11 Acoustics of Enclosed Spaces: Architectural Acoustics

11.1 Introduction

Although people have gathered in large auditoriums and places of worship since the advent of civilization, architectural acoustics did not exist on a scientific basis until a young professor of physics at Harvard University accepted an assignment from Harvard's Board of Overseers in 1895 to correct the abominable acoustics of the newly constructed Fogg Lecture Hall. Through careful (but by present-day standards, rather crude) measurements with the use of a Gemshorn organ pipe of 512 Hz, a stopwatch, and the aid of a few able-bodied assistants who lugged absorbent materials in and out of the lecture hall, Wallace Sabine established that the reverberation characteristics of a room determined the acoustical nature of that room and that a relationship exists between quality of the acoustics, the size of the chamber and the amount of absorption surfaces present. He defined a reverberation time T as the number of seconds required for the intensity of the sound to drop from a level of audibility 60 dB above the threshold of hearing to the threshold of inaudibility. To this day reverberation time still constitutes the most important parameter for gauging the acoustical quality of a room. The original Sabine equation

$$T = \frac{0.049V}{\sum_i S_i\,\alpha_i}$$

is deceptively simple, as effects such as interference or diffraction and behavior of sound waves as affected by the shape of the room, presence of standing waves, normal modes of vibration, are not embodied in that equation. Here V is the room volume in cubic feet, S_i the component surface area and α_i the corresponding absorption coefficient. On the basis of his measurements Sabine was able to cut down the reverberation time of the lecture hall from 5.6 s through the strategic deployment of absorbing materials throughout the room. This accomplishment firmly established Sabine's reputation, and he became the acoustical consultant for Boston Symphony Hall, the first auditorium to be designed on the basis of quantitative acoustics.

In this chapter we shall examine the behavior of sound in enclosed spaces, and develop the fundamental equations that are used in optimizing the acoustics of auditoriums, music halls, and lecture rooms. We shall also study the means of improving room acoustics through installation of appropriate materials. This chapter concludes with descriptions of a number of outstanding acoustical facilities.

11.2 Sound Fields

The distribution of acoustic energy, whether originating from a single or multiple sound sources in an enclosure, depends on the room size and geometry and on the combined effects of reflection, diffraction, and absorption. With the appreciable diffusion of sound waves due to all of these effects it is no longer germane to consider individual wave fronts, but to refer to a *sound field*, which is simply the region surrounding the source. A *free field* is a region surrounding the source, where the sound pattern emulates that of an open space. From a point source the sound waves will be spherical, and the intensity will approximate the inverse square law. Neither reflection nor diffraction occurs to interfere with the waves emanating from the source. Because of the interaction of sound with the room boundaries and with objects within the room, the free field will be of very limited extent.

If one is close to a sound source in a large room having considerably absorbent surfaces, the sound energy will be detected predominantly from the sound source and not from the multiple reflections from surroundings. A free field can be simulated throughout an entire enclosure if all of the surrounding surfaces are lined with almost totally absorbent materials. An example of such an effort to simulate a free field is the extremely large anechoic (echoless) chamber at Lucent Technologies Bell Laboratories in Murray Hill, New Jersey, shown in the photograph of Figure 11.1. Such a chamber is typically lined with long wedges of absorbent foam or fiberglass and the "floor" consists of either wire mesh or grating suspended over wedges installed over (and covering entirely) the "real" floor underneath. Precisely controlled experiments on sound sources and directivity patterns of sound propagation are rendered possible in this sort of chamber.

A *diffuse field* is said to occur when a large number of reflected or diffracted waves combine to render the sound energy uniform throughout the region under consideration. Figure 11.2 illustrates how diffusion results from multiple reflections. The degree of diffusivity will be increased if the room surfaces are not parallel so there is no preferred direction for sound propagation. Concave surfaces with radii of curvature comparable to sound wavelengths tend to cause focusing, but convex surfaces will promote diffusion. Multiple speakers in amplifying systems auditoriums are used to achieved better diffusion, and special baffles may be hung from ceilings to deflect sound in the appropriate directions.

Sound reflected from walls generates a *reverberant field* that is time dependent. When the source suddenly ceases, a sound field persists for a finite interval as the result of multiple reflections and the low velocity of sound propagation. This residual acoustic energy constitutes the reverberant field. The sound that reaches a

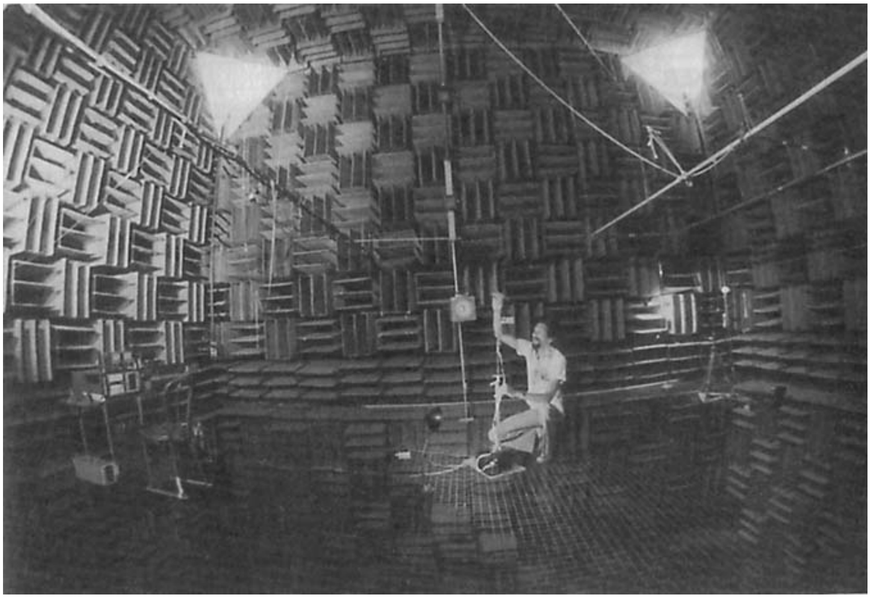

FIGURE 11.1. Photograph of the large anechoic chamber at the Lucent Bell Laboratory in Murray Hill, NJ. Dr. James E. West, a former president of the Acoustical Society of America, is shown setting up test equipment. (Courtesy of Lucent Technologies.)

listener in a fairly typical auditorium can be classified into two broad categories: the direct (free field) sound and indirect (reverberant) sound. As shown in Figure 11.3, the listener receives the primary or direct sound waves and indirect or reverberant sound. The amount of acoustic energy reaching the listener's ear by any single reflected path will be less than that of the direct sound because the reflected path is longer than the direct source–listener distance, which results in greater divergence; and all reflected sound undergo an energy decrease due to the absorption of even the most ideal reflectors. But indirect sound that a listener hears comes from a great number of reflection paths. Consequently, the contribution of reflected sound to the total intensity at the listener's ear can exceed the contribution of direct sound particularly if the room surfaces are highly reflective.

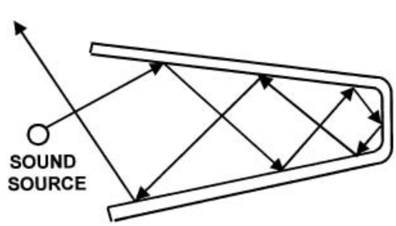

FIGURE 11.2. Sound diffusion resulting from multiple reflections.

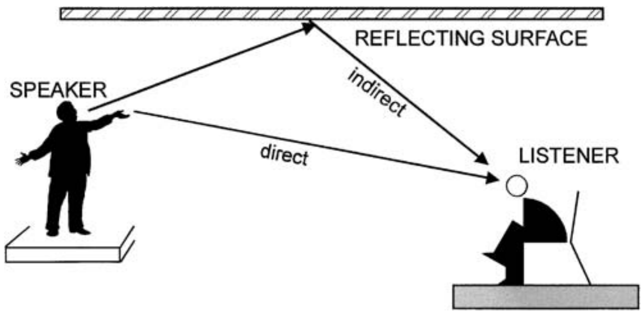

FIGURE 11.3. Reception of direct and indirect sound.

The phases and the amplitudes of the reflected waves are randomly distributed to the degree that cancellation from destructive interference is fairly negligible. If a sound source is operated continuously the acoustic intensity builds up in time until a maximum is reached. If the room is totally absorbent so that there are no reflections, the room operates as an anechoic chamber, which simulates a free field condition. With partial reflection, however, the source continues to add acoustic energy to the room, that is partially absorbed by the enclosing surfaces (i.e., the walls, ceiling, floor and furnishings) and deflected back into the room. For a source operating in a reverberation chamber the gain in intensity can be considerable—as much as ten times the initial level. The gain in intensity is approximately proportional to the reverberation time; thus it can be desirable to have a long reverberation time to render a weak sound more audible.

11.3 Reverberation Effects

Consider a sound source that operates continuously until the maximum acoustic intensity in the enclosed space is reached. The source suddenly shuts off. The reception of sound from the direct ray path ceases after a time interval r/c, where r represents the distance between the source and the reception point and c the sound propagation velocity. But owing to the longer distances traveled, reflected waves continue to be heard as a reverberation which exists as a succession of randomly scattered waves of gradually decreasing intensity.

The presence of reverberation tends to mask the immediate perception of newly arrived direct sound unless the reverberation drops 5–10 dB below its initial level in a sufficiently short time. Reverberation time T, the time in seconds required for intensity to drop 60 dB, offers a direct measure of the persistence of the reverberation. A short reverberation time is obviously necessary to minimize the masking effects of echoes so that speech can be readily understood. However, an extremely short reverberation time tends to make music sound harsher—or less "musical"—while excessive values of reverberation time T can blur the distinction between

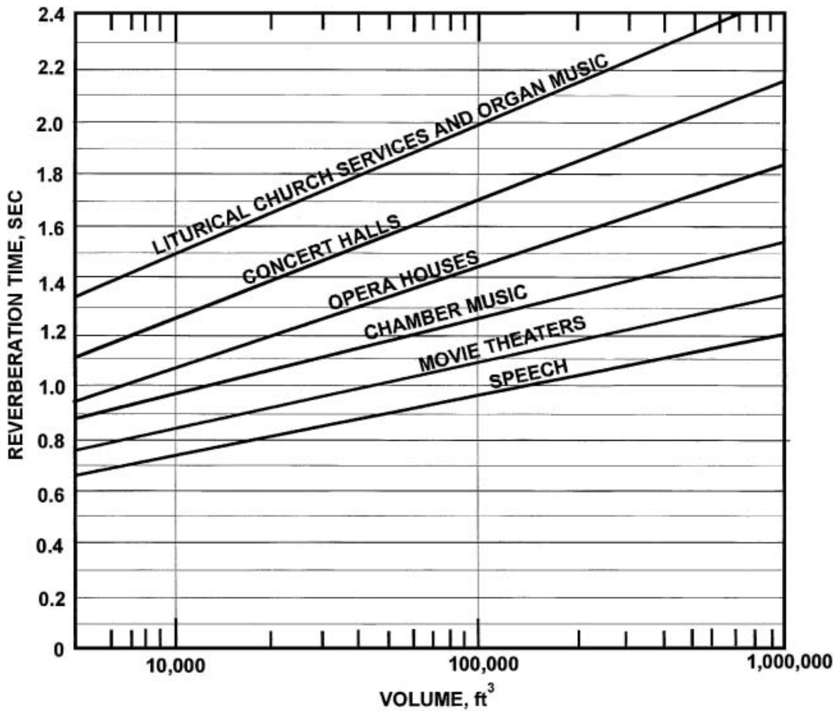

FIGURE 11.4. Typical reverberation times for various auditoriums and functions.

individual notes. The choice of T, which also depends on the room volume, therefore represents an optimization between two extremes.

Figure 11.4 represents the accumulation of optimal reverberation time data as functions of intended use and enclosure volume. Lower values of T occur from increased absorption of sound in the surfaces of the enclosures. Hard surfaces such as ceramic tile floors and mirrors tend to lengthen the reverberation time. In addition to reverberation time, the ability of a chamber or enclosure to screen out external sound minimizes annoyance or masking effects. The acoustic transmission of walls, treated in Chapter 12, constitutes a major factor in enclosure design. A short reverberation time with its attendant high absorption tends to lessen the ambient noise level generated by external sounds that penetrate the walls of the enclosure.

11.4 Sound Intensity Growth in a Live Room

We now apply the classic ray theory to deal with a sound source operating continuously in an enclosure, which will yield results in fairly good agreement with

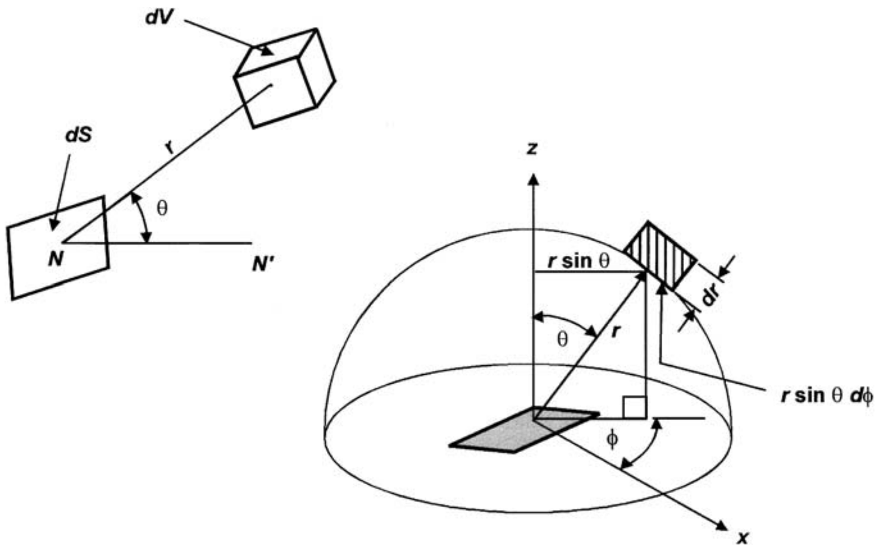

FIGURE 11.5. Geometric configuration for setting up the relationship between energy density and intensity of sound.

experimental measurements. The process of absorption in the medium or the enclosing surfaces prevents the intensity from becoming infinitely large. Absorption in the medium is fairly negligible in medium- and small-sized enclosures, so the ultimate intensity depends upon the absorption power of the boundary surfaces. If the enclosure's boundary surfaces have high absorption the intensity will quickly achieve the maximum which exceeds only slightly the intensity of the direct ray. If the enclosure has highly reflective surfaces, i.e., low absorption, a "live" room ensues; the growth of the intensity will be slow and appreciable time will have elapsed for the intensity to reach its maximum.

After a sound source is started in a live room, reflections from the wall become more uniform in time as the sound intensity increases. With the exception of close proximity to the source, the energy distribution can be considered uniform and random in direction. In reality a signal source having a single frequency will result in standing-wave patterns, with resultant large fluctuations from point to point in the room. But if the sound consists of a uniform band of frequencies or a pure tone warbling over at least a half octave, the interference effects of standing waves are obliterated.

Referring to Figure 11.5, we establish the relationship between *intensity* (which represents the energy flow) and energy density of randomly distributed *acoustic energy*. In the figure dS represents an element of the wall surface and dV the volume element in the medium at a distance r from dS. The distance r makes an angle θ with the normal NN' to dS. Let the average acoustic energy density E (in W/m^3) be assumed uniform throughout the region under consideration. The acoustic energy in incremental volume dV is $E\ dV$. The surface area of the sphere of radius r encompassing dV is $4\pi r^2$. The projected area of dS on the sphere is $\cos\theta\ dS$. The portion of the total energy contained in dV is given by the ratio $dS \cos\theta/4\pi r^2$. The energy from dV that strikes dS directly becomes

$$dE = \frac{E\ dV\, dS \cos\theta}{4\pi\ r^2}. \tag{11.1}$$

Now consider the volume element dV as being part of a hemisphere shall of radius $r$and thickness dr. The acoustic energy rendered to S by the complete shell is found by assuming a circular zone of radius $r \sin\theta$ (with θ treated as a constant) in Figure 11.5 and integrating over the entire surface of the shell. The volume of the resultant element is $2\pi r \sin\theta\ r\ dr\ d\theta$. From $\theta = 0$ to $\theta = \pi/2$, and Equation (11.1) yields

$$\Delta E = \frac{E\ dS\ dV}{2} \int_0^{\pi/2} \sin\theta \cos\theta\ d\theta = \frac{E\ dS\ dr}{4}$$

This energy arrives during time interval $t = dr/c$. Hence, the rate of acoustic energy impinging dS from all directions is

$$\frac{\Delta E}{t} = \frac{Ec\ dS}{4}$$

or $Ec/4$ per unit area, which is therefore the intensity I of the diffused sound at the walls. This is also equal to one fourth of a plane wave of energy intensity I incident at a normal angle onto a plane. The intensity $I$of the diffuse sound at the wall becomes

$$I = \frac{Ec}{4} \tag{11.2}$$

11.5 Sound Absorption Coefficients

All materials constituting the boundaries of an enclosure will absorb and reflect sound. A fraction α of the incident energy is absorbed and the balance $(1 - \alpha)$ is reflected. Reflection is indicated by the reflection coefficient r defined as

$$r = \frac{\text{amplitude of reflected wave}}{\text{amplitude of incident wave}}$$

Because the energy in a sound wave is proportional to the square of the amplitude, the sound absorption coefficient α and the reflection coefficient are related by

$$\alpha = 1 - r^2$$

The value of the sound absorption coefficient α will vary with the frequency of the incident ray and the angle of incidence. Materials comprising room surfaces are subject to sound waves that impinge upon them from many different angles as are-sult of multiple reflections. Hence, published data for absorption coefficients

Table 11.1. Absorption Coefficients.

	Octave-Band Center Frequency (Hz)					
	125	250	500	1000	2000	4000
Brick, unglazed	0.03	0.03	0.03	0.04	0.05	0.07
Brick, unglazed, painted	0.01	0.01	0.02	0.02	0.02	0.03
Carpet on foam rubber	0.08	0.24	0.57	0.69	0.71	0.73
Carpet on concrete	0.02	0.06	0.14	0.37	0.60	0.65
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
Floors, concrete or terrazzo	0.01	0.01	0.015	0.02	0.02	0.02
Floors, resilient flooring on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Floors, hardwood	0.15	0.11	0.10	0.07	0.06	0.07
Glass, heavy plate	0.18	0.06	0.04	0.03	0.02	0.02
Glass, standard window	0.35	0.25	0.18	0.12	0.07	0.04
Gypsum, board 0.5 in.	0.29	0.10	0.05	0.04	0.07	0.09
Panels, fiberglass, 1.5 in. thick	0.86	0.91	0.80	0.89	0.62	0.47
Panels, perforated metal, 4 in. thick	0.70	0.99	0.99	0.99	0.94	0.83
Panels, perforated metal with fiberglass insulation, 2 in. thick	0.21	0.87	1.52	1.37	1.34	1.22
Panels, perforated metal with mineral fiber insulation, 4 in. thick	0.89	1.20	1.16	1.09	1.01	1.03
Panels, plywood, 3/8 in.	0.28	0.22	0.17	0.09	0.10	0.11
Plaster, gypsum or lime, rough finish on lath	0.02	0.03	0.04	0.05	0.04	0.03
Plaster, gypsum or lime, smooth finish on lath	0.02	0.02	0.03	0.04	0.04	0.03
Polyurethane foam, 1 in. thick	0.16	0.25	0.45	0.84	0.97	0.87
Tile, ceiling, mineral fiber	0.18	0.45	0.81	0.97	0.93	0.82
Tile, marble or glazed	0.01	0.01	0.01	0.01	0.02	0.02
Wood, solid, 2 in. thick	0.01	0.05	0.05	0.04	0.04	0.04
Water surface	nil	nil	nil	0.003	0.007	0.02
One person	0.18	0.4	0.46	0.46	0.51	0.46
Air	nil	nil	nil	0.003	0.007	0.03

Note: The coefficient of absorption for one person is that for a seated person (m^2 basis). Air absorption is on a per cubic meter basis.

are for "random" incidence as distinguished from "normal" or "perpendicular" incidence.

The angle–absorption correlation appears to be of somewhat erratic nature, but at high frequencies the absorption coefficients in some materials is roughly constant at all angles. At low frequencies the random-incidence absorption tends to be greater than for normal incidence. However, as Table 11.1 shows, α varies considerably with frequency for many materials, and the absorption coefficients are generally measured at six standard frequencies: 125, 250, 500, 1000, 2000, and 4000 Hz. Absorption occurs as the result of incident sound penetrating and becoming entrapped in the absorbing material, thereby losing its vibrational energy that converts into heat through friction. Ordinarily the values of α should fall between zero for a perfect reflector and unity for a perfect absorber. Measurements of $\alpha > 1.0$ have been reported, owing possibly to diffraction at low frequencies and other testing condition irregularities.

Let $\alpha_1, \alpha_2, \alpha_3, \ldots \alpha_i$ denote the absorption coefficient of different materials of corresponding areas S_1, S_2, S_3, ... S_i forming the interior boundary planes (viz. the walls, ceiling and floor) of the room as well as any other absorbing surfaces (e.g. furniture, draperies, people, etc.). The average absorption coefficient α for an enclosure is defined by

$$\overline{\alpha} = \frac{\alpha_1\, S_1 + \alpha_2\, S_2 + \alpha_3\, S_3 + \cdots + \alpha_i\, S_i}{S_1 + S_2 + S_3 + \cdots + S_i} = \frac{A}{S} \tag{11.3}$$

where A represents the total absorptive area $\sum \alpha_i S_i$, and S the total spatial area.

11.6 Growth of Sound with Absorbent Effects

The rate W of sound energy being produced equals the rate of sound energy absorption at the boundary surfaces of the room plus the rate at which the energy increases in the medium throughout the room. This may be expressed as a differential equation governing the growth of acoustic energy in a live room:

$$V\frac{dE}{dt} + \frac{AcE}{4} \equiv W \tag{11.4}$$

The solution for E in Equation (11.4) is

$$E = \frac{4W}{Ac}\left(1 - e^{-(Ac/4V)t}\right) \tag{11.5}$$

with the initial condition that the sound source begins operating at $t = 0$. From the relationship of Equation (11.2) the intensity becomes

$$I = \frac{W}{A}\left(1 - e^{-(Ac/4V)t}\right) \tag{11.6}$$

and from Equation (3.58) the energy density is

$$E = \frac{p^2}{2\,\rho_0\, c^2} \tag{11.7}$$

The mean square acoustic pressure becomes

$$p^2 = \frac{4W\,\rho_0\, c}{A}\left(1 - e^{-(Ac/4V)t}\right) \tag{11.8}$$

Equation (11.8) is analogous to the one describing the growth of direct current in an electric circuit containing an inductance and a resistance. The time constant of the acoustic process is $4V/Ac$. If the total absorption is small and the time constant is large, a longer time will be necessary for the intensity to approach its ultimate value of $I_\infty = W/A$. The ultimate values of the energy density and mean square acoustic pressure are given by

$$E_\infty = \frac{4W}{Ac}, \qquad p_\infty^2 = \frac{4W\rho_0 c}{A}$$

A number of caveats pertain to the use of Equation (11.8). In order that the assumption of an even distribution of acoustic energy be cogent, a sufficient time t must have elapsed for the initial rays to undergo several reflections at the boundaries. This means approximately 1/20 of a second should have elapsed in a small chamber; and the time must approach nearly a full second for a large auditorium. The final energy density, being independent of the size and shape of the room, should be the same at all points of the room and dependent only upon the total absorption A. But Equation (11.6) does not hold for spherical or curved rooms which can focus sounds; neither is Equation (11.8) applicable to rooms having deep recesses nor to oddly shaped rooms or rooms coupled together by an opening, and nor to rooms with some surfaces of extraordinarily high absorption coefficients α (these cause localized lesser values of energy densities).

11.7 Decay of Sound

We can now develop the differential equation describing the decay of uniformly diffuse sound in a live room. The sound source is shut off at time $t = 0$, meaning $W = 0$ at that instant. E_0 denotes the uniformly distributed energy density at that instant. From Equation (11.4)

$$\frac{AcE}{4V}dt = dE \tag{11.9}$$

and the solution to Equation (11.9) becomes

$$E = E_0\, e^{-(Ac/4V)t} \tag{11.10}$$

The intensity I at any time t after the cessation of the sound source is related to the initial intensity I_0 by

$$\frac{I}{I_0} = e^{-(Ac/4V)} \tag{11.11}$$

Applying the operator 10 log to both sides of Equation (11.11) results in

$$\Delta IL = 10 \log e^{-(Ac/4V)t} = \frac{10}{2.3} \ln e^{-(Ac/4V)t} = -\frac{1.087Act}{V} \tag{11.12}$$

where ΔIL denotes the intensity level change in decibels. The intensity level in a live room decreases with elapsed time at a constant decay rate D (in dB/s),

$$D = \frac{1.087Ac}{V}.$$

Following Sabine's definition, we define the reverberation time T as the time required for the sound level in the room to decay by 60 dB:

$$T = \frac{60}{D} = \frac{55.2V}{Ac} \tag{11.13}$$

Expressing volume V in m^3 and area S used to compute A in m^2, and setting sound propagation speed $c = 343$ m/s, Equation (11–13) becomes

$$T = \frac{0.161V}{A} \tag{11.14}$$

Equation (11.14) becomes for English units

$$T = \frac{0.049V}{A} \tag{11.15}$$

where volume V is rendered in ft^3 and A in ft^2 (or *sabins*, with 1 sabin equal to 1 ft^2 of absorption area αS). (One *metric sabin* is equal to 1 m^2 of absorption area.) It becomes apparent here that the reverberation time for a room can be controlled by selecting materials with the appropriate acoustic absorption coefficients. The absorption coefficient of a material can be measured by the introduction of a definite area of the absorbent material in a specially constructed live room or *reverberation* (or *echo*) *chamber*. A photograph of such a chamber is given in Figure 11.6.

FIGURE 11.6. Photograph of a reverberation chamber. (Courtesy of Eckel Industries, Inc.)

Example Problem 1: Reverberation Prediction

A room 8 m long, 4 m wide, and 2.8 m high contains four walls faced with gypsum boards. The only exceptions to the wall area are a glass window 1 m by 0.5 m and a plywood-paneled door 2.2 m by 0.6 m. In addition the door has a gap underneath, 1.5 cm high. In order to estimate the reverberation time of the room at 500 Hz we make use of the data in Table 11.1. Predict the reverberation time T.

Solution

The absorption area (in m^2) is found as follows:

$$\begin{aligned} A = \Sigma\, S_i\, \alpha_i &= [2(8 \times 2.8) + 2(4 \times 2.8) - 2.2(0.6) - (0.015)(0.6) - 1.0(0.5)] \\ &\quad \times 0.05 + (2.2)(0.6)(0.17) + (0.015)(0.6)(1) + 1.0(0.5)(0.18) \\ &\quad + (4)((8)(0.81) + (4)(8)(0.81) = 32.70 \text{ m}^2. \end{aligned}$$

Applying Equation (11.14):

$$T = \frac{0.161 \times 8 \times 4 \times 2.8}{32.7} = 0.82 \text{ s}.$$

The gap at the bottom of the door is treated as a complete sound absorber with a coefficient of unity. From the above estimated value of the reverberation time of 0.44 s and a chamber volume of 89.6 m^3, the room may be suitable for use as a classroom according to Figure 11.4.

11.8 Decay of Sound in Dead Rooms

The derivation of Equation (11.14) was based on the assumption that a sufficient number of reflections occur during the growth or decay of sound and also that the energy of the direct sound and the energy of the fractional amount of sound reflected were both sufficient to ensure a uniform energy distribution. In the case of anechoic chambers, where the absorption coefficient of the materials constituting the boundaries is very close to unity, it is apparent that the derivations of the preceding equations for growth and decay of sound are not applicable. The only energy present is the direct wave emanating from the sound source. The reverberation time must be zero, whereas application of Equation (11.14) would yield a finite reverberation time of $0.161V/S$, where S is simply the total area of the interior surfaces of the chamber. Thus, it is apparent that Equation (11.14) would be increasingly in error as the average sound absorption coefficient increases. If the average value of the absorption coefficient exceeds 0.2, Equation (11.14) will be in error by approximately 10%.

A different approach to ascertaining the decay of sound in a dead room, which was developed by Eyring (1930), is to consider the multiplicity of reflections as a set of image sources, all of which are considered to exist as soon as the real source begins. Let $\bar{\alpha}$, found from the relationship $\bar{\alpha} = (\sum \alpha_i S_i)/\sum S_i$ denote the average sound absorption coefficient of the room's boundary materials. The growth of acoustic energy at any point in the room results from the accumulation of successive increments from the sound source, from the first-order (single reflection) images with strengths $W(1 - \bar{\alpha})$, from the second-order (secondary reflection) images with strengths $W(1 - \bar{\alpha})^2$, and so on, until all the image sources of appreciable strengths have rendered their contributions. When the true sound source is stopped, the decay of the sound occurs with all the image sources stopped simultaneously along with the source. The energy decay in the room occurs from successive losses of acoustic radiation from the source, then from the first-order images, the second-order images, and so on.

Eyring derived the following equation for the growth in acoustic energy density:

$$E = -\frac{4W}{cS\ln(1-\overline{\alpha})}\left[1 - e^{cS\ln(1-\overline{\alpha})t/4V}\right] \tag{11.16}$$

The above equation is very similar to Equation (11.5) excepting that the total room absorption is given by

$$\alpha = -S\ln(1-\overline{\alpha}) \tag{11.17}$$

Here S is the total area of the boundary surfaces of the room. In a like fashion the analogy to Equation (11.6) for the decay of sound energy is given by

$$E = E_0\, e^{cS\ln(1-\overline{\alpha})t/4V}$$

and the decay rate in dB/s is expressed as

$$D = -\frac{1.08cS\ln(1-\overline{\alpha})}{V}$$

with the reverberation time expressed by

$$T = \frac{0.161V}{-S\ln(1-\overline{\alpha})}$$

For small values of absorption ($\alpha \ll 1$) the term $\ln(1 - \alpha)$ may be replaced by α, the first term in an infinite series. This results in recovering the Sabine formula for live rooms. It should also be noted that the coefficient 0.161 for the Sabine and the Eyring formulas, which is based on the speed of sound at 24°C, will vary according to air temperature. The coefficient becomes somewhat higher at lower air temperatures and vice versa.

Another formula for determining the reverberation time of a room lined with materials of widely ranging absorption coefficients was developed by Millington and Sette (Millington, 1932; Sette, 1993). The Millington–Sette theory indicates that the total room absorption is given by

$$A = \sum -S_i\ln(1-\bar{\alpha}_i)$$

which yields the reverberation time

$$T = \frac{0.161V}{\sum - S_i\ln(1-\bar{\alpha}_i)}$$

11.9 Reverberation as Affected by Sound Absorption and Humidity in Air

We have previously not considered the effect of absorption of sound and humidity in air on reverberation times. The volume of air contained in very large auditoriums or a place of worship can absorb an amount of acoustic energy that cannot be neglected as in the case with smaller rooms. If a room is small, the number of reflections from the boundaries is large and the amount of time the sound wave spends in the room is correspondingly small. In this situation acoustic energy absorption in the air is generally not important. In very large room volumes the time a wave spends in the air between reflections becomes greater to the extent that absorption of energy in air no longer becomes negligible. The reverberation equations must now include the effect of air absorption, particularly at higher frequencies (>1 kHz).

Sound waves lose some energy through viscous effects during the course of their propagation through a fluid medium. The intensity of a plane wave lessens with distance according to the equation

$$I = I_0 e^{-2\beta x} = I_0 e^{-mx}.$$

Here $m = 2\beta$ represents the attenuation coefficient of the medium. Some texts use α rather than β to denote the attenuation constant of the medium; we eschew its use in order to avoid confusion with α used in this chapter to denote the absorption coefficient of a surface. During time interval t, a sound wave travels a distance $x = ct$, and the preceding equation may be revised to read

$$I = I_0\, e^{(\beta/4V+m)ct}$$

The expression for the reverberation time becomes

$$T = \frac{0.161V}{A + 4mV} \tag{11.18}$$

where the constant $m$is expressed in units of m^{-1}. The total surface absorption A is given either by Equation (11.3) or (11.17) depending whether that room fits into the category of being an acoustically live or dead chamber. As the room volume V becomes larger, the second term in the denominator of Equation (11.18) increases in magnitude, as air absorption becomes more significant, due to increasing path lengths between the walls. Since m also increases with frequency, air absorption also becomes more manifest at higher frequencies (above 1 kHz) than at lower frequencies. The values of m are given in Figure 11.7[1] as a function of humidity for various frequencies at a normal room temperature of 20°C. More details, also given in tabular form, for a range of air temperatures and humidities are given in the NASA report (1967), prepared by Cyril M. Harris, listed at the end of this chapter. It is seen from Figure 11.7 that the effect of humidity reaches a maximum

[1] The plot of Figure 11.7 applies to *indoor* sound propagation, not to outdoor propagation that includes meteorological effects not present indoors.

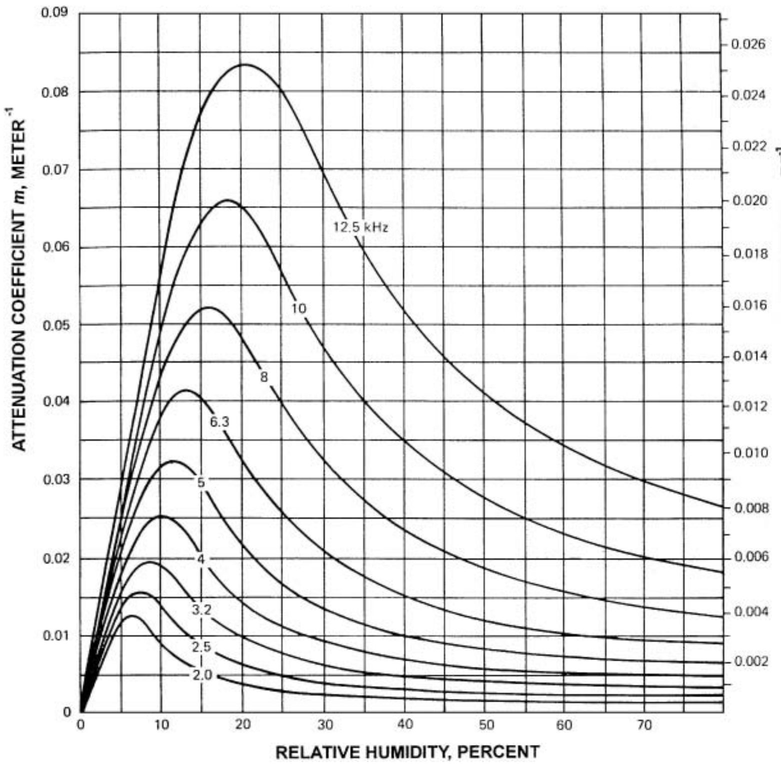

FIGURE 11.7. Values of the total attenuation coefficient m versus percent relative humidity for air at 20°C and normal atmospheric pressure for frequencies between 2 kHz and 12.5 kHz. Values are rendered in both SI units and U.S. Customary System units. (After Harris, 1966, 1967.)

in each of the given frequencies in the 5–25% relative humidity range and then trails off at higher humidities.

Example Problem 2

Find the air absorption at a frequency of 6300 Hz and 25% relative humidity for a room volume of 20,000 m^3.

Solution and Brief Discussion

From Figure 11.7 the value of m is equal to 0.026 m^{-1}. The air absorption $A_{air} = 4mV$ in Equation (11.18) is equal to $4 \times (20{,}000\ m^3) \times 0.026\ m^{-1} = 2080\ m^2$. If we consider absorption at 500 Hz the effect of air absorption would be negligible in comparison.

11.10 Early Decay time (EDT10)

A modification of the reverberation time T_{60} is the *early decay time*, or EDT10, which represents the time interval required for the first 10 dB of decay to occur, multiplied by 6 to produce an extrapolation to 60 dB decay Originally proposed by Jordan (1974), EDT10 is based on early psychoacoustical research, and according to Cremer and Muller (1982), "the latter part of a reverberant decay excited by a specific impulse in running speech or music is already masked by subsequent signals once it has dropped by about 60 dB."

11.11 Acoustic Energy Density and Directivity

In order to account for uneven distribution of sound in some sources, we express the sound intensity I (W/m^2) due to a point source of power W (W) in the direct field (i.e., reflections are not considered) as

$$I = \frac{WQ(\theta, \phi)}{4\pi r^2} \tag{11.19}$$

where r is the distance (m) from the source and $Q(\theta,\phi)$ is the *directivity factor*. The directivity factor $Q(\theta,\phi)$ equals unity for an ideal point source that emits sound evenly in full space. For an ideal point source above an acoustically reflective surface, in an otherwise free half-space, $Q(\theta,\phi)$ equals 2. The sound or acoustic energy density is the sound energy contained per m^3 at any instant. In the direct field in full space, the direct sound energy density D_D (W s/m^3) is given by

$$D_D = \frac{I}{c} = \frac{WQ(\theta, \phi)}{4\pi r^2 c} \tag{11.20}$$

where c is the speed of sound in m/s.

11.12 Sound Absorption in Reverberant Field: The Room Constant

The product IS gives the rate of acoustic energy striking a surface area S; and IS cos θ gives that rate for the incidence angle θ. In an ideal reverberant field, with equal probability for all angles of incidence, the average rate of acoustic energy striking one side of the surface is given by $IS/4$. The power absorbed by the surface having an absorption coefficient α is

$$\text{Power absorbed} = \frac{\alpha IS}{4} = \frac{\alpha c D_R S}{4}$$

where D_R denotes the reverberant sound field density. In a fairly steady state condition the power absorbed is balanced by the power supplied by the source to the reverberant field. This is the portion of the input power W that remains after one reflection:

$$\text{Power supplied} = W(1-\alpha)$$

The steady-state condition results in

$$\frac{c\, D_R\, S\alpha}{4} = W(1-\alpha)$$

which we rearrange to obtain the energy density in the reverberant field,

$$D_R = 4W\frac{1-\alpha}{\alpha c S} = \frac{4W}{cR} \tag{11.21}$$

where the room constant R is by definition

$$R = \frac{\alpha S}{1-\alpha}.$$

In most cases, the boundaries of the actual enclosure and other objects inside the enclosure are constructed of different materials with differing absorption coefficients. The room constant R of the enclosure is then described in terms of mean properties by

$$R = \frac{S_T\,\overline{\alpha}}{1-\overline{\alpha}}$$

where

$$\begin{aligned} R &= \text{the room constant (m}^2\text{)} \\ S_T &= \text{total surface area of the room (m}^2\text{)} \\ \bar{\alpha} &= \text{mean sound absorption coefficient} = \sum \alpha_i S_i / S_T. \end{aligned}$$

11.13 Sound Levels due to Direct and Reverberant Fields

Near a point of nondirectional sound source, the sound intensity is greater than from afar. If the source is sufficiently small and the room not too reverberant, the acoustic field very near the source is independent of the properties of the room. In other words, if a listener's ear is only a few centimeters away from a speaker's mouth, the room surrounding the two persons has negligible effect on what the listener hears directly from the speaker's mouth. At greater distances from the source, however, the direct sound decreases in intensity, and, eventually the reverberant sound predominates.

If we are more than one-third wavelength from the center of a point source, the energy density of a point r is given by Equation (11.20) for the direct sound field. Combining the Equations for the direct and the reverberant sound intensities, i.e., Equations (11.20) and (11.21), we get the total sound intensity I given by

$$I = W\left[\frac{Q(\theta,\phi)}{4\pi r^2} + \frac{4}{R}\right] \tag{11.22}$$

It is assumed that reverberant sound comes from nearly all directions in a fairly even distribution. The modes generated by standing waves must be rather insignificant; otherwise the assumption of uncorrelated sound is not valid and Equation (11.22) will not truly constitute the proper model for the actual sound field.

The sound pressure level within the room can now be found from

$$\begin{aligned} L_p &= 10\log\left(\frac{I}{I_{\text{ref}}}\right) = 10\log\left[W\left(\frac{1}{4\pi r^2}+\frac{4}{R}\right)\right]+120 \\ &= L_W + 10\log\left(\frac{1}{4\pi r^2}+\frac{4}{R}\right) \end{aligned} \tag{11.23}$$

for an ideal point source emanating equally in all directions. The reference sound intensity I_{ref} is equal to 10^{-12} W/m^2, and the sound power level L_W of the source is defined as

$$L_W = 10\log\left(\frac{W}{10^{-12}}\right)$$

which is given in dB re 1 pW. For an ideal point source over an acoustically reflective surface

$$L_P = 10\log\left[W\left(\frac{1}{2\pi r^2}+\frac{4}{R}\right)\right]+120 = L_W + 10\log\left(\frac{1}{2\pi r^2}+\frac{4}{R}\right) \tag{11.24}$$

Equations (11.23) and (11.24) are based on the fact that the absorption coefficients do not vary radically from point to point in the room and the source is not close to reflective surfaces. If the sound power and room absorption characteristics are assigned for each frequency band, the sound pressure level L_P can be determined for each frequency band in dB/octave, dB/one-third octave, and so on. If the sound power level is A-weighted, and if the room constant is based on frequencies in the same range as the frequency content of the source, the sound power level will be expressed in dB(A).

Example Problem 3

Predict the reading of a sound pressure level meter 12 m from a source having a sound power level of 108 dB(A) re 1 pW in a room with a room constant $R = 725$ m^2 (at the source frequencies). The source is mounted directly on an acoustically hard floor.

Solution

We apply Equation (11.24) as follows:

$$\begin{aligned} L_p &= L_W + 10\log\left(\frac{1}{2\pi r^2}+\frac{4}{R}\right) \\ &= 108 + 10\log[(2\pi \times 12^2\ \text{m}^2)^{-1} + 4/725\ \text{m}^2] \\ &= 86.2\ \text{dB}(A) \end{aligned}$$

If the meter would be placed at $R = 3$ m from the source, the SPL meter reading for L_p will increase to 91.7dB(A).

11.14 Design of Concert Halls and Auditoriums

Ideally, the main objective of auditorium design is to get as many members of the audience as close as possible to the source of the sound, because sound levels decrease with increasing distances from the sound source. A good visual line of sight usually results in good acoustics, so stepped seating becomes desirable for larger rooms seating more than 100 people. Reverberation should be controlled in order to provide optimum reinforcement and equalization of sound. For speech the room design should provide more in the way of direct sound augmented by reflections, while the clarity of articulation of successive syllables must be sustained. Rooms for music typically have longer reverberation times because the requirements for articulation are not as stringent, and more enhancement of the sound is desirable.

The aim of the design of a listening type of facility is to avoid the following acoustic defects (Siebein, 1994).

- *Echoes*, particularly those from the rear walls of the facility. Echoes can be lessened or eliminated by placing absorbent panels or materials on the reflecting walls or introducing surface irregularities to promote diffusion of the sound.
- *Excessive loudness* can occur from prolonged reverberation. Again, the proper deployment of absorbent materials should alleviate this problem.
- *Flutter echo* results from the continued reflection of sound waves between two opposite parallel surfaces. This effect can be especially pronounced in small rooms; and this can be contravened by splaying the walls slightly (so as to avoid parallel surfaces) or using absorbent material on one wall.
- *Creep* is the travel of sound around the perimeter of domes and other curved surfaces. This phenomenon is also responsible for whispering gallery effects in older structures with large domed roofs.
- *Sound focusing* arises when reflections from concave surfaces tend to concentrate the sound energy at a focal point.
- *Excessive* or *selective absorption* occurs when a material that has a narrow range of acoustical absorption is used in the facility. The frequency that is absorbed is lost, resulting in an appreciable change in the quality of the sound.
- *Dead spots* occur because of sound focusing or poorly chosen reflecting panels. Inadequate sound levels in specific areas of the listening facility can result.

11.15 Concert Halls and Opera Houses

Three basic shapes exist in the design of large music auditoriums, namely (1) rectangular, (2) fan-shaped, and (3) horseshoe, all of which are illustrated in floor plans of Figure 11.8. A fourth category is the “modified arena”, nearly elliptical in shape. The Royal Albert Hall (constructed in 1871) in London, the Concertgebouw

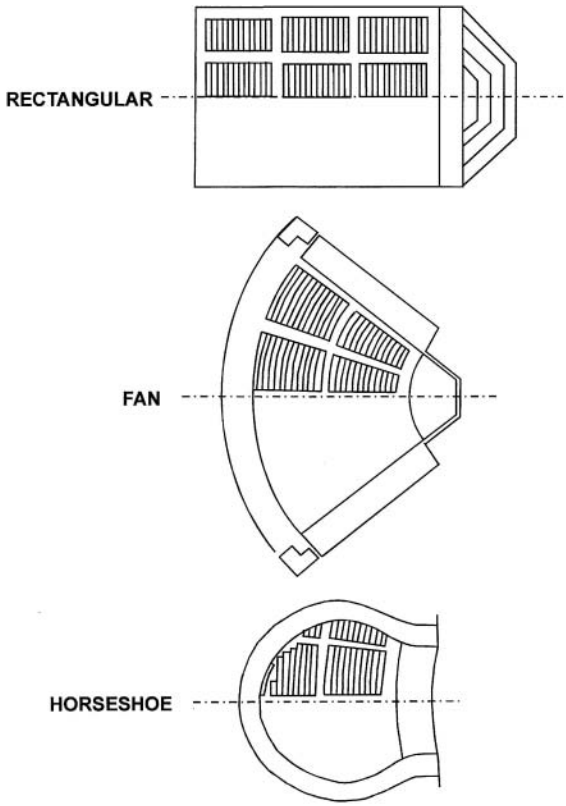

FIGURE 11.8. Three basic hall configurations: rectangular, fan shaped, and horseshoe shaped.

(1887) in Amsterdam, the Sidney Opera House (opened in 1973), and the Colorado Symphony's Boettcher Concert Hall in Denver (opened in 1978 and acoustically remodeled in 1993) are examples of this type of facility.

The *rectangular hall* is quite traditional, and it has been built to accommodate both small and large audiences. But these halls will always generate cross reflections (flutter echoes) between parallel walls. Sound can also be reflected from the rear walls back to the stage, depending on balcony layout and the degree of sound absorption. These reflections can help in the buildup of sound and provides a reasonable degree of diffusion in halls of modest interior dimensions. A considerably larger hall can result in standing wave resonances and excessive flutter echoes.

It is interesting to note that the first music hall to be designed from a scientific viewpoint, by none other than Wallace C. Sabine, is the Boston Symphony Hall (1900), views of which are given in Figures 11.9 and 11.10. The structure contains a high, textured ceiling and two balconies extending along three walls. Volume is 602,000 ft^3; seating capacity 2631; and the reverberation time in the 500–1000 Hz

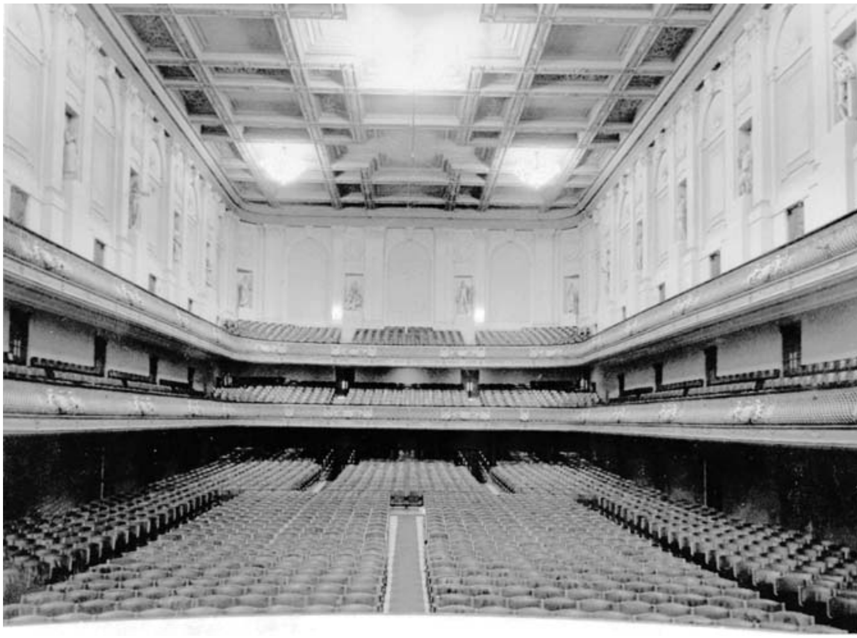

FIGURE 11.9. A view of the Boston Symphony Hall from the stage. Wallace Clement Sabine was the principal acoustical consultant for this facility. Reprinted with permission from Leo Beranek, *Concert Halls and Opera Houses: Music, Acoustics, and Architecture*, 2nd ed. (New York, NY: Springer, 2004), 48.

FIGURE 11.10. A stage area of the Boston Symphony Hall. Reprinted with permission from Leo Beranek, *Concert Halls and Opera Houses: Music, Acoustics, and Architecture*, 2nd ed. (New York, NY: Springer, 2004), 48.

range is 1.8 s (occupied). Another example of a great rectangular hall is the venerable Grösser Musikvereinssaal (1870) in Vienna which has a reverberation time of 2.05 (occupied) in a volume of 530,000 ft^3. Its superior acoustics can be attributed to its relatively small size, high ceiling, irregular interior surfaces and the plaster interior (Beranek, 2004).

A *fan-shaped hall* accommodates, through its spread, a larger audience within closer range from the sound source (stage). It features nonparallel walls that eliminate flutter echoes and standing waves; and most audience members can obtain a pleasing balance between direct and reflected sounds. A disadvantage in terms of early time delay gap is the distance from the side walls. Often it is necessary to add a series of inner reflectors or canopies hanging from ceilings over the proscenium area to maintain articulation and other acoustical characteristics. Many architects in the United States have resorted to the fan-shaped hall design in order to accommodate larger audiences while retaining an appreciable degree of both visual and aural coupling to the stage area. Relatively modern examples of this design are the Dorothy Chandler Pavilion (1964) in Los Angeles; the Orchestra Hall (completed in 1904 and most recently renovated in 1997, Chicago; the Eastman Theater (opened in 1922 as a movie theater and converted into a concert hall in 1930) in Rochester, New York; and the Kleinhaus Music Hall (1940, designed by Eliel and Eero Saarinen) in Buffalo, New York.

Over a number of centuries *horseshoe-shaped structures* have been used as the preferred design for opera houses and concert halls of modest seating capacity. This design provides for a greater sense of intimacy, and the textures of convex surfaces promote adequate diffusion of sound. The multiple balconies allow for excellent line of sight and short paths for direct sound. The La Scala Opera House (Figures 11.11a, b) in Milan is probably the most notable example of the horseshoe design. It was opened in 1778. This edifice, formally known as Teatro alla Scala, was closed in 2001 for 3 years to undergo a badly needed renovation. A tubular structure and a 17-story fly tower designed by Mario Botta were added to provide stagecraft storage, dressing rooms, and rehearsal rooms. In addition to repairing the ravages of time on the structure, modern stage machinery and new wiring were installed, as well as a new heating, ventilation and air conditioning system. The acoustics were improved by the prominent acoustician, Higini Arau from Barcelona. Other celebrated examples of the horseshoe design are the Carnegie Hall (completed 1897, renovated 1983–1995) in New York City and the Academy of Music (the first opera house in the United States, opened in 1857) in Philadelphia.

Nearly all concert halls have balconies, which were designed to accommodate additional seating capacity within a smaller auditorium volume, so that listeners can sustain an intimate relationship with the stage. The depths of the balconies generally do not exceed more than twice their vertical "window" (opening) to the stage. In fact a smaller ratio is desirable to minimize undue sound attenuation at the rear wall. A rule of the thumb in contemporary acoustical design: the depth of the balcony should not exceed 1.4 times its outlook to the stage at the front of the balcony.

In all types of auditorium design, ceilings constitute design opportunities for transporting sound energy from the stage to distant listeners. In Figure 11.12, it is

(a)

(b)

FIGURE 11.11. (a) View looking toward the stage of the La Scala Opera House. The theater was closed down for three years for seriously needed renovations; (b) a classic example of the horseshoe configuration. (Courtesy of Dr. Antonio Acerba Cantier Escala)

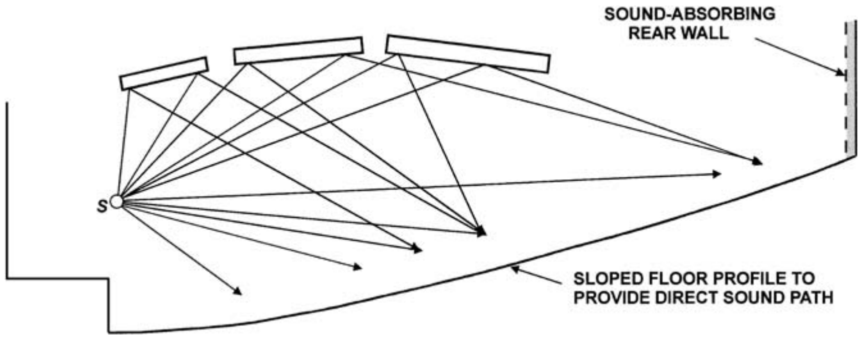

FIGURE 11.12. Transmission of sound to all areas of an auditorium through the ceiling and floor profiling.

shown how a ceiling can convey sound to the listeners without imposing a great time difference between direct and ceiling-reflected sound. Floor profile is also important in establishing the proper ratio of direct to indirect sound. Splays on the side walls have proven effective in promoting diffusion and uniformity of loudness. Rear walls generally should be absorbent to minimize echoes being sent back to the stage.

Many concert halls have been built throughout the world, some with outstanding acoustics and others resulting in dismal sound. Among the features common to all of the aurally superior halls are a limited audience capacity (generally 2800 seats or less), extreme clarity of sound so that the audience can clearly distinguished the individual instruments of the orchestra without loss of fullness or blending of tones associated with reverberation. A good hall also allows the orchestra to hear itself.

Table 11.2 contains a summary of the characteristics of a number of prominent musical facilities all over the world. Figure 11.13 shows the stage view of the Orchestra Hall in Minneapolis (1974), which is patterned after the classical rectangular design. The hall contains a slanted floor with 1590 seats and an additional 983 seats are located in the three-stepped balcony tiers, making for a total audience capacity of 2573. A random pattern of plaster cubes covers the ceiling, providing effective diffusion of sound throughout the hall. This overlay of cubes also continues down the back wall, behind the stage, as shown in Figure 11.13. Wood paneling partially covers the walls, and both the flooring of the stage and audience areas are wood. This concert hall is notable for its clarity, dynamic range, and balance.

Another acoustic success among the contemporary musical facilities is the Kennedy Center for the Performing Arts which opened in Washington, D.C. in 1971. The Center consists of a single structure that contains a 2759-seat concert hall, a 2319-seat opera house (Figure 11.14), and a 1142-seat theater. The location was environmentally challenging, for the Center is situated on a site near the Potomac River, in close proximity to the Washington Ronald Reagan National Airport on the other side of the river. Both commercial and private aircraft fly as

TABLE 11.2. Reverberation Times of Leading Concert Halls and Auditoriums.

			Reverberation Time[a]	
	Volume (ft^3)	Seating Capacity	Occupied	Unoccupied
United States				
Baltimore, lyric Theatre	744,000	2616	1.47	2.02
Boston Symphony Hall	662,000	2631	1.8	2.77
Buffalo, Kleinhans Music Hall	644,000	2839	1.32	1.65
Cambridge, Kresge Auditorium	354,000	1238	1.47	1.7
Chicago, Aric Crown Theatre	1,291,000	5081	1.7	2.45
Cleveland, Severance Hall	554,000	1890	1.7	1.9
Detroit, Ford Auditorium	676,000	2926	1.55	1.95
New York, Carnegie Hall	857,000	2760	1.7	2.15
Philadelphia Academy of Music	555,000	2984	1.4	1.55
Purdue University Hall of Music	1,320,000	6107	1.45	1.6
Rochester, New York, Eastman Theatre	900,000	3347	1.65	1.82
Austria				
Vienna, Grosser Musikvereinssaal	530,000	1680	2.05	3.6
Belgium				
Brussels, Palais des Beaux-Arts	442,000	2150	1.42	1.95
Canada				
Edmonton and Calgary, Alberta Jubilee Halls	759,000	2731	1.42	1.8
Vancouver, Queen Elizabeth Theatre	592,000	2800	1.5	1.9
Denmark				
Tivoli Koncertsal	450,000	1789	1.3	2.25
Finland				
Turku, Konserttisali	340,000	1002	1.6	1.95
Germany				
Berlin, Musikhochschule Konzertsaal	340,000	1340	1.65	1.95
Bonn, Beethovenhalle	555,340	1407	1.7	1.95
Great Britain				
Edinburgh, Usher Hall	565,000	2760	1.65	2.52
Liverpool Philharmonic Hall	479,000	1955	1.5	1.65
London, Royal Albert Hall	3,060,000	5080	2.5	3.7
London, Royal Festival Hall	755,000	3000	1.47	1.77
Israel				
Tel Aviv, Frederic R. Mann Auditorium	750,000	2715	1.55	1.97
Italy				
Milan, Teatro Alla Scala	397,300	2289	1.2	1.35
Netherlands				
Amsterdam, Concertgebouw	663,000	2206	2.0	2.4
Sweden				
Gothenburg, Konserthus	420,000	1371	1.7	2.0
Switzerland				
Zurich, Grosser Tonhallesaal	402,500	1546	1.6	3.85
Venezuela				
Caracas, Aula Magna	880,000	2660	1.35	1.8

[a] At 500–1000 Hz.

FIGURE 11.13. The stage area of the Minnesota Orchestra Hall in Minneapolis. (Courtesy of the Minnesota Orchestral Association.)

low as a few hundred feet directly over the roof, and occasionally helicopters pass by along the Potomac River at rooftop levels. In addition, vehicular traffic runs across the river and directly beneath the plaza of the Center.

To deal with these external noise sources, the Center was constructed as a box-within-a-box, so that each of the auditoriums is totally enclosed within an outer shell. The columns within each auditorium are constructed to isolate interior

FIGURE 11.14. The interior of the John F. Kennedy Opera House in Washington, DC. (Courtesy of the John F. Kennedy Center. Photograph by Scott Suchman.)

ceilings, walls, and floors from both airborne and mechanical vibrations. The double-wall construction generally consists of 6-in. solid high-density blocks separated by a 2-in. air gap. The huge windows in the Grand Foyer facing the river consist of 1.27-cm (1/2-in.) and 0.64-cm (1/4-in.) thick glass sheets separated by a 10-cm (4-in.) air gap. Resilient mounts are used to isolate interior noise sources (e.g. transformers, air-conditioning units, etc.) The ductworks are acoustically lined, flexible connectors are used, special doors are installed at all auditorium entrances together with "sound locks" between foyer and the auditoriums.

The rectangular concert hall encompasses a volume of 682,000 ft^3 and accommodates an audience of 2759. Large contoured wall surfaces and a coffered ceiling abet the diffusion of sound at low and high frequencies. The 11 massive crystal chandeliers, each weighing 1.3 metric ton, donated by the Norwegian government, also contribute to the diffusion. The balconies are purposely shallow to prevent reduction of sound below the balcony overhang. Unoccupied, the concert hall has a reverberation time of 2.2 s at 500 Hz and 2.0 s at 1 kHz; the corresponding values are 2.0 s and 1.8 s for the fully occupied hall.

Located in downtown Seattle, Washington, Benaroya Hall opened in September 1998, contains two spaces for musical performances: a 2500-seat main auditorium (Figure 11.15) and a more intimate 540-seat recital hall. The main auditorium, the S. Mark Taper Foundation Auditorium, is a classic rectangular configuration with the stage enclosed in a permanent acoustic shell. LMN (Loschky, Marquartdt and

FIGURE 11.15. Seattle Symphony performing in Benaroya Hall in Seattle, WA under the direction of Music Director Gerard Schwarz. (Courtesy of the Seattle Symphony, photograph by Craig Raymond.)

Nesholm) Architects and the acoustical consultant, Cyril M. Harris, combined the shoebox design with state-of-the-art materials to achieve maximum warmth and balance.

The location of Benaroya Hall in a busy sector of Seattle posed special challenges to the designers. They had to contend with a railroad tunnel running diagonally beneath the auditorium and a nearby underground bus tunnel. A slab of concrete more than 2m (6 ft) thick, 24 m (80 ft) wide, and 131 m (430 ft) long was poured under the hall to swallow the sound from the tunnels. In order to combat other exterior noises, the designers essentially encased a building within a building. The auditorium, weighing 12 million kilograms, rests on 310 rubber pads, which absorb vibration from the tunnels. The pads are 38 cm^2 and are composed of four layers of natural rubber sandwiched with 0.32-cm (1/8-in.) steel plates.

All electrical, plumbing, and other noise-generating equipment are located outside the auditorium box. Any penetration of the box is made with flexible connections. Water is known to transmit sound very well, so the fire sprinkler system is left dry and it will flood with water only when a fire is detected. The ventilation system is connected to the outside by a sound trap, which channels air through narrow openings between perforated aluminum boxes of sound insulation. Very large vents collect air below the floor and move it slowly behind the auditorium to another sound trap and from thereon to fans. The basic idea of the ventilation system is to move a high volume (2400 m^3/min) of air at low speed, eliminating noise created by fast-moving air in conventional systems.

Instead of frame construction, the walls are built of precast concrete panels. The heavy mass helps to cut down building vibration and provides a stiff, hard surface to reflect concert sound. Side walls, back walls, and ceiling are covered with paneling shaped like truncated pyramids to reflect sound at various angles to aid diffusion. Randomly spaced wood blocking behind the angled paneling creates framed sound boxes that reflect both high and low frequency sounds so that no tone is eliminated from the music. Side walls are covered with particle boards veneered with a dense, fine-grained hardwood from a single makore tree. The ceiling is suspended from the roof by hundreds of metal strips. The ceiling is coated with 3.8 cm (1.5 in.) of plaster in irregularly shaped panels to diffuse sound. The plaster is sufficiently dense to prevent the ceiling from vibrating. House lighting is imbedded in the ceiling to minimize sound leakage. Access to the light bulbs is achieved above the ceiling through heavy, removable plaster caps.

11.16 Band Shells and Outdoor Auditoriums

Over the past several decades there has been an increasing trend toward outdoor concerts, either at band shells or in semi-open structures. These types of structures are more economical to construct than full-fledged indoor concert halls, and they also meet the criteria of providing an informal setting for audiences seeking entertainment in a usually rural environment, away from the metropolitan areas.

FIGURE 11.16. The Hollywood Bowl in Los Angeles, California, after its reconstruction in 2004. (Courtesy of the Los Angles Philharmonic, photograph by Mathew Imaging.)

It is generally not possible for a large orchestra to play effectively in open air. The use of a band shell becomes necessary, as this permits the members of a musical group to hear each other and directs the music toward the audience area. The band shell site should also be carefully selected. Ideally, the region should be isolated from the noise of passing traffic and overhead aircraft. The topology of the land also ranks important in providing the proper acoustics. If the land can be contoured properly, there can be appreciably less attenuation of the sound than would be the case if the band shell were located on flat ground.

The Hollywood Bowl (Figure 11.16) is an example of an orchestra shell strategically located in a natural hollow. The only reflected sound is that reflected from the shell, but the stage distances are sufficiently short so that the sound is heard without any discernable time delay gap. However, shells can never equal the dynamic range of sound power and sonority that are achieved in an enclosed reverberant concert hall. The use of high quality amplification systems is therefore often necessary at many outdoor concerts.

The Bowl, the summer home of the Los Angeles Philharmonic, has undergone a number of changes throughout its years of operation. The previous shell design—the fourth since 1922—has been subjected to a great deal of criticism from performers and audiences alike. In 2004, the shell and stage area was reconstructed and made 30% larger to accommodate a full orchestra. The new shell is provided with a set of adjustable reflectors above the orchestra, mounted on a 27.5 m × 18.2 m (90 ft × 60 ft) elliptical structure that also supports an improved lighting

FIGURE 11.17. A view looking toward the interior of the Tanglewood Music Shed in Lenox, MA. The ceiling canopy reflects sound, adding dynamic range and brilliance. (Photograph by Kim Knox Beckhius.)

system. Advantage was also taken of the opportunity to install a completely new sound system.

Another type of "outdoor" structure is the "music shed," a semi-enclosed structure specifically designed for musical performances. The Tanglewood Music Shed in the Berkshires region of Massachusetts (Figure 11.17) is the summer home of the Boston Symphony Orchestra, and the quality of its acoustics exceeds that of any band shell for an audience of 6000 persons. An additional 6000 people on the lawn adjacent to the shed can listen to the music through the open segments of the pavilion. The canopy of the interior projects and diffuses sound through a volume of 42,500 m^3 (1,500,000 ft^3). The ceiling is constructed of 5-cm thick wooden planks; the side and rear walls are of 2 cm fiberboard; and the floor is simply packed earth. When occupied, the shed embodies a reverberation time of 2 s in the frequency range of 500–1000 Hz, which is quite excellent considering the rustic nature of its construction. The Tanglewood shed is the precursor to similar structures at Wolf Trap in Vienna, Virginia, the Performing Arts Center at Saratoga Spring, New York (the summer home of the Philadelphia Symphony Orchestra), and the Blossom Music Center near Cleveland, Ohio.

On July 16, 2004, the Jay Pritzker Pavilion (Figure 11.18), a radical outdoor concert facility located at Chicago's Millennium Park near the banks of Lake Michigan, presented its first musical program. Working in concert with the eminent architect Frank Gehry, the consulting firm TALASKE of Oak, Park, Illinois dealt with the acoustic challenge of outdoor orchestral performances. TALASKE

FIGURE 11.18. The Jay Pritzker Pavilion in Chicago. The radical design also features a trellis system from which loudspeakers are suspended. (Courtesy of TALASKE, graphics by DOXA.)

established the overall shape and arrangement of the stage enclosure and, in the course of design, refined the shapes and finishes of the interior surfaces. The stage of the pavilion (Figure 11.19), the permanent home of the Grant Park Music Festival Orchestra and Chorus, can accommodate a 100 plus member orchestra and

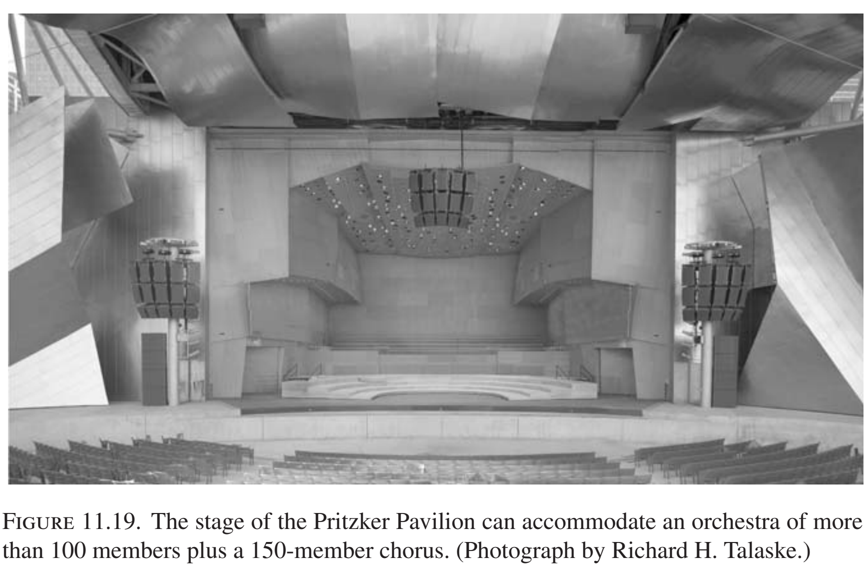

FIGURE 11.19. The stage of the Pritzker Pavilion can accommodate an orchestra of more than 100 members plus a 150-member chorus. (Photograph by Richard H. Talaske.)

also a 150-member chorus. The walls and the ceiling of the stage incorporate contours and angles precisely planned to enhance cross-room reflections so that musicians can hear each other from one side of the stage to the other and from front to back. When an orchestra performs at high volume it is often difficult for musicians playing in the high register to hear instruments in the lower registers. To circumvent this problem, the orchestra risers feature a platform system, researched and designed by the acousticians and musicians together, that allows the musicians to feel the vibration created by the cellos and bass instruments. The riser system, essentially a floating floor with rigid interconnections and resilient materials just below the floor surface, acts to maximize cross stage vibrations. This enables a more precise ensemble and tonal balance.

To compensate for the lack of reflected sound in an outdoor setting and the lack of envelopment, modern technology was recruited to improve the sound environment by blending architecturally created and electronically processed sound. The Jay Pritzker Pavilion, which replaces the Petrillo Music Shell constructed in 1931 and still in use on a more limited basis, is the first orchestral facility that distributes sound by using an overhead steel trellis structure from which loudspeakers are suspended (Figure 12.20). The trellis area that incorporates the network of speakers is 100 m (325 ft) wide by 180 m (600 ft) long. These loudspeakers are strategically suspended at predetermined heights and orientations, in concentric circles outward from the stage. A distributed sound reinforcement system provides direct or "frontal" sound to the audience. A separate acoustic enhancement system,

FIGURE 11.20. View from the stage of the Pritzker Pavilion. The trellis overhead of the audience contains a distributed loudspeaker system.

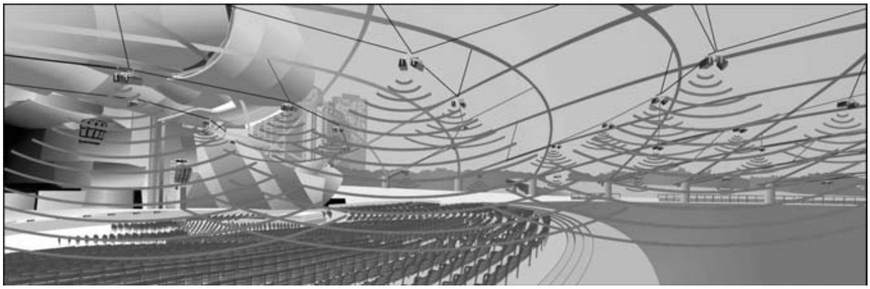

FIGURE 11.21. Distribution of sound from the speakers distributed on a trellis system overhead of the audience at the Pritzker Pavilion. (Courtesy of TALASKE, graphics by DOXA.)

LARES^{TM} (Lexicon Acoustic Reinforcement and Enhancement System), delivers lingering, enveloping sound characteristics through supplemental loudspeakers. It uses a time varying technique to maintain stability by shifting the output in time enough to maintain stability but not enough to introduce tonal coloration. This system simulates reflections and reverberations using specialized electronics and digital processing. Time delay creates the impression that sound is coming from the stage rather than from the speakers. These two systems work together to deliver sound throughout the seating area and the Great Lawn (cf. Figure 11.21). The system makes use of eight microphones deployed around the stage area for the enhancement system and 30 or so microphones for reinforcement, 143 amplifiers housed in three separate equipment rooms, 90 CobraNet^{TM} channels, and 185 speakers. The beauty of this system is that the acoustic environment can be adjusted to work equally well for orchestral music, staged opera, blues, jazz, rock, and other music. The fixed seating area accommodates 4000 concertgoers, and the Great Lawn has a capacity for a lawn crowd of up to 7000 stretching more than a city block behind the front area.

Backstage facilities include warm-up rooms that are shared with the Harris Theater for Music and Dance. A bridge near the concert area was configured to help mask the traffic noises at the pavilion.

11.17 Subjective Preferences in Sound Fields of Listening Spaces

Beyond the Sabine realm of architectural acoustics, which is based on the relatively simple but effective formula $T_{60} = (0.161V)/\,\Sigma A_i\alpha_i$, other considerations come into play in determining optimal configurations for listening spaces (Ando, 1998). This involves combining the elements of psychoacoustics, modeling of the auditory-brain system, and mapping of subjective preferences. The physical properties of source signals and sound fields in a room are considered, in particular the autocorrelation function (ACF) that contains the same information as power density spectrum but it is adjusted to account for hearing sensitivity. Effective duration of the normalized ACF is defined by the delay τ_e at which the envelope of the normalized ACF becomes one-tenth of its maximal value. The response of the ear includes the effects of time delay due not only to the room's acoustical characteristics, but also the spatially incurred differences in the signals reaching the right and the left ears. The difference in the sounds arriving at the ear is measured by the "interaural cross-correlation function" or IACF, which is defined by

$$\mathrm{IACF}(\tau) = \frac{\int_{t_1}^{t_2} p_L(t) p_R(t+\tau)}{\sqrt{\int_{t_1}^{t_2} p_L^2 \int_{t_1}^{t_2} p_R^2 \, dt}} \tag{11.25}$$

where L and R denotes entry to the left and right ears, respectively. The maximum possible value of IACF is unity. The time $t = 0$ is the time of the arrival of the *direct* sound from the impulse radiated by a source. Integration from 0 to t_2 ms includes the energy of the direct sound and any early reflections and reverberant sounds occurring during the t_2 interval. Because there is a time lapse of about 1 ms for a sound wave to impinge the other side of the ear after impinging one side, it is customary to vary τ over the range from –1 ms to +1 ms. In order to obtain a single number that measures the maximum similarity of all waves arriving at the two ears with the time integration limits and the range of τ, it is customary to choose the maximum value of IACF, which is then called the interaural cross-correlation coefficient (IACC), i.e.

$$\mathrm{IACC} = |\mathrm{IACF}(t)|_{\max}$$

Different integration periods are used for IACC. The standard ones include IACC_A ($t_1 = 0$ to $t_2 = 1000$ ms), and $\mathrm{IACC}_{\mathrm{E(arly)}}$ (0–80 m), IACC $_{\mathrm{L(ate)}}$ (80–1000 m). The early IACC is a measure of the apparent source width ASW and the late IACC is a measure of the listener envelopment LEV. IACC is generally measured by recording on a digital tape recorder the outputs of two tiny microphones located at the entrances to the ear canals of a person or a dummy head, and quantifying the two ear differences with a computer program that performs the operation of Equation (11.25). $\mathrm{IACC_A}$ is determined with a frequency bandwidth of 100 Hz–8 kHz and for a time period of 0 to about 1 s.

Subjective attributes for a sound field in a room have been developed experimentally with actual listeners. The simplest sound field is considered first, a situation which consists of the direct sound and a single reflection acting in lieu of a set of reflections. The data obtained are based on tests in anechoic chambers (which allowed for simulation of different concerts halls) with normal hearing subjects listening to different musical motifs. From these subjective tests the optimum design objectives are established, namely the listening level, preferred delay time, preferred subsequent reverberation time (after the early reflections), and dissimilarity of signals reaching both ears (involving IACC).

These factors are each assigned scalar values and then combined to yield a subjective preference that can vary from seat to seat in a concert hall. Some rather interesting results of investigations include the fact that the right hemisphere of the brain is dominant for "the continuous speech." while the left hemisphere is dominant when variation occurs in the delay time of acoustic reflection. The left hemisphere is usually associated with speech and time-sequential identifications, while the right hemisphere is allied with nonverbal and spatial identifications. A proposed model for the auditory-brain system was developed (Ando, 1998) that incorporates a subjective analyzer for spatial and temporal criteria and entails the participation of the left and the right hemispheres of the brain. The power density spectra in the neural processes in the right and left auditory pathways yield sufficient information to establish autocorrelation functions.

It is obvious that different individuals are likely to have different subjective preferences with respect to the same musical program, so seating requirements can differ, with respect to the preferred listening level and to the initial time delay, and even lighting. For example, evaluations were conducted for a performance of Handel's *Water Music* with 106 listeners providing the input on their preferences with respect to listening level, reverberation time, and IACC. The information thus obtained can provide insight into how the acoustic design of a concert hall and a multipurpose auditorium can be accomplished. Procedures for designing sound fields include consideration of temporal factors, spatial factors, the effect of sound field on musicians, the conductor, stage performers, listener, and the archetypal problem of fusing acoustical design with architecture. Multipurpose auditoriums present bigger challenges, some of which have been met very well and many which have not. In the design procedure, a number of factors other than acoustical include measurable quantities such as temperature, lighting levels, and so on, and other less tangible determinants that can be aesthetically evocative.

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Problems for Chapter 11

1. The amplitude of the reflected wave is one-half that of the incident wave for a certain angle and frequency. What is the reflection coefficient? What is the corresponding sound absorption coefficient?
2. Find the average sound coefficient of 125 Hz and 2 kHz for a wall constructed of different materials as follows:

Area (m^2)	Material
85	Painted brick
45	Gypsum board on studs
35	Plywood paneling
4	Glass window

3. Find the time constant of a room that is $8 \times 10 \times 2.5$ m and an average sound absorption coefficient of 0.34.
4. A room has dimensions 3.5 m high $\times$ 30 m length $\times$ 10 m width. Two of the longer walls consist of plywood paneling; the rear wall is painted brick and the front has gypsum boards mounted on studs. The ceiling is of acoustic paneling and the floor is carpeted. For 250 Hz and 125 people in the audience:
   (a) Determine the reverberation time from the Sabine equation.
   (b) Compute the room constant R.
   (c) Comment on the suitability of the room for use as an auditorium.
5. An auditorium has dimensions 6.0 m high $\times$ 22.0 m length $\times$ 15.5 m width. The floor is carpeted and one of the longer walls has gypsum boards mounted on studs the entire length, while the other three walls and the ceiling are constructed of plaster. For 500 Hz:
   (a) determine the reverberation time from the Sabine equation.
   (b) compute the room constant R.
6. In Problem 5 there is a set of two swinging doors 2.5 m in height and 2.5 m in total width. However, there are open cracks along the bottom and between the doors 1 cm wide. In addition there is an open window on the gypsum-boarded wall that is 0.8×1 m. Find the effect of these openings on the reverberation time of the auditorium.
7. A room has dimension 4.0 m high $\times$ 18.0 m length $\times$ 9.5 m width. The floor is carpeted and one of the longer walls has gypsum boards mounted on studs along the entire length, while the other three walls are constructed of painted brick. The ceiling is plastered. For 500 Hz:

(a) determine the reverberation time from the Sabine equation.
(b) compare the reverberation time obtained from the Eyring formula with that obtained in part (a) above.
(c) compute the room constant R.

8. An isotropic source emitting sound power level of 106 dB(A) re 1 pW is operating in the room of Problem 4. What sound pressure (on the A-weighted scale) will be registered by a meter 5.2 m from the source? If 125 people are *not* seated in the room, how will it affect the reading? Assume that no one blocks the path of direction propagation between the meter and the signal source.
9. Why does an anechoic chamber provide what we call a free field?