Chapter Ten

Room Acoustics

Sound behaves very much like light in several respects even though the wavelength of sound is several orders of magnitude longer than the wavelength of light. As we saw in the previous chapter, the wavelength of sound ranges from about ¼ IN to 37 FT compared with less than one ten-thousandth of an inch for light (see Chapter Six)[1]. Sound can bend around obstructions and if the obstruction is very much smaller than the wavelength of the sound then there may be virtually no acoustic shadow on the other side of the obstruction.

In a simplified manner, the path of sound in an enclosed space can be simulated with light rays emanating from a point source. Even though the light rays cannot reproduce the spherical progression of sound they do provide an accurate indication of how the sound waves will be reflected from surfaces in their path. While this kind of model analysis falls far short of a comprehensive representation of the acoustic environment that is likely to be experienced in the space, it does provide valuable information at a relatively low level of effort. The required model can be constructed out of inexpensive cardboard and a commonly available laser pointer can be used as a light source.

Building designers may resort to this kind of model analysis because the bounding surfaces of an interior space have a profound influence on the quality of sound in that enclosed space. For this reason, before we can explore the acoustics of building spaces for speech and music we must digress briefly to consider the principles of sound reflection and diffraction.

10.1 Reflection and Diffraction of Sound

When a sound wave strikes the boundary surface of an enclosed space part of the sound energy is reflected back into the space, part is transmitted through the boundary, and part is absorbed by conversion to heat within the material (Figure 10.1). This relationship between reflected, transmitted and absorbed sound is very conveniently expressed in terms of numerical coefficients as either percentages or fractions (i.e., decimal values) of one whole, as follows:

Reflection Coefficient (ρ) = (reflected sound energy)/(incident sound energy)

Absorption Coefficient (α) = (absorbed sound energy)/(incident sound energy)

Transmission Coefficient (τ) = (transmitted sound energy)/(incident sound energy)

However, the Absorption Coefficient is often considered to include the Transmission Coefficient when we are concerned simply with the proportion of sound that remains in the space. Under these circumstances the distinction between absorption and transmission is of little consequence. Similarly, when we are concerned with the transmission of noise from one space to another we typically do not distinguish between reflected and absorbed sound since neither of these reaches the external side of the noise barrier.

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[1] The wavelength of the visible portion of the electromagnetic spectrum (i.e., light) extends from about 450 to 750 millimicrons, which is equivalent to a range of 0.00011 to 0.00019 IN.

The property of a boundary that exerts most influence on its sound reflection, absorption and transmission characteristics is the density of the material that the boundary is composed of. This statement requires further discussion. As explained in the previous chapter, sound is a physical force that generates pressure. Therefore, the transmission of sound from one side of a barrier, such as a wall, to the other side requires the actual movement (i.e., vibration) of the wall. In other words, the pressure created by the sound waves on the internal side of the wall creates a force that causes the wall to deflect slightly. This deflection, however slight, will produce sound waves on the other side of the wall. Successive sound waves produce successive deflections, so that the wall starts to vibrate. Under most sound conditions experienced in buildings this vibration is so slight that it cannot be seen. However, quite often it can be felt when the wall is touched lightly with one hand and the sound source is low frequency such as a truck passing by. Naturally, the heavier the wall the more sound energy will be required to set it into vibration.

There are of course other factors involved. For example, the stiffness (i.e., rigidity) of the wall also plays a role. A stiff wall is set in vibration more easily than a pliable wall, which will tend to dampen the mechanical reaction to the incident sound waves through its sluggishness. These and other factors involved in the transmission of sound through a barrier are discussed in more detail in the next chapter.

What is perhaps less intuitively obvious is why a heavy wall can be expected to have a lower coefficient of absorption. The reason is that a dense material is also less porous and this reduces the ability of sound waves to penetrate into the material and set air pockets within the material into vibration thereby converting some of their mechanical energy into heat energy due to friction. As a result, since a heavy wall transmits less sound and absorbs less sound a greater proportion of the incident sound is reflected.

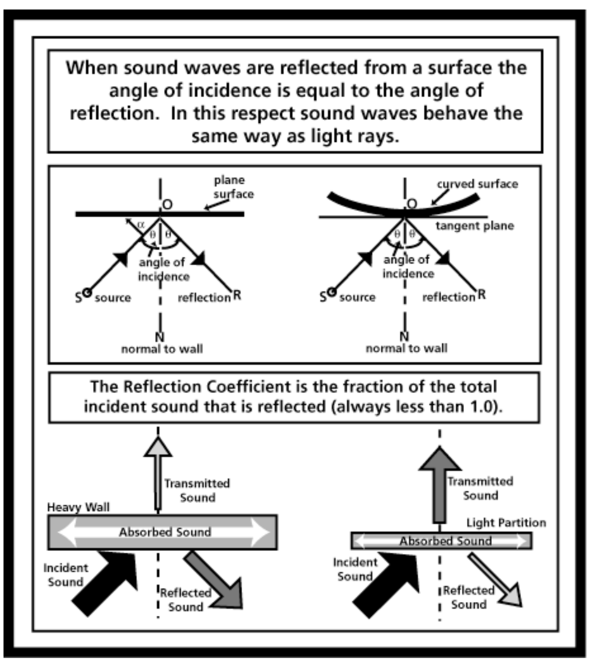

Figure 10.1: Reflection of sound

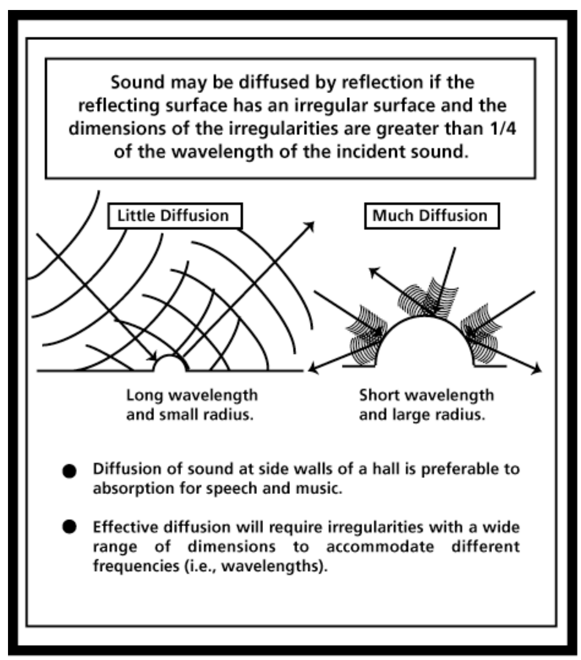

Figure 10.2: Reflection on irregular surfaces

When sound is reflected from a hard, smooth surface the law of reflection, which also applies to light, is closely obeyed. Typical reflection patterns for plane and curved walls are shown in

Figure 10.1. In the case of a curved wall the reflection occurs as if a tangent plane exists at the point where the incident sound ray (i.e., wave) meets the wall. If the surface is irregular, as shown in Figure 10.2, then the degree to which the reflections are diffused depends on the relationship between the size of the irregularities and the wavelength of the incident sound. As a rule of thumb, the amount of diffusion will be significant if the wavelength of the incident sound is less than four times the radius of curvature of the protrusions.

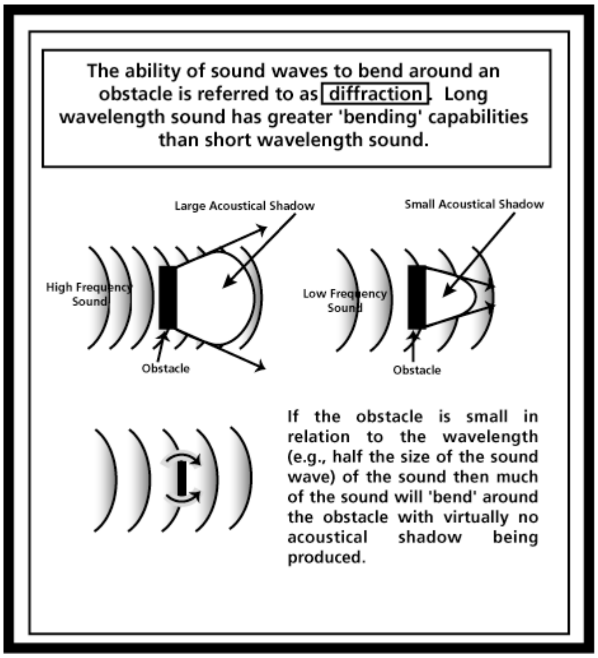

Figure 10.3: Acoustical shadows

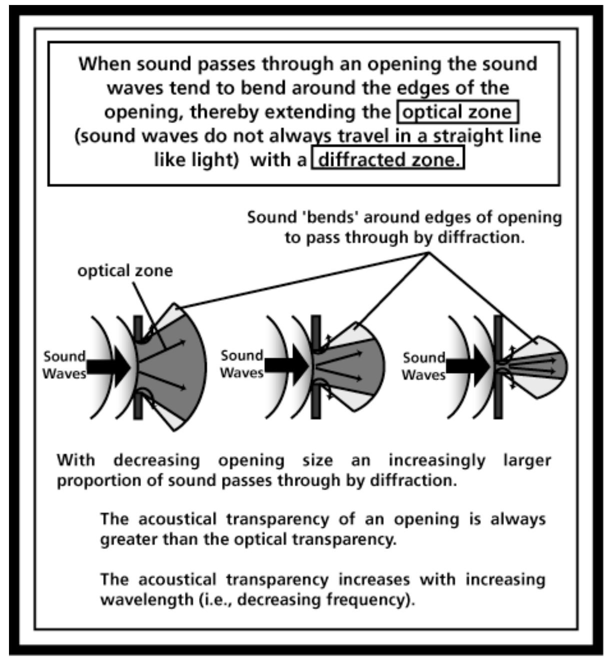

Figure 10.4: Diffraction around openings

If the surface is not a continuous boundary but takes the form of a reflecting panel or shield that is placed in the path of the incident sound then the behavior of the sound depends very much on its wavelength. If the dimensions of the panel are not several times larger than the wavelength then much of the sound will tend to bend or *diffract* around the obstacle in its path (Figure 10.3). In this respect the behavior of sound appears to be quite different from the interaction of light with an obstacle. However, in reality this is not true. It is simply a matter of scale. Due to the much shorter wavelength of light (see the discussion at the beginning of this chapter) the physical dimensions of any surface that might be encountered in a building will be several orders of magnitude larger.

In halls for speech and music reflecting panels are often used to improve and control the quality of the acoustic environment. To be effective the dimensions of these panels must be at least five times larger than the wavelength of the sound to be reflected. As an example, let us consider a lecture hall. What are the minimum dimensions of a suspended panel designed to reflect to the front section of the audience the typically mid-frequency speech sounds produced by the lecturer?

mid-frequency = 500 to 2000 cps

wavelength of sound = [(speed of sound) / (frequency of sound)]

speed of sound in air = 1100 FT/sec

wavelength at 500 cps = [1100 / 500] = 2.2 FT wavelength at 2000 cps = [1100 / 2000] = 0.6 FT

Therefore, to effectively reflect sound waves at the lower end of the mid-frequency range (i.e., 500 cps) the dimensions of the panel need to be at least 11 FT by 11 FT.

Conversely, the acoustical shadow that will be produced behind a large reflecting surface may be highly undesirable or very useful, depending on the circumstances. In the case of an auditorium the acoustic shadow that is likely to occur under deep balconies must be avoided by the designer, because of its unfavorable effect on the listening conditions of that section of the audience. In the case of halls for music, involving a much wider range of wavelengths, the situation is exacerbated because low frequency sound is able to penetrate more deeply under balconies than high frequency sound. Therefore, there is an additional danger that the audience positions under a deep balcony will receive an unbalanced frequency spectrum.

A useful application of the acoustic shadow phenomenon is the construction of walls along freeways to shield adjacent communities from vehicular traffic noise. Unfortunately, much of the noise produced by trucks and fast-moving cars tends to be in the low-frequency range between 100 and 300 cps (i.e., from 11 FT down to 3.7 FT). Therefore, an effective traffic noise barrier needs to be quite high and this is a rather costly proposition. It needs to be able to resist substantial horizontal forces due to wind[2], and possibly earthquakes if located in a seismic region.

Sound is also subject to diffraction when it passes through a small opening (Figure 10.4). The proportion of the transmitted sound that passes through the opening by diffraction increases with both decreasing frequency and decreasing size of the opening. In other words, for low frequency sound and small openings almost all of the sound is transmitted by diffraction (i.e., bends around the edge of the opening). Consequently, the amount of sound that is transmitted through a small opening is always greater than what would seem possible based on the visual appearance of the opening. In technical terms, the sum total of sound passing through a small opening is made up of two components, namely: the optical zone component; and, the diffraction zone component.

10.2 Absorption of Sound

Although a small amount of sound is absorbed during its passage through air by the friction of oscillating molecules, this absorption is negligible at low frequencies (i.e., frequencies less than 1,000 cps) and quite small even at higher frequencies. In halls for speech and music the absorption of the audience is usually the largest single factor, due mainly to the absorption provided by clothing. Accordingly, the acoustic conditions in an auditorium will change significantly in relation to the number of people present and their location. This problem has been largely overcome in modern auditoria and concert halls by the use of highly absorbent seating, with each individual seat providing, ideally, the same absorption whether occupied by a person or not.

There remains then for consideration the absorption that inevitably takes place whenever sound waves strike the wall surfaces of a room. The surface on which the sound impinges may move slightly due to air pressure changes, or if it is porous, air may penetrate to some depth. In either case energy will be expended and this naturally causes a reduction of sound energy. As a result, the reflected sound wave is bound to be weaker than the incident wave, and the degree of absorption will depend on the Absorption

[2] The horizontal force is given by the formula: force = [(wind speed)^2/400]. Therefore, a wind speed of 50 mph will produce a horizontal force of more than 6 LB/SF. For a 30 FT high wall this equates to nearly 200 LB/FT.

Coefficient (α) of the surface. The latter provides a simple numerical scale that relates the incident sound energy to the sound energy absorbed by the surface. Perfect reflection and complete absorption (e.g., an open window) are rated on this scale as zero and unity, respectively. While the Absorption Coefficient varies with the angle of incidence of the sound waves, in practice it normally suffices to state the average value taken over all angles (Figure 10.5).

Construction and/or Material	125 cps	500 cps	2000 cps
Walls:			
clay bricks (glazed)	0.03	0.03	0.05
clay bricks (painted)	0.01	0.02	0.02
concrete (painted)	0.10	0.06	0.09
gypsum board (1/2") (on two sides of studs)	0.27	0.05	0.03
glass window	0.35	0.18	0.07
Floors:			
wood floor on joists	0.15	0.10	0.06
wood parquet on concrete	0.04	0.07	0.06
concrete or terrazzo	0.01	0.01	0.02
linoleum on concrete	0.02	0.03	0.03
carpet (heavy) on concrete	0.05	0.14	0.60
Miscellaneous:			
water surface (pool)	0.01	0.01	0.02
curtain fabric (10 oz.)	0.03	0.11	0.24
curtain fabric (18 oz.)	0.14	0.55	0.70
metal roof deck	0.40	0.15	0.04
wood roof deck	0.24	0.14	0.13
glass pane (1/4" or thicker)	0.15	0.04	0.02

Figure 10.5: Absorption Coefficients

If the surface area of a wall (i.e., any surface) is 80 SF and the Absorption Coefficient is 0.25 at 125 cps then the sound absorption provided by that wall is 20 sabins (i.e., 80 x 0.25) at a frequency of 125 cps.

Example: The Sound Absorption Coefficients in a large conference room measuring 20 FT (length) by 15 FT (width) by 8 FT (height) are:

Surface Material	Location	125 cps	500 cps	2000 cps
clay bricks (painted)	walls	0.01	0.02	0.02
carpet (1/4" pile)	floor	0.04	0.15	0.50
pegboard (1/4" over 4" fiberglass)	ceiling	0.80	1.00	0.38

What is the total sound absorption in the empty room?

wall area = 2(20 x 8) + 2(15 x 8) = 560 SF
wall sound absorption = 560 x 0.01 = 5.6 sabins at 125 cps
floor area = 20 x 15 = 300 SF
floor sound absorption = 300 x 0.04 = 12.0 sabins at 125 cps
ceiling area = 20 x 15 = 300 SF
ceiling sound absorption = 300 x 0.80 = 240.0 sabins at 125 cps

Total sound absorption (125 cps) = 5.6 + 12.0 + 240.0 = 257.6 sabins.
Total sound absorption (500 cps) = 11.2 + 45.0 + 300.0 = 356.2 sabins.
Total sound absorption (2000 cps) = 11.2 + 150.0 + 114.0 = 275.2 sabins.

Figure 10.6: Absorption of a wall surface

The unit of sound absorption is the *sabin* named after the American acoustician Wallace Sabine (1869-1919). As discussed earlier in this chapter, the *Sound Absorption Coefficient* is a measure of the proportion of incident sound that is absorbed by a surface expressed as a decimal number (so that perfect absorption is equal to 1.0). Figure 10.6 shows a calculated example of the total sound absorption in *sabins* provided by a conference room with acoustically *hard* (i.e., reflective) wall surfaces, a carpeted floor and an acoustic ceiling.

Two additional measures of sound absorption are commonly used:

Noise Reduction Coefficient = The average of the Sound Absorption Coefficients at the four frequencies of 250 cps, 500 cps, 1000 cps, and 2000 cps.

For example, a material with Sound Absorption Coefficients of 0.31 (at 250 cps), 0.52 (at 500 cps), 0.83 (at 1000 cps), and 0.91 (at 2000 cps), will have the following Noise Reduction Coefficient (NRC):

$$NRC = [(0.31 + 0.52 + 0.83 + 0.91) / 4] = 0.65$$

Average Absorption Coefficient = Total absorption of all surfaces in a room (i.e., area times Absorption Coefficient divided by the total surface area).

The well-known *conservation of energy* principle in physics tells us that energy cannot be simply lost. Therefore, the absorption of sound must involve the conversion of sound energy to one or more other forms of energy. Fundamentally, sound absorbing mechanisms and devices are divided into three categories: porous absorbers; panel absorbers; and, volume absorbers.

> In ***Porous Absorbers*** incident sound waves force the air in the pores of a material to vibrate producing friction, which converts sound energy into heat energy. When soft materials such as carpet or heavy fabric are exposed to sound waves the air contained in the pores of the material is set into motion, causing the transformation of some sound energy into heat energy. However, if the porous surface of the material is covered with paint or a plastic membrane then the absorption may be seriously affected and it is therefore normal practice to choose materials with a natural finish.
>
> In ***Panel Absorbers*** incident sound waves set a solid panel mounted in front of an air space into vibration. The mechanical action of the panel dampened by the air pocket absorbs the sound energy. Examples include the diaphragmatic action of airtight membranes such as paper, oilcloth and plywood panels, which are mounted at some distance from a solid wall. The incident sound waves will cause the membrane and the air in the cavity behind it to vibrate thereby dissipating sound energy. While the absorption will be a maximum whenever the vibrating panel is in resonance, it may also be increased by filling the air-cavity with a resilient material.
>
> In ***Volume Absorbers*** incident sound waves acting through a small opening set a contained volume of air into vibration. The vibration of air inside the container and the surging of the air through the small opening converts sound energy into heat energy.

10.2.1 Porous Absorbers

It follows from the table of common building materials shown in Figure 10.5 that rough, soft, porous and light materials absorb most sound, while smooth, hard, dense and heavy materials absorb least. We may therefore generalize that the structure of the material as well as the surface finish will affect the degree of absorption. In this respect a distinction can be drawn between *rigid* porous surfaces and *flexible* porous surfaces. Porous surfaces that are rigid may be characterized by the following three properties:

- Porosity, which is a measure of the amount of air-filled space in the material.
- Flow resistance, which is best described as the resistance of the material to the direct passage of air. If P_1 and P_2 are the air pressures on the two opposite sides of a material, then the flow resistance F_R is given by:

  $$F_R = [(\text{flow velocity}) \times (P_1 - P_2) / (\text{thickness of material})] \quad \ldots\ldots\ldots \quad (10.1)$$

  (Where the *flow velocity* is assumed to be in the direction of P_1 to P_2.)
- Structure factor, which provides a measure of the amount of air-filled space in sealed pores, cul-de-sac pores, or the equivalent. Accordingly, a material in which all pores run straight through from one side to the other would have a very low structure factor of around one.

Illustrations of these interrelated properties are contained in Figure 10.7, while Figure 10.8 describes some typical practical applications. Flow resistance is of considerable importance and warrants some further explanation. When sound travels through air, not only is some sound energy dissipated (i.e., absorbed), but the sound also experiences a very small characteristic impedance as it furnishes the forces required to alternately squeeze and accelerate the air molecules during its motion. Due to the general principle of conservation of energy, the energy stored during the application of these forces will be regained in the next half cycle of wave motion. If the sound waves reach the end of one medium, then part of the sound energy will continue in the second medium while the remainder is reflected back into the first medium. Depending on the compatibility of the characteristic impedances of the two media, either a large or small fraction of the total incident sound energy will continue to travel through the second medium. Naturally, if there is a large difference between the characteristic impedances of the two media, most of the energy is likely to be reflected. This explains the deleterious effect on absorption that a thin surface layer of paint will have on a porous material. The paint will tend to seal the pores thereby increasing the flow resistance of the surface with the result that more sound energy will be reflected.

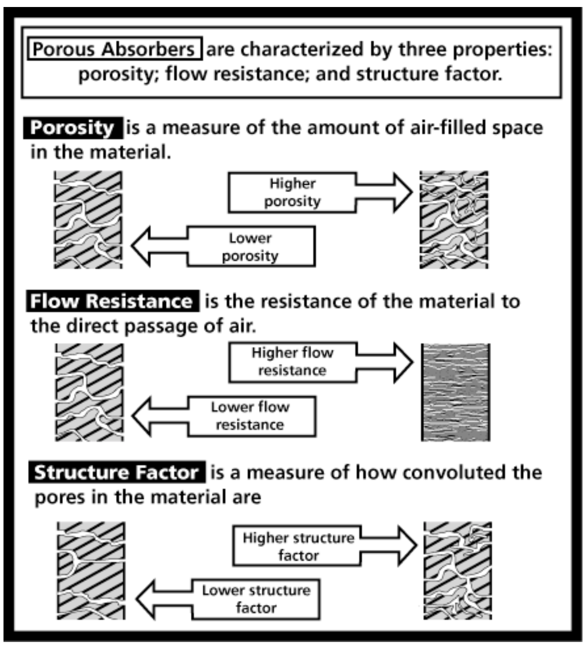

Figure 10.7: Porous absorbers in theory

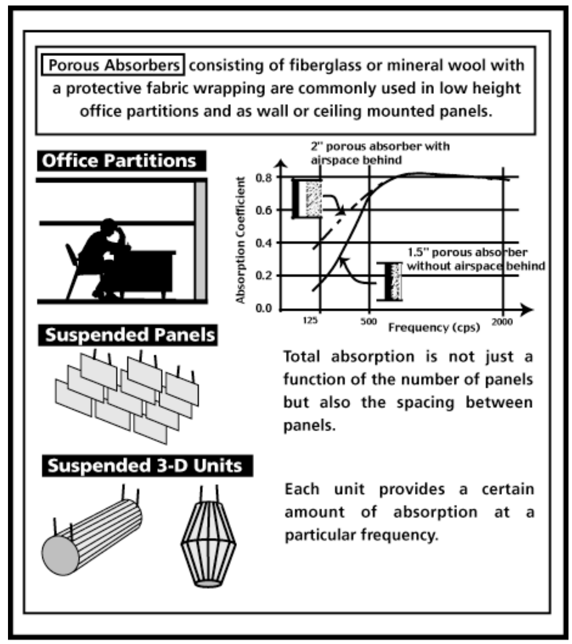

Figure 10.8: Porous absorbers in practice

Flexible porous surfaces fall into two categories. If the material has a low flow resistance (i.e., the pores penetrate through from one surface to another) it will have properties very similar to those of *rigid* porous materials, although by virtue of its flexibility the material will tend to move under the pressure of the incident sound waves. This movement will tend to increase the effective density of the porous material, as well as improve its absorptive capacity due to energy losses. On the other hand, in the case of a *flexible* porous material containing sealed pores (e.g., plastic foams) the flow resistance will be much higher and the absorption due to elastic movement is likely to dominate.

Impervious surfaces, whether rigidly mounted or not, will normally allow a fair amount of vibration under the action of sound pressures. This vibration will be a maximum when the frequency of the incident sound waves is the same as the natural frequency of the surface. Under these conditions the amplitude of the vibration will be reinforced and the amount of sound energy dissipated (i.e., absorbed) will be correspondingly large. This phenomenon is referred to as *resonance* and will be discussed in more detail in the following Section.

In summary, a good *porous absorber* is a material with many interconnected and continuous open pores. Such pores will produce a great deal of friction as the air, set into vibration by the incident sound waves, pumps back and force within the material. Accordingly:

- Only open-cell materials are good *porous absorbers*. Plastic foams such as expanded polyurethane, which are excellent thermal insulators, are poor sound absorbers due to their closed cell structure.
- Open-cell plastic foams such as melamine and polyester are good sound absorbers due to their interconnected cell structure.
- General building construction advantages of open-cell plastic foams include light weight, fiber-free composition, and moldability. Disadvantages include their combustibility and the emission of toxic fumes during a fire.
- The Absorption Coefficient of a *porous absorber* typically increases with thickness. This is mainly due to the increased volume of air that can be set into vibration, thereby facilitating the dissipation of sound energy as heat.

10.2.2 Panel Absorbers

We have mentioned previously the ability of materials to be set in motion by incident sound waves. Naturally, the resultant elastic vibration will cause a certain amount of sound energy to be dissipated. Although the velocity of sound in air is constant for all frequencies, the velocity of vibration that a sound wave may produce in a material will vary with the frequency of the sound. It follows that at some particular frequency the velocity of the sound wave in air will be identical to the velocity of the resultant vibration of the incident surface. At this critical frequency, the transfer of sound energy from air to surface is most efficient and the absorption is likely to be very high. This condition is referred to as *resonance* and for a *panel absorber* the largest amount of absorption will occur for frequencies where the vibrating panel is in resonance. In architecture, the entire structure of a high-rise building, an individual room subjected to standing waves[3], or a *panel absorber*, are all examples of situations where *resonance* can play a critical role.

The resonance frequency (f_{RES}) of a *panel absorber* is a function of the mass of the panel (LB/SF) and the depth of the air space (IN) between the panel and the construction element on which it is mounted (e.g., a wall). It can be calculated as follows:

$$\mathbf{f_{RES} = [170 / ((panel\ mass) \times (air\ space\ depth))^{1/2}]\ \ cps\ \ \ldots\ldots\ldots\ldots\ \ (10.2)}$$

Since *panel absorbers* have a relatively low Absorption Coefficient, they are rarely used in buildings to absorb noise at the source, such as in the mechanical spaces. However, they are used to considerable advantage in halls for speech and music where the amount of sound that must be absorbed is less of a concern then considerations related to sound quality and visual appearance. Since the air that is contained between the back of the panel surface and the wall acts as a damping device, much like mattress springs would if the panel were to be attached to the wall by

[3] Standing waves can occur between two parallel walls in a building space when the distance between these walls is some exact multiple of the wavelength of the ambient sound. Accordingly, the walls of rectangular rooms for music are typically slightly off-set from 90°. For a detailed treatment of standing waves see Mehta et al 1999 (pp. 383-392) and Louden 1971 (pp. 101-104).

such metal springs, it follows that the amount of absorption provided can be fine-tuned in at least two ways.

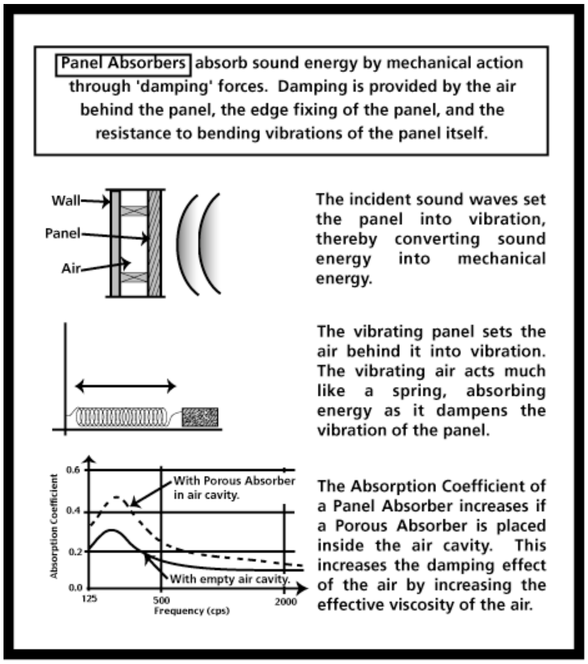

Figure 10.9: Mechanical action of a panel absorber

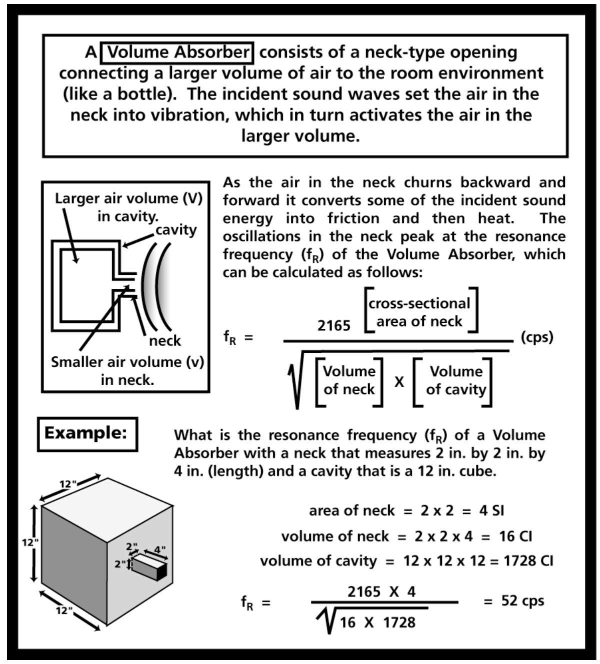

Figure 10.10: Volume absorber or Helmholtz Resonator

First, the placement of fiberglass or mineral wool in the cavity will increase the Absorption Coefficient by virtue of the increase in damping. Second, the depth of the air space behind the panel will influence the resonant frequency of a panel absorber. A relatively thin cushion of air will result in a stiffer spring action, while a thicker cushion of air will produce a more pliable spring action. Applying equation 10.2 we can see by inspection that combinations of large panel mass and deep air cavity will lead to lower resonant frequencies. However, even at resonant frequencies the Absorption Coefficient provided by a panel absorber rarely exceeds 0.3, unless it is made of a very inelastic (i.e., limp) material such as thin lead sheeting.

10.2.3 Acoustic Ceilings

The sound absorption characteristics of a typical suspended acoustical tile ceiling combine the characteristics of porous absorbers and panel absorbers. An acoustical tile, whether it has regularly spaced holes or is textured in some way, is essentially a rigid board (i.e., panel) made of a porous material. From this point of view, it has the characteristics of a porous absorber. However, since acoustical tiles are usually suspended on a structural frame below the floor above or below the roof, with a large air cavity in between, they also have the properties of a panel absorber.

Manufacturers normally quote the absorption characteristics of acoustical tiles in terms of the Noise Reduction Coefficient discussed at the beginning of Section 10.2. This is an average value since the Noise Reduction Coefficient is based on four frequencies within the range of 250 to 2000 Hz or cps (i.e., 250, 500, 1000, and 2000 cps).

10.2.4 Volume Absorbers

A further category of absorption is provided by perforated or slotted panels backed by porous materials, according to the Helmholtz principle. Absorption devices based on this principle are commonly referred to as volume absorbers. As shown in Figure 10.10, the incident sound waves are forced through a narrow opening into a larger space containing air. The latter is set into vibration if the natural period of vibration in this space is the same as the frequency of the particular sound. We might say that the air in the space has its own resonance, and when the sound wave emerges from the aperture it is forced to vibrate. This causes air to surge in and out of the aperture, resulting in the absorption of sound energy due to friction.

$$\mathbf{f_{RES} = [(2165 \times area\ of\ neck) / ((neck\ volume) \times (air\ space\ volume))^{1/2}]\ \ cps\ ...\quad (10.3)}$$

It is normal practice to further impede the movement of the vibrating air by the addition of some porous material to the neck, or by loosely filling the air space with fibrous material. This brings us to an interesting point, namely, resonators can also act as stores for energy rather than just dissipaters. This phenomenon, which occurs when the viscosity of air in the aperture is sufficiently large, has been developed by Parkin and Morgan (1965) as a means of prolonging the reverberation time for frequency groups in the Royal Festival Hall, London.

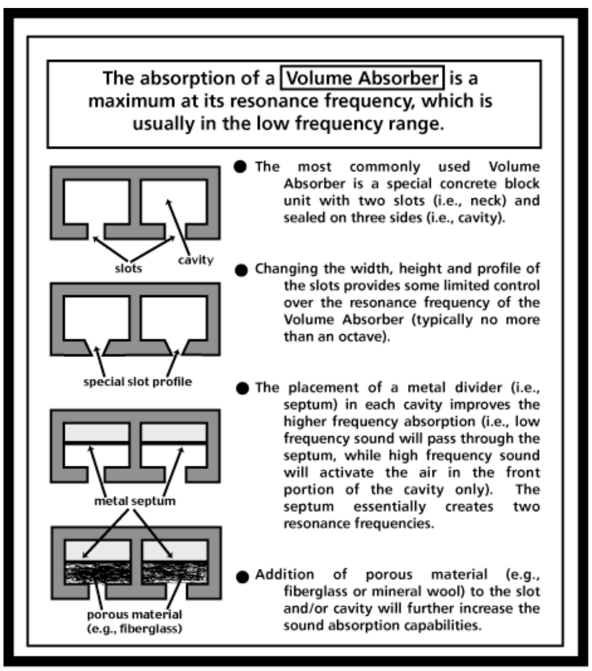

Figure 10.11: Different configurations of a concrete block absorption unit

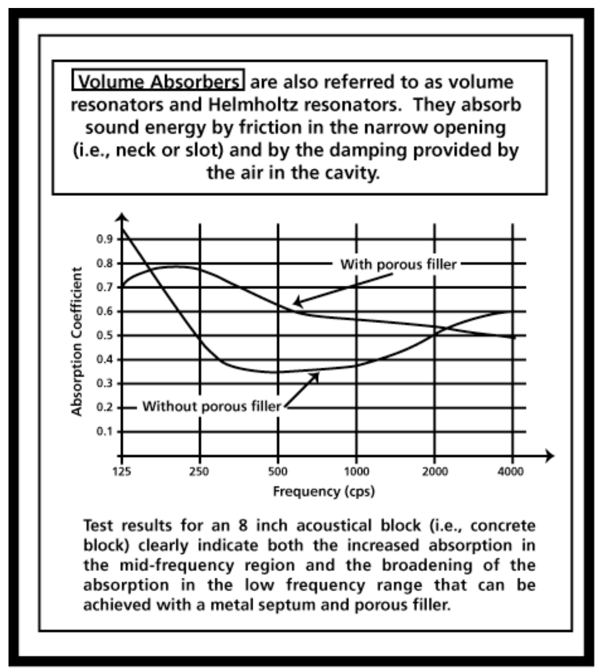

Figure 10.12: Role of resonance in a volume absorber

The most common application of volume absorbers in buildings is in the form of walls constructed with slotted hollow concrete blocks, also referred to as *acoustical blocks* (Figures 10.11 and 10.12). Due to the thickness of the wall of the concrete block the slotted opening becomes the neck of a Helmholtz resonator, while the hollow portion of the block serves as the large air space. The addition of a *septum* inside the larger air space, as shown in Figure 10.11, essentially creates two resonant frequencies. The rear cavity will absorb the lower frequency sound and the front cavity will absorb the higher frequency sound.

10.3 Speech Communication

In an enclosed space, sound originating from a source will spread out on the surface of a sphere of continually increasing diameter until some part of this surface reaches the enclosing shell, where some of the sound is absorbed or transmitted and the remaining portion is reflected according to simple geometrical rules. In quick succession, this remaining portion of the sound is inter-reflected from surface to surface within the enclosed space. During each reflection the sound energy is reduced by absorption and transmission until eventually all of the energy is dissipated. During this process of sound decay, the occupants of the space will receive some of the sound directly and some indirectly by reflection. It is logical that the listeners will hear the direct sound only once, while reflections will reach them from various directions for a period of time that is largely determined by the Absorption Coefficients of the reflecting surfaces.

Although there appears to be little change in the loudness of a sound as its direction changes, the listener can distinguish among sounds that arrive from different directions. As discussed previously, the brain is able to correlate the separate signals from each ear on a selective time delay basis. This phenomenon is an integral part of the appreciation of sound, a fact that is simply demonstrated by blocking one ear for short periods of time when exposed to music in a fairly reverberant space. Reverberation is here defined as the acoustic sensation that is produced by the slow decay of sound. Naturally, if the enclosed space is influenced by extraneous background noise, the weaker reflections will not be heard and listening may become more difficult. The listening process may also be hindered by excess reverberation within the enclosed space. If we assume that about 10 speech sounds are produced each second during normal speech, it is apparent that excessive reverberation (i.e., the prolonged inter-reflection of each speech sound) will tend to interfere with each new speech sound. Accordingly, it is very important that the listener should receive as much sound as possible within the first few milliseconds (e.g., 30 to 50 milliseconds). This has led to the prominent development of Reverberation Time into a major acoustical criterion (see Section 10.4 later).

10.3.1 Speech Interference Level

Experience has shown that unwanted sound (i.e., noisiness) rather than sound intensity (i.e., loudness) is the major cause of annoyance. Noisiness seems to increase at a greater rate than loudness, whenever the pitch of a sound is raised or the complexity of the spectrum is increased. Furthermore, the reaction to noisiness is time dependent. It has been demonstrated that annoyance levels are higher if unwanted sound persists beyond 200 milliseconds. Due to the fact that individuals vary in their reaction to noise, we are forced to assess these subjective reactions on a statistical basis.

In regard to interference with speech or music, it has been found that individual variation is not great and it is therefore possible to predict with reasonable accuracy the effect of ambient noise. However, when it is a question of annoyance, we are unfortunately faced with a wide range of responses. Nevertheless, it is highly desirable and indeed possible to make some estimate of the response of a group of persons to a particular background noise level.

The criterion of Speech Interference Level is widely used in specifying the permissible levels of background noise that will not interfere with speech communication. Background noise will increase our threshold of hearing and as a result we may be able to distinguish only a few of the sounds necessary for satisfactory speech intelligibility. The energy of the various speech sounds is distributed over a frequency range of below 100 cps to above 10,000 cps. Fortunately, a complete frequency range is not required for reliable intelligibility, and it can be shown that a high percentage of the information in speech is contained in the frequency range from 200 cps to 6,000 cps. Indeed, measurement and calculation may be simplified even further by using a three-octave band analysis. The bands normally chosen are 600-1,200 cps, 1200-2,400 cps, and 2,400-4,800 cps. The average of the sound levels in these three bands is described as the Speech Interference Level.

However, it should be pointed out that Speech Interference Levels were developed as a simplification of the more complex method of assessing articulation (ANSI 1969, 1989). Early researchers in this field measured the intelligibility ratio using syllables, words, and sentences. A speaker would constitute a source and communicate a predetermined list of sounds, while a number of listeners would attempt to write down the sounds as they heard them. The intelligibility ratio was then defined as the percentage of syllables correctly recognized by the listeners. Word and sentence intelligibility can be measured similarly, although in these cases scores are likely to be higher because of the inherent redundancies of normal speech.

10.3.2 Background Noise

The concept of Speech Interference Level (abbreviated to SIL in Tables 10.1, 10.2 and 10.5, and Figure 10.13) was briefly discussed in Section 9.7, previously. For convenience Table 9.3 is reproduced hereunder as Table 10.1. This table lists Speech Interference Levels that barely allow reliable conversation at the distances and noise levels indicated. The values listed in Table 10.1 are based on average male speakers and average listeners and will thus involve some variations due to individual differences. This variation has been estimated to be of the order of 10 dBA. While it is mainly noise in the three octave bands (i.e., 600-1200, 1200-2400, and 2400-4800 cps) that will interfere with speech, noise at lower frequencies will also interfere if it is sufficiently loud (Houtgast and Steeneken 1984, Steeneken and Houtgast 1985).

Table 10.1: Maximum background noise levels (i.e., SIL) for reliable speech communication (according to Beranek)

Distance Between Speaker & Listener	Speaker's Voice Level Normal	Raised	Loud	Shouting
1 FT	66 dB	72 dB	78 dB	84 dB
2 FT	60 dB	66 dB	72 dB	78 dB
4 FT	54 dB	60 dB	66 dB	72 dB
6 FT	50 dB	56 dB	62 dB	68 dB
12 FT	44 dB	50 dB	56 dB	62 dB
24 FT	38 dB	44 dB	50 dB	56 dB

Experience in office environments has shown that communicating persons (i.e., speakers and listeners) are keenly aware of the deleterious effect of background noise on speech intelligibility. Beranek found that a large percentage of office workers communicate *often* to *very often* and that the Speech Interference Level for continuous noise should therefore not exceed 40dB. Furthermore, office staff seem to desire a Loudness Level (LL) that does not exceed the Speech

Interference Level by more than 22 phons (Figure 10.13). It appears therefore that the objective measure of Speech Interference Level is in some cases inadequate and that the subjective LL should be also taken into account.

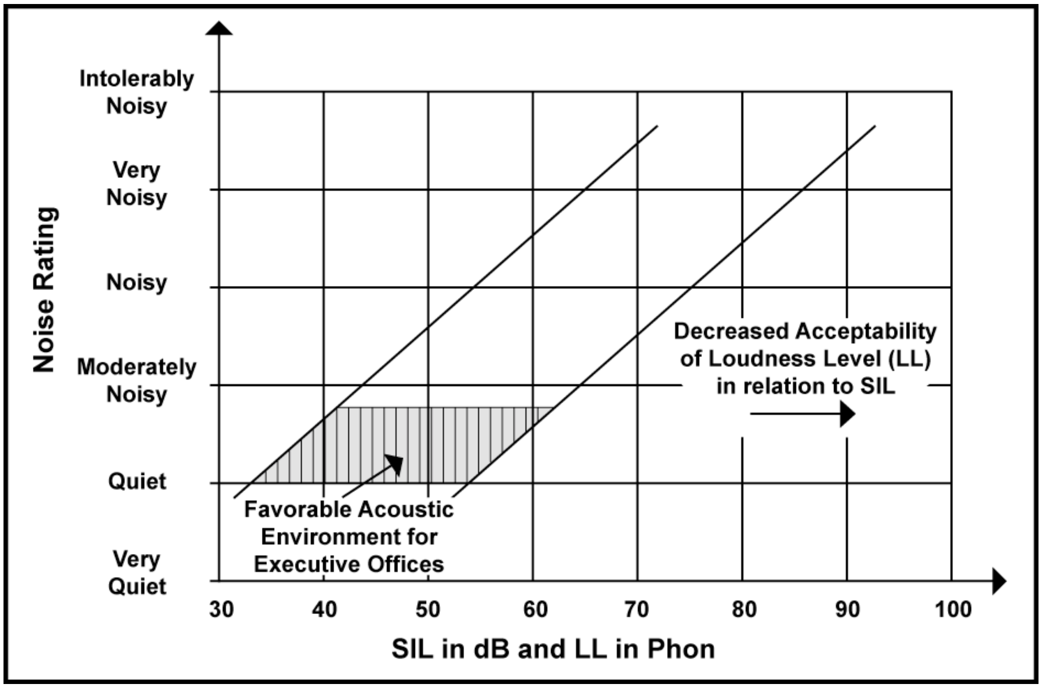

Figure 10.13: The relationship between SIL and LL (according to Beranek)

Accordingly, in 1956 Beranek proposed Noise Criteria Curves (NC or NCA[4]) as a design criterion for the specification of maximum permissible or desirable background noise levels in various occupancies (Beranek 1960). It is assumed that the background noise is nearly steady. For intermittent noise, short duration loud noise may be permitted to exceed the sound pressure levels indicated by these curves without creating undue interference with speech communication. Since this type of specification is only possible in conjunction with a complete frequency analysis of the background noise, the Noise Criteria Curves are plotted on a chart of eight octave bands. The measured octave band levels for each of the eight octaves are plotted on the chart and the corresponding NC value is noted. The noise is then rated according to the highest NC value in any band.

This led Beranek to propose the following NC ranges for the maximum background noise levels in various types of office spaces.

NC-20 to NC-30: Executive offices and large conference rooms (50 persons) that require a *very quiet* environment for clear communication at some distance.

NC-30 to NC-35: Private and semi-private offices, reception rooms, and small conference rooms (20 persons) that require a *quiet* environment to facilitate communication in a normal voice at a distance of 10 to 30 FT

[4] NC curves are based on the *linear* measure of Sound Pressure Level (SPL), while NCA curves are based on the *A weighted* scale that makes some allowance for the reduced sensitivity of the human ear at lower sound frequencies. (see Section 9.8).

(e.g., around a 15 FT table).

NC-35 to NC-40: Medium-sized offices and industrial business offices in which communication in a normal voice at a distance of 6 to 12 FT around an 8 FT table should be possible.

NC-40 to NC-50: Large engineering, banking, lobbies, and waiting rooms in which communication in a normal voice at a distance of 3 to 6 FT around a 5 FT table should be possible.

NC-50 to NC-55: Unsatisfactory for conferences of more than two or three persons since satisfactory speech communication in a normal voice is restricted to a distance of about 2 FT.

Above NC-55: A *very noisy* environment that is not recommended for any office space.

Noise measurements made for the purpose of judging the acceptability of the noise in an office environment in comparison with these suggested criteria should be made with the office in normal operation, but with no one talking at the particular work station or conference table where the noise is being measured. The equivalent background noise in an unoccupied office space should be between 5 and 10 NC units lower.

Beranek's recommendations for maximum allowable background noise levels have been generally followed to the present day, even though individual acoustic engineers may have presented their own suggestions in slightly different forms. For example, on the basis of the NCA version of the NC curves the Swiss acoustic engineer Willi Furrer (1964) recommended the noise ratings for speech intelligibility shown in Table 10.2.

Table 10.2: Maximum background noise levels (i.e., SIL) for reliable speech communication (according to Furrer)

Noise Criteria Curve (NCA)	Maximum Speech Intelligibility Distance Normal Voice	Raised Voice
40 NCA	23 FT	40 FT
45 NCA	13 FT	25 FT
50 NCA	7 FT	12 FT
55 NCA	4 FT	7 FT
60 NCA	2 FT	4 FT

The English acoustic engineers, Parkin and Humphreys (1958), proposed four criteria for permissible noise levels in rooms used for speech and music. Even though they decided to present their recommendations on the basis of octave bands, the relationship to Noise Criteria Curves and therefore the influence of Beranek is evident. These maximum recommended noise levels (Table 10.3) refer to intruding noise, assumed to be of a meaningless nature. Criterion A is fairly close to the threshold of hearing for continuous noise and applies to situations where almost complete silence is required (e.g., concert halls). Criterion B may be accepted as a compromise if Criterion A is unattainable and also applies to broadcasting studios, opera houses, and larger theaters. Criterion C applies to classrooms, music rooms, conference rooms, and assembly halls, while Criterion D refers to courtrooms and churches.

Table 10.3: Maximum background noise levels for various occupancies recommended by Parkin and Humphreys

Frequency (octave band in cps or Hz)	Criterion A	B	C	D
37 to 75 cps	53	54	57	60
75 to 150 cps	38	43	47	51
150 to 300 cps	28	35	39	43
300 to 600 cps	18	28	32	37
600 to 1200 cps	12	23	28	32
1200 to 2400 cps	11	20	25	30
2400 to 4800 cps	10	17	22	28
4800 to 9600 cps	22	22	22	27

10.3.3 Masking Sound Principles

Masking is concerned with the effect of one noise on another. It is a common experience to have one sound completely drowned out when another, louder noise occurs. For example, during the early hours of the evening the normal domestic refrigerator motor may not be heard, because of the usual background noise level occurring at that time. However, late at night when there is much less background noise, the motor noise of the same refrigerator will become relatively louder and possibly annoying. Actually, the noise level produced by the motor is the same in the two instances; however, the apparent noise level is louder at night because there is less background or masking noise present. Similarly, speech that may be perfectly intelligible in a relatively quiet sound environment will become less intelligible as the background noise level becomes louder until ultimately, complete masking will take place. While it is possible to determine directly the amount of noise reduction (i.e., attenuation) required of a particular wall separating a noisy and quiet sound environment, this applies only when there is no other noise in the receiving room. Obviously, if there is some other noise present, then the intruding noise will be partially masked. It is therefore possible to influence the Speech Interference Level of a particular acoustic environment through an artificially produced *sound blanket* (i.e., background sound).

The masking of one tone by another was described by Wegel and Lane (1924) in the early 1990s. It was observed that the masking of one pure tone by another is most apparent when the two tones are of approximately the same frequency. On the basis of their experiments dealing with monaural listening, Wegel and Lane made certain predictions as to the manner in which pure tones will mask speech communication. The typical masking effect of a pure tone at a frequency of 400 cps is shown in Figure 10.14. The number on each curve refers to the sound level above the level of the masking tone. On studying these curves, it is readily appreciated that the masking effect of a pure tone is greatest near the frequency of the total component, falling off steeply on either side. As a means of comparison, curves of masking versus frequency for masking by a narrow band are shown in Figure 10.15, based on the research of Egan and Hake (1950). Clearly, these curves are quite similar to the curves shown in Figure 10.14 apart from the elimination of the sharp dips, which are known to be caused by beats.

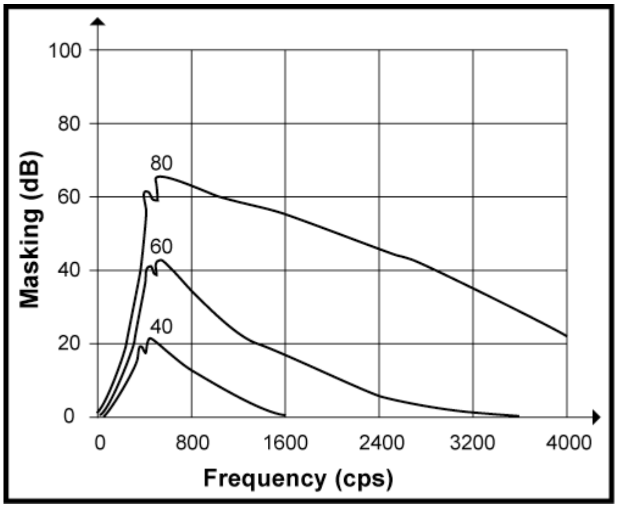

Figure 10.14: Masking effect of a pure tone

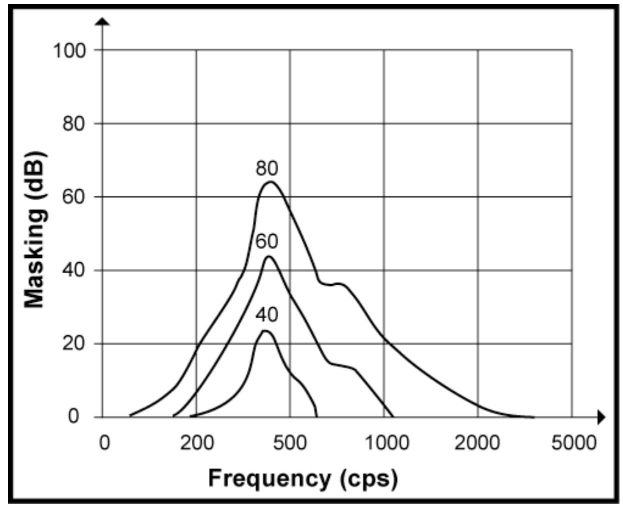

Figure 10.15: Masking effect of a narrow band

In 1959 Bryan and Parbrook (1959, 1960) published results of their own experiments on masking sound and compared these with the earlier publications of Egan and Hake (1950) and Wegel and Lane (1924). By interpolating the shift in the masked threshold for the harmonics of a masking tone of frequency 400.cps, they found some divergence between the results of these two research groups. Not only are the slopes of the masking curves different, but there is also a discrepancy of some 15 dB in the masking threshold. Bryan and Parbrook's experimental results are generally in good agreement with those of Egan and Hake.

Hempstock (1967) draws attention to the *critical band* principle, which may be applied when the masking stimulus is less than 50 dB above the threshold of audibility. The *critical band* is here defined as the width of a band of uniform noise that contains the same power as a tone at its masked threshold. By masking a pure tone with noise of increasing bandwidth, while keeping the central frequency of this noise similar to that of the pure tone, it is possible to determine variations in the degree of masking. However, a stage is reached at which the degree of masking remains unaltered with any further increase in bandwidth. Hence a critical band-width may be found for the specific frequency of the pure tone being masked.

10.3.4 Artificial Sound Blankets

Experience has shown that high level intruding noise can result in substantial physiological stress in the occupants of a building space. However, it must be recognized that the resultant stress situation can be amplified or reduced by the state of mind, motivation, occupation, and degree of familiarization of each individual person (Carr 1967, Stevens 1951). Further, it is necessary to distinguish between continuous steady-state noise, continuous variable noise, and random impulsive noise. As long as the level is not too high (e.g., below 80 dB), continuous steady-state sound appears to be able to be tolerated by most persons for prolonged periods without a negative psychological impact. For example, aircraft passengers can communicate quite comfortably with adjacent passengers in the presence of an ambient steady state noise level of 75 dBA to 80 dBA. Also, background noise levels of 55 dBA to 60 dBA may be quite acceptable in general office areas under most conditions.

The annoyance generated by intruding sound is a complicated, subjective issue. Broadly speaking it has been found that the degree of annoyance produced is more closely related to the nature of the noise and its relationship with every day experience, rather than the Loudness Level. The noise produced by a dentist's drill is a lucid illustration of this supposition.

In hot-humid climates, a serious conflict arises between thermal and acoustical comfort requirements in buildings that are not air-conditioned. Open windows of substantial size are essential for uninterrupted cross-ventilation. The intruding noise level may not be sufficiently high to completely mask speech communication and yet cause considerable reduction in the intelligibility of telephone conversations. Faced with this situation, Australian acoustic consultants Carr and Wilkinson (1967) developed a customized window unit with three specific features designed to mitigate both the thermal and acoustic environment within the building (Figure 10.16). First, double glazing reduces the intruding noise level. Second, continuous and controlled air movement is provided by a tangential fan assembly and third, the mechanical operation of the fan provides steady-state masking sound. To validate the design of their window unit, Carr and Wilkinson measured the noise level in an executive office, exposed to external traffic noise, before and after installation of the window unit (Figure 10.17). They found that the fan provided a broadband masking sound equivalent to approximately NC 40, and described the spectrum of this masking sound as nondescript and tolerable. Although extraneous traffic noise was still audible, it was not considered to be disturbing or annoying by the occupant of the office.

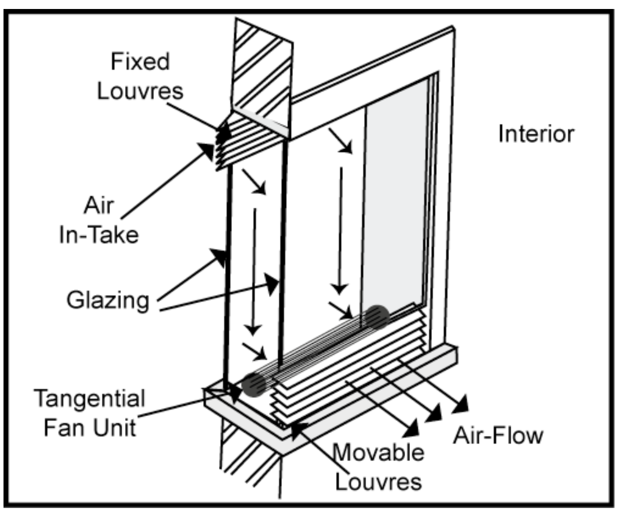

Figure 10.16: Window-fan masking unit (Carr and Wilkinson)

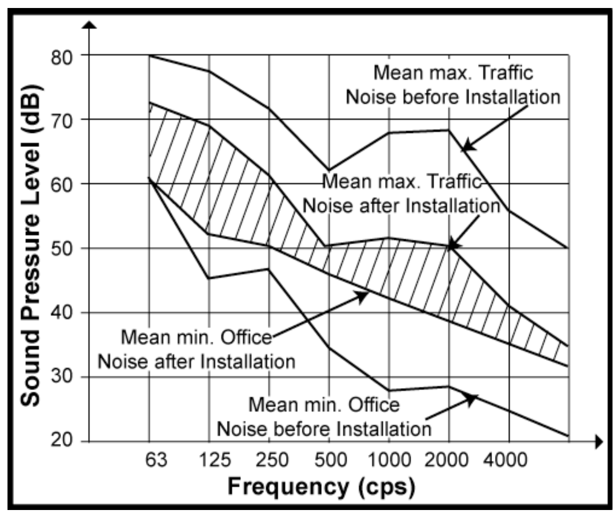

Figure 10.17: Noise conditions before and after installation of window-fan unit

From a more general point of view, artificial sound blankets in office spaces would normally have two objectives. First, to provide speech privacy in those places and at times when discussions are likely to deal with confidential business or personnel matters and second, to reduce the degree of annoyance produced by intruding noise. However, in principle, it may be argued that the introduction of additional noise into an environment that already suffers from noise pollution should be avoided. Therefore, the application of masking sound is normally reserved for situations where orthodox methods of sound insulation are likely to be very expensive or inadequate. In these situations, the sound conditioning system provides a diffuse, nondescript background sound free from any distinctly annoying tones, tuned to provide the maximum masking effect at the lowest possible volume. It is common practice to gradually increase the volume from entrance foyer to main office and provide for similar adjustments in various sections of the office space (Pickwell 1970).

10.3.5 Open-Plan School Buildings

During the 1950s there was considerable interest in the application of artificial masking sound systems to overcome inherent speech interference problems in open-plan building designs. In particular, masking sound was seen as a means of achieving an acceptable level of acoustic privacy in open-plan school buildings. We will examine this proposition in some detail.

One of the objectives of the designer of an educational building complex, such as a school, is to arrange the internal spaces in a manner that will allow the curriculum to be effectively developed and delivered. Within the total enclosed space, it will therefore be necessary to arrange a number of smaller group areas, each intended to develop a subsection of the curriculum at specific times. The most skillful division into group areas will be that which allows full use of every space for the greatest part of each day. However, because of the importance of maintaining favorable hearing conditions at all times in learning situations, acoustics has long been considered a fundamental environmental factor essential to the efficient functioning of school buildings.

Traditional practices for the design of school buildings called for the architect to separately plan the acoustical environment of classrooms individually. Under these circumstances it is possible to meet the speech privacy requirements of each classroom through sound insulation. The background noise levels in unoccupied schoolrooms can be held to a range of 35 to 40 dBA for ordinary classrooms and as low as 25 dBA for language rooms, music rooms, and special classrooms (Knudsen and Harris 1962). To achieve these low background noise levels the architect has to rely heavily on:

- ***Building layout:*** Special attention must be paid to site planning in relationship to external noise centers. Adequate siting, grading and landscaping may contribute considerably to noise attenuation. Similarly, within the school building, classrooms can be arranged in a manner that will minimize the sound insulation requirements.
- ***Noise insulation:*** By avoiding direct air-paths and applying proven noise reduction solutions such as massive wall construction, internal background noise could be held to stringent levels such as 40 dBA for speech communication rooms and 45 dBA for music rooms (Knudsen and Harris 1962, Doelle 1965).

However, these building design and construction measures are not compatible with the notion of open-plan school buildings. A notion that desires spaces to be integrated into visually undivided large units without permanent enclosures and continuing through curtains, folding doors, glazed screens, or other forms of removable partitioning. In other words, the realization of open-plan design objectives presents the architect with acoustical problems of severe complexity. Every school activity is a potential noise source. During the 1950s and 1960s a small number of open-plan school buildings were constructed in the US. Even though these schools were judged to be reasonably acceptable from a noise control point of view, such school designs have not been favored in more recent years. The reasons may be more related to a need for standard modular classroom units to meet the demands of rapid expansion within economic constraints, then the failure of open-plan concepts dues to lack of noise control.

The noise levels listed in Table 10.4 serve as a guide to the average and extreme intensity ranges that may be expected in typical school building spaces (Caudill 1954).

Table 10.4: Average and extreme noise levels generated in typical school spaces

School Area	Mean Noise Level	Extreme Noise Level
Laboratories	70 to 75 dB	85 dB
Recitation areas	60 to 65 dB	75 dB
Activity areas	65 to 70 dB	90 dB
Individual instruction	50 to 55 dB	70 dB
Band practice	70 to 75 dB	95 dB
Group singing practice	65 to 70 dB	85 dB
Indoor play areas	80 to 85 dB	95 dB
Cafeteria	75 to 80 dB	90 dB
Outdoor playgrounds	75 to 80 dB	85 dB
External street traffic	80 to 85 dB	105 dB

Movable partitions are intrinsically poor sound attenuators and any endeavor to increase their sound attenuation performance is accompanied by a substantial cost factor. Furthermore, when large spaces are subdivided into temporary cells, we must allow for pedestrian access and ventilation openings, thus creating direct air-paths and thereby nullifying a high percentage of expensive insulation treatment. The general noise levels within a school building that are listed in Table 10.4 may be broken down into three main noise groups (Knudsen and Harris 1962), as follows:

1. ***Speech noise*** characterized primarily by the three-octave frequency band ranging from 600 to 4800 cps.
2. ***Impact noise*** consisting mostly of low frequency sound generated by footsteps, foot shuffling, scraping of furniture, and dropping of objects.
3. ***Mechanical noise*** generated by mechanical and electrical services, and external vehicular traffic.

If we assume a continuous masking sound to have a spectrum typical of school activities, then in relation to the level of the masking noise the threshold of audibility will be raised by proportional amounts for each octave band. For example, suppose a noise of 70 dBA in the 600 to 1200 cps octave band is to be reduced to practical inaudibility in an adjoining room. In the absence of any masking in the second room the required reduction will be about 59 dBA to 11 dBA (where 11 dBA is the threshold of hearing). However, if a masking noise level of y dBA in the 600 to 1200 cps octave band is present in the second room, the amount by which the threshold of audibility is raised in this octave band may be calculated as x dBA. The numerical value of x is dependent on the spectrum and level. For example, if y is equal to 35 dBA, then the threshold of audibility will be raised to about:

11 dBA + 35 dBA = 46 dBA

The required sound attenuation of the wall separating the two adjoining rooms is likewise reduced to approximately 23 dBA.

In the absence of any objective measure available to determine the annoyance caused by noise, Ingerslev (among others) has suggested a criterion based on *acceptable noise levels* (Ingerslev 1952). This criterion assumes the existence of a background noise level due to occupancy. In an area where approximately 20 to 30 students are present, the background noise level (while no special activity is taking place) due to movement, breathing, and so on, will be approximately 50 dB (or 48 dBA). It is further suggested by Ingerslev that noise of a level 20 dB (or 18 dBA) below the background noise level will be inaudible if the two noise levels have the same spectrum. Accordingly, in reference to the previous example, intruding noise from an adjoining area is unlikely to cause any annoyance if its level is 30 dBA or less.

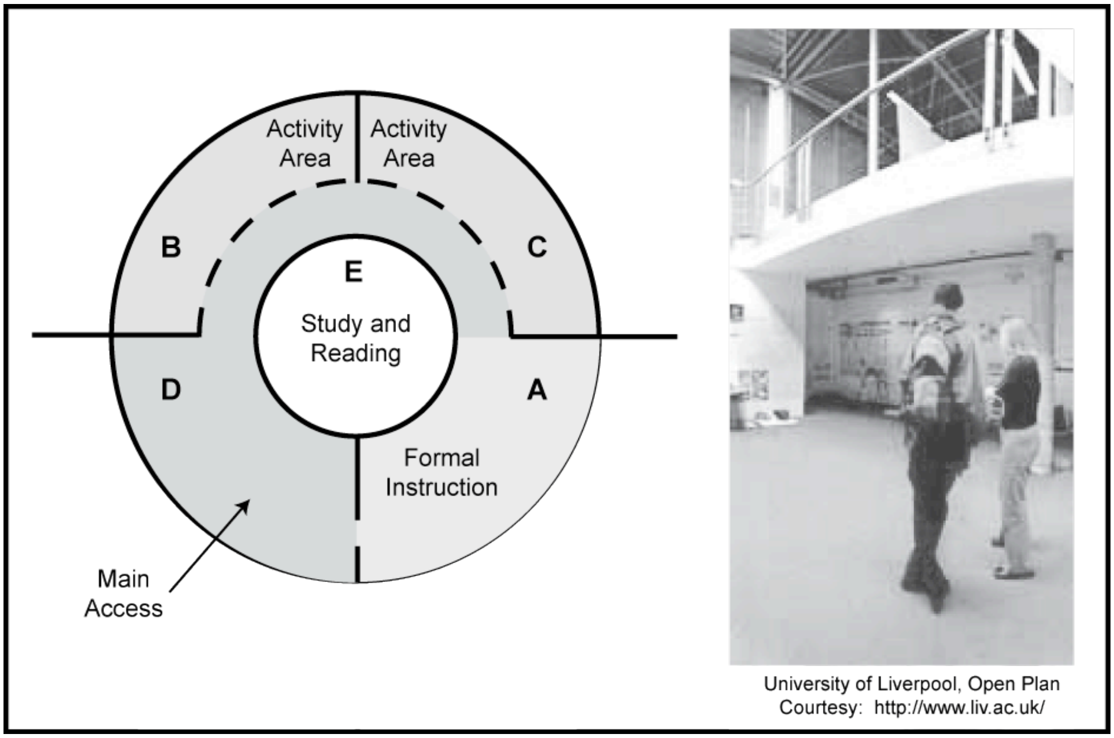

Figure 10.18: Conceptual layout of an open-plan school

These conditions may be further illustrated by reference to Figure 10.18, where one large school space is subdivided into a number of smaller group activity areas. Allowing for the various activities specified to take place in areas A, B, C, D, and E, the noise level that will intrude into anyone of these areas may be designated as x dBA. At the same time, the background noise level due to occupants seated within this same area may be designated as y dBA. Therefore, for the intruding noise level x dBA to be practically inaudible (or at least not annoying):

x dBA must be less or equal to [y dBA - 20 dBA] ………………….... (10.4)

By superimposing a masking noise level of z dBA, the overall noise level will be only slightly increased.[5] If *m* dBA is the actual masking noise level due to the combination of *y* and *z* dBA, then equation 10.4 can be rewritten in terms of *m* dBA as follows:

$$\boldsymbol{m} \textbf{ dBA} = \boldsymbol{x} \textbf{ dBA} + \textbf{20 dBA} \quad \textbf{(10.5)}$$

In equation 10.5, *x* dBA is the intruding noise level and *m* dBA is the total masking noise level due to the combined presence of the noise levels due to occupancy and masking sound. This general relationship provides a simplified basis for the determination of the superimposed (artificial) masking noise level *z* dBA, with the limitation that the total masking level *m* dBA must be of a sufficiently low level so as not to interfere with speech communication. In this respect it must be emphasized that the numerical value of *m* that is calculated with equation 10.5 is such that the intruding noise will be inaudible in relation to the total masking sound level. However, normally it would not be considered necessary to follow the stringent requirement of *inaudibility* in practice.

Table 10.5: Suggested maximum SIL values for open-plan schools employing masking sound

Maximum SIL	Communication Environment	School Area
30 dB	Very quiet room suitable for lecturing and music involving large groups of students.	Large Hall Auditorium
35 dB 40 dB	Quiet room with normal voice communication up to 30 FT possible. Satisfactory for discussions involving up to 12 students. Normal voice communication up to 12 FT possible.	Library Reading Room Interview Room Staffroom
45 dB	Normal voice communication up to 6 FT and raised voice communication up to 12 FT.	Small group instruction
50 dB	Satisfactory for discussions involving up to 6 students.	Individual study and instruction
55 dB	Unsatisfactory for discussions involving more than 4 students. Normal voice communication up to 3 FT and raised voice communication up to 6 FT.	Laboratory Art Studio

Taking into account the tolerable Speech Interference Levels listed in Table 10.1, the following maximum acceptable masking noise levels (*m* dBA) are suggested for typical instruction and study areas in schools:

Individual instruction ………………. 50 dBA
Small group instruction ……………. 45 dBA
Group activity ……………………… 50 dBA
Library and reading………………… 40 dBA

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[5] The reader will remember that SPLs in decibels cannot be added by simple arithmetic. The addition of two equal SPLs produces an increase in SPL of approximately 3 dB.

These levels are considerably in excess of the background noise levels recommended for orthodox classrooms in non-open-plan school buildings (e.g., 25 dB by Doelle (1965), 35 dB by Knudsen and Harris (1962), and 30 dB by Beranek (1962). In defense of this apparent diversion from accepted standards, at least two pertinent arguments may be presented. First, the standards stated by these authors are related to non-open-plan principles of school design, where requirements and methods of sound control follow a very different pattern. Second, studies conducted during the past 50 years have generally included the suggestion that there are three basic teaching spaces, namely: large group instruction; small group instruction; and, individual study. It can be argued that the acoustical environment required in each of these spaces is determined by the number of students involved and the activities being performed. In an endeavor to be more precise on this issue, Table 10.5 provides a more detailed interpretation of these arguments.

The presence of an artificially produced sound blanket will no doubt substantially decrease the sphere of interaction of sound radiated from two or more noise sources. Yet it is equally true that if the masking noise level is too high then the whole purpose of its presence will be defeated, since it is likely to cause annoyance of its own accord. Noise reduction at the source will allow the total masking noise level to be kept down to a minimum. In the case of school buildings designed according to open-plan principles, where heavy partitions are not feasible in most locations, noise reduction at the source will be limited to the treatment of walls, ceilings and floors with sound absorbing material.

10.4 Halls for Speech and Music

Design of larger halls (i.e., auditoria) for speech and music is a complex undertaking. It requires the designer to balance several factors that influence the clarity of speech and the quality of music, respectively. Both the direct sound and the sound that reaches the audience after it has been reflected from the enclosing surfaces of the hall need to be considered. However, the reflected sound must reach the listener within a critical time window, otherwise it may interfere with the next direct sound instead of reinforcing the previous direct sound from which the reflections originated.

Sound travels about 55 FT in 0.05 sec, a time interval that the human ear can barely detect. Therefore, for perfect sound reinforcement the first reflection or echo should reach the listener within 0.05 sec of the direct sound. Also, all subsequent reflections of the same direct sound should be sufficiently weak (i.e., much lower SPL than the direct sound) so as not to be audible above the next direct sound. This time-based relationship between a direct sound and its first reinforcing reflection can be expressed in a simple geometric equation that governs the dimensions of a hall in which the communicating sound is not reinforced by electronic means (see also Figure 10.21 later). If z is the direct distance (FT) between the sound source and the listener, x is the distance (FT) between the sound source and the reflecting surface, and y is the distance (FT) between the reflecting surface and the listener, then:

$$x + y - z \leq 55 \text{ (FT)} \qquad (10.6)$$

The limit suggested by equation 10.6 may be extended in practice depending on the loss of intelligibility that might be tolerated or even desirable. For example, in the case of a church where speech is likely to be slow and well-articulated this upper limit could be extended to 60

FT. Also, the blending of one sound into the next sound may be called for in the case of romantic music.

To obtain good sound reinforcement for speech communication, reflections should preferably come from two directions so that any directional effect is either eliminated or very much reduced. Furthermore, if wall or ceiling mounted reflectors are used to provide this reinforcement it is important that they should be of large enough dimensions. As discussed previously in Section 10.1, to be effective the length and width dimensions of the reflecting surface should be five times the wave length of the sound. In the case of speech this would suggest that the walls of the hall should be at least 11 FT high (i.e., the wavelength for a sound frequency of 500 cps is 2.2 FT, and 5 times 2.2 FT is equal to 11 FT).

10.4.1 Audience Absorption

Sound is absorbed not only by any surface in its path, but also by any medium through which it travels. As discussed in Chapter 9, when sound travels through air it sets the air molecules into harmonic motion very much like a pendulum. The propagation of the sound wave through the air medium is then made possible by the progressive movement of these molecules as they bump into the next molecules in line. Naturally, this uses up some energy. As shown in Figure 10.19, the amount of energy dissipated is very small, particularly at frequencies below 2,000 cps and above 10,000 cps under normal relative humidity conditions (i.e., above 40%).

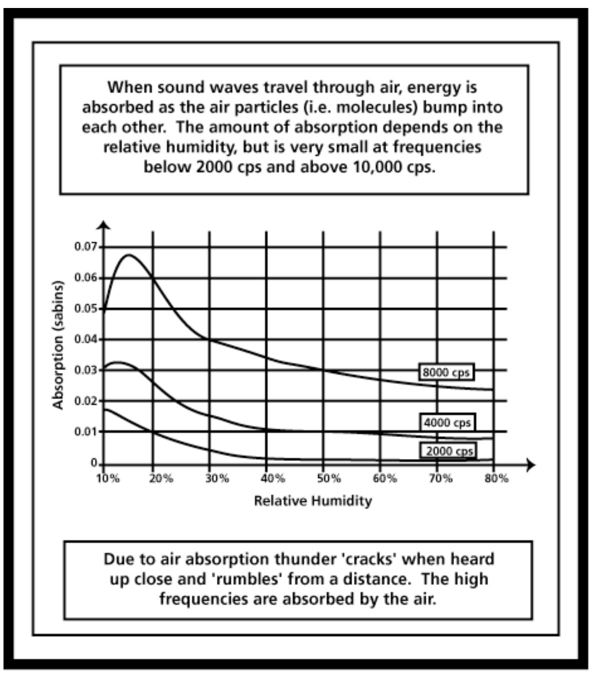


Figure 10.19: Sound absorption in air

Due to the many factors involved, audience Absorption Coefficients apply only to average conditions.

Type of Seats (occupied and unoccupied)	Frequency (cps)					
	125	250	500	1000	2000	4000
Occupied Seats:						
heavily upholstered	0.76	0.83	0.88	0.91	0.91	0.89
medium upholstered	0.68	0.75	0.82	0.85	0.86	0.86
lightly upholstered	0.56	0.68	0.79	0.83	0.86	0.86
Empty Seats:						
heavily upholstered	0.72	0.79	0.83	0.84	0.83	0.79
medium upholstered	0.56	0.64	0.70	0.72	0.68	0.62
lightly upholstered	0.35	0.45	0.57	0.61	0.59	0.55

Approximate Absorption Coefficients (per SF of seating)

Figure 10.20: Absorption due to audience (according to Beranek 1996)

By far the largest contributor to absorption in an auditorium is the audience (i.e., as much as 75% for concert halls). This stands to reason when we consider that the audience represents an appreciable portion of the exposed surface area of a hall (i.e., usually close to 25%) and the clothing worn by each person is composed of soft, semi-porous material with multiple folds and creases. Two alternative methods are commonly used to quantify audience absorption: per audience seat; or, per square foot of floor area covered by the audience. The area method is normally preferred. Using this method, the sound absorption at a particular frequency is calculated by multiplying the applicable Absorption Coefficient by the area occupied by the audience seats (Figure 10.20).

Several factors influence audience absorption, including: the type of upholstery; the kind of clothing worn by the audience; and, the slope (if any) of the floor. Experiments have shown that at shallow angles of incidence (i.e., when the sound waves graze an absorbing surface such as clothing) the absorption is higher. Accordingly, the effectiveness of sound absorption decreases as the angle of incidence of the sound waves approaches 90° and therefore the audience absorption is higher in a hall with a relatively flat floor.

10.4.2 Psycho-Acoustic Considerations

The establishment of acoustical criteria arose basically out of the need to relate the physical measurement of sound to the subjective perception of sound, in an effort to protect persons from noises that are potentially harmful or annoying and optimize the enjoyment of sounds (e.g., music) produced for entertainment purposes.

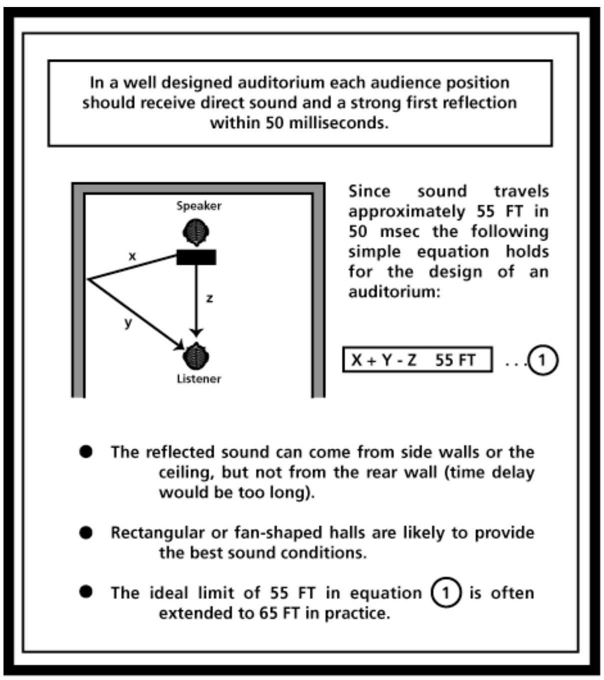


Figure 10.21: Auditorium size limits

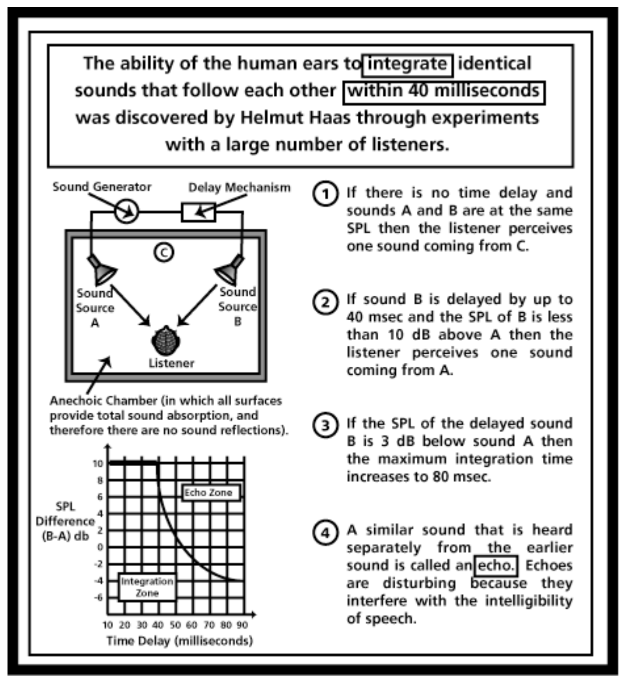


Figure 10.22: The Haas Effect

The acoustic performance of an auditorium, particularly one used primarily for musical entertainment such as a concert hall, is naturally a function of the auditory perception and mental processes of the individual members of the audience. These aspects fall under the subject matter area of psycho-acoustics and include the following:

1. The average adult under 40 years of age is conscious of sounds ranging in frequency from about 30 cps (i.e., wavelength of 37 FT) to 15,000 cps (i.e., wavelength of 1 IN), if the sound pressure level is sufficiently high. Persons above this age tend to be less sensitive to higher frequencies (i.e., the SPL may need to be raised by 10 dB or more

for the higher frequencies to be heard).

2. The change in frequency of a pure tone that can be barely detected by a person will depend on the frequency of the tone. At low frequencies smaller changes in frequency can be detected (i.e., about 0.3%), while below 20 dB the ear loses its ability to detect changes in frequency altogether.
3. Similarly, the ability to detect a change in loudness depends on both the frequency and SPL. The maximum sensitivity to a change in sound pressure level occurs at a frequency of around 4,000 cps, when a change of only 0.5 dB can be detected.
4. If two similar sounds from different sources are separated in time by less than 35 milliseconds, they will appear to the listener to be coming from the source of the first sound. This is known as the Haas effect (Figure 10.22).
5. If a sound is followed by an identical sound more than 50 msec later intelligibility is reduced. This would correspond to a difference in path length between a direct and a reflected sound of more than 50 FT.
6. When a sound undergoes a continuous change for more than 0.8 sec a listener will find it difficult to ascertain the precise nature of the change.
7. Tones that are of very short duration (i.e., less than 10 msec) are perceived simply as clicks without any apparent pitch[6]. On the other hand, if a note lasts longer than 100 msec there tends to be no improvement in pitch quality.
8. After 0.5 sec a tone attains a maximum loudness. Beyond this point time fatigue occurs, with the result that the tone appears less loud.
9. It appears that a tone becomes less distinct after a duration of 0.15 sec. At the same time SPLs in excess of 90 dB tend to overload the ear and produce distortion.

10.4.3 The Concept of Reverberation Time

In a previous section we discussed at length the ability of surfaces to absorb and reflect incident sound. It is logical that in a room, successive reflections from the bounding surfaces will continuously reduce the sound energy until the sound can no longer be heard. Toward the end of the 19th Century Wallace Sabine established a measure of the rate of decay of sound in a finite room. This measure, which is referred to as *Reverberation Time*, relates the total absorption in the room (A sabins) to the volume of the space (V CF) for a sound decay of 60 dB in the following formula:

Reverberation Time (RT) = [(k x V) / A] (sec) ……………………. (10.7)

Therefore, if a sound of 90 dB is created in a previously quiet room, then the time taken for this sound to die down to 30 dB after the source has ceased is the Reverberation Time of that room. In equation 10.7:

k = a constant (0.05 if volume *V* and area *A* are in CF and FT-sabin respectively; or, 0.16 if *V* and *A* are in metric units (i.e., cm and m-sabin)).

[6] Pitch is the subjectively perceived frequency of a sound. The actual objectively measured frequency may differ due to overtones in the sound.

A = the total absorption in sabins, which is found by multiplying each individual area by its Absorption Coefficient and summating these into one numerical value:

$$A = \Sigma(\alpha \times a)$$

where: α are the Absorption Coefficients corresponding to the individual surface areas a.

Reverberation Time provides a measure of the liveliness or noise-sensitivity of a room. A correct Reverberation Time will ensure that sounds do not persist in a room to an extent that would interfere with intelligibility. For a given room-volume a short Reverberation Time corresponds to high absorption power, while a long Reverberation Time corresponds to small (total) absorption power. Since the total intensity of the reverberant sound produced by a continuous source is inversely proportional to the total absorption, a 50% reduction in Reverberation Time will require a doubling of the total absorption power and hence the reverberant sound level will be decreased by 3 dB.

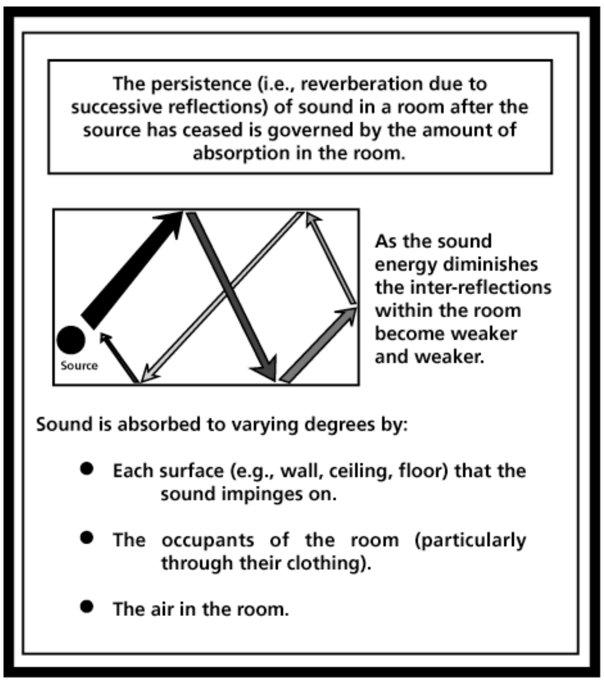
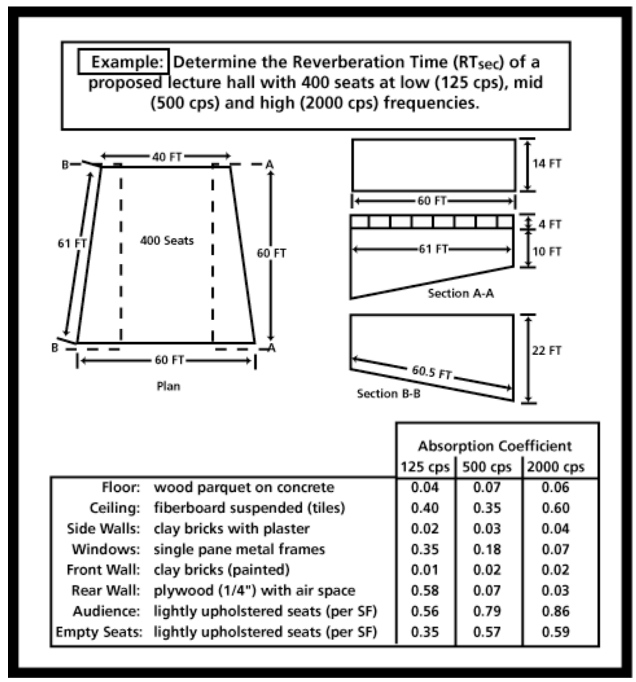


Figure 10.23: Impact of absorption on reverberation in a hall



		Absorption Coefficient		
		125 cps	500 cps	2000 cps
Floor:	wood parquet on concrete	0.04	0.07	0.06
Ceiling:	fiberboard suspended (tiles)	0.40	0.35	0.60
Side Walls:	clay bricks with plaster	0.02	0.03	0.04
Windows:	single pane metal frames	0.35	0.18	0.07
Front Wall:	clay bricks (painted)	0.01	0.02	0.02
Rear Wall:	plywood (1/4") with air space	0.58	0.07	0.03
Audience:	lightly upholstered seats (per SF)	0.56	0.79	0.86
Empty Seats:	lightly upholstered seats (per SF)	0.35	0.57	0.59

Figure 10.24: Reverberation Time calculation example

During the 20th Century, Reverberation Time emerged as an important acoustic criterion not only for speech communication but also for concert halls. Although there has been some disagreement regarding the relative significance of auditorium size and type of performance in determining an optimum Reverberation Time in any given situation, it is still generally agreed that Reverberation Time is a critical parameter in the design of halls for speech and music.

Example: Calculate the Reverberation Time of a proposed lecture room with the surface finishes stipulated in Figure 10.24. We begin by calculating the approximate area of each surface that has a different finish and therefore also a different Absorption Coefficient:

$$\mathbf{a}_{floor} = [60 \text{ x } (40 + 60)/2] = 3000 \text{ SF}$$

$$\mathbf{a}_{ceiling} = [60.5 \text{ x } (40 + 60)/2] = 3025 \text{ SF}$$

$$\mathbf{a}_{side\ walls} = [2 \text{ x } (61 \text{ x } (14 + 22)/2)] = 2196 \text{ SF}$$

$$\mathbf{a}_{windows} = [61 \text{ x } 4] = 244 \text{ SF}$$

$$\mathbf{a}_{front\ wall} = [40 \text{ x } 22] = 880 \text{ SF}$$

$$\mathbf{a}_{rear\ wall} = [60 \text{ x } 14] = 840 \text{ SF}$$

$$\mathbf{a}_{audience} = 2100 \text{ SF (assume 70\% of floor area)}$$

To calculate the total absorption in the lecture hall for any given frequency we simply multiply the area of each surface by the corresponding Absorption Coefficient. Therefore, at a frequency of 125 cps:

$$\begin{aligned}\mathbf{A_{125}} &= [(3000 \text{ x } 0.04) + (3025 \text{ x } 0.40) + (2196 \text{ x } 0.02) + (244 \text{ x } 0.35) + (880 \text{ x } 0.01) + \\ &\quad (840 \text{ x } 0.58) + (2100 \text{ x } 0.56)] \\ &= 3131 \text{ (FT-sabin)}\end{aligned}$$

Similarly, the total absorption for frequencies of 500 cps and 2000 cps can be calculated as follows:

$$\begin{aligned}\mathbf{A_{500}} &= [(3000 \text{ x } 0.07) + (3025 \text{ x } 0.35) + (2196 \text{ x } 0.03) + (244 \text{ x } 0.18) + (880 \text{ x } 0.02) + \\ &\quad (840 \text{ x } 0.07) + (2100 \text{ x } 0.79)] \\ &= 3115 \text{ (FT-sabin)}\end{aligned}$$

$$\begin{aligned}\mathbf{A_{2000}} &= [(3000 \text{ x } 0.06) + (3025 \text{ x } 0.60) + (2196 \text{ x } 0.04) + (244 \text{ x } 0.07) + (880 \text{ x } 0.02) + \\ &\quad (840 \text{ x } 0.03) + (2100 \text{ x } 0.86)] \\ &= 3949 \text{ (FT-sabin)}\end{aligned}$$

Next, it is necessary to calculate the volume of the lecture hall:

$$\mathbf{V} = [(40 + 60)/2 \text{ x } (14 + 22)/2 \text{ x } 60] = 54000 \text{ CF}$$

It is now possible to determine the actual Reverberation Time (RT) for each of the three frequencies by substituting the appropriate values for total absorption in equation 10.7:

$$\mathbf{RT_{125}} = [(0.05 \text{ x } 54000) / 3131] = 0.86 \text{ sec}$$

$$\mathbf{RT_{500}} = [(0.05 \text{ x } 54000) / 3115] = 0.87 \text{ sec}$$

$$\mathbf{RT_{2000}} = [(0.05 \text{ x } 54000) / 3949] = 0.68 \text{ sec}$$

The optimum reverberation time is based on the volume and function of the hall. In this case, around 0.8 sec should ensure adequate speech intelligibility. The average Reverberation Time for the three frequencies should be within 10% of the optimum value (in this case the average Reverberation Time is 0.80 sec). A slightly higher Reverberation Time for the 125 cps frequency band is desirable.

If the absorption of the floor is relatively small then the shading due to the audience may be conveniently neglected. Otherwise, the absorption of the floor surface may be reduced by 40% at 500 cps. It must be noted that the Reverberation Times calculated above all assume a capacity audience. A closer examination of the Absorption Coefficients listed in Figure 10.24 for occupied and empty seats shows that these vary considerably. As might be expected they are consistently lower for the empty seats (i.e., 62% lower at 125 cps, 72% at 500 cps, and 69% at 2000 cps). For the sake of comparison, the Reverberation Times of the half empty lecture hall are calculated to be:

$$\mathbf{RT_{125}} = [(0.05 \text{ x } 54000) / 2921 = 0.92 \text{ sec}$$

$$\mathbf{RT_{500}} = [(0.05 \text{ x } 54000) / 2884] = 0.94 \text{ sec}$$

$$\mathbf{RT_{2000}} = [(0.05 \text{ x } 54000) / 3676] = 0.73 \text{ sec}$$

Furrer (1964) has demonstrated the significance of audience absorption by means of acoustical tests conducted in the Musiksaal at Basel. From the graphs shown in Figure 10.25 it appears that prior to renovation, at a frequency of 1000 cps, the reverberation time of the empty hall with wooden seats was some three times longer than when fully occupied. At the same time, the provision of upholstery produced a significantly shorter reverberation time for the empty hall. The dependence of reverberation time on audience absorption suggests a need to allow for variable audiences. This may be at least partly achieved by ensuring that each individual unoccupied seat provides approximately the same absorption as a seated person. The desired effect may be further increased by perforating the underside of the seats (i.e., the underside would provide maximum absorption when the seat is empty, if the seats are collapsible).

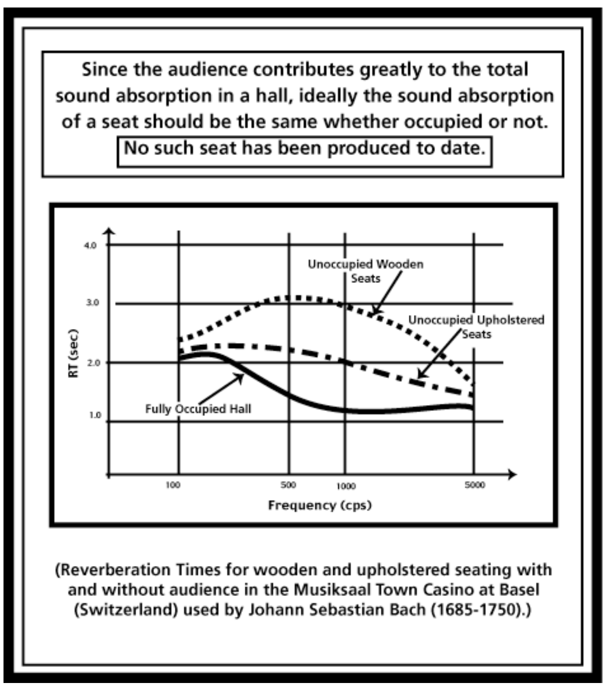


Figure 10.25: Impact of absorption on Reverberation Time

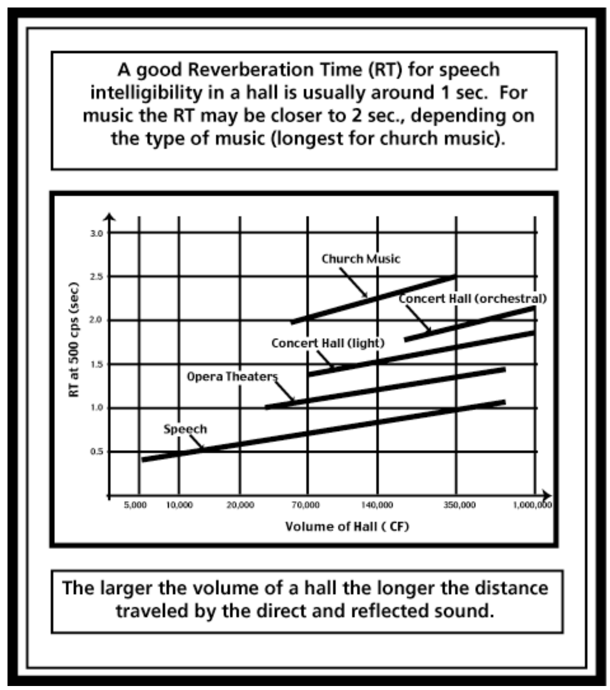


Figure 10.26: Reverberation Times for different kinds of halls

Strictly speaking, the concept of Reverberation Time proposed by Sabine is based on assumptions that are unlikely to be achieved in practice. These assumptions include: uniform intensity of sound throughout the hall; and, equally absorbent surfaces and reflections at all angles. Nevertheless, the agreement between theory and practice has been generally considered to be satisfactory.

10.4.4 Concert Halls

For the evaluation of musical perception, it is of particular interest to refer to existing concert halls for at least two reasons. First, since there are a substantial number of halls well-known for their superior acoustics, and second, because some halls are known to have played an important part in the history of music. In the past there have been strong ties between individual composers and particular halls. The relationship that is known to have existed between Johann Sebastian Bach (1685-1750) and St. Thomas Church in Leipzig (Germany) is but one example.

According to surveys the Reverberation Time of concert halls that are known to have good acoustics is close to 1.8 sec (Parkin et al. 1952, Beranek 1966 and 1962). For example:

Beethoven Halle, Bonn (Germany) ……………….	1.8 sec
Symphony Hall, Boston (US) ………………………	1.8 sec
Musiksaal, Basle (Switzerland) …………………….	1.7 sec

On the basis of systematic investigations, it has been generally agreed that the optimum reverberation time for average and large halls (i.e., 70,000 CF to 485,000 CF) is independent of the volume. However, this does not invalidate the notion that large halls require a slightly longer Reverberation Time Figure 10.26). At the same time Kuhl (1963) has suggested optimum reverberation times of 1.5 sec for classic and modern music (e.g., Mozart and Stravinsky) and 2.1 sec for romantic music (e.g., Brahms).

Discussions among acoustic engineers questioning the adequacy of Reverberation Time as the principal design criterion for concert halls appear to have started in the 1930s. In an effort to define acoustical conditions more precisely further criteria were proposed by Wente (1935), Mason and Moir (1941), Somerville (1951), Meyer (1954), Somerville and Head (1957), and Nickson and Muncey (1964). Since then, it has become common practice for acoustic consultants to take into account a small number of criteria in the design of concert halls. These now generally include the following:

- The correct Reverberation Time for each frequency. Ideally, all frequencies should decay at the same exponential rate. However, since the lower frequencies appear to be softer, it has been proposed that a longer Reverberation Time is permissible at the lower frequencies (i.e., 15% at 250 cps and 30% at 125 cps). Nevertheless, there must be no sudden change in slope of the decay curve.
- Speech requires a relatively short Reverberation Time from 0.5 sec for small rooms to 1.0 sec for larger halls. Music requires a longer Reverberation Time ranging from 1.6 sec to over 2 sec for very large halls (Figure 10.26). For music the *Bass Ratio* should be between 1.1 and 1.25 for longer Reverberation Times (i.e., greater than 1.8 sec) and between 1.1 and 1.45 for lower Reverberation Times (i.e., less than 1.8 sec)[7].
- Walls, ceiling and overhead reflectors (if any) are required to be in such a geometrical arrangement that each audience position receives one substantial echo within 35 milliseconds of the direct sound. All reflections that arrive at the listener within this period are often referred to as the *first reflection*.

---

[7] The Bass Ratio is a ratio of Reverberation Times (RT) as follows: $(RT_{125} + RT_{250}) / (RT_{500} + RT_{1000})$.

- Avoidance of sound reflecting, concave surfaces that could lead to sound concentrations or undesirable echoes.
- Exclusion of all external noise by adequate insulation. This may require rather expensive systems of discontinuous construction. If a concert hall is located close to an airport or directly below a regular flight path, sound attenuation in excess of 70 dB may be required for all external walls. Double and triple wall and ceiling constructions are common practice for broadcasting and television studios (Kuhl and Kath 1963). At the same time care must be taken to avoid ventilation noise, because the amount of low frequency sound generated by air flow within ducts increases exponentially with increasing air speed[8]. Accordingly, it is desirable to control the speed of air in the ventilation ducts before consideration is given to the insertion of noise attenuators in the duct network.
- The acoustical performance of a concert hall should be adjustable to allow some degree of fine tuning for variable audience size and type of music.

Unfortunately, there exists no definable or measurable criterion for the intelligibility of music. The matter is further complicated by the fact that we are concerned with the subjective assessment of not just one, but three groups of persons, namely: the audience; the conductor; and, the musicians. It is normal for the listener in the audience to attribute any minor acoustical faults to the musicians or the conductor, and not the auditorium. Moreover, the listener is often conditioned to stereophonic sound reproduction and expects to be treated to the same or better quality in the concert hall. While compact discs usually provide a crisp and clear reproduction, the concert hall will tend to warmth and richness. This may not always be appreciated.

Table 10.6: Framework for rating concert halls (according to Beranek 1962)

Desirable Aspect or Quality	Maximum Score	Optimum Feature to Score Maximum Points
Intimacy	40	One reflection within 20 milliseconds.
Liveliness	15	Correct Reverberation Time.
Warmth	15	Correct Reverberation Time at low and middle frequencies.
Loudness of direct sound	10	No more than 60 FT from conductor to listener in mid-hall.
Loudness of reflected sound	6	Reverberation Time is approximately equal to $3 \times 10^{-6} \times volume$ (sec).
Diffusion	4	Coffered ceiling.
Balance and blend	6	Correct orchestra layout.
Ensemble	4	A performer can hear all others.

[8] It can be shown that the acoustic power of low frequency noise varies as the sixth power of the air velocity.

Beranek (1962) has suggested a method of rating the acoustics of concert halls. Each hall is given a rating on a scale subdivided into 100 arbitrary units and incorporating eight desirable aspects (Table 10.6). These qualities are based primarily on Reverberation Time in relationship to the type of music to be played. Adjustments were necessary for opera, where the required speech clarity leads to a somewhat shorter Reverberation Time. Beranek used this method to rate the acoustics of a considerable number of halls in several countries, taking into account the opinions of musicians, conductors, and knowledgeable members of the audience. The following elaboration in respect to some of the criteria in Table 10.6 may be helpful:

> ***Diffusion:*** Sound diffusion is preferable to sound absorption for side walls. Good diffusing surfaces are rough surfaces with many recesses. However, the roughness must be in the order of magnitude of the wavelengths to be reflected (i.e., to be diffused). Therefore, recesses of 3 FT to 10 FT will be required for music. Rear walls could be corrugated if they are concave. Also, absorption treatment is commonly applied to rear walls because reflections from rear walls will arrive too late for useful sound reinforcement.
>
> ***Surface material:*** Wood paneling is a poor reflector of sound at low frequencies; however, plaster and gypsum board is fine if thick enough (i.e., two or more layers of ½ IN sheeting). Where absorption treatment is applied, the absorbing material must be effective over a wide range of frequencies.
>
> ***Orchestra layout:*** Ideally, all instruments and singers should be grouped within a 20 FT radius to avoid *time delay* problems. In churches the choir should be in one group and somewhat elevated in respect to the congregation. The ceiling height above the choir should be at least 20 FT to avoid *reflection* problems. To the conductor it is of importance that the musicians should be able to play their instruments without any special effort. Ideally, the conductor should perceive the music on identical terms with the audience. It is essential that the musicians should not be disturbed by any extraneous or audience noise. At the same time the musicians would like to hear the orchestra and themselves as a balanced unit.
>
> ***Dimensions:*** As discussed previously, the size of an auditorium without electronic sound reinforcement is governed by the requirement that every audience position should receive a strong first *reflection* within 50 milliseconds of the direct sound (i.e., $x + y - z \leq 65$ *FT*, and ideally 55 FT). The volume per seat should be as small as practical (i.e., 100 to 130 CF per seat) and the ceiling height should be ⅓ to ⅔ of the width dimension (i.e., ⅔ for smaller halls).

In regard to the most desirable physical shape of a concert hall there are basically three shapes that come under consideration, namely: rectangular; fan-shape; and; horseshoe. Of these, the rectangular shape seems to hold the most potential. The main disadvantage of the fan-shaped hall is that the rear wall and balcony front are normally curved, thereby producing a serious risk of echoes. The rectangular hall is free from this risk and in addition has the advantage of strong cross-reflection between parallel walls, which may lead to added *fullness* of sound.

It is an essential property of a good concert hall ceiling that it should produce an equal distribution of sound. There are therefore serious objections to sharply curved shells. The Sydney Opera House is a prominent example of the problems that can arise when a particular roof shape is selected for other than acoustical reasons.

10.4.5 The Sydney Opera House

The Sydney Opera House, designed by the Danish architect Jørn Utzon, is often described as one of the eight *wonders of the world*. This description is based more on its external architectural expression than its acoustical excellence, although the performance of its principal concert hall has received the approval of such renowned acoustic experts as Beranek (Beranek 2004). Located on the Bennelong peninsula, which juts out into Sydney's picturesque harbor, the gleaming concrete shells of the Opera House present a memorable view that has become an international icon for both the city of Sydney and Australia as a whole (Figure 10.27).

Figure 10.27: Sydney Opera House

Today the Sydney Opera House complex consists of five halls, as follows:

Concert Hall - with 2,679 seats is the principal auditorium for concerts and contains the world's largest mechanical tracker action organ with over 10,000 pipes.

Opera Theater - with 1,547 seats, is the main auditorium for opera and ballet performances.

Drama Theater - with 544 seats, was not part of Utzon's original design.

Studio Theater - with 364 seats, was not part of Utzon's original design

Playhouse - with 398 seats, was added long after the completion of the original construction in 1999.

What makes this imposing building complex particularly interesting from a design point of view is the apparent conflict between its exciting structural solution and the requirements of good acoustic performance that have been discussed previously in this chapter. The internal concave curvature of the concrete shells would lead to sound concentrations in the audience if these surfaces were allowed to serve as the primary reflecting elements. At face value it would appear that this potential problem could be fairly easily overcome by reflecting the direct sound before it can reach the curved shell walls. It would seem that this could be accomplished by means of suspended reflecting panels. However, as was discussed in Sections 10.1 and 10.4, for such panels to be effective they would need to be at least five times as large (in each dimension) as the wavelength of the incident sound. Therefore, for effective low frequency sound reflection the panels would need to be quite large (with dimensions greater than 20 FT by 20 FT for frequencies below 250 cps) and consequently very heavy. For adequate structural integrity, a suspended plywood panel of that size would probably require an effective thickness of 2 IN (e.g., ⅞ IN plywood sheets on either side of a timber frame) with a corresponding weight of around 12 LB/SF or over 2 ton for a 400 SF reflector panel.

Unfortunately, the suspension of such a relatively heavy object from a lightweight shell structure presents a formidable structural problem. Thin concrete shells obtain their structural integrity from double curvature and continuity. For example, it is virtually impossible to crush a whole egg in the palm of one hand. This is due to the fact that the load is applied evenly over the surface of the egg and even though the egg shell is very thin it can support a substantial distributed load. However, the same egg shell can be easily pierced with a pointed object such as a knitting needle. In other words, thin concrete shells are surprisingly strong in supporting evenly distributed loads and very weak in supporting concentrated loads such as those represented by a heavy suspended plywood panel.

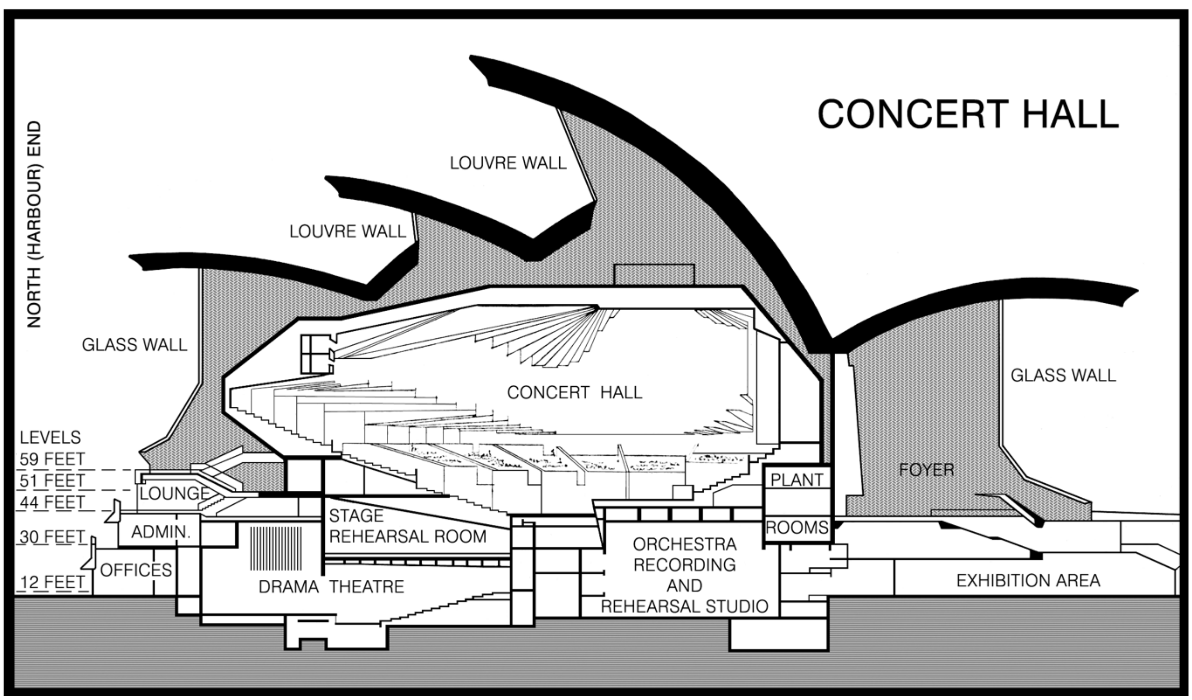


Figure 10.28: Sectional view of the main Concert Hall (Sydney Opera House)

The solution to this dilemma in the case of the Sydney Opera House was to build another enclosure for the concert and opera halls within the external concrete shell perimeter (Figure 10.28). This solution was welcome for another non-structural reason. The location of the Opera House in the Sydney Harbor exposed it to the unusually high sound levels generated by fog horns that warn ships of approaching danger. It would be incongruous and unacceptable to have a particularly quiet portion of a concert performance suddenly interrupted by the excruciating sound of a fog horn. The wide air cavity between the external concrete shells and the internal enclosures of the halls provided the means for applying the necessary noise insulation.

Throughout history significant public building projects have often been the subject of a great deal of controversy and political intrigue. The Sydney Opera House project has been no exception in this regard (Flyvbjerg 2005). Planning for an opera house in Sydney essentially began in the late 1940s with recognition of the need for a more suitable venue for large theatrical productions than was at that time provided by the Sydney Town Hall. In 1954 the advocates for a new building gained the support of the state governor, who authorized an international architectural competition. The renowned American architect, Eero Saarinen, served as a prominent member of the selection committee. It is rumored that at the time the committee commenced the task of evaluating the 233 entries, Saarinen was not able to travel to Australia due to other commitments. So as not to delay the evaluation process, he asked the committee to go ahead and reduce the competition entries to a small number of the best designs. He would then assist the committee with the selection of the winning entry as soon as he was able to come to Sydney.

When Saarinen eventually joined the committee, he was apparently not satisfied with the selections made by the committee and asked to see all of the 233 entries. He was particularly impressed by the submission of the Danish architect, Jørn Utzon, which had not been included in the committee's selection of finalists. Saarinen persuaded the committee that if they wanted Sydney to have a landmark Opera House then this innovative design of sail-like concrete shells would serve magnificently. The committee accepted his recommendation and announced the Utzon design as the winning entry in 1955. Subsequently, Jørn Utzon arrived in Australia in 1957 to take up residence in Sydney and complete the final design drawings and commence construction.

The initial cost estimate for the design and construction of the Sydney Opera House was (AU) $7 million. However, this was a political cost estimate based on what the current state government felt to be acceptable by the public rather than a true estimate of actual costs. The correspondingly dubious construction time for the entire project was announced as five years, with a targeted completion date of January 1963. In fact, the final cost was (AU) $102 million and the Sydney Opera House was formally completed 10 years after that original deadline, in October 1973. From the very start the Opera House became an icon of controversy with various public groups in strong opposition and support of the entire venture. Increasing cost estimates, as well as design and construction delays fueled the controversy. With a change of state government in 1965, the relationship between architect and client deteriorated to the point where the government refused to honor a progress payment of architectural fees and Utzon consequently resigned and returned to Denmark in 1966 (Duek-Cohen 1967, Baume 1967). Thereafter, the design and construction of the Opera House was completed by a government appointed triad of three Australian architects[9] (Hall 1973, Drew 2000).

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9 The New South Wales Government appointed Peter Hall as principal design architect, David Littlemore as construction supervisor, Lionel Todd as chief of documentation, and E. H. Farmer ex officio in his capacity as the State Architect of New South Wales, Australia.

The significant changes to the design after the resignation of Utzon were related to the interior design and finishes, the addition of the Drama and Studio Theaters, the structural solution of the expansive glass walls at the front of the shells, and the enclosure of the podium down to the water level. Apart from the addition of the two theaters, which completely changed the layout of the floor plan, Utzon's entire interior acoustic design concept was replaced by a different solution that includes 21 very large torus shaped acrylic circular reflectors that are suspended from an enormous circular ceiling some 82 FT above the stage. Vernon Jordan of the German firm V. L. and N. V. Jordan served as the acoustical consultant (Jordan 1973). The measured Reverberation Times of the Concert Hall recorded after the completion of construction are just over 2 sec (Table 10.7).

Table 10.7: Measured Reverberation Times (sec) of the Sydney Opera House

Occupancy	125 cps	250 cps	500 cps	1000 cps	2000 cps
Unoccupied	2.45	2.46	2.45	2.55	2.60
Occupied	2.10	2.20	2.10	2.30	2.20

It is of interest to note that due to the unusually high ceiling the volume per audience seat is also much higher than the 100 to 130 CF/seat normally recommended. For this hall it is 324 CF/seat.

10.5 Questions Relating to Chapter 10

Answers to the following multiple-choice questions with references to the appropriate text (by page number) may be found at the back of the book.

1. **Absorption of sound by a surface occurs mainly due to:**
   - A. The inverse square law.
   - B. An interaction between the incident and reflected sound waves.
   - C. A transfer of energy.
   - D. The texture of the surface.
   - E. None of the above statements (i.e., A, B, C, and D) are correct.

2. **The Absorption Coefficient (α) provides a simple numerical scale that relates the incident sound intensity to the sound absorbed by a surface. Complete absorption is rated on this scale as:**
   - A. 1.0
   - B. 0
   - C. 10
   - D. 100
   - E. None of the above statements (i.e., A, B, C, and D) are correct.

3. **The separate factors that will result in absorption include the following:**
   - A. Friction at the surface, low temperature, and diaphragmatic action of porous containers.
   - B. Diaphragmatic action of airtight containers, high temperature, and friction within porous material.
   - C. Low modulus of elasticity of rigid material and friction at the surface or within porous material.
   - D. Friction at the surface or within porous material and diaphragmatic action of airtight membranes.
   - E. None of the above statements (i.e., A, B, C, and D) are correct.

4. **Absorption of sound by a non-porous membrane will be a maximum if:**
   - A. The membrane is rigid.
   - B. The membrane is flexible.
   - C. The membrane is under compression.
   - D. Resonance occurs.
   - E. The membrane faces an air cavity of at least 8 inches width on one side of it.

5. **Sound and light are both forms of energy and therefore have some similar characteristics. Which of the following statements is not correct?**
   - A. They have similar wavelengths.
   - B. When they are reflected by a large flat surface the angle of incidence is equal to the angle of reflection.
   - C. Sound cannot be transmitted through a vacuum, but light can be transmitted through a vacuum.
   - D. All of the above statements (i.e., A, B, and C) are correct
   - E. All of the above statements (i.e., A, B, C, and D) are incorrect.

6. **The property of a boundary surface that has most influence on sound absorption, reflection, and transmission characteristics is:**
   - A. The texture of its surface.
   - B. Its rigidity.
   - C. Its density
   - D. All of the above statements (i.e., A, B, and C) are correct
   - E. All of the above statements (i.e., A, B, C, and D) are incorrect.

7. **In halls of speech and music reflecting panels are often used to improve the quality of the acoustic environment. For effective reflection the dimensions (i.e., length and width) of the panel should be at least:**
   - A. Equal to the wavelength of the sound.
   - B. About one fifth of the wavelength of the sound.

C. Five times the wavelength of the sound.
D. Three times the wavelength of the sound.
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

8. **Porous sound absorbers are characterized by the three properties of porosity, flow resistance, and structure factor. Which (if any) of the following definitions are incorrect:**

A. Porosity is a measure of the amount of air-filled space in the material.
B. Flow resistance is a measure of the resistance of the material to the direct flow of air.
C. Structure factor is a measure of the stiffness of the material.
D. All of the above statements (i.e., A, B, and C) are correct
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

9. **Panel absorbers are mounted in front of an air space. Which (if any) of the following statements is <u>incorrect</u>?**

A. The air acts as a damping mechanism.
B. The resonance frequency of a panel absorber is a function of the mass of the panel and the depth of the air space.
C. The deeper the air space the lower the resonance frequency of the panel absorber.
D. Due to the damping capabilities of the air space, panel absorbers normally have a higher Absorption Coefficient than porous absorbers.
E. All of the above statements (i.e., A, B, C, and D) are correct.

10. **Which of the following statements (if any) are <u>true</u> about acoustic ceiling tiles?**

A. They are essentially panel absorbers and therefore have to be applied over large areas to be effective.
B. The holes in acoustic tiles must be of varying diameter to provide effective sound absorption over a wide range of frequencies.
C. Manufacturers normally quote the absorption characteristics of acoustic tiles in terms of the Noise Reduction Coefficient (NRC), which is based on the average Absorption Coefficient of two frequency bands in the range of 500 to 1000 cps.
D. All of the above statements (i.e., A, B, and C) are correct
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

11. **In volume absorbers sound passing through a small opening sets the air in a relatively large volume in vibration. Which of the following statements (if any) applies to volume absorbers?**

A. The larger the air space the lower the resonance frequency.

B. The smaller the cross-sectional area of the neck opening the lower the resonance frequency.
C. Loosely filling the air space with fibrous material increases the sound absorption.
D. All of the above statements (i.e., A, B, and C) are correct
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

12. **Which of the following statement (if any) are <u>incorrect</u> in respect to speech communication?**
    A. Most of the energy in speech sounds is distributed over a frequency range of 200 to 6000 cps.
    B. Speech Interference Level is the average of the sound pressure levels in the three octave bands of 300-600 cps, 600-1200 cps, and 1200-2400 cps.
    C. About 10 speech sounds are produced each second during normal speech.
    D. All of the above statements (i.e., A, B, and C) are correct
    E. All of the above statements (i.e., A, B, C, and D) are incorrect.

13. **For good speech communication the background noise level should be at least *X* dB below the communicating sound pressure level. What is the <u>correct value of *X*</u>?**
    A. X = 2 dB
    B. X = 4 dB
    C. X = 6 dB
    D. X = 8 dB
    E. All of the above statements (i.e., A, B, C, and D) are incorrect.

14. **In halls for speech and music each audience position should receive both direct sound and at least one strong reflection (i.e., echo). However, if a strong reflection comes too late then it will interfere with the next direct sound. What time period can separate the direct and reflected sound for a person to perceive them as a single sound.**
    A. 2 sec
    B 1 sec
    C. 0.1 sec
    D. 0.05 sec
    E. All of the values in the above statements (i.e., A, B, C, and D) are incorrect.

15. **Reverberation Time is defined as the time taken by a sound to decay by *X* db. What is the value of *X*?**
    A. X = 80 dB
    B. X = 70 dB
    C. X = 60 dB

D. X = 50 dB
E. All of the values in above statements (i.e., A, B, C, and D) are incorrect.

16. **The Reverberation Time of a lecture hall that can seat 200 persons should be about *X* sec. What is the value of *X*?**
    A. X = 0.5 sec
    B. X = 1.0 sec
    C. X = 2.0 sec
    D. X = 3.0 sec
    E. All of the values in above statements (i.e., A, B, C, and D) are incorrect.

17. **The Reverberation Time of an auditorium is mainly a function of:**
    A. The volume and total sound absorption of the auditorium.
    B. The slope of the floor.
    C. The curvature of the rear wall.
    D. All of the above factors (i.e., A, B, and C) are taken into account in the calculation of Reverberation Time.
    E. All of the above statements (i.e., A, B, C, and D) are incorrect.

18. **The audience has a major impact on the Reverberation Time of all concert halls, because:**
    A. The audience increases the background noise level (mostly due to coughing, talking, and movement).
    B. The audience can contribute as much as 75% of the total sound absorption in the hall.
    C. The audience blocks much of the sound reflected by side walls from reaching seats in the middle of the hall.
    D. All of the above statements (i.e., A, B, and C) are correct
    E. All of the above statements (i.e., A, B, C, and D) are incorrect.

19. **The Reverberation Time of a concert hall that can seat 1,800 persons should be about *X* sec. What is the value of *X*?**
    A. 2 sec
    B 1 sec
    C. 0.1 sec
    D. 0.05 sec
    E. All of the values in the above statements (i.e., A, B, C, and D) are incorrect.

20. **Important considerations for the design of a concert hall include:**
    A. The internal shape of the hall (i.e., sharply curved shapes should be avoided).
    B. The external appearance of the hall.

C. The ceiling height of the hall.
D. All of the above statements (i.e., A, B, and C) are correct
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

21. **Which (if any) of the following did the world renowned acoustics engineer Leo Beranek consider to be the most important criterion for good concert hall acoustics?**
A. A coffered ceiling.
B. The correct Reverberation Time at low frequencies.
C. A hall volume of less than 200 CF per audience seat.
D. One reflection within 20 milliseconds.
E. All of the above statements (i.e., A, B, C, and D) are incorrect.

22. **Which (if any) of the following statements (i.e., A, B, C, and D) is <u>incorrect</u>?**
A. In the case of concert halls sound diffusion is a better design approach than sound absorption for side walls.
B. Ideally, every audience position in an auditorium should receive a strong first reflection within 35 milliseconds of the direct sound.
C. Wood paneling is a poor reflector of sound at high frequencies, but a good reflector at low frequencies.
D. In churches the ceiling height above the choir should be at least 20 FT.
E. All of the above statements (i.e., A, B, C, and D) are correct.