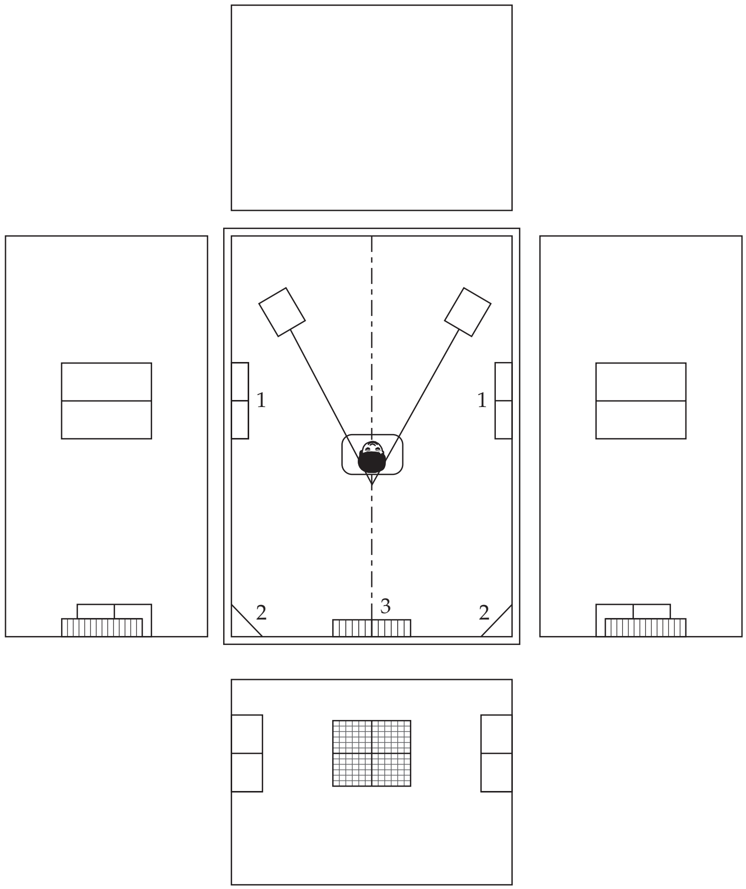


Figure 21-5 Home studio: Initial treatment.

design the number of absorptive units (1) on each side wall is the same as in the earlier initial-treatment design, but there are now four absorptive panels on the ceiling. This minimizes unwanted early reflections from the ceiling. In addition, the floor reflections could be reduced materially with a 6- × 6-ft rug with thick pile. In addition, there are now four bass traps (2) in each of the two rear corners. If the ceiling height was limited to 8 ft and the modules were used, this would be the maximum possible number in a corner. The diffusive panels (3) are left unchanged.

A comprehensive treatment is shown in Fig. 21-7. This more costly design essentially increases the number of all of the treatment units to the required level. Three absorptive panels (1) are now placed on each side wall, six on the ceiling and four on the front wall between the loudspeakers. Four bass traps (2) are now in each of the four corners of the room. Another option, if space permits, is to build a gypsum-board closet in a corner; its lightweight wall construction would let the closet double as a bass trap. Eight of the diffusive panels (3) are now mounted on the rear wall. This diffusion will help provide improved definition of the stereo image, accurately perceived depth of sound stage, and a feeling of being enveloped by the sound.

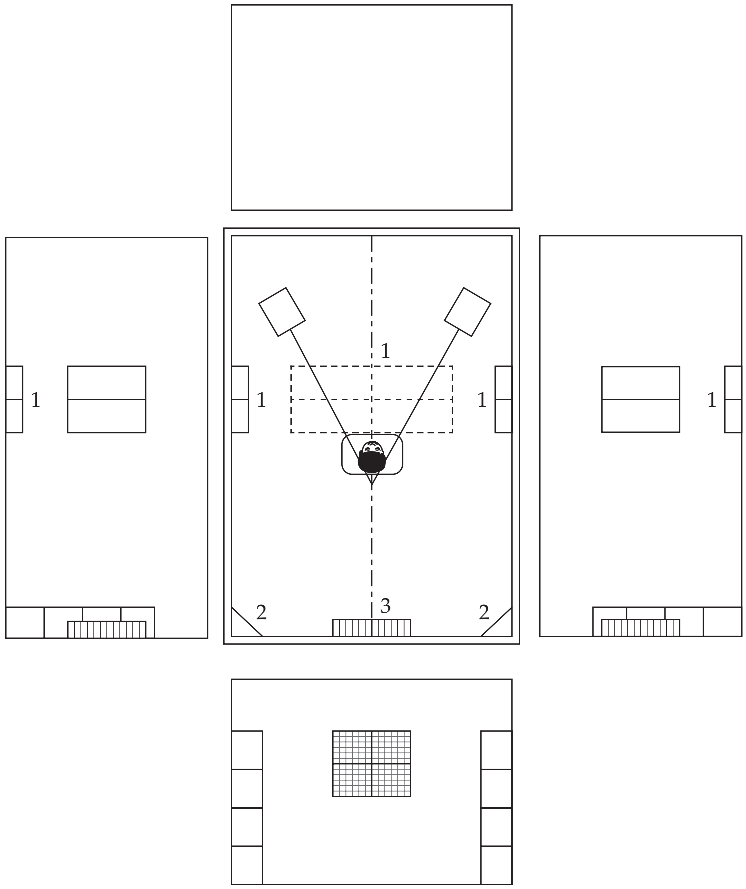


Figure 21-6 Home studio: Intermediate treatment.

Overall, with this additional treatment, potentially troublesome acoustical problems should be eliminated. The room should provide neutral acoustics so that mixes done under these acoustical conditions should be transferable to other listening environments.

Recording in the Home Studio

The room treatments described thus far (see Figs. 21-5 to 21-7) were designed for mixing with loudspeakers as the sound source; these designs essentially replicate a conventional RFZ control room. However, to complete this home studio design, the same room must also be used as a recording studio for music performance. In this case, musicians are the source of sound. Fortunately, these rooms would also provide good recording venues.

If the musicians are placed between the loudspeakers and the microphone is placed at the listening and mixing sweet spot, the recording would be relatively free of early

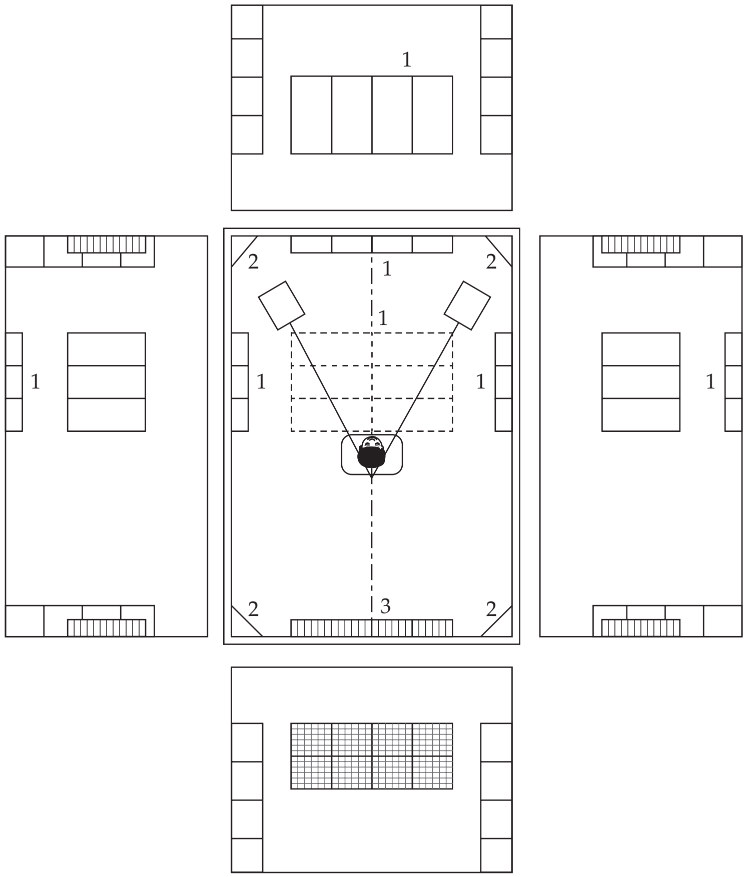


Figure 21-7 Home studio: Comprehensive treatment.

reflections. The side wall, ceiling, and floor absorbers were placed according to loudspeaker directionality. With a broader source, such as one or more musicians located between the two loudspeakers, the early sound would not be treated completely by these absorbers, but it would be partially treated. In other words, some early reflections would remain; however, this is not necessarily bad. It can certainly be argued that some early reflections are expected when we listen to live music performance. Thus the requirement is different from our original goal of preserving a flat response from the monitor loudspeakers. In any case, this musician/microphone configuration would yield good recorded results.

Also, the acoustical asymmetry of the room as it has been treated will give a degree of variability during recording; placing musicians and microphones in different locations in the room will change the nature of the recorded sound. For example, the mixing chair could be pushed to one side and this main area of the room used by musicians. The absorbers on side walls, front wall, and ceiling will give a moderate reverberatory condition. The bass traps in the corners will give a "tight" bass sound. The large panel of diffusers will provide a sense of openness that will contribute more acoustical space to recordings, and the sense of space will also be greatly appreciated by the musicians. In short, all the elements are present for a good recording studio, even though the treatment has been placed primarily with playback in mind.

The studio designs presented here (progressively, from basic to comprehensive) will provide a good home studio that is suitable for both recording and mixing. The acoustical elements that have been added can be constructed at low cost, and are highly adaptable to the space available in most homes.

Garage Studio

A detached garage would provide a degree of isolation from the main house as well as sufficient floor space for a home studio. A typical two-car garage might measure 24 × 24 ft, yielding a floor area of 576 ft^2. As shown in Fig. 21-8, a partition (1) could be placed diagonally across one corner, yielding a control room of about 176 ft^2 and a studio of 400 ft^2. These are minimal room sizes but are workable. (In the previous example in this chapter, a rectangular space is used, and the control room and studio are integrated into one room; this open plan could be implemented in a single-car garage.)

In a two-car garage, the subdividing partition should be a staggered-stud wall or a double wall with multiple layers of gypsum board on both sides. An observation window (2) should be double pane with each pane independently set into each wall. Microphone panels and other fixtures on the common wall should be staggered

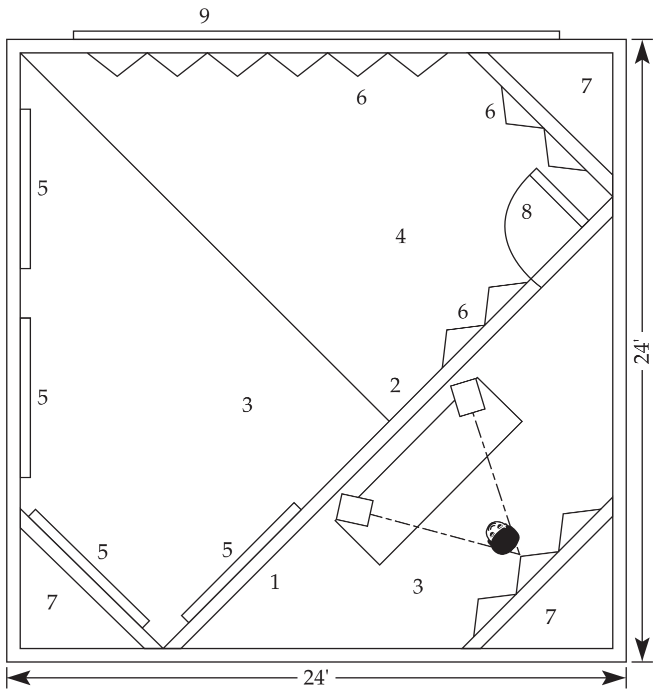


Figure 21-8 Floor plan of a garage studio with separate control room and studio.

(not located back to back), and all openings and conduits should be sealed. Although it is probably not practical, and the benefit would be fairly minimal, the studio and control room could be further isolated by cutting the concrete slab to yield two independent slabs. Ideally the control room should be constructed as a room within the garage, with separate walls, floating floor, and ceiling. Consideration should be given to heating and ventilating each room; clearly, there must not be any common ductwork between the studio and control room.

Both rooms should be relatively dead with reverberation times below 0.5 sec. Carpet (3) and pad can be laid on a portion of the concrete slab, and tile (4) or parquet on another portion. A mixture of absorptive panels (5), reflecting panels, and diffusers (6) should be placed on the walls and ceiling. The control room has axial symmetry with the console facing the diagonal wall; in such a small space, the monitor loudspeakers should be placed in a near-field configuration. A combination resonator/diffuser (7) can be placed in the rear corner as well as in the corners of the studio. Any single doors (8) should be solid core and tightly weather-stripped. Clearly, any open louvers and glass windows should be sealed, using the wood covers as described earlier in this chapter.

For greater recording diversity, even in such a small studio, one end of the studio should be more absorptive, whereas the other end is more live. For example, generally, drums will be recorded in the dead end of the studio. Isolation in drum tracks can be enhanced by suspending a thick absorptive panel over the drums. Moreover, a bass trap can be located in the space above the absorbing panel; a bass trap can be constructed by hanging rows of glass-fiber boards vertically. Conversely, an acoustical guitar would be recorded in the live end of the studio. The walls there are reflective and diffusive, and the floor is left uncovered.

On the downside, a garage might provide little sound isolation. It might be of lighter construction than the main house and may be situated close to a neighbor's house. Isolation can be improved by adding a layer of cement board to the exterior and a double drywall layer on the inside; the latter should be mounted on isolators or independent framing. All seams should be taped or sealed with nonhardening acoustical sealant. Because garage doors use lightweight construction, they offer little isolation. A new wall should be constructed behind the garage door (9), leaving the garage door as a facade.

Key Points

- A small home studio may be designed so that one room is used for both recording and mixing. Alternatively, if space allows, the control room and recording studio may be separated by an isolation wall.
- The reverberation times found in most homes are acceptable, or within a range that can be adjusted to be acceptable, for a home studio. Wall treatment can be used to fine-tune the reverberation time and also to control reflection paths.
- In many cases, when available, for noise reasons, a basement may be the best location for a home studio. It economically offers a high degree of isolation from the neighboring external environment, if not always from the upstairs house.
- Windows are an acoustical weak link; the transmission loss of most windows can be increased with a simple wood cover constructed, for example, of several layers of 3/4-in particle board and sealed at the periphery.

- The home studio designs presented here use the concept of a reflection-free zone (RFZ) in which early reflections are attenuated or eliminated. The RFZ concept is discussed in more detail in Chap. 24.
- Many homes are constructed with drywall partitions that naturally provide diaphragmatic absorption in the 70- to 250-Hz region. Thus bass traps might not be needed. However, if low-frequency reverberation time is too long, additional absorption might be needed.
- Three studio designs show the use of absorptive panels on the side walls, ceiling, and the front wall between the loudspeakers; bass traps in the corners of the room; and diffusive panels on the rear wall.
- In these designs, for recording, if musicians are placed between the loudspeakers and the microphone is placed at the mixing position, the recording would be relatively free of early reflections. Alternatively, musicians and microphones could be placed in different locations in the room; the acoustical asymmetry of the room will give a degree of variability.
- A two-car garage could be subdivided by a partition to create a separate control room and recording studio with minimal yet workable dimensions. The dividing partition must provide isolation, and consideration must be given to exterior noise.

CHAPTER 22

Acoustics of Small Recording Studios

Because of their wide availability and favorable economics, small studios play an important role in the recording industry. Many musicians use small studios to develop their skills and to make both demo and commercial recordings. Music recorded in small studios can certainly have high sound quality that makes it suitable for commercial release. Many small studios operated by not-for-profit organizations produce material for educational, promotional, and religious purposes. Small studios are required for the production of campus and community radio, television, and cable programs. These small studios have limited budgets and limited technical resources. The operators of these studios are often caught between a desire for top quality and the lack of means. This chapter is primarily intended for owners of small studios who wish to optimize their acoustical environment with a keen eye on budget although the acoustical principles expounded are widely applicable, even to the design of large studios.

What is a good recording studio? There is only one ultimate criterion—the acceptability of the sound recorded in it for its intended audience. In a commercial sense, a successful recording studio is one that is fully booked and making money. If listeners like the recording, the studio passes the test. There are many factors influencing the acceptability of a studio beside sound studio quality, but sound quality is vital, at least for success on a substantial, long-term basis.

This chapter focuses on the acoustics of the "studio" room in a small recording studio, that is, the room where music is performed and recorded. The room's ambient acoustical response, as perceived by musicians and picked up by microphones, is paramount. Chapter 24 discusses the acoustics of control rooms, that is, the room in recording studios where sound is played back and mixed. In control rooms, the acoustics of loudspeaker playback is most critical.

Ambient Noise Requirements

To be useful, any studio must have low ambient noise levels. In fact, studios rank among the quietest work spaces—when they are not being used. This can be quite difficult to achieve. In many cases, a room is "quiet," except for a few occasional noises. However, this is unacceptable. A studio must be quiet at all times.

Many noise and vibration problems can be avoided entirely by choosing a site in a quiet location. Clearly, noisy sites must be avoided. Is the site close to a train track or busy intersection? Is it under the approach path (or worse yet, the takeoff path) of an airport? The maximum external noise spectrum must be reduced to the background noise criteria goal within the space by the transmission loss of the walls, windows, and doors. Floating floors and other relatively costly construction methods may be required. Whenever possible, choosing a quiet site is much more cost-effective than isolating a studio sited at a noisy location.

If the space being considered is within a larger building, other possible noise and vibration sources within the same building must be identified and evaluated. Is there a machine shop on the floor above? A noisy elevator? A dance studio? A reinforced concrete building, while providing high transmission-loss partitions, also efficiently conducts noises throughout via structural paths. The structure of the walls determines the degree to which external noise is attenuated. This is equally true of the ceiling and floor structures in regard to attenuation of noise from above and below.

A professional studio, even a small one, might require support activities such as tech support, duplication, accounting, sales, reception, and shipping and receiving, each with its own noise generation potential. Review of such noise sources will helpfully concentrate attention on transmission loss characteristics of internal walls, floors, and ceilings.

The HVAC (heating, ventilating, and air-conditioning) system must be designed to provide the required noise criteria goals. Noise and vibration from motors, fans, ducts, diffusers, and grilles must all be minimized for satisfactorily low ambient noise levels. Noise control and HVAC design are discussed in Chaps. 16 through 19.

Acoustical Characteristics of Small Studios

Sound picked up by a microphone in a studio consists of both direct and indirect sound. The direct sound is the same as would exist in a free field or in an anechoic chamber; in this case, it is the sound that travels in a straight line from the source to the microphone. Everything that is not direct sound is indirect sound, that is, reflected sound. The indirect sound, which immediately follows the direct, is the sound that results from all the various non–free-field effects characteristic of an enclosed space. The latter is unique to a particular room and comprises the room's acoustical response.

Direct and Indirect Sound

Figure 22-1 shows how sound level varies with distance from a source, for different room absorbencies. The source could be someone talking, a musical instrument, or a loudspeaker. Assume a sound-pressure level of 80 dB measured 1 ft from the source. If all surfaces of the room are 100% reflective, as shown by contour A, the room is a reverberation chamber and the sound pressure level would be 80 dB everywhere in the room because no sound energy is absorbed. Relatively, there is almost no direct sound; it is almost all indirect. Contour B represents the fall off in sound-pressure level with distance from the sound source with all surfaces 100% absorptive. In this case all the sound is direct; there is no indirect component. The best anechoic chambers approach this condition. This is a true free field, and for this condition the sound-pressure level decreases 6 dB for each doubling of distance.

Role of Room Treatment

Between the indirect "all-reverberation" case of contour A and the direct "no-reverberation" case of contour B lie a multitude of other possible "some-reverberation" cases that depend on room treatment. In the area between these two extremes lies the

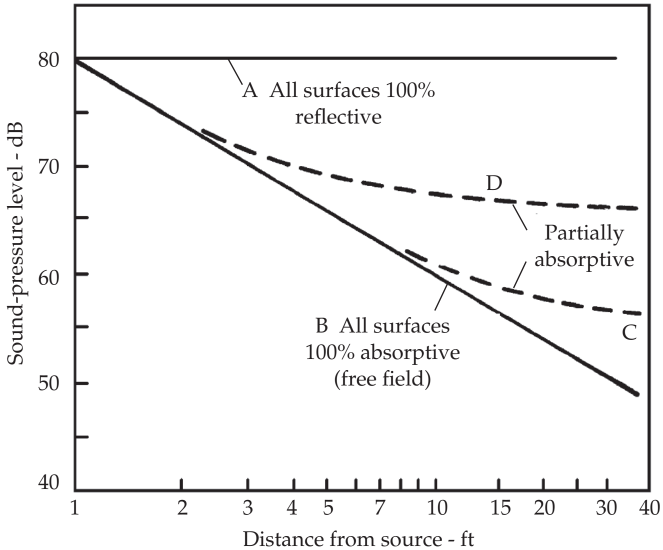


FIGURE 22-1 The sound-pressure level in an enclosed space varies with distance from the source of sound, and also varies with the absorbency of the space.

real world of studio acoustics. The room represented by contour C is much deader than that of D. In practice, the direct sound is observable a short distance away from the source, but farther away, the indirect sound dominates. A transient sound picked up by a microphone would, for the first few milliseconds, be dominated by the direct component, after which the indirect sound arrives at the microphone as reflections from room surfaces. These surface reflections are spread out in time because of the different path lengths traveled.

Another component of indirect sound results from room resonances, which in turn are the result of reflected sound. Specific frequencies contained in the sound from the source excite these resonances, bringing into play all the effects described in Chap. 13. When the source excitation ceases, each mode dies away at its own natural frequency and at its own rate. Sounds of very short duration might not last long enough to fully excite room resonances.

Distinguishing between reflections and resonances acknowledges that neither a reflection concept nor a resonance concept alone will fully describe room acoustical behavior throughout the entire audible spectrum. Room mode analysis is used to characterize the effects of resonances which dominate the low-frequency region in which wavelengths of sound are comparable to room dimensions. The ray concept is used to analyze the effects of reflections, which determine room response at higher frequencies and their shorter wavelengths. The 300- to 500-Hz region is a transition zone. By remembering the limitations of each analysis, the components of sound in a small studio can be characterized.

Indirect sound also depends on the materials of construction, such as doors, windows, walls, and floors. These too are set into vibration by sound from the source, and they too decay at their own particular rate when excitation is removed. For example, if

Helmholtz resonators are used in room treatment, sound energy not absorbed by them is reradiated.

The sound of a studio, embracing the direct sound plus all of the components of the indirect sound, has a counterpart in the sound produced by musical instruments. In fact, it is helpful to consider a studio as a musical instrument. It has its own characteristic sound, and skill is required to extract its full potential.

Room Modes and Room Volume

The short dimensions of a small room almost guarantee acoustical anomalies resulting from excessive spacing of room resonant frequencies. A cubical room distributes modal frequencies in the worst possible way, by piling up all three fundamentals, and each trio of multiples with maximum gaps between modes. Having any two dimensions in a multiple relationship also results in this type of problem. For example, a height of 8 ft and a width of 16 ft mean that the second harmonic of 16 ft coincides with the fundamental of 8 ft. This emphasizes the importance of proportioning the room for the best distribution of axial modes. This is particularly important in small rooms; as we will see in later chapters, large rooms have the inherent advantage of relatively more modes in the low-frequency region, and hence smoother low-frequency response.

If there are fewer low-frequency axial modes than desired in the room under consideration, and this is always the case in a small room, mode frequencies must be distributed as uniformly as possible. This can be achieved by choosing a favorable room dimension ratio. For example, Sepmeyer suggests a dimension ratio of 1.00:1.28:1.54. Table 22-1 shows the selected dimensions, based on ceiling heights of 8, 12, and 16 ft, resulting in room volumes of approximately 1,000, 3,400, and 8,100 ft^3.

Mode Analysis for Different Room Sizes

Axial mode frequencies for three rooms (1,000, 3,400, and 8,100 ft^3) were calculated after the manner of Table 13-6 and plotted in Fig. 22-2. A visual inspection of Fig. 22-2 shows the increase in the number of axial modes as volume is increased. This results in the benefit of closer low-frequency mode spacing, and hence smoother low-frequency response.

The number of axial modes below 300 Hz varies from 15 for the small studio to 31 for the large studio, as shown in Table 22-2. The low-frequency response of the two smaller studios at 45.9 and 30.6 Hz is inferior to the large studio at 22.9 Hz. The average spacing of modes, based on the frequency range from the lowest axial mode to 300 Hz, is also listed in Table 22-2. The average spacing varies from 16.9 Hz for the small studio

	Ratio	Small Studio	Medium Studio	Large Studio
Height	1.00	8.00 ft	12.00 ft	16.00 ft
Width	1.28	10.24 ft	15.36 ft	20.48 ft
Length	1.54	12.32 ft	18.48 ft	24.64 ft
Volume	—	1,000 ft^3	3,400 ft^3	8,100 ft^3

Table 22-1 Dimensions and Volumes for Three Studio Sizes

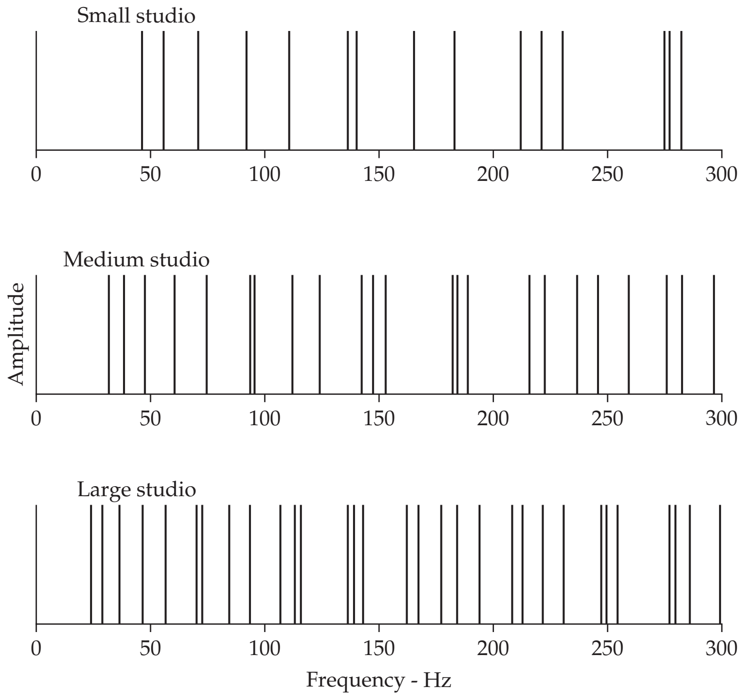


Figure 22-2 Comparison of the axial-mode resonances of a small (1,000 ft^3), a medium (3,400 ft^3), and a large (8,100 ft^3) studio, all having the proportions 1.00:1.28:1.54.

to 9.2 Hz for the large studio. Room volume is thus an important factor in recorded sound quality; small studios have an inherent limitation in this regard.

As we have seen in previous chapters, in addition to axial modes, a room will support tangential and oblique modes. Although they are weaker in energy, a thorough analysis would consider their effect. In particular, as with axial modes, a more even distribution will help smooth the low-frequency response of the room. Conversely, an irregular distribution of tangential and oblique modes, particularly if they coincide with axial modes, will increase irregularities in the room's low-frequency response and may create audible timbral defects.

Parameter	Small Studio	Medium Studio	Large Studio
Number of axial modes below 300 Hz	15	23	31
Lowest axial mode (Hz)	45.9	30.6	22.9
Average mode spacing (Hz)	16.9	12.0	9.2
Assumed reverberation time of studio (sec)	0.3	0.5	0.7
Mode bandwidth ($2.2/RT_{60}$)	7.3	4.4	3.1

Table 22-2 Studio Resonances and Mode Bandwidths for Three Studio Sizes

The reverberation times listed in Table 22-2 are assumed, nominal values for the respective studio sizes. Given these reverberation times, the mode bandwidth is estimated from the expression $2.2/RT_{60}$. Mode bandwidth varies from 3.1 Hz for the large studio to 7.3 Hz for the small studio. The advantage of closer spacing of axial modes in the large studio tends to be offset by its narrower mode bandwidth. So, we see conflicting factors at work as we realize the advantage of the mode skirts overlapping each other. In general, however, the greater number of axial modes for the large studio, coupled with the extension of room response in the low frequencies, produces a response superior to that of the small studio.

A studio having a very small volume has fundamental response problems in regard to room resonances; larger studio volume yields a smoother low-frequency response. Frequency-response anomalies encountered in studios having volumes less than 1,500 ft^3 are sometimes so severe as to make small studio rooms impractical for recording music. This points out the need for careful consideration in the planning of any small studio.

Reverberation Time

Reverberation is the composite effect of all three types of indirect sound: reflections from room surfaces, room resonances, and effects of room construction materials. Measuring reverberation time does not directly reveal the nature of these individual components of reverberation. Herein is the weakness of reverberation time as an indicator of a room's acoustical quality. The important action of one or more of the reverberation components may be obscured by the compositing process. This is why reverberation time is one indicator of acoustical conditions, but not the only one.

Reverberation in Small Rooms

Some acousticians feel it is inaccurate to apply the concept of reverberation time to relatively small rooms. It is true that a genuine reverberant field may not exist in small spaces. The Sabine reverberation equation is based on the statistical properties of a random sound field. If such an isotropic, homogeneous distribution of energy does not prevail in a small room, is it proper to apply the Sabine equation to compute the reverberation time of the room? The answer is a purist "no," but a practical "yes." Reverberation time is a measure of decay rate. A reverberation time of 0.5 sec means that a decay of 60 dB takes place in 0.5 sec. Another way to express this is 60 dB/0.5 sec = 120 dB/sec decay rate. Whether the sound field is diffuse or not, sound decays at some particular rate, even at the low frequencies at which the sound field is least diffuse. The sound energy stored at the modal frequencies also decays at some measurable rate, even though only a few modes are contained in the band being measured. It would seem to be practical to utilize the Sabine equation in small-room design to estimate absorption needs at different frequencies. At the same time, it is important to remember the limitations of the process.

As noted, technically, the term "reverberation time" should not be associated with relatively small spaces in which the sound field is not random. However, a designer must calculate the amount of absorbent needed to establish the acoustical character of a room. While reverberation time is useful for this purpose, the values of reverberation time in a small studio do not have the same significance as in a large space. We must remember that in a small room, the assumptions underlying the Sabine equation are not met; as a result, the equation's predictions, while useful, may be less than accurate.

Optimal Reverberation Time

If a room's reverberation time is too long, speech syllables and music phrases are masked and a deterioration of speech intelligibility and music quality results. If reverberation time is too short, music and speech lose character and suffer in quality, with music suffering more. These effects are not definite and precise; there is no specific optimum reverberation time for any room because many other factors are involved. For example, is it a male or female voice, slow or fast talker, English or German language (they differ in the average number of syllables per minute), vocal or instrumental music, solo flute or string ensemble, hard rock or a waltz? In spite of such variables, there is a body of experience from which we can extract helpful information. Figure 22-3 is an approximation rather than a true optimum, but following it will result in reasonable, usable conditions for many types of recording. In particular, the shaded area of Fig. 22-3 represents a compromise in studios used for both speech and music recording.

Diffusion

Diffusion contributes a feeling of spaciousness through the spatial multiplicity of room reflections. Diffusion can also help control the effect of resonances. Splaying walls and the use of geometrical protuberances have a modest diffusing effect. Distributing the absorbing material is a useful means of not only achieving some diffusion but increasing the absorbing efficiency as well.

Modular diffraction grating diffusing elements provide efficient diffusion and are easily installed in small studios. For example, 2- × 4-ft modular units offer diffusion and

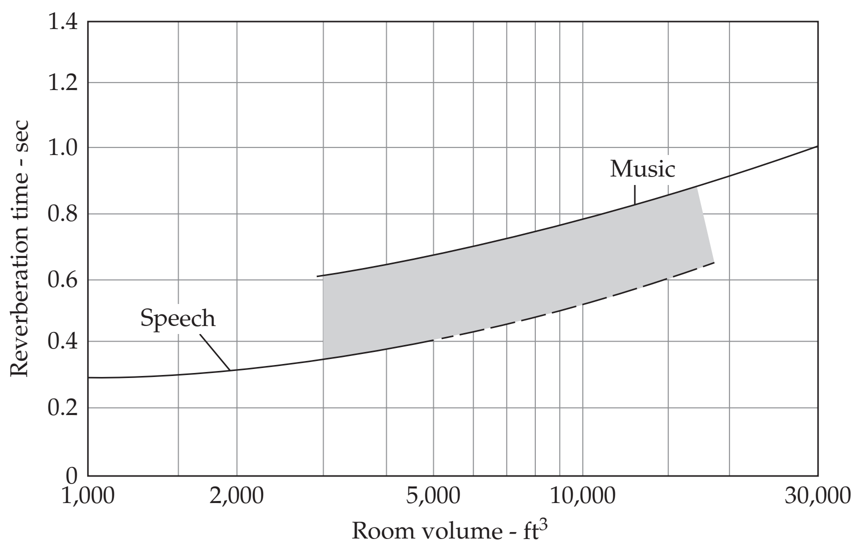


FIGURE 22-3 Suggested reverberation times for recording studios. The shaded area is a compromise region for studios in which both music and speech are recorded.

broadband absorption (e.g., 0.82 coefficient at 100 Hz), all within a 2-in thickness. In practice, it would be difficult to provide too much diffusion in a studio room.

Noise

A small studio is subject to the same rules of acoustical isolation as any other room. Walls, floors, and ceilings must be constructed for high transmission loss and decoupling from external noise and vibration sources. This helps ensure the low ambient noise levels required for good recording quality. Likewise, it isolates neighboring spaces from the potentially loud music levels present in the studio. The partition between the studio and the control room is particularly critical in a recording studio. Among other functions, the partition must provide good isolation so that engineers in the control room will only hear sound from the monitor loudspeakers, that is, what is actually being recorded, and will not be misled by other live sound leaking through the partition.

Small Studio Design Example

The following is an example of the methodology that can be used to design the principal acoustical aspects of a small studio. Consider a room measuring 25 × 16 × 10 ft. Let us assume that the primary interest is conventional music produced by small vocal and instrumental ensembles. For a volume of 4,000 ft^3, we may select a reverberation time of about 0.6 sec as our design goal. The proportions are not optimum for modal response. A more suitable ratio would call for a room length of 23.3 ft instead of the existing 25 ft. We could shorten the room, perhaps using that volume for an acoustical device such as a large bass trap, but it is never advisable to sacrifice floor space unless absolutely necessary. And in this case, the potential axial-mode problems stemming from a 25-ft length are acceptable and can be addressed by good design. In many practical studio designs, compromises such as this are inevitable, and not always worth the cost to remedy. Indeed, it is the designer's job, from an acoustical and budgetary standpoint, to understand when a compromise is acceptable and when it is not.

We can consider many different materials and structures capable of giving the low-frequency absorption needed in a small studio. Because it is primarily a music studio, we may consider paneling, and because we are seeking good diffusion and brilliance for a recording studio, we might consider polycylindrical module diffusers.

Absorption Design Goal

We begin the design process with some general thinking about absorption. A reverberation time of 0.6 sec is desired. What absorption is required to achieve this? For a studio of 4,000 ft^3 volume, we compute this to be 327 sabins. Moreover, for consistent reverberation over the audio band, we will need 327 sabins at each of six frequency points. Polycylindrical modules can provide a large part of the low-frequency absorption and provide some needed diffusion as well. How many polycylindrical modules are needed to achieve this? In the 100- to 300-Hz region, the absorption coefficient is 0.3 to 0.4; we choose an average of 0.35 as we will use polycylindrical modules of different sizes. To estimate the area of modules required, we use the relationship:

$$A = S\alpha \tag{22-1}$$

where A = absorption units, sabins
S = surface area, ft^2
α = absorption coefficient

Thus, we observe that the required surface area $S = 327/0.35 = 934$ ft^2. For broadband absorption, the polycylindrical modules should be of different chord lengths; we shall select three of the four shown in Fig. 12-26: modules A, B, and D.

Not all of the low-frequency absorption will come from polycylindrical modules. The ceiling alone has 400 ft^2, so it would appear that we have enough surface area to accommodate 934 ft^2 of polycylindrical modules. For high-frequency absorption let us consider common acoustical tile. Ideally, we should have some of each acoustical material applied to every surface of the studio. Practically, we cannot achieve this ideal. For one thing, the only practical acoustical material for the floor is carpet and having the entire floor covered with carpet would give more high-frequency absorption than required, as detailed below.

A floor area of 400 ft^2 with a carpet absorption coefficient of 0.7 yields 280 sabins in the high-frequency region for the carpet alone, and the polycylindrical modules have coefficients of 0.2 in the high-frequency region also. Even if we use only 500 ft^2 of modules, they would contribute another 100 sabins. Carpet plus polycylindrical modules would give 280 + 100 = 380 sabins and we only want 327 sabins. So, carpet, at least over the entire floor, cannot be used. One option is to carpet a smaller portion of the floor; having a reflective floor area, and an absorptive floor area, might be desirable because it helps create different acoustical environments in the studio. However, applied carpet makes those environments more permanent whereas a rug could be used as needed. For this studio design, the necessary added high-frequency absorption can be obtained with acoustical tile, as described below.

Proposed Room Treatment

The proposed room treatment is shown in Fig. 22-4. On the north and south walls, the cross section of the ceiling-mounted polycylindrical modules is visible. A large polycylindrical module A is in the center, flanked with a module D on each side. Between the D modules and the outside B modules is space for lighting fixtures running the length of the room. This polycylindrical module treatment on the ceiling opposes the hard, reflective floor covered with vinyl tile and should prevent flutter echo in the vertical dimension.

On the west wall, two A polycylindrical modules, three B modules, and one D module are arranged vertically, which places them perpendicular to the ceiling-mounted modules as they should be. Four vertical tiers of 3/4-in acoustical tile are worked into this west wall; two of the tiers are placed at the corners where absorbers are especially effective as described in the following text. This breaks up the west wall so that we would expect no flutter echo between it and the east wall. Untreated walls are surfaced with plaster.

The south wall has a single A and a single B polycylindrical module arranged horizontally so that the axis of each set of modules is perpendicular to the other two sets. Two horizontal tiers of acoustical tile are placed between 4 and 6 ft from the floor (which is head height for a standing person). High-frequency absorbers are placed at this height on all walls, except the west wall; given this arrangement, a performer would ideally face the east wall.

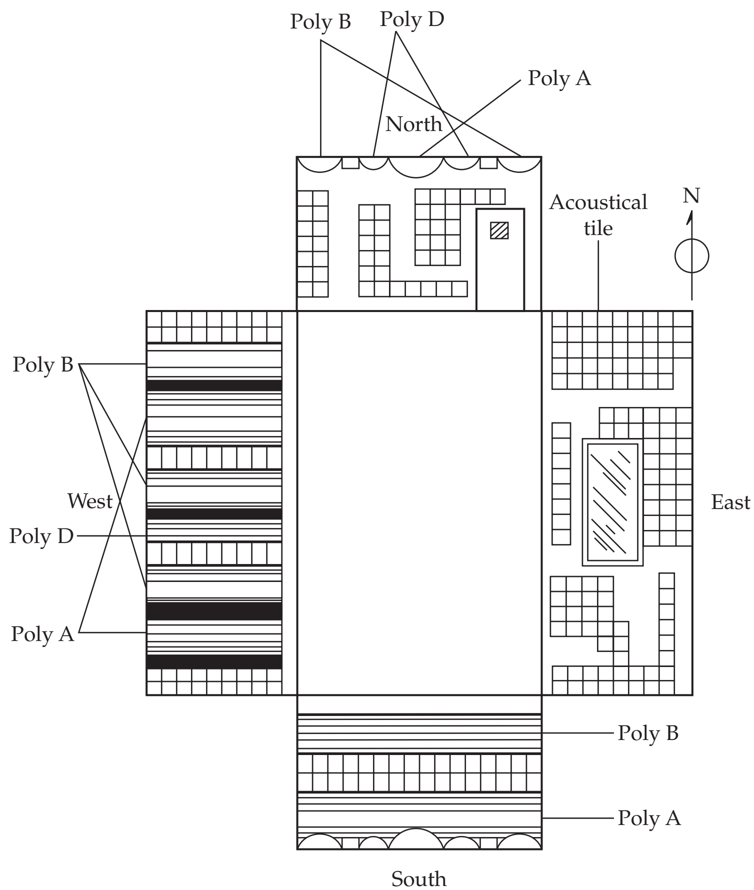


FIGURE 22-4 Fold-out plan of a music studio measuring 25 × 16 × 10 ft. Polycylindrical modules (Poly A, B, C, and D) and acoustical tiles are used for absorption. The axes of the polycylindrical modules on the walls and ceiling are mutually perpendicular.

With the polycylindrical modules as selected, and using the appropriate absorption coefficients, the absorption at the six standard frequencies was calculated, as shown in Table 22-3. The absorption attributed to the polycylindrical modules was calculated in two ways: with the modules empty, and with the modules stuffed with glass fiber. The empty condition shows a slight deficiency of absorption at low frequencies and considerable extra absorption at middle frequencies. Stuffing the polycylindrical modules with glass fiber will satisfactorily increase absorption at low frequencies; this condition is selected. Using the calculations of Table 22-3, the absorption contributed by the stuffed polycylindrical modules is plotted in Fig. 22-5.

The 3/4-in acoustical tile offers an absorption coefficient of 0.73 at high frequencies, and following the same procedure as with the polycylindrical modules, we find that we need 200/0.73 = 274 ft^2 of acoustical tile. Additional tile is then added to the room, distributing it around the studio in patches, until 274 ft^2 is obtained. Using the calculations

Size: 25 × 16 × 10 ft
Volume: 4,000 ft^3
Floor: vinyl tile
Walls: plaster

Material		**Area (ft^2)**	**125 Hz**		**250 Hz**		**500 Hz**		**1 kHz**		**2 kHz**		**4 kHz**	
			α	Sα	α	Sα	α	Sα	α	Sα	α	Sα	α	Sα
EMPTY	Poly A	232	0.41	95.1	0.40	92.8	0.33	76.6	0.25	58.0	0.20	46.4	0.22	51.0
	Poly B	271	0.37	100.3	0.35	94.9	0.32	86.7	0.26	75.9	0.22	59.5	0.22	59.6
	Poly D	114	0.25	28.5	0.30	34.5	0.33	37.6	0.22	25.1	0.20	22.8	0.21	23.9
	Total			223.9		222.2		200.9		159.0		128.8		134.5
FILLED	Poly A	232	0.45	104.4	0.57	132.2	0.38	88.2	0.25	58.0	0.20	46.4	0.22	51.0
	Poly B	271	0.43	116.5	0.55	149.1	0.41	111.1	0.28	75.9	0.22	59.6	0.22	59.6
	Poly D	114	0.30	34.2	0.42	47.9	0.35	39.9	0.23	26.2	0.19	21.7	0.20	22.6
	Total			255.1		329.2		239.2		160.1		127.7		133.4
Acoustical tile, 3/4"		274	0.09	24.7	0.26	76.7	0.78	213.7	0.84	230.2	0.73	200.0	0.64	175.4
Absorption with polys empty, sabins				248.6		296.9		414.6		389.2		328.8		309.9
Absorption with polys filled, sabins				279.8		405.9		452.9		390.3		327.7		308.8
Reverberation time with polys empty, sec				0.79		0.68		0.47		0.50		0.60		0.53
Reverberation time with polys filled, sec				0.70		0.48		0.43		0.50		0.80		0.63

Table 22-3 Room Conditions and Absorption and Reverberation-Time Calculations for a Small Studio

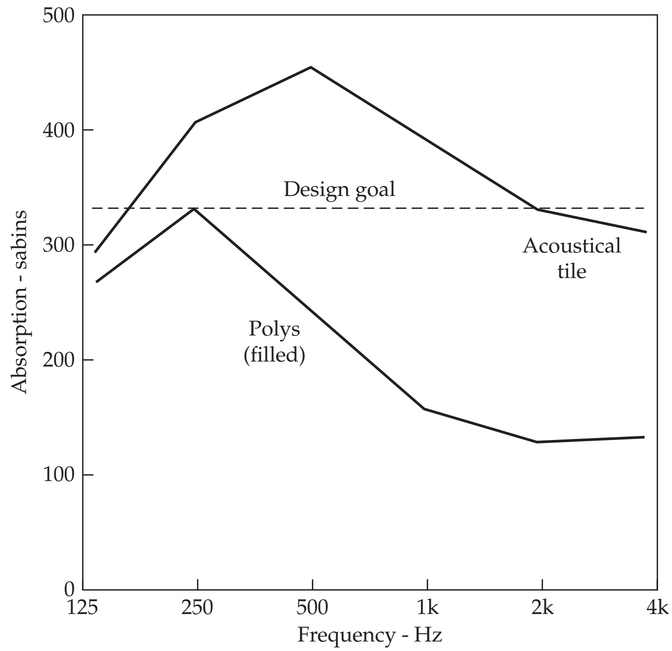


FIGURE 22-5 Relative contributions of the polycylindrical modules (stuffed with glass fiber) and acoustical tile to the overall absorption in the music studio of Fig. 22-4.

of Table 22-3, the absorption attributed to acoustical tile is plotted in Fig. 22-5 along with the absorption of the polycylindrical modules.

We must always be aware of the lack of precision in absorption calculations such as these. The curves appear to be precise, but care must be taken to avoid frequency-response anomalies due to axial modes. And, for example, additional diffusion may be required. In short, successful acoustical design requires more than calculation; it also requires experience and a good ear.

A final and partly unrelated note: when designing a small studio, it is easy to budget for major construction such as wall and floor/ceiling constructions, but overlook details such as sound lock treatment, doors and their sealing, cable runs through walls, lighting fixtures, observation windows, and other features common to studios. All of these can create serious acoustical problems if not handled properly and must be addressed in the acoustical design, and budget.

Previously, we have considered reverberation and how to compute and measure it (Chap. 11), dissipative and tuned absorbers (Chap. 12), the complications of room resonances (Chap. 13), the need for diffusion and how to obtain it (Chaps. 9 and 14), noise control (Chaps. 16 through 18), and air-handling equipment (Chap. 19). All of these are also integral parts of any studio design, big or small.

Key Points

- The noise and vibration levels in a commercial recording studio must be below those of almost any other work space, and below the levels tolerable in a home studio.
- Room modes dominate the low-frequency region while specular reflections determine high-frequency room characteristics. The 300- to 500-Hz region is a transition zone. Each analysis has limitations but can provide useful design guidance.
- In small rooms, it is particularly important to choose a favorable room dimension ratio to distribute axial mode frequencies.
- Frequency-response anomalies encountered in studios having volumes less than 1,500 ft^3 are sometimes so severe as to make small studio rooms impractical for recording music.
- In a small room, the assumptions underlying the Sabine equation are not met; as a result, the equation's predictions, while useful, may not be accurate.
- There is no specific optimum reverberation time for any room because many other factors are involved. However, reasonable assumptions can result in reverberation times that are usable for many types of recording.
- Diffusion contributes a feeling of spaciousness through the spatial multiplicity of room reflections, and can also help control the effect of resonances. In practice, it would be difficult to provide too much diffusion in a studio room.
- In many practical studio designs, compromises are inevitable, and not always worth the cost to remedy. Indeed, it is the designer's job, from an acoustical and budgetary standpoint, to understand when a compromise is acceptable and when it is not.
- One must always be aware of the lack of precision in absorption calculations. The curves appear to be precise, but care must be taken to avoid frequency-response anomalies due to axial modes and other conditions.

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CHAPTER 23

Acoustics of Large Recording Studios

Large recording studios designed for the recording of popular music are the flagships of the professional music recording industry. Large studios are often very ambitious architectural and acoustical projects and can cost several million dollars before equipment costs are considered. To facilitate the work done there, large studios usually contain several very specific architectural features such as drum and vocal booths. A studio such as this can be used to record a wide variety of music such as pop, rock, hip-hop, rap, R&B, country, soul, and jazz. Classical music, particularly music with large performing ensembles, usually requires a larger acoustical space and one with a longer reverberation time, such as a concert hall.

Any recording studio, large or small, is critical in terms of acoustical requirements. This is the place where music performances are recorded for preservation. Along with the music performance, anomalies and defects in the studio's acoustics are also unfortunately captured and replayed countless times. Thus the acoustical treatment of a studio must be carefully designed and constructed so that the acoustics are pleasing and complementary to the music, and also relatively neutral so the acoustics do not intrude on the music.

The reverberation time in a large studio intended for recording popular music should be relatively short. For example, a broadband reverberation time of less than 0.4 sec is typical. The short reverberation time yields relatively dead tracks particularly when close micing is used. It also provides better isolation between simultaneously recorded tracks. This makes it easy to add digital reverberation and other effects to individual tracks during the mixdown. Too little absorption simply makes the studio too live. However, too much absorption is problematic because musicians need some reverberation for comfortable performance. Proper absorption is thus a key element of the studio design.

The high sound-pressure levels typical in a large studio (and its control room) must be contained. High levels in the studio that are audible in the control room can adversely affect decisions made during tracking; clearly, ideally, only the control room monitors should be heard. Conversely, loud monitor levels in the control room could be picked up by microphones in the studio. Thus, the wall between the studio and control room, as well as walls between different recording spaces in the studio, must provide good isolation. Although not as critical, some isolation is needed between the main studio space and other rooms such as drum and vocal booths.

In addition, precautions must be taken to avoid creating noise intrusion issues for those outside the studio complex. Isolating the high-level sounds of music is a very real technical challenge. If the sound levels in the studio or control room reach 120 decibels (near the threshold of pain), a 50-dB wall is needed to reduce music levels to a tolerable 70 dB. In many cases, a design goal of less than 70 dB would be required, necessitating walls with even greater transmission loss. For this reason, in the proposed studio design presented here, sound isolation is a high priority. To achieve a high degree of sound isolation, many professional recording studios utilize massive masonry construction. However, whereas frame walls would provide diaphragmatic absorption at low frequencies, masonry walls do not. Thus it is important for the room to provide adequate absorption of low-frequencies using bass traps or other means.

Design Criteria of a Large Studio

For this studio design example, we shall assume that the space to be used for the studio is on the ground floor of an existing concrete building. Furthermore, the studio is located in a corner of the building; it has two exterior walls and two interior walls, and there are neighbors on the same ground floor and on the floor above. The isolation criteria of the floor and walls are carefully considered; to help maintain isolation between rooms, a sound lock is included. In addition, treatment is designed to provide a suitable diffusion and reverberation time.

The control room in this facility is acknowledged only by placement of an observation window in the south wall. Control room design is a separate project and will be carried no further in this chapter. Control rooms are considered in Chap. 24. To minimize the passage of sound between rooms and to maintain low background noise levels, noise control and heating, ventilating, and air-conditioning (HVAC) design is very important. These topics are discussed in Chaps. 16 through 19.

Floor Plan

The floor plan for this studio facility is shown in Fig. 23-1; the overall dimensions are 37 ft, 3 in × 32 ft, 3 in × 16 ft. The facility includes a drum booth, a vocal isolation booth, an equipment storage room, a baffle storage area, and a sound lock that opens into the control room as well as the exterior. A control room window is included in the south wall. Approximately half of the studio floor area is carpeted, with wood parquet on the other half. Different floor coverings help provide different acoustical characteristics and thus variability when recording. However, because the floor coverings are permanent, some flexibility is lost. The volume of the studio is relatively large at about 15,220 ft^3.

Wall Sections

Figure 23-2 locates sections D-D, E-E, F-F, and G-G (sections A-A, B-B, and C-C are reserved for drum-booth walls, which are discussed later). Sections D-D and G-G are exterior walls; E-E and F-F are interior walls. The inside walls for the drum booth, vocal booth, and equipment room are not indicated pictorially but comprise simple single layers of 5/8-in gypsum board on each side of metal channels. Using Fig. 23-2 as a guide, each of the wall sections can be examined individually.

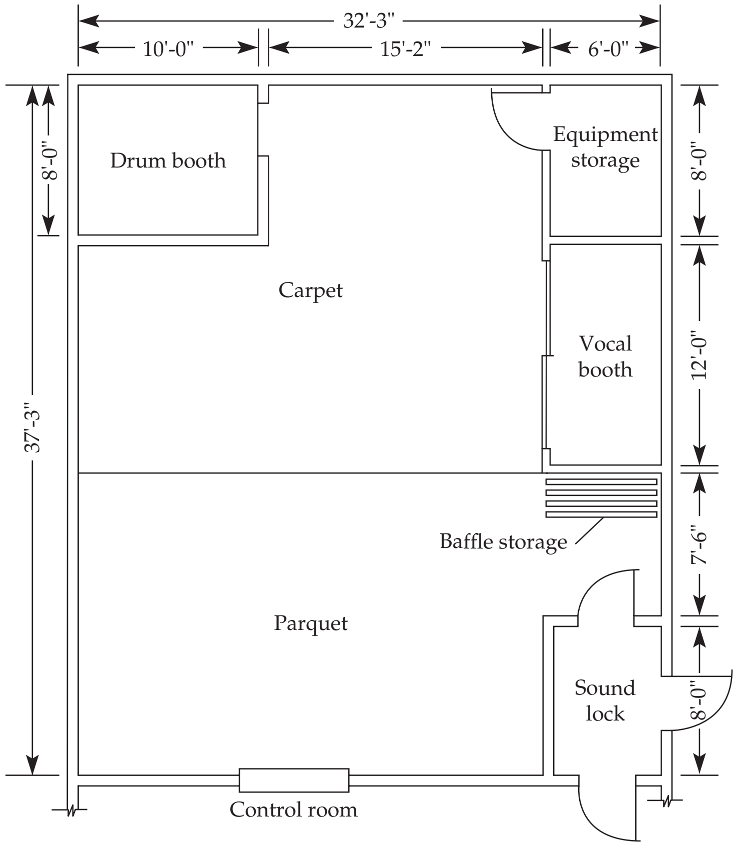

Figure 23-1 Floor plan of a large multitrack music recording studio.

Section D-D

As noted, isolation is a critical element in this studio design. Thus, a floating floor is incorporated in the design. Sections D-D and E-E are both shown in Fig. 23-3A. The floor (5) of 4-in concrete, common to both, is floated on pads of compressed glass fiber or neoprene. The concrete is not poured until the 3/4-in plywood form is in place and covered with a plastic membrane to prevent concrete from seeping through cracks and forming a solid bridge to the supporting structure. Glass-fiber perimeter boards separating the floating floor from the walls are also shielded by a plastic membrane. The floating concrete floor must be completely isolated from the structure to be effective. The pads must be selected and sufficient in number to bear the weight of the concrete floor as well as the floating walls, and be deflected about 15%, so that the full resiliency of the pads can be realized. However, for optimal performance, the floor's mass and static deflection of the pads must be carefully analyzed; in particular, the floor's resonant frequencies must be calculated. As a lower-cost alternative, a wood floating floor may be used. Floating floors are discussed further in Chap. 17.

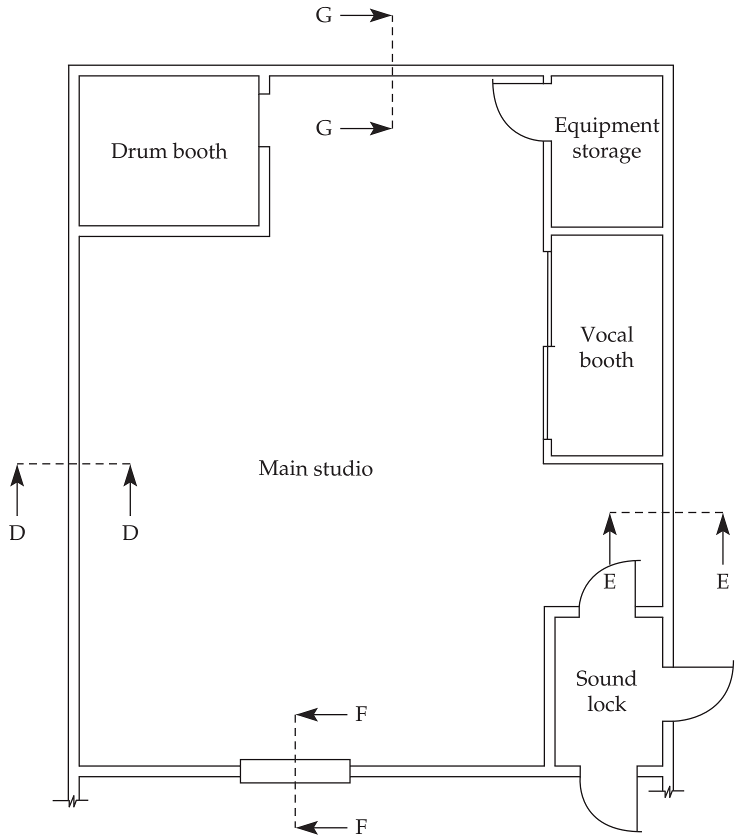

Figure 23-2 Locations of four sectional views of a large recording studio.

All of the interior walls of the recording studio are supported on the floating floor. The upper ceiling (1), hung from the structural building on resilient isolation hangers (6), is made of two 5/8-in gypsum-board layers separated by a septum of dense vinyl. This is used to increase the acoustical isolation to occupants on the second floor. The floating floor, the four walls, and this acoustical ceiling constitute a space within a space for attenuation of the music within.

The lower acoustical ceiling (2) uses a standard suspended T-frame ceiling with a 16-in airspace to conform to the standard C-400 (400 mm) mounting. The ceiling uses Tectum Lay-In ceiling panels measuring 1 × 24 × 24 in, laid in the T-frame with 6-in of glass-fiber backing resting on the top of the panels. The section also shows wall stabilizers (9) designed to isolate the walls from the structure. Tectum panels are manufactured by Armstrong World Industries.

A spacing wall (3) is built alongside the concrete west wall of the building with a 2-in spacing. This spacing is loosely filled with 1-1/2-in glass fiber. The layer of the new wall is 5/8-in gypsum board attached to steel studs and channels. The inner space of the wall is filled with 4 in of low-density building insulation, and the inner layer is

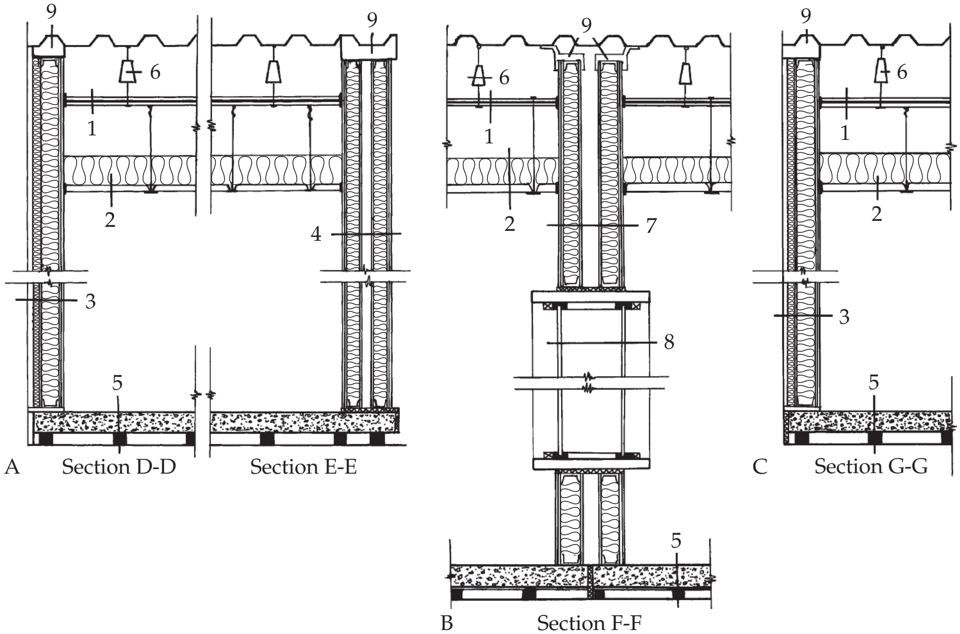

Figure 23-3 Wall sections of a large recording studio. (A) Sections D-D and E-E, exterior and interior walls. (B) Section F-F, interior wall between studio and control room. (C) Section G-G, exterior wall.

a double layer of 5/8-in gypsum board. As with any acoustically sensitive construction, all cracks and joints should be filled with acoustical sealant, the goal being to make the inner space hermetically independent. Cracks that allow passage of air will also allow sound to pass through the barrier, defeating the purpose of the carefully designed walls, floor, and ceiling. All metal wall runners should be set in acoustical sealant.

Resilient hangers (6), floating floor (5), and spacing wall (3) from the wall of the building are all employed to isolate the studio space from the structure. The reason is that the concrete structure is an excellent transmitter of noises both into and out of the studio. The primary justification for this expense is to isolate neighboring facilities from the studio.

Section E-E

The only feature of section E-E (see Fig. 23-3A) not discussed in terms of section D-D is the double wall (4); it is an inner wall that runs the length of the studio. It is a double wall spaced 2 in. Each leaf of the double wall comprises standard metal studs and channels with a single 5/8-in gypsum-board layer on the inside and a double layer of the same on the outside. Loosely stuffed 4-in glass fiber is placed inside the walls. Although not specified, it would be advantageous to increase the transmission loss by a few decibels by mounting the outer layer of 5/8-in gypsum board facing the other occupants on resilient channels.

Sections F-F and G-G

Section F-F is shown in Fig. 23-3B. This shows the double wall (7) that separates the studio from the control room. It must provide high transmission loss so that sound levels from the studio do not affect monitoring in the control room, and conversely so that monitoring levels do not leak into the open microphones in the studio. The double wall is the same as wall (4) shown in section E-E, but with 6-in spacing.

The wall incorporates a double-pane glass observation window (8). Because most windows usually have lower transmission loss than the surrounding walls, care must be taken in the window design so that it does not compromise the isolation integrity of the wall. The window uses two heavy (or laminated) panes of different thicknesses, each isolated from the frame by rubber edging. The frame, in turn, is isolated from the wall structure by pads of compressed glass fiber or neoprene. Each pane is mounted in a separate wall built on separated slabs. In some designs, a third interior load-bearing wall is used; alternatively pipe columns can be used for support. If a third glass pane is added, this can mildly improve high-frequency isolation; however, care must be taken to ensure that low-frequency isolation is not diminished. Window design is discussed further in Chap. 18.

The surfaces inside the two window panes are lined with open-cell foam or glass-fiber boards. In some designs, this cavity also functions as a Helmholtz resonator. If desired, the window panes can be angled downward; this may usefully deflect sound downward and prevent an unwanted reflection in the room. Moreover, the angle can reduce light reflections. In addition, the angle moves the upper wall sections farther apart; among other things, this can provide room for soffit-mounted monitor loudspeakers. When designing soffit mounts for loudspeakers, care must be taken to ensure that the monitors are properly isolated, and sound will not enter the studio.

Section G-G is shown in Fig. 23-3C. All features of this section have been previously discussed.

Studio Treatment

The north wall of the studio is covered with 2-in Tectum panels with standard C-40 (40 mm) mounting, as indicated in Fig. 23-4. With C-40 mounting (not to be confused with C-400 mounting), the panels are furred out from the wall on 2- × 2-in strips (net about 1-1/2 in) on 24-in centers. This modest spacing from the wall improves the absorption of the panel so that it is highly absorptive above 250 Hz.

The south wall contains the window to the control room, as well as a functional artistic touch. A recording studio can be excellent acoustically and very poor aesthetically. Some might associate drabness with poor sound quality. Musicians are aesthetes, and it is a fair guess that they, their guests, and clients would appreciate one wall of the studio devoted to a decorative absorptive piece. For this reason, the south wall of the studio (see Fig. 23-4) would be an ideal place for decorative acoustic panels. Numerous manufacturers offer customizable panels; custom images, colors, sizes, shapes, and finishing are available. The noise reduction coefficient (NRC) is nominally 0.9. Clearly, if budget or taste does not allow a decorative commercial treatment, low-cost absorption can be provided with glass-fiber panels such as described in Chap. 12.

The east wall is broken up by many doors and other openings. The wall can only accommodate an absorptive panel measuring 4 to 6 ft in width running almost the length of the studio. This is shown in Fig. 23-5.

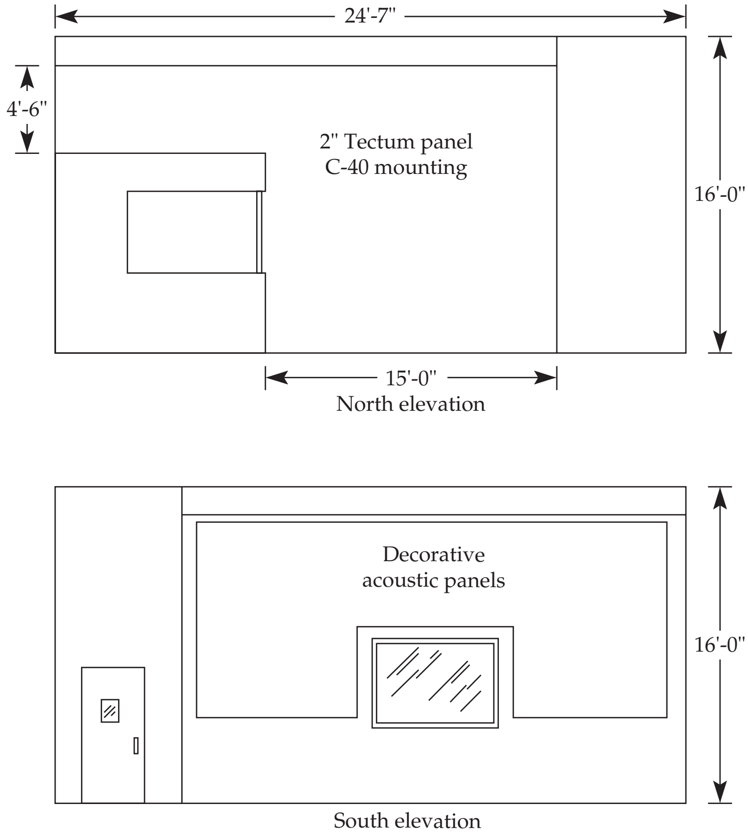

Figure 23-4 North and south interior walls of a large recording studio.

The west wall contains an array of doors, 10-ft high and 24-in wide, that cover 24 ft of the wall (see Fig. 23-5). Each door is hinged, and is absorbent on its inner side and reflective on the outer side. When each door is open, it not only exposes its own absorbent back side but also reveals another absorbent area the size of the door. When all doors are swung open, the entire 10- × 24-ft area is absorbent.

The ultimate purpose of the doors on the west wall is not to alter the reverberation time because, as we will see, this 240-ft^2 area has only a minor effect on the overall reverberation time. Rather, the doors make available reflective and absorbent areas so that musicians can set up near one or the other to achieve the acoustical ambience they desire. Note that the Tectum panel wall on the north end of the studio is absorbent, but this area of swinging panels is the only other absorbent area close to musicians.

The floor is about half carpet and half parquet wood (see Fig. 23-1). A curved line of demarcation between the two areas would look better, but the style should be drawn by the occupants-to-be. The suspended ceiling is a very absorbent surface, even if the space behind is shared with air-conditioning ducts and electrical equipment.

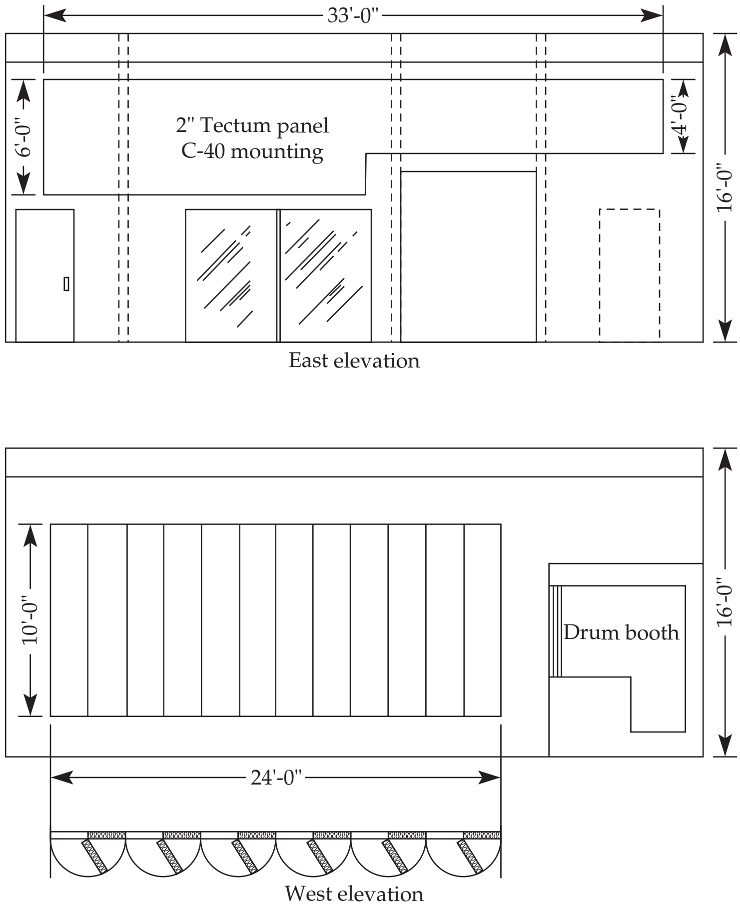

Figure 23-5 East and west interior walls of a large recording studio.

Drum Booth

The purpose of a drum booth is to reduce the high sound levels from the drum kit at the positions of the other instruments (and microphones) in the studio and to provide a relatively dead local environment for recording drums. An isometric sketch of the proposed drum booth is shown in Fig. 23-6 and details of its construction are given in Fig. 23-7.

The drum booth measures 8 × 10 × 11 ft with openings on the south and east sides to give a clear view of the studio as well as the control room window. The booth features a wood floor of 1-1/2-in tongue-and-groove decking on top of a 2- × 8-in structure deadened by loading with sand. The entire floor is isolated from the building structure by pads of compressed glass fiber or neoprene such as used to float the floor. The floor must be constructed to be solid and essentially nonresonant. If desired, for improved isolation, double-pane glass windows can be placed in some portions of the openings.

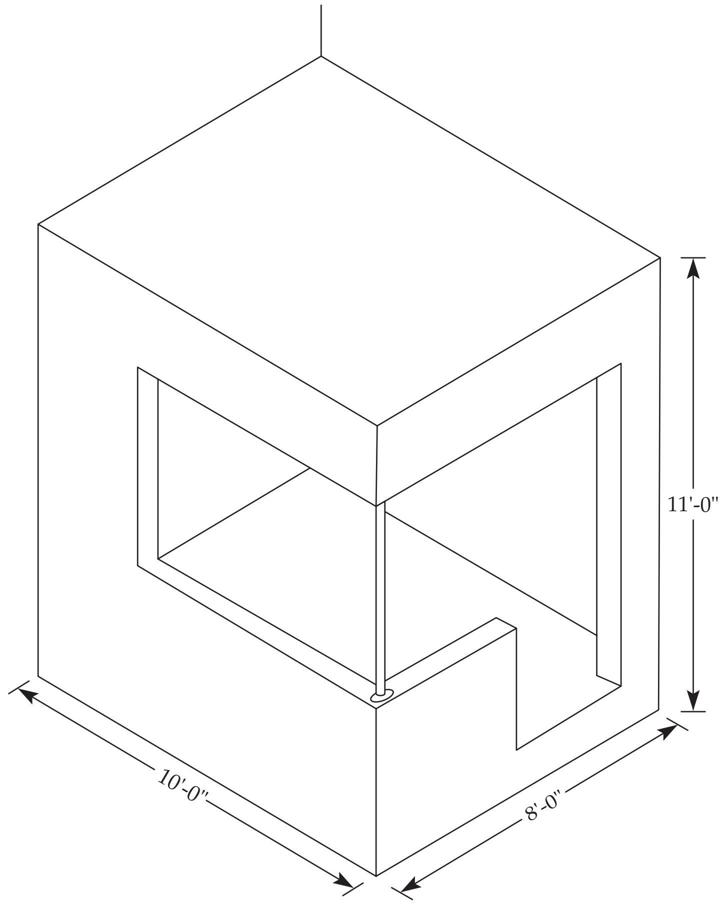

Figure 23-6 An isometric sketch of a drum booth.

For isolation purposes, a solid door or sliding glass door could be installed. Alternatively, if isolation is not required, the entryway to the booth can remain open.

To deaden the ambient sound field in the drum booth, the ceiling is made highly absorbent by packing the 2- × 12-in frame with glass-fiber building insulation. The north and west walls of the booth are reflecting surfaces of 3/4-in plywood, but made into a tuned absorptive structure with 1/2-in holes spaced 7-in apart. These two walls have a tuned absorption peaking in the region of 80 Hz, but little absorption above 150 Hz. This absorption helps deaden the kick drum, yet provides the drummer with adequate personal acoustical return.

Vocal Booth

The vocal booth is a space measuring 6 × 12-ft with a sliding glass door opening to the studio which provides entry as well as an excellent view both ways. This sliding glass door, when closed, will provide about 20 to 25 dB of isolation in addition to the

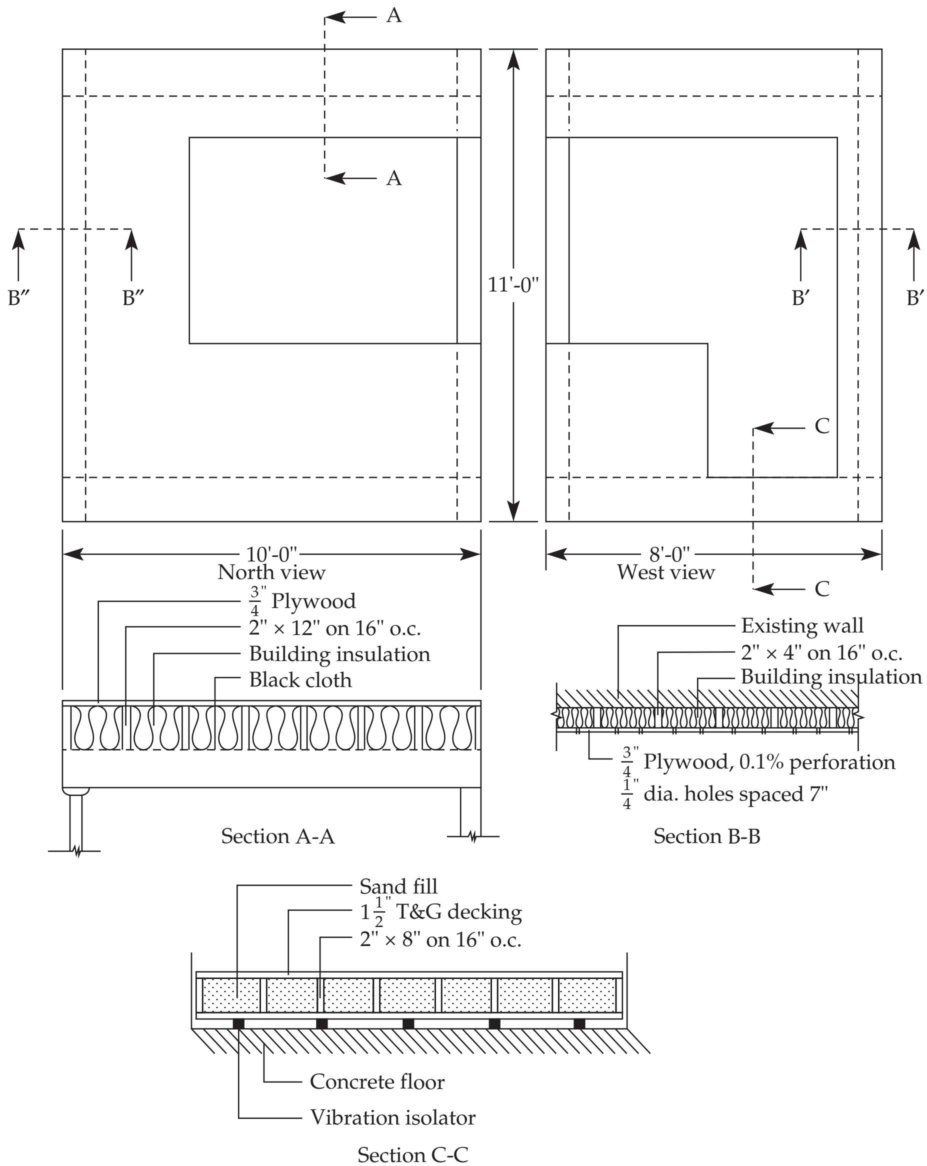

FIGURE 23-7 Drum booth design details.

inverse-square (or "6 dB/distance-double") law. The interior of the vocal booth should be treated with absorptive material such as glass-fiber panels. In addition, treatment should be added to control axial modes. Absorption such as from bass traps in the corners would be best for this. Isolation booths are discussed in more detail in Chap. 25.

Sound-Lock Corridor

The studio's floor plan (see Fig. 23-1) shows a sound-lock corridor. The inner door leads to the studio area, a second door leads to the control room, and a third door leads to the exterior. The purpose of a sound lock is clear; some isolation is provided even when one door is opened. When both doors are closed, the sound lock provides isolation well above that of a single door. To provide this additional isolation, the interior of the sound lock should be absorptive. Although studio wiring is outside the scope of this book, it may be advisable to place a microphone/headphone panel in a wall of the sound lock; this corridor can provide a uniquely dead recording space. Sound locks are discussed further in Chap. 18.

Reverberation Time

Multichannel recording demands close microphone placement or direct feeds to achieve sufficient channel separation. To assist close micing, a studio should present a short reverberation time. Moreover, reverberation-time calculations are of interest to balance the absorption of different areas and materials. In this recording studio design almost every available ceiling and wall area have been covered with absorbent material. In other words, it has been made quite dead for a room of 15,220-ft^3 volume. Reverberation-time calculations also allow us to inspect the distribution of absorption about the room. Table 23-1 shows all the reverberation-time calculations for each absorbent unit contributing to the overall acoustics of the space.

For each of the six standard frequencies, the absorption coefficient α for a given material is multiplied by the area S of that material to obtain the $S\alpha$ product in sabins. Summing all these unit absorbances in sabins for a given frequency gives the total absorption A for that frequency from which the reverberation time may be estimated from the equation:

$$\mathrm{RT}_{60} = \frac{0.049V}{A} \tag{23-1}$$

where RT_{60} = reverberation time, sec
V = volume of room, ft^3
$A = S\alpha$ = total absorption of room, sabins

Given that V = 15,220 ft^3, and using the surface areas S and absorption coefficients α of the surface treatments, we can use the equation to calculate RT_{60} at different frequencies. The calculated reverberation times have been plotted in Fig. 23-8 for the swinging panels both open and closed. These panels will probably never be moved for the purpose of adjusting reverberation time. It is not possible to sense the difference between the 500-Hz reverberation times of 0.32 and 0.35 sec brought about by the swinging panels. Rather, the adjustment of these panels is primarily to vary local acoustics near the panels to suit individual musicians. A musician can easily sense playing alongside a hard wall versus a soft wall.

Material	Area (ft^2)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	Sα	α	Sα	α	Sα	α	Sα	α	Sα	α	Sα
Suspended ceiling: C-400 (16 in) Tectum ceiling tile 1- × 24- × 24-in lay-in 6-in glass-fiber backing	956	1.01	965	0.89	850	1.06	1,013	0.97	927	0.93	897	1.13	1,680
East-wall panel: Tectum wall panel C-40 mounting, 1-1/2 in	164	0.42	69	0.89	146	1.19	195	0.85	139	1.08	177	0.94	154
West-wall adjustable panels:													
Fully opened	240	1.0	240	1.0	240	1.0	240	1.0	240	1.0	240	1.0	240
Fully closed	0		0		0		0		0		0		0
South-wall panel: decorative acoustic panel	210	0.16	34	0.47	99	1.10	231	1.14	239	1.05	221	1.04	218
North wall: fully covered with Tectum wall panel C-40 mounting, 1-1/2 in	252	0.42	106	0.89	224	1.19	300	0.85	357	1.08	272	0.94	237
Drum booth ceiling	80	1.0	80	1.0	80	1.0	80	1.0	80	1.0	80	1.0	80
North and west walls: 0.1% perforated 4 in deep	128	0.8	102	0.3	38	0.2	26	0.15	19	0.15	19	0.1	13
Floor, heavy carpet	448	0.08	36	0.24	108	0.57	255	0.69	309	0.71	318	0.73	327
Floor, parquet	465	0.02	9	0.03	14	0.03	14	0.03	14	0.03	14	0.02	14
Total absorption, sabins			1,641		1,799		2,354		2,324		2,238		2,365
Reverberation time with west doors open, sec			0.45		0.41		0.32		0.32		0.33		0.32
Reverberation time with west doors closed, sec			0.53		0.48		0.35		0.36		0.37		0.35

Table 23-1 Absorption and Reverberation-Time Calculations for a Large Recording Studio

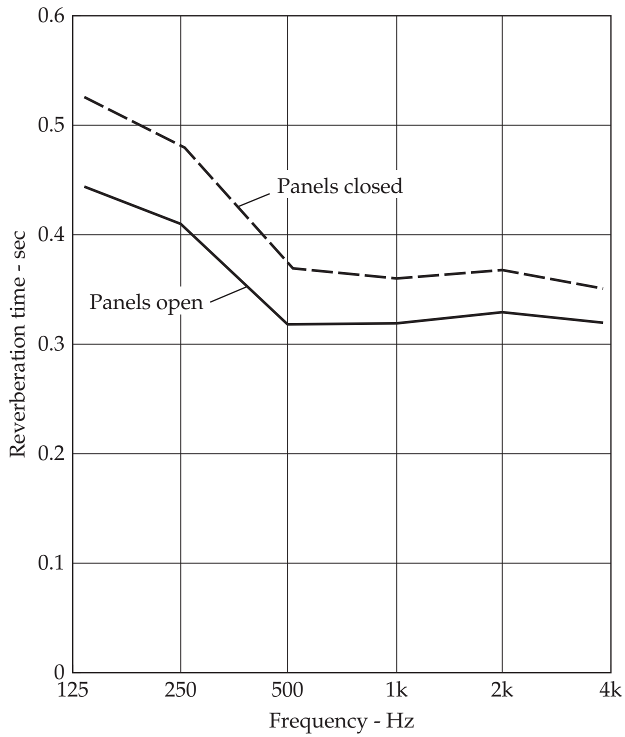

FIGURE 23-8 A reverberation-time plot of a large recording studio.

Key Points

- The reverberation time in a large studio intended for recording popular music should be relatively short; too little absorption makes the studio too live. However, too much absorption is problematic because musicians need some reverberation for comfortable performance.
- High sound levels in the studio that are audible in the control room can adversely affect tracking; ideally, only the control room monitors should be heard. Conversely, loud monitor levels in the control room could be picked up by microphones in the studio. Thus, the wall between the studio and control room must provide good isolation.
- The floor plan for this studio includes a drum booth, vocal isolation booth, equipment storage room, baffle storage area, and sound lock; the volume is about 15,220 ft^3. The control room design is not considered here.
- A floating floor comprises a concrete slab floated on pads of compressed glass fiber or neoprene. The pads must bear the weight of the concrete floor as well as the walls which are supported on the floating floor. The upper ceiling is hung from the structural building on resilient isolation hangers.

- The floating floor, the perimeter walls, and the acoustical ceiling constitute a space within a space for attenuation of the music within.
- All cracks and joints in partitions should be filled with acoustical sealant. Cracks that allow passage of air will allow sound to pass through the barrier, defeating the isolation goals.
- The control-room window uses two panes of glass, each isolated from the frame by rubber edging. The frame is isolated from the wall structure by pads of compressed glass fiber or neoprene. Each pane is mounted in a separate wall built on separated slabs.
- A drum booth is used to reduce the high sound levels from the drum kit at the positions of the other instruments (and microphones) in the studio and to provide a relatively dead local environment for recording drums.
- The floor of the drum booth is isolated from the building structure by pads of compressed glass fiber or neoprene; the floor must be solid and essentially nonresonant.
- To deaden the ambient sound field in the drum booth, the ceiling of the booth is made highly absorbent. Walls of the booth can be made into a tuned absorptive structure peaking in the region of 80 Hz.
- A sound lock provides some isolation even when one door is opened. When both doors are closed, the sound lock provides isolation well above that of a single door. The interior of the sound lock should be absorptive.
- In this studio design, swinging panels provide only slight variability in overall reverberation time. However, these panels can be used to adjust local acoustics near the panels to suit individual musicians.

CHAPTER 24

Acoustics of Control Rooms

The acoustical design of professional control rooms is highly specialized and unique in the field of architectural acoustics. Contemporary control-room designs can provide outstanding acoustical performance. A control room has the very specific purpose of providing high-quality loudspeaker playback to a single listening (mixing) position. This is conceptually similar to a high-end home listening room, but with one important difference. In a listening room, the goal is primarily to provide a pleasurable listening experience; while accurate playback is desired, it is secondary to enjoyment. A professional control room is a work environment where playback accuracy is critical. The mixing engineer bases all recording and mixing decisions on the sound delivered by the monitor loudspeakers via the room acoustics. If that sound is not accurate, recording decisions will be flawed. For example, if a control room delivers a boosted bass response at the mixing position, the engineer will be misled by that, and mistakenly compensate for it by decreasing bass. As a result, the recordings mixed in the room will have a bass deficiency.

This chapter focuses on the acoustics of control rooms, that is, the room in recording studios where sound is played back and mixed. In this room, the acoustics of the loudspeaker/room interface are critical. Assisting the design goal of accurate sound is the fact that the location of the listener with respect to the monitor loudspeakers is fixed and known. The previous chapter discussed the acoustics of the studio room in a recording studio, that is, the room where music is performed and recorded via microphones.

Initial Time-Delay Gap

The initial time-delay gap (ITDG) is an acoustical metric used to examine early reflections in a sound field. ITDG is defined as the time between the arrival of the direct sound at a given seat, and the arrival of the first early reflections. In smaller, more intimate rooms, with reflective surfaces closer to the listener, ITDG is small. In larger rooms, ITDG is greater. In an extensive analysis of concert halls, Beranek noted that those halls rated the highest by qualified listeners had certain technical similarities, including similar ITDG values. Highly rated halls had a well-defined ITDG of about 20 msec. In inferior halls, the ITDG was flawed by uncontrolled reflections. The term was originally coined by Beranek to describe this characteristic of concert halls, but the principle can be applied to other kinds of spaces, including recording studios and control rooms.

The factors generating the initial time-delay gap of an untreated control room are illustrated in Fig. 24-1A. The direct sound travels from the loudspeaker to the mixing position. Soon after, reflections from surfaces near the loudspeaker in the front of the room arrive at the mixing position; they are the first reflections to arrive because their path lengths are shorter than other subsequent reflections. These early reflections create comb-filter effects in the frequency response at the mixing position. Later, the sound reflected from the floor, ceiling, or other objects arrives at the mixing position. This time gap between the arrival of the direct and reflected components is determined by the geometry of a particular room.

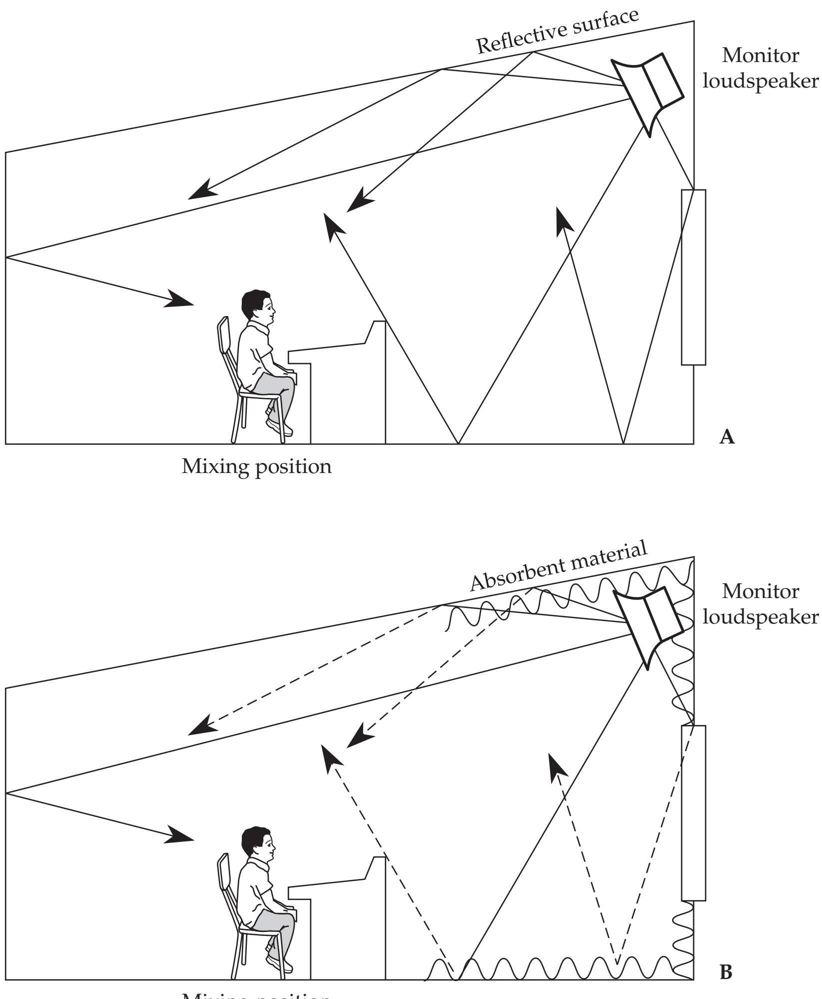

FIGURE 24-1 The initial time-delay gap describes the time between the arrival of direct sound at the mixing position and the arrival of early reflections. (A) In an untreated control room, the early reflections from the monitor loudspeakers in the front of the room can cause comb-filter effects at the mixing position. (B) In a treated room, with absorbent placed near the monitor loudspeakers, these early reflections are attenuated.

Don Davis and Chips Davis (not related) analyzed the effects of the initial time-delay gap of recording studios and control rooms with a measuring technique known as time-delay spectrometry. Time-delay spectrometry revealed the comb-filter effects associated with early reflections from surfaces near the monitor loudspeakers in control rooms (as well as from the mixing console surface). They observed that one way to minimize comb-filter effects was to eliminate or reduce the early reflections in the control room. This clarification of the problem led to the solution of placing absorbing material on the surfaces surrounding the monitor loudspeakers in the front part of the control room, as shown in Fig. 24-1B.

In simplified form, Fig. 24-2 shows the energy-time relationships essential for a properly designed and treated control room with absorbing material in the front of the room. At time = 0, the signal leaves the monitor loudspeaker. After an elapsed transit time, the direct sound reaches the mixing position. There follows some insignificant low-level clutter (to be neglected if 20 dB below the direct signal), after which the first return from the rear of the room arrives. Subsequent delayed reflections constitute the end of the initial time-delay gap and the first signs of an exponential reverberant decay.

Live End–Dead End

Using time-delay spectrometry analysis, Davis and Davis experimentally mounted absorbing material on the surfaces of the entire front part of a control room. The results were encouraging; sound clarity in the control room was improved, as was stereo imaging. Also, the ambience of the room was perceptually more spacious. Giving the control room a precise initial time-delay gap gave listeners the impression of a much larger room.

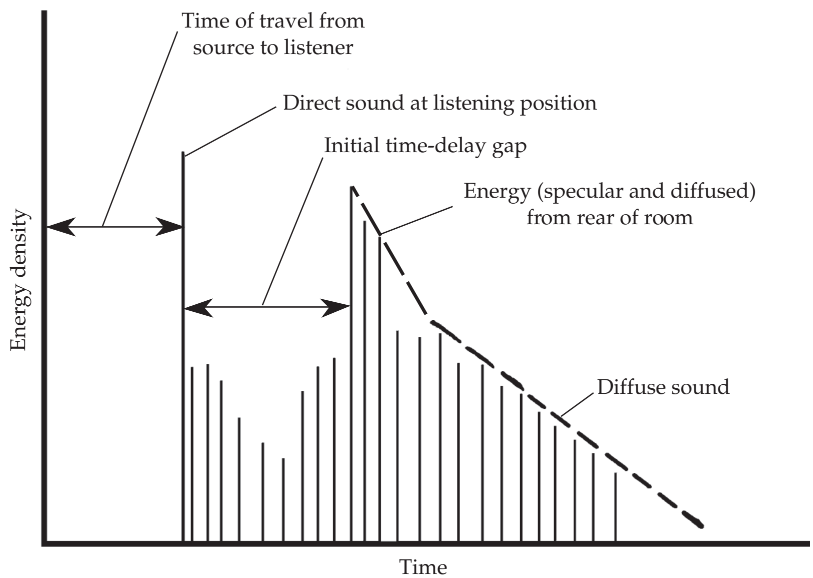

Figure 24-2 In a treated control room, the initial time-delay gap is well defined. Early reflections from the front of the room, although present, have been attenuated by absorbent placed in the front of the room.

The deadening of the front end of a control room near the observation window is a straightforward procedure—just cover the surfaces with absorbent. There is a resulting improvement of sound quality. Once the front end is made absorbent, attention naturally shifts to the rear end of the control room. If the rear end is deadened, the room will almost certainly be too absorbent overall. Therefore, to obtain a suitable reverberation time, the rear end of the control room is made live. This control-room configuration is known as live end–dead end (LEDE). Late reflections from the rear of the room do not present the same comb-filtering problems as front-end early reflections. Moreover, if the delay time of the rear-end reflections is within the Haas fusion zone, the human auditory system integrates these delayed reflections from the rear wall into the early-arrival sound. The ambience from the rear of the control room is perceived as coming naturally from the sound sources in the front of the room. However, specular reflections from the rear wall surfaces should be avoided; the rear wall surfaces should provide diffused reflections.

The principle of the LEDE control room was originally designed for stereo playback; it naturally lends itself to the assumption that the monitor loudspeakers will be in the front of the room. In rooms configured for multichannel playback, with surround loudspeakers, the LEDE design is more complex. However, the principle of avoiding early reflections is still important. Following the principle, surfaces near the loudspeakers should be deadened to prevent comb-filter effects. Other surfaces should be reflective while providing good diffusion. Specular reflections should be avoided. A further discussion on control-room design is given later; for simplicity, stereo playback is assumed. However, the principles can be extended and modified for surround-sound playback.

Specular Reflections versus Diffusion

In a specular reflection, incident energy is reflected from a surface such that the angle of reflection equals the angle of incidence, much like light reflects from a mirror. The surface is flat and smooth relative to the wavelength of sound so there is no diffusion. In a diffuse reflection, the incident energy is reflected such that it is uniformly scattered in all directions. This is caused when the reflecting surface has irregularities with a size on the order of the wavelength of sound being diffused. As with all diffusion, this is wavelength dependent. For example, a surface with irregularities of about 1 ft will diffuse sound that has a wavelength of about 1 ft (1 kHz). However, for sound with much longer wavelengths, the same 1-ft irregular surface will appear smooth, and that lower frequency sound will be specularly reflected by the entire wall. Also, sound with much shorter wavelengths will specularly reflect from the individual irregularities, whatever angle they are positioned at.

An example of specular reflection is shown in Fig. 24-3A; energy returns over a relatively narrow angle. If the same sound energy is incident on a reflection phase-grating diffuser, as shown in Fig. 24-3B, the returning energy is scattered over a wide area. In particular, a one-dimensional diffuser spreads its reflected energy in a horizontal hemidisc. By orienting other one-dimensional units, vertical hemidiscs of diffusion are easily obtained. This is in contrast to the specular panel, which distributes reflected energy in only a portion of half-space determined by the location of the source and the size of the panel.

In specular reflection, all of the acoustical energy from a given point on the reflector surface arrives in a single instant of time. With a reflection phase-grating diffuser, the

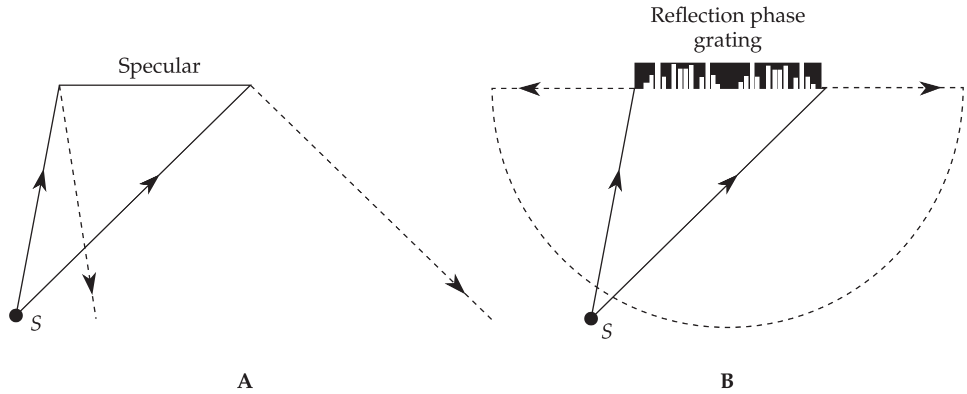

Figure 24-3 A comparison of reflection of sound energy. (A) A flat surface yields specular reflection. (B) A reflection-phase grating yields energy dispersed in a hemidisc.

reflected wavelets are not only spread in space, they are also spread in time. Each element of the diffuser returns energy, the respective reflected wavelets arriving at different times. This temporal distribution of reflected (diffused) energy results in a rich, dense, nonuniform mixture of comb filters that the human auditory system perceives as a pleasant ambience. This is in contrast to the sparse specular reflections that combine to form unpleasant wideband frequency-response deviations. For this reason, diffusing elements are placed in the live rear of a live end–dead end control room.

Another feature of the reflection phase grating, illustrated in Fig. 24-4, makes it especially desirable for the live rear end of a control room. Consider the example of

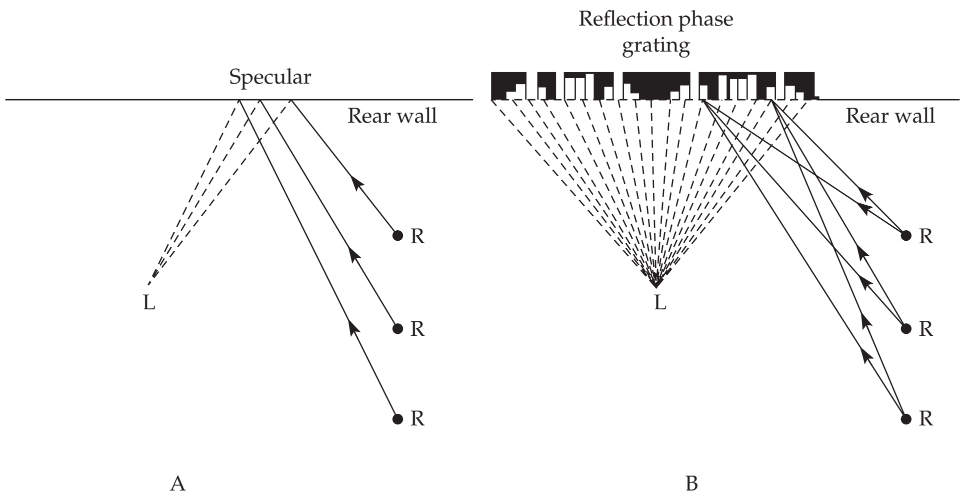

Figure 24-4 Side-wall reflections R in a control room return energy to the mixing position L. (A) If the rear wall is specular, only a single point on the rear wall returns energy to the mixing position. (B) With a reflection phase-grating diffuser, every element returns energy to the mixing position. (*D'Antonio.*)

three side-wall reflections R that impinge on the rear wall, returning energy to the mixing (listening) position, L. If the rear wall is specular, there is only one point on the surface returning energy from each source to the listener. In contrast, each element of the surface of the grating diffuser sends energy toward the listener. Energy from all sound sources (direct or reflected) falling on the diffuser is scattered to all observation positions. Instead of a small "sweet spot" at the mixing console, a much wider area of good sound quality results.

Low-Frequency Resonances in Control Rooms

As we have seen, room modes largely determine the low-frequency response of rooms, particularly small rooms such as most control rooms. This can be examined with a time-energy-frequency analysis. Figure 24-5 is a three-dimensional plot showing the relationship of time-energy-frequency at the mixing position in a control room. The vertical scale is 6 dB between marks. This is not sound-pressure level, but rather true energy. The frequency scale ranges from 9.64 to 351.22 Hz, a region expected to be dominated by modes. Time ranges from the rear of the plot, 2.771 msec per step, or about 100 msec for the entire traverse. The dramatic ridges in Fig. 24-5A are the modal (standing wave) responses of the control room, which together make up the low-frequency acoustical response of the room.

In addition to the modal response, this analysis contains information concerning a second phenomenon of control room design. This is the interference between the direct low-frequency wave from a monitor loudspeaker and its reflection from the rear wall. If the mixing position is 10 ft from the rear wall, the reflection lags the direct sound by the time it takes sound to travel 20 ft. The delay is t = (20 ft)/(1,130 ft/sec) = 17.7 msec. The frequency of the first comb-filter notch is $1/2t$ or 1/(2)(0.0177) = 28.25 Hz (see Chap. 10). Subsequent notches, spaced $1/t$ Hz, occur at 85, 141, 198, 254 Hz, and so on.

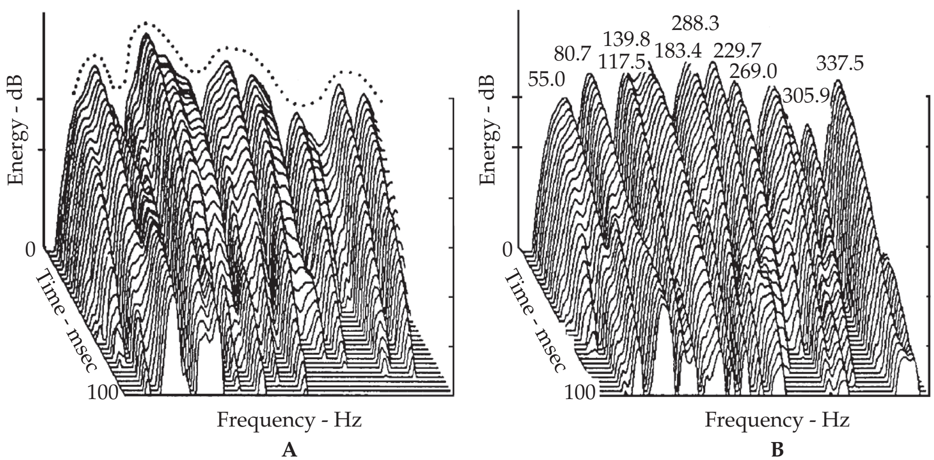

Figure 24-5 A three-dimensional energy-time-frequency plot of the sound field of a control room showing the room modes and their decay. (A) The untreated room condition shows uneven response. (B) Modal interference can be reduced by installing a low-frequency diffuser at the rear of the control room. (*D'Antonio.*)

The depth of the notches in the frequency response depends on the relative amplitudes of the direct and reflected components. One way to control the depth of the notches is to absorb the direct sound so that the reflection is low. However, this removes potentially useful sound energy from the room. Another approach is to build large diffusing units capable of diffusion at these low frequencies. The result of this approach is shown in Fig. 24-5B. A 10-ft-wide, 3-ft-deep, floor-to-ceiling diffuser is placed behind the midband diffusers. This provides a smoother frequency response and smoother decay. These modes decay about 15 dB in the 100-msec time sweep. This means a decay rate of roughly 150 dB/sec, which corresponds to a reverberation time of 0.4 sec, but the variation of decay rate between the various modes is great.

Initial Time-Delay Gaps in Practice

An energy-time display allows an evaluation of the distribution in time of acoustical energy in a room and the presence or absence of an initial time-delay gap. Figure 24-6 shows three energy-time displays of widely differing spaces, each of high acoustical quality. Figure 24-6A shows the energy-time display for a control room in the Master Sound Astoria recording studio in New York. The well-defined initial time-time gap, the exponential decay (a straight line on a log-frequency scale), and the well-diffused reflections are the hallmarks of a well-designed control room.

Figure 24-6B shows the energy-time display for the Concertgebouw concert hall in Haarlem, the Netherlands. A well-defined initial time-time gap of about 20 msec qualifies it as a high-quality concert hall; this hall was designed by Beranek.

There are many high-quality control rooms in the world, as well as many high-quality concert halls. However, there have been relatively few small listening or recording rooms that could be classified as quality spaces for the simple reason that their small volumes are dominated by normal modes and their associated problems. Figure 24-6C shows the energy-time display for a high-quality small listening room belonging to Audio Electronics Laboratory in Albertson, New York. A beneficial, initial time-delay gap of about 9 msec is made possible by a live end–dead end design in which the narrator and microphone are placed in the dead end of the room facing the live end.

Loudspeaker Placement, Reflection Paths, and Near-Field Monitoring

The management of reflections is a major concern in control room design. Davis and Davis recommended making the entire front end dead by applying absorbent to the surfaces. Berger and D'Antonio devised a method that does not depend on absorption but rather on the shaping of surfaces to nullify the negative effect of reflections. In this approach, the front end of the control room may be live, but with a very specific shape to provide specific reflection paths. This configuration places the listener in a reflection-free zone (RFZ). The placement of a loudspeaker close to solid boundaries can greatly affect its output. If it is placed close to an isolated solid surface, its power output into a one-half space is doubled, providing an increase of 3 dB. If the loudspeaker is placed close to the intersection of two such surfaces, the power is confined to a one-quarter space, and there is an increase of 6 dB. If placed close to the intersection of three such solid surfaces, the power radiates into a one-eighth space, and there is an increase of 9 dB.

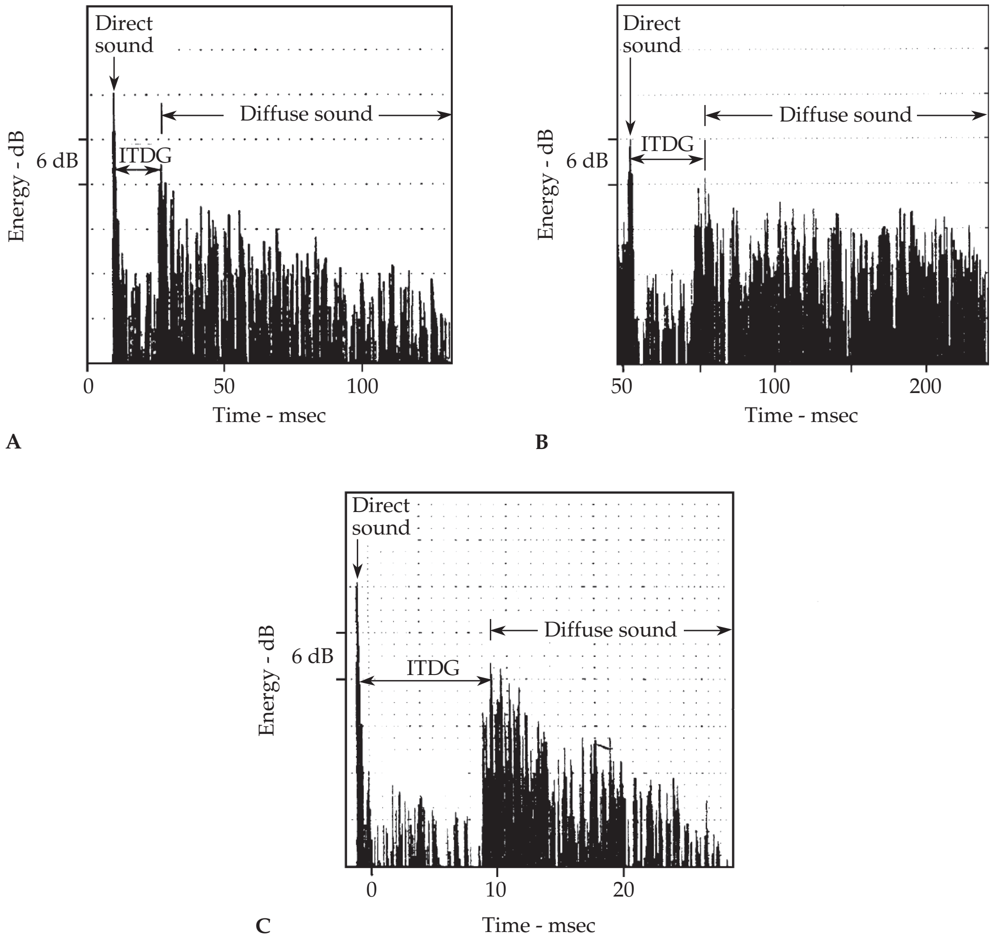

Figure 24-6 Illustrations of initial time-delay gaps for widely different types of spaces, each of high acoustical quality. (A) The control room of Master Sound Astoria, New York. (B) The Concertgebouw concert hall, Haarlem, the Netherlands. (C) A small listening room, Audio Electronics Laboratory, Albertson, New York. (*D'Antonio.*)

In control rooms as well as home listening rooms, it is common to place loudspeakers some specific distance away from the trihedral surfaces of the room. If the distances from the loudspeaker to the surfaces are appreciable in terms of wavelength of sound, problems are introduced. The overall power boost effect may be reduced, and the frequency response might be affected due to the constructive and destructive interference.

Specifically, when reflections from a room's boundaries combine with the direct sound, the interaction will either enhance or cancel sound at the listening position in varying degrees, depending on the amplitude and phase relationship between the reflection and the direct sound. This response with its peaks and dips is known as the speaker-boundary interference response (SBIR). This interference occurs across the entire frequency spectrum but is more significant at low frequencies. The typical effect is a low-frequency emphasis followed by a notch. SBIR is particularly prominent when a loudspeaker is placed in a corner of a room and the walls immediately surrounding it produce strong reflections.

A point source in a trihedral corner has a flat response at an observation point if that observation point is in a reflection-free zone. There are no reflections to contribute to interference effects. D'Antonio extends this observation and recommends placing control-room monitor loudspeakers in trihedral corners formed from splayed surfaces. By splaying the room boundaries, a reflection-free zone can be created around the listener. By splaying the walls and ceiling, it is even possible to extend the reflection-free zone across the entire console, several feet above, and enough space behind, to include the producer behind the mixing position.

One reflection that is uniquely problematic in control rooms is the reflection from the top surface of the mixing console. The console is placed between the monitor loudspeakers and the mixing engineer, and its angle often allows a reflection to strike the mixing position. This strong reflection arriving from the same direction as the direct sound, combined with the direct sound, can result in a comb-filtered signal at the mixing position. This can result in timbral deviations of the signal. To some extent, the comb-filter response at the mixing position is alleviated by the many other reflections naturally occurring in the room, arriving at different times and from different directions. Even so, the console reflection is a stubborn problem in many control rooms.

As we have seen, early reflections can add distortion to what mixing engineers hear from loudspeakers placed a distance away. The potentially high-quality sound of large monitor loudspeakers can be distorted by early reflections, negating their potential perceived accuracy and thus degrading the quality of the work. This problem of room interaction can be avoided by using near-field monitors; they relatively emphasize direct sound and discriminate against reflected sound, potentially providing a more accurate perceived sound.

Near-field monitors are small loudspeakers that are typically placed on the instrument bridge of the mixing console and supplement or sometimes replace large wall-mounted monitors. Such near-field monitors allow mixing engineers to hear relatively more direct sound and may give them a confidence they might not get from distant monitors. Some engineers prefer smaller loudspeakers on the premise that they more closely duplicate how the program will sound to consumers on their home loudspeakers. Others use them a great proportion of the time out of necessity; with near-field monitors, the listener hears more direct sound; therefore, a room with poor acoustics will have less effect. Another advantage to near-field monitors is their small size; in many control rooms, there simply isn't enough space to mount large loudspeakers and be seated the required distance away from them. Also, many near-field monitors are self-powered, simplifying amplifier placement and wiring. However, many small loudspeakers have inherent performance limitations compared to large loudspeakers, and thus provide lower sound quality. For example, bass response or sound-pressure output may be limited. If space and budget permits, the best solution is probably large monitors placed in a correctly designed control room, with near-field monitors available as an alternate listening reference.

The Reflection-Free-Zone Control Room

In their 1980 paper, Don Davis and Chips Davis specified that there should be ". . . an effectively anechoic path between the monitor loudspeakers and the mixer's ears." The most obvious way to achieve an anechoic condition is through absorption, hence the "dead end" designation in a live end–dead end room. However, a room designed with a reflection-free zone also achieves the goal of an anechoic path.

To design a control room with a reflection-free zone, image sources must be considered. The contribution of a reflection from a surface can be considered as coming from a virtual source on the other side of the reflecting plane on a line perpendicular to that plane through the observation point and at a distance from the plane equal to that to the observation point. With splayed surfaces in three dimensions, all the virtual sources must be visualized; this is necessary to establish the boundaries of the reflection-free zone.

A floor plan of a reflection-free zone control room is shown in Fig. 24-7. The monitor loudspeakers are flush-mounted as close as possible to the trihedral corner formed with the ceiling intersection. Next, both the front side walls and the front ceiling surfaces are splayed to send reflections away from the volume enclosing the listener. It is possible to create an adequate reflection-free zone at the mixing position by proper splaying of walls. In this way, an anechoic condition is achieved without recourse to absorbents. If absorbent is needed to control specific reflections, it can be applied to the splayed surfaces.

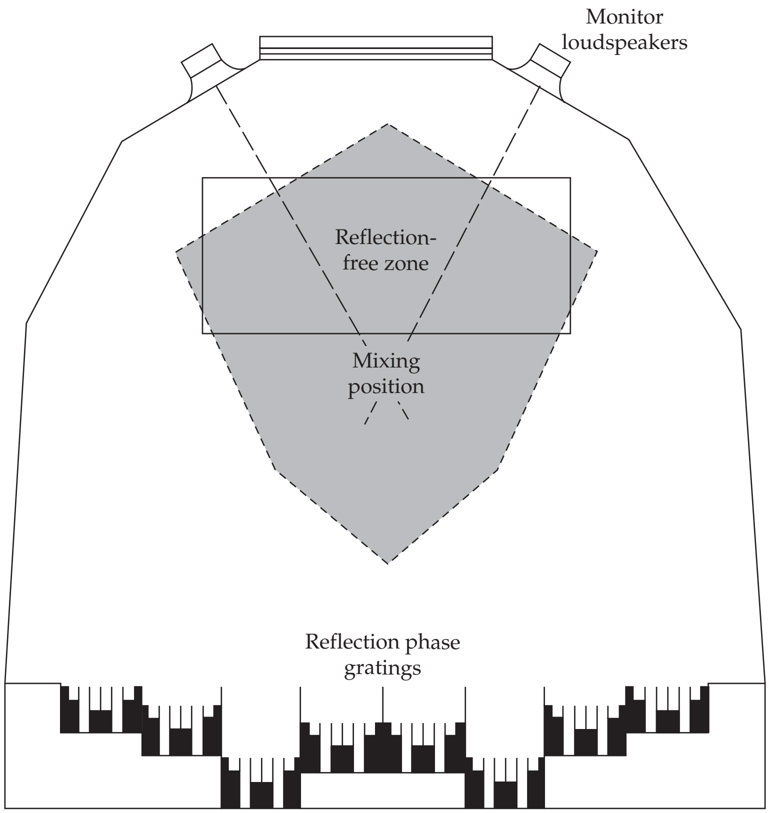

FIGURE 24-7 Plan view of a control room with a live, rather than a dead, front end. Front-end reflections at the mixing position are avoided by shaping the surfaces to create a reflection-free zone. In this design, the rear of the room is dominated by a mid- and high-band fractal diffuser and a low-frequency diffuser. (*D'Antonio.*)

The rear end of the reflection-free zone control room is provided with a complement of reflection phase-grating diffusers, also shown in Fig. 24-7. Although not fundamentally part of the RFZ design, the figure also shows how the self-similarity principle can be employed in the form of fractals. A high-frequency quadratic residue diffuser is mounted at the bottom of each low-frequency diffuser well. In this way, wideband sound energy falling on the rear wall is diffused and directed back to the mixing position. This sound is diffused both in space and time by the hemidisc pattern of the diffusers. An elevation view of the reflection-free zone control room is shown in Fig. 24-8.

Control-Room Frequency Range

Some of the salient constructional features of control rooms address specific acoustical issues. The range of frequencies handled in a control room is very great, and every frequency-dependent component must perform its function over that range. The commonly accepted high-fidelity range of 20 Hz to 20 kHz is a span of 10 octaves, or three decades. This represents a range of wavelengths from about 57 ft to 0.6 in. The control room must be built with this fact in mind.

The lowest modal frequency is associated with the longest dimension of a room. The room's response falls sharply below this frequency, which can be estimated by $1{,}130/2L$, in which L is the longest dimension in feet. For example, for a rectangular room following Sepmeyer's proportions of 1.00:1.28:1.54, with dimensions of $18.48 \times 15.36 \times 12$ ft, the low-frequency cutoff is 30.6 Hz. Very-low-frequency sound can exist in the room below the frequency of this lowest axial mode, but there is no resonant support because there are no modes in that region.

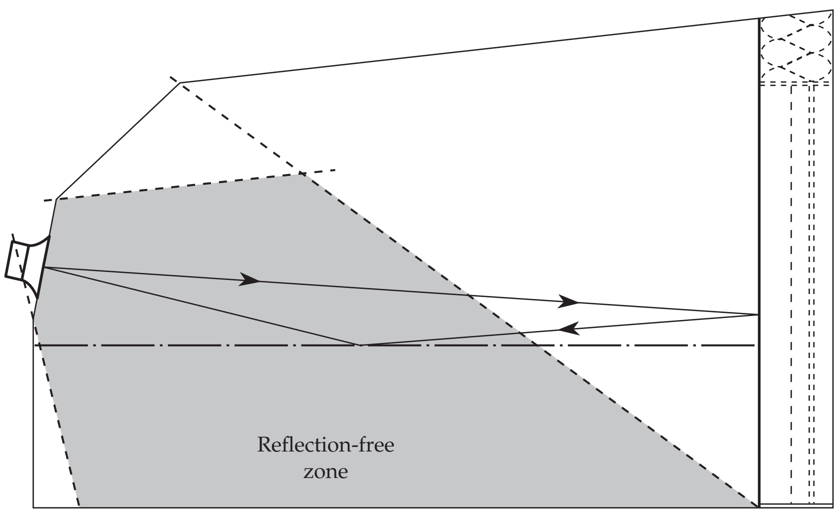

FIGURE 24-8 Elevation view of the reflection-free zone of the control room of Fig. 24-7. (*D'Antonio.*)

From 30.6 to about 100 Hz (for a room of this size), normal modes dominate and wave acoustics must be applied. In this room, 100 to about 400 Hz is a transition region in which diffraction and diffusion prevail. Above 400 Hz, true specular reflection and ray acoustics provide correct modeling. These frequency zones determine the construction of the control room.

Outer Shell and Inner Shell of the Control Room

The construction of a control room essentially comprises a massive outer shell to contain and distribute low-frequency modal energy, and an inner shell for reflection control. The size, shape, and proportions of the outer shell determine the number of modal frequencies and their specific distribution, as discussed in Chap. 13. There are two schools of thought: One prefers splaying the walls of the outer shell to break up modal patterns, whereas the other prefers a rectangular shape and its typical modal patterns. Only a modest deviation from a rectangular shape toward a trapezoidal shape is feasible. Such a shape does not eliminate modal patterns; it just distorts them into a potentially unpredictable form. Bilateral (side to side) symmetry is strongly preferred in a control room floor plan. Room symmetry provides balance for both low- and high-frequency sound when reproduced by either stereo or multichannel playback systems. To contain the low-frequency sound energy associated with control-room activities, thick walls, possibly even 12-in-thick concrete, are required.

The purpose of the control-room inner shell is, among other things, to provide the proper reflection pattern at the mixing position. For example, in a reflection-free zone configuration, construction of the inner shell can be relatively light. For the inner shell, shape is more important than mass. However, care must be taken to avoid any buzzing or rattling from lightweight partitions.

Design Criteria of a Control Room

The essential designs of three types of control rooms are presented below. The design and construction (and equipping) of a professional-quality control room is a very expensive undertaking. In the majority of projects, certain compromises must be made. This is the case in the examples presented here; areas of compromise will be discussed.

A survey of many facilities may suggest that control rooms employ very different design philosophies. To some extent, this is very true. Many control rooms use LEDE principles to overcome the negative consequences of first-order reflections and LEDE control rooms use fundamentally different acoustics than non-LEDE control rooms. Many control rooms use a reflection-free zone so that first-order reflections do not pass through the mixing position. Still other control rooms use designs that integrate principles from both approaches, or take entirely different approaches. While not acoustically significant, many control rooms are cosmetically very different. Despite this diversity, because they are tasked with identical functions, most control rooms share many important design criteria.

Although generally not as spacious as the studio room itself, professional control rooms are generally large. Control rooms may be 20 to 25 ft across at the widest point and high ceilings, perhaps 15 ft or more, are preferred. However, despite their large enclosed volume, control rooms have fairly short reverberation times. For example, a broadband reverberation time of about 0.3 sec is typical. Longer reverberation times could mask details in the monitored sound. In most cases, an expansion ceiling (sloping upward from the monitors) is used, as opposed to a compression ceiling (sloping downward). Most control rooms are designed with axial symmetry; the left and right halves of the room are identical. This is important to ensure proper mixing of the symmetrical stereo or surround panorama. Control rooms typically employ bass traps or other means to control low-frequency modes, and thus provide a more consistent low-frequency response. Other broadband absorption is used to provide a moderate reverberation time.

Diffusers are often mounted on the side and particularly the rear walls of control rooms; this is especially true when the front end of the room is absorptive. In most cases, rear-wall diffusers successfully provide an immersive sound field. However, some designers argue that a totally diffuse rear wall contributes too much ambience and thus can degrade playback imaging. Therefore, in some room designs, some absorption is placed in the middle of the rear wall, with diffusers on either side. This alternative design, and ones like it, should be particularly considered when a surround-sound monitoring system is installed. Generally, surround-sound playback requires a more balanced front/back distribution of reflecting, absorbing, and diffusing elements in the control room.

In professional control rooms, main monitor loudspeakers are often flush mounted in soffits; near-field monitors are placed on the console meter turret. In either case, loudspeakers are angled so the center lines intersect at a point about 1 to 2 ft behind the mixing engineer's head; this is done so that the center lines pass through the ears. The front stereo loudspeakers are placed at angles of about ±30°. Surround loudspeakers can be placed in a variety of positions, and mounting should allow a variety of standard configurations; for example, placement at ±110° to ±120° is often used.

As with most acoustically sensitive spaces, a control room must be quiet; a balanced noise criteria (NCB) of 15 to 20 is typical. This is relatively demanding, but not as stringent as the NCB specification for many studio rooms. In contrast to a studio with open microphones, a control room can tolerate some noise. On the other hand, a control room must be isolated from other rooms. Because of the open microphones in the studio, and potentially loud monitoring levels in the control room, the wall separating them must provide a high degree of isolation. In particular, the typically large window in this wall must be designed and constructed with great care to ensure a high transmission loss. To a lesser extent, noise from adjoining rooms such as maintenance rooms must be minimized as should noise intrusion from the control room to rooms such as offices.

Design Example 1: Control Room with Rectangular Walls

The control room shown in Fig. 24-9 is constructed in a room with rectangular walls and no inner shell walls; this might be considered to be a budget design. In this design, absorptivity is intentionally minimized; too much absorption would reduce acoustical signal energy and thus necessitate higher monitor amplifier power. Instead, to control reflections, diffusing elements are employed. The Abffusor functions as both a diffuser and an absorber (see Fig. 14-11). Abffusor panels (3) are mounted on the ceiling between the loudspeakers and the listening position. Additional Abffusor panels are mounted on the front and side walls to intercept reflections that would otherwise go toward the mixing position. Alternatively, other types of diffusers could be used. The monitor loudspeakers (1) are aimed in the normal stereo playback configuration.

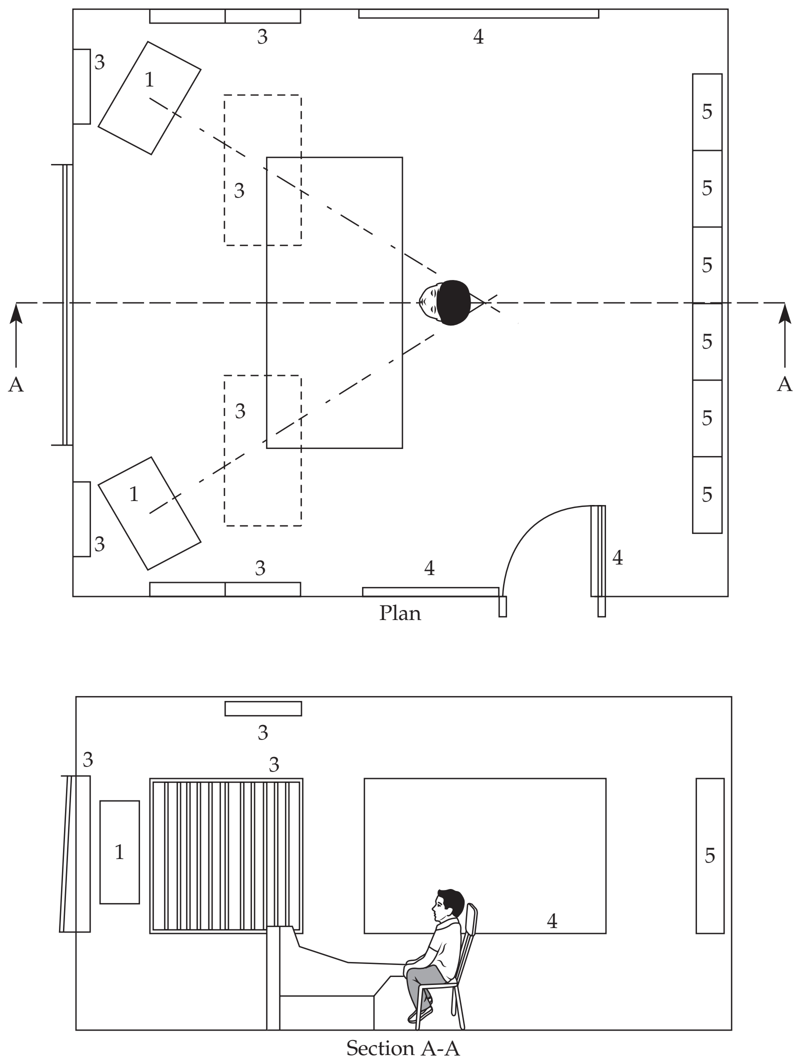

Figure 24-9 A control room (Design Example 1) with rectangular side walls.

As noted, this control room is designed within a room with rectangular walls. The mixing position is thus located between two parallel wall surfaces. Flutter echoes between the side walls would constitute a significant acoustical defect. These could be defeated with absorbent (which we wish to avoid). Instead, Flutterfree panels (4) of surface-mounted hardwood molding are installed on both side walls (see Fig. 14-12). These panels provide high-frequency diffusion that will eliminate flutter. The diffuser group at the rear of the control room comprises six QRD-734 diffusers (5). These 2- × 4-ft panels are mounted so that half of them have vertical wells and half have horizontal wells; this gives diffusion in both the horizontal and vertical planes (see Fig. 14-9). The Abffusor, Flutterfree, and QRD-734 panels are manufactured by RPG Acoustical Systems.

Design Example 2: Double-Shell Control Room with Splayed Walls

The control room shown in Fig. 24-10 is built within a room with rectangular walls, but has splayed side walls; this is a fairly sophisticated design using a reflection-free zone (RFZ) approach. The room is strategically designed to direct early reflections

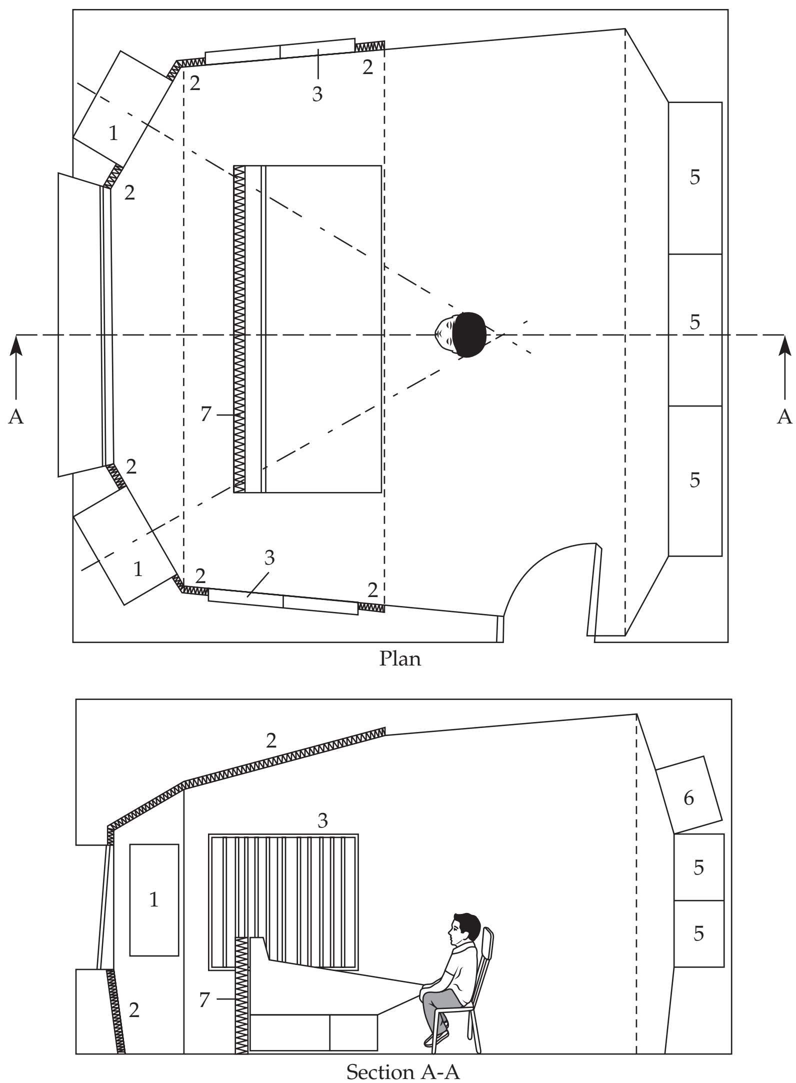

Figure 24-10 A control room (Design Example 2) with splayed side walls within a rectangular structure.

with room shaping and also reduce the level of those reflections with absorption. The splayed shape fits well into a rectangular room that comprises an outer shell. The inner walls can be constructed of relatively lightweight materials providing necessary treatment while the outer walls use heavy materials that provide sound isolation. The side walls are splayed sufficiently to create a reflection-free zone at the mixing position and to eliminate side-wall flutter echo. If the side-wall splaying insufficiently redirects the early-reflection paths, Abffusor or similar panels (3) can be placed on the side walls. The loudspeaker's (1) center lines intersect close behind the mixing engineer's head.

Absorbers (2) are placed on the ceiling, front, and side walls; the absorbers comprise fabric-covered graduated-density glass fiber. This treatment extends toward the rear of the room as far as the armrest of the mixing console. The back of the console and the wall under the front window constitute a quasi-cavity which may resonate at its own frequency. To control this resonance and to avoid "bass buildup," absorbers (7) are also mounted on the back of the console. Additional absorbers (2) can be placed on the wall under the window. Abffusor panels (3) are mounted on the side walls.

The rear-wall diffuser array comprises three rows of three diffuser panels each. Two rows of three diffuser panels such as QRD-734 diffusers (5) are set horizontally with their wells vertical. On top of them is one row of three QRD-734 diffusers (6) set horizontally with their wells horizontal; these three diffusers are angled slightly downward. This combination gives good diffusion in both the horizontal and vertical planes. Depending on room design, the amount of rear-wall diffusion can be increased or reduced. The Abffusor and QRD-734 panels are manufactured by RPG Acoustical Systems.

Design Example 3: Single-Shell Control Room with Splayed Walls

The control room shown in Fig. 24-11 combines design elements of the previous two examples, with a few notable differences; this is a more ambitious design. This room features slightly splayed side walls at a single angle. The splaying is not sufficient to eliminate the need for side-wall absorption/diffusion, but it is enough to eliminate flutter echo between the side walls. Diffuser panels such as Abffusors (3) are mounted on the side walls to intercept the early reflections otherwise directed at the mixing position. Other Abffusors are mounted on the ceiling to accomplish the same thing. Alternatively, other types of diffusers could be employed. The monitor loudspeakers (1) are recessed into soffits and mounted flush. A producer's desk has been placed behind the mixing position.

The rear of the control room is large enough to accommodate a sizable treatment element; thus a large diffuser array (8) is placed there. Various types of diffusers could be used; one possible candidate is an array of Diffractal units (see Figs. 14-14 and 14-15). This design mounts diffusers within diffusers (using the fractal principle) to provide very broadband diffusion. In addition to broadband diffusers, a low-frequency diffuser could be constructed from ordinary concrete blocks, or from DiffusorBlox (see Fig. 14-19). Diffusers with low-frequency response to 100 Hz would certainly address most potential modal problems in the room. The Abffusor and Diffractal panels are manufactured, and DiffusorBlox concrete blocks are licensed, by RPG Acoustical Systems.

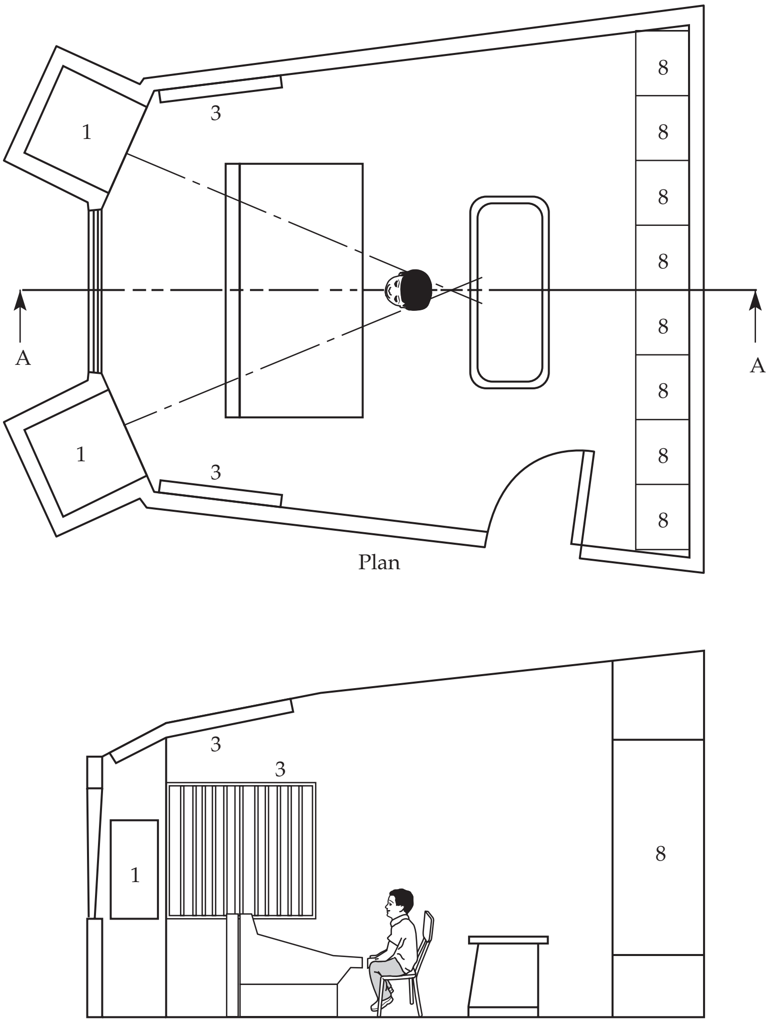

FIGURE 24-11 A control room (Design Example 3) with splayed side walls.

Figure 24-12 clearly shows the effect of rear-wall diffusers. These measurements were taken in two very similar control rooms; Fig. 24-12A is without rear diffusers and Fig. 24-12B is with rear diffusers. In the top record, the early energy is confined primarily in the groups of specular reflections (1) and (2) arriving at about 17 and 21 msec respectively. The reflection from the console (3) is also very prominent. The lower record shows the effect of the rear-wall diffuser. The valley of low reflections (4) is the ITDG of about 17 msec; after that come the highly diffuse reflections that add body and ambience to the direct signal. As noted, in some control room designs, some of the rear-wall diffusion area is reduced and replaced by absorption.

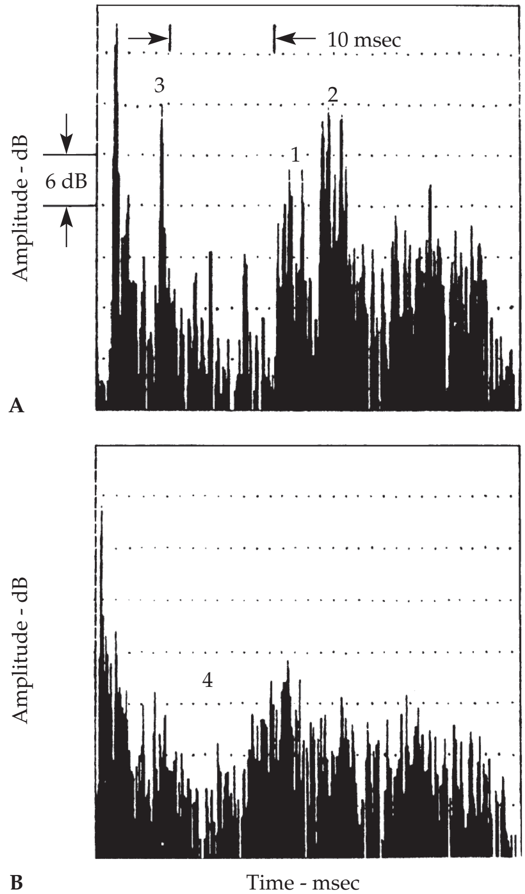

Figure 24-12 An echogram showing the effect of diffusers in similarly-designed control rooms. (A) Response without rear-wall diffusers. (B) Response with rear-wall diffusers. (*D'Antonio*.)

Key Points

- The acoustical design of professional control rooms is highly specialized and unique in the field of architectural acoustics with the specific purpose of providing high-quality loudspeaker playback to a single listening (mixing) position.
- Playback accuracy is critical because the mixing engineer bases all recording and mixing decisions on the sound delivered by the monitor loudspeakers via the room acoustics.
- The initial time-delay gap (ITDG) is used to examine early reflections in a sound field. It is defined as the time between the arrival of the direct sound at a given seat, and the arrival of early reflections.

- In a live end–dead end (LEDE) control-room configuration, the front of the room is absorptive and the rear of the room is reflective. Sound clarity and stereo imaging are improved, and the ambience of the room is perceptually more spacious.
- In a LEDE control room, if the delay time of the rear-end reflections is within the Haas fusion zone, the human auditory system integrates these delayed reflections from the rear wall into the early-arrival sound in front.
- With a reflection phase-grating diffuser, the reflected wavelets are not only spread in space, they are also spread in time. The human auditory system perceives this temporal distribution of diffused energy as a pleasant ambience. In contrast, sparse specular reflections form unpleasant wideband frequency-response deviations. For this reason, diffusing elements are used liberally in control-room designs and placed, for example, in the live rear of a live end–dead end control room.
- An energy-time display allows an evaluation of the distribution in time of acoustical energy in a room and the presence or absence of an initial time-delay gap.
- In a reflection-free zone (RFZ) control room, the front end of the control room may be live, but with a very specific shape to provide specific reflection paths to place the listener in a reflection-free area.
- A point source in a trihedral corner has a flat response at an observation point if that observation point is in a reflection-free zone.
- In an RFZ control room, the monitor loudspeakers are placed in trihedral corners formed from splayed surfaces. Properly configured, there are no reflections at the mixing position to contribute to interference effects.
- Reflection phase-grating diffusers are placed in the rear end of an RFZ control room.
- The construction of a control room essentially comprises a massive outer shell to contain and distribute low-frequency modal energy, and an inner shell for reflection control.
- Bilateral (side to side) symmetry is strongly preferred in a control room floor plan. Room symmetry provides balance for both low- and high-frequency sound when reproduced by either stereo or multichannel playback systems.
- Generally, surround-sound playback requires a more balanced front/back distribution of reflecting, absorbing, and diffusing elements in the control room.
- Despite a great diversity, because they are tasked with identical functions, most control rooms share many important design criteria.

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CHAPTER 25

Acoustics of Isolation Booths

As the name suggests, an isolation booth is used to acoustically isolate a small space from the surrounding environment. There are several types of isolation booths and they go by different names: announce booths, voice-over booths, and more generally—isolation booths. One might reasonably ask what the difference is, or even whether a meaningful difference exists, between these types of booths. Announce booths are strictly intended for speech applications; the narration may be broadcast live, or may be recorded for later replay. Voice-over booths similarly function for speech applications and are generally used to record speech that is placed over a video track. Isolation booths, depending on their size, can be used for both vocal and limited instrumental music recording; they are typically an adjunct room in a music recording studio.

From a design standpoint, announce booths, voice-over booths, and isolation booths are very similar. Because they are used exclusively for speech, announce booths and voice-over booths do not need to consider low-frequency response that is below the speech-frequency range, or high-frequency response that is above it. Because isolation booths can be used for both vocal and instrumental music recording, their design must consider the entire audio band. Other considerations, such as low ambient noise, are common to all types of booths. In practice, a well-designed booth can be used for any of these purposes.

Because these booths share the function of isolating one or a few sources, for simplicity they might all be referred to as isolation booths. In this chapter, we refer to all booths as isolation booths, and pertinent differences in their designs are noted. Three types of booth designs are presented, each an example of a different approach to room treatment. To minimize the variables in the comparisons to follow, a room having the dimensions of 6×8 ft with an 8-ft ceiling height is used for each of the three design examples.

Applications

Historically, announce booths were among the earliest commercial acoustical spaces. They originated in the early days of radio broadcasts and were used for on-air identification announcements. Today, announce booths and voice-over booths are found in most radio and TV broadcasting studios. For example, a roving reporter and companion sound/camera operator cover an outdoor event, recording the reporter on a portable device. Returning to the studio, the producer calls for additional narration by the reporter to cover scenes in which the reporter does not appear. These postproduction segments of the reporter's voice are recorded in a booth. The booth's acoustics must ensure that the sound in live and post scenes matches. Otherwise, the ambient information added by the booth's acoustics can affect the sound of the postrecording, making it noticeably different from the live sound when edited together.

Most large recording studios have one or more isolation booths, usually placed adjacent to the main studio. During a recording session, a musician, such as a vocalist, might be placed in the booth to isolate their track from the rest of the band. In many multitrack recording sessions, instruments are recorded separately for later remix; likewise the vocals are recorded separately. The various instruments and vocals could be recorded in the main studio, but it may be acoustically advantageous to record those tracks, particularly the vocal tracks, in an isolation booth. This would yield a more controlled, and generally less reverberant, ambient environment. The booth must provide an acoustical environment that is both neutral and natural sounding over a wide frequency range. The tracks recorded there must allow postproduction effects such as reverberation to be added, so that the tracks can be successfully added to the final music mix.

Design Criteria

Many isolation booths are very small; they are literally booths. Their small size presents particular acoustical challenges. Small closet-sized recording spaces can be notorious for their poor acoustics. Boundary surfaces may be treated with carpet, acoustical tile, or other materials that absorb high-frequency energy well, but provide very little low-frequency absorption. Thus important mid and high frequencies may be overly absorbed, whereas low-frequency room modes are left untreated. Because of the small dimensions of the booth, these room modes are relatively high in frequency, few, and widely spaced. Room-mode analysis should be performed to determine if these will affect frequency response in the speech range.

For voice recording, a booth may be acceptable if room-mode problems are lower than the speech range. If so, mid- and high-frequency absorption may be applied, to the extent needed to control flutter echoes in the booth. Consideration should be given to potential reflections from glass in an observation window, as well as reflections from a script stand.

In some cases, as noted, voice booths are relatively absorbent, and the voice is recorded intentionally dry, and reverberation is added as needed in postproduction. However, making a small booth too absorbent for both high and low frequencies can produce an eerie effect on the narrator. Such an anechoic space gives essentially no acoustical feedback, which the narrator may need for voice orientation and adjustment. For these reasons and others, an anechoic space is unsuitable for voice-over or announce booths, unless the dry booth acoustics are simply ignored and only headphones are used.

Another approach to treating an isolation booth is to obtain a relatively live sound. This provides a more natural ambient environment for people working in the booth without headphones and will yield a more natural recorded sound. However, recordings made in a booth with too much ambience may be difficult to match to other recordings. Ultimately, the intended application of the booth should determine how the booth is acoustically treated.

Isolation Requirements

As with any recording space, isolation booths must be acoustically isolated to provide a quiet space. Booths used exclusively for voice work need only be isolated in the speech-frequency range of 100 to 4,000 Hz; noise below and above that range can be removed from the audio signal using electrical filters. Booths used for music recording will require isolation across the entire audio frequency range.

The noise levels surrounding the booth will dictate what degree of isolating construction is needed. For voice work, where the external ambient noise level is low, simple gypsum-board stud walls may suffice. With higher external noise levels, more sophisticated gypsum-board walls can also provide excellent isolation in the speech-frequency region. Because of the low volume levels in a booth used for voice, only very modest isolation is needed to prevent noise intrusion to external spaces. For example, several voice studios could be built adjacent to one another and not suffer from interference.

Booths used for both voice and music must provide isolation across the broad frequency range. Their design must follow the guidelines used for music recording studios. In particular, greater low-frequency isolation may be required and higher sound levels may necessitate more stringent isolation overall.

As noted, many recording studios have isolation booths adjacent to the main studio. If the booth is used simultaneously with the main studio, the partition separating them must provide considerable isolation; for example, soundproof doors and double-pane windows must be used. If the booth is not used simultaneously with the main studio, isolation between the two can be lax; for example, sliding-glass doors would suffice.

When designing a small booth, it is important to consider HVAC needs. Temperatures can become uncomfortably warm in a small booth in a short period of time; some form of ventilation is needed. Because of the small confines, any air diffuser would be close to the microphone; particular attention must be given to minimizing noise from that source. In speech booths, low-frequency HVAC noise below the speech frequency region can be eliminated with high-pass filters in the audio signal path.

The Small-Room Problem

As discussed in Chap. 22, rooms smaller than 1,500 ft^3 are almost certain to produce timbral anomalies. Thus the small size of most isolation booths presents special problems. The sound field in a small room such as an isolation booth is dominated by modal resonances. The small dimensions dictate that resonances will be relatively higher in frequency compared to those in larger rooms. As noted, a room having the dimensions of 6×8 ft with an 8-ft ceiling height will be used for each of the three examples in this chapter. From a mode standpoint, these dimensions are not ideal; however, dimensions similar to these are often used because the footprint is minimal yet workable, and an 8-ft ceiling height is widely used in smaller rooms.

Table 25-1 tabulates all the frequencies of the axial modes of this $6 \times 8 \times 8$-ft space to 300 Hz beyond which modal timbral defects of the sound are minimal. To simplify the normal mode considerations of this small room, only the axial modes of the first order are calculated. The lowest resonance frequency of this room is the axial mode associated with the longest dimensions, the 8-ft length and height of the room. The room's length forms the (1, 0, 0) mode and the height forms the (0, 0, 1) mode, both at 71 Hz;

Room Dimensions = 8.0 × 6.0 × 8.0 ft					
	Axial Mode Resonances				
	Length $L = 8.0$ ft $f_1 = 565/L$ (Hz)	Width $W = 6.0$ ft $f_1 = 565/W$ (Hz)	Height $H = 8.0$ ft $f_1 = 565/H$ (Hz)	Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
f_1	70.6	94.2	70.6	70.6	0
f_2	141.3	188.3	141.3	70.6	23.6
f_3	211.9	282.5	211.9	94.2	47.1
f_4	282.5	376.7	282.5	141.3	0
f_5	353.1		353.1	141.3	47.0
				188.3	23.6
				211.9	0
				211.9	70.6
				282.5	0
				282.5	0
				282.5	70.6
				353.1	

Mean axial mode spacing: 25.7 Hz.
Standard deviation: 28.7 Hz.

TABLE 25-1 Axial Modes of Isolation Booth Design Examples 1, 2, and 3

this is the lowest-frequency mode in the room. There will be no resonant support for sound of lower frequency than this. Because the length and the height of the room are the same at 8 ft there will be coincidences at 71 Hz and at every multiple of 71 Hz. The axial-mode spacing column shows coincidences at 71, 141, and 212 Hz, and a triple coincidence at 282 Hz. The effect of the modes at these frequencies will be stronger. Changes in voice timbre may be encountered at these frequencies. The triple coincidence at 282 Hz is at a frequency that is high enough to lead us to hope for a minimal timbral problem. Such coincidences could have been avoided (or at least minimized) by choosing more favorable room proportions. Two identical dimensions, and dimensions that are multiples of each other, should be avoided.

Although the mean (average) difference between adjacent axial modal frequencies is 25.7 Hz, the gaps shown in the mode spacing column in Table 25-1 suggest that the room will have a very irregular response below 300 Hz. This means that extra absorption must be provided to control these axial modes. Rooms smaller than 6 × 8 × 8 ft have even greater problems; the axial modal frequencies are higher, potentially having an even more degrading effect on sound quality in the mid-frequency range.

Design Example 1: Traditional Isolation Booth

This first design example serves only to illustrate the acoustical problems in a traditional isolation booth. It is not a design intended to be built. Of course, existing traditional booths are very different, and presenting one to represent them all is a fragile fiction.

However, the design presented here, using carpet and acoustical tile, is commonly found in a great many existing isolation booths.

This assumed traditional isolation booth is illustrated in Fig. 25-1. It is a small $6 \times 8 \times$ 8-ft rectangular space with common 2×4 framing covered on both sides with 1/2-in gypsum board. With all its deficiencies, if it is located inside a larger room, this structure would provide adequate isolation from outside noise except in extreme cases. A heavy carpet with 40-oz pad covers the floor and the carpet (without the pad) is run up the wall to the wainscot strip. The ceiling is covered with acoustical tile, as well as the walls above the wainscot strip.

Axial Modes

The sound pressure distribution of the axial modes in this $6 \times 8 \times 8$-ft room is illustrated in Fig. 25-2. The small pressure graphs are moved outside the space for clarity. The left portion of the figure shows the resonance between the east and west walls creating the width mode (0, 1, 0) at 94 Hz. When this resonance is excited, the sound pressure over the entire east and west wall surfaces is high, with a null plane a,b,c,d,a extending from floor to ceiling.

The lower right portion of Fig. 25-2 shows the lengthwise mode (1, 0, 0) at 71 Hz. When this resonance is excited, the sound pressure is maximum over the entire surfaces of the north and south walls. There is a null plane e,f,g,h,e bisecting the room.

The upper right portion of Fig. 25-2 shows the vertical mode (0, 0, 1) at 71 Hz, at resonance between the floor and the ceiling. When this resonance is excited, sound pressure is high over the entire floor and ceiling surface with a null plane i,j,k,l,i bisecting the room.

These pressure maxima and minima dominate the acoustics of the space. The announcer could, for example, sit with his or her head positioned in or near all three null planes, yielding a poor acoustical response. The acoustical response would fluctuate if the announcer's head moves about and likewise the response of the microphone would fluctuate considerably if it is moved about in search of the best-sounding position. What the announcer hears and the microphone "hears" may be quite different.

Reverberation Time

A reverberation-time calculation is helpful in studying the frequency distribution of absorbance in a room. Table 25-2 shows the calculations that apply to the booth's carpet and acoustical tile. The absorption is $A = S\alpha$ where S is the surface area and α is the absorption coefficient. The result shows a long reverberation time below 250 Hz, as shown in Fig. 25-3 (solid line). However, other elements of the room will add additional absorption. In particular, the gypsum board and wood floor provide absorption in this needy region. It is easy to overlook this "free" absorption in a room, but its effect is very important.

The gypsum-board panels and the wood floor will contribute diaphragmatic absorption. The relatively large areas of wall panels and floor mean that their resonance points and peaks of absorption will be at low frequencies. Absorption at low frequencies is just what is needed to control the axial modes in this small room, and to minimize the great variations of sound pressure in the room caused by the axial modes (see Fig. 25-2). With the diaphragmatic absorption effective in the 70- to 250-Hz region, the modal maxima would be decreased and the null areas would be raised, but this would not completely overcome the uneven response caused by the axial modes.

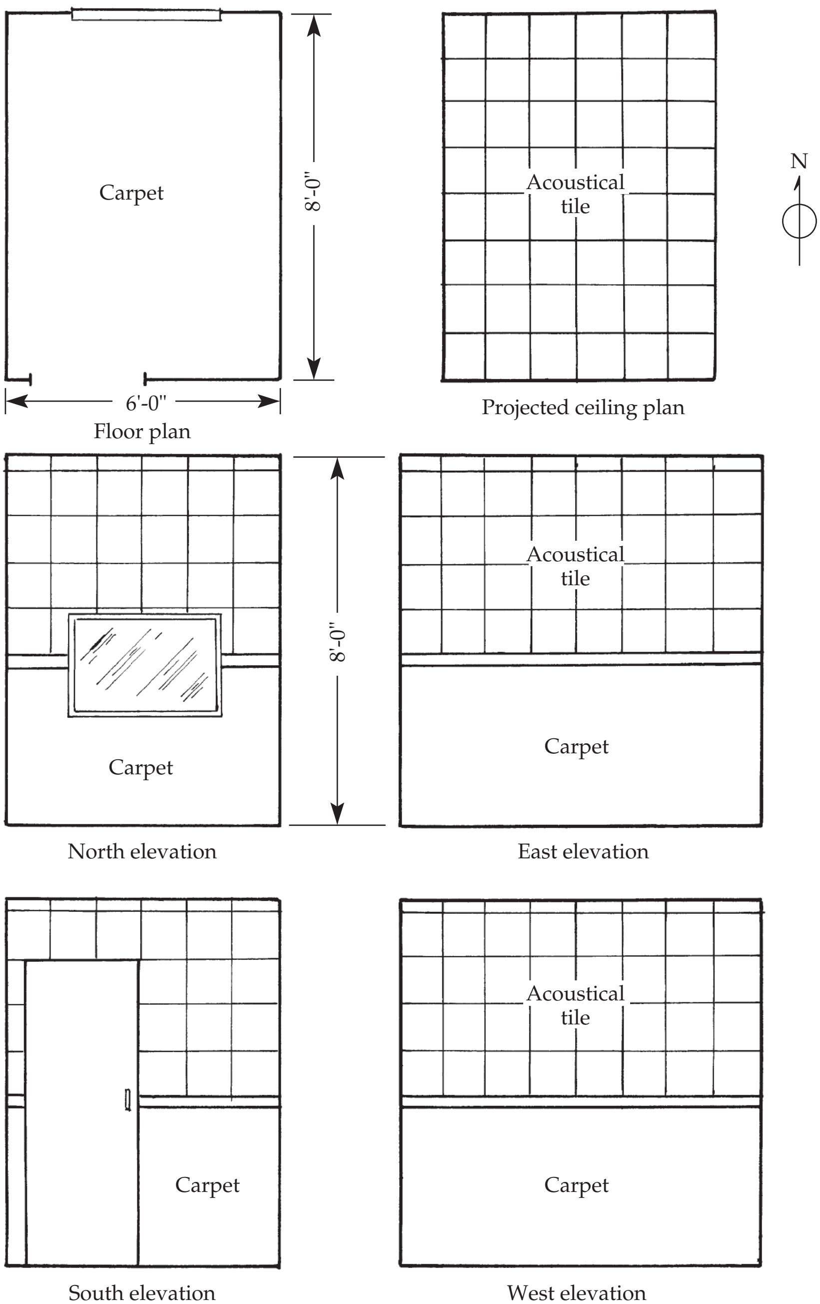


Figure 25-1 An isolation booth (Design Example 1) using a traditional treatment plan. This design illustrates the problems in a traditional booth; it is not recommended as a buildable design.

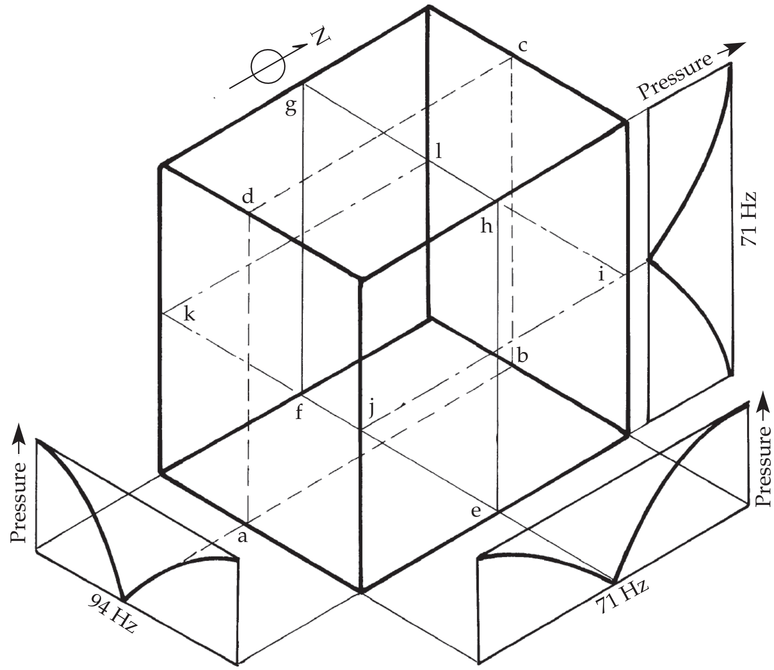


FIGURE 25-2 An axial-mode pressure diagram for the example 6 × 8 × 8-ft room.

By considering the absorption due to the walls and the floor in addition to that of the carpet and acoustical tile, the reverberation time is lowered materially as shown in Fig. 25-3 (dashed line). However, without depending too much on the precision of the calculated reverberation time, the 0.1 sec of Fig. 25-3 is very low. A deadness may characterize this room; it may possibly have too much absorption for acoustical comfort and effectiveness. Another issue to consider is diffusion. The room currently lacks diffusion. As with any room, the room's acoustics would be improved if diffusion was used to spatially distribute the sound energy more evenly through the room.

In summary, we can consider the various issues that would characterize the traditional isolation booth of this first design example: The axial modes would produce an irregular frequency response in the sound field; however, the diaphragmatic absorption of walls and floor may help to partly smooth the response; excessive mid- to high-frequency absorption would make the room too dead; diffusion is needed to thoroughly distribute sound energy. In short, better design approaches are needed.

Design Example 2: Isolation Booth with Cylindrical Traps

This second design example of an isolation booth is shown in Fig. 25-4. This room design features quarter-round and half-round cylindrical traps. These alone constitute the acoustical treatment of the room. Specifically, 16-in quarter-round traps are mounted in the four wall corners to provide absorption to control normal modes. A series of 9-in

Material	Area (ft^2)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$
Carpet, heavy 40-oz pad	140	0.08	11.2	0.24	33.6	0.57	79.8	0.69	96.6	0.71	99.4	0.73	102.2
Acoustical tile 1/2 in	155	0.07	10.9	0.21	32.6	0.66	102.3	0.75	116.3	0.62	96.1	0.49	76.0
Absorption, sabins			22.1		66.2		182.1		212.9		195.5		178.2
Reverberation time, sec			0.85		0.28		0.10		0.09		0.10		0.11
Drywall	224	0.29	65.0	0.10	22.4	0.05	11.2	0.04	9.0	0.07	15.7	0.09	20.2
Floor, wood	48	0.15	7.2	0.11	5.3	0.10	4.8	0.07	3.4	0.06	2.9	0.07	3.4
Total absorption, sabins (carpet, tile, drywall, floor)			94.3		93.9		198.1		225.3		214.1		201.8
Reverberation time, sec			0.20		0.20		0.10		0.08		0.09		0.09

Table 25-2 Absorption and Reverberation-Time Calculations for a Traditional Isolation Booth (Design Example 1)

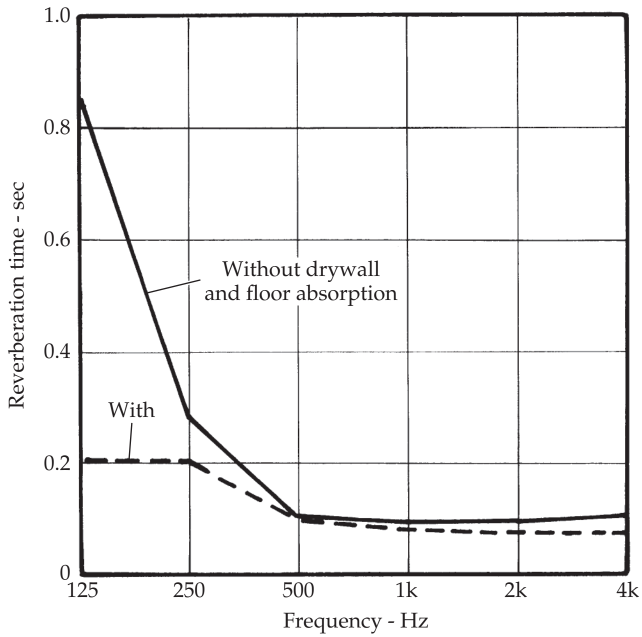


FIGURE 25-3 Reverberation time for an isolation booth (Design Example 1).

half-round cylindrical traps alternate on the four walls and ceiling with hard, reflective drywall strips as wide as the traps. The half-rounds on the walls and ceiling provide absorption and, in conjunction with the strips of reflective wall surface between the cylindrical traps, also provide diffusion. The door and window should also be covered in this way to avoid compromising the sound field. Limited visibility is available through glass windows in some of the reflective strips. The floor remains untreated. The perspective drawing of the isolation booth with two walls removed, as shown in Fig. 25-5, might help in understanding the conventional drawings of Fig. 25-4.

The TubeTrap is an example of a commercially available cylindrical tube trap. The construction of a half-round TubeTrap is shown in Fig. 25-6. It is a rigid, easily handled unit with a fabric cover. The sound absorption characteristics of the various forms of the cylindrical trap are shown in Fig. 25-7. Note that in this figure the absorption per linear foot is given directly in sabins, or absorption units, rather than the usual specifications of absorption coefficient and surface area. The TubeTrap is manufactured by Acoustic Sciences, Inc.

In a typical room, too-few early reflections from bare areas of floor, walls, and ceiling are dominant and interact with each other, forming comb-filter distortion in the sound. In this room example, discrete reflections from the strips of wall between the cylindrical traps or even the script stand, which may otherwise cause comb-filter anomalies, are overwhelmed by the density of diffuse reflections from the many cylindrical surfaces in the room. They are not audible as timbral defects. Instead, the density of reflections yields a natural ambience and overall smooth response.

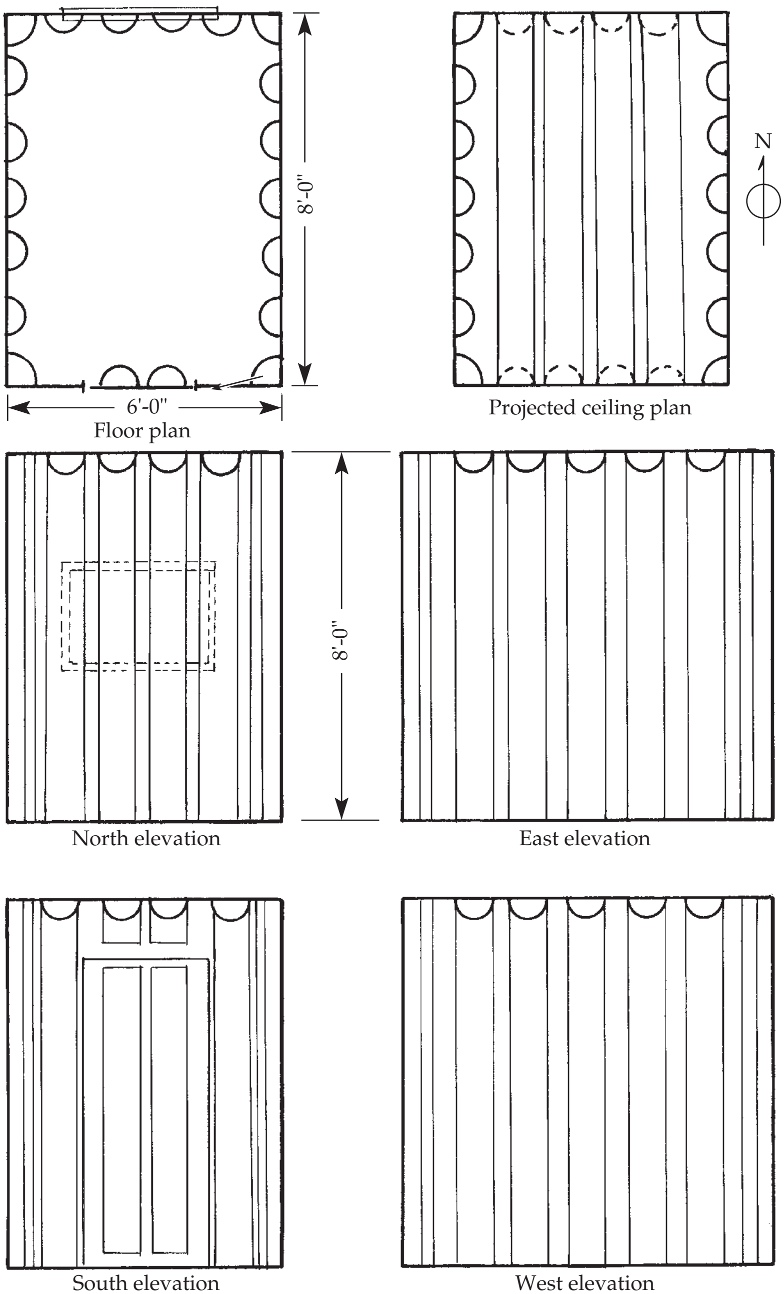


Figure 25-4 An isolation booth (Design Example 2) with a treatment plan using cylindrical traps.

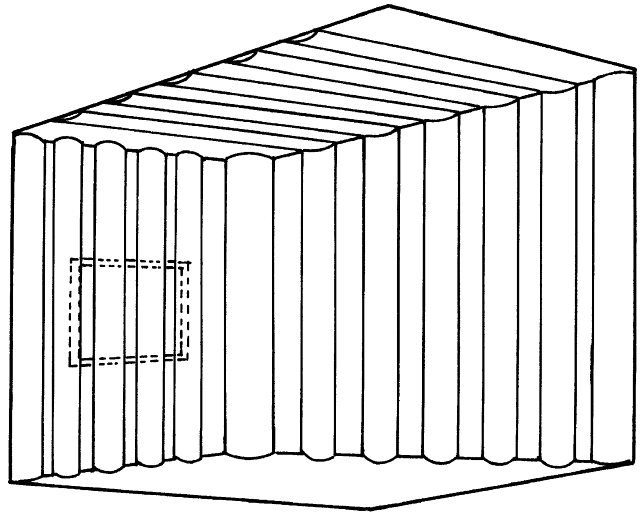


Figure 25-5 A perspective sketch for an isolation booth (Design Example 2).

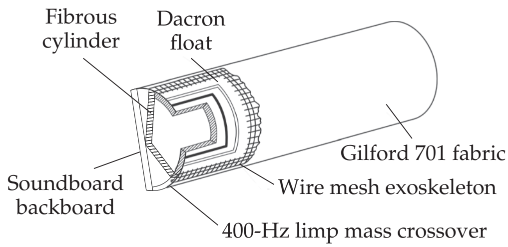


Figure 25-6 TubeTrap construction details.

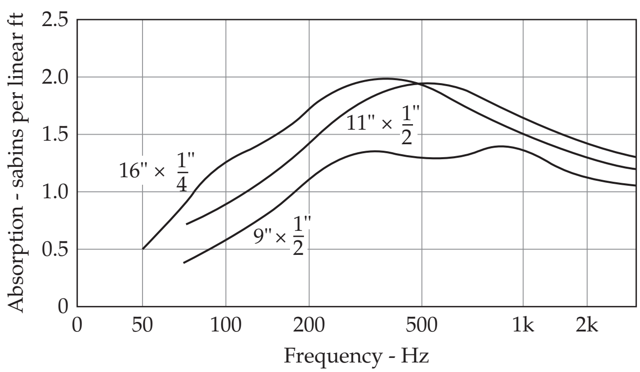


Figure 25-7 TubeTrap absorption.

Acoustical Measurements

Figure 25-8 shows an isolation booth that was constructed on the same principles as the booth in this design example (see Fig. 25-4). It uses a multiplicity of cylindrical tube traps as its sole treatment, but its footprint differs from the rectangular design. A set of acoustical measurements were made, giving a detailed view of the acoustical functioning of the room. An energy-time-curve (ETC) of the room's response is shown in Fig. 25-9. The horizontal time axis extends from 0 to 80 msec. The spike at the left extreme is the arrival of the direct signal at the measuring microphone, followed by a smoothly decaying ambient sound field. This decay is uniform and quite fast, corresponding to a reverberation time of 0.06 sec (60 msec). Figure 25-10 is an expanded view of Fig. 25-9, showing the first 20 msec of the same ETC plot. There is a distinct initial time-delay gap of 3 msec between the arrival of the direct sound and the arrival of a smooth, dense fill of reflections from the cylindrical trap grids on the walls and ceiling.

The time-energy-frequency (TEF) "waterfall" record of the room is shown in Fig. 25-11. The vertical amplitude scale is 12 dB/division. The horizontal frequency scale ranges from 100 Hz to 10 kHz. The diagonal scale shows time, ranging from zero at the rear to 60 msec at the front. This measurement shows no prominent room modal resonances, only a dense series of smooth decays over time throughout the entire 100 Hz to 10 kHz frequency range. These plots demonstrate that the acoustical treatment of the room yields a response that is smooth over both time and frequency.

Reverberation Time

The reverberation-time calculations for the isolation booth in this example are shown in Table 25-3 and the results are plotted in Fig. 25-12. The absorption is $A = S\alpha$ where S is the surface area and α is the absorption coefficient. The very low calculated reverberation time of about 70 msec approximately agrees with the measured decay of Fig. 25-9 and is what is expected in a room such as this. A similarly low reverberation time was

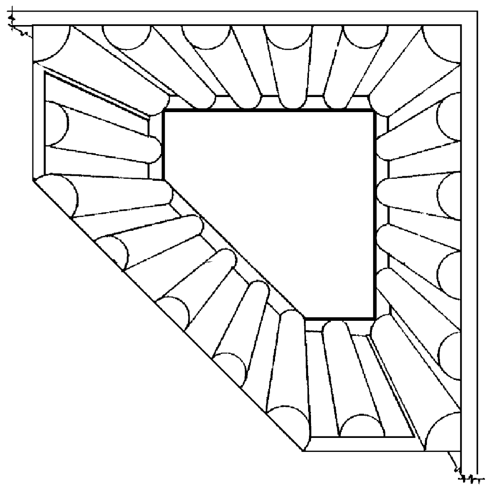


FIGURE 25-8 View of an isolation booth. The ceiling, walls, door, and the observation window are all covered with spaced half-round tube traps. (*Acoustic Sciences Corporation.*)

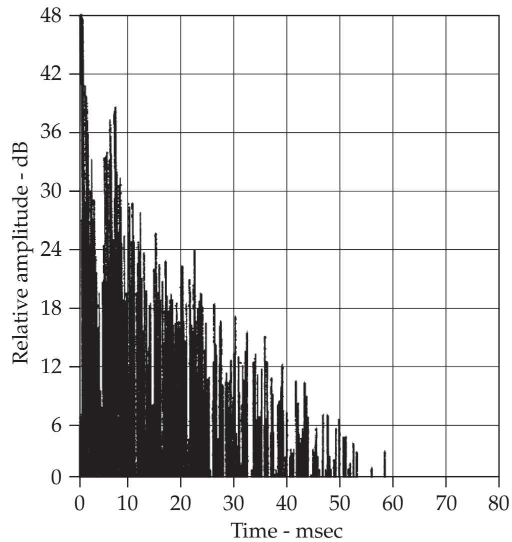


Figure 25-9 Energy-time-curve (ETC) measurement of the isolation booth of Fig. 25-8. About 1,000 reflections/sec are produced in such a space, resulting in a dense, but rapidly decaying ambient field. The horizontal time base extends to 80 msec. (*Acoustic Sciences Corporation.*)

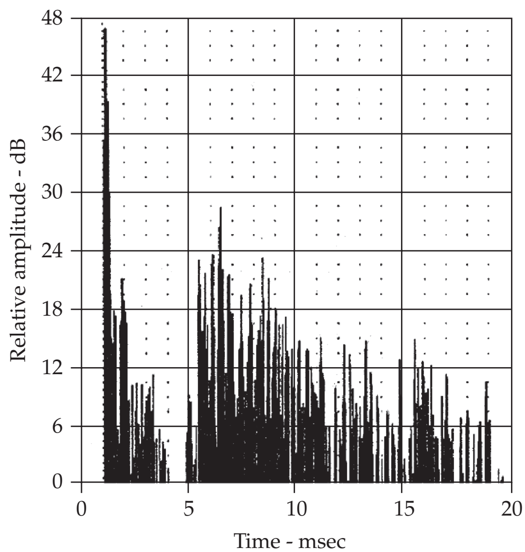


Figure 25-10 The early part of the energy-time-curve of Fig. 25-9. In this case, the expanded time base extends to 20 msec. A well-defined initial time-delay gap is present. (*Acoustic Sciences Corporation.*)

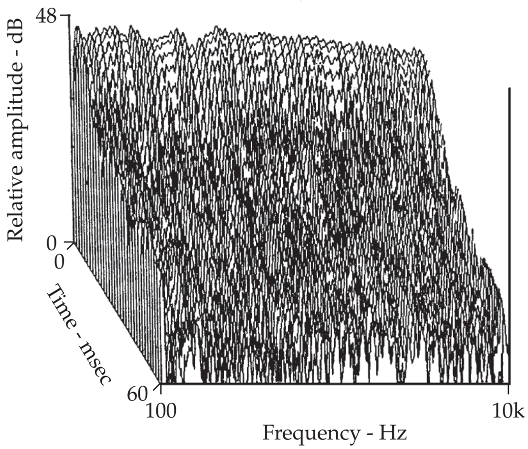


Figure 25-11 Time-energy-frequency (TEF) response of the isolation booth of Fig. 25-8. The vertical scale is energy (12 dB/division), the horizontal scale is frequency (100 Hz to 10 kHz), and the diagonal scale is time (zero at the back to 60 msec at the front). The broadband decay of sound is smooth, balanced, and dense. (*Acoustic Sciences Corporation.*)

calculated for the traditional booth (see Fig. 25-3), which was overloaded with carpet and acoustical tile. The difference is that the dead characteristics of the cylindrical trap room are improved with highly diffused sound.

This isolation booth would yield voice recordings that are accurate and clean-sounding. Recordings made at one microphone position would generally match those recordings made at different microphone positions. Moving the microphone would only change the fine structure of the ambient sound (some thousand reflections per second) and would have little effect on the quality of sound. However, the announcer sitting in such a dead booth would have little acoustical feedback from the room. The announcer's own voice would sound somewhat unnatural to himself or herself. This can be easily rectified by providing the announcer with headphones and a quality playback of their own voice.

Design Example 3: Isolation Booth with Diffusers

The first isolation booth example used absorption excessively. The second example used cylindrical traps and would yield satisfactory recording quality of voice, albeit with a short reverberation time. This third design example uses diffusing elements to also produce satisfactory small-room conditions with a short but reasonable reverberation time.

The plan shown in Fig. 25-13 is based on the same $6 \times 8 \times 8$-ft room dimension. The T-bar suspended ceiling is filled with twelve 2×2-ft Formedffusor quadratic-residue units (4). These are diffusing elements that also offer mid- to low-frequency absorption. To control axial modes, two Modex Corner bass traps (1) are mounted in each corner, for a total quantity of eight traps. They are placed in corners because modal pressures are highest there, thus their effect is greater. These units contain a membrane with high

Material	Length (ft)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		A/ft	*A*	*A*/ft	*A*	*A*/ft	*A*	*A*/ft	*A*	*A*/ft	*A*	*A*/ft	*A*
9-in TubeTraps, half-round	156	0.8	124.8	1.3	202.8	1.3	202.8	1.4	218.4	1.1	171.6	1.0	156.0
16-in TubeTraps, quarter-round	32	1.4	44.8	1.9	60.8	1.9	60.8	1.6	51.2	1.3	41.6	1.1	35.2
	Area (ft^2)	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$
Drywall	224	0.29	65.0	0.10	22.4	0.05	11.2	0.04	9.0	0.07	15.7	0.09	20.2
Floor	48	0.15	7.2	0.11	5.3	0.10	4.8	0.07	3.4	0.06	2.9	0.07	3.4
Total absorption, sabins			241.8		291.3		279.6		282.0		231.8		214.8
Reverberation time, sec			0.078		0.065		0.067		0.067		0.081		0.088

Table 25-3 Absorption and Reverberation-Time Calculations for an Isolation Booth with Cylindrical Traps (Fig. 25-8)

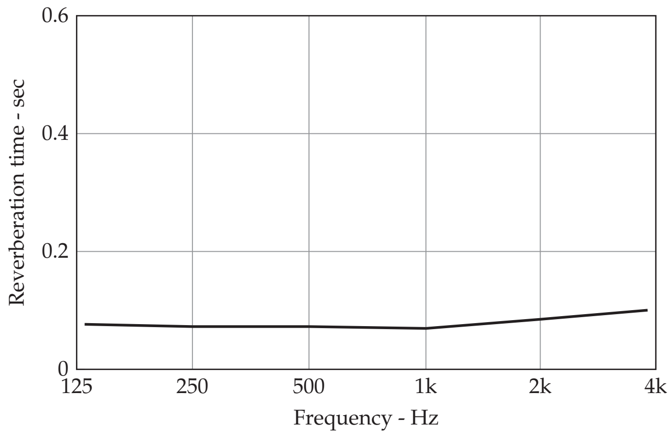


Figure 25-12 Reverberation time for the isolation booth of Fig. 25-8.

internal losses mounted in a trapezoidal cavity that fits into 90-degree corners; they measure $2 \times 2 \times 1$ ft. The traps are available in tuning frequencies of 40, 63, and 80 Hz. This design uses 80-Hz units; the absorption measurements are shown in Fig. 25-14. These bass traps provide absorption near the dominant axial modes of the room at 71 Hz and 94 Hz.

Abffusor units (2) are mounted on the east and west walls, three to a side. These are positioned to minimize potential flutter echoes and to distribute the diffusion effect. These units measure 2×2 ft and have a wide-band absorption characteristic, as shown in Fig. 25-15. Two 2×2-ft Skyline units (5) are mounted on the door in the south wall. Two others are mounted above and below the observation window in the north wall. These have maximum diffusion in two dimensions with a minimum of absorption. These diffusing surfaces are described in Chap. 14.

The only remaining diffuser is mounted over the observation window of the north wall. This 2×2-ft Diviewsor (3) is a quadratic-residue diffuser based on the prime 7 sequence. It is a conventional type-734 diffuser, described in Chap. 14, except that the panels are of transparent Plexiglas. This allows visual communication through the Plexiglas window. The Formedffusor, Modex Corner, Abffusor, Skyline, and Diviewsor modules are manufactured by RPG Acoustical Systems.

Reverberation Time

The isolation booth in this example features a great deal of diffusion. However, there must also be sufficient absorption so that the sound is not too bright. Table 25-4 shows the reverberation time calculations for this isolation booth with diffusing modules. The absorption is $A = S\alpha$ where S is the surface area and α is the absorption coefficient. No sound absorption material has been added to the room apart from that intrinsic to the diffusing elements and the structure itself. Both the Abffusors and the Formedffusor provide absorption in addition to their main function as diffusers. From the calculations of Table 25-4 come the reverberation-time data plotted in Fig. 25-16.

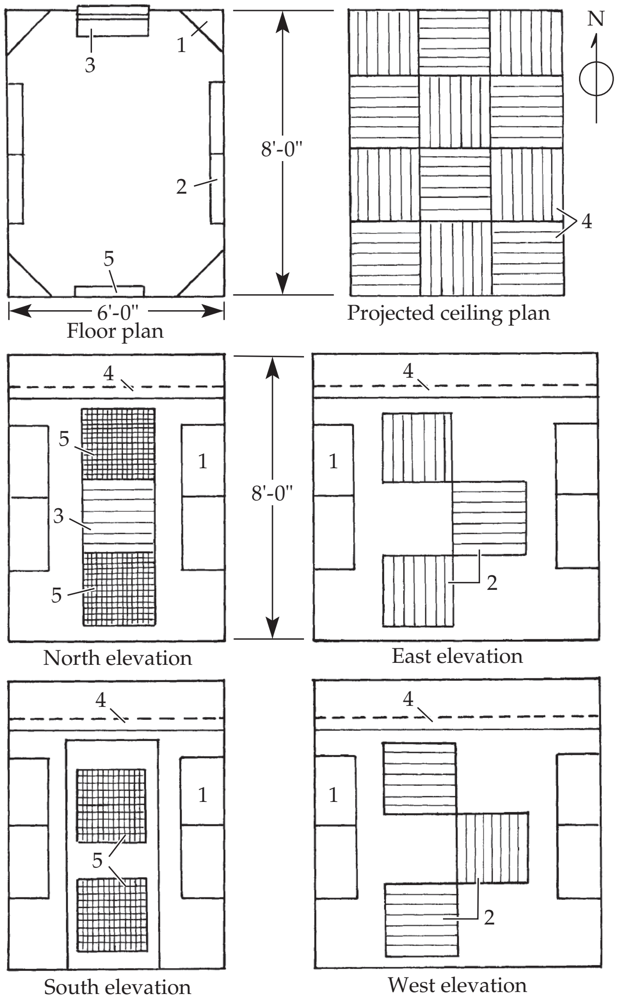


FIGURE 25-13 An isolation booth (Design Example 3) treatment plan using diffusers.

The reverberation time of approximately 0.3 sec is appropriate for such a small space. It is suggested that anyone building a booth like this and treating it with diffusing elements alone should listen analytically with the floor bare and then if desired, add a 5×7-ft rug or perhaps carpet the entire floor to bring the reverberation time down to about 0.2 sec.

Evaluation and Comparison

Three isolation booth designs have been presented. The same small $6 \times 8 \times 8$-ft space has been treated in a traditional manner, and in two more contemporary ways using various acoustical treatment devices. The numerical results are summarized in Table 25-5.

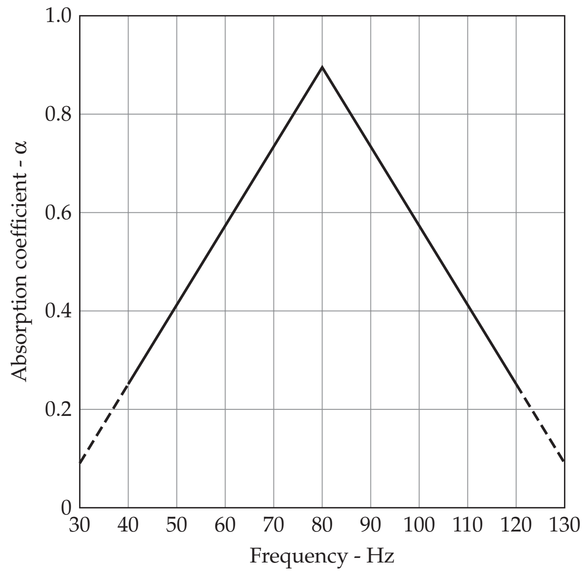


FIGURE 25-14 Absorption characteristic of an 80-Hz Modex Corner bass trap.

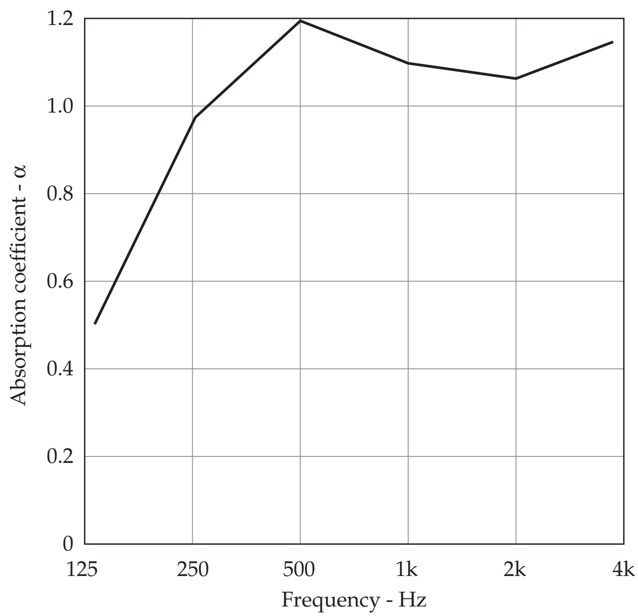


FIGURE 25-15 Absorption characteristic of an Abffusor.

Material	Area (ft^2)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	Sα	α	Sα	α	Sα	α	Sα	α	Sα	α	Sα
Modex Corners	32	0.18	5.8	0.1	3.2	0.07	2.2	0.05	1.6	0.03	1.0	0.02	0.6
Abffusors	24	0.82	19.7	0.90	21.6	1.07	25.7	1.04	25.0	1.05	25.2	1.04	25.0
Formedffusors	48	0.53	25.4	0.37	17.8	0.38	18.2	0.32	15.4	0.15	7.2	0.18	8.6
Drywall	224	0.29	65.0	0.10	22.4	0.05	11.2	0.04	9.0	0.07	15.7	0.09	20.2
Floor	48	0.15	7.2	0.11	5.3	0.10	4.8	0.07	3.4	0.06	2.9	0.07	3.4
Total absorption, sabins			123.1		70.3		62.1		54.4		52.0		57.8
Reverberation time, sec			0.15		0.27		0.30		0.35		0.36		0.33

Table 25-4 Absorption and Reverberation-Time Calculations for an Isolation Booth with Diffusers (Design Example 3)

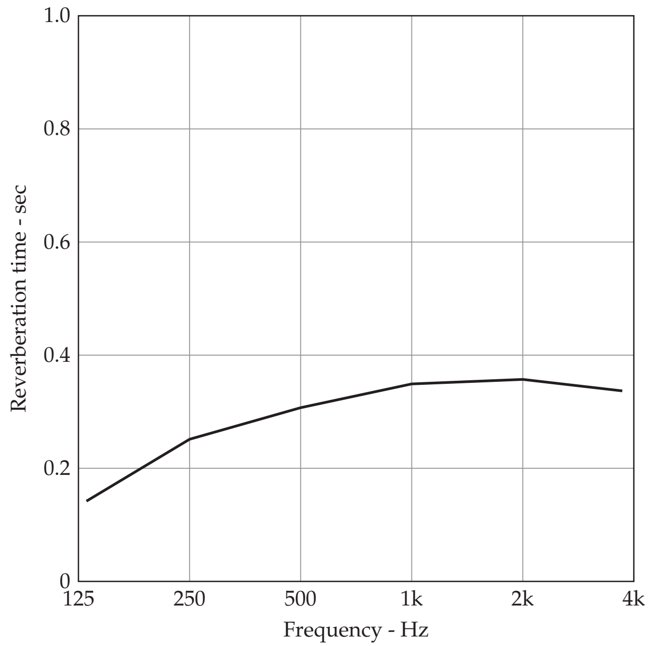


Figure 25-16 Reverberation time for an isolation booth (Design Example 3).

The first design example uses carpet and acoustical tile. The room has excessive mid- to high-frequency absorption and lacks diffusion. The sound field will suffer from an irregular low-frequency response although the diaphragmatic absorption of walls and floor may help to partly smooth the response. The mid- to high-frequency absorption will yield a very short reverberation time.

The second design example uses cylindrical traps as its sole treatment to provide absorption, as well as diffusion in conjunction with the reflective wall surfaces. The room construction provides additional absorption. The sound field will be satisfactory

Example	Acoustic Treatment	Normal Mode Treatment	Reverberation Time at 500 Hz	Microphone Placement Sensitivity	Announcer Hearing Own Voice
1	Acoustical tile, carpet	*(only)	0.1 sec	Poor	Needs headphones
2	Cylindrical traps	*plus Cylindrical traps	0.07 sec	No problem	Needs headphones
3	Diffusers	*plus Diffusers	0.3 sec	No problem	Natural

*Drywall and floor diaphragmatic absorption.

Table 25-5 Comparison of Design Examples 1, 2, and 3 with good diffusion, pleasant ambience, and overall smooth response. However, the reverberation time will be very short.

The third design example uses diffusing units as its sole treatment; however, some diffuser units as well as the room construction provide absorption. The sound field will be satisfactory with thorough diffusion, smooth response, and a reasonable reverberation time. The announcer should experience a sense of being in a larger space, the sound of the voice will be natural, and the microphone pickup should be relatively independent of microphone location.

Live End–Dead End Isolation Booth

An alternative to familiar isolation booth designs uses a live end–dead end (LEDE) design. This design uses the same approach as the live end–dead end design used in some control rooms where monitoring is performed, as discussed in Chap. 24. A live end–dead end isolation booth would require more space than a typical booth, although not much. The objective is to obtain a clean direct sound free from early reflections, followed by a normal ambient decay. The microphone is placed in the room's absorbent end so that no reflections reach it, except those delayed and diffused from the room's more distant live end. All walls of the dead end, as well as the floor there, must be absorptive. The observation window is placed in the live end of the room. A short initial time-delay gap will exist between the time of arrival of the direct sound and the arrival of the first highly diffused sound from the live end. A natural-sounding voice recording can be obtained with this somewhat unusual arrangement. This type of treatment could certainly be used in an isolation booth for the purpose of musical instrument recording.

Key Points

- In an isolation booth, because of the small dimensions, room modes are relatively high in frequency, few, and widely spaced. Room modes at voice frequencies must be analyzed and controlled if necessary. Rooms smaller than 1,500 ft^3 are almost certain to produce timbral anomalies and should be designed to minimize anomalies.
- Mid- and high-frequency absorption may be applied to control flutter echoes; care must be taken so that voice frequencies are not overly absorbed. Consideration should be given to potential reflections from glass.
- Anechoic conditions are unsuitable for an isolation booth unless the dry acoustics are ignored and artificial reverberation is added. Even then, unless only headphones are used, some natural ambience is needed to support the user's voice orientation and adjustment.
- Another design approach is to produce a relatively live sound in the booth. A high density of diffuse reflections would yield voice recordings that are accurate and clean-sounding.
- The traditional treatment in booth #1 comprises carpet and acoustical tile. The axial modes would produce an irregular frequency response; however, the diaphragmatic absorption of walls and floor may partly smooth the response; excessive mid- to high-frequency absorption would make the room too dead; diffusion is needed to improve spatial distribution of sound energy.

- The acoustical treatment in booth #2 comprises quarter-round and half-round cylindrical traps. The density of diffuse reflections from the many cylindrical surfaces would yield a natural ambience and smooth response. Because of the highly diffused sound field, recordings made at one microphone position would generally match those recordings made at different microphone positions.
- In booth #2, the reverberation time would be very short. An announcer would have little acoustical feedback from the room and the announcer's own voice would sound somewhat unnatural to himself or herself. This can be rectified with headphones and quality playback of the voice.
- The acoustical treatment in booth #3 comprises multiple types of diffusing elements to produce satisfactory small-room conditions with good diffusion, smooth response, and a reasonable reverberation time.
- In booth #3, because of the diffused sound and longer reverberation time, the announcer should experience a sense of being in a larger space, the sound of the voice will be natural, and the microphone response should be consistent over different positions.
- If booth space permits, a live end–dead end (LEDE) design approach could yield a clean direct sound free from early reflections, followed by a normal ambient decay. The microphone would be placed in the room's absorbent end.

CHAPTER 26

Acoustics of Audiovisual Postproduction Rooms

In a room dedicated to one or a few related functions, the acoustical treatment can be tailored to those few applications, achieving excellent results. However, in many cases, rooms must be used for a diversity of functions. It is not practical to design separate rooms for each function; instead, one room must be designed to work acoustically well, with some compromises, for many functions. This is often the case for small studios and workrooms used for audiovisual production and postproduction. For example, in these rooms, these activities may be considered: digital sampling, MIDI, editing, sound effects (Foley), dialog replacement, voice-over, signal processing, composing, video production, postrecording, and equipment evaluation.

As with larger studios, these rooms demand accuracy in the sound field and comfortable working conditions. However, these needs must often be met on a very tight budget. As such, the design of these workrooms is a challenge to any designer. With careful planning, room treatment can be devised to yield a postproduction room that is suitable for these and similar activities. On the other hand, such a room is not a substitute for an acoustically purpose-built room.

Design Criteria

Audiovisual postproduction rooms must support widely varying functions; however, their design must always consider several essential factors. Any function requiring audio evaluation, even if audio is not often being recorded, demands good general acoustics in terms of room frequency response, reverberation, diffusion, and so on. Room treatment must be designed to provide this. Most audio work will be performed by recording engineers, preferably monitoring their work with high-quality loudspeakers. Thus, the room must allow good audio playback in stereo or multichannel mode. Headphone monitoring would alleviate this requirement. These workrooms are primarily intended for postproduction work, hence recording of production sound would be rare. The primary need is that background noise be kept at an appropriately low level so that the engineer can evaluate sound from the monitors with no interference. Noise from outside the workroom must be considered; acoustical isolation suitable for the work is required. Conversely, other rooms must be isolated from loud sound inside the workroom. In addition, noise from inside the room must be considered;

noise emanating from production equipment (such as from computer cooling fans) or HVAC must be minimized.

The two room examples to follow illustrate how these design factors can be addressed. The examples take somewhat different approaches, to demonstrate that there is flexibility in the execution of any acoustical project. Also, although these rooms are intended to serve multiple purposes, the designs could easily form the basis of a room dedicated to one purpose; for example, they could be used as project recording studios.

Many of the criteria applicable to an audiovisual workroom are covered in earlier chapters. To avoid redundancy, references are made to appropriate earlier coverage. A review of this referenced material is very much a part of this chapter.

Design Example 1: Small Postproduction Room

This design example will propose a layout and relatively basic acoustical treatment for a small- to medium-sized postproduction room. The room dimensions, selected from Table 22-1, are a length of 18.5 ft, width of 15.3 ft, and height of 12 ft. This yields a floor area of 283 ft^2 and a volume of 3,397 ft^3. The room proportions of 1.00:1.28:1.54 are as close to "optimum" as is feasible (see Table 13-5 and Fig. 13-6).

Appraisal of Room Resonances

The room proportions are favorable in this example, but it is still important to verify axial mode spacing. Only axial modes are considered because the tangential modes are inherently reduced 3 dB, and the oblique modes are reduced 6 dB, with respect to the more powerful axial modes. The length, width, and height axial mode resonance frequencies to 300 Hz are listed in Table 26-1. These constitute the low-frequency acoustics of the room. Listing these frequencies in ascending order and noting the difference in frequency between adjacent modes gives a good appraisal of possible modal problems.

Spacings of 25.5 Hz and 30.5 Hz are the only ones exceeding Gilford's suggested limit of 20 Hz (see Chap. 13). Experience suggests that these two spacings are possible sources of frequency-response deviations. However, this is by no means a major flaw of the room. Favorable room proportions can only minimize the potential for modal problems, not eliminate them.

Most room resonances can be controlled with bass traps. In this example, some bass trap absorption in the 50- to 300-Hz region may be necessary. If the audiovisual work space is of frame and drywall construction, an impressive amount of low-frequency absorption will be inherently present in the frame structure. The floor and wall diaphragms vibrate, and absorb low-frequency sound in the process. This is demonstrated in subsequent calculations. However, additional low-frequency absorption might be needed.

Proposed Treatment

The room volume of 3,397 ft^3 exceeds the usual "small room" category and allows the room's acoustics to approach the condition of thoroughly mixed sound. Therefore, the concept of reverberation time can be applied accurately in this room. We use the

Room Dimensions = 18.5 × 15.3 × 12.0 ft					
	Axial Mode Resonances			Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
	Length $L = 18.5$ ft $f_1 = 565/L$ (Hz)	Width $W = 15.3$ ft $f_1 = 565/W$ (Hz)	Height $H = 12.0$ ft $f_1 = 565/H$ (Hz)		
f_1	30.5	36.9	47.1	30.5	6.4
f_2	61.1	73.9	94.2	36.9	10.2
f_3	91.6	110.8	141.3	47.1	14.0
f_4	122.2	147.7	188.3	61.1	12.8
f_5	152.7	184.6	235.4	73.9	17.7
f_6	183.2	221.6	282.5	91.6	2.6
f_7	213.8	258.5	329.6	94.2	16.6
f_8	244.3	295.4		110.8	11.4
f_9	274.9	332.4		122.2	19.1
f_{10}	305.4			141.3	6.4
				147.7	5.0
				152.7	30.5
				183.2	1.4
				184.6	3.7
				188.3	25.5
				213.8	7.8
				221.6	13.8
				235.4	8.9
				244.3	14.2
				258.5	16.4
				274.9	7.6
				282.5	12.9
				295.4	10.0
				305.4	

Mean axial mode spacing: 12.0 Hz.
Standard deviation: 7.1 Hz.

TABLE 26-1 Axial Modes of a Small Postproduction Room (Design Example 1)

Sabine equation to calculate an estimate of the absorption needed in the room:

$$RT_{60} = \frac{0.049V}{A} \tag{26-1}$$

where RT_{60} = reverberation time, sec
V = volume of room, ft^3
$A = S\alpha$ = total absorption of room, sabins

Assuming a desired reverberation time of 0.3 sec, the amount of absorption can be calculated: $A = 0.049V/\mathrm{RT}_{60} = (0.049) \times (3{,}397)/0.3 = 555$ sabins. This is an approximate absorption value that will result in reasonable acoustical performance, and it can be varied at a later time to meet specific needs.

The proposed treatment to provide the required absorption in the audiovisual room is shown in Fig. 26-1. This is a fold-out plan in which the four walls, graphically hinged along the edges of the floor, are laid out flat. It is assumed that the room is of frame construction, with a wooden floor, 1/2-in drywall on all walls, and carpet covering the floor. The drop ceiling panels are glass fiber. The materials needed to supply absorption are described in Table 26-2. Moreover, this table shows the values needed to determine the absorption contributed by each element, at six standard frequencies, using absorption coefficients for the different materials found in App. C.

Some of the treatment materials shown in Table 26-2 are less common. These elements include polycylindrical diffusing/absorbing modules, a bass trap under the polycylindrical modules, and diffuser modules on one wall and in the ceiling.

Polycylindrical modules are inexpensive to build and their construction is relatively simple (see Figs. 12-26 to 12-28). A polycylindrical module can be reasonably effective as a diffuser; a module can yield a normal incidence diffusion pattern somewhat similar to that of a quadratic residue diffuser (see Fig. 14-21). Instead of polycylindrical

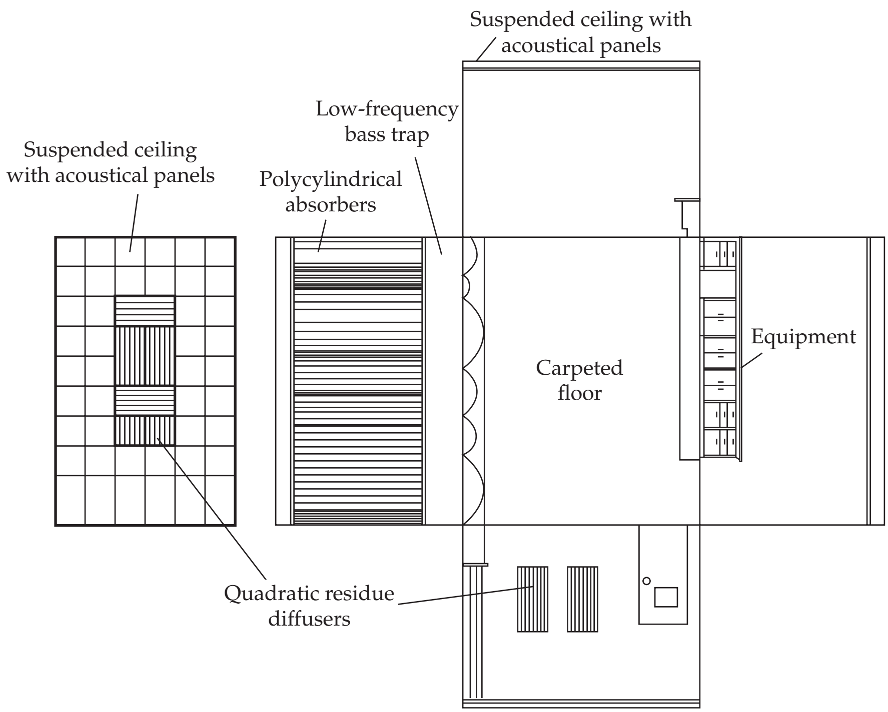


FIGURE 26-1 Proposed treatment of a small postproduction room (Design Example 1). Elements include carpeted floor, suspended ceiling with acoustical tile, low-frequency Helmholtz resonator bass trap, polycylindrical sound absorbers, and quadratic residue diffuser panels.

Material	Area (ft²)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	Sα	α	Sα	α	Sα	α	Sα	α	Sα	α	Sα
Drywall	812	0.10	81.2	0.08	65.0	0.05	40.6	0.03	24.4	0.03	24.4	0.03	24.4
Wood floor	284	0.15	42.6	0.11	31.2	0.10	28.4	0.07	19.9	0.06	17.0	0.07	19.9
Drop ceiling	234	0.69	161.5	0.86	201.2	0.68	159.1	0.87	203.6	0.90	210.6	0.81	189.5
Carpet	284	0.08	27.7	0.24	68.3	0.57	161.9	0.69	196.0	0.71	201.6	0.73	207.3
Polycylindricals	148	0.40	59.2	0.55	81.4	0.40	59.2	0.22	32.6	0.20	29.6	0.20	29.6
Bass trap	37	0.65	24.1	0.22	8.1	0.12	4.4	0.10	3.7	0.10	3.7	0.10	3.7
Diffusers	50	0.48	27.8	0.98	49.0	1.2	60.0	1.1	55.0	1.08	54.0	1.15	57.5
Total absorption, sabins			424.1		504.1		513.6		535.2		540.9		531.9
Reverberation time, sec			0.39		0.33		0.32		0.31		0.31		0.31

Table 26-2 Absorption and Reverberation-Time Calculations for a Small Postproduction Room (Design Example 1)

modules, this wall could be covered with quadratic residue diffuser modules, which would provide good diffusion as well as absorption. Cost should determine which to use. If labor costs to build the diffuser modules are too high, factory-built modules may be preferred.

The bass trap beneath the polycylindrical modules is a simple perforated panel Helmholtz-type absorber tuned to give peak absorption in a low-frequency band. It can be covered with an open weave material for appearance. Before the bass trap is constructed, but after the other treatments are installed, the room could be measured. The bass trap could then be designed with its peak absorption frequency optimally selected. Full design particulars for a perforated panel bass trap resonator are given in Chap. 12. The bass trap might not be sufficient to control low-frequency modes in the room. If more bass absorption is needed, the two corners opposite the door could be treated with one of the four types of corner absorbers (see Fig. 20-4).

Two quadratic residue diffuser modules with a total area of 16 ft^2 are mounted, spaced apart, on the wall near the door to discourage a potential lengthwise flutter echo. In addition, diffuser modules with a total area of 50 ft^2 are laid into the suspended ceiling framework. These diffuser modules are described in detail in Chap. 14.

Design Example 2: Large Postproduction Room

This design example will propose a layout and acoustical treatment for a relatively large audiovisual postproduction room. In the design of this room, attention is given to eliminating early reflections at the listening position so that wall-mounted monitors can be trusted to convey an accurate sound field. As we have seen, early reflections are problematic, and require treatment to avoid their deteriorating effect. Although the room design prioritizes wall-mounted monitors, near-field monitors can additionally be used.

The dimensions selected for the room are a length of about 28 ft excluding the irregular space behind the loudspeakers, width of 19.2 ft, and height of 12 ft. The irregular shape yields a working floor area of about 510 ft^2 and a volume of about 6,150 ft^3. The proportions are 1.00:1.60:2.33. This follows Sepmeyer's C ratio (see Table 13-5).

Appraisal of Room Resonances

The room proportions dictate axial mode spacing which in turn dictates the smoothness of the room's low-frequency response. The length, width, and height axial mode resonance frequencies to 300 Hz are listed in Table 26-3. The mean (average) axial mode spacing is 9.5 Hz, although the real differences vary from 0 to 20.2 Hz. A near-coincidence occurs at both 141 Hz and 282 Hz, and a definite coincidence at 235 Hz.

Only the 141-Hz coincidence threatens an audible timbral anomaly; the other coincidences are high enough in frequency to avoid degrading effects. The 141-Hz coincidence is associated with both the 12-ft ceiling height and the 28-ft length of the room. The chances of the 141-Hz anomaly being audible are slight because the tangential and oblique modes (even though lower in energy) will tend to fill in the gaps between the axial modes. An advantage of a larger room like this is that problems of mode spacing tend to recede.

Monitor Loudspeakers and Early Sound

The general plan of the room is shown in Fig. 26-2. A workstation with monitor loudspeakers and a small mixing console is placed along the front (north) wall. A workbench and an array of reflection phase-grating diffusers are placed along the

Room Dimensions = 27.96 × 19.2 × 12.0 ft					
	Axial Mode Resonances			Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
	Length $L = 27.96$ ft $f_1 = 565/L$ (Hz)	Width $W = 19.2$ ft $f_1 = 565/W$ (Hz)	Height $H = 12.0$ ft $f_1 = 565/H$ (Hz)		
f_1	20.2	29.4	47.1	20.2	9.2
f_2	40.4	58.9	94.2	29.4	11.0
f_3	60.6	88.3	141.3	40.4	6.7
f_4	80.8	117.7	188.3	47.1	11.8
f_5	101.0	147.1	235.4	58.9	1.7
f_6	121.2	176.6	282.5	60.6	20.2
f_7	141.5	206.0	329.6	80.8	7.5
f_8	161.7	235.4		88.3	5.9
f_9	181.9	264.8		94.2	6.8
f_{10}	202.1	294.3		101.0	16.7
f_{11}	222.3			117.7	3.5
f_{12}	242.5			121.2	20.1
f_{13}	262.7			141.3	0.2
f_{14}	282.9			141.5	5.6
f_{15}	303.1			147.1	14.6
				161.7	14.9
				176.6	5.3
				181.9	6.4
				188.3	13.8
				202.1	3.9
				206.0	19.3
				222.3	13.1
				235.4	0.0
				235.4	7.1
				242.5	20.2
				262.7	2.1
				264.8	17.7
				282.5	0.4
				282.9	11.4
				294.3	8.8
				303.1	

Mean axial mode spacing: 9.5 Hz.
Standard deviation: 6.4 Hz.

TABLE 26-3 Axial Modes of a Large Postproduction Room (Design Example 2)

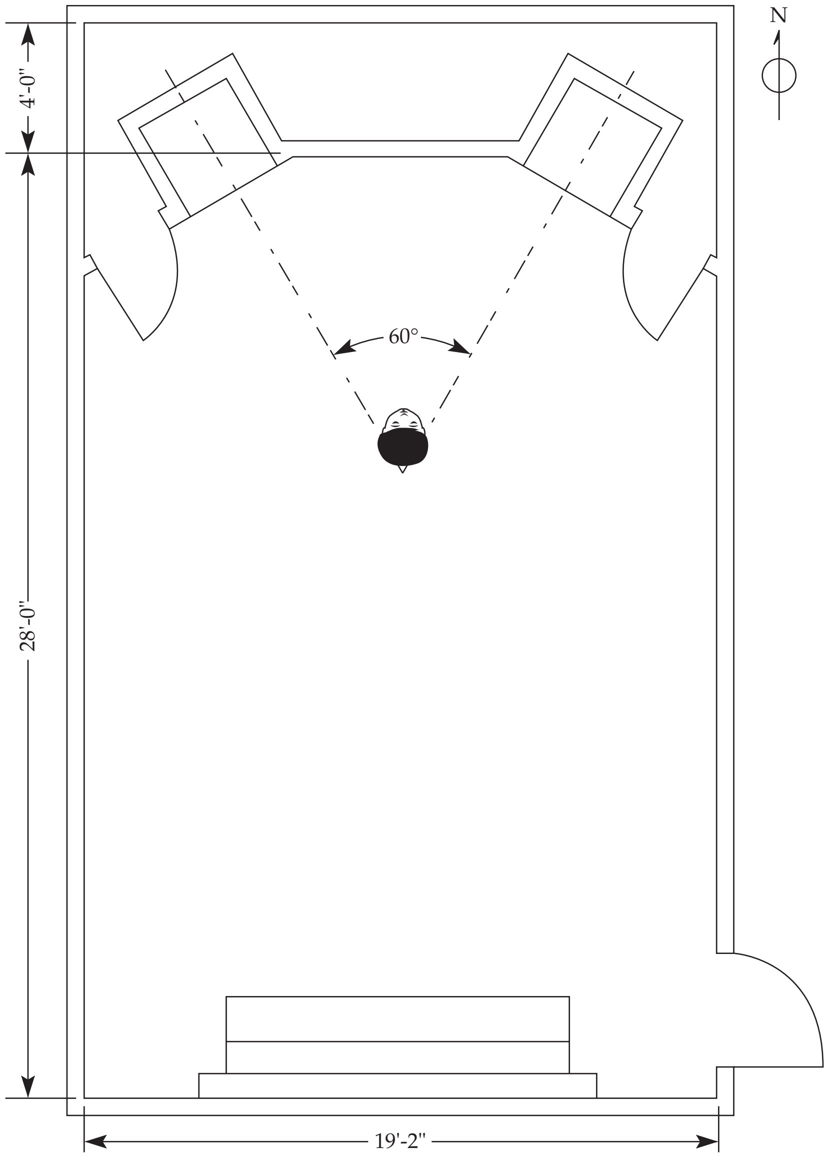


FIGURE 26-2 Plan view of a large postproduction room (Design Example 2) showing placement of built-in monitors and workstation on the rear wall.

rear (south) wall. To give the mixing engineer the best possible sound from the monitors, two soffits are centered on 60° lines converging slightly behind the listening position. This ensures that the direct sound from the monitors passes by the listener's ears. The function of the soffits is to place the faces of the monitors flush with the angled wall. The corners of a stand-alone loudspeaker cabinet radiate sound through a wide angle by diffraction from the corners. This diffracted sound from the loudspeaker edges contributes to the early sound problem directly, and indirectly by reflections from the wall behind the loudspeakers. By making the faces of the monitor cabinets flush with the wall in which they are set, the diffraction effects from this source are eliminated.

Loudspeaker-cabinet diffraction issues can be minimized by flush-mounting the cabinets. However, the room creates other time-delay problems that must be addressed. A consideration of the favorable and unfavorable sound paths from the monitors is shown in Fig. 26-3. The solid lines indicate direct rays arriving at the listening position

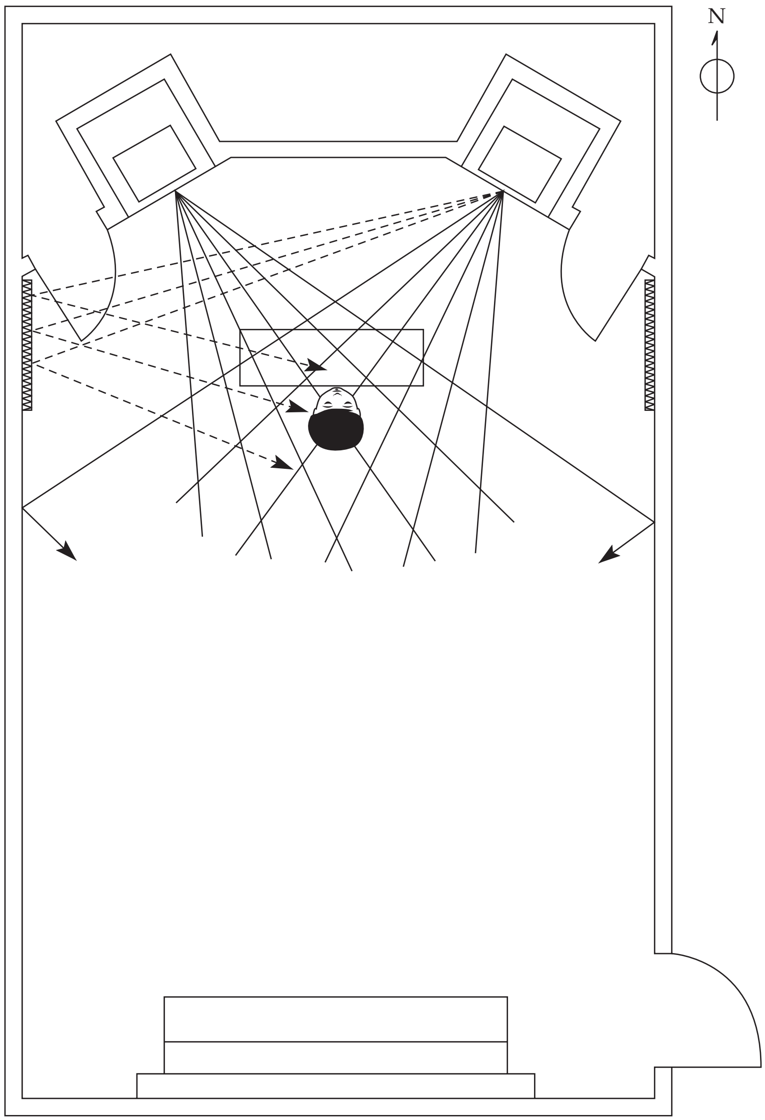


Figure 26-3 Plan view of a large postproduction room (Design Example 2) showing direct sound versus early reflections at the mixing position.

as desirable sound. The broken lines are reflected from the walls, floor, and ceiling. Because they arrive at different times, they result in comb-filter distortion. These reflections can be controlled by locating areas of absorbent material at strategic positions on the side walls and on the ceiling over the mixing engineer. Alternatively, the reflections could be controlled by placing diffusers in these same locations. For this design example, absorption patches will be used. Reflections from the floor are absorbed by a carpet. One early-arrival reflection comes from the face of the mixing console; this is unavoidable.

Late Sound

After the initial time-delay gap problems are minimized by the use of soffits and patches of absorbent or diffusers, the direct sound from the monitors should be free of timbral changes produced by early reflections. This direct sound allows accurate perception of the monitor playback, but this is only the first part of the complete sound field. The late sound, made up of reflections from the rear of the room, completes the sound field and must be considered.

A view of both the early and the late sound is shown in Fig. 26-4. The direct sound reaching the ears of the engineer at the workstation establishes both the zero level and the zero time of this graph. Following the arrival of the direct sound, a period of relative silence occurs as the sound passes the engineer on its way to the rear wall. This is not a completely silent period; there are always minor reflections of low level but they can be neglected. These miscellaneous reflections are down perhaps 30 dB, and hence not significant in the overall perception.

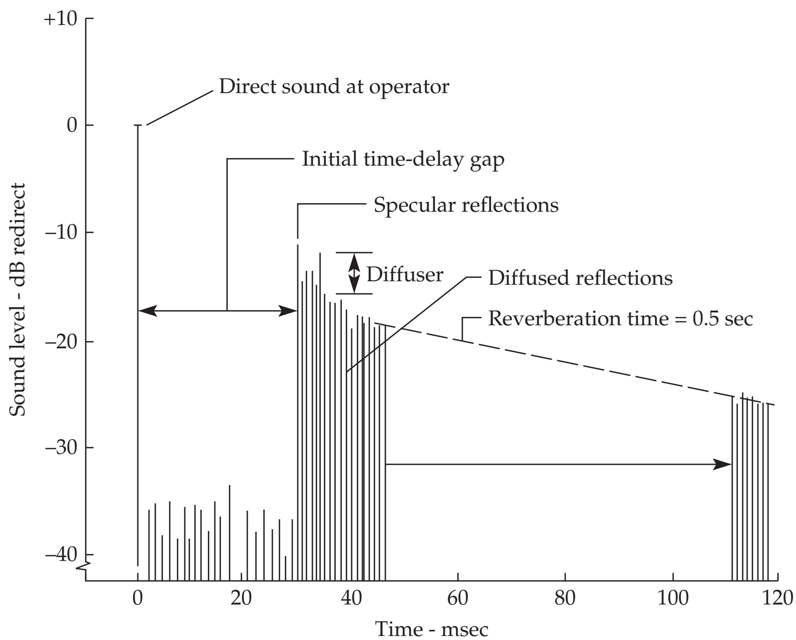


FIGURE 26-4 Early reflections and reverberation in a large postproduction room (Design Example 2) showing the initial time-delay gap.

Using the room dimensions, we can estimate the level and time values of the late reflections. The direct sound must travel about 9 ft from the loudspeakers to the engineer, and depending on the path, must travel an additional 34 to 40 ft to return to the engineer as reflections from the rear wall and the face of the rear workbench. The level of the reflection from the rear wall will be about 20 log 9/40 = 12.9 dB and the level of the reflection from the workbench will be about 20 log 9/34 = 11.5 dB. These rough estimates, based on inverse-square law propagation, are accurate enough for present purposes.

The delays of these rear reflections are about 34 ft/1,130 ft/sec = 30 msec, and 40 ft/1,130 ft/sec = 35 msec. Therefore, the reflection from the workbench is 11.5 dB with 30 msec of delay, and the reflection from the rear wall is 12.9 dB with 35 msec delay. These specular reflections are also shown in Fig. 26-4.

An array of reflection phase-grating diffusers is located on the rear wall. Various types of diffuser units can be used; in this design, they will diffuse sound in both the horizontal and vertical directions. Due to normal diffuser action, the level of the return from the diffusers will be 6 to 8 dB less than what falls on the face of the diffusers. The flood of rear-wall reflections, both specular and diffused, decays at a rate determined by the reverberation time of the room. Our goal is a reverberation time of about 0.5 sec, and that is plotted in Fig. 26-5.

A substantial portion of the sound falling on the rear wall hits the diffusers and is diffused in both the horizontal and the vertical directions. The reflections from the workbench and the rear wall not covered with diffusers combine with the diffused sound and return to the engineer's ears delayed in time.

To summarize the theory of this design: the direct sound from the monitors sweeps past the engineer to the rear wall and workbench and returns to the engineer's ears, arriving about 30 to 35 msec later than the direct sound. The initial time-delay gap in the room characteristic is very important to sound quality, enabling the engineer to clearly hear the direct sound and also natural room ambience that does not degrade the accuracy of the direct sound. Moreover, because of the Haas effect, the engineer does not hear the ambience as a separate event, and instead localizes the ambience as coming from the front of the room with the direct sound.

Proposed Treatment

Starting with no specific room treatment, the major absorption component is the 1,148 ft^2 of gypsum drywall with its diaphragmatic absorption. The gypsum-board absorption peak is in the low-frequency region, thus it is logical to consider adding heavy carpet with pad (with its primary absorption in the higher frequencies) to compensate for it. This justifies placing carpet over the entire floor surface. The only other absorbing areas, and these are of minor effect because of their small size, are the two wall and one ceiling 4 × 4-ft panels to intercept the early reflections. The absorption for each material (gypsum board drywall, carpet with pad, and small panels) is $A = S\alpha$ where S is the surface area and α is the material's absorption coefficient. As noted, excluding the irregular space behind the loudspeakers, the room volume is about 6,150 ft^3. The Sabine equation is used to compute the reverberation times at six standard frequencies. This data is collected in Table 26-4. The reverberation times are also plotted in Fig. 26-5.

The gypsum board yields a 250-Hz reverberation time of 1.03 sec; this is too high, given the desired reverberation time of 0.5 sec. However, the reverberation times at

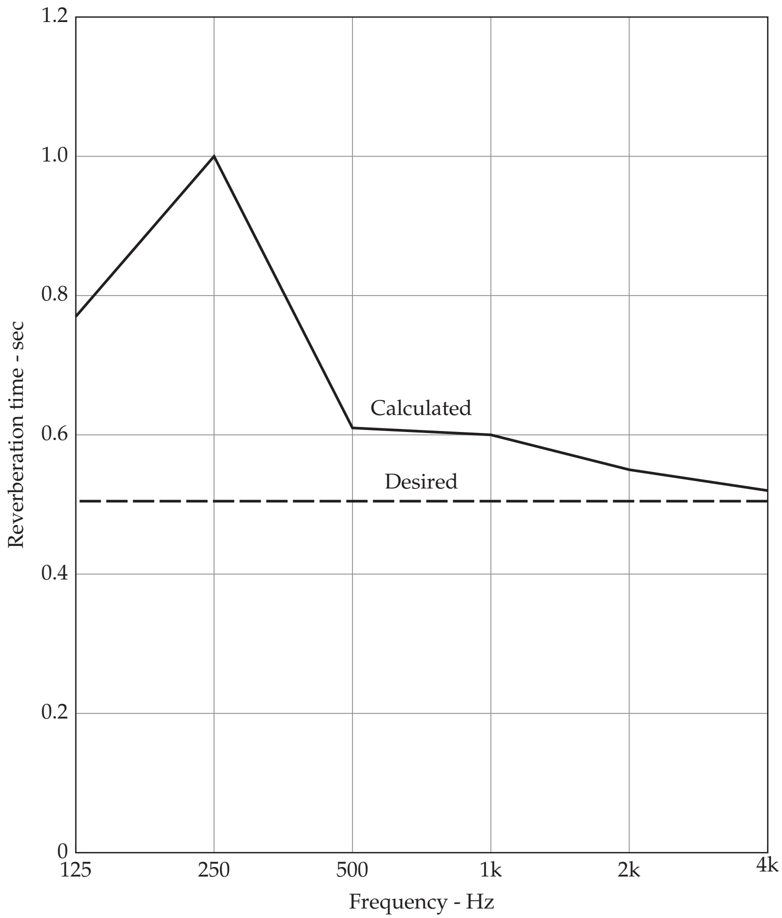


Figure 26-5 Calculated reverberation times for the large postproduction room (Design Example 2) from Table 26-4, and the desired design goal.

500 Hz and above are not excessive. About 400 ft^2 of Helmholtz resonators tuned to 250 Hz would decrease the 1.03-sec peak time closer to 0.5 sec, but this modification would be unwise at this stage. There are too many uncertainties in gypsum board construction to be that specific. A wiser approach would be to wait until the structure is built and the carpet laid, and then to measure the reverberation time to see precisely what low-frequency correction is needed. The correction will be relatively small and the treatment can be specifically tuned as needed.

A goal of a reverberation time of 0.5 sec has been indicated and can be achieved, but the client might very well prefer 0.4 sec (a slightly dryer acoustic) or 0.6 sec (a slightly livelier acoustic). Such a preference and the measured values would provide the basis for the optimal correction.

Material	Area (ft^2)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	Sα	α	Sα	α	Sα	α	Sα	α	Sα	α	Sα
Drywall	1,148	0.29	332.9	0.10	114.8	0.05	57.4	0.04	45.9	0.07	80.4	0.09	103.3
Carpet, heavy 40-oz pad	589	0.08	47.1	0.24	141.4	0.57	335.7	0.69	406.4	0.71	418.2	0.73	430.0
Wall and ceiling panels, 2-in flat on wall	48	0.24	11.5	0.77	37.0	1.13	54.2	1.09	52.3	1.04	49.9	1.05	50.4
Total absorption, sabins			391.5		293.2		497.3		504.6		548.5		583.7
Reverberation time, sec			0.77		1.03		0.61		0.60		0.55		0.52

TABLE 26-4 Absorption and Reverberation-Time Calculations for a Large Postproduction Room (Design Example 2)

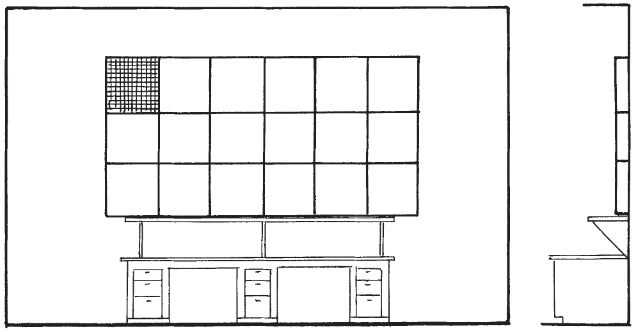


Figure 26-6 South elevation of a large postproduction room (Design Example 2) showing the workbench and diffusers.

Workbench

Figure 26-6 shows the south elevation of the room; the rear workbench is placed there, and the wall is largely covered by an array of 18 diffusing modules each measuring 2×2 ft. Beneath this diffusing area, a workbench is suggested. Any reflections from this bench will simply add to the rear wall specular and diffuse reflection mixture. If placed anywhere else in the room, reflections from the bench would clutter the initial time-delay gap with spurious reflections. Moreover, if placed along one wall, reflections from the bench would compromise the bilateral symmetry of the room's acoustics.

The engineer at the workstation may need an assistant to do the many jobs to keep the workstation efficiently occupied. Certain equipment will be necessary to make this assistant efficient. A shelf at the bottom of the diffusing panel is intended to hold such equipment. A few such pieces will have a negligible effect on the functioning of the wall diffusers.

Mixing Engineer's Workstation

Figure 26-7 shows the north elevation of the workroom. This drawing details the position of the mixing engineer to the monitor loudspeakers and the video display. The line of sight is high enough that there is room for equipment without blocking the view, but care must be exercised in this regard. In particular, the monitors and display should not be placed too high relative to the sitting position of the engineer.

Racks of auxiliary equipment can be mounted under the desk on either side of the engineer's feet. Unless absolutely necessary, another workstation should not be placed behind the engineer; its forward face and equipment on it such as monitors would send comb filtering reflections back to the engineer's ears. Doors have been suggested to make the space behind the monitors available for storage. Figure 26-8 presents a side view of the engineer's position along the west elevation. The relative position of the side wall and ceiling early-sound panels is shown.

The monitor loudspeakers should be flush-mounted in the front wall; this would avoid a potential source of comb-filter reflections and is an important element in this room design. The monitor loudspeakers could be mounted vertically, or inclined downward to place the engineer on the axis of the monitor. An on-axis downward inclination

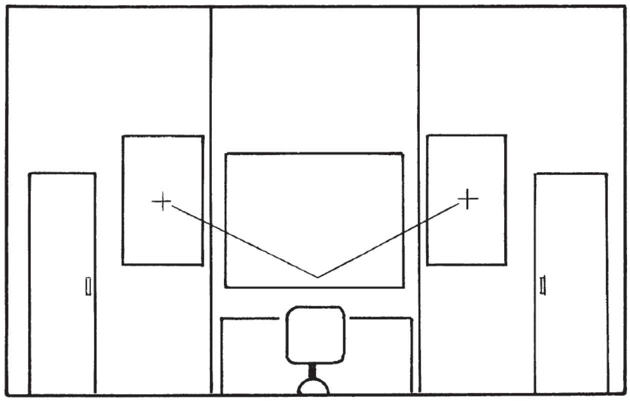


Figure 26-7 North elevation of a large postproduction room (Design Example 2) showing placement of monitors.

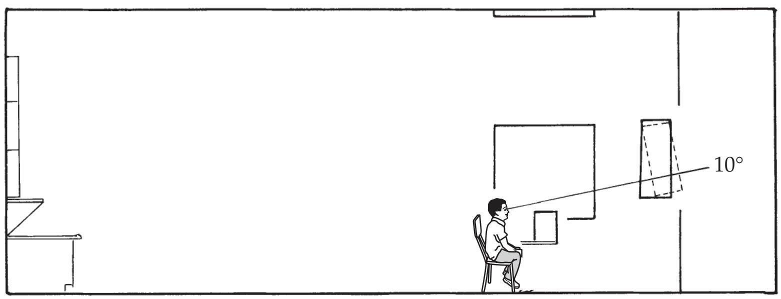


Figure 26-8 West elevation of a large postproduction room (Design Example 2) showing downward inclination of the monitors and ceiling absorber panel.

is preferable. The partition could be made in two sections, one inclined at the same horizontal and the same vertical angle as the face of the left loudspeaker, and the other inclined to coincide similarly with the face of the right loudspeaker. This would cause some minor problems with the doors and the video display which could be resolved. As noted, on-axis mounting is preferable, but in high-quality monitors sound 10° off-axis is not significantly degraded. The simplest design would accept vertical mounting and 10° off-axis sound.

Video Display and Lighting

A possible video display position is indicated in the north elevation (Fig. 26-7). A flat-panel display could be easily mounted on the front wall. Alternatively a video projector could be placed near the engineer's desk. The projector, video monitors, and other equipment on the desk are all possible culprits in producing early sound reflections that could distort the sound the engineer hears. As a last resort, absorbing materials over the top and back of desk equipment might be necessary to control these reflections. Also, ambient noise from the cooling fans in a projector or computers must be minimized.

It is suggested that the ceilings and upper walls be painted flat black and that shaded light fixtures be hung from the ceiling. Track lighting to highlight the rear workbench and diffusers could provide sufficient working illumination as well as a dramatic touch. Similar track lighting could provide working illumination at the workstation.

Key Points

- An audiovisual postproduction room must support widely varying functions; while the room can be designed to work acoustically well, there must be compromises. Such a room is not a substitute for an acoustically purpose-built room.
- The room design must provide treatment that allows monitoring with high-quality loudspeakers, and provide measures to ensure a quiet and acoustically isolated space.
- If the audiovisual room is of frame and drywall construction, considerable low-frequency absorption will be inherently present in the diaphragmatic structure. However, additional low-frequency absorption might be needed.
- Treatment in audiovisual room #1 is designed to provide a reverberation time of about 0.3 sec. Design elements could include polycylindrical diffusing/absorbing units, a bass trap, and diffuser modules in the ceiling and walls.
- Treatment in audiovisual room #2 allows for wall-mounted monitors while minimizing early reflections. Late reflections are mainly diffused and yield a correct initial time-delay gap.
- Treatment in audiovisual room #2 is designed to provide a reverberation time of about 0.5 sec. To achieve this, additional low-frequency absorption should be added according to measurements taken in the room.
- In audiovisual room #2, the monitor loudspeakers should be flush-mounting to avoid a potential source of comb-filter reflections. The monitors could be mounted vertically, or inclined downward to place the engineer on the axis of the monitor; the latter is preferred.

CHAPTER 27

Acoustics of Teleconference Rooms

For many businesses, conference rooms are an essential component of their corporate infrastructure. Conference rooms can host meetings with in-house employees, and with employees and clients visiting from other locations. However, for the latter, the expense of travel and loss of time required for face-to-face meetings naturally encourages teleconferencing. This has promoted the development of many types of audio/video communication systems designed to bring people together electronically irrespective of distance, and at a low cost.

The availability of audio/video communication systems is one part of a teleconferencing solution; also needed is an acoustically suitable space in which to use them. A dedicated teleconference room can mean the difference between an unprofessional makeshift hookup and a professional business environment. From an acoustical standpoint, the goal of a teleconference room is to connect people to another or many locations on a global basis, in a professional setting, while ensuring comfortable conversations with natural and understandable speech quality.

Design Criteria

Speech intelligibility is the single most important requirement in a meeting room of any kind. When speech is conveyed over data lines, local acoustical intelligibility is an even greater concern because of distortion caused by the transmission channel or present at the remote location. Given a clear channel, and satisfactory local equipment, good room acoustics at both ends of the communication line will help to optimize speech intelligibility.

Speech is most intelligible in acoustically dead spaces. Conversely, intelligibility is very poor in highly reverberant spaces. However, adequate acoustical design of a teleconference room involves many things other than sound absorption. The background noise level must also be low. Specifically, it is well to keep the background noise level below the NCB-20 contour (see Fig. 19-1). To achieve this low level of background noise, attention must, among other things, be given to HVAC (heating, ventilating, air conditioning) noise. A low-velocity HVAC system is necessary, along with wrapped and lined ducts, and avoidance of certain fittings such as noisy air diffusers. Nearby noisy operations external to the teleconference room and the sound attenuation of the walls of the room must be brought into conformance. Once these considerations are attended to, attention is directed to the treatment of the interior of the teleconference room.

Shape and Size of the Room

The size of the teleconference room must be determined by the number of participants to be accommodated at one time. For 12 people plus the director, a space measuring 21 ft × 14.42 ft × 9 ft is selected for this design example; this fits the room proportions of 1.00:1.60:2.33. Because the room will be used for speech, and not music, it is not overwhelmingly important to achieve a flat low-frequency response. Rather, the room proportions should be selected to ensure good modal distribution at relatively higher speech frequencies. A listing of the normal modes of this space is given in Table 27-1.

Room Dimensions = 21.0 ft × 14.42 ft × 9.0 ft					
	Axial Mode Resonances			Arranged in Ascending Order (Hz)	Axial Mode Spacing (Hz)
	Length $L = 21.0$ ft $f_1 = 565/L$ (Hz)	Width $W = 14.42$ ft $f_1 = 565/W$ (Hz)	Height $H = 9.0$ ft $f_1 = 565/H$ (Hz)		
f_1	26.9	39.2	62.8	26.9	12.3
f_2	53.8	78.4	125.6	39.2	14.6
f_3	80.7	117.5	188.3	53.8	9.0
f_4	107.6	156.7	256.1	62.8	15.6
f_5	134.5	195.9	313.9	78.4	2.3
f_6	161.4	235.1		80.7	26.9
f_7	188.3	274.3		107.6	9.9
f_8	215.2	313.5		117.5	8.1
f_9	242.1			125.6	8.9
f_{10}	269.0			134.5	22.2
f_{11}	296.0			156.7	4.7
				161.4	26.9
				188.3	0.0
				188.3	7.6
				195.9	19.3
				215.2	19.9
				235.1	7.0
				242.1	9.0
				251.1	17.9
				269.0	5.3
				274.3	21.7
				296.0	

Mean axial mode spacing: 12.8 Hz.
Standard deviation: 7.8 Hz.

Table 27-1 Axial Modes of a Teleconference Room

The values in the mode spacing column, showing the calculated frequency distance between axial modes, appear reasonable, with a single degeneracy at 188.3 Hz. The chance that this degeneracy would cause an audible anomaly is small because experience has shown that few anomalies are detected at this relatively high frequency. Furthermore, the presence of tangential and oblique modes will tend to minimize the audible effect of the degeneracy at this relatively high frequency. The volume of the room is 2,725 ft^3.

Floor Plan

Figure 27-1 shows the floor and ceiling plans of the suggested teleconference room and many of its acoustical elements: loudspeakers (1), video display (2), Skyline diffusers (3), Modex Corner low-frequency membrane absorbers (4), Helmholtz low-frequency absorbers (5), and carpet (6).

A conference table with a wedge shape is suggested so that each seated participant has a reasonably good view of the video display or any activity at the head of the table. The 30-in-high shelf at the front of the room supports loudspeakers as well as a flat-panel display. Alternatively, the loudspeakers could be wall-mounted, and a retractable projection screen could be installed in the ceiling. Sliding doors make the space underneath this shelf available for storage. The 30-in-high shelf continues along the sides of the room. To eliminate any possibility of flutter echoes between the east and west walls, and to provide supplemental diffusion for the room, 14 Skyline modules are mounted on each side above the shelf. These are omnidirectional diffusers based on a primitive root using 156 unique block heights. Skyline modules are manufactured by RPG Acoustical Systems.

Ceiling Plan

The room uses a standard T-frame suspended ceiling with a 16-in airspace to conform to the standard C-400 (400 mm) mounting, as shown in Fig. 27-1. In the center of this

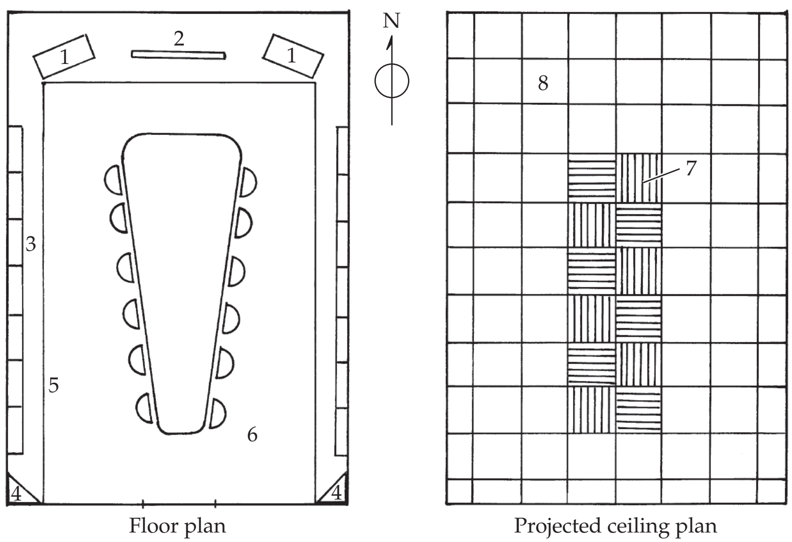


Figure 27-1 Floor and ceiling plans for a teleconference room.

frame, directly over the conference table, are 12 Abffusor units (7) which both diffuse and absorb sound. The rest of the ceiling rack is filled with Tectum Lay-In ceiling panels (8) having the dimensions 1-1/2 × 24 × 24 in. An overhead video projector can be mounted here if desired. Abffusor modules are manufactured by RPG Acoustical Systems. Tectum panels are manufactured by Armstrong World Industries.

Elevation Views

The north, east, south, and west elevation sketches are shown in Fig. 27-2. These sketches help to relate the two types of diffusers. Because reverberation time must be very short to ensure speech intelligibility, all available wall space between the 30-in-high shelf top and the suspended ceiling is covered with Tectum wall panel on a D-20 mounting (mounted on 3/4-in furring strips). The Skyline modules (3) are cemented to this Tectum wall panel.

Referring to the south elevation shown in Fig. 27-2, two Modex Corner membrane sound absorbers (4) are placed on the 30-in-high shelf on either side of the door. These help control potential low-frequency anomalies caused by low-frequency room modes. If such effects persist, two similar stacks of the same bass units should be placed in the corners behind the loudspeakers. It is not expected that additional units will be needed; because speech intelligibility is the principal concern, the room's low-frequency response is not critical. Modex Corner bass absorbers are manufactured by RPG Acoustical Systems.

A partial sectional view through a wall is shown in Fig. 27-3. The 30-in-high shelf along the side walls is only about 12-in wide to avoid intruding on the limited space of the room. On both sides of the room, a Helmholtz low-frequency absorber having an absorption peak at about 250 Hz is mounted beneath this shelf. This type of volume resonator absorber is discussed in Chap. 12. Its purpose is described below.

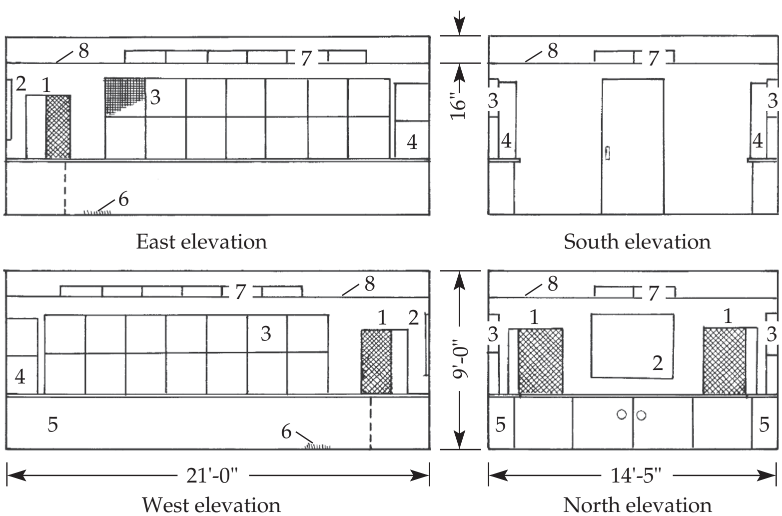


Figure 27-2 Four elevation views of a teleconference room.

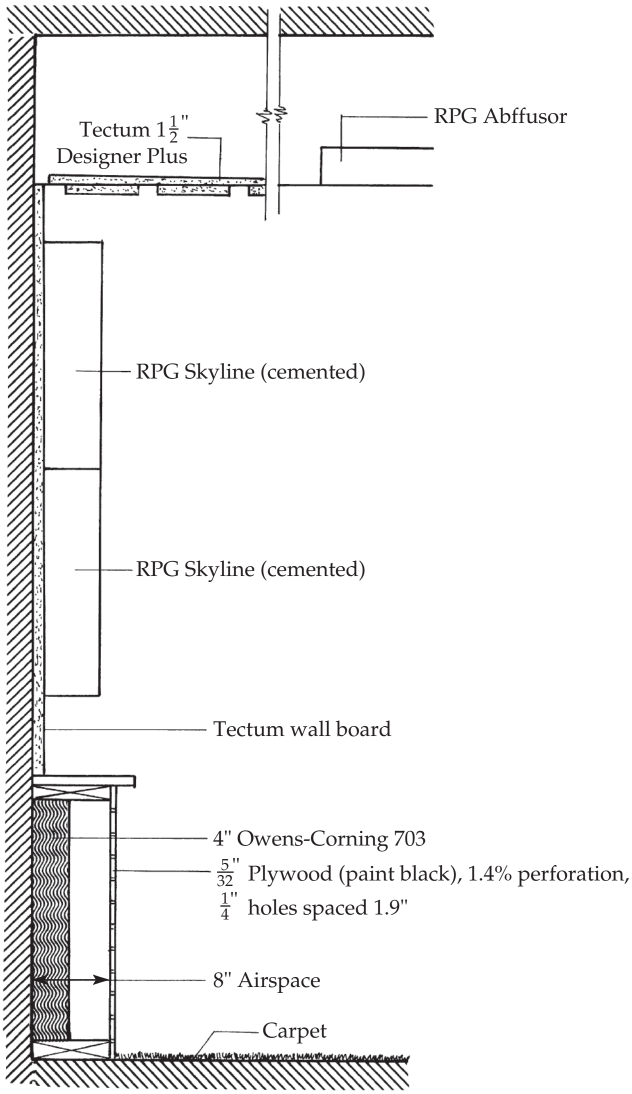


FIGURE 27-3 Wall section for a teleconference room.

Reverberation Time

Reverberation-time calculations for the teleconference room are shown in Table 27-2. The absorption units (sabins) are shown for six standard frequencies for the following elements: (a) 12 ceiling-mounted Abffusors, (b) Tectum Lay-In ceiling panels, (c) Tectum wall panels, and (d) heavy carpet with pad. These calculations lead directly to the plotted graph of Fig. 27-4 labeled "without compensation."

Material	Area (ft^2)	125 Hz		250 Hz		500 Hz		1 kHz		2 kHz		4 kHz	
		α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$	α	$S\alpha$
Drywall, ½-in on 16-in centers	940	0.29	272.6	0.10	94.0	0.05	47.0	0.04	37.6	0.07	65.8	0.09	84.6
Abffusor, ceiling	48	0.82	39.4	0.90	43.2	1.07	51.4	1.04	49.9	1.05	50.4	1.04	49.9
Tectum, ceiling, C-400 mounting	246	0.35	86.1	0.42	103.3	0.39	95.9	0.51	125.5	0.72	177.1	1.05	258.3
Carpet, heavy w/pad	203	0.08	16.2	0.27	54.8	0.39	79.2	0.34	69.0	0.48	97.4	0.63	127.9
Tectum walls, D-20 mounting	360	0.07	25.2	0.15	54.0	0.36	129.6	0.65	234.0	0.71	255.6	0.81	291.6
Total absorption, sabins			439.5		349.3		403.1		516.0		646.3		812.3
Reverberation time w/o compensation, sec			0.30		0.38		0.33		0.26		0.21		0.16
Helmholtz low-frequency compensation	97	0.80	77.6	0.90	87.3	0.68	66.0	0.28	27.2	0.18	17.5	0.12	11.6
Total absorption, sabins			517.1		436.6		469.1		543.2		663.8		823.9
Reverberation time w/ compensation, sec			0.26		0.31		0.28		0.25		0.20		0.16

Table 27-2 Absorption and Reverberation-Time Calculations for a Teleconference Room

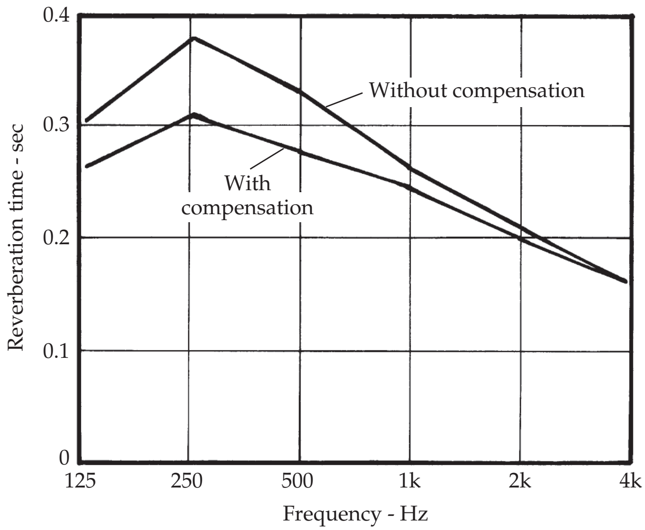


Figure 27-4 Reverberation time for a teleconference room.

The calculated reverberation time varies from 0.16 to 0.38 sec, which is a reasonable range. A value of 0.2 sec would represent a room with somewhat greater absorption, and this design should ensure good speech intelligibility with regard to room acoustics. The room has adequate absorption at 125 Hz due to the diaphragmatic action of wall and ceiling gypsum board. The 0.38-sec peak at 250 Hz suggests the possibility of adding a peak of absorption at this frequency. For this reason, Helmholtz absorbers having an absorption peak at about 250 Hz are placed under the shelf. The 97 ft^2 available brings the reverberation time at 250 Hz down somewhat, but not as much as desired. Even though it is impractical to equalize reverberation time to a reasonably flat 0.2 sec, reverberation time between 0.2 and 0.3 sec should make the room suitably absorptive for good speech intelligibility. The two bass traps that are used to control very low-frequency room modes are not included in the reverberation-time calculation.

Key Points

- Speech intelligibility is the most important requirement in any meeting room. When speech is conveyed over data lines, local acoustical intelligibility is an even greater concern.
- Given a clear channel, and satisfactory local equipment, good room acoustics at both ends of the communication line will help to optimize speech intelligibility.
- A teleconference room must be relatively absorptive. Speech is most intelligible in acoustically dead spaces. Conversely, intelligibility is very poor in highly reverberant spaces.
- The background noise level should be below the NCB-20 contour. To achieve this low noise level, attention must, among other things, be given to HVAC noise.

- The size of the teleconference room must be determined by the number of participants to be accommodated at one time. The room proportions should be selected to ensure good modal distribution, particularly at speech frequencies.
- In this design example, the calculated reverberation time varies from 0.16 to 0.38 sec, which is a reasonable range. A reverberation time between 0.2 and 0.3 sec would represent a room with somewhat greater absorption and should make the room suitably absorptive for good speech intelligibility.

CHAPTER 28

Acoustics of Large Halls

In many ways, large halls represent the pinnacle of acoustical design. The sheer size of large halls gives them great civic importance, and in most cases, their construction costs dwarf those of any other acoustical spaces.

Concert halls, opera houses, auditoriums, drama theaters, places of worship, and other large halls can range in size from ample spaces to grand edifices. The seating capacity may range from several hundred to several thousand. The nature of the application determines the acoustical priorities, and can also create diverse demands. For example, in a place of worship, a sermon may demand clear speech intelligibility; a liturgical service may require that chant be properly heard; another religious service may incorporate music performances, both from onstage performers and singing from the congregation. Clearly, the type of presentation will profoundly influence the acoustical design of the space. In addition, many halls designed primarily for speech must integrate a sound playback system with the natural acoustics of the space.

Concert halls providing good acoustics for live music performance are revered, whereas halls with poor acoustics are reviled. The opening of any new concert hall is a great public event, and the evaluation of the quality of its acoustics can make or break the career of an acoustician. Even after extensive computer modeling, it is only on opening night, with a symphony orchestra on stage and a full audience in the house, that the acoustical quality of the hall can be fully ascertained. Different types of music performance spaces, such as concert halls, opera houses, and chamber music halls place particular demands on the design, with different acoustical priorities. For best results, a hall must be tailored for its specific purpose. A well-known axiom is that a multipurpose music hall is a no-purpose hall.

Although classical music is often recorded in concert halls, their acoustical design is optimized for live performance, as opposed to recording. In some cases, halls are temporarily acoustically modified, or the orchestra is seated in a nontraditional way, to provide sound that is specifically beneficial for recording.

Very generally, we may consider the design of large halls in two aspects: spaces primarily intended for speech and those primarily intended for music. Clearly, the former emphasizes speech intelligibility whereas the latter requires musical sonority.

Design Criteria

In some ways, even the largest hall is no different from the smaller rooms we have considered. In other words, the basic acoustical criteria are the same. A large hall must have a low ambient noise level from internal and external sources; it must provide a reasonable level of acoustical gain; it must provide appropriate reverberation time; it must provide reverberation with a pleasing tonal quality; it must avoid artifacts such as echoes. Of course, all of these acoustical needs must be reconciled with both aesthetic and practical architectural concerns. Above all, the criteria must be evaluated in terms of the structure's intended purpose. For example, while a low level of noise intrusion is important in any hall, it is particularly important in a place of worship—by definition, a place of refuge from the outside world. From these major design criteria, every small detail of the design is derived. For example, again citing a place of worship, care must be taken to design aisles and doors so that those leaving and entering during services will not disturb others.

Reverberation and Echo Control

As we observed in Chap. 11, reverberation is an important parameter that helps define the sound quality of an acoustical space. This is perhaps especially true in large halls such as concert halls, performance theaters, auditoriums, and many places of worship. Reverberation time RT_{60} is closely linked to the intended purpose for any room, and to room volume. Figure 28-1 shows recommended values for mean reverberation time between the two octave bandwidths 500 and 1,000 Hz when a room is occupied between

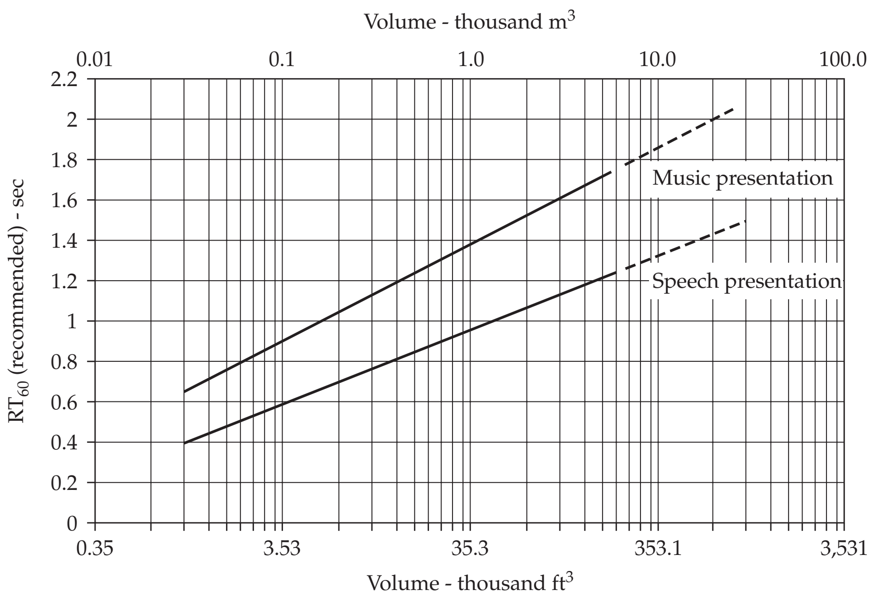


Figure 28-1 The recommended mean reverberation time between 500 and 1,000 Hz, for speech and music, with respect to room volume. (*Ahnert and Tennhardt.*)

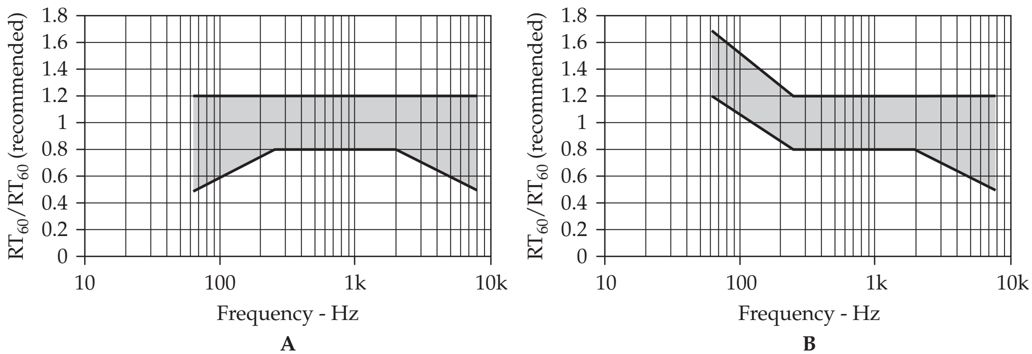


Figure 28-2 The frequency-dependent tolerance range of reverberation time, as referenced to recommended reverberation time. (A) Speech. (B) Music. (*Ahnert and Tennhardt*.)

80 and 100%. As shown, halls designed for speech have shorter mean reverberation times than halls designed for music performance. Also, the recommended mean reverberation time increases as a function of room volume.

The question of reverberation is more than reverberation time. The frequency response of the reverberant field must also be considered. Figure 28-2 shows the frequency-dependent tolerance ranges of reverberation time referenced to the recommended mean reverberation time described in Fig. 28-1. The response of these tolerances is different for speech (Fig. 28-2A) and music (Fig. 28-2B). In particular, music requires significantly longer reverberation times at low frequencies. This imparts warmth to the sound quality of the reverberant field and is sometimes characterized as the bass ratio, as described later. The converse is true for speech; reverberation time decreases at low frequencies. This improves speech intelligibility.

Large enclosed spaces are all potentially subject to the problem of discrete echoes. The long path lengths and multiplicity of seating positions near and far from the sound source can easily create echo problems. Architects and acousticians must be alert to surfaces that might produce reflections of sufficient level and delay to be perceived as discrete echoes. This would be a gross defect in the hall design for which there is little tolerance, audible to everyone with normal hearing.

Reverberation time affects the audibility of echoes. As an example, Fig. 28-3 presents acceptable echo levels for speech under reverberant conditions. The heavy broken line shows the decay rate representing the reverberation time RT_{60} of 1.1 sec in this example. The shaded area represents combinations of echo level and echo delay, experimentally determined, which result in echoes disturbing to people. The upper edge is for 50%, the lower edge is for 20% of the people disturbed by the echo. Concert hall reverberation time is typically around 1.5 to 2.0 sec; many places of worship are closer to 1 sec to favor speech.

Other measurements of this type, made in spaces with other reverberation times, show that the shaded area indicating the level/delay region causing troublesome echoes is close to being tangent to the reverberation decay line. This presents the possibility of estimating the echo threat of any large room by plotting the reverberation time. For example, the lighter broken line in Fig. 28-3 is drawn for a reverberation time RT_{60} of 0.5 sec. An echo-interference area just above this decay line can be very roughly inferred in this way.

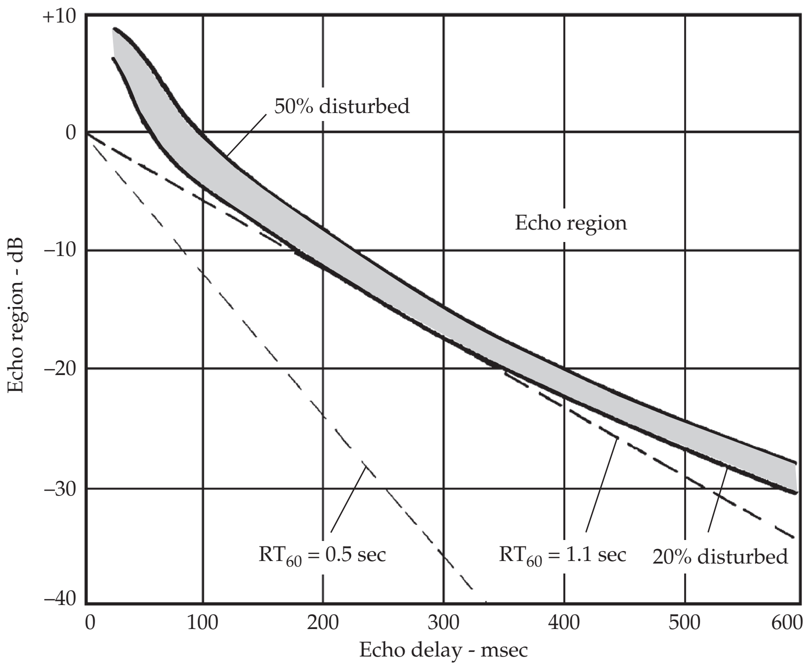


FIGURE 28-3 Acceptable echo levels for speech under reverberant conditions (reverberation time 1.1 sec). (*Nickson, Muncey, and Dubout*.)

Acoustical consultants and architects design music halls to give lateral reflections of appropriate levels and delays to add a sense of spaciousness to the music. This application emphasizes the importance of informed manipulation of reflections in large spaces to achieve desirable results. Lateral reflections are discussed in more detail later.

Air Absorption

Because of air absorption, sound is attenuated as it travels through air. The attenuation is significant when sound path lengths are long. Air absorption should certainly be considered in large halls. Its effect is only at higher frequencies. The resulting decrease in reverberation time at higher frequencies (e.g., at 2,000, 4,000, and 8,000 Hz) should be documented. Air absorption can be approximated from:

$$A_{\text{air}} = mV \tag{28-1}$$

where A_{air} = air absorption, sabins
m = air attenuation coefficient, sabins/ft^3
V = volume of room, ft^3

The value of the air attenuation coefficient m varies with humidity. With humidity between 40 and 60%, the values of m at 2,000, 4,000, and 8,000 Hz are 0.003, 0.008, and 0.025 sabins/ft^3, respectively.

It is also important to note that people are absorptive. Some acousticians use a rule of thumb that each seated person adds 5 sabins at 500 Hz. In a small hall holding a few number of people, particularly one with a short reverberation time, the amount of absorption attributed to the audience may be relatively small. However, if a small hall has a long reverberation time, an audience may contribute a relatively significant amount of total absorption. In a large performance hall, with many hundreds of people in attendance, audience absorption will play a significant role; concert halls can sound quite different depending on whether they are empty or full.

Hall Design for Speech

In halls designed for speech applications, many of the same acoustical criteria that are important for any room design will still apply. However, in halls used primarily for speech, some criteria must be modified, and additional important requirements arise. In particular, it is essential that the hall acoustics provide good speech intelligibility in every seat in the hall. In addition, the design must consider whether a sound system will be used.

Volume

In halls designed for unamplified speech, it is often necessary to limit the overall room volume. This is because a large volume requires more speech power than a small room. In a very large room, for example, 1,000,000 ft^3 in volume, an unamplified voice probably cannot be satisfactorily heard with even the best acoustical design. Even if highly reflective surfaces yield adequate acoustical gain, the resulting long reverberation time would overly degrade intelligibility. A smaller room can be designed to more easily balance speech power and reverberation. A small room with lower absorption and thus moderate reflectivity can provide adequate acoustical gain and thus an adequate speech level. The result is satisfactory intelligibility. In a relatively large auditorium designed for speech, a room volume ranging from 100 to 200 ft^3 per seat is suggested. This volume minimization is contrary to rooms designed for music, where a relatively large volume is desirable.

In a face-to-face conversation, an unamplified talker may generate a sound-pressure level of only about 65 dBA. This level decreases 6 dB for every doubling of distance. Less significantly, sound is also attenuated as it travels through the hall because of air absorption. To support audible levels, the audience area must be placed as close as possible to the talker. This minimizes sound attenuation, provides a more direct sound path, and also improves visual recognition which improves intelligibility. Very generally, the maximum distance from a talker in an auditorium to the most distant seat should be about 80 ft.

Hall Geometry

The talker-to-audience distance can be minimized by carefully considering the room geometry. In particular, as seating capacity increases, the lateral dimensions of the room must increase, and the side walls should be splayed. A rectangular shoebox-type hall, with the stage across one narrow end, may be excellent for music where an audience can be seated farther away and a greater ratio of reverberant sound is desirable. However, a rectangular geometry is only suitable for a relatively small speech hall. Otherwise, with larger seating capacities, much of the audience is placed far away from the stage, at the far end of the hall.

This limitation can be overcome to some extent by widening the hall. However, for greater seating capacity, the side walls should be splayed from the stage. Figure 28-4 shows a rectangular floor plan, and two examples of halls with splayed side walls. Splayed side walls allow greater seating area that is relatively close to the stage. Care must be taken with wall angles to avoid any flutter echoes, such as the case in Fig. 28-4B for the marked reflection. The splayed walls can usefully reflect sound energy to the rear of the hall (Fig. 28-4B and C). A side-wall splay may occupy the entire length of the side walls, or be limited to the rear portions of the side walls. A side-wall splay may range from 30° to 60°; the latter is considered a maximum angle,

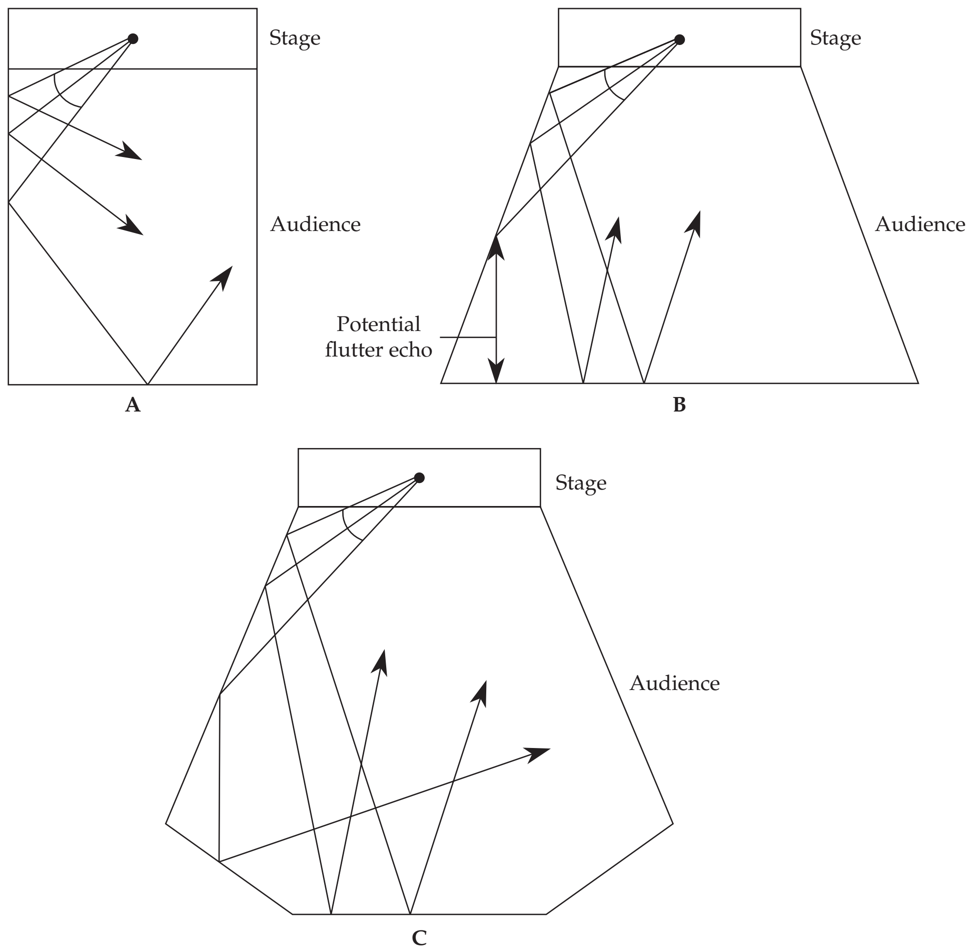


Figure 28-4 For greater seating capacity, the side walls can be splayed from the stage. (A) Rectangular floor plan. (B) Splayed side walls with flat rear wall. Note the potential flutter echo. (C) Splayed side walls with extended rear walls.

given the directionality of speech. In addition, the rear walls may be extended outward in the center of the hall (Fig. 28-4C) or the rear wall may be concave with equal radius from the stage, to create a fan-shaped room. Any concave room geometries require careful design to avoid sound focusing. Very generally, fan-shaped halls are not used for music performance.

Absorption Treatment

In small speech halls, the majority of absorption is provided by the audience; therefore, the room surfaces can be relatively reflective. In larger halls, where there is greater room volume per seat, relatively greater room absorption may be needed. Beneficially, a reflective front stage area provides strong early reflections (short path-length differences) that are integrated (by the precedence effect) with the direct sound and enhance it. On the contrary, strong late reflections and reverberation, such as from rear walls, would not be integrated and may produce echoes. To accommodate this, the stage area and front of the hall are made reflective, and absorption is placed in the seating area and rear of the hall. To preserve speech intelligibility, although early reflections are useful, the overall reverberation time must be short, perhaps less than 0.5 sec. When the front stage area is reflective, an important benefit is an increase in overall speech gain. However, the same reflections may produce comb-filter effects; this possible source of timbral change must be considered.

Ceiling, Walls, and Floor

In many large halls, ceiling reflectors, sometimes called clouds, are used to direct sound energy from the stage to the seating area. Both planar and convex reflectors may be used. The size of the reflectors determines the range of frequencies that are reflected; the larger the panel, the lower the cutoff frequency of the reflected sound. Both dimensions of a square reflecting panel should be at least five times the wavelength of the lowest frequency to be reflected. For example, a square panel measuring 5 ft on a side will reflect frequencies of 1 kHz (wavelength is 1 ft) and higher. Panels must also be solid and stiff, and securely mounted to avoid resonances. When ceilings are high, care must be taken to ensure that path-length differences in the seating area between direct and reflected sound are not too great, and particularly should not exceed 20 msec. In some cases, clouds are made absorptive, to avoid late reflections.

A sloping (raked) floor allows a more direct angle of incidence which in turn yields less absorption. Generally, the slope of an auditorium floor should not be less than 8°. The floor of a lecture-demonstration hall might have a 15° angle of inclination. Staggering of seats is also recommended.

Because of its potential to create undesirable late reflections, the rear wall of a large hall requires special attention. Reflections from the rear wall would create a long path-length difference (the path length of the reflection from the rear wall minus the path length of the direct sound) to a listener at the front of the hall. This can result in audible echoes, particularly because of the otherwise low reverberation level. As noted, a reflective concave rear wall would also undesirably focus sound. For these reasons, the rear wall of a large hall is usually absorptive. In some cases, when added absorption is undesirable because of decreased reverberation time, reflective diffusers can be placed on the rear wall.

Speech Intelligibility

Speech intelligibility is the highest design priority for any hall intended for spoken word. This is the case in many places of worship, auditoriums, and drama theaters. Sound systems are often used to overcome acoustical limitations and to provide intelligibility in even very large spaces. In a hall where amplification is not used, a room design providing high speech intelligibility begins by recognizing that a normal voice, as noted, will generate a long-term average sound-pressure level of about 65 dBA. Peak sound-pressure levels may be 12 dB higher than the long-term average. Different loudness levels of talkers may range from about 55 to 75 dBA.

Speech Frequencies and Duration

Very generally, the most significant speech frequencies range from 200 Hz to 5 kHz. The great majority of speech power is below 1 kHz, and the maximum speech energy range is 200 to 600 Hz. Speech vowels mainly occupy low frequencies, whereas consonants occupy higher frequencies. Consonants are most important in intelligibility. The frequencies above 1 kHz, specifically in the 2- to 4-kHz range, are primarily responsible for speech intelligibility. The three bands at 1, 2, and 4 kHz provide 75% of speech intelligibility content.

Generally, consonants have a duration of about 65 msec, and vowels a duration of about 100 msec. Syllables may have a duration of 300 to 400 msec, and words a duration of 600 to 900 msec, depending on rate of delivery. A relatively short reverberation time is required so that reverberation from a previous element does not mask a subsequent element. Early reflections (less than 35- to 50-msec delay) tend to be integrated with the direct sound and increase perceived loudness and assist intelligibility. Late reflections (greater than 50-msec delay) particularly degrade speech intelligibility. Also, a high speech-to-noise dynamic range is needed for good speech intelligibility. A slower rate of talker delivery and the ability to articulate the speech also play a large role in intelligibility. For example, in highly reverberant spaces, slowing the delivery rate from five syllables per second to three per second can significantly improve intelligibility.

Subject-Based Measures

Speech intelligibility in a room is often estimated using subject-based measures, that is, by using live experimentation. A talker reads from a list of words and phrases, and listeners in the room write down what they hear. The list includes examples of important speech sounds. Between 200 and 1,000 words are used per test. For example, Table 28-1 lists some English words used in subject-based intelligibility testing. The higher the percentage of correctly understood words and phrases, the better the speech intelligibility.

In some cases, listening difficulty is measured. When the level of speech is the same as the noise level, intelligibility can be adequate, but listeners can still have difficulty in understanding what is being said, and considerable attention is required. When the speech level is raised by 5 or 10 dB over the noise, intelligibility is not greatly improved, but listeners report much less difficulty in hearing.

Analytical Measures

Various analytical measures have been devised to assess speech intelligibility. The articulation index (AI) uses acoustical measurements to estimate speech intelligibility and conversely, speech privacy. AI uses weighting factors in five octave bands from 250 Hz

aisle	done	jam	ram	tame
barb	dub	law	ring	toil
barge	feed	lawn	rip	ton
bark	feet	lisle	rub	trill
baste	file	live	run	tub
bead	five	loon	sale	vouch
beige	foil	loop	same	vow
boil	fume	mess	shod	whack
choke	fuse	met	shop	wham
chore	get	neat	should	woe
cod	good	need	shrill	woke
coil	guess	oil	sip	would
coon	hews	ouch	skill	yaw
coop	hive	paw	soil	yawn
cop	hod	pawn	soon	yes
couch	hood	pews	soot	yet
could	hop	poke	soup	zing
cow	how	pour	spill	zip
dale	huge	pure	still	
dame	jack	rack	tale	

Table 28-1 Examples of Words Used in Subject-Based Intelligibility Testing

to 4 kHz (in some cases, 1/3-octave bands are used). Each weighting factor accounts for our hearing sensitivity in that band; for example, the weighting factor is highest at 2 kHz, because our hearing sensitivity is greatest there. AI is calculated by multiplying the signal-to-noise (S/N) ratio in each octave band by the weighting factor in each octave band, and summing the result. When the S/N ratio is greater than 30 dB, a value of 30 dB is used. When the S/N ratio is negative, a value of 0 dB is used. In some cases, a correction factor accounting for reverberation is subtracted from the AI value. AI ranges from 0 to 1.0; the higher the value, the better the intelligibility.

Another objective measure used to assess speech intelligibility is known as percentage articulation loss of consonants, known as %Alcons. As its name implies, %Alcons focuses on the perception of spoken consonants. %Alcons can be approximately measured as:

$$\%\mathrm{Alcons} \approx 0.652\left(\frac{r_{lh}}{r_h}\right)^2 \mathrm{RT}_{60} \tag{28-2}$$

where %Alcons = percentage articulation loss of consonants, percent
r_{lh} = distance from sound source to listener
r_h = reverberation radius, or critical distance for directional sound sources
RT_{60} = reverberation time, sec

Subjective Intelligibility	%Alcons
Ideal	≤3%
Good	3–8%
Satisfactory	8–11%
Poor	>11%
Worthless	>20%*

*Limit value is 15%.

Table 28-2 Subjective Weighting for Results of %Alcons Testing

Subjectively, %Alcons scores can be related to speech intelligibility, as shown in Table 28-2. Other methods used to estimate speech intelligibility include the speech transmission index (STI), speech intelligibility index (SII), and rapid acoustics speech transmission index (RASTI). The latter may be correlated to %Alcons by: RASTI = 0.9482 – 0.1845ln(%Alcons).

According to one criterion, satisfactory speech intelligibility can be achieved by designing for an appropriate reverberation time. In particular, reverberation time at 500 Hz, with the room two-thirds occupied, should be selected so that at the most distant listening position, the ratio of the reflected sound energy to the direct sound energy is no greater than 4. This corresponds to a 6-dB difference between the energy densities and should provide a low (5%) consonant articulation loss.

When a room's purpose is primarily for spoken word, it may be useful to specify the acceptable ambient noise level in terms of the speech interference level (SIL). SIL is calculated as the average of the sound-pressure levels (SPL) in octaves centered at 500, 1,000, 2,000, and 4,000 Hz. For calculation purposes, the female SIL is set at 4 dB below that of the male SIL.

Concert Hall Acoustical Design

The design of a large hall for music performance presents special challenges. Perhaps the first complexity rests with the music itself. Clearly, symphonic music, chamber music, and opera each require very different acoustics, as well as size and room functionality. Moreover, different styles of music, such as baroque, classical, and popular have different acoustical requirements. Finally, different music cultures, such as Eastern and Western, require different design criteria. Perhaps the most difficult aspect of hall design is the ambiguity of the goal itself. While it is possible to measure many specific aspects such as reverberation time, there is no measurement, or even specific consensus, on what "good" music acoustics are. The diversity of the requirements, subjectivity of the goal, lack of objective benchmarks, and differences of opinion all conspire to make concert hall design an art as well as a science.

Reverberation

Very generally, the problem of music hall acoustics may be considered in two parts: early sound and late reverberant sound. Early sound is sometimes considered in terms of early reverberation decay time, intimacy, clarity, and lateral spaciousness. Late reverberant sound can be considered as late reverberation decay time, warmth, loudness, and brilliance.

Reverberation can further be considered in two parts: early and late reverberation. The ear is very sensitive to early reverberation. This is partly because in most music, later reverberation is partly masked by the following music notes. Early reverberation largely defines our subjective impression of the entire reverberation event. The early decay time (EDT) is defined as the time required for sound to decrease 10 dB, multiplied by 6. (Multiplying by 6 allows comparison with late reverberation time RT_{60}.) Unlike dense late reverberation, early reverberation comprises relatively few primary reflections. These reflections arrive within the Haas fusion zone and are integrated with the direct sound, reinforcing it. This early reverberation can affect the clarity of sound. The greater the energy in the early reverberation, the better the clarity. Late reverberation can affect our perception of the liveness of sound. More late reverberant energy can increase liveness or fullness. As late reverberant energy increases, the clarity relatively decreases.

Clarity

Clarity, measured in decibels, is sometimes defined as the difference between the sound energy in the first 80 msec, and the late reverberation energy arriving after the first 80 msec. This is sometimes referred to as C_{80}. In some cases, a $C_{80}(3)$ value is used, which averages clarity at 500, 1,000, and 2,000 Hz. In large halls with good clarity, the value of $C_{80}(3)$ ranges between −4 dB and +1 dB.

In some hall designs, to achieve good clarity, as well as good liveness, the reverberation decay is tailored as a two-part slope. The early decay has a steep slope and thus a short EDT (clarity), whereas the late decay has a shallower slope and thus a longer RT_{60} (liveness). When there is a single decay slope, the reverberation time is simply measured as RT_{60}. For large concert halls, RT_{60} in mid frequencies is generally between 1.8 and 2.2 sec.

Brilliance

Brilliance is another metric used to quantify music hall acoustics. Brilliance describes sound that has presence and clearness. Brilliance is achieved with adequate high-frequency energy from reflecting surfaces. On the other hand, a hall with good brilliance should not sound too bright or harsh. Brilliance can be estimated by comparing a high-frequency EDT to an averaged mid-frequency EDT. In particular:

$$\frac{EDT_{2,000}}{EDT_{Mid}} = \frac{EDT_{2,000}}{EDT_{500} + EDT_{1,000}} \tag{28-3}$$

Similarly, $EDT_{4,000}/EDT_{Mid}$ can be calculated. Some acousticians recommend that $EDT_{2,000}/EDT_{Mid}$ should be at least 0.9, and $EDT_{4,000}/EDT_{Mid}$ should be at least 0.8.

Gain

A good music hall should also provide adequate acoustical gain at all seating positions. Gain (G) can be thought of as the difference in sound-pressure level between the sound-pressure level in the center of the hall from an arbitrary source, minus the sound-pressure level 10 m away from the same source in an anechoic environment. The former gain is a function of direct sound and reflected sound at a particular seating position in the hall, while the latter contains only direct sound at 10 m. Gain thus depends on the volume of the hall and the reverberation time RT_{60} or the early delay time EDT. In particular:

$$G_{Mid} = 10 \log\left(\frac{RT_{Mid}}{V}\right) + 44.4 \tag{28-4}$$

or

$$G_{Mid} = 10 \log\left(\frac{EDT_{Mid}}{V}\right) + 44 \tag{28-5}$$

where G_{Mid} = gain as averaged at 500 and 1,000 Hz, dB
RT_{Mid} = reverberation time averaged at 500 and 1,000 Hz, sec
EDT_{Mid} = early reflection time averaged at 500 and 1,000 Hz, sec
V = volume, m^3

In many concert halls with good acoustics, G_{Mid} lies between 4.0 and 5.5 dB. However, in halls with different, more specialized applications, G_{Mid} will vary over a wider range.

Seating Capacity

Given a room volume, the optimal number of seats may be determined. This relationship partly depends on the type of music that will be performed. Furthermore, the total floor area of the hall (stage and audience) may be estimated from the number of seats. When measured in square meters, the floor area may be estimated as 0.7N where N is the number of seats. When measured in square feet, floor area is about 7.5N. The number of seats may be determined from this equation:

$$N = \frac{0.0057\,V}{RT_{Mid}} \tag{28-6}$$

where N = number of seats
V = room volume, ft^3 or m^3
RT_{Mid} = reverberation time averaged at 500 and 1,000 Hz, sec

Note: In metric units, change 0.0057 to 0.2.

Volume

Room volume is profoundly influenced by the intended purpose of the hall. Among other things, volume affects reverberation time and required absorption. Total room volume may be specified; for example, a concert hall may be specified at 900,000 ft^3. In some cases, volume is specified as a minimum volume per listener seat; for example, a concert hall may range from 200 to 400 ft^3 per seat. When a balcony is incorporated into a hall design, volume per seat is generally reduced.

Diffusion

As with any room designed for the enjoyment of music, whether the performance is heard live or via recording, music concert halls must include diffusion in their designs to ensure that the acoustical environment is suitable. Some of the most highly acclaimed concert halls with the best reputations for sound quality have sound fields that are highly diffuse. Research has shown that the surface diffusivity index (SDI), a qualitative characterization of surface diffusion, highly correlates with the acoustical quality index (AQI) of a hall. There is debate as to whether their diffusion was a deliberate part of their design, or was a happenstance of the design aesthetic when they were built. For example, the Grosser Musikvereinssaal in Vienna, inaugurated in 1870, has excellent acoustics. The architectural style of the period led to highly diffuse surface finishes of geometrical shapes and relief ornamentation. In more modern times, because of rising building costs, increased seating requirements, and changes in interior design fashions, ornate features have been largely replaced by flat plaster, concrete, drywall, and cinder block surfaces. In some cases, halls with this more modern construction lacked adequate diffusion, and sound quality suffered. In other modern halls, the importance of a highly diffuse sound field has been recognized and established as a primary design goal. This diffusion can be achieved with highly modulated surfaces using number-theoretic, and shape-optimized diffusers.

Spaciousness

The characteristics of the early sound field are also important in establishing a sense of spaciousness so the listener feels enveloped by sound in the hall. A sense of spaciousness can be created by early reflections from side walls, often called lateral reflections, which occur within 80 msec of the arrival of the direct sound. It is important that these reflections arrive at the listener from either side at angles of approximately 20° to 90° relative to the front of the listener. The geometry of halls that are relatively narrow, as in a rectangular shoebox-type design, lends itself to lateral reflections. In fan-shaped halls, these reflections arrive more from the listener's front; this can decrease the effect. In some designs, a reverse lateral shape can provide good lateral reflections. Spaciousness is also augmented by providing adequate diffusion; the ornate decorations in many older concert halls accomplish this.

Apparent Source Width

Apparent source width (ASW) can be used to describe the perceived breadth of a sound source such as an orchestra, which is wider than the physical source. ASW is improved, that is, the source appears wider when there is a high level of early lateral reflections (before 80 msec) that contribute early spatial impressions. Listener envelopment (LEV) is sometimes used to describe the feeling of being surrounded by and immersed in sound that fills a large space. It is improved by late-arrival lateral reflections (after 80 msec).

Initial Time-Delay Gap

Beranek made an intensive study of concert halls around the world. He noted that those halls rated the highest by qualified listeners had certain technical similarities. Among them was an acoustical metric known as initial time-delay gap (ITDG). This is the time between the arrival of the direct sound at a given seat and the arrival of critically important early reflections. Concert halls that rated high on the sound quality scale had a well-defined ITDG of about 20 msec or less, measured in the center of the hall. Halls with an ITDG flawed by uncontrolled reflections were rated inferior by qualified listeners. ITDG was discussed in Chaps. 11 and 24.

In smaller, more intimate halls, with reflective surfaces closer to the listener, the initial time-delay gap is small. In larger halls, ITDG is greater for most listeners. However, ITDG depends on seating position; a listener sitting near a side wall, for example, would experience a smaller ITDG. Smaller ITDG values, because they denote a more intimate sound field, are desirable. Large halls that are relatively narrow, as in a rectangular shoebox-type design, can have a relatively small ITDG. In some cases, a small ITDG is achieved by segmenting the audience into smaller areas and placing reflecting walls near them. Alternatively, side balconies, terraces, and other side protrusions can be used to provide early reflections. Since these architectural features also provide lateral reflections, they can uniquely impart both a sense of intimacy and spaciousness.

Bass Ratio and Warmth

A sense of acoustical warmth is desirable in most halls. This is often attributed to longer reverberation times at low frequencies. One way to characterize this is through the bass ratio (BR). Bass ratio is calculated by summing the reverberation times at 125 and 250 Hz, and dividing by the sum of reverberation times at 500 and 1,000 Hz:

$$BR = \frac{RT_{60/125} + RT_{60/250}}{RT_{60/500} + RT_{60/1,000}} \tag{28-7}$$

An acoustically warm hall with longer bass reverberation times would thus yield a bass ratio greater than 1.0. According to one study, for halls with RT_{60} less than 1.8 sec, the bass ratio should lie between 1.1 and 1.45. For halls with a higher RT_{60}, the bass ratio should lie between 1.1 and 1.25. (For speech, a bass ratio between 0.9 and 1.0 is preferred.)

Concert Hall Architectural Design

The architectural design of a concert hall requires close collaboration between the architect and the acoustician. This is particularly true in the sound chamber itself; the stage and seating area of this inner shell must meet very specific acoustical requirements for both performing musicians and the audience while providing amenities and safety for all the occupants. Overarching these practical needs are the more intangible aesthetic requirements; a large concert hall must present a space where the pleasure of hearing music is never compromised.

Balcony

In some halls, a balcony can be used to decrease the distance from the stage to some seating areas and to provide good sight lines. Care must be taken to avoid acoustical shadowing in the seating areas underneath the balcony, as shown in Fig. 28-5. Very generally, the balcony overhang depth should be less than twice the height of the balcony underside. Ideally, the depth should not be more than the height. Deep balconies in particular can create acoustical shadows in the seats underneath the balcony. In addition, reflecting surfaces on the ceiling and side walls, as well as the underside of the balcony, should be designed to add as much reflected sound as possible to the seating areas on the balcony and under it, to supplement the direct sound from the stage.

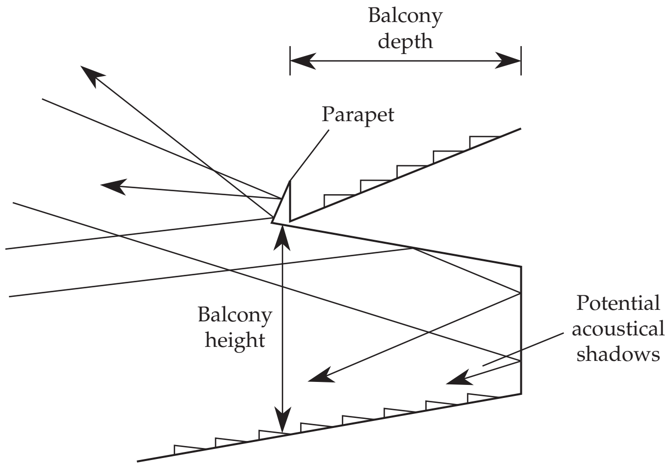


Figure 28-5 Ideally, balcony depth should not be more than the height to avoid acoustical shadowing underneath the balcony. The underside of the balcony should be reflective to supplement the direct sound. The front of a balcony parapet should be designed to prevent undesirable reflections.

The front of a balcony parapet should be designed to avoid reflections that could affect sound quality in the seating areas in the front of the hall; this is particularly true when the plan view of the balcony has a concave shape.

Ceiling and Walls

Ceiling height is usually determined by the overall room volume that is required. Very generally, ceiling height should be about one-third to two-thirds of the room width; the lower ratio is used for large rooms, and the higher ratio is used for small rooms. A ceiling that is too high may result in a room volume that is too large and may also create undesirable late reflections. To avoid potential flutter echo, a smooth ceiling should not be parallel to the floor.

In many halls, the ceiling geometry itself is designed to direct sound to the rear of the hall, or to diffuse it throughout the hall, as shown in Fig. 28-6. A ceiling may have several segments, each sized and angled to reflect sound to different seating areas; for

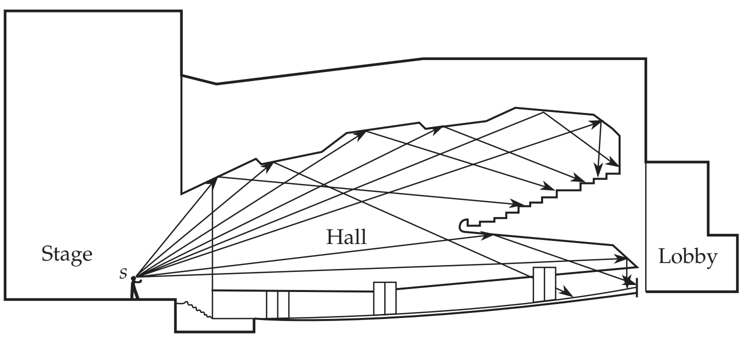


Figure 28-6 The ceiling geometry should direct reflected sound throughout the hall. Several ceiling segments may be sized and angled to reflect sound to particular seating areas in the hall.

example, the ceiling near the stage might reflect sound to the near rows, whereas the ceiling farther from the stage might reflect sound to the farther rows. In some hall designs, the ceiling can be raised or lowered, often as several independent sections, to vary the acoustics of the hall. For example, when the ceiling is lowered, the hall's acoustics are more intimate. In some concert halls, large side doors can be opened or closed to the offstage wings, varying the volume of the sound chamber.

Concave surfaces such as domes, barreled ceilings, and cylindrical arches should be avoided because of the undesirable sound foci they create. The rear wall must avoid any large, unbroken concave geometry. Side walls must avoid parallelism. This can be avoided by tilting or splaying wall surfaces. These angles can also be advantageously used to direct reflected sound to the audience seating area, and to provide diffusion. Any surface that unavoidably introduces concave geometry or an undesirable angle should be covered with absorptive material.

Raked Floor

In halls designed for either music or speech, a raked (sloping) floor is desirable; this is particularly true for large halls. A raked floor improves sight lines, and also improves fidelity in the seating area. When sitting on a raked floor, the listener receives more direct sound than would be available on a flat floor; in either case, the stage should be raised. In some designs, the floor slope is constant with respect to distance from the stage; risers are equal. In other designs, the slope increases with distance from the stage; risers are not equal. In other designs, the floor slope is constant near the stage and then changes to another constant but steeper slope further from the stage. In some halls, a balcony with a raked floor can be used to add seating capacity while decreasing distance from the stage to the audience. It is common for balconies to use relatively steeply sloping floors.

A raked floor is desirable in halls where audience sound absorption must be minimized. A more direct angle of incidence allows less absorption than a shallow angle that passes sound over a larger audience area. The frequency response of sound passing over an audience can be greatly affected by its angle of incidence. At low angles of incidence, a dip in low-frequency response around 150 Hz, as deep as 10 to 15 dB and extending for two octaves, occurs at the audience head position. Also, this effect is most pronounced for direct and early arrival sound. A raked floor minimizes these problems. Otherwise, one approach is to design a low-frequency boost in early-reverberation energy. Some evidence shows that strong ceiling reflections are also helpful in this regard.

Virtual Image Source Analysis

As we observed in Chaps. 6 and 13, a reflection from a boundary surface such as wall, floor, or ceiling can be considered as a virtual image source. The virtual source is acoustically located behind the reflecting surface, just as when viewing an image in a mirror. Moreover, when sound strikes more than one surface, multiple reflections will be created. Thus, images of the images will exist. In a rectangular room, there are six surfaces and the source has an image in all six surfaces, sending energy back into the enclosure, resulting in a highly complex sound field.

Using this modeling technique, we can ignore the boundary surfaces themselves, and consider sound as coming from many virtual sources spaced away from the actual source, arriving at time delays based on their distance from the source. Moreover, the magnitude of the images depends on the absorbency of the reflecting surface, and the number of times the image has been reflected.

This technique can be used to examine the spatial sound field of large spaces, as shown in Fig. 28-7. Four large spaces were tested by sounding an impulse on the stage and measuring the resulting reflecting sound field with a 6-point microphone array. In this figure, the center of each circle represents the location of the virtual source image, and the circle size represents its intensity. The distance of the circles from the intersection of the axis represents the arrival time delay. The measuring array is located at the intersection of the axes. The virtual sources created by the reflections from the boundary surfaces spatially document the sound fields of the four enclosures. In the case of the stadium (Fig. 28-7A), there are only a few reflecting surfaces, and the images are widely spaced, revealing probable echoes. In the opera house (Fig. 28-7B), the virtual sources are relatively dense, with a number of strong images overhead. The relative sizes of the large concert hall (Fig. 28-7C) and small concert hall (Fig. 28-7D) are evident from the spatial distribution of the images.

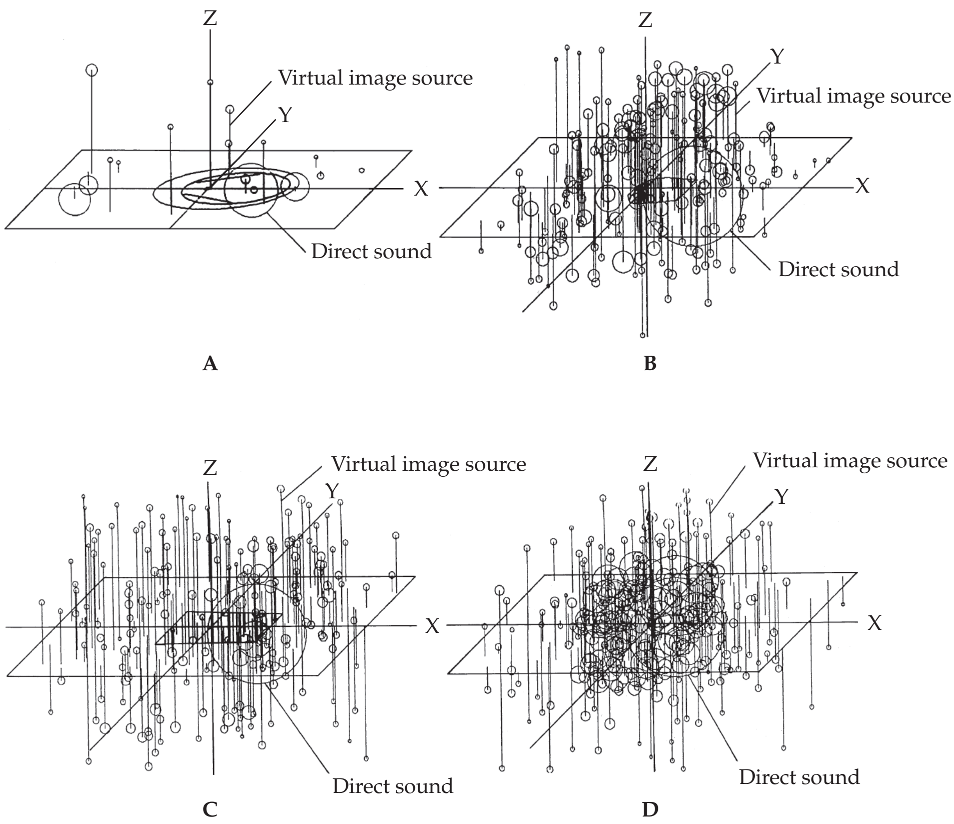


Figure 28-7 The virtual image sources created by the reflections from boundary surfaces spatially document the sound fields of four enclosures. (A) Stadium. (B) Opera house. (C) Large concert hall. (D) Small concert hall. (*JVC Corporation.*)

Hall Design Procedure

In practice, an acoustician may begin a hall design by carefully considering the site, and determining ambient noise levels and structure layout. Numerical design may begin by specifying a value for G_{Mid}. For example, a value of 5.0 dB may be assumed. Furthermore, a value of RT_{Mid} or EDT_{Mid} is assumed; this value will depend greatly on the type of music to be performed in the hall. For example, assume an RT_{Mid} value of 2.0 sec. Given these values, the hall volume may be calculated, as well as the number of seats and total floor area. If these results do not meet the design criteria, the value of G_{Mid} may be adjusted. Using these calculated guidelines, the acoustician and architect may decide on a hall configuration, such as rectangular or fan shaped. Again using the guidelines, elements such as balconies and terraces can be included, taking into account factors such as ITDG. Absorption is added according to the assumed RT and EDT values. The design procedure may seem simple in principle, but in practice, considerable skill and perhaps some luck are required to create a concert hall with excellent acoustics.

Case Studies

The design of a large concert hall ranks among the most complex of architectural tasks. Hall design is often unique and demonstrates creativity, and daring, on the part of the architect and acoustician. This physical uniqueness guarantees that each concert hall will have a unique sound character that is unlike any other. This has been true for hundreds of years, and continues to this day. Two renowned concert halls are considered here.

Orchestra Hall in Chicago is an example of a traditional rectangular hall, with steeply raked balconies. Figure 28-8A and B shows architectural plans for the hall. This hall was constructed in 1904 and acoustical deficiencies were identified shortly after its opening, particularly in the stage area and the shell over the stage. As a result, the hall has been renovated several times. In 1966 much of the plaster ceiling was replaced by a perforated aluminum screen in an effort to increase effective room volume, and thus lengthen the reverberation time. However, upholstery was added to the balcony and gallery and the occupied reverberation time did not increase, and the unoccupied reverberation time was severely reduced, affecting rehearsals. In 1981 the hall was again renovated; among other improvements, absorption was reduced in the main floor seating area as well as on surfaces surrounding the stage, upper hall surfaces were hardened, hall volume was opened, diffusing plaster was placed on rear wall surfaces, and a new pipe organ was installed. Figure 28-8C shows unoccupied reverberation time before and after the 1981 renovation; the increase in low-frequency reverberation can be seen. Subsequently, another renovation was completed in 1997.

The Philharmonie in Berlin is an example of large concert hall using a vineyard configuration. In this approach, the large hall is segmented into a number of smaller multilevel audience areas separated by low walls. This contributes lateral reflections to provide intimacy within the larger space. This hall, completed in 1963, places the audience around the orchestra and is notable for a tentlike ceiling with reflecting clouds below it. Figure 28-9A and B shows architectural plans of the Philharmonie, with sketches of the ground plan and longitudinal section. Figure 28-9C plots reverberation time of the hall in three states: unoccupied; occupied by orchestral musicians as in a recording session; and fully occupied by orchestral musicians, chorus, and audience. In all three cases, the rise in low-frequency reverberation is evident.

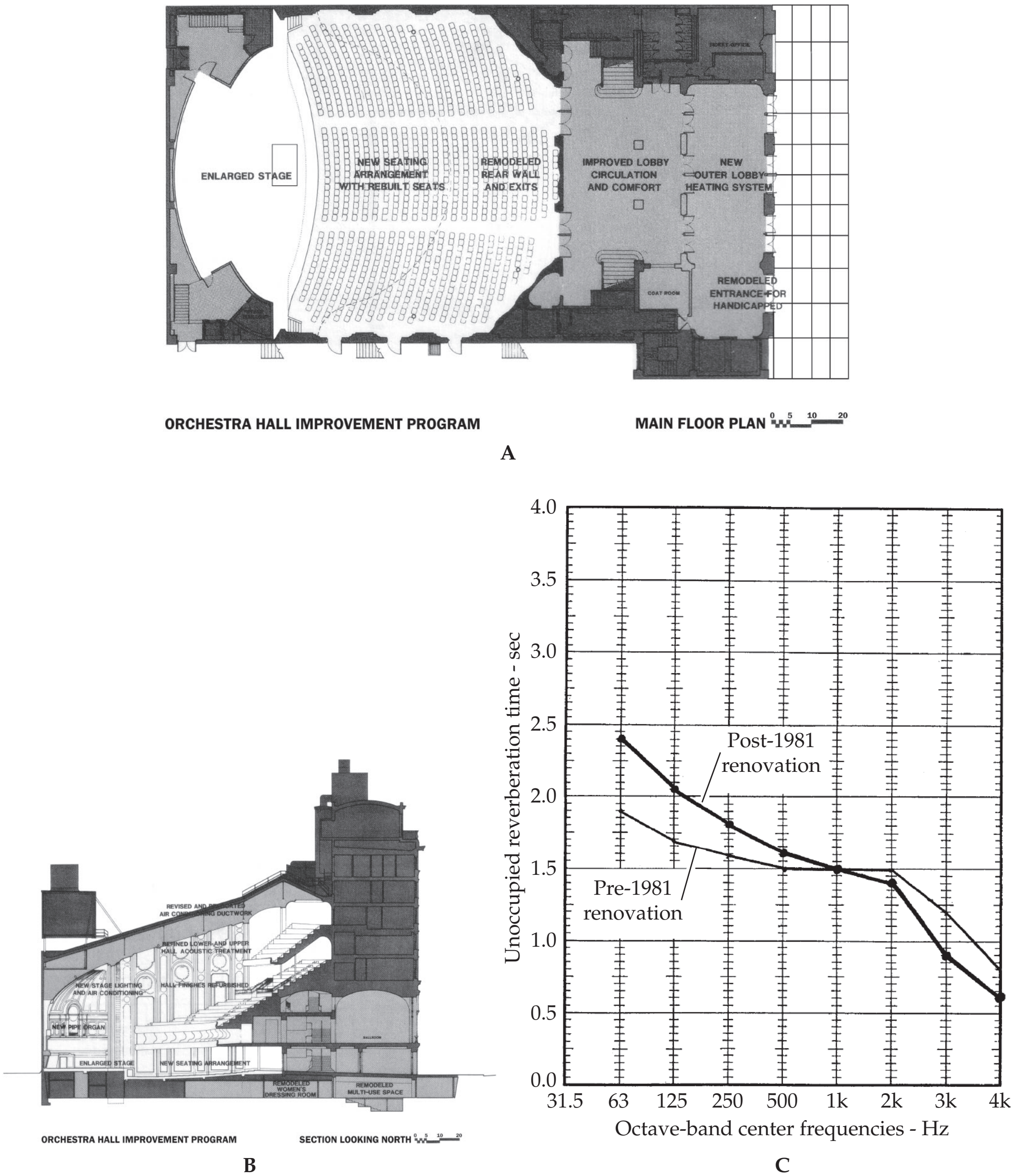
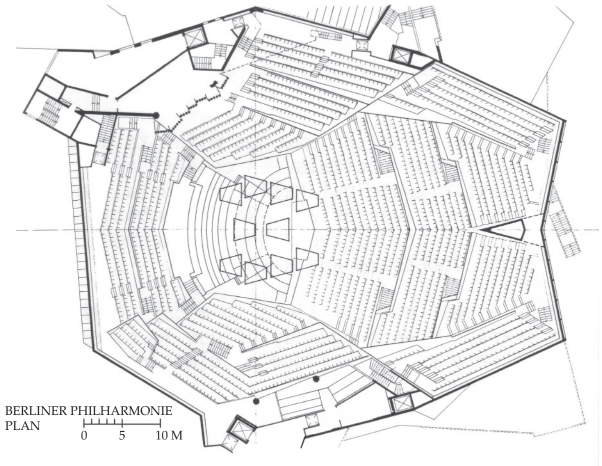


FIGURE 28-8 Architectural plans and reverberation-time measurements for Orchestra Hall in Chicago. (A) Ground plan. (B) Longitudinal section. (C) Unoccupied reverberation time before and after the 1981 renovation. (*Reprinted with permission from the Acoustical Society of America, Halls for Music Performance: Two Decades of Experience, 1962–1982, Richard H. Talaske, Ewart A. Wetherill, and William J. Cavanaugh, eds., 1982.*)



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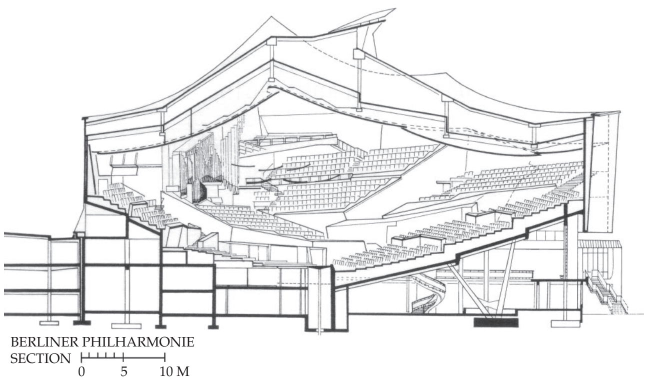

B

Figure 28-9 Architectural plans and reverberation-time measurements for the Philharmonie in Berlin. (A) Ground plan. (B) Longitudinal section. (C) Reverberation time. (*Reprinted with permission from the Acoustical Society of America, Halls for Music Performance: Two Decades of Experience, 1962–1982, Richard H. Talaske, Ewart A. Wetherill, and William J. Cavanaugh, eds., 1982.*)

Postscript

Everyone with at least a passing interest in audio is familiar with the Dolby name and logo. For many years, this company has been instrumental in the advance of audio technology. Ray Dolby, the founder of the company, was an audio engineer who strove for high sound quality in the recording and playback of music. He was also a classical music lover, and carried that same interest in sound quality to the concert hall. He was a patron of the San Francisco Symphony and when their home, Davies Symphony Hall, was being renovated, he worked with acousticians to ensure the best possible result. After the work was completed, he was determined to find the seat in the hall that was best attuned to his tastes. So, during symphony rehearsals, he auditioned one seat after another until he found the one that gave him the best possible sound quality. Thereafter, he always sat there.

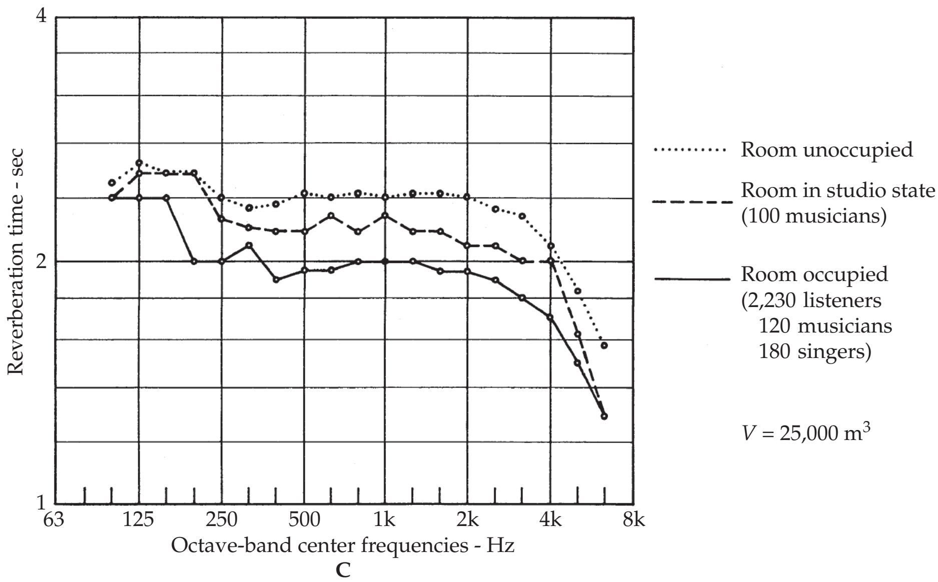

Figure 28-9 (*Continued*)

If Dr. Dolby had continued his search, he might have identified the best seat for full orchestra, another for chamber music, and yet another for orchestra and chorus. He might have found the best seat for Mozart, one for Beethoven, and one for Stravinsky. Different conductors might have necessitated different seats. Theoretically speaking, there are countless subtleties in how we perceive music in a concert hall.

We conclude that in concert halls, as well as recording studios, home theaters, and any acoustical spaces, the pursuit of perfect sound quality is an intriguing journey, and one that will never end.

Key Points

- Two types of large halls may be considered: spaces intended for speech and those intended for music. The former emphasizes speech intelligibility whereas the latter requires musical sonority.
- The basic acoustical criteria of large halls is no different from smaller rooms. A large hall must have a low noise level, provide acoustical gain, provide appropriate reverberation time and tonal quality; and must avoid acoustical artifacts.
- Halls designed for speech have shorter mean reverberation times than halls designed for music. The recommended mean reverberation time increases as a function of room volume.
- The frequency response of the reverberant field must be tonally pleasing and also appropriate for the function of the space. Music performance requires longer reverberation times at low frequencies compared to speech.

- The long path lengths and multiplicity of seating positions near and far from the sound source can create echo problems. This would be a gross defect in the design.
- Because attenuation due to air absorption is significant when sound path lengths are long, it should be considered in large halls. Its effect is only at higher frequencies.
- In halls designed for unamplified speech, it is often necessary to limit the overall room volume because a large volume requires more speech power than a small room.
- Hall geometry must be designed according to the intended function. As seating capacity increases, the lateral dimensions of the hall must increase, and side walls should be splayed.
- When the front stage area of an auditorium is reflective, speech gain is improved. However, the reflections may produce comb-filter effects, a possible source of timbral change.
- In many large halls, ceiling reflectors (clouds) are used to direct sound energy from the stage to the seating area. The size of the reflectors determines the range of frequencies that are reflected.
- Reflections from the rear wall of a large hall can create audible echoes because of long path-length differences. Thus the rear wall is usually absorptive. When added absorption is undesirable, reflective diffusers can be placed on the rear wall.
- Speech intelligibility is the highest design priority for any hall intended for spoken word. Sound systems are often used to overcome acoustical limitations and to assist intelligibility.
- Speech intelligibility can be estimated using subject-based measures, or with analytical acoustical measures such as the articulation index (AI) and percentage articulation loss of consonants, known as %Alcons.
- Early sound can be considered in terms of early reverberation decay time, intimacy, clarity, and lateral spaciousness. Late reverberant sound can be considered as late reverberation decay time, warmth, loudness, and brilliance.
- The greater the energy in the early reverberation, the better the clarity. Late reverberation can affect perception of the liveness of sound. More late reverberant energy can increase liveness or fullness. As late reverberant energy increases, clarity relatively decreases.
- Brilliance describes sound that has presence and clearness. It is achieved with adequate high-frequency energy from reflecting surfaces.
- A sense of spaciousness can be created by early reflections from side walls, called lateral reflections, which occur within 80 msec of the arrival of the direct sound.
- According to Beranek, concert halls rated high in sound quality by qualified listeners had a well-defined ITDG of about 20 msec or less. Smaller ITDG values, because they denote a more intimate sound field, are desirable.

- Bass ratio can be used to characterize a sense of acoustical warmth that is often attributed to longer reverberation times at low frequencies.
- A balcony can decrease the distance from the stage to some seating areas. Care must be taken to avoid acoustical shadowing in the seating areas underneath the balcony.
- A raked floor improves fidelity in the seating area because the listener receives more direct sound than would be available on a flat floor.

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APPENDIX A

Overview of TDS and MLS Analysis[1]

Acoustics is often described as being both an art and a science. On one hand, most acousticians will agree that the pursuit of sound quality will never omit the subjective human response to sound and indeed the ear will always be the final arbiter of sound quality. However, as science has become more capable, objective means have assumed a larger role in assessing the quality of an acoustical space.

As we observed in previous chapters, basic tools such as a sound-level meter can provide a convenient measure of the fundamental nature of a sound, but the scope of the data is limited. Here we consider the more advanced techniques of time-delay spectrometry (TDS) and maximum-length sequence (MLS) measurements that can impart a deeper understanding of a room's influences on a sound's properties.

Both dedicated and software-based systems are widely used to perform a variety of acoustical measurements. Common measurement parameters include:

- Room reverberation time
- Room resonances
- Energy-time curves (specific reflection levels)
- Impulse response
- Loudspeaker delay times
- Pseudoanechoic loudspeaker frequency response in normal rooms
- Reverberant loudspeaker frequency response

These measurements can be invaluable in analyzing, troubleshooting, and improving room acoustics and sound reproduction.

Basic Measurement Instruments

A sound-level meter is probably the most basic sound measurement device and is useful for measuring the sound-pressure level (SPL) at a particular location. Although an SPL measurement may be useful for many purposes, it does not give an indication of the sound quality. Measuring sound quality, at a minimum, also requires the measurement

[1]Contributed by Geoff Goacher and Doug Plumb.

of frequency response, for example, using octave or 1/3-octave bands. There are a number of instruments and methods for measuring frequency response. Another important requirement for sound-quality measurements is time response. This is frequently used for measuring reverberation time, speech intelligibility, and other time-domain phenomena. Again, there are a variety of instruments used for measuring this factor.

One problem that has traditionally plagued the measurement of these room acoustics parameters is the presence of acoustical background noise. Background noise typically resembles pink noise, which has a power spectrum of –3 dB/octave across the frequency spectrum. Background noise is typically more prevalent at lower frequencies because it easily travels through walls and structures.

It was found that this noise could only be overcome in room acoustics measurements by exciting the room with stimulus energy so great that the background noise becomes insignificant. In the past, this was accomplished by using loud impulses such as gunshots as a test signal for impulse response and reverberation-time measurements. This method has technical flaws and provides only a rudimentary look at how sound behaves in rooms. Ultimately, it was concluded that to effectively study room acoustics and establish a correlation between certain acoustical parameters and sound quality, instruments with both time and frequency measurement capabilities as well as significant noise immunity were required.

Time-Delay Spectrometry Techniques

In response to this need for better measurement techniques, in the late 1960s, Richard C. Heyser refined a measurement method known as time-delay spectrometry (TDS). By the early 1980s, a TDS measurement system had been implemented commercially as a time-energy-frequency (TEF) analyzer. TEF uses time-delay spectrometry to derive both time and frequency response information from measurements.

The basic operating principle of time-delay spectrometry resides in a variable-frequency sweep excitation signal and a receiver with sweep tuning synchronized with that signal. A critical third element is an offset facility that can introduce a time delay between the swept excitation signal and the receiver. The latter is not needed in electric circuits in which signals travel near the speed of light, but it is vital in acoustical systems in which sound travels at only 1,130 ft/sec.

The coordination of the outgoing sine-wave sweep and the sharply tuned tracking receiver is shown in Fig. A-1. In this illustration, the horizontal axis is time and the vertical axis is frequency. The signal begins at A (t_1, f_1) and sweeps linearly to B (t_3, f_2). After a time delay (t_d) the receiver sweep begins at C (t_2, f_1) and tracks linearly at a rate identical to that of the signal to D (t_4, f_2). The receiver, being delayed by the time t_d, is now perfectly tuned to receive the swept-signal sine wave after it has traveled in air for the time t_d. At any instant during the sweep, the receiver is offset f_0 Hz from the signal.

The advantages of this arrangement of swept exciting signal and offset-swept tuned receiver are manifold. For example, it takes a certain amount of time for the signal to reach a wall, and it takes a certain amount of time for the reflection to travel back to the measuring microphone. By offsetting the receiver an amount equal to the sum of the two, the receiver examines only that specific reflected component. The offset may be viewed as a frequency offset. When the time offset in the previous example is set to accept the reflection from the wall, the unwanted reflections arrive at the microphone when the analyzing receiver is tuned to some other frequency; hence they are rejected.

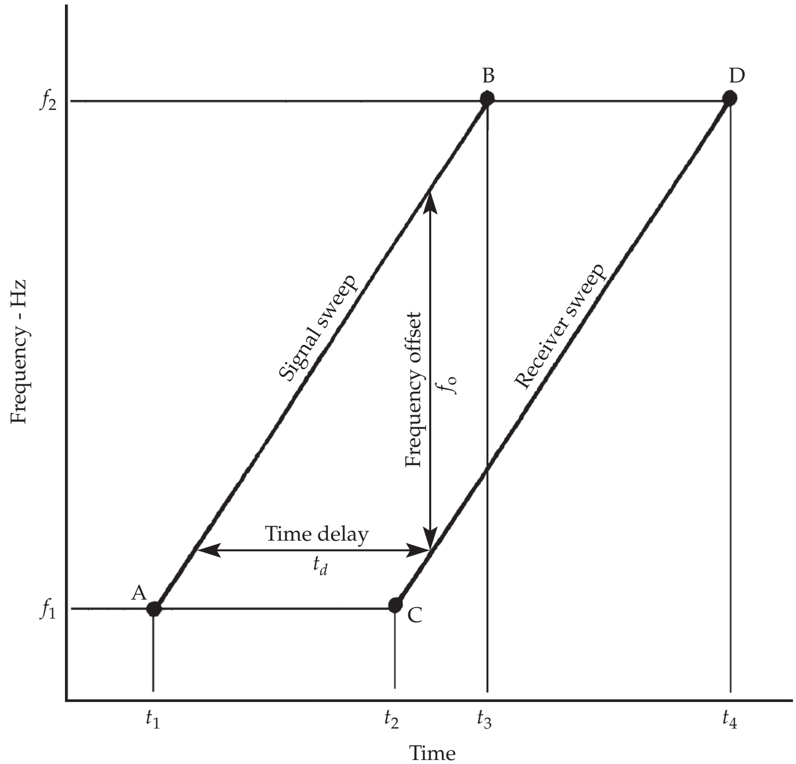

FIGURE A-1 The basic principles of Heyser's time-delay spectrometry (TDS) measurements. The outgoing signal is swept linearly from A to B. After a delay that is appropriate to select the reflection desired, the receiver is swept from C to D. Only energy from the desired reflection is accepted; the receiver is detuned to all others.

In this way the receiver is detuned for noise, reverberation, and all unwanted reflections in the measurement. The frequency response of only that particular desired reflection or sound is displayed for study.

Also note that the full signal energy is applied over the testing time throughout the swept spectrum. This is in contrast to applying a brute-force signal to a system, which often results in driving elements into nonlinear regions. TEF analysis was a breakthrough in measurement tools for evaluating the effects of room acoustics on perceived sound quality. One of its primary benefits is that the original TDS signal from a test can be stored and then repeatedly modulated with delayed versions of the sine and cosine chirp to get multiple snapshots of the time versus frequency content. These frequency-response slices allow for generation of three-dimensional waterfall graphs of energy versus time versus frequency.

Another feature of TEF measurements is that it allows for pseudoanechoic measurements in ordinary rooms by only measuring the initial sound plus the first few (or no) reflections. This technique can be used to take loudspeaker measurements and ignore the room effects by removing reflections from the measurement. The effect of reflections can therefore be studied.

The ability to remove undesired information from time-response sound measurements is often referred to as gating. Gating permits audio analyzers to remove reflections from a measurement that occur beyond a certain point in time after the room has initially been stimulated, and hence the term, pseudoanechoic measurements.

One disadvantage of gating is the loss of resolution that occurs from truncating the time response. For example, room reflections often occur within 5 msec of the direct sound in a typical room with 10-ft ceilings. In a pseudoanechoic measurement made in such a room, all data that are taken 5 msec after the initial stimulus would have to be removed, limiting the resolution of the measurement. For example, a 5-msec gate limits the lowest frequency that can be measured to 1/5 msec = 200 Hz. Frequency-response curves are smoothed as a result and frequency effects that appear with resolution finer than 200 Hz appear with reduced detail.

TDS swept tone measurements have two major advantages. First, the postprocessing causes unwanted harmonics to be rejected from the measurement, making the measurement accuracy less dependent on system linearity. Devices that add a large distortion component can be reliably measured for frequency response. Second, the swept tone can be used over long durations of time, injecting much energy into the room. This has the effect of increasing measurement signal-to-noise ratio.

The major disadvantage of TDS is that a new measurement must be taken each time a change in the time resolution is desired. If it was desired to take the above measurement again with a time window of, for example, 10 msec, another measurement would have to be taken using a different sweep rate.

Despite the tremendous advantages that were brought about by TDS measurement capabilities, acousticians began searching for a measurement system having the advantages of TDS with the additional capability that a single measurement could later be reprocessed with a different time window. Therefore, only one physical measurement would be required for a complete view of the response. This led to the development of maximum-length sequence measurement techniques.

Maximum-Length Sequence Techniques

The maximum-length sequence (MLS) measurement technique was refined in the 1980s and has been found to be an excellent measurement method for room acoustics. MLS measurements use a pseudorandom binary sequence to excite a system and/or room with a test signal resembling wideband white noise. The noise-rejection capabilities of MLS are such that the noise excitation can be played at a low level while maintaining a good signal-to-noise ratio and measurement accuracy. An MLS analyzer can also play a test signal for a long period of time, injecting enough energy into a room to reduce the effects of acoustical background noise to an acceptable level. Like TDS, MLS measurements also have very good distortion immunity.

The binary sequence used to produce the noise signal is precisely known and generated from a logical recursive relationship. Recordings of the test signal played through a system are used to generate an impulse response of the system using a method known as the fast Hadamard transformation.

The ideal impulse can never be realized in practice, but the function can be approximated perfectly up to a set frequency limit. This limit is due to the aperture effect. An example would be a system excited by a pulse with time duration of 1/1,000 sec. The aperture effect will reduce the frequency content above 1,000 Hz when the frequency response is calculated from the recorded data.

Sampling rates of 44.1 kHz and higher in digital measurements prevent the aperture effect from affecting measurements in the audible frequency range. The sampling theorem proves that all of the information that is contained below 22.05 kHz can be recovered completely. This ideal is realized in practice with PC-based measurement tools. There is no reason or need to increase this sampling rate when taking measurements for the audible frequency range.

The fact that the impulse response of the system can be determined when using MLS-type stimulus represents a significant advantage of MLS systems over TDS systems. The impulse response can be used to calculate all remaining linear parameters of a system. These include the various forms of frequency-response measurement as well as time measurement. Knowledge of the impulse response of a system permits the construction of a square-wave response, triangular-wave response, or any other form of time response required. All intelligibility calculations can easily be done with only knowledge of the impulse response. When the complete time response of a system is known, the frequency response can be calculated directly using a Fourier transform. Frequency response, by definition, is the Fourier transform of a system's time response obtained by impulse excitation.

Summary

The development of advanced acoustical measurement techniques such as TDS and MLS makes it possible to perform highly accurate measurements at low cost. These measurements include fractional-octave frequency response, resonance identification, energy-time curves, and reverberation time. These types of data can provide an invaluable aid to any sound contractor, acoustical consultant, audiophile, recording engineer, or loudspeaker builder.

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APPENDIX B

Room Auralization[1]

In the past, acousticians have used a wide range of techniques to design room acoustics. Despite the limitations of many techniques, they have produced spaces with superb acoustical performance. However, original acoustical designs or modifications to existing designs could not be tested prior to construction, except by evaluating scale models.

In contrast to older methods, it would be remarkable if designers could listen to the sound field in a proposed space before the space was built. In this way, we might avoid some acoustical problems and evaluate different design options and surface treatments. This process of acoustical rendering, analogous to visual rendering used by architects, has been called auralization. By definition, auralization is the process of rendering audible, by physical or mathematical modeling, the sound field of a source in a space, in such a way as to simulate the binaural listening experience at a given position in the modeled space. This has been described by Kleiner et al. and Dalenbäck et al. Today, such auralization software provides acousticians with tools to predict and simulate the performance of music halls, recording studios, listening rooms, and other critical acoustical spaces.

History of Acoustical Modeling

In the broadest of terms, to characterize a room it would be necessary to find a way to follow the reflection path history of complex sound reflections as they proceed in time through the room.

This reflection path history of sound level versus time as picked up at a location in the room is called an echogram, and the process is called ray tracing. In this approach, we follow each ray through its reflection path history and catalog those rays that pass through a small volume at the listening position. This approach is easy to visualize, and if we use audio slow motion and slow down sound from 1,130 ft/sec to 1 in/sec, we can color-encode each sound ray and record the event.

Consequently, in the late 1960s, acousticians began using ray tracing to determine an echogram and estimate reverberation time. By the 1980s, ray tracing had been widely used. In ray tracing, the total energy emitted by a source is distributed according to the radiation characteristics of the source into a specified number of directions. In the simplest form, the energy of each ray is equal to the total energy divided by the number of rays. Depending on the type of surface, each ray from each boundary in its reflection history is either specularly reflected, in which the angle of incidence equals the angle of

[1]Contributed by Peter D'Antonio.

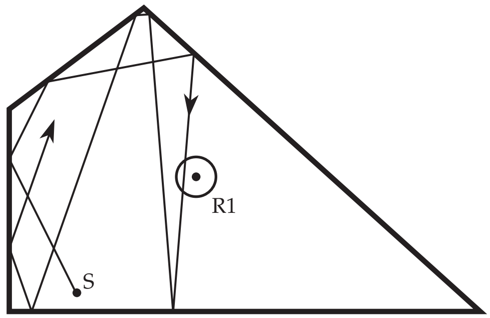

Figure B-1 Ray tracing example showing emitted ray from source S entering the circular cross-section detection area of receiver cell R1 after three specular reflections.

reflection, or diffusely reflected, in which the direction of the reflected ray is randomized. The reflected energy is diminished by absorption and also by the spherical attenuation due to propagation. (In ray tracing this is achieved automatically because of the fixed receiver size.) The number of rays passing through a receiver cell determines the sound-pressure level.

An example of ray tracing is shown in Fig. B-1. A ray emitted from source S is shown reflecting from three surfaces before it passes through the circular cross-section detection area of receiver R1. The energy contributions of the various rays to a certain receiver cell are added within prescribed time intervals, resulting in a histogram. Because of time averaging and the strongly random character of the ray arrivals, the histogram will only be an approximation of the true echogram. Ray tracing is straightforward, but its efficiency is achieved at the expense of limited time and spatial resolution in the echogram. The number of rays and the exact angles of emission from the source determine the accuracy of the sampling of room details and the reception of rays within the receiver volume.

In the late 1970s, another approach, called the mirror image source method (MISM), was developed to determine the echogram. In this approach, a virtual image of the actual source is determined by reflecting the source perpendicularly across a room boundary. Thus the image source is located at a distance equal to twice the perpendicular distance *d* to the reflecting boundary. The distance between S1 and R1 in Fig. B-2 is equal to the reflection path from S to R1. The reflections of all real and virtual images across the room boundaries create a set of mirror images. The arrival times in the echogram are now simply determined by the distances between these virtual images and the receiver. When applied to a rectangular room, all mirror sources are visible from every position in the room and the calculation is fast. In irregular rooms, however, this is not the case and validity tests are performed. For example, in Fig. B-2, it can be seen that receiver R1 can be reached by a first-order reflection from surface 1, but receiver R2 cannot. This means R1 is "visible" from S1 and R2 is not. Therefore, validation of each image is required. Since each further reflection order multiplies the number of image sources to be tested (by the number of walls minus one), the number of sources to validate grows very rapidly.

Two approaches were developed in the late 1980s to minimize validation of sources. One was the hybrid method that used specular ray tracing to find potentially valid

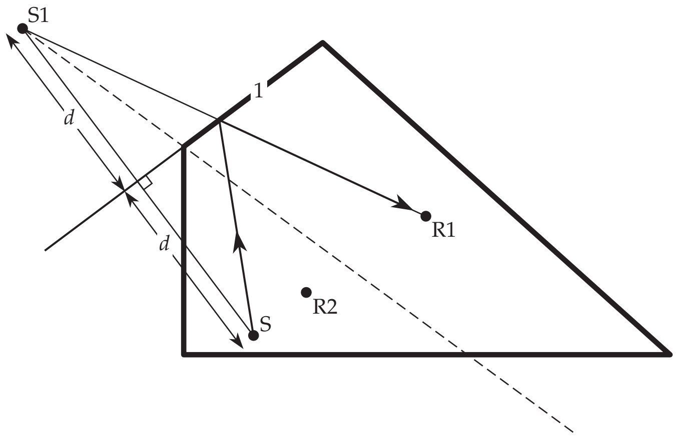

Figure B-2 Mirror image source model (MISM) construction showing real source S, virtual source S1, boundary 1, and receiver R1.

reflection paths and validated only those, whereas another used cones or triangular-based pyramidal beams instead of rays. First- and second-order cone tracing examples are shown in Figs. B-3 and B-4. If the receiver point lies within the projection of the beam, a likely visible image source has been found and validation is not performed. Both of these methods can calculate a more detailed specular echogram than ray tracing and are also faster than MISM at the expense of leaving out reflection paths for the late part in the echogram (when the ray-cone separation is becoming larger than individual walls). However, it was only in ray tracing that diffuse reflection can be included in an efficient way, so the improved algorithms left out a very central acoustical phenomenon. Contemporary geometrical room modeling programs utilize a more complex combination of methods such as MISM for low-order reflections, and randomized cone tracing for higher-order reflections. Randomization is thus reintroduced to handle diffuse reflection.

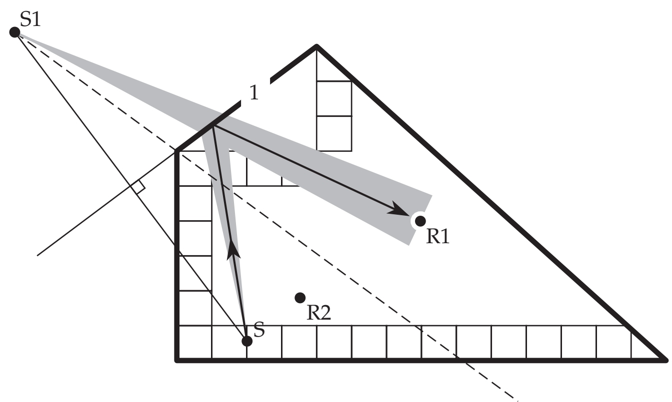

Figure B-3 Equivalence of MISM and cone tracing for first-order reflection from surface 1.

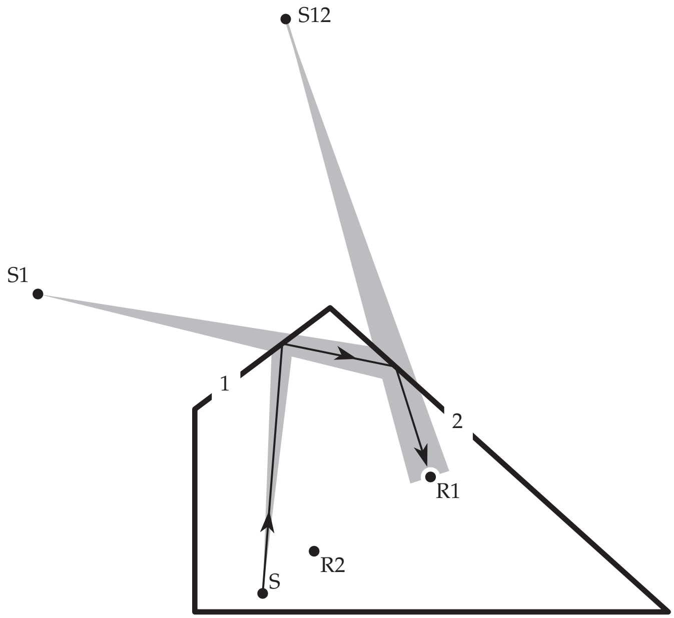

Figure B-4 Second-order reflection of source S from surfaces 1 and 2 to receiver R1 using cone tracing. S1 and S12 are virtual sources.

The Auralization Process

The auralization process begins with predicting octave band echograms based on a 3D CAD model of a room, using geometrical acoustics. Since geometrical acoustics is limited to high frequencies roughly above 125 Hz, there are attempts to model low frequencies using boundary element wave acoustics to extend the range of applicability. Frequency-dependent material properties (absorption and scattering) are assigned to room surfaces, and frequency-dependent source directivities are assigned to sound sources. From these echograms a great number of numerical objective measures, such as reverberation times, early and late energy ratios, and lateral energy fraction, can be estimated.

Figure B-5 illustrates the information needed to generate an echogram. Everything about the room needs to be defined, such as sound sources, the room's geometry and surface treatments, and listeners. Each source, be it a natural source or loudspeaker, is defined by its octave-band directivity at least at 125, 250, and 500 Hz, and 1, 2, and 4 kHz. The directivity balloons (a 3D version of the polar diagrams shown in Fig. B-5) describe how the sound is directed in each octave band. The surface properties of the room's boundaries are described by their absorption and scattering coefficients.

Scattering Coefficients

A method to determine scattering coefficients has been standardized as *ISO 17497-1:2004 Acoustics—Sound-Scattering Properties of Surfaces—Part 1: Measurement of the Random-Incidence*

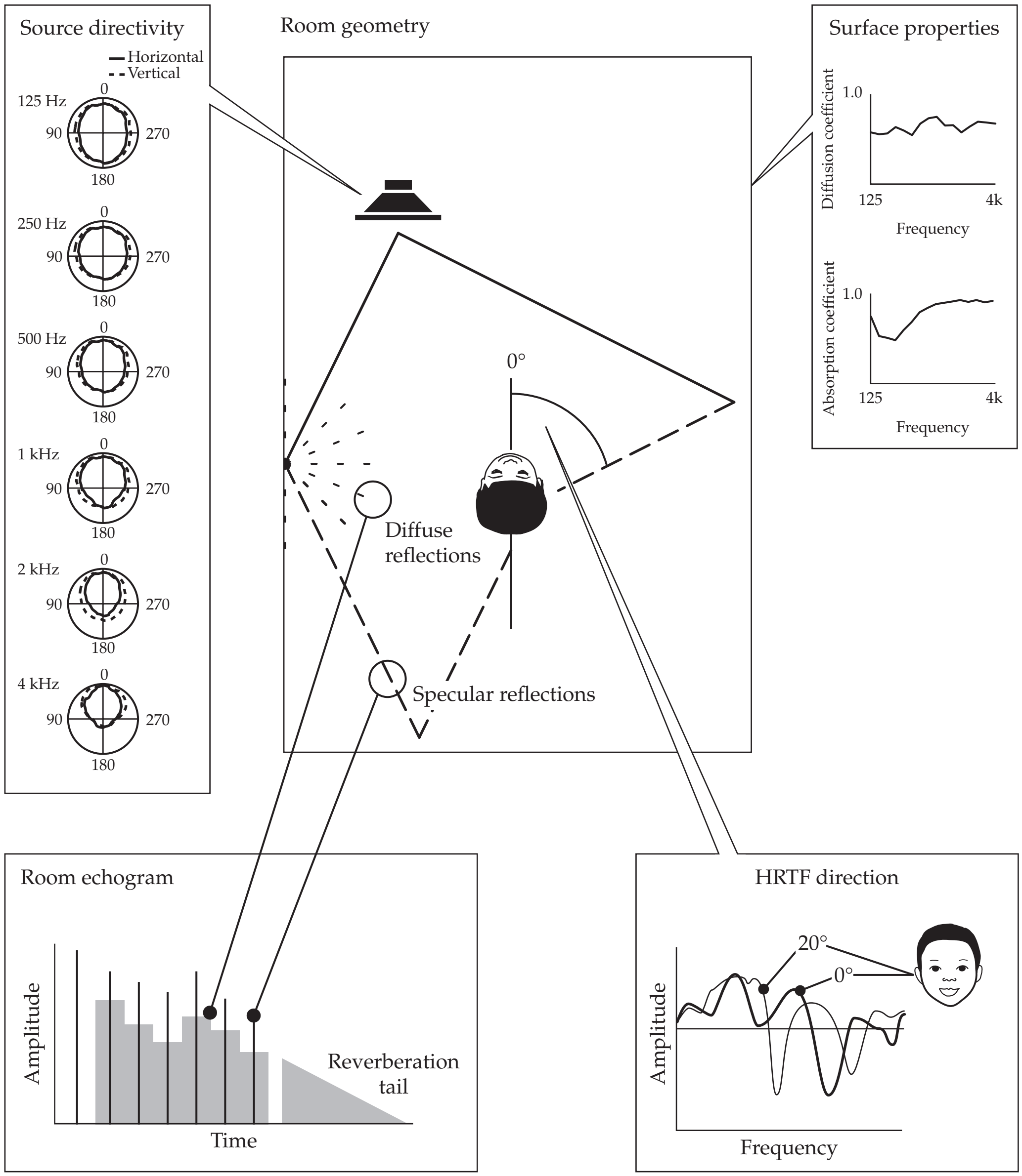

FIGURE B-5 Simulation of a room echogram using geometrical acoustics and descriptions of the room geometry, the surface properties, the source directivity, and the receiver's HRTFs.

Scattering Coefficient in a Reverberation Room. The method requires measuring the reverberation time with a suitably sized sample in a reverberation chamber mounted on a circular rotatable turntable in the stationary and rotating condition. In brief, the diffusion coefficient *d* is a measure of the uniformity of the reflected sound. The purpose of this coefficient is to enable the design of diffusers and to also allow acousticians to compare the performance of surfaces for room design and performance specifications. The scattering coefficient *s* is a ratio of sound energy scattered in a nonspecular manner to the total reflected sound energy. The purpose of this coefficient is to characterize surface scattering for use in geometrical room modeling programs.

Both the coefficients are simplified representations of the true reflection behavior. It is necessary to devise simple metrics, rather than trying to evaluate the reflection characteristics for all possible source and receiver positions, because the amount of data is otherwise too large. The coefficients attempt to represent the reflection by a single parameter, maximizing the information carried by that single number. The difference between diffusion and scattering coefficients is the emphasis on which information is most important to be preserved in the data reduction. For diffuser designers and acoustical consultants, it is the uniformity of all reflected energy which is most important; for room acoustical modeling, it is the amount of energy scattered away from specular angles. The difference between the definitions may appear subtle, but in practice it is significant.

Receiver Characterization

Each receiver is described by an appropriate response. If a binaural analysis is to be carried out, the listener is characterized by the head-related transfer function (HRTF). These frequency responses describe the difference in the response at each ear with and without a listener present. An example of HRTF for sound arriving at 0° and at 20° is shown in Fig. B-5. Thus in the generation of the echogram, each reflection contains information about its arrival time, its level, and its angle with respect to the receiver. The specular reflections are indicated as lines and the diffuse reflections as histograms.

Echogram Processing

Figure B-6 illustrates one method for converting an echogram into an impulse response by adding phase information. For each reflection the magnitude is determined by fitting the octave band levels (A through F), determined by a combination of the loudspeaker directivity, the absorption coefficient, and the scattering coefficient. The phase for early specular reflections is often determined by minimum-phase techniques using the Hilbert transform. The impulse response for each reflection is then obtained by an inverse fast Fourier transform (IFFT).

There are several ways in which one can process the predicted echograms for final audition. These include binaural processing for headphones, including headphone equalization or loudspeakers with crosstalk cancellation, monaural processing, stereo processing, 5.1-channel processing, or B-format processing (for ambisonic replay). Figure B-7 illustrates the method for binaural processing. The impulse response of each reflection in turn is modified by the HRTF for the appropriate angle of arrival and transformed into the time domain using an IFFT. Thus each reflection is transformed into a binaural impulse response, one for the left ear and one for the right. This is done for each of the reflections from 1 to *N*, and the resultant binaural impulse responses are added to form the total left and right room impulse responses. We are now able to auralize the room. This is achieved by convolving the binaural impulse responses with anechoic or nonanechoic music, as illustrated in Fig. B-8.

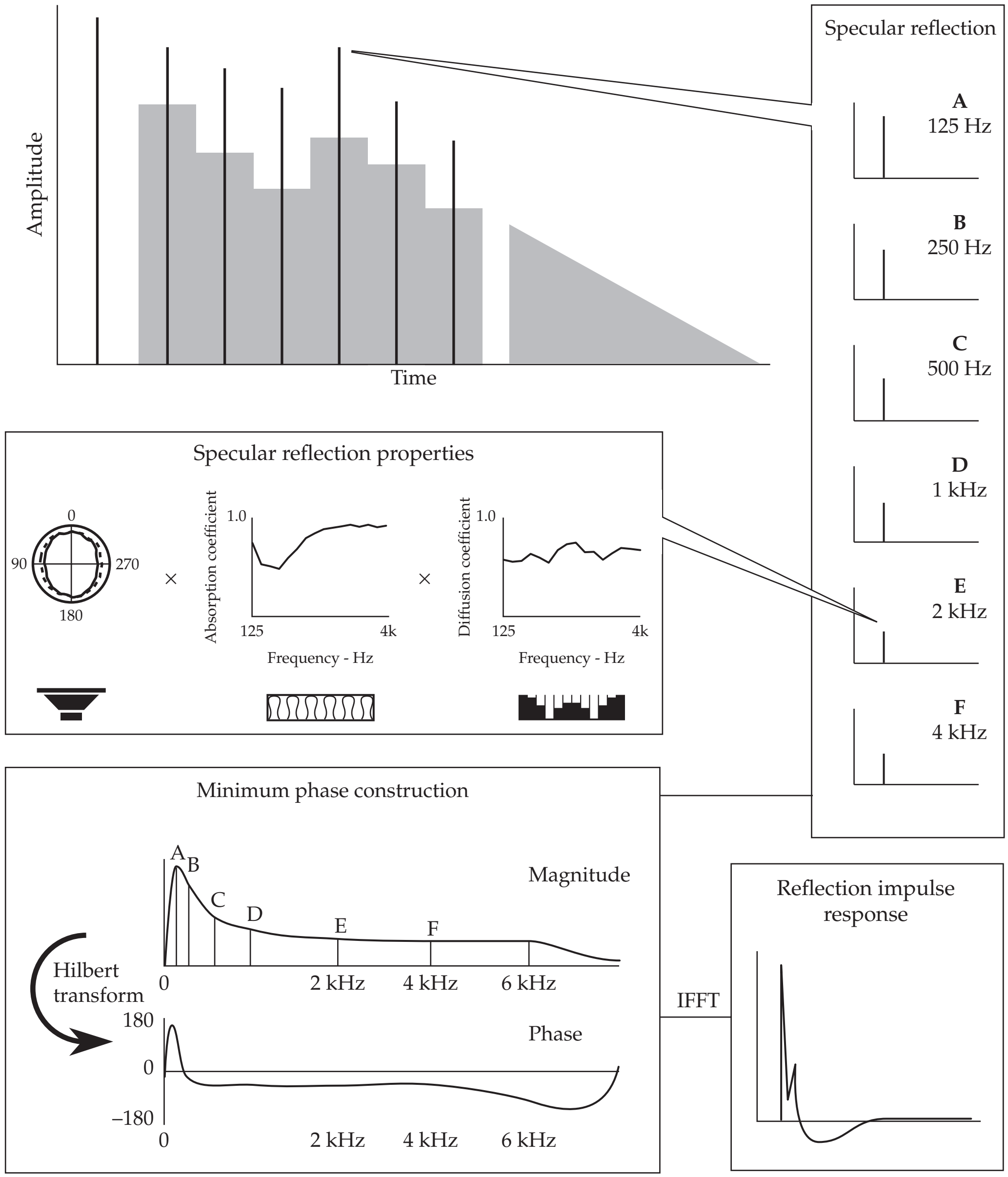

Figure B-6 Transfer function construction is used to convert an echogram reflection into an impulse response.

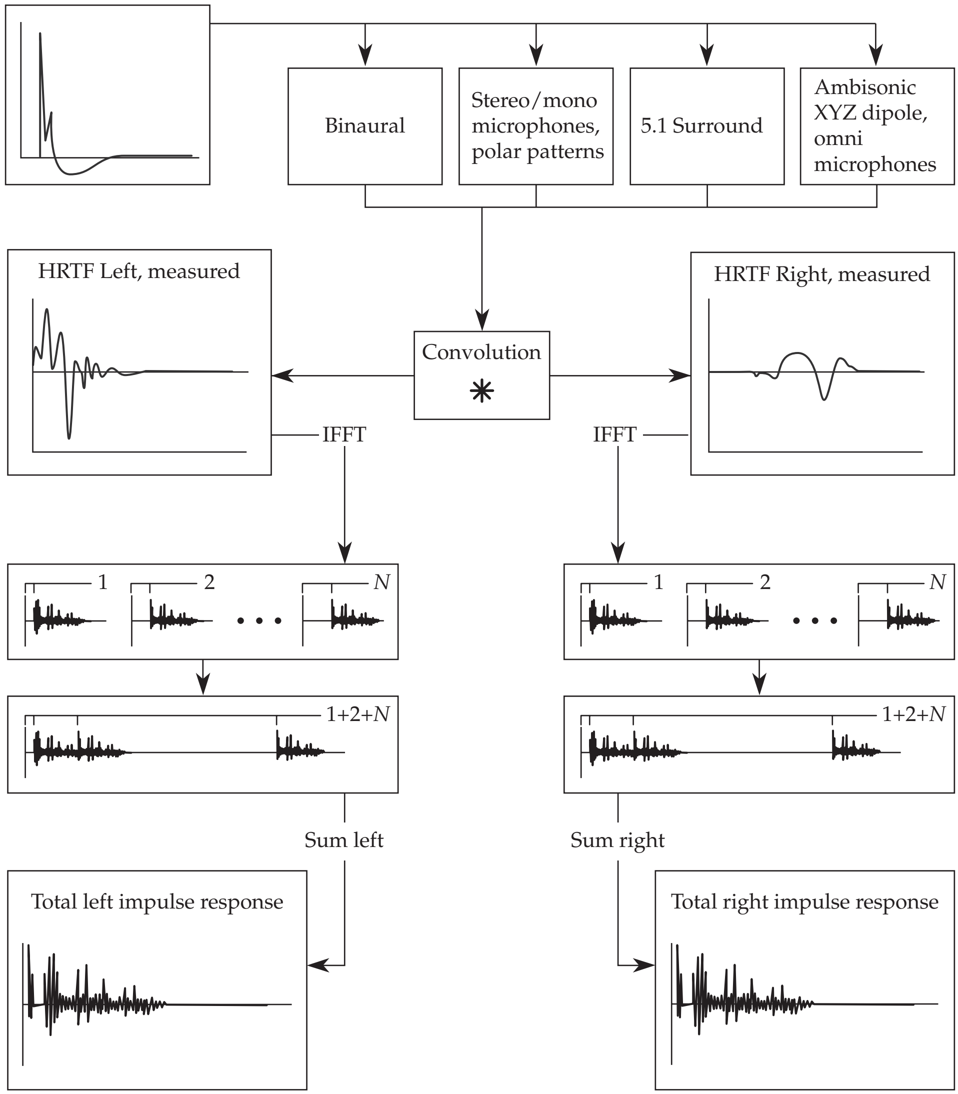

FIGURE B-7 For binaural auralization, the postprocessing step converts the room impulse response into a left- and right-ear binaural impulse response through convolution with the HRTFs.

Room Model Data

Auralization models are created either in AutoCAD or with a text program language, which allows the user to set variables such as length, width, and height. Changing these variables allows one to easily change the model. Objects can also be created with variable orientations, to model the effect of different designs.

The CATT-Acoustic program (www.catt.se) is one example of a 3D auralization modeling program. It generates the early part of the echogram showing the specular

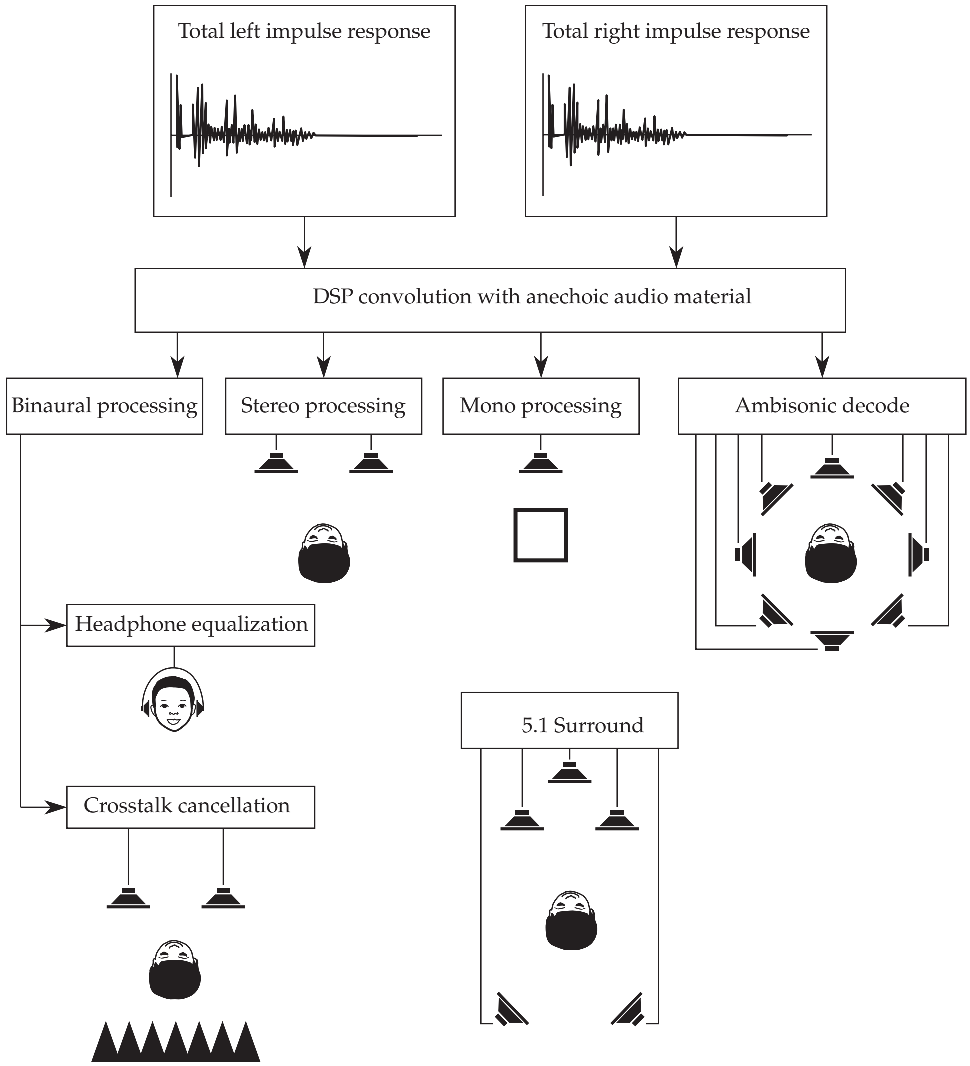

FIGURE B-8 Auralization of the room is achieved by convolving the binaural impulse responses with music.

and diffuse reflections, the emitted angles of the sound from the source, the arrival angles at the receiver, and an isometric view of the room. Reflection paths from source to receiver can be viewed in the room illustration. In this way, one can determine the boundary responsible for a possibly problematic reflection.

Figure B-9A illustrates the objective parameters that can be determined directly from the echogram after backward integration; in this example, a listening room with five matching monopole loudspeakers in the 5.1-channel ITU surround configuration (described in Chap. 20) is modeled. The early decay time is the time it takes to reduce the sound-pressure level from its steady-state level by 10 dB. T-15 and T-30 refer to the time

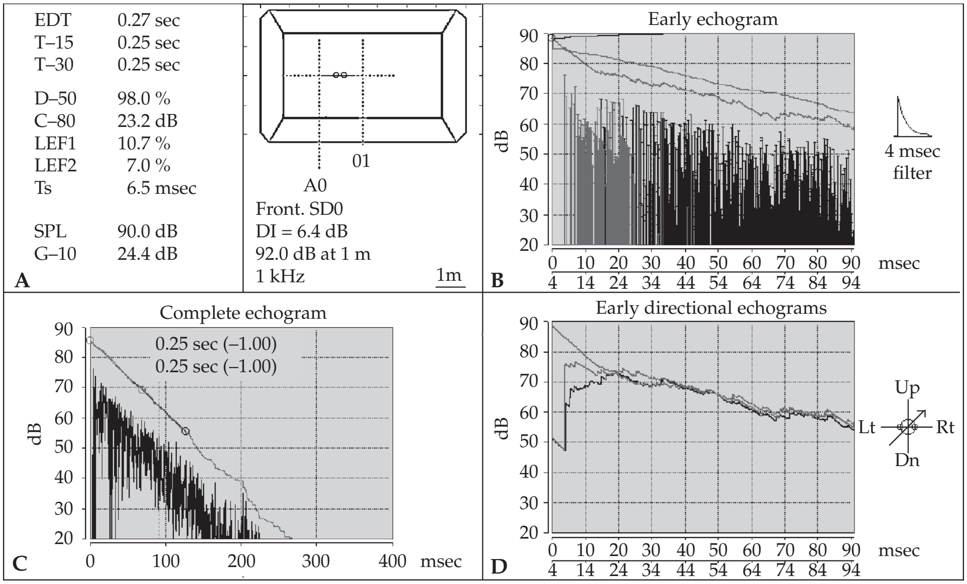

Figure B-9 Data display from 3D room model. (A) List of the objective parameters. (B) Early echogram with backward integration. (C) Complete echogram with T-15 and T-30 from backward integration. (D) Directional echograms (up/down, left/right, and front/back). (*CATT-Acoustic.*)

it takes to decrease the sound-pressure level from –5 to –20 and from –5 to –35 dB, respectively. There are several objective parameters based on ratios of early- to late-arriving sound. The definition criterion, referred to as D-50 in the program, is a measure of speech intelligibility. It is a percent ratio of the sound arriving in the first 50 msec following the direct sound to the total sound. The music clarity index, referred to as C-80, is based on a ratio of the first 80 msec of sound to the rest of the sound. Lateral energy fractions, referred to as LEF1 and LEF2, are measures of the impression of spaciousness. They are based on the ratio of laterally arriving reflections between 5 and 80 msec to all of the omnidirectionally arriving reflections after 80 msec.

The center-of-gravity time Ts is a measure constructed to describe where the sound is concentrated in the echogram. Ts has a low value if the arriving sound is concentrated in the early part of the echogram and a high value if the early reflections are weak or if the decay is slow. Sound-pressure level is 10 times the 10-based log of the ratio of the pressure squared over a reference pressure squared. G-10 is the sound pressure at a location in a room as compared to what the direct sound from an omnidirectional source would give at a 10-m distance. G-10 is a measure that is used for characterizing how loud the sound will be, and can be compared between rooms.

Figures B-9B, C, and D show additional information: early echogram with backward integration, complete echogram with T-15 and T-30 from the backward integration, and directional echograms (up/down, left/right, and front/back).

Room Model Mapping

Another useful function of geometrical modeling programs is the ability to map the objective parameters, so we can see how a parameter varies over the listening area.

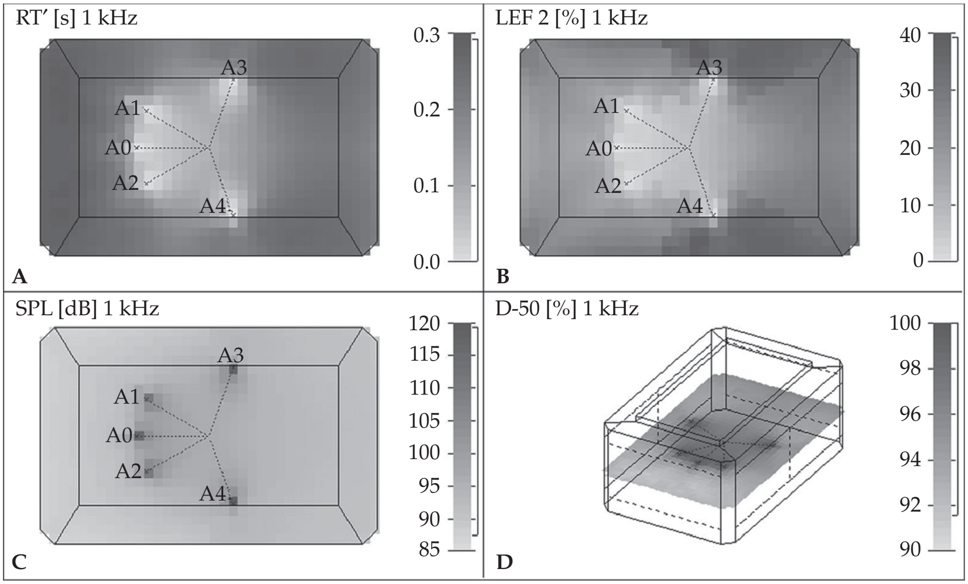

FIGURE B-10 Parameters are mapped at 1 kHz across the room for five matching loudspeakers in the 5.1-channel ITU surround configuration, left/center/right with monopole surrounds. (A) Perceived reverberation time RT′. (B) Lateral energy fraction LEF2. (C) Sound-pressure level SPL. (D) Energy ratio D-50. (*CATT-Acoustic*.)

(These maps are normally in color, giving a much better visual resolution than the grayscale versions presented here.) Figure B-10 shows a mapping of perceived reverberation time RT′ (which is related to the early decay time), the LEF2, SPL, and D-50 for the five matching monopole loudspeakers in ITU format.

Although these parameters are mostly useful for large-room analysis, they also give information in the case of smaller listening rooms. For example, they can illustrate how the effective listening area is affected by loudspeaker placement and surface treatment. For example, note that the RT′ value in Fig. B-10 is very low between the surround speakers (showing a relatively direct sound). In contrast, if dipole loudspeakers had been used for the surrounds (as in a THX configuration) the mapping would show the listener receiving less direct sound from the surrounds (and more diffuse sound). This demonstrates the ability of the mapping technique to illustrate different loudspeaker configurations in a room.

Another informative mapping plot shows how a given parameter changes with time. In Fig. B-11, the sound-pressure level is plotted in four time intervals so that we can view how the sounds from the various loudspeakers combine over time to energize the room. If a reflection-free zone is created, it will appear (in Fig. B-11B) as low SPL values around the listening position. In the same map, the effect of absorptive surfaces behind the front loudspeakers can be seen as a lower sound-pressure level. By selecting the time windows well, the efficiency of a design can be studied.

Binaural Playback

In addition to analyzing acoustical parameters, we can also listen to what the room might sound like with the specified acoustical design and loudspeaker configuration.

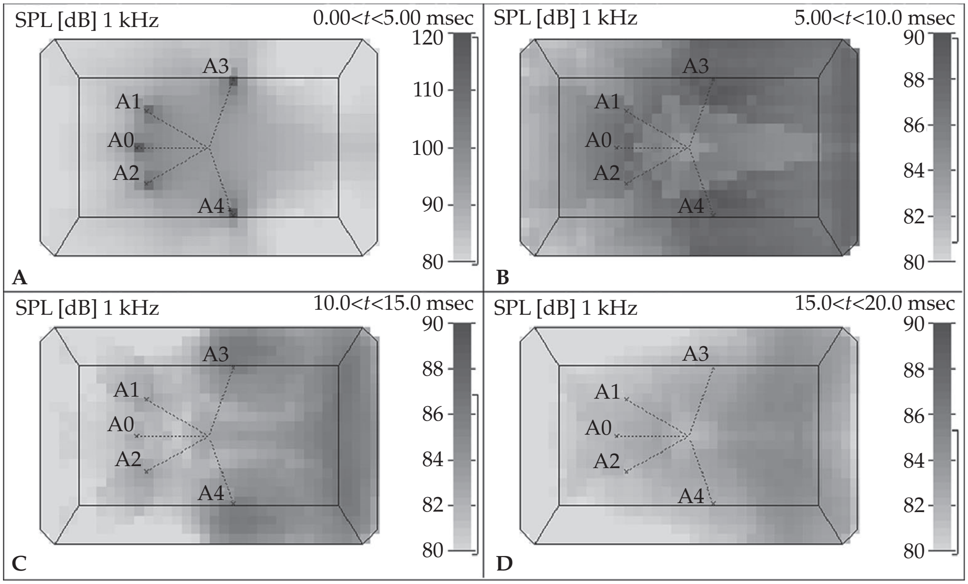

Figure B-11 The sound fields from the five loudspeakers combine over time to energize the room. (A) t < 5 msec. (B) 5 < t < 10 msec. (C) 10 < t < 15 msec. (D) 15 < t < 20 msec. (*CATT-Acoustic*.)

To do this, we combine the binaural impulse responses of the left and right ear with music. For binaural auralization, the binaural impulse responses are derived from the room impulse response by convolution of each reflection with HRTFs, which are stored in the program from a mannequin and also from a real person. The next step is to select an anechoic or nonanechoic music sample and convolve it with the binaural impulse responses. An anechoic music sample can be used to auralize the sound. Since the music is anechoic, it does not contain any room signature. Therefore, convolving with the room response allows us to audition the room at various source and listening positions. A nonanechoic source might be used to determine the extent to which a sound reproduction room influences what is heard by adding its own room signature to the sound. Thus by comparing the nonanechoic music sample with that heard in the room, we can make an evaluation of the influences that are introduced.

Summary

The modeling approach described here uses geometrical acoustics to generate the room impulse response of a virtual room. This impulse response can be postprocessed for several types of auralization, including monaural, stereo, 5.1-channel, binaural, and ambisonic. Real rooms can also be auralized if their impulse response is measured and applied to the model.

It should be reiterated that geometrical acoustics methods presently used are more accurate at higher frequencies and they cannot accurately represent lower frequencies. Geometrical acoustics can be combined with wave acoustics to extend the modeling to lower frequencies. Despite the many approximations involved, auralization is a valid and useful tool, especially in the hands of experienced acousticians.

APPENDIX C
Selected Absorption Coefficients

Material	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz
Porous Type						
Drapes: cotton, 14 oz/yd^2						
Draped to 7/8 area	0.03	0.12	0.15	0.27	0.37	0.42
Draped to 3/4 area	0.04	0.23	0.40	0.57	0.53	0.40
Draped to 1/2 area	0.07	0.37	0.49	0.81	0.65	0.54
Drapes: medium velour, 14 oz/yd^2						
Draped to 1/2 area	0.07	0.31	0.49	0.75	0.70	0.60
Drapes: heavy velour, 18 oz/yd^2						
Draped to 1/2 area	0.14	0.35	0.55	0.72	0.70	0.65
Carpet: heavy on concrete	0.02	0.06	0.14	0.37	0.60	0.65
Carpet: heavy on 40-oz hair felt	0.08	0.24	0.57	0.69	0.71	0.73
Carpet: heavy with latex backing on foam or 40-oz hair felt	0.08	0.27	0.39	0.34	0.48	0.63
Carpet: indoor/outdoor	0.01	0.05	0.10	0.20	0.45	0.65
Acoustical tile, ave, 1/2-in thick	0.07	0.21	0.66	0.75	0.62	0.49
Acoustical tile, ave, 3/4-in thick	0.09	0.28	0.78	0.84	0.73	0.64
Miscellaneous Building Materials						
Concrete block, coarse	0.36	0.44	0.31	0.29	0.39	0.25
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Concrete floor	0.01	0.01	0.015	0.02	0.02	0.02
Floor: linoleum, asphalt tile, or cork tile on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Floor: wood	0.15	0.11	0.10	0.07	0.06	0.07
Glass: large panes, heavy glass	0.18	0.06	0.04	0.03	0.02	0.02
Glass, ordinary window	0.35	0.25	0.18	0.12	0.07	0.04

(*Continued*)

Material	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz
Owens-Corning Frescor: painted, 5/8-in thick, mounting 7	0.69	0.86	0.68	0.87	0.90	0.81
Plaster: gypsum or lime, smooth finish on tile or brick	0.013	0.015	0.02	0.03	0.04	0.05
Plaster: gypsum or lime, smooth finish on lath	0.14	0.10	0.06	0.05	0.04	0.03
Gypsum board: 1/2-in, 2- × 4-in studs, 16-in centers	0.29	0.10	0.05	0.04	0.07	0.09
Resonant Absorbers						
Plywood panel: 3/8-in thick	0.28	0.22	0.17	0.09	0.10	0.11
Polycylindrical						
Chord 45-in, height 16-in, empty	0.41	0.40	0.33	0.25	0.20	0.22
Chord 35-in, height 12-in, empty	0.37	0.35	0.32	0.28	0.22	0.22
Chord 28-in, height 10-in, empty	0.32	0.35	0.3	0.25	0.2	0.23
Chord 28-in, height 10-in, filled	0.35	0.5	0.38	0.3	0.22	0.18
Chord 20-in, height 8-in, empty	0.25	0.3	0.33	0.22	0.2	0.21
Chord 20-in, height 8-in, filled	0.3	0.42	0.35	0.23	0.19	0.2
Perforated Panel						
5/32-in thick, 4-in depth, 2-in glass fiber						
Perf: 0.18%	0.4	0.7	0.3	0.12	0.1	0.05
Perf: 0.79%	0.4	0.84	0.4	0.16	0.14	0.12
Perf: 1.4%	0.25	0.96	0.66	0.26	0.16	0.1
Perf: 8.7%	0.27	0.84	0.96	0.36	0.32	0.26
8-in depth, 4-in glass fiber						
Perf: 0.18%	0.8	0.58	0.27	0.14	0.12	0.1
Perf: 0.79%	0.98	0.88	0.52	0.21	0.16	0.14
Perf: 1.4%	0.78	0.98	0.68	0.27	0.16	0.12
Perf: 8.7%	0.78	0.98	0.95	0.53	0.32	0.27
With 7-in airspace with 1-in mineral fiber, 9-10 lb/ft^3 density, 1/4-in cover						
Wideband, 25% perf or more	0.67	1.09	0.98	0.93	0.98	0.96
Mid peak, 5% perf	0.60	0.98	0.82	0.90	0.49	0.30
Low peak, 0.5% perf	0.74	0.53	0.40	0.30	0.14	0.16
With 2-in airspace filled with mineral fiber, 9-10 lb/ft^3 density						
Perf: 0.5%	0.48	0.78	0.60	0.38	0.32	0.16

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Appendix A Overview of TDS and MLS Analysis

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Glossary

absorption In acoustics, when some energy is taken away from sound, for example, the changing of sound energy to heat.

absorption coefficient The fraction of sound energy that is absorbed at any surface. It has a theoretical value between 0 and 1 and varies with the frequency and angle of incidence of the sound.

acoustics The science of sound that deals with the production, control, transmission, and reception of sound. It can also refer to the effect a given environment has on sound.

active sound absorber A resonant sound absorbing structure.

AES Audio Engineering Society.

algorithm Procedure for solving a mathematical problem.

ambience The distinctive acoustical characteristics of a given space.

amplifier, line An amplifier designed to operate at intermediate levels. Its output is usually on the order of one volt.

amplifier, output A power amplifier designed to drive a loudspeaker or other load.

amplitude The instantaneous magnitude of an oscillating quantity such as sound pressure. The peak amplitude is the maximum value.

amplitude distortion A distortion of the wave shape of a signal.

analog An electrical or acoustical signal whose frequency and level vary continuously in direct relationship to the original signal.

anechoic Without reflection or echo.

anechoic chamber A room designed to suppress internal sound reflections, used for acoustical measurements and testing.

antinodic Point of maximum vibration in a vibrating body.

arrival gap The time between the arrival of the direct signal and reflections. *See also* initial time-delay gap.

articulation A quantitative measure of the intelligibility of speech, can be expressed as the percentage of speech items correctly perceived.

artificial reverberation Reverberation generated usually by algorithm to simulate the sound field of real or imagined acoustical spaces.

ASA Acoustical Society of America.

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers.

attack The beginning of a sound, the initial transient of a musical note.

attenuate To reduce the level of an electrical or acoustical signal.

attenuator A device, usually a variable resistance, used to control the level of an electrical signal.

audio frequency An acoustical or electrical signal of a frequency that falls within the audible range of the human ear, usually taken as 20 Hz to 20 kHz.

audio spectrum *See* audio frequency.

auditory area The sensory area lying between the threshold of hearing and the threshold of feeling or pain.

auditory cortex The region of the brain receiving nerve impulses from the ear.

auditory system The human hearing system made up of the external ear, the middle ear, the inner ear, the nerve pathways, and the brain.

aural Having to do with the auditory mechanism.

A-weighting A frequency response adjustment of a sound-level meter that makes its reading approximately conform to human perception of sound.

axial mode A room mode produced by reflections from one pair of parallel surfaces of a room.

baffle A movable barrier (also known as a gobo) used in the recording studio to improve separation of acoustical signals from different sources. Also, the surface or board upon which a loudspeaker is mounted.

band-pass filter A filter that attenuates signals both below and above the desired passband.

bandwidth The frequency range passed by a given device or structure. Commonly measured as the width between –3-dB points.

basilar membrane A membrane inside the cochlea that vibrates in response to sound, exciting the hair cells.

bass The lower range of audible frequencies.

bass boost The increase in level of the lower range of frequencies, usually achieved by electrical circuits or acoustical reinforcement.

bass trap A structure designed to absorb low-frequency sound energy.

beats Periodic fluctuations resulting from superimposing signals of slightly different frequency.

binaural Listening with two ears or a recording and playback configuration emulating hearing with two ears.

bit The elemental “low” or “high” state of a binary number system.

boomy Colloquial expression for excessive bass response in a recording, playback, or sound-reinforcing system.

byte A grouping of bits in a digital system. One byte usually comprises 8 bits of data.

capacitor An electric component that passes alternating current but blocks direct current. Also called a condenser, it is capable of storing electric energy.

clipping The amplitude of the peaks of an electrical signal limited by electronic circuits or by overloading or exceeding the level capabilities of an electronic device. It is a distortion of the signal.

cochlea The portion of the inner ear that changes the mechanical vibrations of the cochlear fluid into electrical signals. It can be modeled as the frequency-analyzing portion of the auditory system.

coincidence effect Sound energy falling on a wall having a frequency coincident with the natural period of the wall sustains the wall vibration. This results in a decrease in the transmission loss for the wall near that frequency.

comb filter A distortion produced by combining an electrical or acoustical signal with a delayed replica of itself. The result is constructive and destructive interference that results in peaks and nulls introduced into the frequency response. When plotted to a linear frequency scale, the response resembles a comb, hence the name.

compression Reducing the dynamic range of a signal by digital or analog circuits that reduce the level of loud passages.

condenser *See* capacitor.

correlogram A graph showing the correlation of one signal with another.

cortex *See* auditory cortex.

crest factor Peak value divided by RMS (root-mean-square) value.

critical band In human hearing, a narrow band over which frequency components within the band mask a given tone or noise. Critical bandwidth varies with frequency but is usually between 1/6 and 1/3 octave wide.

crossover frequency The frequency corresponding to the −3-dB point of a network that divides signal energy. For example, in a loudspeaker with multiple radiators, the frequency where the response from different radiators crosses.

crosstalk The signal of one channel or circuit interfering with another.

cycles per second The frequency of an electrical signal or sound wave. Measured in hertz (Hz).

dB *See* decibel.

dBA A sound-level meter reading with an A-weighting network simulating the human-ear response at a loudness level of 40 phons.

dBB A sound-level meter reading with a B-weighting network simulating the human-ear response at a loudness level of 70 phons.

dBC A sound-level meter reading with a C-weighting network simulating the human-ear response at a loudness level of 100 phons.

dBZ A sound-level meter reading with a zero weighting used to describe a flat frequency response from 10 Hz to 20 kHz ±1.5 dB.

decade Ten times any quantity or frequency range. The range of the human ear is about three decades.

decay rate A measure of the decay of acoustical signals, expressed as a slope in decibels/second.

decibel The bel is the logarithm of the ratio of two powers, and the decibel is one-tenth of a bel. Abbreviated dB. The human ear responds logarithmically and it is expedient to deal in logarithmic units in audio systems.

delay line A digital or analog device employed to delay one audio signal with respect to another.

diaphragm Any surface that vibrates in response to sound or is vibrated to emit sound, such as in microphones and loudspeakers. Also applied to wall and floor surfaces vibrating in response to sound or in transmitting sound.

dielectric An electrically insulating material. The material between the plates of a capacitor.

diffraction The spatial distortion (bending) of a wavefront caused by the presence of an obstacle in the sound field. The angle of diffraction is a function of wavelength relative to the size of the obstacle.

diffuser A device or physical surface causing the diffusion of sound, for example, through reflection phase-grating means.

diffusion The process of diffusing or scattering of sound.

diffusion coefficient The ratio of scattered intensity at 45° to the specular intensity.

digital audio A numerical representation of an analog signal. Pertaining to the application of digital techniques to audio tasks.

distance double law In pure spherical divergence of sound from a point source in free space, the sound-pressure level decreases 6 dB for each doubling of the distance. This condition is rarely encountered in practice but can be used to estimate sound changes with distance.

distortion Any change in the waveform or harmonic content of an original signal as it passes through a device. The result of nonlinearity within the device.

distortion, harmonic The change in the harmonic content of a signal when it passes through a nonlinear device.

DSZ Diffused sound zone.

dynamic range All audio systems are limited by inherent noise at low levels and by overload distortion at high levels. The usable region between these two extremes is the dynamic range of the system. Expressed in decibels.

dyne The force that accelerates a one-gram mass at the rate of 1 cm/sec. A standard reference level for sound pressure is 0.0002 dyne/cm^2, also expressed as 20 micropascals, or 20 μPa.

ear canal The external auditory meatus; the canal between the pinna and the eardrum.

eardrum The tympanic membrane located at the end of the ear canal that is attached to the ossicles of the middle ear.

early sound Direct and reflected components that arrive at the ear from a source during the first 50 msec or so. Such components are replicas of the original sound and arrive at different times producing comb-filter distortion. Also the basis of desirable effects such as spaciousness and defining of the stereo image.

echo A delayed return of sound that is perceived by the ear as a discrete repetition.

echogram A record of the very early reverberatory decay of sound in a room.

EES Early-early sound. Structureborne sound may reach the microphone in a room before the airborne sound because sound travels faster through denser materials.

EFC Energy-frequency curve.

EFTC Energy-frequency-time curve.

eigen function A defining constant in the wave equation.

ensemble Musicians must hear each other to function properly; in other words, ensemble must prevail. Diffusing elements surrounding the stage area contribute to ensemble.

equal loudness contour A contour representing a constant loudness for all audible frequencies. The contour having a sound-pressure level of 40 dB at 1,000 Hz is arbitrarily defined as the 40-phon contour.

equalization The process of adjusting the frequency response of a device or system to achieve a flat or other desired response.

equalizer A device for adjusting the frequency response of a device or system.

ETC Energy-time curve.

Eustachian tube The tube running from the middle ear into the pharynx that equalizes middle-ear and atmospheric pressure.

external meatus The ear canal terminated by the eardrum.

feedback, acoustical Unwanted interaction between the output and input of an acoustical system, for example, between the loudspeaker and the microphone of a system.

FFT Fast Fourier transform. An iterative program that computes the Fourier transform in a shorter time.

fidelity As applied to sound quality, the faithfulness of a signal to the original.

filter, band-pass A filter that passes all energy between a low-frequency and a high-frequency cutoff frequency.

Filter, FIR Digital filter of finite impulse response type.

filter, high-pass A filter that passes all energy above a cutoff frequency.

Filter, IIR Digital filter of infinite impulse response type.

filter, low-pass A filter that passes all energy below a cutoff frequency.

flanging The term applied to the use of comb filters to obtain special sound effects.

flanking sound Sound traveling by circuitous paths which reduces the effectiveness of a barrier.

floating floor A massive floor that is resiliently connected to the structure for the purpose of increasing transmission loss.

flutter echo A repetitive echo set up by parallel reflecting surfaces.

Fourier analysis Application of the Fourier transform to a signal to determine its spectrum.

fractal Numerous natural phenomena exhibit a macroscopic property or shape which is repeated microscopically at progressively smaller scales. These possess the property of self-similarity. Fractals applied to quadratic-residue diffusers result in extended range.

frequency The measure of the rapidity of alterations of a periodic signal, expressed in hertz.

frequency response The changes in the amplitude or sensitivity of a circuit or device with frequency.

FTC Frequency-time curve.

fundamental The basic pitch or frequency of a musical note or a harmonic series of frequencies.

fusion zone All reflections arriving at the observer's ear within 20 to 40 msec of the direct sound are integrated, or fused together, with a resulting apparent increase in level and a change in character. *See also* Haas effect.

gain The increase in power level of a signal produced by an amplifier.

graphic-level recorder A device for recording signal level in decibels versus time. The level in decibels versus angle can also be recorded for directivity patterns.

grating, diffraction An optical grating consists of minute, parallel lines used to break down light into its component optical colors. The principle is also used to achieve diffraction of acoustical waves.

grating, reflection phase An acoustical diffraction grating used to diffuse sound.

Haas effect Delayed sounds are integrated by the auditory apparatus if they fall on the ear within 20 to 40 msec of the direct sound. The level of the delayed components contributes to the apparent level of the sound, and sound is localized at the first-arriving source. Also called the precedence effect. *See also* fusion zone.

hair cell The sensory elements of the cochlea that transduce the mechanical vibrations of the basilar membrane to nerve impulses that are sent to the brain.

harmonic distortion *See* distortion, harmonic.

harmonics Integral multiples of the fundamental frequency. The first harmonic is the fundamental, and the second harmonic is twice the frequency of the fundamental, and so on.

hearing loss The loss of sensitivity of the auditory system, measured in decibels below a standard level. Some hearing loss is age related; some is caused by exposure to high-level sound.

Helmholtz resonator A reactive, tuned, sound absorber. A bottle is an example of such a resonator. It can employ a perforated cover or slats over a cavity.

henry The unit of inductance.

hertz The unit of frequency, abbreviated Hz. Cycles per second. The frequency of an electrical signal or sound wave.

high-pass filter *See* filter, high-pass.

HRTF Head-related transfer function. Describes how sound is changed by reflection and diffraction at the head, torso, and outer ear.

HVAC Heating, ventilating, and air-conditioning.

IEEE Institute of Electrical and Electronics Engineers.

image source A virtual reflected source located at an image point.

impact insulation class (IIC) A single value rating used to quantify impact noise in a floor/ceiling.

impedance The opposition to the flow of electric or acoustical energy measured in ohms.

impedance matching Maximum power is transferred from one circuit to another when the output impedance of the one is matched to the input impedance of the other. Maximum power transfer may be less important than low noise or voltage gain in many electronic circuits.

impulse A very short, transient, electrical or acoustical signal.

in phase Two periodic waves reaching peaks and going through zero at the same instant are said to be “in phase.”

inductance An electrical characteristic of circuits, especially of coils, that introduces inertial lag because of the presence of a magnetic field. Measured in henrys.

initial time-delay gap The time gap between the arrival of the direct sound and the first sound reflected from the surfaces of the room. *See also* arrival gap.

insulation As referred to sound barriers, insulation (isolation) refers to the sound transmission loss of a particular wall, floor/ceiling, and so on.

intensity Acoustical intensity is sound energy flux per unit area. The average rate of sound energy transmitted through a unit area normal to the direction of sound transmission.

interference The combining of two or more signals results in an interaction called interference. This may be constructive or destructive. Another use of the term refers to undesired, intrusive signals.

intermodulation distortion Distortion produced by the interaction of two or more signals. The distortion components are not harmonically related to the original signals.

inverse-distance law In a free field, the sound-pressure level decreases by 6 dB as the distance from the source is doubled.

inverse-square law In a free field, the intensity of a point-source sound is inversely proportional to the square of the distance from the source. *See also* spherical divergence.

isolation Refers to the isolation of an entire studio or room from outside noise.

ITDG Initial time-delay gap.

JAES *Journal of the Audio Engineering Society*.

JASA *Journal of the Acoustical Society of America*.

kHz 1,000 Hz.

law of the first wavefront The first wavefront falling on the ear determines the perceived direction of the sound.

LEDE *See* live end–dead end.

level A sound-pressure level in decibels means that it is calculated with respect to the standard reference level of 20 μPa. The term "level" associates that figure with the appropriate standard reference level.

LFD Low-frequency diffusion.

linear A device or circuit is linear if a signal passing through it is not distorted.

live end–dead end (LEDE) An acoustical treatment plan for rooms in which the front end of the room (with loudspeakers) is highly absorbent and the back end (with seating) is reflective and diffusive.

logarithm An exponent of 10 in the common logarithms to the base 10. For example, 10 to the exponent 2 is 100; the log of 100 is 2.

loudness A subjective term for the sensation of the magnitude of sound.

loudspeaker An electroacoustical transducer that changes electrical energy to acoustical energy.

masking The amount or the process by which the threshold of audibility for one sound (masked) is raised by the presence of another (masking) sound.

mass-air-mass resonance A resonating system composed, for example, of the mass of two spaced glass panes and the air between them. There is usually a dip in the transmission-loss curve at the frequency at which this system is resonant.

maximum length sequence *See* sequence, maximum length.

mean free path For sound waves in an enclosure, it is the average distance traveled between successive reflections.

microphone An acoustical-to-electrical transducer by which sound waves in air may be converted to electrical signals.

middle ear The cavity between the eardrum and the cochlea housing the ossicles connecting the eardrum to the oval window of the cochlea.

millisecond One-thousandth of a second, abbreviated ms or msec.

mixer A resistive device that is used to combine signals from multiple sources.

MLS (maximum length sequence) *See* sequence, maximum length.

modal resonance *See* mode.

mode A room resonance, a standing wave. Axial modes are associated with pairs of parallel walls. Tangential modes involve four room surfaces and oblique modes all six surfaces. The effect of modes is greatest at low frequencies and for small rooms.

monaural *See* monophonic.

monitor Loudspeaker used in the control room of a recording studio.

monophonic Single-channel sound.

multitrack A system of recording multiple tracks. The signals recorded on the various tracks are then mixed down to obtain the final recording.

NAB National Association of Broadcasters.

node Point of no vibration.

noise Unwanted interference that is electrical or acoustical in nature. Random noise is a desirable signal used in acoustical measurements.

noise criteria Standard spectrum curves by which a given measured noise may be described by a single NC or NCB number.

nonlinear A device or circuit is nonlinear if a signal passing through it is distorted.

normal mode A room resonance. *See also* mode.

null A low or minimum point on a graph. A minimum pressure region in a room.

oblique mode A room mode produced by reflections from all of the six surfaces of a rectangular room.

octave The interval between two frequencies having a ratio of 2:1.

oscilloscope An indicating instrument used to display waveforms, for example, on a time axis.

ossicles A linkage of three tiny bones providing the mechanical coupling between the eardrum and the oval window of the cochlea comprising the malleus, incus, and stapes (hammer, anvil, and stirrup).

out of phase The offset in time between two related signals.

oval window A tiny membranous window on the cochlea to which the foot plate of the stirrup ossicle is attached. The sound from the eardrum is transmitted to the fluid of the inner ear through the oval window.

overtone A component of a complex tone having a frequency higher than the fundamental frequency.

panel absorber A panel mounted with an enclosed airspace that vibrates and absorbs sound energy.

partial One of a group of frequencies, not necessarily harmonically related to the fundamental, which appear in a complex tone.

pascal A unit of pressure, a measure of perpendicular force per unit area, equivalent to 1 N/m^2.

passive absorber Any sound absorber that dissipates sound energy through frictional loss.

perforated absorber A panel absorber with perforated holes in the panel and an enclosed airspace functioning as a Helmholtz absorber.

PFC Phase-frequency curve.

phase The time relationship between two waveforms or signals.

phase shift The time or angular difference between two waveforms or signals.

phon The unit of loudness level of a tone, related to the ear's subjective impression of signal strength.

pink noise A noise signal whose spectrum level decreases at a 3-dB/octave rate. This gives the noise equal energy per octave. It is useful for sound analyzers with constant percentage bandwidths.

pinna The exterior ear.

pitch A subjective term for the perceived frequency of a tone.

place effect The theory that pitch perception is related to the pattern of excitation on the basilar membrane of the cochlea.

plenum A building cavity that is absorbent-lined to reduce noise, through which conditioned air is routed.

polar pattern A graph showing the 360° directional characteristics of a microphone or loudspeaker.

polarity The relative position of the high (+) and the low (–) signal leads in an audio system.

PRD Primitive root diffuser.

preamplifier An amplifier designed to optimize the amplification of weak signals, such as from a microphone.

precedence effect *See* Haas effect.

pressure zone As sound waves strike a solid surface, the particle velocity is zero at the surface and the pressure is high, thus creating a high-pressure layer near the surface.

primitive root sequence *See* sequence, primitive root.

psychoacoustics The study of the ear-brain auditory system and its perception and reaction to auditory stimuli.

pure tone A tone with no harmonics, a sine wave. All energy is concentrated at a single frequency.

Q-factor Quality factor. A measure of the losses in a resonance system. The sharper the tuning curve, the higher the Q-factor.

QRD Quadratic residue diffuser.

quadratic residue sequence *See* sequence, quadratic residue.

random noise A noise signal, commonly used in measurements, which has constantly shifting amplitude, phase, and frequency and a uniform spectral distribution of energy.

ray At higher audio frequencies, sound may be considered to travel in straight lines as a beam in a direction normal to the wavefront.

reactance The opposition to the flow of electricity posed by capacitors and inductors.

reactive absorber A sound absorber, such as the Helmholtz resonator, which utilizes the effects of mass and compliance as well as resistance.

reactive silencer A silencer in air-conditioning systems that uses reflection losses to provide attenuation.

reflection For surfaces large compared to the wavelength of impinging sound, sound is reflected much as light is reflected, with the angle of incidence equaling the angle of reflection.

reflection-free zone A room design with reflecting surfaces to provide an anechoic path between the monitor loudspeakers and the listening position.

reflection-phase grating A diffuser of sound energy using the principle of the diffraction grating.

refraction The bending of sound waves traveling through layered media with different sound velocities.

resistance That quality of electrical or acoustical circuits that results in dissipation of energy through heat.

resonance A natural periodicity, or the reinforcement associated with this periodicity. A resonant system vibrates at maximum amplitude when tuned to its natural frequency.

resonator silencer A silencer in air-conditioning systems that uses tuned stubs and their resonating effect to provide attenuation.

response *See* frequency response.

reverberation The decay of sound in an enclosure after the source has stopped. Caused by multiple reflections from the room boundaries.

reverberation chamber A room with reflective boundaries used for measuring sound absorption coefficients.

reverberation time The time required for the sound in an enclosure to decrease a certain amount, usually by 60 dB from an initial steady-state level. The latter is abbreviated as RT_{60}.

RFZ *See* reflection-free zone.

ringing High-Q electrical circuits and acoustical devices have a tendency to oscillate (or ring) when excited by a suddenly applied signal.

room mode The normal modes of vibration of an enclosed space. *See also* mode.

round window The tiny membrane of the cochlea that opens into the middle ear that serves as a pressure release for the cochlear fluid.

RPG Reflection-phase grating. A diffuser of sound energy using the principle of the diffraction grating.

RPG cloud Reflection-phase gratings arranged for overhead mounting.

RT_{60} Time for sound to decay by 60 dB. It can be measured at different frequencies, but 500 Hz is often used as a default. *See also* reverberation time.

sabin The unit of sound absorption. One square foot of open window (100% absorption) has an absorption of 1 sabin. One square meter of open window has an absorption of 1 metric sabin.

Sabine Wallace Clement Sabine, originator of the Sabine reverberation formula.

SBIR *See* speaker-boundary interference response.

Schroeder plot A reverberation decay computed by the mathematical process defined by Manfred Schroeder.

semicircular canals The three sensory organs for balance that are a part of the cochlear structure.

sequence, maximum length A measurement method using a pseudorandom binary sequence to excite a system and/or room with a test signal resembling wideband white noise. Also a mathematical sequence used in determining the well depth of diffusers.

sequence, primitive root A mathematical sequence used in determining the well depth of diffusers.

sequence, quadratic residue A mathematical sequence used in determining the well depth of diffusers.

signal-to-noise ratio The difference measured in decibels between the nominal or maximum operating level and the noise floor. Abbreviated S/N.

sine wave A pure wave, periodic wave related to simple harmonic motion.

slap back A discrete reflection from a nearby surface.

sone The unit of measurement for subjective loudness.

sound absorption coefficient The practical unit between 0 and 1 (nominally) expressing the absorbing efficiency of a material. Coefficient values are determined experimentally.

sound-level meter A microphone-amplifier-meter device calibrated to read sound-pressure level above the reference level of 20 μPa.

sound-power level A power expressed in decibels above the standard reference level of 1 pW.

sound-pressure level A sound pressure expressed in dB above the standard sound pressure of 20 μPa.

sound spectrograph An instrument that displays the time, level, and frequency of a signal.

speaker-boundary interference response Acoustical distortion occurring when loudspeaker reflections combine with the direct sound, either enhancing or canceling it in varying degrees.

spectrum The distribution of the energy of a signal with respect to frequency.

spectrum analyzer An instrument for measuring and/or recording the spectrum of a signal.

specular reflection A mirror-like reflection of sound (or light) from a surface with an incoming direction and outgoing direction and an equal angle between them.

specularity A term expressing the efficiency of diffraction-grating types of diffusers.

spherical divergence Sound diverges spherically from a point source in free space.

splaying Walls are splayed when they are constructed “off square,” that is, a few degrees from the normal rectilinear form.

standing wave A resonance condition in an enclosed space in which sound waves traveling in one direction interact with those traveling in the opposite direction, resulting in a stable condition. *See also* mode.

STC Sound transmission class. A single-number system of designating sound transmission loss of partitions.

steady state A continuous condition devoid of transient effects.

stereo A stereophonic system of two channels.

subwoofer A low-frequency loudspeaker typically used in conjunction with other high-frequency satellite loudspeakers.

superposition Many sound waves may traverse the same point in space, the air molecules responding to the vector sum of the different waves.

T_{60} *See* RT_{60}.

tangential mode A room mode produced by reflections from four of the six surfaces of a rectangular room.

TDS *See* time-delay spectrometry.

TEF *See* time-energy-frequency.

threshold of feeling (pain) The sound-pressure level that produces a perceived physical sensation in the ears at approximately 120 dB above the threshold of hearing.

threshold of hearing The lowest level sound that can be perceived by the human auditory system. This is close to the standard reference level of sound pressure, 20 μPa.

timbre The quality of a sound related to its harmonic structure.

time-delay spectrometry (TDS) A measurement method for obtaining anechoic results in echoic spaces.

time-energy-frequency (TEF) An analysis method employing time-delay spectrometry.

tone A tone results in an auditory sensation of pitch.

tone burst A short signal used in acoustical measurements making it possible to differentiate desired signals from spurious reflections.

tone control An electrical circuit to allow adjustment of frequency response.

transducer A device for changing electrical signals to acoustical or vice versa, such as a microphone or loudspeaker.

transient A short-lived aspect of a signal, such as the attack of a percussive instrument.

treble The higher frequencies of the audible spectrum.

volume The colloquial equivalent of sound level.

watt The unit of electrical or acoustical power.

wave A regular variation of an electrical signal or acoustical pressure.

wavelength The distance a sound wave travels in the time it takes to complete one cycle.

weighting Adjustment of sound-level meter response to achieve a desired measurement.

white noise Random noise having uniform distribution of energy with respect to frequency.

woofer A low-frequency loudspeaker that is part of the main loudspeaker.

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