Chapter Eleven

Noise Control and Insulation

The control of noise in the external environment and inside buildings is at least as important to the quality of life as the design of building spaces for the enjoyment of music and the promotion of good speech communication conditions. So, what is *noise* as opposed to sound? By definition any sound that is annoying, distracting and generally unwanted is commonly referred to as noise. The term *unwanted* appears to be of particular significance in determining whether a particular sound is perceived to be tolerable or annoying. For example, persons are usually not annoyed by noise originating from their own activities, but may be greatly annoyed if a similar noise is produced by the apparently unnecessary activities of others.

11.1 Noise Control by Legislation

In the years following World War II the average noise level in highly industrialized countries grew at an alarming rate. During the period of 1935 to 1955, Knudsen (1955) estimated a yearly increase in the average noise level of one decibel, which indicates an approximate doubling of loudness in about one decade. To counteract this trend many local, state and national government authorities around the world felt compelled to take some action. As a first step, particularly at the national level, the responsible government agencies recognized the need for an assessment of the nature and extent of the problem. This led to the funding of several research studies and field surveys aimed at providing a basis for legislation to control the creation and mitigation of noise (Aldersey-Williams 1960, Piess et al. 1962, Karplus and Bonvallet 1953, HMSO 1963).

In the built environment we are normally concerned not with noise sources that may produce permanent damage to the hearing mechanism, but with the more complicated and less precise aspects of annoyance. It is now generally recognized that:

- Persons are unlikely to be annoyed by noise originating from their own activities.
- It is possible for individuals to become accustomed to certain noises.
- Annoyance is a function of the sound pressure level as well as the frequency spectrum of the noise.
- There are some noises (e.g., such as those that produce fear or that disturb sleep) to which persons are unable to adapt even after prolonged exposure (Nickson 1966).

The matter is further complicated by the highly subjective nature of individual reactions to noise. Not only do individuals exhibit different tolerances and conditioning abilities, but their reactions also vary with the particular circumstances. At times it is indeed difficult to assess whether the reaction produced has been activated by physiological or psychological stimuli.

Unexpected impulsive sounds may increase the pulse rate and cause muscular contractions. Accordingly, on the basis of the definition of health stipulated by the World Health Organization, namely, "… *health is a state of complete physical, mental and social well-being, and not merely an absence of disease and infirmity*", it must be accepted that noise can constitute a health hazard even though there may be no risk of actual physical hearing damage.

Outdoor noise sources can be generally divided into three main groups, namely, industrial, residential, and traffic noise. Of these residential and in particular traffic noise seem to constitute the major source of annoyance, since interference from industrial noise has now been largely eliminated by the implementation of town planning legislation. However, one aspect of industrial noise has gained importance in recent years, and that is noise from building construction sites in urban areas. Most large multistory buildings have construction periods exceeding 12 months and generate a great deal of noise at least during the first half of this period, while site works and the erection of the structural frame are in progress.

Most of the noise produced in residential areas is due to traffic and activities, such as children at play, motor mowers, radios, and television. The interference of noise produced by tenants in multi-unit dwellings is of particular concern. For this reason, building codes in most countries stipulate minimum sound transmission loss values for party walls between adjacent apartments and condominiums.

Type of External Noise Source	Maximum SPL	Distance from Source	US State or Country
Diesel-powered tractors, trucks and buses.	95 dBA	25 FT	California
Motorcycles.	90 dBA	25 FT	California
Regular passenger cars and sports vehicles.	85 dBA	25 FT	California
All vehicles travelling at more than 35 mph.	94 dBA	25 FT	New York
All new vehicles with more than two wheels.	85 dBA	17 FT	England
New motorcycles and other mechanically propelled two-wheeled vehicles.	90 dBA	17 FT	England

Figure 11.1: Legislated maximum noise levels

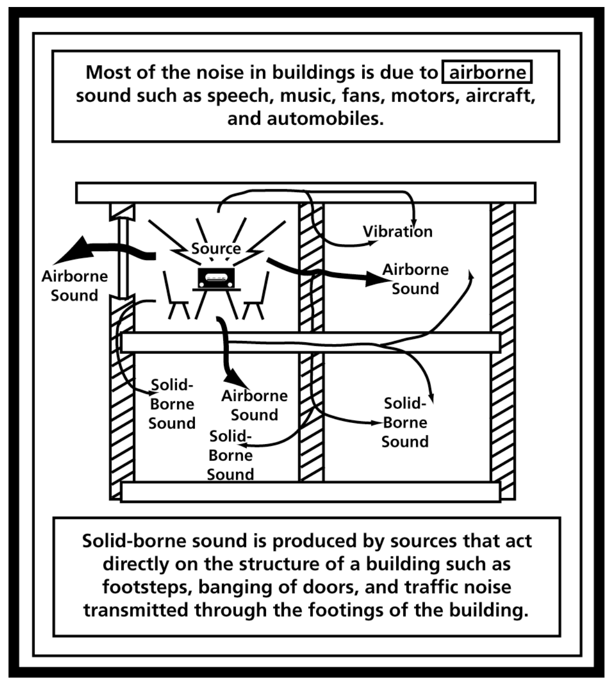

Figure 11.2: Transmission of air-borne and solid-borne noise through a building

In respect to external noise sources, the approach to legislation is based on the control of maximum noise levels. This applies in particular to vehicular traffic noise although, as can be seen in Figure 11.1, there is some variation between countries and even between states in the US. For example, the difference between California and New York is 9 dB, which constitutes a doubling of loudness. The recommendations made by the Ministry of Transport in England, based on the Wilson Report (HMSO 1963) are generally more stringent. In Europe, prior to the formation of the European Economic Union, the situation was similarly disjointed with each country progressively drawing up its own requirements and methods of measurement, covering one or more types of vehicles. Today, the EEU has established standards in virtually all fields including noise control that must be observed by its members.

11.2 Air-Borne and Solid-Borne Sound

The attenuation of sound in air varies directly with the frequency of the sound and inversely with the relative humidity of the air. For example, an attenuation constant of up to 9 dB per 100 FT may be obtained for a sound of 10,000 cps. frequency under ambient atmospheric conditions, if the relative humidity is 20%. Similarly, sound is refracted by both temperature and wind gradients, giving rise to so called *shadow zones*. These may be created when a blanket of air at high temperature is located near the surface of the ground, or up-wind from a sound source, particularly if the topography is sufficiently rough to reduce the wind velocity near the surface of the ground. Unfortunately, the converse also applies down-wind from a noise source, or in the case of a temperature inversion. Under these conditions, the noise that would normally disperse into the atmosphere is refracted back toward the ground. It is therefore apparent that whenever noise measurements are taken externally over long distances, the weather conditions should be accurately recorded and considered during the analysis of the test results.

In the case of buildings, as shown in Figure 11.2, an air-borne sound may travel directly to the ear of a listener, or it may be instrumental in setting up vibrations in the surrounding structure and partitions, which will in turn create compression waves in the surrounding air. Since sound waves in air can be transmitted with little loss of intensity along apparently insignificant paths, such as keyholes, ventilation grills, badly fitted door jambs, it is essential that sound barriers should be impervious and carefully sealed at the perimeter to eliminate *flanking paths*. With particularly intense sounds it is sometimes possible to physically feel the vibration of a partition, and it is therefore not difficult to understand that this movement of the partition will act on the surrounding air in exactly the same manner as a loudspeaker. Accordingly, for air-borne sound insulation to be effective, it will be necessary to reduce the ability of an insulating partition to be set in vibration by incident sound waves.

Solid-borne sound is produced by sources that act directly on the structure of a building, in the form of impacts or vibrations. Impact sources include footsteps, scraping of furniture, and slamming of doors, while vibration sources include traffic noise transmitted through the footings of a building and machinery such as air-conditioning compressors and fans. All of these are transmitted through and from the structure. In fact, the presence of solid-borne sound may sometimes be detected by listening with one ear pressed tightly against a wall or floor surface. For example, in dense urban areas such as New York or London the vibrations generated by a subway train traveling at some speed well below the ground may be felt throughout the structure of the buildings above its path. Broadly speaking, solid-borne sound insulation will rely on methods of dampening the impact of objects (e.g., carpets effectively dampen the impact of footsteps and moving furniture) and reducing or isolating the vibration of the source (e.g., flexible mountings).

11.3 Air-Borne Noise Insulation

Air-borne noise may be reduced by the use of absorption, by means of effective insulation barriers, or by a combination of both of these. Since the degree of interference of noise with voice communication is closely related to the relative sound pressure levels of the interfering noise and the background noise level due to occupancy, it might be expected that sound reduction at the source by means of absorption would constitute a viable method of air-borne noise insulation. Unfortunately, the only instance where absorption alone will be an economical proposition is found in large offices where there are many noise sources and the occupants are scattered over a large area. Even here it is rare to achieve a reduction in air-borne noise in excess of 6 dB (Lawrence 1968). However, in the case of marginal noise problems affecting speech communication this improvement may be adequate.

In Chapter 10 (Section 10.1) we discussed the relationships that exist among the Coefficients of Absorption (α), Transmission (τ) and Reflection (ρ). In all cases a value of 1 indicates complete effectiveness (i.e., full absorption, transmission and reflection), while a value of 0 implies total ineffectiveness (e.g., an open window is presumed to provide full absorption and absolutely no insulation or reflection. Since values of the Transmission Coefficient (τ) for common building elements such as walls and floors tend to be very small (i.e., between 10^{-2} and 10^{-8}) and therefore rather awkward to use, the sound insulation capabilities of a barrier are normally measured on a logarithmic scale in terms of Transmission Loss (TL) values.

$$\textbf{Transmission Loss (TL)} \quad = \quad \mathbf{10 \log (1 / \tau)} \;\ldots\ldots\ldots\ldots\ldots\ldots \quad \mathbf{(11.1)}$$

If we know the Transmission Coefficient of a material then we can calculate the theoretical TL value. For example, a ¼ IN thick glass window pane has a Transmission Coefficient of 0.00078 (i.e., 7.8×10^{-4}). Therefore, applying equation 11.1 the TL value becomes:

$$\text{Transmission Loss (TL)} \quad = \quad 10 \log (1/0.00078) = 10 \log (1{,}282) = 31 \text{ dB}$$

Similarly, we can extrapolate equation 11.1 to calculate the Transmission Coefficient that will result in a given TL value. For example, a desired TL of 30 dB should give us a required TL value that is very close to that of the ¼ IN thick glass pane used in the previous calculation.

$$\begin{aligned} 30 &= 10 \log (1 / \tau) \\ \log (1 / \tau) &= 30 / 10 \\ (1 / \tau) &= 10^3 \\ \tau &= 10^{-3} = 0.001 \text{ (which is close to 0.00078)} \end{aligned}$$

Why is a good sound absorption material not also a good sound insulation material? If, for example, a material has an Absorption Coefficient of 0.95 at a particular frequency (i.e., at that frequency 95% of the sound will be absorbed) then surely 95% of the sound will also not be transmitted. Let us test this hypothesis. If 95% of the sound is absorbed then the Transmission Coefficient for this material at that frequency is:

$$\text{Transmission Coefficient } (\tau) \quad = \quad (1 - 0.95) = 0.05 \text{ (or } 5 \times 10^{-2})$$

Substituting in equation 11.1, we obtain:

$$\text{Transmission Loss (TL)} \quad = \quad 10 \log (1 / 0.05) = 10 \log (20) = 13 \text{ dB}$$

The reason why the hypothesis is wrong is because the Absorption Coefficient (α) is a linear measure, while the Transmission Coefficient (τ) is a logarithmic measure. For a more practical example let us consider heavy curtain fabric (18 oz), shown in Table 10.5 (Chapter 10) to have an Absorption Coefficient of 0.55 at a frequency of 500 cps. The Transmission Coefficient at that frequency is:

Transmission Coefficient (τ) = (1 – 0.55) = 0.45 (or 4.5×10^{-1})

Substituting in equation 11.1, we obtain:

Transmission Loss (TL) = 10 log (1 / 0.45) = 10 log (2.2) = 3.4 dB

These two examples foreshadow another important principle of sound insulation that will be discussed in more detail later. A good sound barrier must be devoid of air paths, because air is a good conductor of sound. This is incompatible with the characteristics of a sound absorption material that depends on porosity in the form of open air pockets and cul-de-sac pores for its absorption capabilities (i.e., to convert sound vibration through friction into heat).

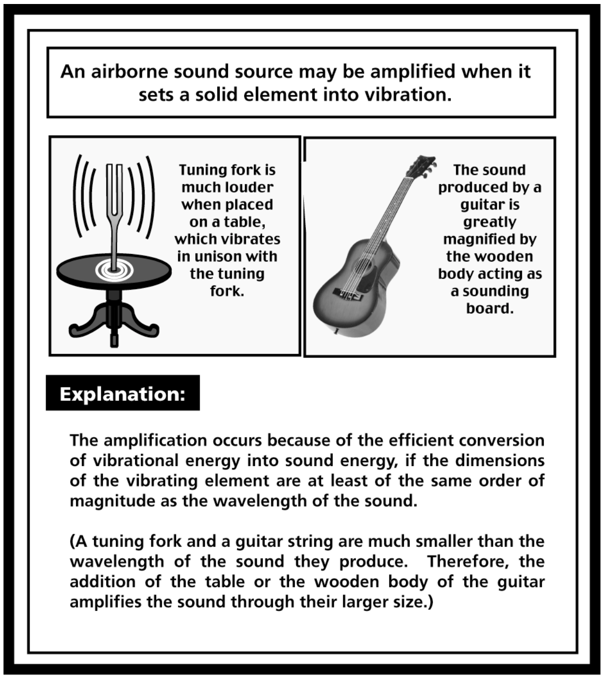

Figure 11.3: The sound-board effect created by placing a tuning fork on a table

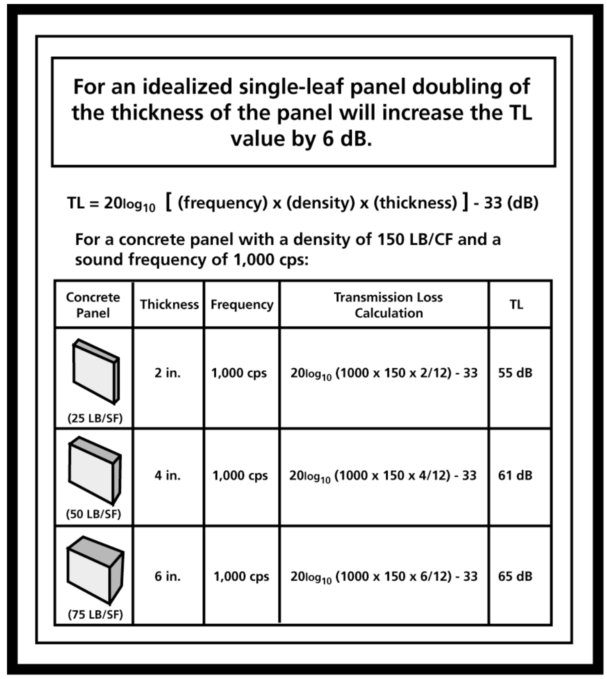

Figure 11.4: Effect of thickness on a single-leaf panel

Before delving into sound insulation in more detail the reader should be aware that air-borne sound can be significantly reinforced by solid components. As shown in Figure 11.3, the sound produced by a tuning fork is greatly amplified when the tuning fork is placed on a solid element such as a table. In a similar manner the strings of a violin or guitar would produce little sound without the wooden body of the instrument acting as a sounding board. However, for the sounding board to be effective it must be of at least the same order of magnitude as the wavelength of the sound produced by the strings. Therefore, at least up to a point, the larger the table the louder the sound produced by the interaction of the relatively small tuning fork with the much larger table.

11.3.1 Single-Leaf Panels and the Mass Law

Although most space dividers in buildings, such as walls and floors, are composed of several materials that are applied in layers (e.g., an external timber wall constructed of a sheet of drywall on the inside and a sheet of stucco on the outside of a timber frame), we will first consider the sound insulation characteristics of single layer panels. Such a single-leaf panel obeys what is commonly referred to as the *Mass Law*. To facilitate mathematical analysis, it is assumed that a single-leaf panel consists of many connected parts that move largely independently of each other when the panel is set in motion (i.e., vibration) by an incident sound wave. This is of course very much an idealized model of the vibration of the panel. In reality the movements of these theoretical parts of the panel are not at all independent of each other. It stands to reason that the stiffer the panel the more each part will be impacted by the movement of its neighbors. However, it has been verified by means of physical tests that the TL of a single-leaf panel increases in direct logarithmic proportion to its surface mass (i.e., mass per SF) and the frequency (f) of the sound to which it is exposed.

In the case of building construction, the surface mass of a component such as a wall is governed by the density (d LB/CF) and thickness (t FT) of the material. Therefore, the Mass Law may be stated as follows:

Transmission Loss (TL) = 20 log (d x t x f) - C (dB) ……………… (11.2)

Where *C* is a constant and equal to *33* in the American system of units and *47* in the Metric system of units (i.e., with d in kg/m^3 and t in m).

Applying equation 11.2 to an 8 IN thick concrete wall with a density of 150 LB/CF, exposed to a sound with a frequency of 500 cps, we obtain:

$$TL_{500} = 20 \log [150 \times (8/12) \times 500] - 33$$
$$TL_{500} = 20 \log [50000] - 33$$
$$TL_{500} = 20 (4.7) - 33 = 94 - 33 = 61 \text{ dB}$$

For an idealized single-leaf panel doubling of the frequency of the sound will increase the TL value by 6 dB.

$TL = 20\log_{10}$ [(frequency) x (density) x (thickness)] - 33 (dB)

For a concrete panel with a density of 150 LB/CF and a thickness of 4 in.:

Concrete Panel	Thickness	Frequency	Transmission Loss Calculation	TL
(50 LB/SF)	4 in.	500 cps	$20\log_{10}$ (500 x 150 x 4/12) - 33	55 dB
(50 LB/SF)	4 in.	1,000 cps	$20\log_{10}$ (1000 x 150 x 4/12) - 33	61 dB
(50 LB/SF)	4 in.	2,000 cps	$20\log_{10}$ (2000 x 150 x 4/12) - 33	67 dB

Figure 11.5: Effect of frequency on a single-leaf panel

For an idealized single-leaf panel doubling of the mass (i.e., density) of the panel will increase the TL value by 6 dB.

$TL = 20\log_{10}$ [(frequency) x (density) x (thickness)] - 33 (dB)

For a sound frequency of 4000 cps and three 1/2 in. thick panels made of corkboard (35 LB/CF), plywood (70 LB/CF) and gypsum (140 LB/CF).

Panel Type	Thickness	Frequency	Transmission Loss Calculation	TL
(1 LB/SF)	1/2 in.	4,000 cps	$20\log_{10}$ (4000 x 35 x 1/24) - 33	42 dB
(3 LB/SF)	1/2 in.	4,000 cps	$20\log_{10}$ (4000 x 70 x 1/24) - 33	48 dB
(6 LB/SF)	1/2 in.	4,000 cps	$20\log_{10}$ (4000 x 140 x 1/24) - 33	54 dB

Figure 11.6: Effect of mass on a single-leaf panel

The direct logarithmic relationship between density, thickness, and frequency in equation 11.2 leads to an interesting and useful conclusion. Doubling of any one of these three quantities has the same numerical impact on the value of TL. In each case, it will result in an increase in the value of TL by the exact amount of '*20 log (2)*', which is equal to *6*. Therefore, as shown in Figures 11.4, 11.5 and 11.6, each doubling of the thickness, frequency or density of a single-leaf panel will increase its theoretical noise insulation (i.e., Transmission Loss value) by 6 dB.

However, as discussed previously, the Mass Law makes assumptions that are only partly realized in practice. While surface mass is certainly the most important factor that determines the sound insulation provided by a single-leaf panel, other factors such as stiffness, boundary conditions, and the angle of incidence of the sound, will influence the degree to which the theoretical TL value predicted by the Mass Law is realized in practice. Figure 11.7 shows schematically the significant increase in TL that will result from measures such as cutting grooves in the surface of a panel, aimed at reducing the overall stiffness of the panel. Similarly, but to a lesser extent the damping that will be provided by the manner in which the panel is fixed at the boundary will also affect its stiffness. In addition, the actual sound insulation provided is also dependent on the angle of incidence of the noise. In practice, wave fronts of sound generally arrive over a range of angles from 0˚ to 80˚, which is referred to as field incidence, while the Mass Law assumes that the sound waves are incident at an idealized angle of 90˚ (i.e., perpendicular) to the panel surface.

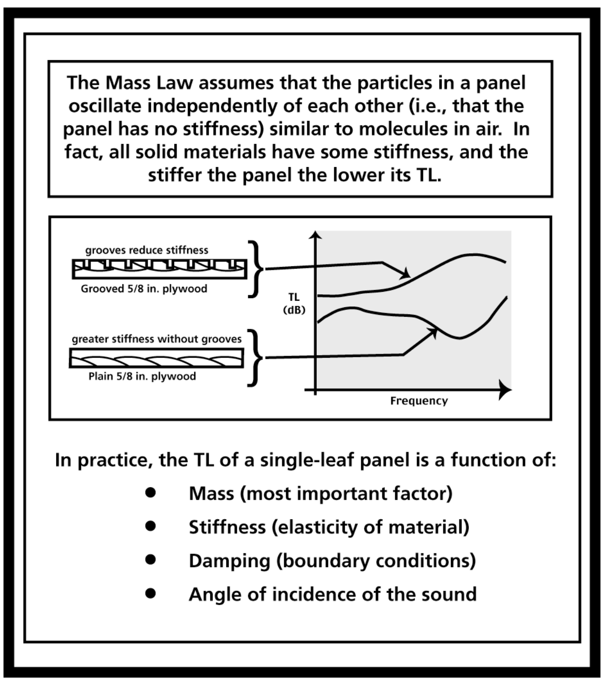

Figure 11.7: The impact of stiffness on the TL value of a single-leaf panel

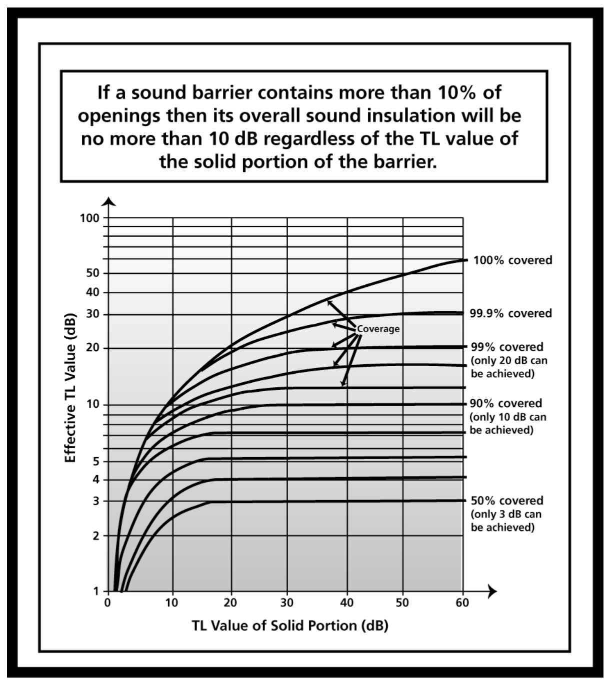

Figure 11.8: The impact of openings on the TL value of any sound barrier

Taking these factors into consideration, the theoretical TL value for each doubling of density, thickness, or frequency, reduces by at least 20% to 5 dB in practice. If the sound barrier is totally sealed so that there are no direct air paths, then stiffness is the factor that contributes most to this discrepancy between theory and practice. However, as shown in Figure 11.8 the presence of any direct air paths will drastically reduce the effectiveness of any sound barrier. If a barrier contains more than 10% of openings its overall TL value will be equivalent to no more than about 10 dB, regardless of the insulation value of the solid portion of the barrier. This has become a serious problem in commercial buildings where it is very difficult to eliminate flanking paths around or over the top of modular partitions. Naturally, continuation of such partitions above the ceiling would interfere with ventilation. A simple method of checking the adequacy of seals around full height partitions in air-conditioned (or mechanically ventilated) buildings is to close off the return-air duct and measure the reduction in the supply air velocity. In the case of partial enclosures (i.e., partial height partitions), the noise insulation provided is mainly governed by reflecting surfaces nearby. Accordingly, the provision of absorptive material on large horizontal overhead surfaces will generally ameliorate conditions (Mariner and Park 1956, Bishop 1957). In summary, the following guidelines should be kept in mind by the building designer:

- Generally, the heavier a wall the more effective its sound insulation capabilities.
- Stiffness reduces the sound insulation capabilities of a barrier. Lead is the most effective sound insulation material, because it is very heavy and limp at the same time. However, lead is so limp that it cannot support its own weight in most structural configurations. Also, lead is potentially toxic (e.g., handling of lead sheets during construction) and expensive. Therefore, it is typically used only as a hidden layer in highest quality modular partitions and doors.
- If a sound barrier has an acoustically weak element, then its overall TL value is likely to be close to the TL value of the weak element.
- The smallest air path, such as a badly fitting door or even a large keyhole will transmit sound to a disproportionate degree directly through the barrier.
- When a sound barrier contains different components such as windows and doors, then the effective TL value of the barrier is *not* directly proportional to the relative areas of the components (i.e., as would be the case for thermal insulation). The effective sound insulation of a barrier consisting of two different components can be determined by reference to a table that relates the difference in TL values of the two components to the percentage area of the smaller component (Table 11.1).

Table 11.1: Reduction in Transmission Loss (TL) of an Assembly of Two Components

Difference	Area of Smaller Component as Percentage of Total Area							
in TL values	50%	20%	10%	5%	2%	1 %	0.5%	0.1%
5	3.0	1.5	1.0	0.5	0.0	0.0	0.0	0.0
6	4.0	2.0	1.0	0.5	0.0	0.0	0.0	0.0
7	5.0	2.5	1.5	1.0	0.5	0.0	0.0	0.0
8	6.0	3.0	2.0	1.0	0.5	0.0	0.0	0.0
9	6.5	4.0	2.5	1.0	0.5	0.5	0.0	0.0
10	7.5	4.5	3.0	2.0	1.0	0.5	0.0	0.0
15	12.0	8.5	6.0	4.0	2.0	1.0	1.0	0.0
20	17.0	13.0	10.5	8.0	5.0	3.0	2.0	0.5
30	27.0	23.0	20.0	17.0	13.0	10.0	8.0	3.0
40	37.0	33.0	30.0	27.0	23.0	20.0	17.0	10.5

The application of Table 11.1 is shown in Figure 11.9, where the overall TL value of 210 SF wall with a TL value of 45 dB containing a 21 SF door with a TL value of 25 dB, is determined to be 34.5 dB. It is of interest to note that although the area of the door is only 10% of the wall area the reduction in the overall TL value is over 23%. Utilizing the graph shown in Figure 11.8, this example is taken one step further in Figure 11.10 where it is assumed that the door has been poorly fitted with a 2 IN gap at the bottom. Even though this direct air path is only 0.3% of the total wall area the impact is quite dramatic with an additional 25% reduction in the overall TL value.

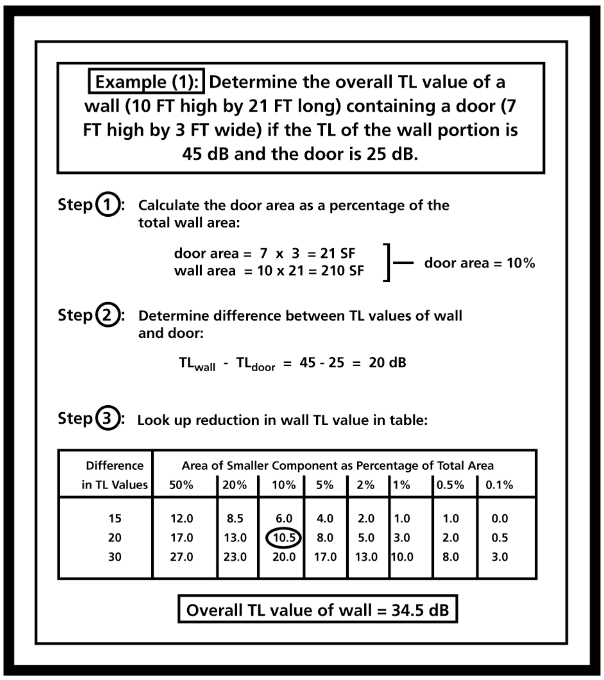

Figure 11.9: Sound insulation impact of a well fitted door

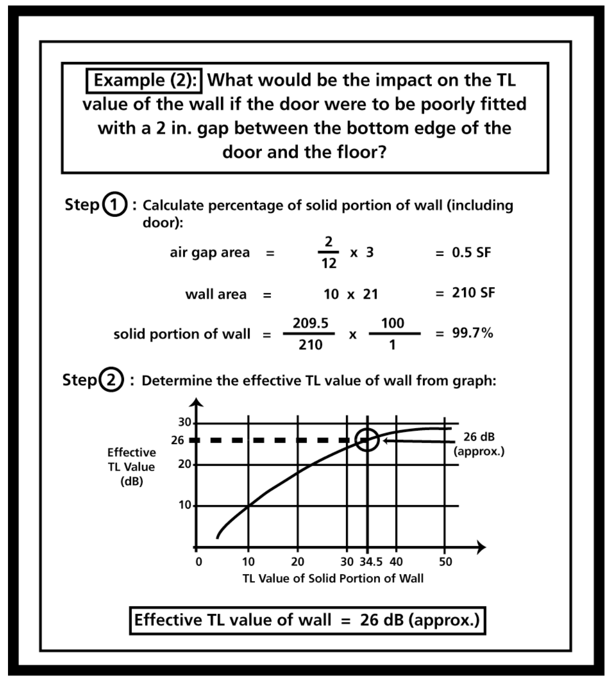

Figure 11.10: Sound insulation impact of a poorly fitted door

Most manufacturers quote TL values of structural walls and partitions as an average of the reduction in decibels of a diffuse sound field passing through the partition mounted in a specified manner for nine frequencies (i.e., 125, 175, 250, 350, 500, 700, 1000, 2000, and 4000 cps). These tabulated values are normally based on laboratory tests using very massive side walls and may therefore not be realized in practice (Harris 1957). In Figure 11.2 are shown two adjacent rooms. Noise created in room A will reach a listener in room B mainly by a direct sound path, but also by a number of indirect sound paths. These secondary paths may decrease the sound insulation of an installed partition by up to 10 dB. Similarly, if transmission loss values are given as the average for a range of frequencies it is of importance to ensure that the frequency range is sufficiently wide.

There are two important aspects of the relationship between the mass and sound transmission loss of a partition that require further explanation.

Resonance: If a barrier is forced to vibrate at its natural frequency, the amplitude of the vibration is likely to be very large. Under these conditions the TL of a partition will be sharply reduced (Figure 11.11). Resonance occurs often at very low frequencies and is largely controlled by stiffness. Although the effects of stiffness and mass both vary with frequency, they are opposed to each other and tend to cancel each other out (Day et al. 1969). It is therefore apparent that the ideal barrier material is one that is heavy, limp and highly damped.

Coincidence Effect: The incidence of sound pressures on the surface of a barrier will set the barrier in vibration, producing bending waves. Although the velocity of sound in air is constant for all frequencies, the velocity of induced bending waves in solids increases with higher frequencies. Naturally, at some particular frequency, the velocity of bending waves in a given barrier will coincide with the velocity of sound in air. This is known as the *Critical Frequency* and gives rise to a more efficient transmission of sound. As shown in Figure 11.12, a significant reduction in sound insulation occurs at frequencies above the Critical Frequency, which suggests that for partition walls the Critical Frequency should be as high as possible.

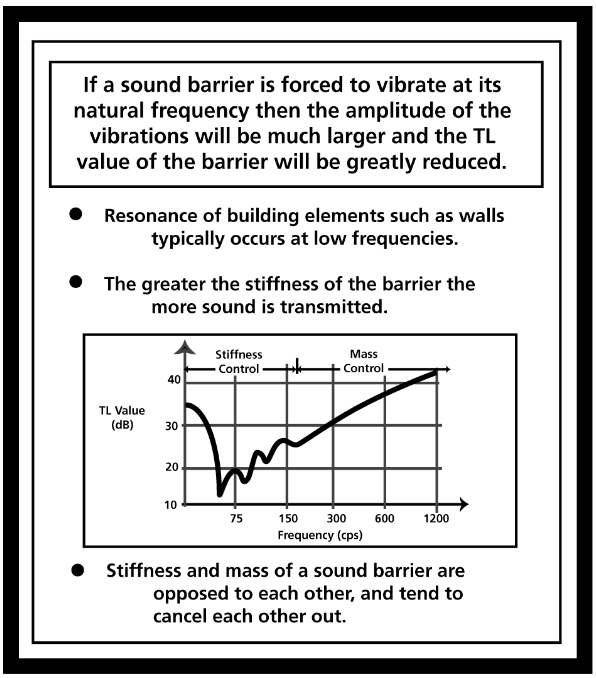

Figure 11.11: Impact of resonance

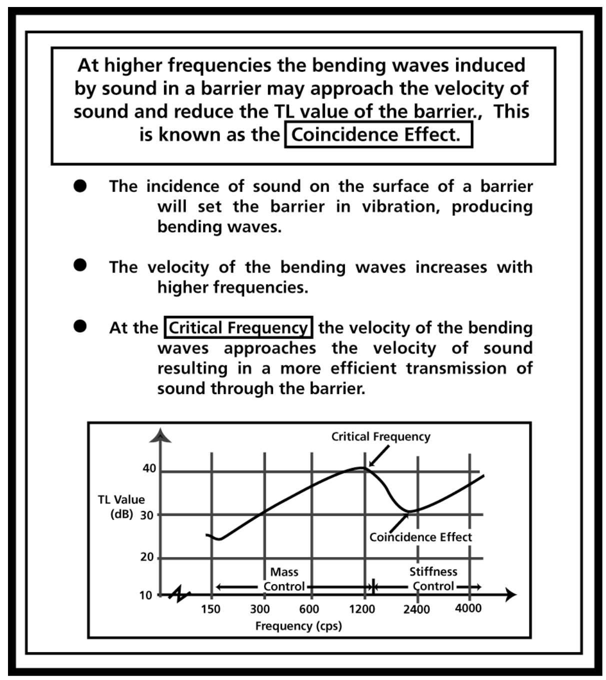

Figure 11.12: The Coincidence Effect

The Critical Frequency of a sound barrier is a function of the density, modulus of elasticity, and the thickness of the material. Therefore, for a given material:

$$\text{Critical Frequency} \quad \infty \quad (\ 1\ /\ \text{panel thickness}\ )$$

Therefore, by increasing or decreasing the thickness of a single-leaf panel it is possible to slightly adjust its Critical Frequency upward or downward beyond the frequency range of the transmitted sound. However, reducing the thickness of a panel to increase the Critical Frequency will also reduce the TL value at the lower frequencies. Fortunately, in practice the Coincident Effect is of concern only for relatively thin partitions such as plywood, glass, and gypsum panels.

Another way of reducing the impact of the Coincidence Effect is to increase the damping of a barrier by adding a viscoelastic layer such as mastic, acrylic sheeting, or one or more layers of lead sheeting. A relatively new type of material with superior damping characteristics is *loaded vinyl*, which consists of a polymer mixed with a fairly heavyweight inorganic material such as calcium carbonate or barium sulfate (ASI 2007). For similar reasons it would seem that a lead sheet hanging loosely as a curtain would satisfy the requirements of an ideal barrier, because it is very heavy and well damped (i.e., limp). For example, a 1/32 IN thick sheet of lead weighs some 2 LB/SF and has a transmission loss value of 32 dB. Thus, a ¼ IN lead sheet weighing 15 LB/SF has a transmission loss value equivalent to a 9 IN thick solid brick wall weighing 105 LB/SF[1].

11.3.2 Sandwich Barriers and Multi-Leaf Walls

As we saw in the previous section, the doubling of thickness of a single-leaf barrier provides only about a 5 dB increase in the TL value. It is therefore more economical to use a multi-leaf barrier consisting of two or more elements. These are also commonly known as sandwich barriers and consist of a combination of materials chosen for both structural and acoustical properties. For example, two sheets of relatively high stiffness and strength such as plywood may be combined with a polystyrene core. Since high stiffness is an undesirable property for a sound barrier, the overall stiffness of a sandwich panel can be reduced by selecting a core material with low shear strength. Thus, at low frequencies when the bending waves are large the core will act as a rigid spacer, while at high frequencies when the bending waves are short, the low shear strength of the core will effectively reduce the stiffness of the panel (Day et al. 1969).

An alternative approach to a sandwich panel is to separate two single-leaf panels with an air space. The cavity wall or double-leaf partition potentially offers the greatest scope for large noise reduction in building construction. Although in theory it should provide at least twice as much transmission loss as each leaf separately, this is not found to be the case in the field. In other words, if the two leaves are completely decoupled then the effective TL value should be the sum of the two individual TL values. However, in practice the two leaves of a double-leaf barrier can never be completely decoupled. Even if all direct ties and common footings are eliminated, the transmission loss of a double-leaf barrier still falls short of the predicted value. Normal 2 IN wide cavities that are very effective for heat insulation purposes, might provide no more than a 2 dB increase in sound insulation. The following explanation applies.

At low frequencies the air in the cavity loosely couples the two leaves, very much like a coil spring, giving rise to a resonant frequency that is determined by the mass of the leaves and the width of the cavity. Thus, during resonance the transmission loss value of the barrier is sharply reduced and it is therefore necessary to ensure that the resonant frequency is very low (i.e., below 100 cps.). In practice this means that if the two leaves are of low mass the cavity will need to be fairly wide. At frequencies above the resonant frequency, the insulation of the cavity barrier will increase more rapidly than that of a solid barrier, although at about 250 cps the danger of cavity resonance arises (Day et al. 1969). This problem can be overcome by partly filling the cavity with sound absorbent material, such as fiber glass, flexible plastic foam, or mineral wool. Absorbent cavity infill has been found to be most effective in lightweight construction, with an expected increase in transmission loss of around 5 dB.

From a general point of view the following three categories of sound insulation barriers, based on the amount of sound transmission loss required, need to be considered by building designers:

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[1] Unfortunately, the tensile strength of lead is insufficient to support even its own weight for the ratio of thickness to length required for this application. However, this limitation can be overcome by spraying lead onto a polythene sheet. The resultant laminated lead sheeting can be applied in situations where high transmission loss and flexibility are required.

Below 40 dB transmission loss: The two leaves of a multi-leaf barrier may be connected by common studs or other framing. Although the use of absorbent infill in the cavity is advisable, it could be reduced to a relatively thin blanket.

Between 40 and 50 dB transmission loss: The required degree of transmission loss will require studs to be staggered, so that there is no direct connection between the two leaves. Experience has shown that even the bridging provided by tie wires in a brick cavity wall may be sufficient to negate the required isolation.

Above 50 dB transmission loss: The method of support of the entire room enclosure (i.e., the perimeter linking) assumes major importance, so that some form of discontinuous construction is normally required.

The sound transmission loss provided by a single window pane follows the same laws discussed in Section 11.3.1 for single-leaf barriers (i.e., mass and stiffness). However, while the assumption of random incidence of sound is reasonable in the case of an internal wall, it does not apply to external windows. Vehicular traffic and other outdoor noise sources normally produce a predominant angle of incidence. Accordingly, TL values provided by manufacturers for window units are often given for different angles, such as 0˚ (i.e., normal), 45˚ and 70˚. Double glazing conforms to the sound insulation pattern of a cavity partition. Therefore, in order to ensure a low resonance frequency, the small mass of the two sheets of glass will need to be supplemented by a wide cavity (i.e., 9 IN or preferably wider). A further increase in transmission loss of around 5 dB may be obtained in the frequency range of 400 cps to 3200 cps by lining the frame surrounding the cavity with absorbent material.

To calculate the effective sound transmission loss of a composite wall (e.g., glass and solid) it is necessary to determine the Transmission Coefficient (r) for each section using equation 11.1, as follows:

$$\text{Transmission Loss (TL)} = 10 \log (1 / r) \quad \ldots\ldots\ldots\ldots \quad (11.1)$$

Rewriting equation 11.1 in terms of $\boldsymbol{r}$ we obtain:

$$r = 1 / [\text{antilog } (TL / 10)]$$

In the case of a solid 9 IN single-leaf brick wall with a TL value of 45 dB and a glazed window unit with a TL value of 25 dB, the Transmission Coefficients for the brick wall (r_w) and the window (r_g) are given by:

$$r_w = 1 / [\text{antilog } (4.5)] = 3.2 \times 10^{-5}$$

$$r_g = 1 / [\text{antilog } (2.5)] = 3.2 \times 10^{-3}$$

If the areas of the brickwork and window unit are 200 SF and 50 SF, respectively, then the weighted average sound Transmission Coefficient (r_{wg}) of the composite wall is calculated to be:

$$r_{wg} = [200\,(3.2 \times 10^{-5}) + 50\,(3.2 \times 10^{-3})] / [200 + 50] = 6.7 \times 10^{-4}$$

Hence the TL value of the composite wall is found from equation (11.1) to be:

$$TL = 10 \log (1 / 0.00067) = \mathbf{31.7\ dB}$$

It is therefore apparent that the effective Transmission Loss value of a composite construction is closer to the lowest transmission loss value of the component elements.

11.3.3 Sound Transmission Class (STC)

Since the Transmission Loss value of a sound insulation barrier varies with the frequency of the incident sound, an accurate evaluation of the sound insulation characteristics of a panel will require a detailed TL-frequency analysis. Apart from the effort involved, the results do not readily lend themselves to the comparison of alternative panel constructions. Also, a simple average of the TL over a range of frequencies is likely to be misleading because it underestimates the impact of the low values.

For this reason, the Sound Transmission Class (STC) was introduced to provide a single value that could be used to rate the sound insulation capabilities of a barrier. It is governed by ASTM standard E-90 (ASTM 1990), which specifies the precise conditions under which a panel must be tested to determine its STC rating. As shown in Figure 11.13, the test must be performed in two reverberation chambers[2] that are separated by a wall with a large rectangular opening.

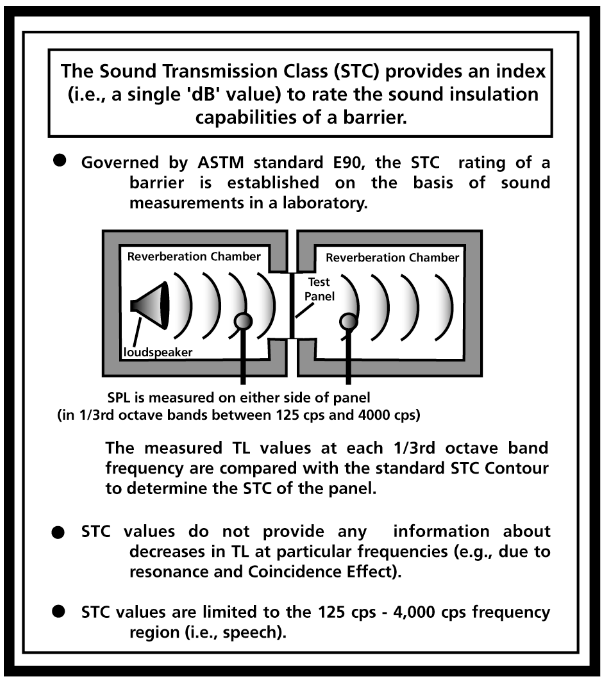

Figure 11.13: Sound Transmission Class

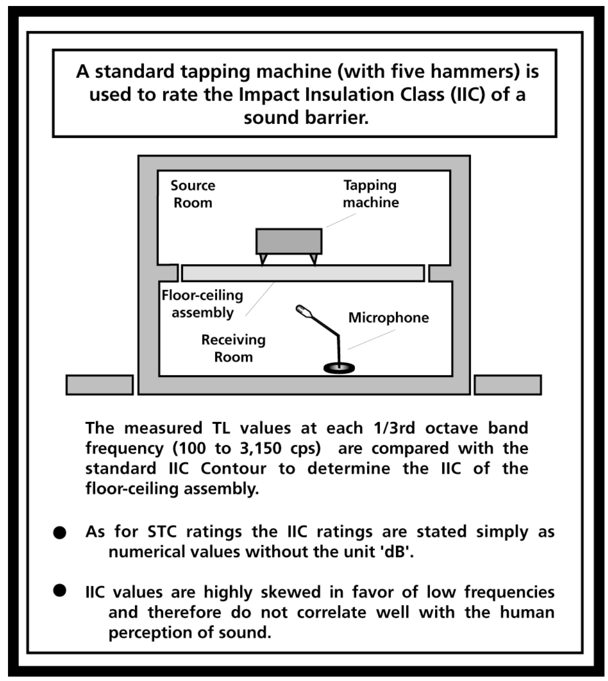

Figure 11.14: Impact Insulation Class

The panel to be tested is placed in the opening and utmost care is taken to carefully seal the perimeter of the panel so that there are no direct air-paths between the two reverberation chambers. A noise source of known frequency distribution is generated on one side of the panel and the transmitted sound is measured on the other side. Measurements are recorded at 16 one-third octave bands between 125 cps and 4,000 cps, and plotted on graph paper with the TL value on the horizontal axis and the frequency on the vertical axis.

The STC of the panel is then obtained by comparison with a standard *STC contour* (i.e., the *contour* is superimposed on the plotted graph) according to the following procedure. The standard *contour*, preferably drawn on tracing paper, is then moved up as high as possible by sliding it across the plotted

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[2] A reverberation chamber is a laboratory with acoustically hard surfaces that facilitate the reflection of sound (i.e., minimize sound absorption). The converse is an anechoic chamber in which all surfaces (including the floor) are highly absorptive to minimize any sound reflection.

test results until neither of two limits are exceeded:

Limit 1: The sum of the test values that exceed the standard *contour* is not greater than 32 dB.

Limit 2: No test value must exceed the standard *contour* by more than 8 dB.

The STC rating of the panel is then given as the TL value of the plotted test results that corresponds to the STC value at 500 cps on the standard *contour*. It is simply referred to as a number without the dB unit.

While the STC rating is a very useful measure of the air-borne sound insulation characteristics of a barrier it nevertheless suffers from two deficiencies. First, it is limited to the range of frequencies between 125 cps and 4,000 cps. This range may be exceeded in certain indoor situations where high pitched noise is encountered. Second, it does not provide any information about the actual shape of the TL-frequency curve and may therefore hide sudden dips in the curve.

11.4 Solid-Borne Noise Insulation

A major acoustical problem confronting architects is the elimination of noise originating through impact on a solid surface, such as footsteps, banging doors, and machinery vibration. The resultant energy is readily transmitted through the structure of the building, and large areas can be set in vibration giving rise to a high degree of radiated air-borne noise. Two factors of modern building construction have highlighted this problem, namely, the generally lower background noise levels in air-conditioned buildings and the use of lighter structures.

11.4.1 Impact Insulation Class (IIC)

While determination of the sound Transmission Loss of a wall is relatively straight-forward for air-borne noise, the matter is very much more complicated in the case of solid-borne noise, where the energy produced depends on the properties of the impacting force and the solid medium. The procedure that has been adopted internationally utilizes a standardized source of impact energy in the form of a mechanical device.

The solid-borne sound[3] equivalent to the STC rating for air-borne sound is the Impact Insulation Class (IIC). According to ASTM guidelines and similar to the STC test described above, the solid-borne sound insulation is measured by fixing the panel to be tested in an opening between two adjoining rooms. The sound source is provided by a standard tapping machine with five equally spaced hammers and the frequency range is set lower the 16 one-third octave bands between 100 cps and 3,150 cps. The lower frequency range is justified because the kind of noise generated by foot steps and vibrating machinery is normally at lower frequencies.

A typical floor-ceiling testing configuration is shown in Figure 11.14, with the tapping machine mounted above the test panel. The measured sound levels below the test panel are plotted on graph paper and compared with a standard *IIC contour*. Using a similar procedure and applying the same two limits that were described in Section 11.3.3 for the STC rating, the IIC rating is determined by the TL value of the plotted test results that corresponds to the IIC value at 500 cps

3 The terms solid-borne sound and structure-borne sound are synonymous.

on the standard *contour*. The only two differences in the procedure are that the *IIC contour* is moved vertically down from the top (while the *STC contour* is moved vertically up from the bottom) across the plotted test curve, and the final IIC rating is obtained by subtracting the corresponding TL value on the 500 cps line of the *IIC contour* from 110 dB. However, just as in the case of the STC rating, the dB unit is also omitted for the IIC rating. The greater the IIC rating the higher the solid-borne noise insulation provided by the barrier.

11.4.2 Methods of Solid-Borne Noise Insulation

Methods of solid-borne or structure-borne insulation differ substantially from those of air-borne insulation, and it is therefore essential that every effort be made to ascertain whether a disturbing noise originates from an air-borne or solid-borne sound source. Although the solution of each individual solid-borne source is unique, there are nevertheless, three well-proven general approaches which should be considered:

- The use of resilient floor covering to reduce impact induced vibration.
- The use of isolating, flexible mountings (or anti-vibration pads) for machinery, in conjunction with the provision of limp or spring-loaded connections between all ducts and vibrating machinery.
- The use of discontinuous systems of construction in the form of floating floors and completely isolated multiple walls.

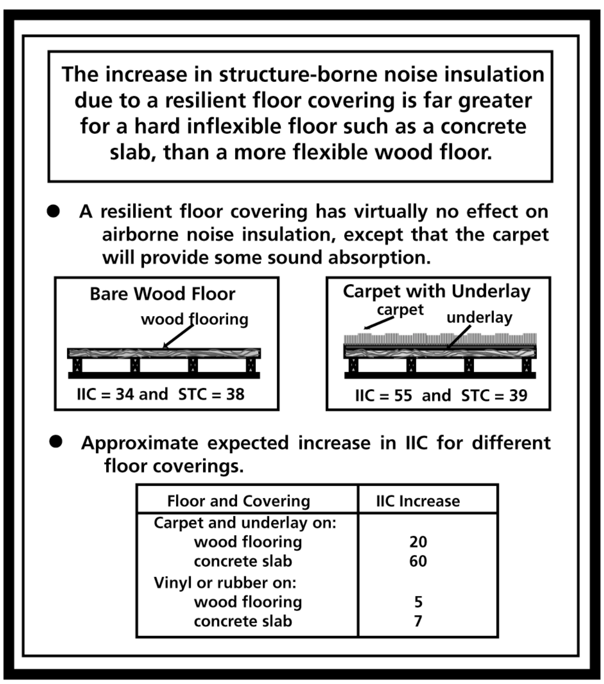

Figure 11.15: Impact of resilient floor coverings on structure-borne noise insulation

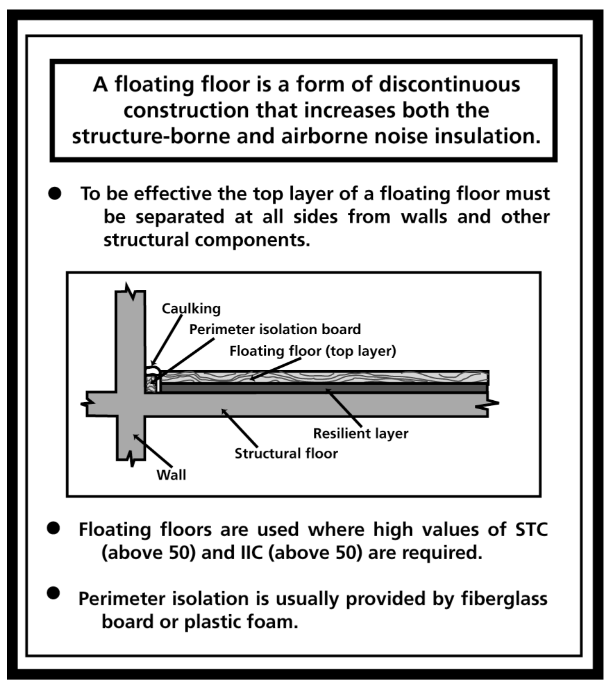

Figure 11.16: Discontinuous construction at the floor level of buildings

The reduction of impact noise from footsteps may be assisted by the use of resilient floor coverings. For example, the increase in IIC that can be obtained by laying a padded carpet on top of a concrete slab is in the vicinity of 60. The increase is less than 10 if the padded carpet is replaced by vinyl tiles. The advantages of carpet are not only high impact noise insulation and effective noise reduction at the source by absorption, but also psychological. Where floors have been carpeted in schools, the general decrease in impact noise has been accompanied by a softening of the normal institutional atmosphere and a greater concern for appearance and manners by the students. However, it should be noted that apart from increased sound absorption the carpet has virtually no effect on air-borne noise insulation.

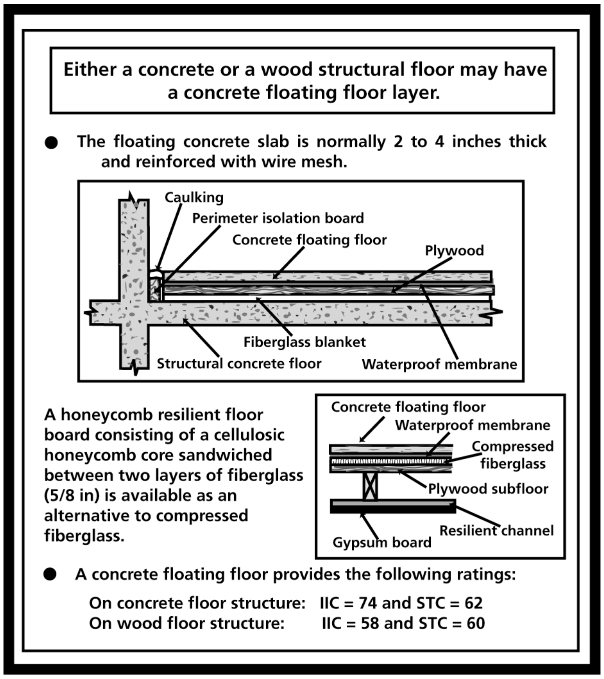

Figure 11.17: Concrete floating floor on concrete structural floor construction

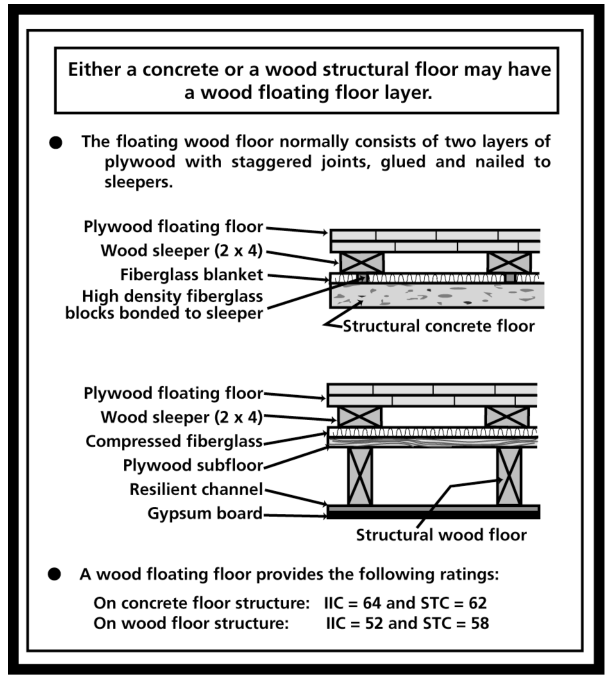

Figure 11.18: Wood floating floor on concrete or wood floor structure

Further impact noise insulation may be obtained by discontinuous floor and ceiling constructions. Floors can be constructed in multiple layers with resilient interlays or flexible mountings, so that the surface layer, which receives the impacts, is virtually a floating mass. Many systems of discontinuous floor construction are commonly in use, and some of these are shown schematically in Figures 11.17 and 11.18. For example, the concrete screed floor shown in Figure 11.17 is approximately 1½ IN thick, poured in-situ on a material that must have sufficient strength to remain resilient under the superimposed weight of concrete and normal building live loads. Compressed fiberglass may be used, but should be turned up at the walls to avoid any direct contact between the surface screed and the structural floor or perimeter walls. Alternatively, a honeycomb resilient floor board is often used. It consists of a cellulose-like honeycomb core sandwiched between two layers of fiberglass. In this case the separation at the wall boundary is achieved with a perimeter isolation board. Unfortunately, most resilient materials tend to compress after some years of use, and their effectiveness is thus reduced. If the base structure is a concrete slab, then the respective IIC and STC ratings are 74 and 62. However, for a timber base structure the ratings, particularly the IIC rating, are lower (i.e., 58 and 60, respectively).

Either a concrete or timber structural base may also have a wooden floating floor (Figure 11.18). In either case the flooring boards are nailed to timber sleepers (typically 2 IN x 4 IN laid on side), which rest directly on the resilient layer such as compressed fiberglass. In the case of a timber structure the gypsum board ceiling may be mounted on the underside of the floor beams using a resilient channel for additional impact isolation. It should be noted that while the STC ratings are only slightly lower for a timber solution (i.e., about 7%), the IIC ratings are significantly lower (i.e., about 23%). In other words, the greater mass and stiffness of concrete is useful for impact noise insulation but only slightly superior to wood construction for airborne noise insulation.

In the case of solid-borne sound produced by vibrating machinery, an attempt should be made to reduce vibration at the source, and if this fails the machine must be isolated from the building structure by means of flexible mountings. It is common practice to place between the source of the vibration and the supporting structure flexible elements, such as springs, rubber pads, or special multiple-layer isolators designed for specific applications. The amplitude of vibration of the source will be proportional to the force causing the vibration, and its own inertia, while the force that is transmitted to the building structure will depend on the dynamic characteristics of the source on its flexible mountings (Bradbury 1963). It is conceivable that an inappropriate choice of mounting stiffness could lead to an increase rather than a reduction of the transmitted force. As a fundamental rule, it is important that the natural frequency of the source on its flexible mounting should be small compared with the frequency of vibration of the source alone. For simple spring devices the degree of isolation is related to the static deflection of the spring when loaded. As a compromise between cost and vibration isolation, a static deflection given by equation (11.3) is often aimed for:

$$\textbf{Deflection (d)} \quad = \quad \mathbf{160 / [f_v^2]} \ \ (\text{IN}) \ \ \ldots\ldots\ldots\ldots\ldots\ldots \ \ \mathbf{(11.3)}$$

Where f_v cps is the lowest appreciable frequency of vibration.

Care must be taken that the spring has sufficient damping to control the vibration of the machine during starting and stopping. At these times, it is to be expected that the vibration of the source will be momentarily equal to the resonant frequency of the system. A steel spring can be damped by external means such as dash pots[4]. Unfortunately, the behavior of anti-vibration pads (e.g., rubber, cork, or felt laminates) cannot be predicted as simply as in the case of springs, since the static and dynamic stiffnesses differ in most materials. For many of these pads, damping is a function of frequency.

11.5 Noise Insulation in Practice

Where Transmission Loss (TL) values in excess of 50 dB are called for, such as recording studios and less noise sensitive building occupancies that are housed in buildings that are required to be located in very noisy environments such as near airports, the most appropriate acoustical solution may be *discontinuous construction*. This essentially requires the construction of two building shells (i.e., one inside the other). In the case of a timber frame this can be accomplished either with a double wall or the staggering of studs in a wider single leaf wall. Figure 11.19 compares the STC values of a normal single-stud wall (approximately 40) with a staggered-stud wall (approximately 50) and a double-wall (approximately 55). As an additional enhancement sound absorbing material can be added in the air cavity. The benefit of the

[4] A dash pot is a mechanical device that prevents sudden or oscillatory motion of a body by the frictional forces of a fluid (typically a piston moving horizontally or vertically within a cylindrical container that is filled with a liquid of known viscosity).

absorptive infill is an STC increase of only 3 for the normal single-stud wall, but a much greater increase in the case of the staggered-stud and double-wall construction alternatives (i.e., 9 and 11, respectively).

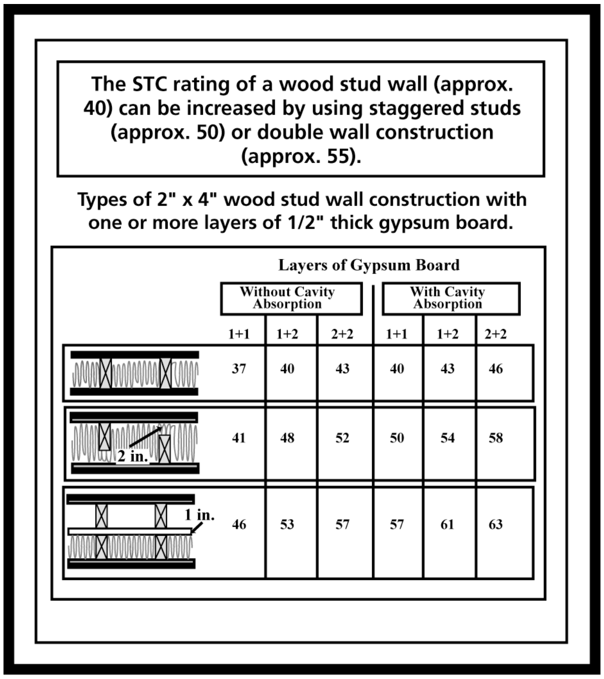

Figure 11.19: Alternative stud wall construction methods

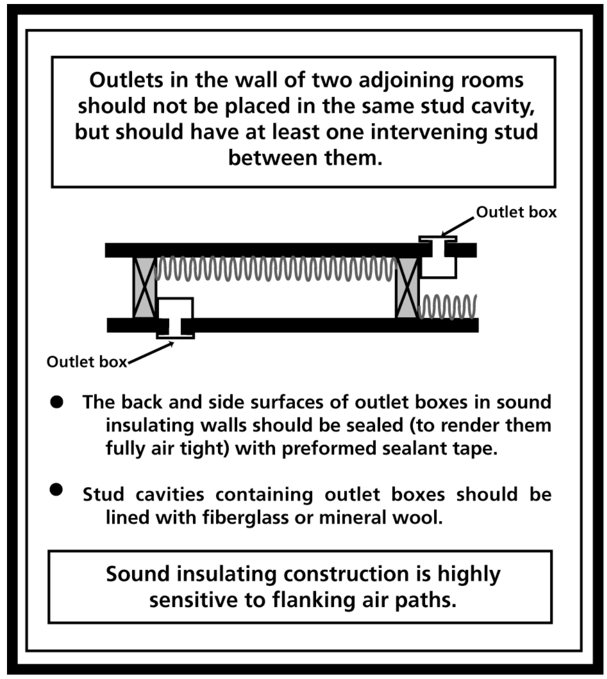

Figure 11.20: Proper sealing of electric power outlet boxes in walls

It is interesting to note that the benefit of adding a second layer of ½ IN thick gypsum board to one side of the wall (i.e., the "1+2" columns in Figure 11.19) is more than twice as much for the discontinuous construction alternatives than the normal single-stud wall, while in all three cases the increase in STC value is virtually the same as that provided by the cavity absorption infill. The reason for this is the extra mass provided by the second layer of gypsum board.

In the case of the floor the required constructional discontinuity can be provided by one of the alternative floating floor constructions described in Section 11.4.2 and depicted in Figures 11.17 and 11.18. However, as noted previously the STC values that can be achieved when the structural floor material is concrete are significantly higher than the corresponding values for a timber structure.

Special care must be taken to avoid air paths in walls that will lead to the direct transmission of noise from one room to another. This is likely to occur whenever electrical outlets for two adjoining rooms are placed opposite each other within the same stud cavity. As shown in Figure 11.20, the outlet boxes should be sealed with preformed tape around all sides and placed in separate stud cavities. In addition, it is considered good practice to line the stud cavities containing outlet boxes with fiberglass or mineral wool. A similar noise transmission vulnerability exists for small built-in cabinets that are desirable in bathrooms. For this reason, whenever noise insulation is an important design criterion, such cabinets should not be provided in a normal single-stud wall. Cabinets in a staggered-stud or double-stud wall need to be backed with gypsum board on the back and all sides.

A very common noise transmission problem is encountered in office buildings with suspended ceilings. Even though special care may have been taken at considerable expense to the building owner to use prefabricated partition walls with a relatively high STC value (e.g., multi-layered with an interstitial lead sheet) the result may be quite disappointing. As shown in Figure 11.21, the sound insulation will only be as good as the weakest link in the chain. In this case the effective STC value will be 20 rather than 40, because the noise will travel more readily through the ceiling than the partition.

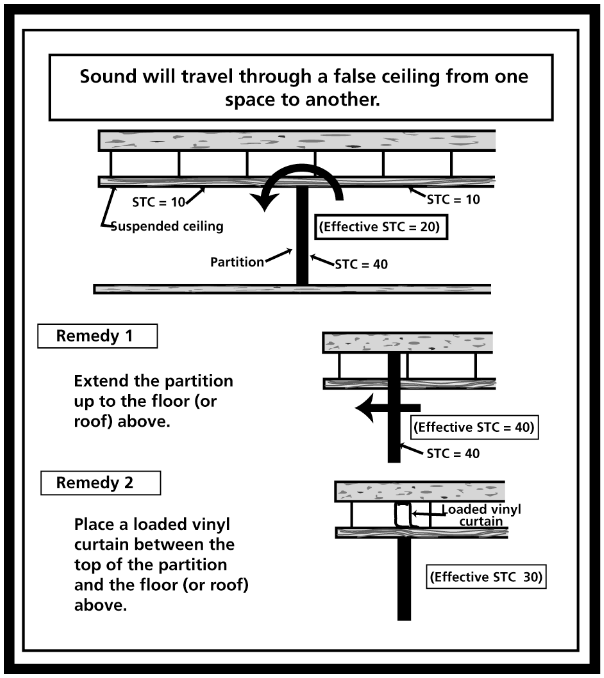

Figure 11.21: Common sound insulation problem between adjacent office spaces

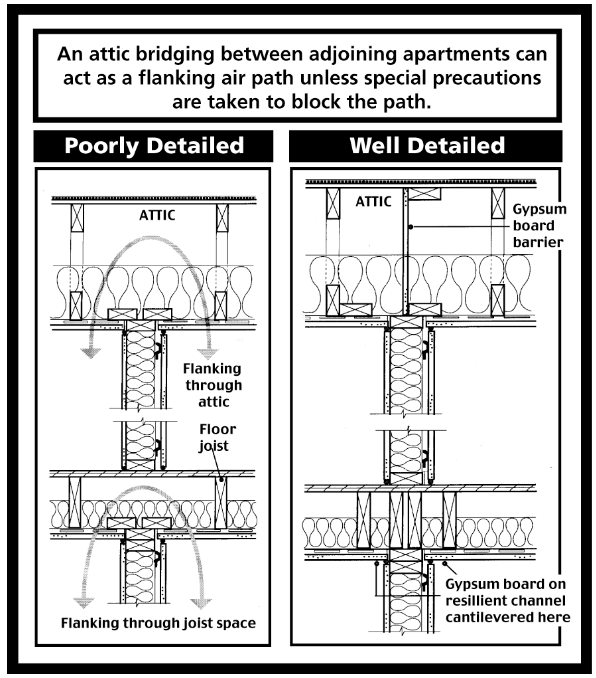

Figure 11.22: Common sound insulation problem between adjacent apartments

Neither of the two remedies suggested in Figure 11.21 are ideal. Extending the partition to the underside of the structural floor above inhibits the flexibility that was intended to be provided through the use of partitions in the first place. The partitions cannot be easily moved to achieve an alternative space layout, because the suspended ceiling is no longer continuous. Adding a loaded vinyl curtain in the space above the partition poses a rather awkward construction challenge and still leaves the ceiling as the weakest link in the chain.

Even more serious is the potential noise transmission problem that can occur between two adjoining apartments that share a common attic space. Here, as shown in Figure 11.22, it is essential that special precautions are taken to fix a solid noise barrier in the attic above the shared wall. Again, it is important that any direct air paths are eliminated by careful attention to the joints where the blocking panel meets the top of the shared wall and the roof above.

Finally, a word of caution about windows in external walls. A typical fixed window with a single ⅛ IN glass pane has a STC value of 29. However, if the window is openable then the STC value for the window in a closed condition is lowered by 3 to 5 points. Sliding glass doors are particularly prone to air flanking paths. While according to the Mass Law the thicker the glass the higher the TL value, this is not borne out in practice. Unfortunately, for thicker glass the critical frequency of the Coincident Effect is lower and this reduces the STC value disproportionally (Table 11.2). While the Mass Law suggests a TL increase of 6 dB with each doubling of thickness, the STC value increases by only 2 points.

Table 11.2: Reduced increase in STC value of glass of different thicknesses

Glass Thickness	Mass Law TL Value	Actual STC Value
⅛ IN	29 dB	29
¼ IN	35 dB	31
½ IN	41 dB	33

Needless to say, the simplest and most effective means of controlling noise inside building is through sound architectural design practices. In particular the following guiding principles are highly recommended:

- Separate noisy rooms from noise-sensitive rooms. For example, in multi-story apartments and condominiums, bedrooms should be separated from community corridors and lobbies.
- Use noisier rooms to buffer noise-sensitive rooms from external noise sources, such as vehicular traffic noise.
- Apply sound absorption to reduce the internally generated noise level in highly reflective (i.e., *live*) rooms. However, be aware of the Law of Diminishing Returns.
- Mitigate very loud internal noise sources through isolation and the treatment of surrounding surfaces with sound absorbing material.
- Ensure that the transmitted noise level is at least 5 dB below the background noise level of the receiving room, so that the background noise level will not be significantly raised.

Table 11.3: Recommended STC values for adjoining rooms

Primary Room	Adjoining Room	STC Value
classroom	classroom	45
	laboratory	50
	corridor/lobby	50
	music room	60
office	office	50
	general office	45
	corridor/lobby	50
	mechanical room	60
conference room	conference room	50
	office	50
	corridor/lobby	50
	mechanical room	60
bedroom	bedroom	55
	corridor/lobby	55
	mechanical room	60

11.6 Common Noise Sources

It appears that any definite attempt to control noise levels in the community must be accompanied by a determined effort in public education. The reluctance of courts of law and laymen to accept objective measurements as a criterion of annoyance is largely due to uncertainty, since the figures quoted have little relevance to most people. Sound level meters, like light meters, are relatively costly instruments and are therefore not readily available to members of the community. The situation is perhaps analogous to the hypothetical substitution of wet-bulb temperature as the principal index for weather forecasts. Thermometers are readily available to all persons. Therefore, while dry-bulb temperature is an inadequate measure of thermal comfort (i.e., neglects air-movement and relative humidity) it nevertheless relates to personal experience. Even though wet-bulb temperature is a more precise thermal measure (particularly for hot-humid climates) it has little relevance to the average person who would be unfamiliar with a sling thermometer. In the 1960s the City of Tokyo in Japan embarked on an interesting public education initiative. To combat the problem of increasing noise levels the City of Tokyo set up, for a trial period, sound level meters in streets and displayed the resultant noise level readings on large screens attached to adjacent buildings.

Chicago, a pioneer in the control of noise by town-planning legislation introduced noise level restrictions in defined areas in the early 1950s. The legislation met with favorable public reaction, no doubt at least partly due to the manner in which it was administered. On the basis of a complaint a city engineer would be sent to investigate the circumstances, without the assistance of a sound level meter. The engineer would inform the apparent offender that a complaint has been lodged and recommend that the noise level be reduced by taking certain steps. If a second complaint was received, the noise would be analyzed with a sound level meter to determine whether the noise infringed the current legislation. If so, the offender would be informed of pending legal action unless the noise was reduced within a specified period of time. Finally, if the offender still refused to act, the city would launch a prosecution that had every hope of being upheld in a court of law. Accordingly, the community became increasingly aware that it is possible to be prosecuted for producing excessive noise, with the result that persons tended to refrain from making unnecessary noise. This in itself was a worthy aim, since the community became educated to appreciate the nuisance value of noise.

11.6.1 Ventilation Noise

Ventilation and air-conditioning installations are normally responsible for two main noise problems, namely noise and vibration produced by the fan and motor assembly, and noise transmitted from one area to another by ducts. For convenience, noise sources in ventilation systems may be grouped into three principal categories:

- ***Mechanical noise:*** Mechanical noise sources such as rotating machinery, bearings, belts, and motors all produce both air-borne and solid-borne noise. The solid-borne noise is later radiated into the air of adjoining rooms from the surface of ducts. Although steps might be taken to reduce the noise levels at the source by improvement in balancing or machining, it is often more economical to merely isolate the ducts from the fan and motor assembly, and the latter likewise from the building structure. Flexible couplings are most effective between fans and ducts, although care must be taken to ensure that the coupling is not stretched during fitting, and remains flexible throughout its service life.

Vortex noise: Vortex noise due to air turbulence around fan blades, grilles and sharp bends forms the major portion of ventilation noise. Accordingly, high velocity systems (i.e., over 3000 FT/min) produce a considerable amount of vortex noise, especially if outlet grilles are fitted with guide vanes rather than open mesh.

Rotational noise: Rotational noise is generated by the mechanical action of fan blades producing fluctuating pressure changes that are transmitted along the main air stream as a series of pure tones. The principal frequency of rotational noise is related to the number of blades and the speed of the fan in revolutions per second. Obstructions close to the fan blades tend to aggravate the noise problem. The rotational noise level of both high-speed axial flow and low speed centrifugal type fans increases approximately linearly with the power (KW), so that for each doubling of power, the noise level increases by some 3 dB. On the other hand, the frequency spectrum of these two types of fans is fundamentally different, and depends as well on the size of the fan. While the centrifugal fan falls off steadily at about 5 dB per octave band toward higher frequencies, the axial fan rises to a peak in the middle frequency range (i.e., 200 to 1000 cps).

The frequency spectrum of the fan is an important consideration, particularly since nearly all noise reducing devices are more effective at higher frequencies. It is therefore apparent that from an acoustical point of view alone the centrifugal fan is more amenable to acoustic treatment despite the fact that it produces potentially more annoying higher frequency noise. The simplest method of obtaining an appreciable amount of sound absorption in ventilation systems is to line the duct walls with an absorbent material. Apart from a high absorption coefficient, such linings should have a smooth surface for low air friction and adequate strength to resist disintegration due to the continuous action of the air stream.

Heating, ventilation and air conditioning (HVAC) systems commonly used in buildings essentially fall into five categories, namely: window air conditioning units; fan coil units; roof-top units; packaged air handling units; and, built-up air handling units.

Window and fan coil units are commonly found in hotel rooms. Window air conditioners are typically located within a bench-like enclosure immediately below the window sill. They include a compressor and a fan as an integrated assembly, and require no ductwork. The air is simply drawn into the unit through an air inlet in the external wall and blown directly into the room through fixed angle vanes. While the noise produced by the compressor and the fan cannot be controlled by building design, it may sometimes have the redeeming quality of masking other undesirable noise sources by raising the background sound level in the room. However, window units are often a source of annoyance particularly if they are not well maintained (i.e., the noise produced by the compressor and fan tends to increase with the age of the unit).

Fan coil units are normally located at ceiling level and incorporate a fan that blows air over coils containing chilled or heated water that comes from a central plant. An alternative to centrally supplied hot water is electric resistance heating that is provided locally within the unit. Although fan coil units are typically located near an external wall with an air inlet, they operate mainly on the basis of recirculated air. The only sources of noise for this type of air conditioning unit are the fan and the possible vibration of the panels that enclose the unit.

By virtue of their name, roof-top air conditioning units are usually found on the roof of a building with the conditioned air delivered to the internal spaces through supply ducts. Return air is drawn into the unit from a ceiling plenum (Figure 11.23 (upper diagram)). These are self-contained systems complete with compressor, evaporative cooling coils, heat exchanger coils to extract heat from the recirculated air, filters, and ducts (i.e., for both supplying and recirculating air). Fan coil units are commonly used in single-story buildings, where the external roof to internal floor area is favorable.

Packaged air handling units (AHU) are much larger and typically prefabricated as fully assembled units in a factory. They are commonly used in medium-rise buildings to serve one or more floors through ductwork. Chilled water is pumped to the unit from a central plant and heating may be likewise supplied centrally in the form of hot water or steam, or provided locally through electric resistance heating. Fresh air is mixed with return air, filtered, adjusted for humidity, blown over chilled or heated coils as required, and then transmitted through the connected ductwork to the conditioned spaces (Figure 11.23 (lower diagram)). Most commonly a packaged AHU incorporates one or more thermostatically controlled mixing boxes that mix cooled air from the AHU with warm return air. The mixing boxes may have their own fans and essentially control the variable fan speed of the AHU through their demand for cool air. To minimize the solid-borne noise that might be produced by the fan-motor assembly it is either mounted directly on vibration isolators, or the entire packaged AHU is mounted on vibration isolation pads.

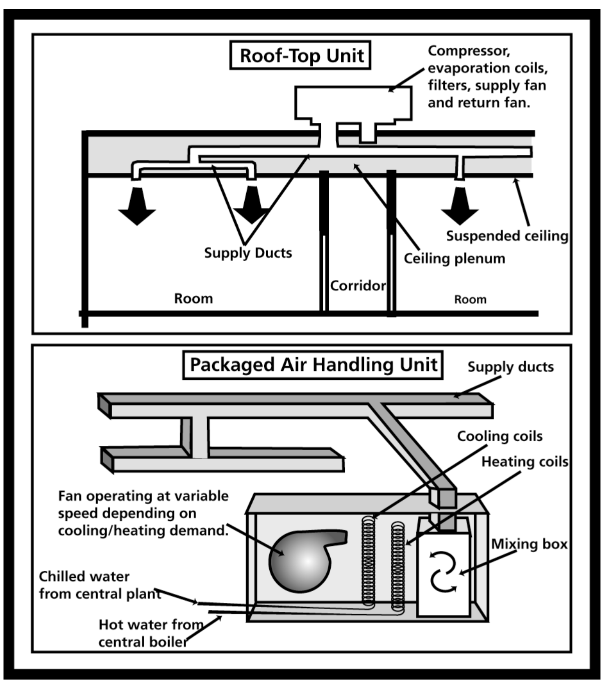

Figure 11.23: Small air-conditioning units

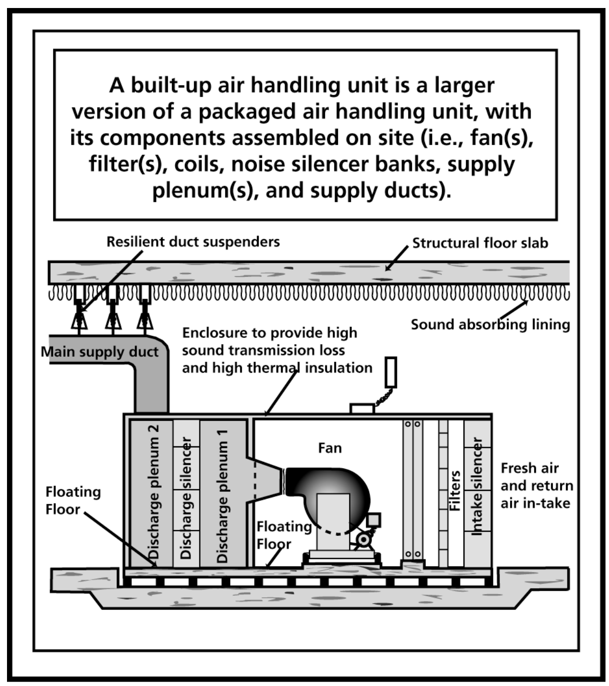

Figure 11.24: Built-up air handling units

A built-up AHU is very similar to a packaged AHU except that it is much larger and therefore assembled on-site. It may include more than one supply fan of the large centrifugal or axial type, and a whole bank of air filters and coils. A complete built-up unit may take up an entire floor of a high-rise building and will require a floating floor construction (Figure 11.24).

Most of the noise in a HVAC system of the roof-top and AHU type is produced by the fan. This noise is then potentially transmitted throughout the building by the air stream through the supply ducts and by the vibration of the duct walls. For this reason, it is common practice to line portions of the inside surface of a duct with sound absorption material such as fiberglass (e.g., 1 to 2 IN thick). The resulting reduction in sound (i.e., referred to as *attenuation* or *insertion loss*) is about 4 dB/FT at 1000 cps, but much less at lower and higher frequencies (i.e., around 1 dB/FT at 300 cps and a little less than 2 dB/FT at 4000 cps).

The air stream itself is also a source of noise. Air turbulence due to excessive air speed[5] or poorly designed duct transitions, such as connections and abrupt changes in diameter, will increase the noise generated by air flow. As a general set of guidelines, duct elbows should have rounded corners, right-angle duct branches should be provided with internal guiding vanes, and changes in duct cross-section should be gradually so that the transitional duct wall does not exceed a 15° angle in the direction of the air stream.

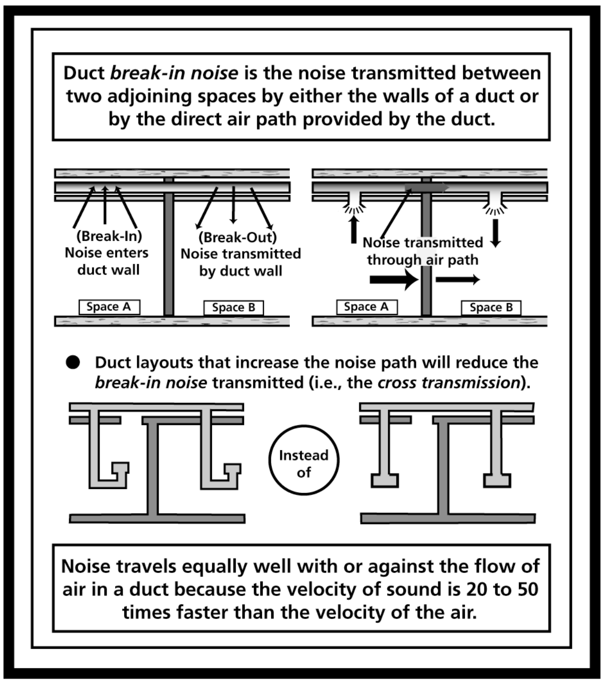

Figure 11.25: Duct *break-in* and *break-out* noise

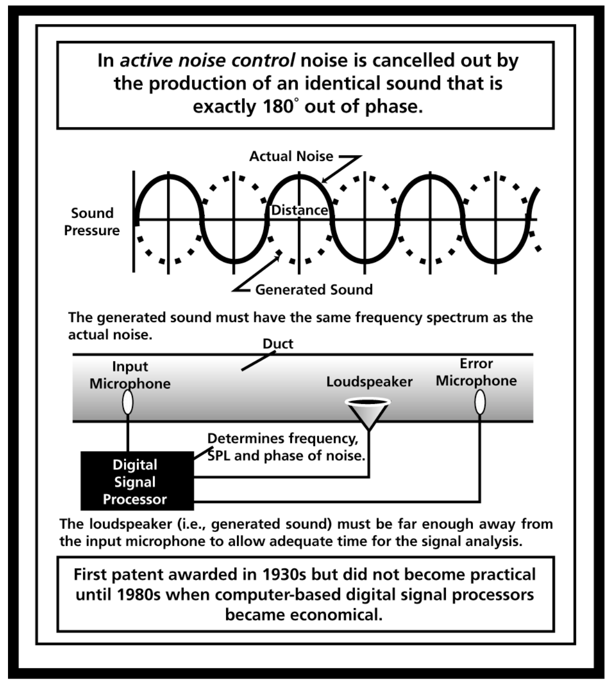

Figure 11.26: Active noise control

The vibration of the duct wall requires further discussion. When the noise generated by the fan is carried by the air stream through the ductwork some of the energy is converted into mechanical vibration of the duct wall. This is referred to as duct *break-out* noise and represents the fan noise that is transmitted by the vibrating walls of a duct and re-radiated into a space as air-borne noise. As might be expected, this noise transmission phenomenon is more serious in high velocity than low velocity systems. Duct shape is also a factor. The more compact (i.e., square) or circular the cross-section of the duct the less break-out noise is generated. Finally, duct stiffness reduces the tendency for the duct walls to vibrate and therefore decreases the break-out noise. To increase its stiffness sections of a duct may be *lagged* with fiberglass wrapping. A typical duct section that combines both the provision for *insertion loss* and break-out noise reduction would include a 1 to 2 IN fiberglass lining inside the sheet metal duct wall and fiberglass sheet wrapping with a loaded vinyl covering around the outside of the duct.

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[5] ASHRAE recommends maximum air speeds for rectangular and circular ducts based on the Noise Criteria (NC) rating of the space (ASHRAE 1989).

A duct can also act as a bridge to transmit noise that is generated in one space into an adjoining space. This is referred to as duct *break-in* noise. As shown in Figure 11.25, this kind of noise transmission can occur either through the walls of a duct or through the direct air path that is provided inside the duct. Since the speed of sound is at least 20 times faster than the flow rate of the conditioned air, the amount of noise transmitted is largely independent of the air flow direction. External insulation of the duct and duct layouts that increase the noise path will reduce the break-in noise transmitted to some degree. Fortunately, duct break-in noise is normally not a major concern because the reduction that can be achieved with these two measures is modest.

The ability to reduce or eliminate a sound through interference with another sound that has an identical frequency spectrum but is 180° out of phase has been considered conceptually by physicists and acoustic engineers for many years. Although a first patent was awarded in the 1930s, a practical implementation of this concept did not become possible until the 1980s when computer-based digital signal processing became economical. While the concept of *active noise control*[6] is quite simple its practical implementation depends on the ability of computers to analyze noise fast enough so that the interfering noise can be generated in time to cancel the offending noise. As shown in Figure 11.26, an active noise control system consists of a microphone located inside the duct, which is connected to a digital signal processor. The latter determines the frequency spectrum and SPL of the noise and generates the required 180° phase shift. The generated sound is then transmitted back into the duct by means of a loudspeaker in time to cancel out the original noise. A second microphone at a small distance behind the microphone provides feedback to the digital signal processor to increase the efficiency of the noise-cancelling operation.

In theory it should be possible to apply active noise control in any sound environment. However, currently its application is limited to the largely low frequency sound in HVAC ducts and sound cancelling earphones that are becoming increasingly popular with airline passengers. Unfortunately, further technical advances are necessary before digital signal processing can become a feasible noise control solution for open sound environments.

The general guidelines for controlling noise sources in ventilation and air-conditioning systems may be summarized as follows:

1. Mount fans and other HVAC equipment on anti-vibration pads.
2. Use flexible connectors for isolating fans from ducts to reduce duct *break-out* noise.
3. Select motors of higher horsepower than is necessary and operate them at less than maximum output.
4. Avoid sharp bends and use vanes inside elbows to reduce noise due to air flow in the main supply ducts.
5. Line duct sections internally with absorbent material, such as fiberglass.
6. Line ducts externally with lagging when they pass through noisy spaces to reduce *break-in* noise.
7. Seal around the edges of ducts when they pass through walls, floors, and ceilings.
8. Insert bends in secondary ducts to decrease cross-transmission (i.e., duct *break-in* noise) between adjoining spaces.

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[6] As opposed to noise insulation, which is referred to as *passive noise control*.

11.6.2 Industrial Process Noise

After World War II systematic surveys of industrial noise produced alarming results. In the US, Karplus and Bonvallet (1953) found that in some 40 different (noisy) plants over a wide range of industries, noise levels of between 90 dB and 100 dB were prevalent in about 50% of machine-operator positions. Their results indicated that noise generated in mechanical plants is particularly significant in the metal and aircraft industries, where chipping and riveting of large plates and tanks were found to represent the loudest individual noise sources. These findings led to subsequent psycho-acoustic surveys to determine the reaction of individual persons to industrial noise. Davies (1962) distributed questionnaires to more than 500 industrial firms engaged in noisy manufacturing processes. He found that:

- Up to 20% of all shop floor employees in these factories worked in surroundings where normal conversation was not possible.
- In the case of offices, under similar circumstances, less than 3% of the workers were affected.
- Methods of noise control were commonly used in the following order of frequency: isolation of machinery mountings (20%); silencers (15%); modifications to machinery (15%); segregation of noise source (5%); and, acoustical treatment and miscellaneous methods (15%).

These and other surveys have shown that industrial machine noise is caused primarily by manufacturing operations that rely on impulsive forces to form metals, abrasive materials to grind surfaces, saws and mills to cut material, and air pressure as a source of power. Forces due to impacts, out of balance disturbances, air flow, resonance, and so on, are set up in the machine, and produce air-borne sound directly or indirectly due to the vibration of components. Fluctuating gas flows found in combustion engines, compressors, pneumatic hammers, and safety valves are known to produce high intensity noise levels. Of less significance are noises due to friction and electrical circuits, such as the humming sound produced by transformers.

Systematic analyses of various typical industrial noise sources and their control were published in the 1950s by Aldersey-Williams (1960), Tyzzer (1953), King (1957), and Geiger (1955). Based on these recommendations and other sources (Beranek 1960, HMSO 1963, Harris 1957) steps were taken by governments, employers, and unions to ameliorate adverse noise conditions in industrial environments. Such measures, which soon became commonplace, included noise reduction at the source, substitution of a quiet process for a noisy process, and the mandatory wearing of hearing protectors by workers.

While at first sight the substitution of a very noisy industrial process with a less noisy process may appear to be more wishful thinking than reality, this was in fact found to be a viable approach. For example, as shown in Figure 11.27, the use of flame gouging instead of chipping on welded construction, and the substitution of press operations for drop forgings, will be accompanied by considerable reductions in noise level. Moreover, since noise resulting from impact is proportional to the rate of deceleration of the impacting parts (i.e., their mass, size, stiffness, and damping) it was found that resilient buffers could be used to advantage between the impacting surfaces (Aldersey-Williams 1960). At the same time, isolation of the impacting pieces from the machine frame reduced the risk of resonance. In the case of grinding operations, significant improvements were achieved by damping the component being ground with the aid of clamps or externally applied loads. For cutting tools, the noise level is directly related to the resonant properties of the assembly, while the frequency spectrum is largely determined by the rate at which the teeth of the cutting edge meet the surface being processed. Although it is almost impossible to influence this type of high frequency noise at the source, some marginal reduction has been achieved by applying large washers to either side of circular saw disks (i.e., thereby damping resonance).

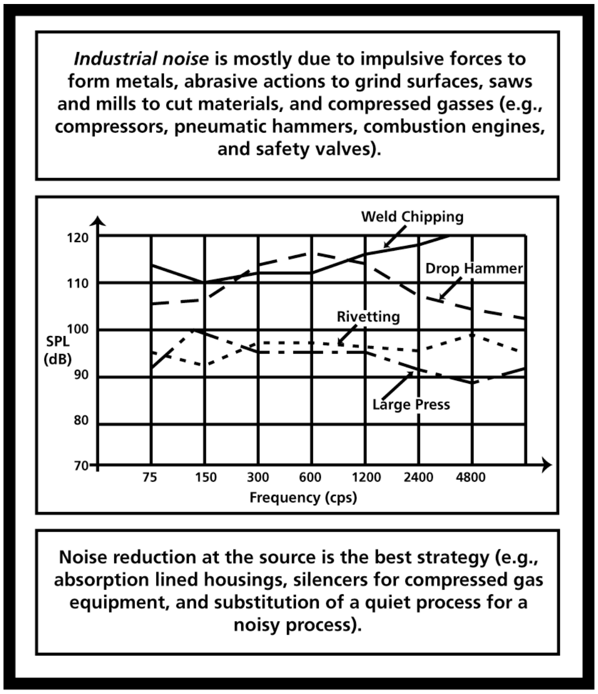

Figure 11.27: Industrial noise sources

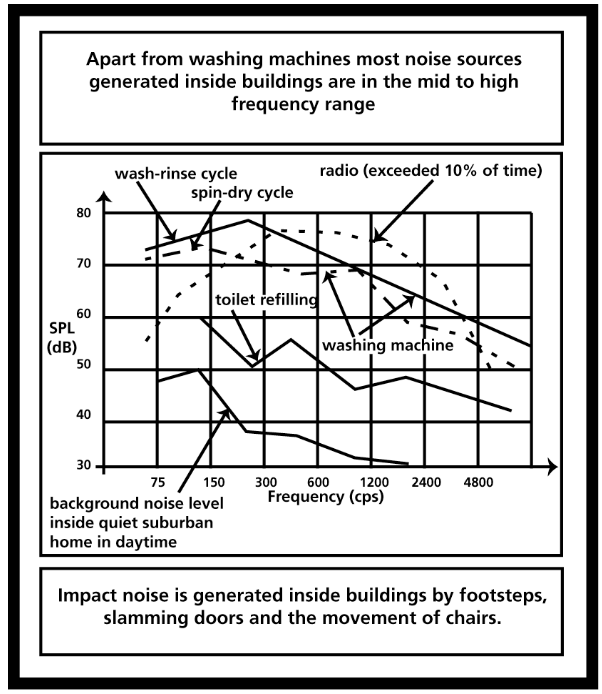

Figure 11.28: Residential noise sources

11.6.3 Residential Noise

Residential buildings are affected by sources of external and internal noise. Several surveys of residential noise sources were undertaken in the late 1950s and early 1960s. Some of the findings of two of these studies focused on internal noise sources, published by Van den Eijk in the Netherlands (1966) and Lawrence in Australia (1968), are combined in Figure 11.28. As might be expected, during this post World War II period radio noise was by far the worst offender. Today, with the prevalent use of earphones, the radio is no longer considered a major source of annoyance. As shown in Figure 11.28, with the notable exception of washing machines, most internal domestic noise sources predominate in the middle and high frequency sections of the spectrum.

Impact noise in residential buildings is usually the result of footsteps, movement of furniture, and door slamming. Unfortunately, many multiple dwellings (i.e., condominiums and apartment buildings) incorporate foyers and staircases with acoustically hard finishes that will readily transmit and reflect impact noise. Appropriate remedial measures include lining the walls and ceilings of these circulation spaces with sound absorbing material and covering floors and staircase treads with carpet. Furthermore, door jambs can be fitted with felt or rubber damping strips to reduce the effect of door slamming.

Referring again to Figure 11.28, a typical push-button water closet cistern can produce noise levels of 60 dB in the toilet compartment. Although this noise level may be reduced to less than 50 dB in an adjoining room it can still exceed the ambient background noise level by a sufficiently large margin to produce undesirable interference, particularly at night. The simplest way of dealing with this noise source is to reduce the rate of refilling the cistern, since the velocity of water is directly proportional to the noise level produced. Another source of annoyance in some older dwellings is the shock noises that can be produced by the sudden closing of a faucet. This is commonly referred to as a *water hammer* and is caused by a large volume of water being cut off suddenly. The remedy is to provide an air damper consisting of a length of pipe containing an air cushion at the dead end, mounted on the top of a tee-pipe section.

Finally, it is good practice to insulate pipes where they pass through or are fixed inside walls, so that solid-borne sound is less likely to be transferred. Under these conditions noise and heat insulation can often be economically combined[7].

11.6.4 Vehicular Traffic Noise

Although aircraft can constitute the most severe external noise source in the vicinity of airports and under direct flight paths, for the majority of city and suburban sites vehicular road traffic will remain the most important and persistent source of external noise. Since road traffic noise is subject to considerable fluctuations due to changes in speed, age, size, and general state of repair of a vehicle, surveys have generally been expressed on a statistical basis. Table 11.4 lists these findings in the form of noise levels recorded for 80% of the time. Accordingly, for 10% of the time noise levels were found to be higher and for the remaining 10% of the time they were found to be lower than the range shown in the Table. Although mean traffic noise levels are acceptable as a guideline, it is important to also know the maximum levels that will occur whenever a vehicle passes the listener.

Table 11.4: Vehicular traffic noise (expected 80% of the time)

Type of Road and Environment	Expected Noise Levels (80% of the Time)	
	Day (8 am – 6 pm)	Night (1 am – 6 am)
Highways and freeways	75 – 85 dBA	65 – 70 dBA
Major heavy traffic roads	65 – 75 dBA	50 – 60 dBA
Residential roads	55 – 65 dBA	45 – 55 dBA
Minor roads	50 – 60 dBA	45 – 50 dBA
Residential side streets	50 – 55 dBA	40 – 45 dBA

It was discussed previously in Chapter 9 (Section 9.3) that a point source of sound in a free field is reduced by some 6 dB for each doubling of the distance between the source and the listener. Unfortunately, traffic noise is more accurately described as originating from a line source, which means that the attenuation with distance is only about 3 dB for each doubling of distance. Further, it should be mentioned that, particularly in city and suburban areas, the noise produced by freely flowing traffic cannot be rightly considered as an accurate design criterion. In these areas, traffic is subjected to stopping and starting due to traffic conditions and intersection controls. When accelerating from a stationary

[7] This does not contradict previous statements in this chapter that draw attention to the fallacy of assuming that thermal insulation will also serve well as sound insulation. In this particular case, the sole acoustic purpose of the pipe wrapping is to isolate pipe vibrations from the structural components of the building.

position, low gears are used and engine speeds (i.e., revolutions) tend to be high. It must therefore be expected that sound pressure levels are higher under these conditions, accompanied by a slight shift of the frequency spectrum toward higher frequencies.

Basically, the noise produced by vehicular traffic can be reduced in four ways, namely: by reducing the number of vehicles; by eliminating conditions that are conducive to the noisy operation of vehicles; by limiting the noise produced by individual vehicles; and, by isolating traffic noise either close to the source or near the listener. Although there appears to be little difference in maximum noise level whether one or more vehicles are passing a given point, the percentage of near maximum readings is greatly increased for heavy traffic flows.

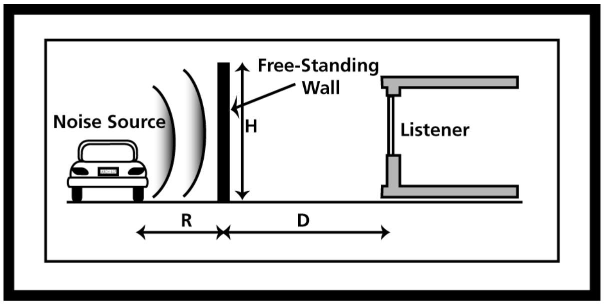

Figure 11.29: Parameters considered in the calculation of the noise reduction provided by external free-standing walls

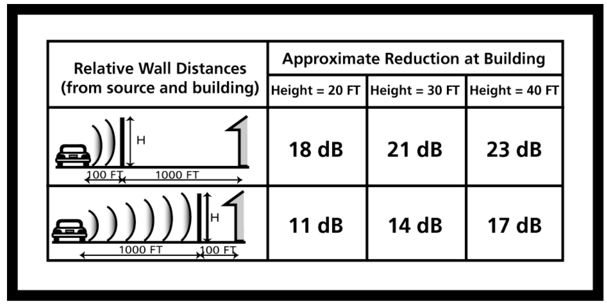

Figure 11.30: Impact on noise reduction of the height and distanced from building of an external free-standing wall

Present restriction of certain suburban roads to passenger and light commercial traffic, while based on utilitarian premises, could be extended as a means of ameliorating noise conditions. In city and suburban traffic, maximum noise levels occur during acceleration in low gear, whether this is due to a sudden increase in traffic speed or grade. Significant improvements in traffic flow tend to be costly and often difficult to achieve in existing cities where most land is privately owned. The reduction of noise produced by individual vehicles has long been considered the most fruitful approach. For petrol engines, the noise level is directly related to the engine load and road speed, while in the case of diesel engines there is much less variation (Lawrence 1968, Priede 1962). However, there are other sources such as tire noise at high speeds on highways and freeways, horns, and the rattling of bodywork in empty heavy commercial vehicles that need to be considered.

In most cases the reduction of traffic noise is achieved mainly at the listener's end by improving the insulation of the external building walls. Since traffic noise has significant low frequency components, systems of construction incorporating heavy cavity walls and double-glazed, plate glass windows with substantial cavities (e.g., 8 in.) are most effective. If a number of closely grouped buildings are involved in the same communal traffic noise problem, consideration might be given to cuttings or a series of free-standing barriers alongside the offending traffic way. Such noise barriers are now quite common in cities to shield residential developments from freeway noise. As shown in Figures 11.29 and 11.30, the approximate noise reduction that can be achieved by such a free-standing sound barrier can be calculated with the following equation:

$$\textbf{noise reduction} = 10 \log_{10} (20 \times (Z / K)) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad \textbf{(11.4)}$$

where: $Z = 2\,[R\,((1 + (H^2 / R^2))^{1/2} - 1) + D\,((1 + (H^2 / D^2))^{1/2} - 1)]$

and: $K = W\,(1 + (H^2 / R^2))$

for: H = height of free-standing noise barrier (FT)
R = distance of barrier from noise source (FT)
D = distance of barrier from listener (FT)
W = wavelength of noise source (FT); – i.e., 1100 (FT/sec) / frequency (cps)

To be effective free-standing sound barriers need to be at least 20 FT high and are therefore required to withstand considerable horizontal wind forces. For a 20 FT high wall subjected to a wind speed of 50 MPH the horizontal design load due to wind forces will be approximately 125 LB per vertical lineal FT of wall[8]. The walls are often slightly concave toward the noise source to increase their effectiveness and structural stability. It is also important that the wall is placed as close to the noise source as possible. As shown in Figure 11.30, the difference in effectiveness of a sound barrier that is located 100 FT from the noise source and one that is placed 1,000 FT from the noise source is 39% for a 20 FT high wall, 33% for a 30 FT high wall, and 27% for a 40 FT high wall.

11.6.5 Trees and Shrubs

Field measurements have verified that trees and shrubs can reduce external noise levels by 5 to 8 dB, which the human ear may perceive as a 50% reduction in noise. In particular, carefully planned tree belts in combination with shrubs can provide a moderate buffer for vehicular traffic noise in residential areas. The following guidelines should be followed:

- Tree belts should be 20 to 100 FT wide depending on the severity of the noise source. Evergreen trees with dense foliage are recommended for year-round noise protection.
- Select taller trees (if permitted) with lower-level shrubs (i.e., 6 to 8 FT high) in front of and between the trees. A soft ground cover such as tall grass is preferable to paving. The trees and shrubs should be planted as close together as the plant species allow.
- The trees and shrubs should be planted as close to the noise source as possible. The same rules as for free-standing sound barrier walls apply in this regard. Ideally, the vegetation belt should be within 20 to 50 FT of the noise source.
- The tree and shrub belt should be at least twice as long as the distance from the area to be protected. In the case of vehicular traffic noise, the vegetation belt should extend a significant distance in both directions parallel to the road or highway.

Wind is also a factor that can influence the effectiveness of a tree belt. If the prevailing winds come from the direction of the noise source, then the ability of a vegetation belt to act as a noise barrier will be somewhat reduced[9].

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[8] The static pressure (P LB/SF or psf) resulting from a wind speed of V mph at 90° to the surface of a wall can be calculated using the equation: $P = 0.00256\ (V^2)$, which may be approximated to $P = V^2/400$ (Mehta 1997). Therefore, for a wind speed of 50 mph the static pressure on the wall is (50 x 50)/400 or 6.25 psf or 125 LB per vertical lineal foot of wall.

[9] The density of foliage and the respective distances of the tree belt from the noise source and the listener appear to be the principal determinants of noise reduction.

11.6.6 Aircraft Noise

During the 1950s and 1960s, as existing airports were expanded and new airports were built to accommodate an increasing volume of commercial passenger and air freight transportation, aircraft noise emerged as one of the most severe and politically activated sound control problems of the late 20th Century. In response a number of surveys and research studies were undertaken to develop subjective rating scales for aircraft noise HMSO 1963, Bishop 1966, Bowsher et al.1966). In the US the Composite Noise Rating (CNR) scale was proposed for relating maximum noise levels with the number of occurrences, while in Germany the *Q* scale was chosen to represent the total weighted noise energy reaching a given point on the ground during any specified period.

It is apparent from these scales that the level of aircraft noise is less important than the number of times the disturbance occurs. Accordingly, in England, on the basis of the Wilson Report (HMSO 1963), the Noise and Number Index (NNI) was introduced, which for the first time allowed an acceptable aircraft noise rating to be established.

Experiments dealing with the subjective judgment of the noise that occurs during aircraft flyovers has resulted in the concept of Perceived Noise (PNdB) level (Kryter and Pearsons 1963). The perceived noise level is calculated from the spectrum of the noise, so that for aircraft flyovers the PNdB value would correspond to the maximum sound pressure level. Experimental data has shown that, if *duration* is defined as the time during which the sound pressure level is within 10 dB of its maximum value, a doubling of duration will produce an increase of 3 dB in the PNdB value (Pearsons 1966). A few years later the perceived noise level was adjusted to allow for the subjective influences of discrete tones as well as the duration of the noise. This became known as the Effective Perceived Noise (EPNdB) level, and has remained to date as the most accurate measure of individual annoyance.

11.7 Questions Relating to Chapter 11

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. **According to the Mass Law the transmission loss of a sound barrier increases by about *X* dB with each doubling of mass.**

   A. X = 2 dB
   B. X = 10 dB
   C. X = 20 dB
   D. X = 50 dB
   E. All of the above (i.e., A, B, C and D) are incorrect.

2. **In practice the actual insulation provided by a sound barrier is related not only to its mass but also to:**

   1. The angle of incidence of the sound.
   2. The frequency of the sound.
   3. The stiffness of the barrier.

4. The presence of direct air paths.
5. The loudness of the sound.
6. The sound pressure level of the background noise.

A. Statements 1, 3 and 4 are correct.
B. Statements 1, 2, 3 and 4 are correct.
C. Statements 1, 2, 3, 4 and 5 are correct.
D. Statements 1, 2, 3, 4, 5 and 6 are correct.
E. Statements 1, 2 and 3 are correct.

3. **If a sound barrier is forced to vibrate at its natural frequency then its insulation value is:**

A. Slightly increased.
B. Slightly reduced.
C. Greatly reduced.
D. Greatly increased.
E. Relatively unaffected.

4. **Which (if any) of the following statements is <u>not correct:</u>**

A. The ideal sound barrier material is one which is heavy, limp and highly damped.
B. Although the velocity of sound in air is constant for all frequencies, the velocity of induced bending waves in solids increases with higher frequencies.
C. The cavity wall or double-leaf partition potentially offers the greatest scope for noise insulation, since it provides close to twice as much transmission loss as each leaf separately.
D. A normal 2-inch air-cavity which is most effective for heat insulation purposes might provide no more than a 2 dB increase in sound insulation.
E. More than one of the above statements (i.e., A, B, C and D) are incorrect.

5. **Which of the following <u>constructional details</u> would you specify for the external wall of a building to achieve a sound transmission loss of 55 dBA.**

A. Minor structural connections between the two leaves of a cavity wall (e.g., tie wires).
B. A thin blanket of absorbent infill in the cavity.
C. Double glazing with a two-inch cavity.
D. Completely discontinuous construction.
E. I would specify A, B and C only.

6. **The <u>sound spectrum</u> of vehicular traffic noise:**

A. Is mainly high frequency.
B. Is mainly low frequency.

C. Contains all frequencies (in the audible range) approximately equally.
D. Varies substantially from one vehicle to another.
E. None of the above statements (i.e., statements A, B, C, and D) are correct.

7. **Which of the following systems of construction would you recommend for a broadcasting studio situated directly under the flight path of a major airport?**

A. Solid double brick walls plastered on both sides.
B. Brick cavity walls and floating floors.
C. Discontinuous wall and floor construction.
D. Discontinuous wall, floor, and roof construction.
E. None of the above should be recommended.

8. **The Mass Law that governs the transmission loss of a sound barrier makes assumptions that are only partially realized in practice. Which of the following does not influence the effective sound insulation provided by a single-leaf partition?**

A. The stiffness of the partition.
B. The angle of incidence of the sound.
C. The boundary conditions (i.e., the manner in which the partition is attached to adjoining walls, the ceiling, and the floor).
D. The height of the partition.
E. All of the above (i.e., A, B, C, and D) influence the sound insulation performance of a single-leaf partition in practice.

9. **If a sound barrier contains more than 10% of openings then its overall sound transmission loss will not be more than *X* dB regardless of the thickness and mass of the solid portion of the barrier.**

A. X = 10 dB
B. X = 15 dB
C. X = 20 dB
D. X = 25 dB
E. All of the above (i.e., A, B, C and D) are incorrect.

10. **The Coincident Effect occurs at a Critical Frequency, which is a function of the following material properties:**

A. Density.
B. Modulus of Elasticity.
C. Thickness.
D. All of the above (i.e., A, B, and C) are correct.

E. None of the above (i.e., A, B, C, and D) are correct.

11. **Although the solution of each individual solid-borne noise problem in buildings is unique, the following well-proven general approach should be considered:**

A. Damping the sound (or vibration) source by reducing its mass (i.e., weight).
B. Isolating the sound (or vibration) source from the building structure.
C. Increasing the stiffness of the platform on which the sound (or vibration) source is mounted.
D. All of the above approaches (i.e., A, B, and C) should be considered.
E. None of the above statements (i.e., A, B, C, and D) are correct.

12. **General guidelines for controlling noise sources in ventilation and air-conditioning systems include:**

A. Use flexible connectors for isolating fans from ducts to reduce duct *break-out* noise.
B. Select motors of higher horsepower than is necessary and operate them at less than maximum output.
C. Avoid sharp bends and use vanes inside elbows to reduce noise due to air flow in the main supply ducts.
D. All of the above guidelines (i.e., A, B, and C) are correct.
E. None of the above statements (i.e., A, B, C, and D) are correct.

13. **If a point source of sound in a free field is reduced by about 6 dB for each doubling of distance between the source and the listener then what would be the equivalent reduction in the case of a line source such as a moving truck on a freeway?**

A. 6 dB
B. 5 dB
C. 4 dB
D. 3 dB
E. All of the above (i.e., A, B, C, and D) are incorrect.

14. **Which (if any) of the following statements are correct? A *reverberation chamber* is an acoustic laboratory:**

A. In which only the ceiling and the floor are treated with sound absorbing material, while the wall surfaces are acoustically *hard* surfaces.
B. In which all surfaces are treated with sound absorbing material.
C. In which all surfaces are designed to reflect sound.
D. All of the above statements (i.e., A, B, and C) are correct.
E. None of the above statements (i.e., A, B, C, and D) are correct.

15. **Sound Transmission Class (STC) was established to provide a single value that could be used to rate the sound insulation capabilities of a barrier. Which (if any) of the following statements is <u>not</u> correct?**
    - A. The Transmission Loss (TL) value of a sound barrier varies with the frequency of the incident sound.
    - B. A simple average of TL values is likely to underestimate the influence of low frequency noise.
    - C. STC rating tests are performed in two anechoic chambers that are separated by a wall with a large rectangular opening. The test specimen (e.g., a wall segment) is placed in the opening and care is taken to seal the perimeter of the specimen to ensure that there are no direct air-paths.
    - D. All of the above statements (i.e., A, B, and C) are correct.
    - E. None of the above statements (i.e., A, B, C, and D) are correct.

16. **Which (if any) of the following statements is <u>not</u> correct in respect to *floating floor* construction?**
    - A. To be effective the top layer of floating floor must be separated at all sides from walls.
    - B. The resilient layer within a floating floor construction should be as thin as possible so as to decouple the *floating* top layer from the structural base layer.
    - C. Floating floors are used where high values of STC (above 50) are required.
    - D. All of the above statements (i.e., A, B, and C) are correct.
    - E. None of the above statements (i.e., A, B, C, and D) are correct.

17. **Which (if any) of the following statements is <u>not</u> correct? In discontinuous timber wall construction staggered studs are often employed because:**
    - A. They are structurally more rigid than normal stud construction and rigidity is highly desirable for sound insulation.
    - B. They result in a wider cavity between the two opposite sides of a wall, which is desirable for sound insulation particularly if the cavity is filled with sound absorbing material.
    - C. They provide a higher STC value.
    - D. All of the above statements (i.e., A, B, and C) are correct.
    - E. None of the above statements (i.e., A, B, C, and D) are correct.

18. **Which (if any) of the following statements relating to free-standing external sound barriers are correct?**
    - A. The barrier should be located as close to the noise source as practical.
    - B. The barrier is unlikely to provide a noise reduction of more than 25 dB.
    - C. The barrier should extend a reasonable distance beyond the noise source at either end.
    - D. All of the above statements (i.e., A, B, and C) are correct.

E. None of the above statements (i.e., A, B, C, and D) are correct.