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Acoustic Design

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Chapter 1

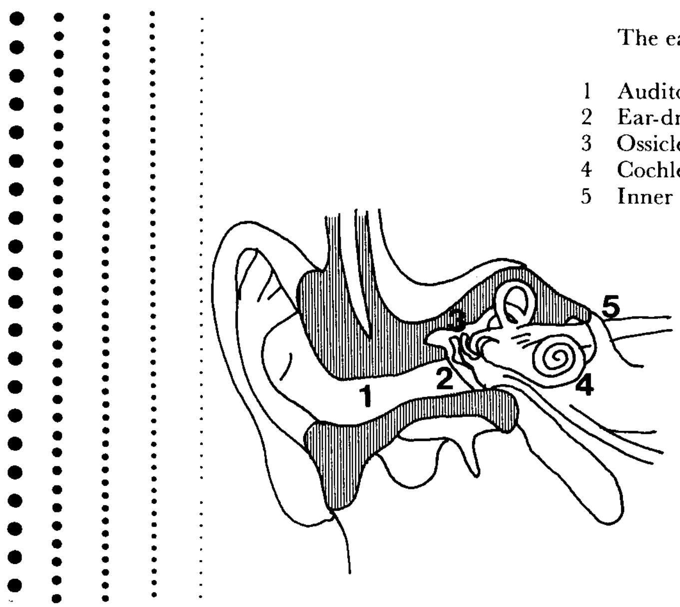
Perception of sound

1.1 Hearing

1.1.1 The ear

The human ear forms a 'microphone to the brain'. Most people are familiar with its component parts and the mechanics of the hearing process (Figure 1.1). What is more mysterious is the interface of ear and brain and the interpretation by the brain of the physical excitation of small, hidden parts of the body.

Air-borne soundwaves enter the ear via the auditory canal and travel down to the ear-drum, or tympanic membrane. The Eustacian tube at the other side equalises general atmospheric pressure conditions either side of the ear-drum. The sound-waves cause the drum to pulse, driving small bones called ossicles which pass the sensation via the oval window to the inner ear. The cochlea within is a hollow bone filled with liquid. The inner ear has about 25,000 nerve endings to relay the vibration or sound information to the brain.



The ear

1 Auditory canal
2 Ear-drum
3 Ossicles
4 Cochlea
5 Inner ear nerve ends

Figure 1.1 Component parts of the human ear.

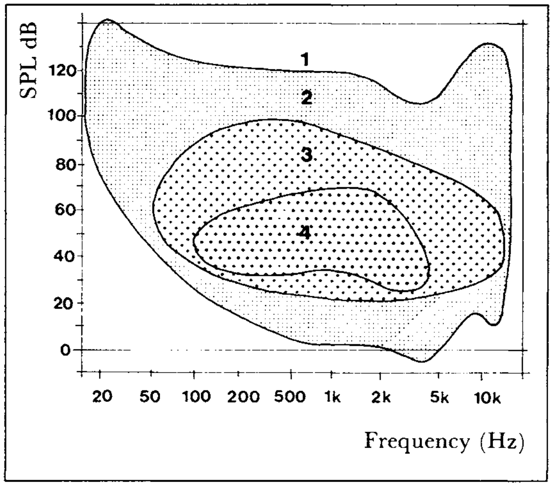


Figure 1.2 Dynamic range of human hearing, 16 – 20,000Hz.

Audible ranges

1 Threshold of feeling
2 Audible range
3 Music
4 Speech

Dynamic and frequency ranges

Each of us has a different 'dynamic range' to our hearing, this being defined as the ratio of just tolerable sound intensity and just audible sound intensity, as indicated in Figure 1.2. The young and healthy enjoy a frequency range of 16–20,000Hz spanning no less than eleven octaves. (Hz, or Hertz, is the number of vibrations or cycles, i.e. to and fro movement, completed per second.) Beyond the age of twenty-five, however, we lose sensitivity in the upper frequencies (Figure 1.3). This effect of progressive hearing loss with age is called presbycusis, a natural process of wear intensified by living and working in noisy conditions.

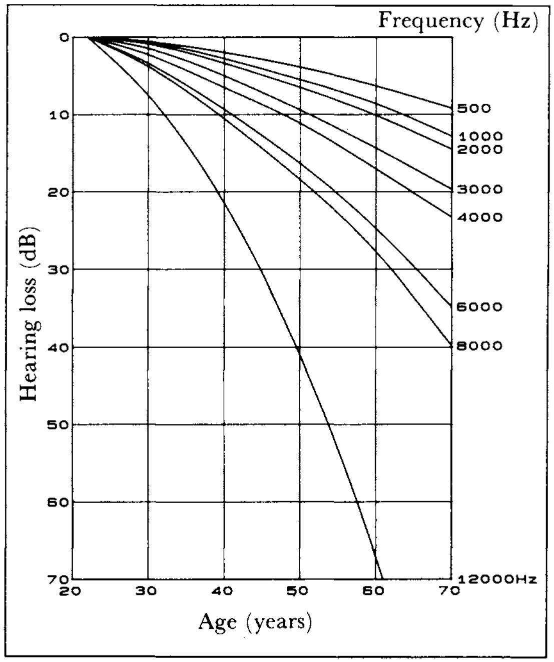


Figure 1.3 Presbycusis – gradual hearing loss with increasing age.

Hearing damage

Physical damage to the ear through high noise levels occurs not through rupture of the ear-drum, although this may happen in the case of explosion overpressure, but through damage to delicate hair cells in the inner ear. Regular noise dosage can also, in some cases, cause resetting of the ossicles. Groups living in cities exhibit angular displacement from what was considered to be the quietest rural, or pre-Industrial Revolution, norm. (This displacement is in the order of 30 per cent of the total displacement potential.) It is thought that many city dwellers, therefore, may have irreversible ossicle displacement sufficient to desensitise the quality of hearing.

Infrasound and ultrasound

Low frequency vibration below the 16Hz lower limit for hearing is known as infrasound. The whole body as much as the ear responds to the vibration although the level has to be very high in order to produce a response. The region above 15,000Hz is the ultrasonic region and the average human is quite insensitive to such high frequencies.

1.1.2 Directivity

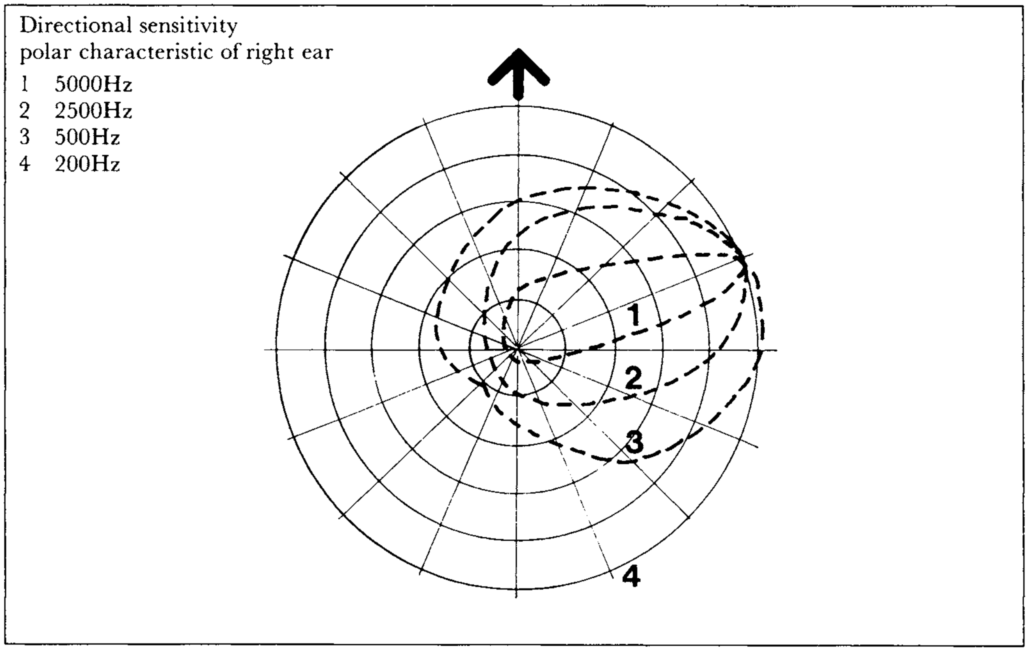
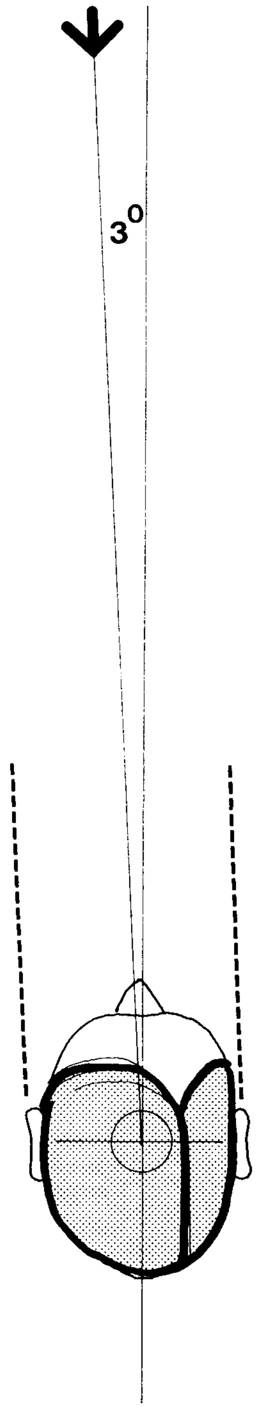


Figure 1.4 The ear as a directional microphone: sound sensitivity field to right ear (L H S mirror image pattern).

Rather than a single ear, we should consider a pair. As with our eyes, we have two coordinated receptors, and for the same reason, so that they can act as a team in collecting information for the brain. Stereophony is to hearing what stereoscopic vision is to sight. The ears have developed on opposite sides of the head which helps in picking up information from sides as well as front (Figure 1.4).

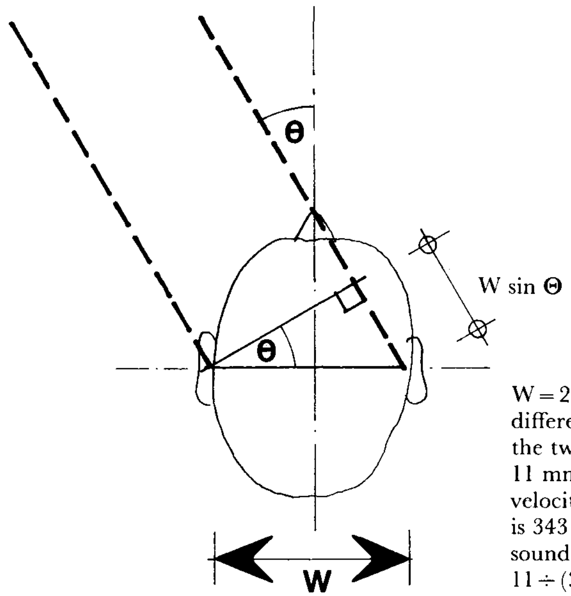


Directional information was essential to our forebears in order for them to locate what they were hunting, or to know how to escape a predator. Since this all took place on the jungle floor, give or take a few metres into the air, this sense is particularly developed in the horizontal rather than the vertical axis.

Acuity

It has been found by experiment that a deviation from frontal sound of only 3° to axis can be clearly detected. Also, it is precisely at slight variations to frontal sound that our sense of direction is at its most acute. We may hear a sound to one side (the individual ear hears best for sound normal to the head – the penalty for glancing frontal incidence is about 5dB). We turn naturally to face a sound attracting our attention and 'home in' on its exact direction. By use of both ears 'focusing' on the sound we then have a good idea of the distance of the sound source. It can be demonstrated (Figure 1.5) that the slight variation in time engendered by this acuity is only 30 msec.

Certain animals have more acute hearing than man, and can hear very weak impulses at higher frequencies. This is due to a shift in their frequency sensitivity range and the use of movable auricles, which can be turned at will to the sound. Man's rotary mechanism has all but disappeared (although some can still 'wiggle their ears') and the shell reduced in size during the evolutionary development of enlarged skull to accommodate a large brain. What remains are the elliptically shaped funnels formed by the ears' protrusion which have little sound-gathering function except at around 3000Hz.



W = 210 mm average (adults) difference in sound paths to the two ears = W sin Θ = 11 mm at 3°. Given that the velocity of sound in air at 20°C is 343 m/sec, time lag in sound at RHS ear is given by: $11 \div (343 \times 10^3) = 32$ msec

Figure 1.5 Focusing effect of both ears hearing a signal with slight time delay at one ear relative to the other.

Sensitivity

The sensitivity of the human ear in the range 1000–5000Hz is likely to be as pronounced as that of any animal, as the minimum perceptible acoustic pressures are so small that any increased sensitivity would pick up only thermal noise, the hiss of molecules' agitation which would mask any specific low-intensity sound signal.

Another evolutionary inheritance is the impression, when hearing two sounds very close together from two locations, that the lead sound is the sole source. In other words, the sound heard slightly before the other determines the perceived location. The time interval between the sounds has to be within 30 msec, otherwise the second sound will be perceived as a double image 'blurring' the original sound, or even a discrete echo if the delay is slightly greater. This hearing phenomenon is called the 'Haas Effect' after its discoverer, a nineteenth-century experimenter. The effect even holds true if the slightly delayed sound is louder than the initial sound, although a difference of over 10dB 'swamps' the lead signal and the effect is lost. This effect is put to good use in sound systems within auditoria.

The messages received by the ear can also be selectively interpreted by the brain, even though other sounds are of similar levels. This is referred to as the 'cocktail party' effect, or ability to overhear a specific conversation amidst many. It is only effective for normal binaural hearing; those deaf in one ear are much more limited in their ability to 'tune in' to a specific sound in many.

1.1.3 Spatial characteristics

The sense of sound movement and nuances of its character are perceived strongly in the horizontal axis, so the finishes of walls in rooms or auditoria have particular influence. An example of this is in traditional 'shoebox' concert halls (i.e., those of rectangular format) where orchestral music is best savoured by 'surround sound' strong reflections off the side walls. It is for this reason that sound-reflective hard plaster is used.

Surfaces

In lecture theatres overhead sound-reflective surfaces are used to reinforce the direct sound and carry speech from the stage without loss of the impression that the speech emanates from the stage. Farther back from the stage, finishes are made sound absorbent to 'damp' the reflected sound which would otherwise 'blur' the spoken word by the long sound path reflections, causing loss of clarity. A discrete second image of the sound is detected if much acoustic energy arrives more than 40 msec after the direct sound. The sound will have travelled an extra 14 m, so it is possible to plot an ellipse, as Figure 1.6, within which surfaces can reflect back in without echo, but outside of which surfaces should reflect incident sound so that it stays outside the ellipse.

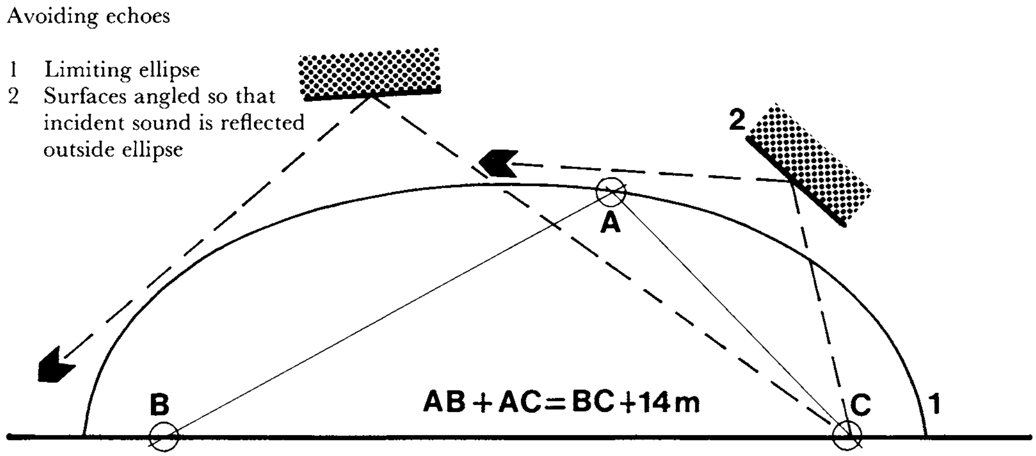


Figure 1.6 Perimeter limits for reflecting surfaces in order to avoid echoes due to long sound path reflections.

1.1.4 Annoyance pattern

Noise heard in the form of unexpected loud sounds can cause pronounced physiological effects – increase in heart rate, change in breathing, even an increase in digestive activities. Evolution has bred into us the association of sudden loud sounds with danger – landslide, avalanche or thunder, for example. We react instinctively with a mild version of our 'fight or flight' response. Repeated arousal of this type is tiring and tends to make the subject irritable and annoyed; the arousal reaction is not easy to decondition.

The human response pattern in noise rating, plotting assessment scale against numbers of respondents making assessments, is a Normal Curve, which is a pattern for natural variations that exist between living things of the same species (height, weight, intelligence, etc, Figure 1.7).

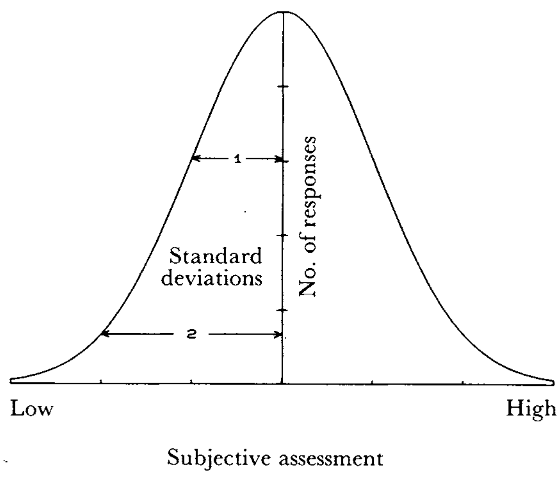
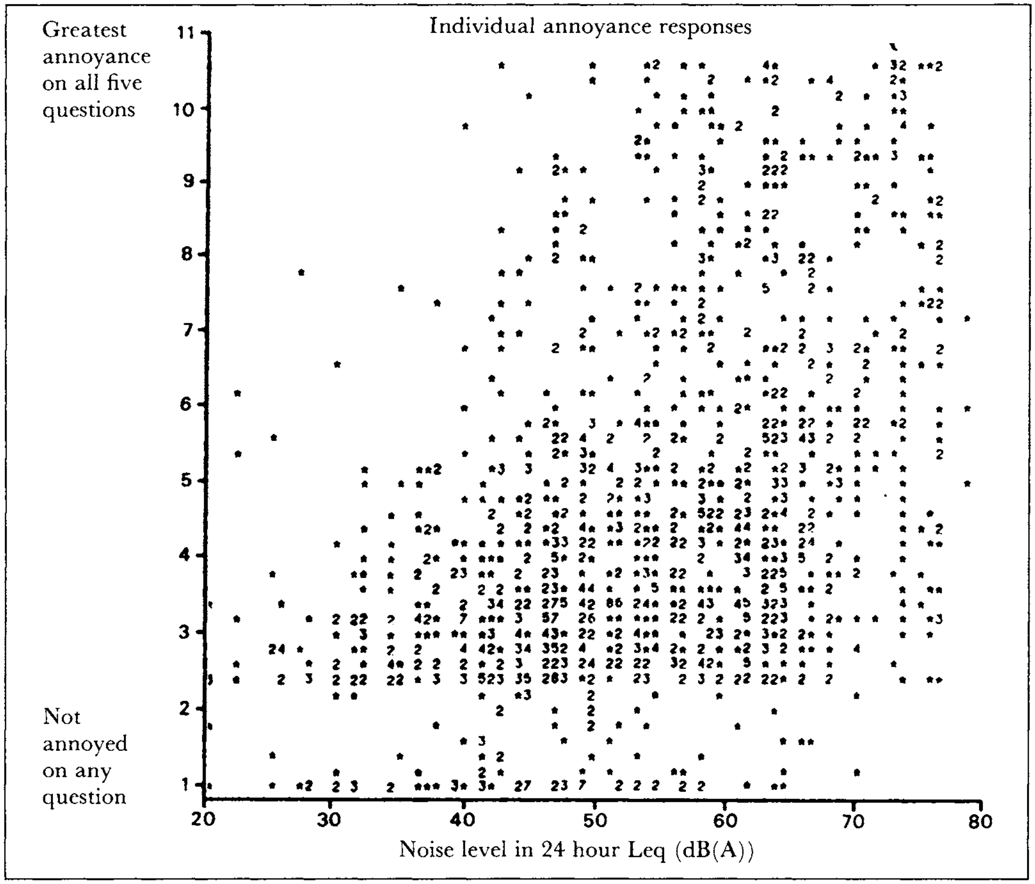


Figure 1.7 Normal curve: the typical pattern for variation in subject response among individuals.

Figure 1.8 An example of the typically wide divergence in subjective response due to a specific noise source.



Railway noise: individuals' annoyance

Individuals

Whilst people agree on objective type data, for example, judging whether one sound is of higher sound pressure level than another, they do not have a consensus of opinion on subjective parameters. Figure 1.8 shows a typical response to a common noise source, railway noise. One of the largest ever surveys of reactions to noise found that the combination of attitudinal differences (favourable or unfavourable) accounted for two and a half times more annoyance variance than the distance from the noise or the intensity of the noise. Even the most carefully considered indices taking full regard of physical factors cannot predict the satisfaction of individuals. Individuals certainly judge noise on a basis of meaning as much as on sound levels. A specific intrusive noise, for example construction site noise, will have a lower sound level annoyance threshold than a noise of a more generally experienced type, endured by people on the whole. Traffic noise is an example of a commonly experienced noise source.

Reaction to a new noise source has been found to depend on sub-group class: some are noise adaptors, others are noise susceptible, and there are those whose annoyance will remain at a particular level as exposure continues. These sub-groups are thought to be in the proportions respectively of 30, 30 and 40 per cent.

Illness
Illness is a particularly sad cause of increased susceptibility to noise. As Hippocrates wrote: 'Noise and stench are to be kept from the sick.' Therapeutic daytime sleep aids recovery, something to be remembered in hospital and rest home design. Although the old lose hearing sensitivity (Figure 1.3), they have waking thresholds from sleep considerably below those for young people. Experiments have been carried out using simulated aircraft noises at levels of 93dBA played to groups aged 7–8, 41–54 and 69–72. These respectively caused 0.9, 19.5 and 72.2 per cent of the subjects to be awakened.

Work output
The effect of noise on work output and decision making has been extensively researched, for example by the US Army. The consensus of research is that noise:
- adversely affects concentration
- reduces efficiency
- causes mistakes
- distracts learning ability.

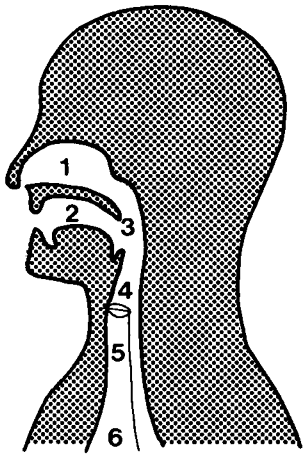
Summary
Important factors (not necessarily in the order of importance) which would appear to influence the subjective disturbance due to sound are:
- sound pressure level
- duration of exposure
- evolution of sound event
- frequency spectrum, i.e., sound character
- individual susceptibility
- personal attitude to noise or sound generator
- mood of listener
- state of health
- activity engaged in during sound event.

1.2 Speech

1.2.1 Vocal cords

Speech is originated by the chest muscles. Contraction squeezes air from the lungs via the larynx to the vocal cords, two flaps with round rim and a split central opening which is vibrated by the passage of air upwards. The opening is usually 250 mm long in men and 150 mm long in women. The system in total forms a series of resonating cavities (nose, throat and mouth) modifying the character of the sound as it comes out (Figure 1.9). The tongue in particular shapes voice sounds, together with the lips, allowing a wide range of voiced sounds.
Breath sounds are those produced by exhaling air without using the vocal cords. This has a hissing quality caused by air flow turbulence as it is driven up, which can be

Figure 1.9 Component parts of the human voice mechanism.

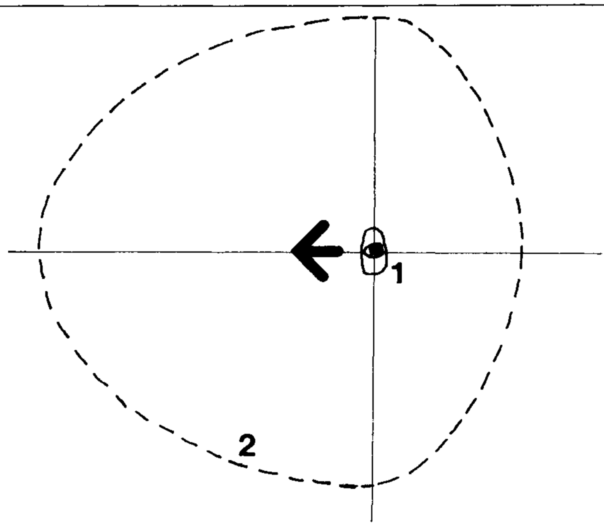


Speech

1 Nasal cavity
2 Tongue
3 Palate
4 Vocal chords
5 Larynx
6 Air from lungs

modified to unvoiced consonants like f, s, p, t and k by the lips, teeth or tongue. Use of vocal cords is of course entirely voluntary, apart from extreme reactions like screams or cries as direct response to powerful stimuli. Other signal or expression sounds are whistling, grunts, laughter, and sighs. Vocal communication is reinforced by body gestures using hands, facial expression and eye contact.

1.2.2 Directionality



Plan

1 Speaker
2 Speech equal sound pressure level contour

Figure 1.10 Sound field around a speaker showing frontal bias in sound projection.

The spoken word is not a sound 'point source' transmitted by the speaker in a uniform radial pattern but is directed forward. This means that a pattern of equal sound pressure levels plotted out from the speaker in the horizontal plane shows shorter distances to side and rear. The difference can be as much as 18dB between an equidistant front and rear location. (The decibel is the basic unit of sound measurement and is defined in section 2.1.4.)
A typical contour for the pattern from a speaker is shown in Figure 1.10. This was

recognised in the past: in the seventeenth century Sir Christopher Wren wrote that the average parish church preacher could not expect to be intelligible further than about 50 ft (15 m) to his front, 30 ft (9 m) to the sides, and 20 ft (6 m) to his back. Given room acoustics more suitable for speech, the frontal characteristics could be extended marginally to:

- up to 15 m = relaxed listening
- 15–20 m = good intelligibility
- 20–25 m = satisfactory
- 30 m = limit of acceptability without electronic amplification.

1.2.3 Speech intelligibility

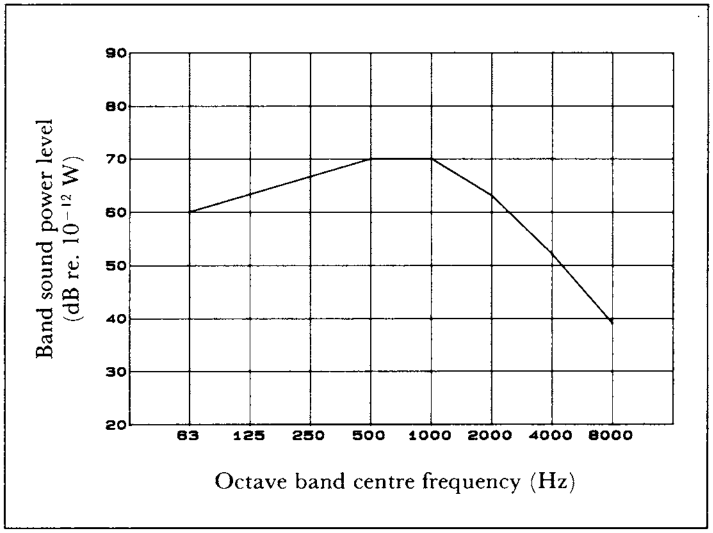


Figure 1.11 Sound power of normal speech v frequency.

The sound power of normal speech (Figure 1.11) is sufficient for conversation within rooms of modest scale, given reasonable background levels; as the background level increases, the speech is masked and sentence comprehension is lost (Figures 1.12 and 1.13); for definition of speech interference level (SIL) see Table 2.10.

In the case of rows of seating in an auditorium, good sightlines also tend to mean reasonable sound reception. The senses act together, that is, it is easier to follow and comprehend speech if one is watching the speaker mouth the words.

In terms of recall, sound cannot compete with sight – people are vision orientated in education. We learn 11 per cent by listening and 83 per cent by sight, and remember 20 per cent of what we hear and 50 per cent of what we see and hear. So give a slide show along with your lecture if you want anyone to remember what you were talking about.

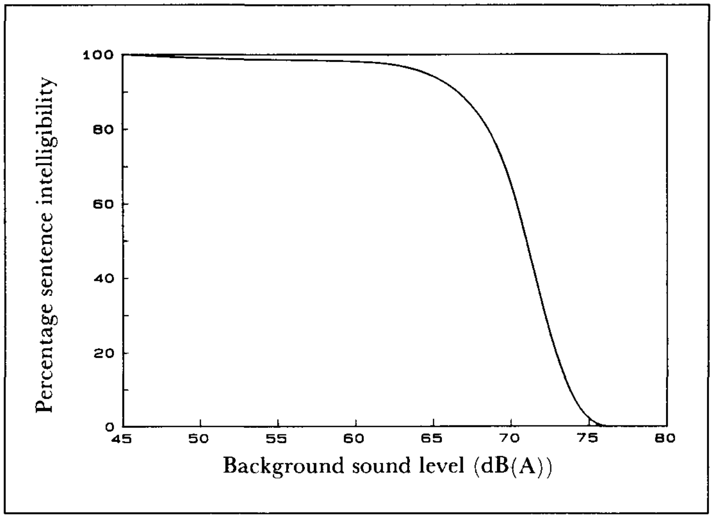


Figure 1.12 Proportion of speech understood in articulation tests v background sound level.

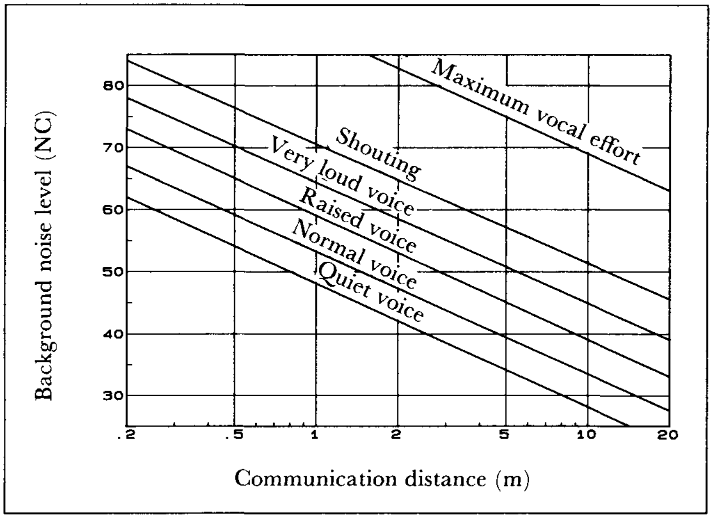
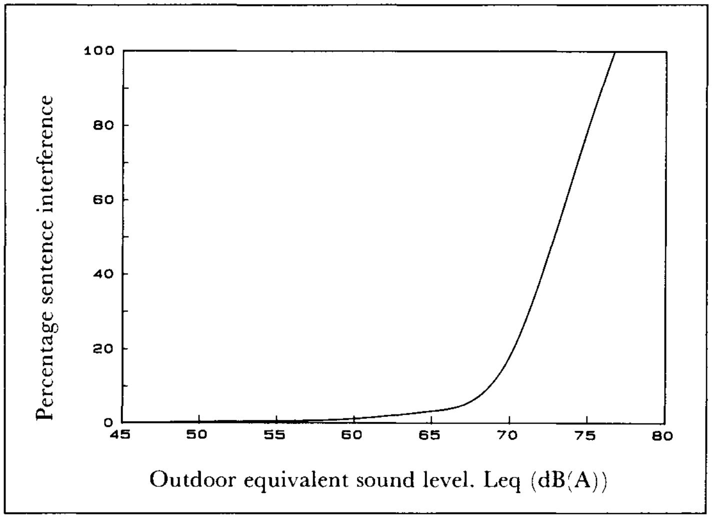


Figure 1.13 Effect of increasing background noise levels in reducing distance speech can be understood.

1.2.4 Speech interference

Noise nuisance frequently occurs through sound from one activity masking sound that is intended to be heard. An example of this is traffic noise interfering with conversation at the roadside (Figure 1.14). The masking effect is most pronounced for sounds of similar frequency; low frequency sounds mask better than high frequency sounds.

Figure 1.14 Interfering effect of high external noise levels on speech.



Intelligibility

The frequency range that contributes to intelligibility of speech is of particular concern – 200–7000Hz, most speech being in the range 500–4000Hz. Intelligibility of speech within a space can be assessed by checking the percentage of syllables, words and sentences understood. When only 30 per cent of syllables are intelligible, conversation can barely be maintained. Tests of a measure called Articulation Index involve lists of random single-syllable words read out to a panel of observers.

In general terms, the factors which affect articulation are background noise level, sound level of the spoken word, room shape (there may be problematic sound reflections), and reverberation time (if too long, successive syllables in speech overlap and mask one another).

On average, speech syllable duration is 0.2 sec and the gap between 0.3 sec, so rooms suitable for speech should have rapid decay characteristics and surfaces positioned for powerful primary reflections to avoid masking of the vulnerable direct sound.

Properties of sound

2.1 Physical Data

2.1.1 Sound waves

Sound is a form of mechanical energy that is propagated through an elastic medium by the vibration of the molecules of the medium. The propagation of energy is called a sound wave and during the passage of the wave individual molecules of the medium vibrate, communicating energy to one another, but suffer no translational movement.

Sound waves are created when some form of vibrating motion is imparted to the medium. In air the sound wave can be caused by a vibrating surface such as the cone of a loudspeaker, a vibrating fluid, such as a resonating air column in an organ pipe, or by the turbulent mixing of air masses, as in the exhaust of a jet. In all cases the wave, which can perhaps be considered most easily as a pressure disturbance, propagates outwards from the source into the surrounding medium.

Types of waves

In air, sound can only travel as longitudinal compressional waves in which the molecules vibrate to and fro in the same direction as the sound wave travels. This is also true for most liquids but in solids sound can travel as compressional, bending and shear waves.

Although our main concern will be with compressional waves we shall have to consider bending waves when dealing with the sound insulation of walls.

2.1.2 Speed of sound

The speed at which a wave travels through a medium is a characteristic of the medium and the type of wave. Table 2.1 gives the velocity of compressional sound waves in some common substances.

The velocity of sound depends upon temperatures and for waves in air the velocity is given by

$$c = \sqrt{\frac{\gamma RT}{M}} \tag{2.1}$$

where γ is the ratio of the specific heats of air at constant pressure and constant volume $= 1.4$
R is the universal gas constant $= 8.31 \times 10^3$ joules/Kmol/°C
M is the molecular weight in kilogrammes $= 28.8$
T is the temperature in degrees kelvin (°K).

If the value of the constants is put into equation 2.1 it reduces to

$$c = 20.05 \times \sqrt{T} \text{ msec}^{-1}$$
or
$$c = 331.3 + 0.61t \text{ msec}^{-1}$$

where t = temp in degrees Celcius (°C).

Thus at 0°C the speed of sound is 331.3 msec^{-1} while at 20°C the speed is 343.5 msec^{-1}. Often, in general acoustics, the speed of sound is given as 340 msec^{-1} which corresponds to a temperature of 14.3°C.

Table 2.1 Speed of longitudinal waves in some common materials

Material	Approximate velocity msec^{-1}
Air (20°C)	343
Helium (20°C)	1007
Water (distilled at 20°C)	1482
Sea water (20°C)	1522
Aluminium	6374
Glass crown	5660
heavy flint	5260
pyrex	5640
Lead	2160
Rubber (natural)	1600
Steel (mild)	5960
Brick	3000
Concrete	3400
Wood	3400

2.1.3 Acoustic pressure

When an acoustic wave passes through a medium there is a local variation in the ambient pressure. This variation is referred to as the acoustic pressure. It is usually quantified by its root mean square (rms) value $\overline{p^2}$ which is given by

$$\overline{p^2} = \left[\frac{1}{T_a}\int_0^{T_a} p^2(t)dt\right]^{\frac{1}{2}}$$

where p(t) is the time varying value of the acoustic pressure and T_a is the period over which its squared value is averaged.

Pressure units

The accepted unit of acoustic pressure is the pascal (Pa) although newtons m^{-2} (Nm^{-2}) and micro-bar (μ bar) are sometimes used

1 pascal = 1 newton m^{-2} = 10 micro-bar.

For sound in air the lowest acoustic pressure that the average human ear can detect is about 2×10^{-5}Pa while an acoustic pressure of about 20Pa produces a painful sensation. As the mean atmospheric pressure is 10^5Pa it can be seen that even very loud sounds cause only a small variation in the steady pressure. Throughout this book any reference made to acoustic pressure will, unless otherwise stated, be rms acoustic pressure.

2.1.4 Sound pressure level

Decibels

In acoustics it is common practice to express the acoustic pressure in units of decibels and to refer to the resulting quantity as the sound pressure level (SPL). The definition of sound pressure level is

$$SPL = 20 \log_{10}\left[\frac{p}{p_0}\right] \text{decibels (dB)} \qquad (2.2)$$

where p is the acoustic pressure and p_0 is a reference pressure which in audio acoustics is chosen to be 2×10^{-5}Pa.

Thus we see that 0dB corresponds to an acoustic pressure of 2×10^{-5}, as $\log_{10} 1$ equals zero, and 120dB corresponds to a pressure of 20Pa as $\log_{10}10^6$ equals 6. The relationship between sound pressure and sound pressure level is shown in Figure 2.1 and the sound pressure levels of some common sounds are shown in Table 2.2.

Anti-logging decibels

The sound pressure level can also be written as

$$SPL = 10 \log\left[\frac{p^2}{p_0^{\,2}}\right] \text{dB} \qquad (2.3)$$

where p^2 is the square of the rms pressure, that is, the mean square pressure. Conversely, the rms pressure and mean square pressure corresponding to a given SPL are obtained as follows

$$p = 10^{\frac{SPL}{20}} \times p_0 \qquad (2.4)$$

$$p^2 = 10^{\frac{SPL}{10}} \times {p_0}^2$$

These relationships we will find useful in section 2.3.1.

Figure 2.1 Relationship between sound pressure and sound pressure level.

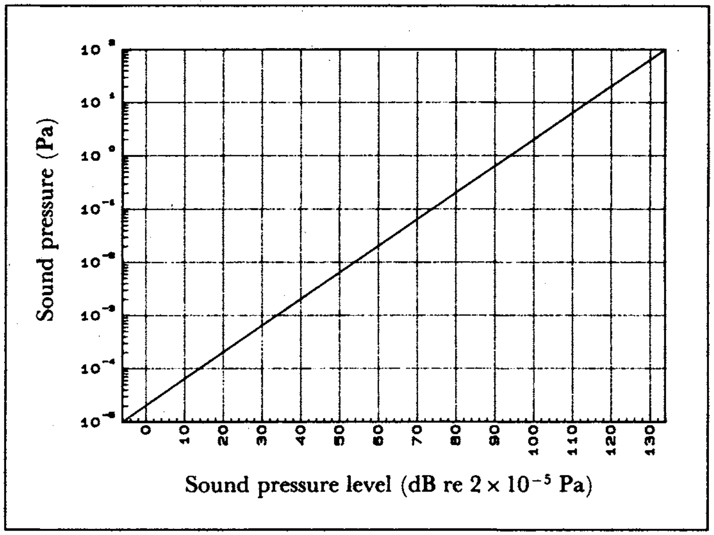


Table 2.2 Approximate sound pressure levels of some common sources	
Sound pressure dB(A)	Source and location
140	Jet aircraft at take-off 25 m
100	Very noisy factory steel riveting at 5 m
90	Heavy diesel lorry at 7 m
	Road drill at 7 m unsilenced
80	Ringing alarm clock at 1 m
70	Inside a small saloon car at about 50 kph
65	Busy general office with typewriters
50	Ordinary conversation at 1 m
40	Quiet living room
35	Quiet bedroom at night
25	Still day in the country with no traffic
15	Background noise in broadcasting studio

Sound level measurement

Sound pressure levels may be measured by the sound level meter. All sound level meters measure by a conversion of acoustic pressure to a proportional alternating current or voltage. A typical sound level meter is shown diagrammatically in Figure 2.2. The frequency weighting network allows different degrees of amplification of the signal at different frequencies, and enables the meter to imitate the ear's response. In addition, filters can be included to pass signals within a specific frequency range and block signals off at other frequencies. Usually octave band frequencies from 31.5Hz up to 16,000Hz can be selected, although for more detailed analysis one-third octave band frequencies can be examined. The measurement of sound is dealt with in more detail in section 2.2.2.

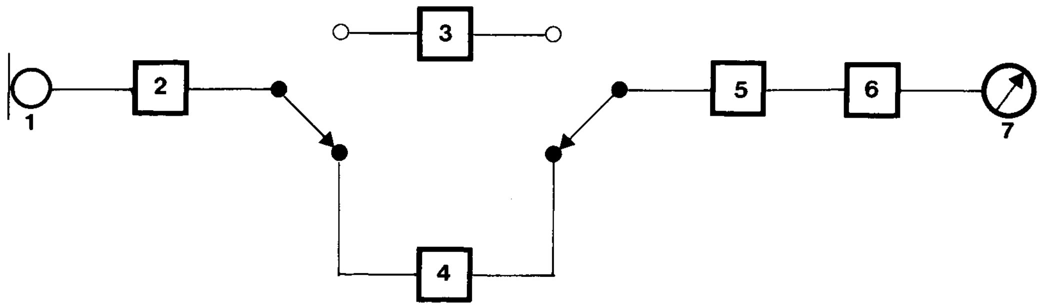


Figure 2.2 Block diagram of sound level meter.

1 Microphone
2 Input amplifier with variable attenuator
3 Weighting networks
4 Octave filters
5 Output amplifier with variable attenuator
6 Averaging circuit
7 Meter

2.1.5 Intensity level

Intensity

Intensity is the energy passing per second through unit area perpendicular to the direction of travel, and intensity level is defined as

$$\text{Intensity level} = 10 \log \frac{I}{I_0} \text{ dB}$$

where I is the rms intensity and I_0 is a reference intensity. In air I_0 is chosen to be $10^{-12}Wm^{-2}$. For well-defined directions of energy flow and away from the source, the intensity is related to the mean square pressure by

$$I = \overline{p}^2/\rho c$$

where ρ is the density of the medium and c is the velocity of sound.

In air, if these conditions hold, the intensity level and sound pressure level are numerically equal and often the two terms are used synonymously. However, as the equality does not generally hold, it is preferable to use intensity level only when actual intensities are involved.

Characteristic impedance
The quantity ρc is referred to as the characteristic impedance of the medium and in air it has a value of 413 $Pam^{-1}sec$.

Measurement of intensity
The measurement of acoustic intensity has, in the past, been most difficult and intensity values were generally obtained from measures of the sound pressure level via the relationship $I = p^2/\rho c$.
This could give large errors when, for example, measurements were made close to a vibrating surface. Recently, instrumentation has become available which measures acoustic intensity directly and it is most useful for determining the sound intensity or power output of sources when several are operating together in close proximity.

2.1.6 Sound power

The power of a source is the amount of energy/sec it emits. Sound power is expressed in watts. Sound power level (SWL) is defined as

$$SWL = 10 \log \frac{W}{W_0} \text{ dB}$$

where W is the rms power and W_0 is a reference power chosen, in air, to be 10^{-12}watts. The sound power equivalent of a sound power level of SWL dB is

$$W = 10^{\frac{SWL}{10}} \times W_0 \text{ watts}$$

2.1.7 Frequency

Wave cycle
If vibrations are given to a medium in a regular manner, then a regular progression of pressure maxima and minima will be observed at any point in the medium. When two adjacent maxima or minima or any two adjacent parts of equal pressure pass the point, we say that one complete cycle of the wave has passed. The number of complete cycles that pass in one second is known as the frequency of the wave. Frequency has units of Hertz (Hz), so that if 1000 cycles occur in one second the wave has a frequency of 1000Hz.

Period
The time for one cycle to pass a given point is known as the period of the wave T and thus frequency and period are related by

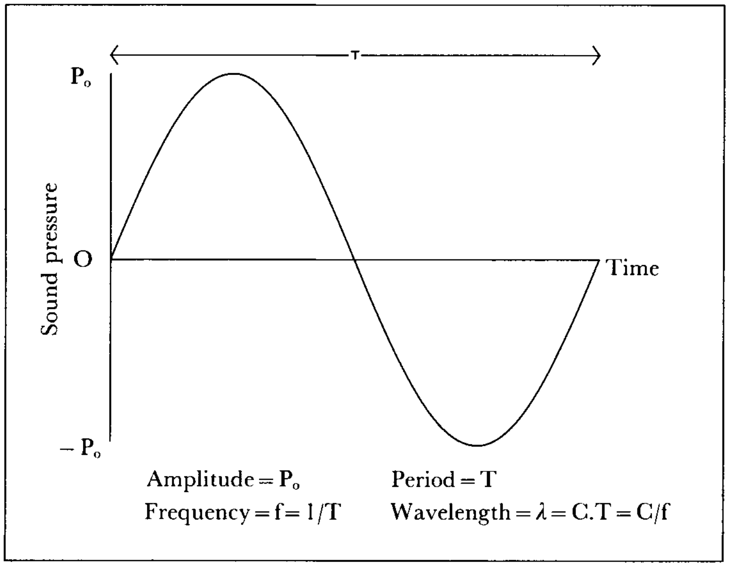
$$f = \frac{1}{T} \text{ Hz for T in seconds}$$

Now in T seconds a pressure disturbance will travel a distance of $c \times T$ metres where c is the velocity of sound in the medium, and this is the distance between two identical adjacent points on the wave or the wavelength of the wave, λ.

Thus $\lambda = cT = c/f$

or $c = \lambda f$

Figure 2.3 Variation of sound pressure with time in a harmonic sound wave.



This is illustrated for a harmonic wave in Figure 2.3.
In air and most media the velocity of compressional waves is independent of frequency and the media are said to be non-dispersive. On the other hand, bending waves in solids travel at velocities that depend on the frequency and for this type of wave the medium is dispersive.
The frequency of a sound is as important as its level because many acoustic phenomena are frequency dependent. For example, the sensitivity of the human ear, the absorption behaviour of a sound absorber and the sound insulation of a partition all vary with frequency.

Pure tone
A pure tone has a simple harmonic pressure fluctuation of constant amplitude and frequency, the amplitude being associated with the subjective quality of loudness, and frequency the subjective quality of pitch. More complicated waves are produced by musical instruments which produce a combination of tones called partials. The lowest frequency partial is known as the fundamental. If higher frequency partials are related integrally to the fundamental (e.g. in stringed instruments) they are called harmonics; if no simple relationship exists they are called anharmonics.

Complex sound
A much more complicated sound wave, such as the noise produced by a jet engine, will consist of many frequencies all present at the same time, each having an amplitude which varies randomly with time. In theory one could isolate each frequency and measure its contribution to the total noise level by measuring its rms pressure over a reasonably long period. In practice, however, this is not feasible and it is common to group frequencies into various bands and measure the contribution of all the frequencies within a band to the total noise level.

2.1.8 Frequency bands

Octave bands
Octave bands are the most frequently used form of frequency sub-division. An octave is generally accepted as being a band of frequencies, the upper frequency of which is twice the lower frequency. However, the International Standard octave band is defined as one in which the upper frequency is $10^{0.3}$ (1.9953) times the lower frequency. This is a very small difference but results in the ratio of the lowest to the highest frequencies of ten octaves having a ratio of 1000 instead of 1024. The accepted octave band centre frequencies are obtained by assuming the centre frequency of one band to be 1000Hz and then successively multiplying or dividing 1000Hz by $10^{0.3}$ to obtain the other centre frequencies. The band limits are obtained by multiplying or dividing the centre frequency by $10^{0.15}$.

One-third octave bands
To obtain more information one-third octave bands can be used to analyse a noise. For one-third octave bands the ratio of the upper to the lower frequency is $10^{0.1}$. Again, 1000Hz is the centre frequency of one band and other centre frequencies are obtained by multiplying or dividing the centre frequency by $10^{0.1}$ (band limits are obtained by multiplying or dividing the centre frequencies by $10^{0.05}$). The band centre frequencies and limits for both octave and one-third octave bands are shown in Table 2.3. The centre frequencies are the preferred frequencies specified in BS 3593:1963. The true values would be calculated as indicated in the text.

Widths of bands
The width of each octave band is 70 per cent of the centre frequency while each one-third octave has a width equal to 23 per cent of its centre frequency and so this type of band is referred to as having a constant percentage bandwidth.

Spectrum
When a noise is analysed into frequency bands each band will have a noise level in decibels. If the band levels are plotted against the centre frequency of the band the resulting graph is called a spectrum. In the spectrum shown in Figure 2.4 it can be seen that the centre frequencies are equally spaced; that is, they have been plotted on a logarithmic scale. This is the usual practice in acoustics.

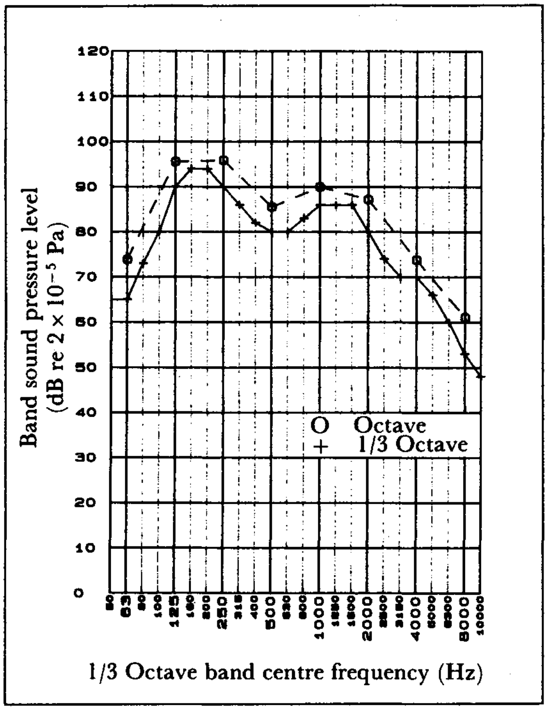


Figure 2.4 Typical noise spectra.

Table 2.3 Preferred octave and third-octave band centre frequencies and band limit frequencies

Band limits Hz	Band centre frequencies Hz octave	third octave	Band limits Hz
22			22
	31.5	25 31.5 40	28 35
44			44
	63	50 63 80	57 71
89			88
	125	100 125 160	113 141
176			176
	250	200 250 315	225 283
353			353
	500	400 500 630	440 565
707			707
	1000	800 100 1250	880 1130
1414			1414
	2000	1600 2000 3500	1760 2250
2825			2825
	4000	3150 4000 5000	3530 4400
5650			5650
	8000	6300 8000 10000	7070 8800
11300			11300
	16000	12500 16000 20000	14140 17600
22500			22500

2.2 Units and Measurement

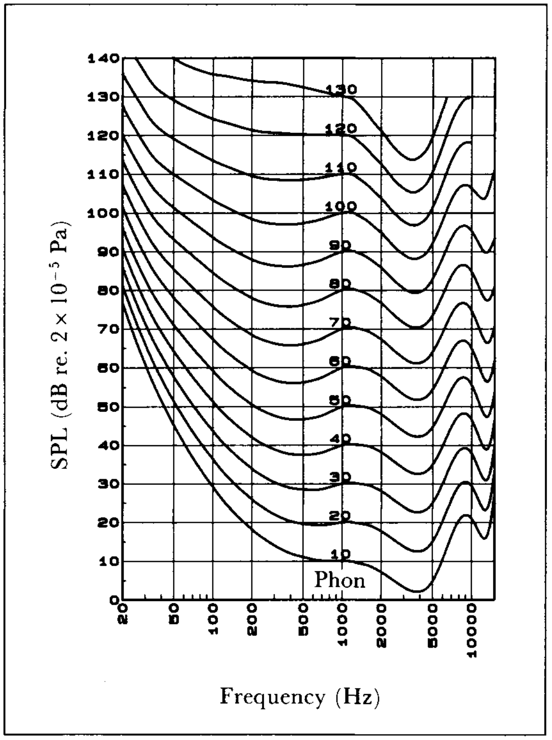
2.2.1 Noise units

It is often convenient to describe a sound field in terms of a single figure descriptor which takes into account various parameters of the field such as level, frequency content or temporal variation. Many such descriptors exist, some of which are used generally to quantify the field from all manner of sources and others which are applied only to noise from specific sources, such as aircraft. In this section some of the more common measures will be presented.

A weighted sound level, L_A

Human hearing is not equally sensitive to all frequencies. In addition the variation with frequency is a function of the sound level. This is shown in Figure 2.5. To try and account for this variation when measuring sound levels, electronic weighting networks are incorporated in the instrument between the microphone and the display. A weighting corresponds most closely to the 30 phon curve, that is the equal loudness contour which passes through 30dB at 1000Hz. The A weighting curve is shown in Figure 2.6 and the specification for the curve is shown in Table 2.4. Sound levels measured using the A weighting network are designated dB(A).

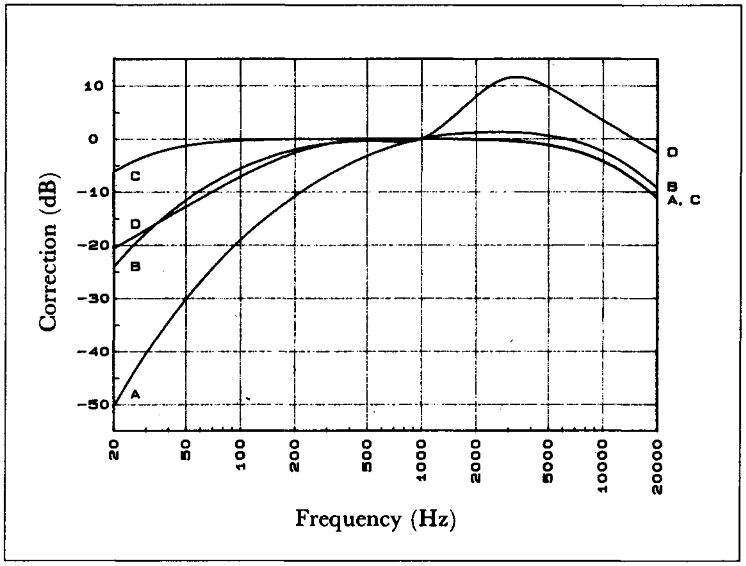
Figure 2.5 Equal loudness contours.



B and C weighted sound pressure levels, L_B and L_C

The B and C weightings are similar to the A weighting in concept, corresponding roughly with the 70 phon and 100 phon curve respectively. They are shown in Figure 2.6 and specified in Table 2.4. Sound levels obtained using the network are quoted as dB(B) and dB(C). The B or C networks are infrequently used compared with the A weighting.

Figure 2.6 Frequency weighting curves.



D weighted sound pressure level, L_D

The D weighting was introduced solely for measuring aircraft noise. It is shown in Figure 2.6 and specified in Table 2.4. Sound levels measured using this network are quoted as dB(D). From the weighting specification it is clear that the weighting emphasises those frequencies in the range 1000–10,000Hz which is consistent with equal noisiness contours.

Equivalent continuous sound level, Leq

This is defined as the level of a notional steady sound which, at a given position and over a defined period of time, would have the same A weighted acoustic energy as the fluctuating noise. So over a period of time T

$$\text{Leq} = 10 \log_{10}\left[\frac{1}{T}\int_0^T \frac{p_A^2(t)}{p_0^{\,2}}\right] \quad \text{dB(A)} \tag{2.5}$$

where $p_{A(t)}$ is the A weighted sound pressure as a function of time and p_o is the reference pressure. Alternatively, Leq can be expressed

Table 2.4 Specification of weighting networks

Frequency Hz	Curve A dB	Curve B dB	Curve C dB	Curve D dB
10	−70.4	−38.2	−14.3	−27.6
12.5	−63.4	−33.2	−11.2	−25.6
16	−56.7	−28.5	−8.5	−23.5
20	−50.5	−24.2	−6.2	−21.6
25	−44.7	−20.4	−4.4	−19.6
31.5	−39.4	−17.1	−3.0	−17.6
40	−34.6	−14.2	−2.0	−15.6
50	−30.2	−11.6	−1.3	−13.6
63	−26.2	−9.3	−0.8	−11.6
80	−22.5	−7.4	−0.5	−9.6
100	−19.1	−5.6	−0.3	−7.8
125	−16.1	−4.2	−0.2	−6.0
160	−13.4	−3.0	−0.1	−4.4
200	−10.9	−2.0	0	−3.1
250	−8.6	−1.3	0	−1.9
315	−6.6	−0.8	0	−1.0
400	−4.8	−0.5	0	−0.3
500	−3.2	−0.3	0	0
630	−1.9	−0.1	0	−0.1
800	−0·8	0	0	−0.4
1000	0	0	0	0
1250	0.6	0	0	1.9
1600	1.0	0	−0.1	5.4
2000	1.2	−0.1	−0.2	8.0
2500	1.3	−0.2	−0.3	10.0
3150	1.2	−0.4	−0.5	11.0
4000	1.0	−0.7	−0.8	10.9
5000	0.5	−1.2	−1.3	10.0
6300	−0.1	−1.9	−2.0	8.5
8000	−1.1	−2.9	−3.0	6.0
10000	−2.5	−4.3	−4.4	3.0
12500	−4.3	−6.1	−6.2	−0.4
16000	−6.6	−8.4	−8.5	−4.4
20000	−9.3	−11.1	−11.2	−8.1

The octave-band weighting is taken to be that of the third octave having the same centre frequency.

$$\mathrm{Leq} = 10 \log_{10} \left[\frac{1}{T} \int_0^T 10^{\frac{L_A(t)}{10}} \, dt \right] \quad \mathrm{dB(A)}$$

where $L_A(t)$ is the A weighted sound level as a function of time. For most noise environments, where the sound level varies with time, the Leq value has to be obtained by measurement. However, in cases where a number of ostensibly constant level noises exist for well-defined periods the Leq can be calculated using

$$\mathrm{Leq} = 10 \log_{10} \left[\frac{1}{T} [10^{L_1/10} \times t_1 + 10^{L_2/10} \times t_2 + 10^{L_3/10} \times t_3 \ldots] \right] \quad \mathrm{dB(A)} \quad (2.6)$$

where L_1 is the constant A weighted sound level that exists for time t_1, etc, and T is the total time over which the value is calculated. Alternatively, the addition can be done using the graph shown in Figure 2.7. In practice Leq is normally calculated over a relatively long time, say, 1, 8, 12 or 24 hours. Perhaps the major disadvantage is that it underestimates short duration high level events; nevertheless, Leq and derivatives from it are widely used.

Example

Measurement and observation of the activity on a construction site yielded the following information:

Plant	Average sound level dB(A)	Typical on time in an 8-hour period (hours)
Cement mixer	75	5
Dumper truck	68	3
Power saw	82	1
Generator	65	8

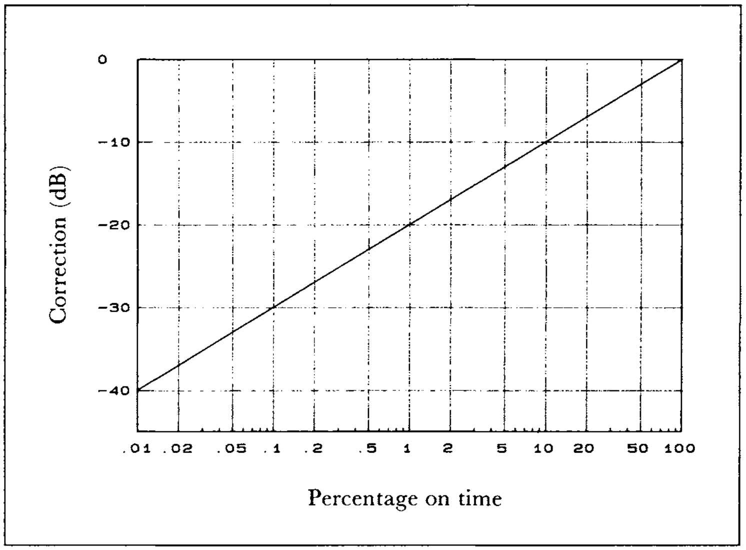
To estimate the 8-hour Leq value we proceed as follows:

Plant	% on time	Correction from Figure 2.7 (dB)	Corrected level dB(A)
Cement mixer	62.5	−2.0	73.0
Dump truck	37.5	−4.3	63.7
Power saw	12.5	−9.0	73.0
Generator	100	0	65

The 8-hour Leq is obtained by adding the corrected levels, assuming they are incoherent noise sources, using either Figure 2.15 or equation 2.12

$$\mathrm{Leq} = 76.6\mathrm{dB(A)}$$

Figure 2.7 Correction to measured noise level for percentage on-time to give the equivalent continuous sound level.



Single event noise exposure level, L_{AX}

This is defined as the continuous sound level which, if maintained constant for one second, contains the same acoustic energy as the actual time varying level of a given noise event. In effect it is an Leq value derived over a period of 1 sec and like Leq the noise level is usually A weighted before the L_{AX} value is computed.

$$L_{AX} = 10 \log_{10} \left[\int_{-\infty}^{\infty} \frac{p_A^2(t)}{p_o^2} \right] \text{dBA}$$

In practice the integration is limited to the duration of the actual noise event, i.e.,

$$L_{AX} = 10 \log_{10} \left[\int_{t_1}^{t_2} 10^{\frac{L_{A(t)}}{10}} dt \right] \quad \text{dB(A)} \qquad (2.7)$$

where t_1 and t_2 denote the beginning and end of the single event respectively.
For noise events that last for more than one second L_{AX} will be higher than the maximum value of $L_{A(t)}$ throughout the event. L_{AX} values are useful for describing short duration events such as aircraft flyovers, single vehicle bypasses, train bypasses, etc., especially if these are to be included in the calculation of an Leq value over a given time period. As L_{AX} gives the energy contribution of a single event, the Leq over a period T is given by

$$\text{Leq} = L_{AX} - 10 \log T \quad \text{dB(A)}$$

The correction 10 log T can be obtained from Figure 2.7 if it is remembered that the L_{AX} has an effective on-time of one second.

For a number of single events

$$\text{Leq} = 10 \log_{10} \left[\frac{1}{T} \sum_{i=1}^{n} 10^{\frac{L_{AXi}}{10}} \right] \quad \text{dB(A)} \tag{2.8}$$

Statistical level, L_N

This is the sound level that is exceeded for N per cent of the measurement time and is obtained from a statistical analysis of the time-varying noise. Invariably the noise is A weighted before analysis. Although it is possible to obtain any value of L_N the most commonly used values are L_{10}, the level exceeded for 10 per cent of the time, L_{50} and L_{90}, the levels exceeded for 50 and 90 per cent of the time respectively. The L_{10} value gives an indication of the higher noise levels but can be significantly lower than the occasional peak levels. It is used in the measurement of noise from road traffic and for freely flowing traffic with a flow of greater than 100 vehicles per hour it has been found that

$$L_{10} \approx \text{Leq} + 3 \quad \text{dB(A)} \tag{2.9}$$

The L_{90} value corresponds to the general noise level in the absence of specific noise sources and it is suggested in BS 4142, 'Method of Rating Industrial Noise Affecting Mixed Residential and Industrial Areas', that it is an appropriate measure of background noise.

Perceived noise level

The perceived noise level, PNL, is a rating for single aircraft flyovers based originally on jury judgements of perceived noisiness, but it is now commonly derived by an extensive calculation procedure. However, for the purpose of monitoring and surveying noise in dwellings and in communities, approximate methods of obtaining the perceived noise level are considered adequate. In these methods the PNL is obtained from measurements of the maximum slow A weighted or D weighted sound level that occurs during a flyover, i.e.

$$\text{PNL} = \text{dB(A)} + 13 \quad \text{PNdB}$$
$$\text{PNL} = \text{dB(D)} + 7 \quad \text{PNdB}$$

Noise and number index, NNI

This index applies specifically to aircraft noise and was developed following a survey around London's Heathrow Airport in 1961. It is defined as

$$\text{NNI} = \text{average peak perceived noise level} + 15\log\text{N} - 80 \tag{2.10}$$

where N is the number of aircraft, having a PNL greater than 80PNdB in the specified period.

The average peak PNL is given by

$$10\log_{10}\left[\frac{1}{N}\sum_{i=1}^{N}10^{L_i/10}\right] \tag{2.11}$$

where L_i is the peak value of PNL occurring during the passage of each aircraft. The NNI is calculated for daytime operations, 0600–1800 hours, over the summer period of mid-May to mid-September.

Noise rating number

This is a unit generally used for specifying acceptable noise levels in buildings and has been adopted by ISO and is widely used in Europe. The noise rating (NR) number is obtained by reference to a set of noise rating curves, shown in Figure 2.8, or Table 2.5.

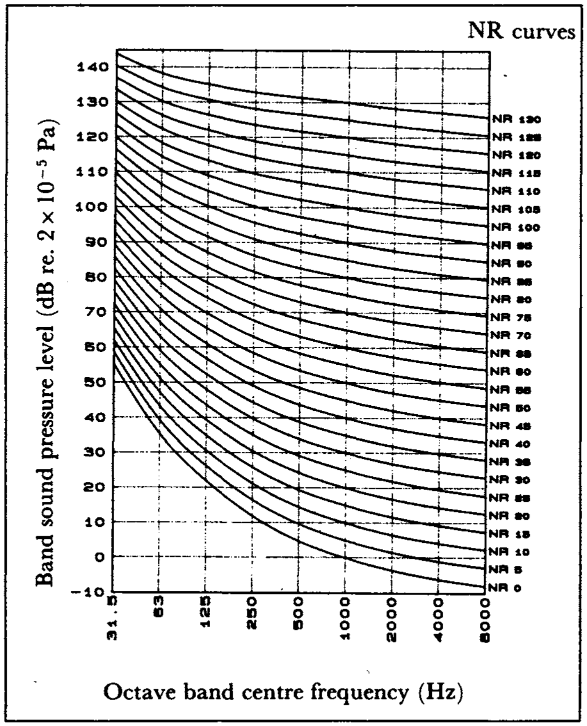


Figure 2.8 Octave band noise rating curves.

The octave band sound levels of the noise are plotted on to the curves and the highest curve that is intercepted by the curve joining the plotted values gives the noise rating number. Interpretation between the noise rating curves is allowed. Noise rating curves for use with one-third octave band sound pressure levels are given in Figure

Table 2.5 Octave band sound pressure levels corresponding to noise rating number NR

	Octave band sound pressure levels (dB)								
	Centre frequencies (Hz)								
NR	31.5	63	125	250	500	1000	2000	4000	8000
0	55.4	35.5	22.0	12.0	4.8	0	– 3.5	– 6.1	– 8.0
5	58.8	39.4	26.3	16.6	9.7	5	1.6	– 1.0	– 2.8
10	62.2	43.4	30.7	21.3	14.5	10	6.6	4.2	2.3
15	65.6	47.3	35.0	25.9	19.4	15	11.7	9.3	7.4
20	69.0	51.3	39.4	30.6	24.3	20	16.8	14.4	12.6
25	72.4	55.2	43.7	35.2	29.2	25	21·9	19.5	17.7
30	75.8	59.2	48.1	39.9	34.0	30	26.9	24.7	22.9
35	79.2	63.1	52.4	44.5	38.9	35	32.0	29.8	28.0
40	82.6	67.1	56.8	49.2	43.8	40	37.1	34.9	33.2
45	86.0	71.0	61.1	53.6	48.6	45	42.2	40.0	38.3
50	89.4	75.0	65.5	58.5	53.5	50	47.2	45.2	43.5
55	92.9	78.9	69.8	63.1	58.4	55	52.3	50.3	48.6
60	96.3	82.9	74.2	67.8	63.2	60	57.4	55.4	53.8
65	99.7	86.8	78.5	72.4	68.1	65	62.5	60.5	58.9
70	103.1	90.8	82.9	77.1	73.0	70	67.5	65.7	64.1
75	106.5	94.7	87.2	81.7	77.9	75	72.6	70.8	69.2
80	109.9	98.7	91.6	86.4	82.7	80	77.7	75.9	74.4
85	113.3	102.6	95.9	91.0	87.6	85	82.8	81.0	79.5
90	116.7	106.6	100.3	95.7	92.5	90	87.8	86.2	84.7
95	120.1	110.5	104.6	100.3	97.3	95	92.9	91.3	89.8
100	123.5	114.5	109.0	105.0	102.2	100	98.0	96.4	95.0
105	126.9	118.4	113.3	109.6	107.1	105	103.1	101.5	100.1
110	130.3	122.4	117.7	114.3	111.9	110	108.1	106.7	105.3
115	133.7	126.3	122.0	118.9	116.8	115	113.2	111.8	110.4
120	137.1	130.3	126.4	123.6	121.7	120	118.3	116.9	115.6
125	140.5	134.2	130.7	128.2	126.6	125	123.4	122.0	120.7
130	143.9	138.2	135.1	132.9	131.4	130	128.4	127.2	125.9

2.9. The third-octave band sound levels corresponding to these curves are obtained by subtracting 4.8dB from the octave band values given in Table 2.5 for each third octave in the octave. A list of NR numbers suitable for different environments is given in Table 2.6.

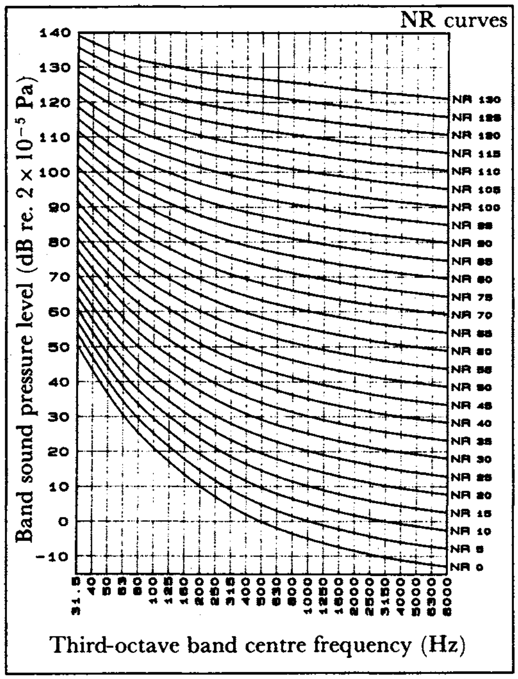


Figure 2.9 Third-octave band noise rating curves.

Table 2.6 Recommended noise ratings for different environments

Environment	Noise rating number
Concert halls, opera halls, recording studios, live theatres (> 500 seats)	20
Live theatres (< 500 seats), cathedrals and large churches, television studios, large conference and lecture theatres (> 50 people), music rooms	25
Bedrooms in private houses, board rooms, top management offices, small conference and lecture rooms, multipurpose halls, small and medium churches, libraries, operating theatres, cinemas, courtrooms	30
Living rooms in private homes, hotel bedrooms, hospital open wards, middle management offices, school classrooms, museums	35
Public rooms in hotels, ballrooms, small restaurants, quality shops, large open plan offices, reception areas, laboratories	40
Halls, corridors and lobbies in hotels, hospitals, etc, recreation rooms, post offices, large restaurants, bars, department stores, shops, computer rooms	45
Cafeterias, canteens, supermarkets, swimming-pools, gymnasia, kitchens, laundry rooms	50

Noise criterion

This is very similar to the noise rating number. It was originally developed in the United States specifically for application in commercial buildings. As with the noise rating scheme, use is made of a set of reference curves shown in Figure 2.10 and Table 2.7. The noise criterion is obtained from the lowest curve which is nowhere exceeded by the octave band sound levels of the noise. In 1971 the preferred noise criteria (PNC) curves were introduced but as yet they are not widely accepted. They are given in Figure 2.11 and Table 2.8. A list of NC numbers appropriate to different environments is given in Table 2.9.

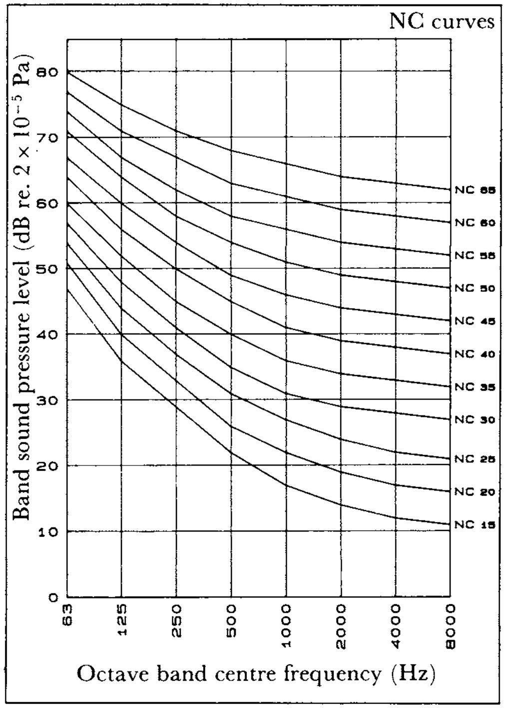
Table 2.7 Octave band sound pressure levels corresponding to noise criterion NC

	Octave band centre frequencies (Hz)							
NC	63	125	250	500	1000	2000	4000	8000
15	47	36	29	22	17	14	12	11
20	51	40	33	26	22	19	17	16
25	54	44	37	31	27	24	22	21
30	57	48	41	35	31	29	28	27
35	60	52	45	40	36	34	33	32
40	64	56	50	45	41	39	38	37
45	67	60	54	49	46	44	43	42
50	71	64	58	54	51	49	48	47
55	74	67	62	58	56	54	53	52
60	77	71	67	63	61	59	58	57
65	80	75	71	68	66	64	63	62

Table 2.8 Octave band sound pressure levels corresponding to preferred noise criterion PNC

	Octave band centre frequencies (Hz)								
PNC	31.5	63	125	250	500	1000	2000	4000	8000
15	58	43	35	28	21	15	10	8	8
20	59	46	39	32	26	20	15	13	13
25	60	49	43	37	31	25	20	18	18
30	61	52	46	41	35	30	25	23	23
35	62	55	50	45	40	35	30	28	28
40	64	59	54	50	45	40	36	33	33
45	67	63	58	54	50	45	41	38	38
50	70	66	62	58	54	50	46	43	43
55	73	70	66	62	59	55	51	48	48
60	76	73	69	66	63	59	56	53	53
65	79	76	73	70	67	64	61	58	58

Figure 2.10 Noise criteria curves.
Figure 2.11 Preferred noise criteria curves.



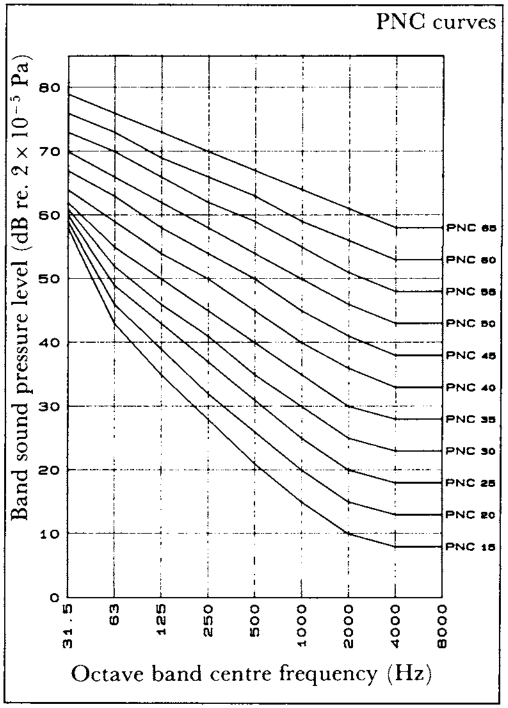


Table 2.9 Recommended noise criteria for different environments

Environment	Noise criterion (NC)
Concert halls, opera houses, large auditoria, large theatres (> 500 seats), large churches, recording studios	10–20
Broadcasting studios, small auditoria, small theatres, small churches, music rehearsal rooms, large meeting and conference rooms, executive offices, hospital wards, operating theatres	20–25
Bedrooms (private to motels), courtrooms, cinemas	25–40
Private offices, small conference rooms, classrooms, libraries, living rooms	30–40
Large offices, reception areas, retail shops, cafeterias, restaurants, laboratories	35–45
Lobbies, drawing offices, general typing areas, swimming pools, gymnasia	40–50
Rooms with computer and office equipment, kitchens and laundries, shops	45–55

Speech interference level, SIL

The speech frequencies that contribute to intelligibility lie between 200 and 7000Hz and masking of these frequencies by noise will obviously give reduced understanding. A simple unit for assessing interference with speech by noise is the speech interference level (SIL) which is defined as the arithmetic average of the sound levels in the three octaves centred on 500, 1000 and 2000Hz. Values of the SIL that allow word intelligibility at given distances are shown in Table 2.10.

Table 2.10 Values of speech interference level for steady continuous noises in which speech communication is barely possible (dB)

Distance between talker and listener	Voice level Normal	Raised	Very loud	Shouting
0.15	71	77	83	89
0.3	65	71	77	83
0.6	59	65	71	77
0.9	55	61	67	73
1.2	53	59	65	71
1.5	51	57	63	69
1.8	49	55	61	67
3.7	43	49	55	61

The values in the table are for an average male speaker. For a female speaker it is suggested that the SIL levels be reduced by 5dB.

The following provides a guide to acceptable background noise when using a telephone:

SIL value dB	*Telephone use*
45	Satisfactory
55	Slightly difficult
65	Difficult
75	Impossible.

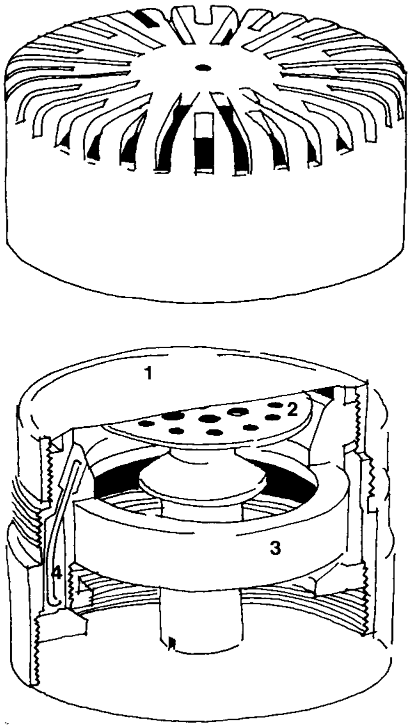
2.2.2 Noise measurement

A knowledge of the noise field is essential to any design, prediction or assessment procedure, and measurement of the parameters of the field often have to be made as a preliminary to any study. The type of measurements that are made will depend upon the use to which they are to be put and this should be considered carefully before any measurement programme is started.

Basic measuring systems
The choice of measurement system will largely be influenced by the characteristics of the noise to be measured and should be chosen to give measurements which properly describe the noise, are accurate and reliable and which can be used with confidence in estimates, calculations and predictions.
All systems are essentially similar in that each will consist of a transducer, an analysis section and an output or display section.

The transducer
The transducer is usually a condenser type microphone. The condenser microphone is chosen because it can combine good frequency response and dynamic range with good long-term stability. This type of microphone works on the principle that the capacitance of two electrically charged plates changes as their separation changes. One of the plates is a very light diaphragm which moves in response to the acoustic pressure, developing a voltage which is conditioned by associated electronic circuits. Such a microphone, a schematic view of which is given in Figure 2.12, requires a large voltage, 200–300V, to be applied across the plates. A prepolarised condenser microphone, which does not require this voltage and is known as an electret microphone, is also available and is incorporated in some instruments.

Figure 2.12 Schematic representation of a condenser microphone.



Microphone (Brüel and Kjær)

1 Diaphragm
2 Backplate
3 Mica isolation
4 Pressure equalising vent

Choosing a microphone

A wide variety of different condenser microphones is available and when choosing the most appropriate, consideration should be given to the range of frequencies and sound levels to be measured, the nature of the sound field and the environment in which it is to operate.

Sound fields

Response of a microphone is influenced by reflection and refraction caused by the presence of the microphone in the sound field. The effect, which causes a higher sound pressure at the microphone diaphragm, depends upon the size of the microphone and is most noticeable at frequencies where the wavelength of the sound is less than the dimensions of the microphone. For example, with a one inch diameter microphone the effect is significant above about 2000Hz, while for a one-half inch microphone the effect does not occur until around 5000Hz. The magnitude of the effect also depends upon the direction of the sound at the microphone and hence depends upon the nature of the sound field. We can distinguish two main types of acoustic field, the free field where sound arrives from a well-defined direction and the diffuse or reverberant field where sound arrives from all directions. To correctly measure sound levels the characteristics of the microphone chosen have to be suitable for the type of sound field.

Microphone characteristics

In general microphone characteristics are given in terms of the free field, pressure, or random incidence response and reference is made to free field, pressure, or random incidence type microphones. It should be realised that any microphone has a free field response, a pressure response and a random incidence response and is named after those of its responses the manufacturer has arranged to be nearly flat as a function of frequency.

Free field microphone

A free field microphone is designed to compensate for the disturbance caused by its own presence in the sound field and is designed to give a flat frequency response to sound waves arriving normal to its diaphragm. At other angles, because the system compensation (attenuation) is too large, the microphone will read values that are too low. A free field microphone should thus always be pointed at the source of noise and is best suited for measurements where the direction of the noise is well defined.

Random incidence microphone

When used in a diffuse field a free field microphone will underestimate the contribution to the total sound level of sound coming from all directions other than normal to the diaphragm. Hence, when measuring diffuse sound fields it is advisable to use a random incidence microphone which is designed to have a flat response when sound waves arrive simultaneously from all directions. A random incidence microphone, on the other hand, will overestimate the pressure in a sound wave incident normally on its diaphragm.

Pressure microphone

The pressure microphone has a uniform frequency response to the sound field as it exists including the effect caused by its presence. The sensitivity of a pressure microphone does not depend upon the angle of incidence of the sound wave but only upon the frequency. A pressure microphone is suited for those measurements where the actual sound pressure that exists, regardless of the fact that the microphone disturbs the field, is required, for example, audiometer calibration and the measurement of the sound pressure at a surface. The response of a pressure microphone is not too different from a random incidence microphone and can be used to measure diffuse sound fields. Also, if the diaphragm is orientated so that sound waves in a free field graze the microphone diaphragm, a pressure microphone will measure the free field sound pressure.

Microphone size

The consequences of a microphone disturbing the sound field it is to measure become less serious the smaller the microphone is, and generally the smaller the microphone the better its frequency response. However, it may have a lower sensitivity and often a compromise must be made between the conflicting requirements. An additional advantage of choosing a small microphone is that it will remain more omni-directional up to a higher frequency.

Analysis section

This is generally the most complex section, containing a wide variety of circuits to condition, weight, integrate or analyse the sound signal. In the simplest case the analysis section will consist only of one of the standard weighting networks but it could just as easily comprise a set of octave, third-octave or narrow band filters. Additionally, the analysis section may compute time dependent quantities such as Leq, L_{AX} or L_{10} or even produce a full statistical analysis of the noise. The most important consideration when choosing the analysis system is to ensure that it will deal adequately with the frequency and sound level range of the noise being measured. To date, developments have been in the field of processing data but before long perhaps 'intelligent' networks will improve both measurements in the field and cumbersome laboratory test procedures.

Output section

The output will generally be a calibrated meter with needle or digital display, although in some cases alphanumeric printers are employed. The meter needle and digital display will normally have a standardised response time that meets relevant international standards. The most frequently encountered are the 'slow' response, which corresponds to a signal averaging time of about 1 sec, and the 'fast' response, which is equivalent to an averaging time of about 0.125 sec. The choice of whether to use the fast or slow time constant depends mainly on the character of the noise field. The aim is to obtain a meter movement that can be averaged easily by eye.

Two further meter time constants are 'impulse', which is used for measuring short duration sounds, and 'peak', which is used when the maximum value of events are required.
It should be realised that the value recorded for an impulsive sound event will depend upon the time constant used and will in general be much less than the true peak value if the impulse is short. In addition, the true peak value will only be recorded if the frequency response of the system is not limited, e.g. by weighting networks. Most measuring systems have a 'linear' setting on the weighting section and this should always be used when measuring peak levels and it is also advisable to use it when measuring octave band levels. However, there is no accepted definition of linear and some linear weightings have a significantly poorer frequency range than others, something that has to be considered if accurate measurements, especially of peak levels, are required.

Calibration
A measuring system should be calibrated every time it is used, both before and after use. For systems measuring noise this can be achieved by using a calibrator or piston phone, devices which produce a known and constant sound pressure level, usually at one frequency. Although such a procedure does not give a thorough check of the system it does usually indicate when problems have occurred. Once a year it is a good idea to subject all equipment to a complete calibration to ensure that their frequency response, dynamic range, noise floor, etc, are all within manufacturers' specifications.

Checklist for use when making sound measurements

- Does the equipment have the required specifications to perform the measurements intended?
- Make a list of the instrumentation used and even record the reference numbers – with more complicated set-ups a sketch is often useful.
- Calibrate the whole measuring system.
- Record the positions at which measurements are made. Again a sketch showing, for example, positions of the sources, other buildings and distances, is often very useful.
- Make noise measurements keeping a careful note of all instrument settings, such as weightings, attenuations, time constants, etc. Always record when any of these are changed during the course of the measurements.
- If tape recordings are made it is always a good idea to record details of settings on to the tape. The advantage of making recordings is that if necessary detailed laboratory analysis, e.g. narrow band analysis to highlight dominant tones in a noise, can be undertaken.
- When making outdoor measurements note the meteorological conditions, especially wind speed and direction. If necessary more detailed information can be obtained from the local meteorological office.
- Remember that your presence close to the microphone can influence the

recorded sound levels. Thus, if possible, do not stand directly behind the microphone. It is sometimes a good idea to have the microphone remotely mounted away from the display which has to be read.

- If possible measure the background noise in the absence of the noise that is to be measured. Unless the sound level (measured when the sound source is operating in the presence of background noise) is more than 10dB above the level of the background noise alone, a correction has to be made to the measured level to obtain the level of the source alone. If the difference is more than 10dB the measured level of source and background noise will give the source noise level to an accuracy of within 0.5dB. For a difference of between 3 and 10dB a correction, obtained from Figure 2.13, should be applied. If the difference is less than 3dB, which means that the source level is less than the level of the background noise, a reliable measure of the source level alone cannot be obtained.

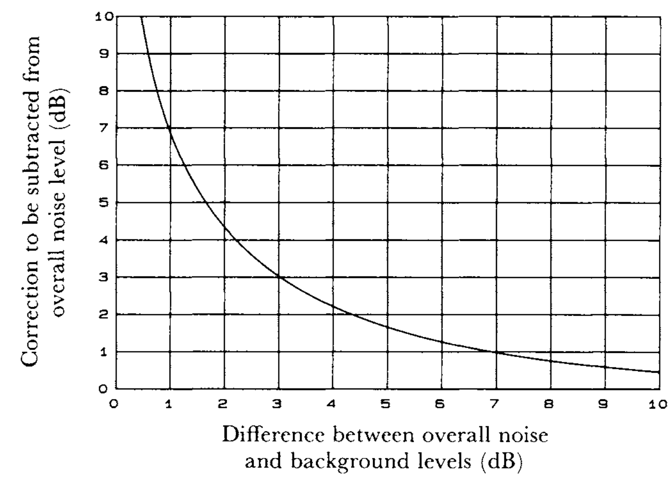


Figure 2.13 Graph to enable correction to be made to the measured noise level to account for the presence of background noise.

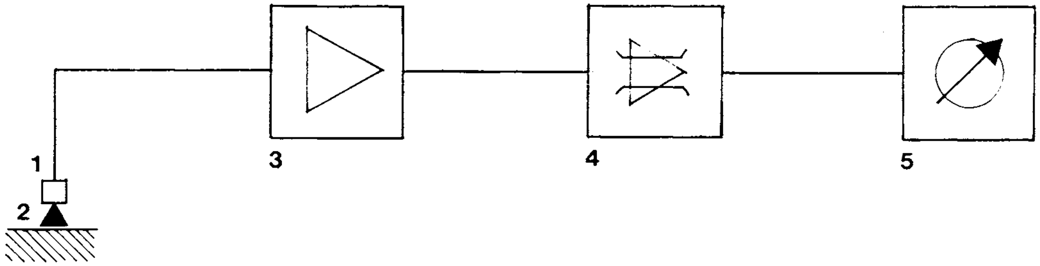
2.2.3 Vibration measurement

Noise in a building is often the result of mechanical vibrations linked directly to a building structure or structural element; a knowledge of the vibration characteristics of a surface can provide valuable information with which to identify the source of the noise. For example, comparing the noise spectrum at one location with the vibration spectrum at the base of a compressor located at some distant location can be very helpful in assessing whether or not the compressor is the source of the noise.

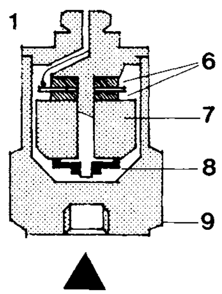
Accelerometers

The vibration of a surface is most often measured using an accelerometer. An accelerometer is an electro-mechanical transducer, the active elements of which are normally pieces of piezo-electric ceramic in the form of thin discs. A relatively heavy mass rests on the discs and is preloaded by means of a stiff spring, the whole assembly being contained within a metal housing with a thick base. When the surface to which the accelerometer is attached vibrates, the mass exerts a variable force on the discs which, because they are piezo-electric, develop a variable charge proportional to the force and therefore to the acceleration of the mass. A schematic diagram of an accelerometer is given in Figure 2.14.

Figure 2.14 Piezo-electric accelerometer.



Detail



Vibration measurement

1 Accelerometer
2 Magnetic attachment to vibrating surface
3 Voltage or charge preamplifier
4 Frequency analyser
5 Read out or recorder
6 Piezo-electric element
7 Mass
8 Spring
9 Base

Selecting an accelerometer

An accelerometer has to be chosen with sufficient sensitivity and an adequate frequency range and, in addition, it should be of such a mass that it does not load the surface to which it is attached. As sensitivity and low weight do not go together the choice may often have to be a compromise. In general the accelerometer should have a weight at least ten times less than the vibrating object.

As with microphones, the environmental conditions can also influence the choice of accelerometer.

Accelerometer preamplifiers

To avoid loading the accelerometer and thus reducing its output and limiting its low frequency response, the correct preamplifier should be selected. Either a voltage or a charge preamplifier can be used although the charge preamplifier is more universally useful as it can be used with any length of accelerometer cable. With the voltage preamplifier it is recommended that only fixed, relatively short, cables be used as both sensitivity and low frequency performance deteriorate as the cable length increases.

Integrating networks
To obtain the vibration velocity or displacements the signal from the accelerometer and preamplifier is conditioned by an integrating network. These tend to reduce the lower frequency performance of the overall system.

Output display
To indicate the measured vibration levels a simple electronic voltmeter is sufficient. The acceleration is obtained by dividing the voltmeter reading by the overall sensitivity of the accelerometer and preamplifier. However, more sophisticated meters are available which allow acceleration values to be read directly.

Use of sound level meter
It is often convenient to use a sound level meter as the signal detector. By a simple attachment the accelerometer can replace the microphone and the sound level meter can be used as both preamplifier and display. If, when fitted with a microphone, the meter is calibrated using a pistonphone or acoustic calibrator, acceleration levels can be obtained from the meter reading using the relationship

$$\text{acceleration} = \frac{\text{Sm}}{\text{Sa}} \times 10^{L/20} \times 2 \times 10^{-5}\ \text{msec}^{-2}$$

where Sm = microphone sensitivity in mV/Pa
Sa = accelerometer sensitivity in mV/msec^{-2}
L = acceleration level in dB read from sound level meter.

Calibration
The accelerometer, preamplifier and display can be calibrated using a calibrator which consists of a battery operated vibrator which produces a reference level of known peak amplitude (usually 9.81 msec^{-2}) at a known frequency. If such a calibrator is available, the procedure outlined for use with the sound level meter will not be necessary.

Accelerometer mounting
The accelerometer should be mounted rigidly to a flat section of the vibrating system. The best method is to screw it to a threaded stud which itself is screwed into the vibrating surface. Other methods include sticking the accelerometer to the surface with, for example, wax or double sided adhesive tape. If the surface is metallic it is often convenient to use a small magnet to which the accelerometer is fixed with a screwed stud.

Cables
To stop unwanted noise being generated it is important that the section of the accelerometer cable close to the accelerometer be fixed to the vibrating surface to reduce the relative movement.

2.3 Sound at a Point

2.3.1 Several sound sources

When a number of sound sources operate simultaneously the resultant sound pressure level is not the sum of the sound pressure levels of the individual sources. At any point the instantaneous sound pressure is the sum of the pressures produced by the individual sources, i.e.

$$p(t) = p_1(t) + p_2(t) + \ldots$$

but what we need to know is the rms pressure and its corresponding sound pressure level and we have to consider two possibilities.

If the sources are coherent – that is, they all produce the same waveform as a function of time – the pressures will have to be summed taking into account the phase relationships between the waves. Thus at some points the waves may arrive in phase, giving a considerable increase in pressure, while at other points they may be out of phase and cancel, giving zero pressure (see section 4.3.3). For waves which are exactly in phase the resultant rms pressure, p_T, is the sum of the rms pressures produced by the individual waves, i.e.

$$p_T = p_1 + p_2 + \ldots$$

For two identical waves p_1 and p_2 will be the same so that $p_T = 2p_1$ and the resultant sound pressure level is given by

$$\mathrm{SPL} = 20 \log \frac{p_T}{p_0} = 20 \log \frac{2p_1}{p_0}$$

$$= 20 \log \frac{p_1}{p_0} + 20 \log 2$$

$$= \mathrm{SPL}_1 + 6$$

Thus, when two identical waves act, the sound pressure is doubled and the sound pressure level increases by 6dB. Pressure doubling also occurs at a hard surface when a sound wave is reflected from the surface.

Incoherent sources

When the sources are incoherent, which is the usual case, there is no fixed phase relationship between the waves and we find that the resultant rms acoustic pressure p_T is given by

$$p_T = \sqrt{p_1^2 + p_2^2 + p_3^2 \text{ etc}}$$

or

$$p_T^2 = p_1^2 + p_2^2 + p_3^2 \text{ etc}$$

that is the resultant mean square pressure is the sum of the mean square pressures produced by the individual waves.

In practice it is usually the sound pressure levels which are known and to obtain the mean square pressures it is necessary to divide each by 10 and take the antilogarithm as shown in equation 2.3. Thus if the sound pressure levels are L_1, L_2, etc, then

$$p_1^2 = 10^{L_1/10} \times p_0^2$$

$$p_2^2 = 10^{L_2/10} \times p_0^2 \text{ etc}$$

and $$p_T^2 = p_o^2[10^{L_1/10} + 10^{L_2/10} + \ldots]$$

and the resultant sound pressure level

$$L_T = 10 \log \frac{p_T^2}{p_0^2}$$

is given by

$$L_T = 10 \log [10^{L_1/10} + 10^{L_2/10} + \ldots] \tag{2.12}$$

For N identical sources

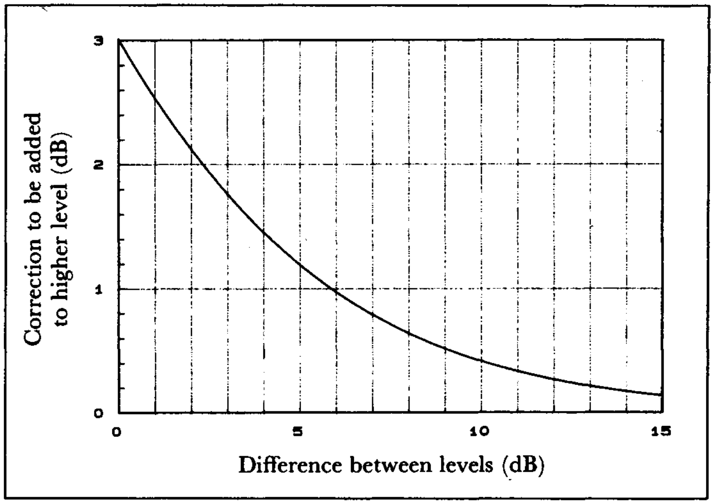
$$L_T = 10 \log [N \times 10^{L_1/10}]$$

$$= 10 \log {}^{L_1/10} + 10 \log N$$

$$= L_1 + 10 \log N \tag{2.13}$$

Thus, for two identical sources, the level is 3dB higher than with one source, 5dB higher with three sources and 6dB higher with four sources. In fact every time the number of sources is doubled the level increases by 3dB. When the sources do not give equal sound levels, equation 2.12 must be used to obtain the resultant levels. However, another way is to use the chart shown in Figure 2.15 and add the levels two at a time. For example, to add 63 and 68dB we note that the difference is 5dB and from the chart we find that the correction to be added to the higher level is 1.2dB. Hence the resultant level is 69.2dB. If we now wanted the resultant of 63, 68 and 70dB we would have to add 69.2dB and 70dB. From the chart the correction is found to be 2.6dB so the resultant of 63, 68 and 70dB is $70 + 2.6 = 72.6$dB.

Figure 2.15 Graph to be used for the addition of sound pressure levels from two incoherent sources.



Reducing number of sound sources

A final point, which is often difficult for people not familiar with decibels to appreciate, is the effect on the sound pressure level of reducing the number of sources operating. For example, in a workshop with several machines it is often felt that because one or two machines may not be operating the level will be much reduced. This is not necessarily so. For example, if there are five similar machines each giving a sound level L_l the resultant level when all five operate together will be $L_l + 10 \log 5$, that is $L_l + 7$dB.

When four of the machines operate the resultant sound pressure level will be $L_l + 10 \log 4$, or $L_l + 6$dB. That is a difference of only 1dB which will be subjectively insignificant. With three machines operating the level will drop by 2dB from that obtained when five are operating which, again, is unlikely to be noticed. As it was necessary to double the number of machines to increase the level by 3dB so it is necessary to halve the number of machines to reduce the level by 3dB which is thought to be the change in sound level required for the average person to detect a change in the loudness of the noise.

2.3.2 Radiation from a source

Point source

A small source radiating uniformly in all directions is the simplest to consider. At a distance r from the source, the power, W, is passing through the surface of a sphere of radius r, that is, through an area of $4\pi r^2$. The intensity, I, on this surface is thus

$$I = \frac{W}{4\pi r^2} \text{ watts/m}^2$$

As the energy flow has a defined direction the intensity is related to the mean square pressure by

$$I = p^2/\rho c$$

Hence the mean square acoustic pressure at a distance r from the source can be expressed as

$$p^2 = \frac{W\rho c}{4\pi r^2}$$

and the sound pressure level, L, is therefore

$$L = 10 \log \frac{W\rho c}{4\pi r^2 p_0{}^2} \quad \text{dB}$$

or $$L = 10 \log W - 20 \log r - 10 \log 4\pi - 10 \log \frac{\rho c}{p_0{}^2} \quad \text{dB}$$

As 10 log 4π is approximately 11dB and 10 log $\rho c/p_0{}^2$ is approximately 120dB the expression for the sound pressure level becomes

$$L = 10 \log W - 20 \log r + 109\text{dB} \tag{2.14}$$

If the power of the source is given in terms of the sound power level then the sound pressure level at distance r is given by

$$L = SWL - 20 \log r - 11 \quad \text{dB} \tag{2.15}$$

If this equation is written as SWL + K then K can be obtained from Figure 2.16.

Inverse square law

If the sound pressure levels at distances r_1 and r_2 from the source are L_1 and L_2 respectively then

$$L_1 = \mathrm{SWL} - 20 \log r_1 - 11$$

and $$L_2 = \mathrm{SWL} - 20 \log r_2 - 11$$

so that

$$L_1 - L_2 = 20 \log r_2 - 20 \log r_1$$

$$L_1 - L_2 = 20 \log \frac{r_2}{r_1} \tag{2.16}$$

$L_1 - L_2$ is the change in sound pressure level that occurs when the distance from the source changes from r_1 to r_2 and in general is found by evaluating 20 log r_2/r_1. In particular if r_2 is equal to $2r_1$, that is, the distance from the source is doubled, the change in sound pressure level is 20 log 2 or 6dB. The decrease in level of 6dB each time the distance from the source is increased or the increase of 6dB in level each time the distance from the source is halved is known as the inverse square law.

2.3.3 Directivity

Directivity factor

The concept of directivity was touched upon in section 1.1.2 but here we give it a more formal definition. The directivity of the source is usually given in terms of the directivity factor or the directivity index. The directivity factor Q is the ratio of the actual intensity at a given point to the intensity at the point had the source radiated uniformly, so that

$$Q = \frac{I}{I_{av}}$$

where $$I_{av} = \frac{W}{4\pi r^2}$$

and I is the intensity at a distance r from the source along some radius.

Hence $$I = \frac{QW}{4\pi r^2}$$

and the mean square acoustic pressure is given by

$$p^2 = \frac{QW\rho c}{4\pi r^2} \qquad (2.17)$$

and so, taking logarithms as before, the sound pressure level, L, can be expressed as

$$L = SWL - 11 - 20 \log r + 10 \log Q \quad dB \qquad (2.18)$$

The quantity 10 log Q is known as the directivity index (DI). Alternatively the sound pressure level can be obtained from

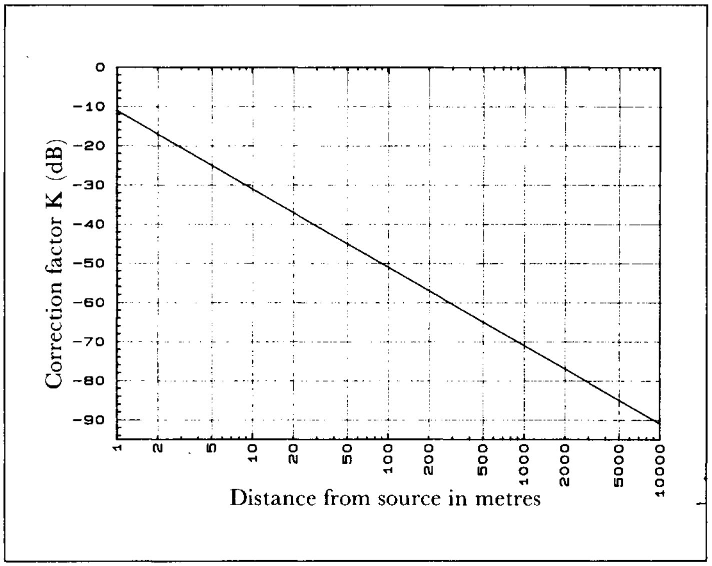
$$L = SWL + K + DI$$

where K is again obtained from Figure 2.16.

As the dependence of the sound level on distance is still contained in the 20 log r term, the sound level will decay at 6dB for each doubling of distance along any radius from the source.

The values of the directivity index for some simple situations are given in Figure 2.17. For example, if a uniformly radiating point source is placed on a hard flat surface the sound power will be radiated above the plane only. Hence all the energy will now effectively radiate into a hemispherical surface. Thus the intensity at any point will be twice the intensity at the point had the source radiated uniformly, in all directions, thus the directivity factor is 2 and the directivity index is 3dB.

Figure 2.16 Graph giving the value of K = −(20 log r + 11) as a function of the distance r, in meters, from a point sound source.



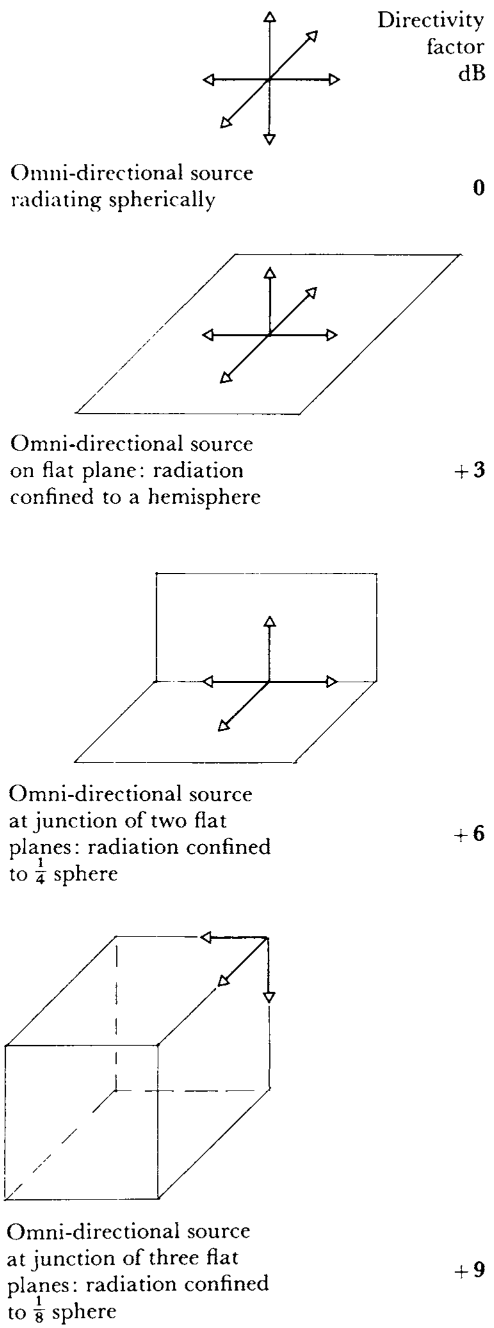


Figure 2.17 Directivity factors for some simple situations.

Example

An extract fan of total sound power level 105dB(A) re 10^{-12} watts is installed in a large external wall of a factory. What will the sound pressure level be at a house 60 m from the fan?

The sound level, L, can be found by using

$$L = SWL - 20 \log r - 11 + Q \quad dB$$

but in this case it must be realised that the fan will radiate noise both externally and internally and it is probably sensible to assume that half the power is radiated in each direction.

Hence, the effective sound power level in the equation is 102dB(A) as a halving of sound power is equivalent to a reduction of 3dB in the sound power level.

From Figure 2.16 the correction $-20 \log r - 11$ for $r = 60$m is -46.6dB(A).

So that

$$L = 102 - 46.6 + Q \quad dB(A)$$

As the fan is effectively radiating into a hemisphere Q is +3dB.

So $\quad L = 102 - 46.6 + 3 \quad dB(A)$

$\quad\quad L = 58.4dB(A)$

By deciding to put Q = +3dB it has been assumed that the fan is radiating into a hemispherical surface. In some situations, for example because the source is also close to the ground, the power will effectively radiate into a quarter sphere and the value of Q would be +6dB.

In addition, if the measurement point is not in line with the fan axis then as the frequency increases the fan is likely to have a directional radiation pattern which will also have to be allowed for.

Variance with frequency

The directivity index will almost certainly vary with frequency, the source becoming more directional as the frequency increases. For frequencies at which the wavelength of the sound is larger than the dimensions of the source the radiation will be more uniform.

Practical noise sources

Practical noise sources are not usually point sources but can be regarded as such at distances from the source exceeding three times the largest source dimension. However, practical sources may inherently radiate more power in some directions than others and the directivity index will normally have to be obtained by making measurements of the sound pressure level around the source.

Directivity index can be positive or negative, so that although the source radiates more energy in some directions than the equivalent uniform source of the same power, it will, in other directions, radiate less power. Thus, there may be situations in which elementary noise control can be achieved by arranging that a directional source radiates most of its power away from noise sensitive areas.

Example

A standby generator has a manufacturer's quoted sound power level of 90dB(A) re 10^{-12} watts. If it is placed on a flat hard surface, what will be the sound pressure level at a distance of 135 m from the generator?

The sound pressure level, L, is given by

$$L = SWL - 20 \log r - 11 + Q \quad dB(A)$$

From Figure 2.16 the correction $-20 \log r - 11$ for

$$r = 135 \text{ m is } -54dB$$

As the generator is on a hard flat surface then, from Figure 2.17, we see that $Q = 3dB$

$$\therefore \quad L = 90 - 54 + 3 \quad dB(A)$$
$$L = 39dB(A)$$

If, instead of the sound power level of the generator, the sound pressure level at a distance of 5 m had been given as 68dB(A), the level, L, at 135 m would have been obtained using

$$L = 68 - 20 \log \left[\frac{135}{5}\right] \quad dB(A)$$

i.e. $L = 39.4dB(A)$

When starting from a measured sound pressure level at a known distance it is important to know under what conditions the measurement was made. For example, if the sound level at 5 m from the generator had been measured under anechoic conditions then the level used in the above equation would have to be increased by 3dB to 71dB(A) to account for the fact that the generator is standing on a hard surface.

Chapter 3

Sound in the built form

3.1 Sound Absorption

3.1.1 Absorption coefficient

Definition

All materials can to a greater or lesser extent absorb sound. That is, they can convert the mechanical energy of molecular vibration into heat. The performance of a material is usually quantified in terms of its sound absorption coefficient α which is defined as

$$\alpha = \frac{\text{sound energy not reflected from material}}{\text{sound energy incident upon material}}$$

For a perfect absorber the value of α would be 1, while for a perfect reflector α would equal zero.

Effect of incidence angle

The value of α for a given material will depend upon the angle at which the sound strikes the surface of the material. However, in most cases this angular variation is not important because in practice sound will tend to strike a surface over a wide range of angles. The value of α, measured under conditions where sound is encouraged to fall at all possible angles on to the material, is known as the random incidence absorption coefficient and is denoted by $\bar{\alpha}$.

Frequency dependence

The behaviour of a sound absorber will also depend upon the frequency of the sound and values of $\bar{\alpha}$ are normally given for octave or third-octave bands covering the frequency range 100–4000Hz. Table 3.1 contains values for some commonly used materials.

In general, third-octave band values of absorption coefficients will not be significantly different from octave band values.

Table 3.1 Coefficients of sound absorption

	Octave band centre frequency (Hz)					
	125	250	500	1K	2K	4K
Ceiling						
Overhead sound absorbers, one 1200 × 450 × 50 mm panel every sq m, parallel pattern	0.28	0.58	0.96	0.91	0.86	0.81
Fissured mineral tiles, 300 mm ceiling void	0.3	0.35	0.4	0.55	0.8	0.7
Metal tiles (5 per cent perforated), 20 mm glass fibre quilt, ceiling void	0.13	0.27	0.55	0.79	0.9	1.0
Metal planks, slots between planks (14 per cent free area) glass fibre behind	0.5	0.7	0.8	1.0	1.0	1.0
13 mm plasterboard ceiling over large air space	0.2	0.2	0.2	0.1	0.05	0.05
13 mm acoustic plaster on metal lathing	0.05	0.2	0.5	0.8	0.8	0.8
Profiled metal deck	0.1	0.3	0.3	0.1	0.1	0.2
Walls and Linings						
Brickwork – fairfaced	0.02	0.02	0.02	0.03	0.04	0.05
– painted	0.01	0.01	0.01	0.02	0.02	0.02
– plastered	0.02	0.02	0.03	0.03	0.04	0.05
Blockwork – fairfaced	0.2	0.5	0.5	0.4	0.5	0.4
Concrete – textured finish	0.01	0.02	0.04	0.06	0.08	0.1
9 mm acoustic plaster on solid wall	0.02	0.08	0.3	0.6	0.8	0.9
Woodwool slabs on solid backing – 50 mm thick	0.2	0.2	0.6	0.8	0.7	0.9
– 100 mm thick	0.3	0.8	0.9	0.7	0.7	0.8
Prescreed woodwool slabs on 600 mm air gap	0.4	0.4	0.7	0.7	0.7	0.8
9 mm plasterboard on 18 mm air space filled with glass fibre to solid backing	0.3	0.2	0.2	0.05	0.05	0.05
12 mm plywood on 30 mm airspace filled with glass fibre, to solid backing	0.4	0.2	0.2	0.1	0.1	0.05
6 mm glass, large panes	0.3	0.3	0.2	0.1	0.05	0.05
Stretched, lightweight fabric wall hanging	0.04	0.1	0.2	0.5	0.6	0.5

Table 3.1 (cont.)	Octave band centre frequency (Hz) 125	250	500	1K	2K	4K
Heavy curtain material hung in folds	0.06	0.16	0.3	0.55	0.65	0.65
Glass fibre quilt to solid backing – 25 mm thick	0.1	0.4	0.6	0.7	0.8	0.8
– 50 mm thick	0.3	0.6	0.8	0.9	0.8	0.8
– 100 mm thick	0.5	0.8	0.8	0.9	0.9	0.9
Glass fibre 100/100 mm airspace	0.5	1.0	0.9	0.8	0.6	0.4
Mineral wool to solid backing – 25 mm thick	0.01	0.3	0.7	1.0	1.0	1.0
– 50 mm thick	0.3	0.8	1.0	1.0	1.0	1.0
Mineral wool 25/25 mm airspace	0.1	0.4	0.7	1.0	1.0	1.0
50/50 mm airspace	0.5	0.7	0.9	0.9	0.9	0.8
Floors						
Thin contract carpet on solid floor	0.01	0.04	0.05	0.18	0.3	0.2
Thick carpet on underlay	0.07	0.23	0.69	1.0	1.0	1.0
Rubber flooring, vinyl sheet	0.02	0.04	0.05	0.05	0.05	0.05
Marble, ceramic tiles	0.05	0.05	0.05	0.05	0.05	0.05
Reinforced concrete, grano	0.02	0.02	0.02	0.04	0.05	0.05
Water (swimming-pool), ice (rink)	0.01	0.01	0.01	0.02	0.02	0.03
Other						
Seated audience, per person*	0.33	0.4	0.44	0.45	0.45	0.45
Standing adults, per person*	0.15	0.38	0.42	0.43	0.45	0.45
Wooden seats*	0.1	0.2	0.3	0.3	0.3	0.35
Upholstered seats*	0.24	0.26	0.27	0.31	0.37	0.38
Shading reduction factors (for floor finishes absorption under seating)	–20%	–30%	–40%	–50%	–60%	–80%
Recess off main space						

$$A = \frac{A_R \times S}{A_R + S}$$

where A = absorption contribution from recess to main space
A_R = absorption in recess
S = area of opening between recess and main space

* These figures are the total absorption in m^2.

Noise reduction coefficient
This is the average of the absorption coefficients in the 250, 500, 1000 and 2000Hz octave bands and is used to quantify the performance of a material.

Absorption
The amount of energy a material absorbs from a sound field will depend upon the absorption coefficient, $\bar{\alpha}$, and the area of material, S, involved. The absorption A is given by

$$A = S \times \bar{\alpha} \text{ m}^2$$

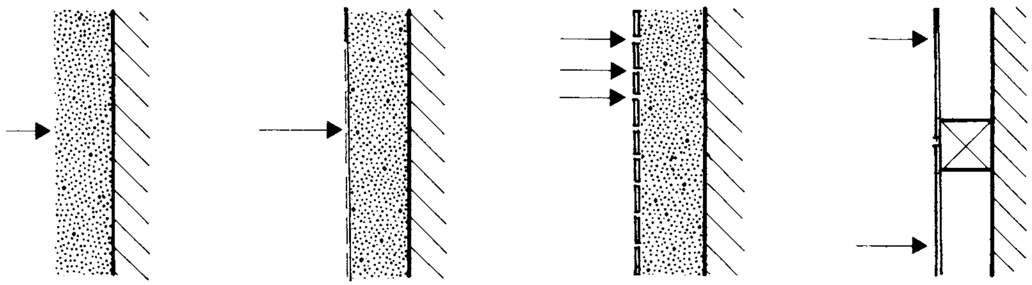
If there are a number of different materials in the enclosure having areas S_1, S_2, S_3, etc, and random absorption coefficients $\bar{\alpha}_1$, $\bar{\alpha}_2$, $\bar{\alpha}_3$, etc, then the total absorption is given by

$$A = S_1\bar{\alpha}_1 + S_2\bar{\alpha}_2 + S_3\bar{\alpha}_3 + \ldots$$

The absorption is usually calculated in this way for each octave or third-octave band of interest. In some cases it is more convenient to measure the total absorption in an enclosure by measuring the reverberation time and then using the Sabine or Norris-Eyring equation (section 3.2.3).

3.1.2 Types of sound absorber

A number of general types of sound absorber are illustrated in Figure 3.1. They are as follows.



Fibrous   Fibrous + membrane   Fibrous + perforated panel   Panel

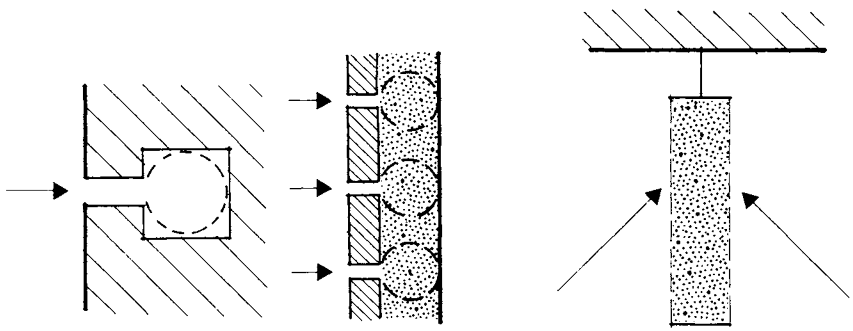


Figure 3.1 Some types of sound absorber.

Fibrous absorbers

These are the most common type of absorber, typical examples being glass fibre and mineral wool blankets. They provide absorption over a wide range of frequencies and are non-flammable, but because of their fibrous nature they must be used with care, especially in ventilation systems where contamination of the air supply can occur. Fibrous absorbents come in a variety of thicknesses and densities so that by choosing correctly one can usually get the absorption required over the necessary frequency range. Generally, quilts for sound absorption are denser than those for thermal insulation. To obtain adequate low frequency performance the thickness of the material has to be increased, say, from 30 mm, which is a good general thickness, to 75–100 mm. Alternatively, spacing a 30 mm absorbent layer away from a rigid backing with a small air space of, say, 50 mm, will also improve the low frequency absorption.

Fibrous layer covered with thin impervious membrane

These are used in wet and/or corrosive environments, in ducts with high flow velocity and in areas where contamination of the air supply must be avoided. The fibrous absorbent blankets are enclosed in very thin impervious membranes; this has the effect of improving the low frequency performance of the material but reduces its absorption at the higher frequencies compared with the same thickness of uncovered material.

Fibrous layer covered with perforated panel

To give protection to the absorbent and to improve its appearance it is often covered with a perforated material of some kind, such as textiles, wood or metal sheet. Provided the perforations are a sufficient proportion of the total area the absorption characteristics will not be changed. For thin panels an open area of about 15–20 per cent is sufficient while for thicker panels the open area should be increased. If the open area is less than 15 per cent there is a tendency for the low frequency performance of the absorbent to improve while its high frequency performance falls off.

Panel or membrane absorber
This consists of a solid membrane of, say, plywood or hardboard over an air space which may contain some fibrous absorbent. The absorption is limited to a small range around the resonant frequency, f_r, of the system, which is given by

$$f_r = \frac{1}{2\pi}\sqrt{\frac{\gamma P}{Md}} \quad \text{Hz}$$

where M is the mass per unit area of the panel, d is the thickness of the cavity, P is atmospheric pressure and γ = the ratio of specific heats of air at constant pressure and constant volume = 1.4.
If M is in kgm^{-2} and d is in metres then

$$f_r = \frac{60}{\sqrt{Md}} \quad \text{Hz}$$

If the fibrous absorbent completely fills the cavity the resonant frequency may be reduced slightly.
This type of absorber is used for low frequency absorption, and modular units are available which find most use in studio design.

Example
An absorber consists of a 0.5 × 0.5 × 0.15 m deep unit. It is constructed from 20 mm blockboard with a 3 mm plywood membrane.
As the density of plywood is 580 kgm^{-3}, the surface mass, M, of the plywood membrane is

$$M = 580 \times 0.003 \ kgm^{-2}$$
$$M = 1.74 \ kgm^{-2}$$

Thus the resonant frequency is given by

$$fr = \frac{60}{\sqrt{1.74 \times 0.15}}$$

$$fr = 117\text{Hz}$$

Resonator absorbers

The simplest form of resonator absorber is the Helmholtz resonator which is an enclosed volume of air connected to the room or enclosure by a small opening or neck. The resonator frequency, f_r, of the system is given by

$$f_r = \frac{c}{2\pi}\sqrt{\frac{\pi a^2}{V\left\{l + \frac{16a}{3\pi}\right\}}}\ \text{Hz}$$

where a is the radius of the neck (m), V is the volume of the enclosed air (m^3), l is the length of the neck (m) and c is the velocity of sound ($msec^{-1}$). The original use of the Helmholtz resonator was to increase the reverberation time at certain frequencies and, unless some absorbing material is introduced in to the neck, no absorption will occur. Lining the neck and cavity with an absorbent material will damp the resonance but increase the range of frequencies over which absorption occurs. This type of absorber is not normally considered for general acoustic treatment but only where a long reverberation time in a narrow frequency range needs to be controlled. They are most effective at controlling low frequencies.

Example

A resonator consists of an enclosed volume of $5 \times 10^{-4} m^3$ connected to the outside by a neck of length 0.03 m and radius 0.014 m.

Its resonant frequency is

$$fr = \frac{340}{2\pi}\sqrt{\frac{\pi(0.014)^2}{5.10^{-4}\left\{0.3 + \frac{16 \times 0.014}{3\pi}\right\}}}$$

$$fr = 250\text{Hz}$$

Multiple resonator

Such a system can be achieved by mounting a membrane of, say, thin metal or plywood with circular or slit perforations over an enclosed air space which may contain some fibrous absorbent. Each hole or perforation acts with an effective volume behind it as a resonator. There is usually no need to separate the resonator cavities from one another by partitions. This type of absorber is identical with the fibrous layer covered with a perforated panel when the open area is less than 15 per cent. Its absorption characteristics will usually peak at a frequency governed by the choice of hole size and spacing but will be reasonably broad band if absorption material is included in the air space.

Functional absorbers
These are slabs of absorbing material such as mineral wool or glass fibre, usually about 1 m^2 in area and 50 mm thick, which can be suspended freely in a room. They are normally very efficient as sound absorbers because sound can impinge on both sides. They are particularly useful in large enclosures such as workshops and large factory spaces.

3.2 Room Acoustics

Diffuse sound field
When a sound source is placed in an enclosure the sound no longer radiates away from the source but is reflected back and forth between the walls of the enclosure. The sound energy grows to a level that depends upon the power of the source and also upon how much energy is removed by absorption at the surfaces of the enclosure. To begin with we shall consider only enclosures in which the sound energy is uniformly distributed, that is, where the sound field is diffuse. This is normally the case for enclosures with conventional aspect ratios but not for flat or shallow enclosures such as open plan offices or large factory spaces or in rooms with shapes that may cause some sound focusing.

Diffusion
Diffusion can also be obtained by large scale modelling of the room surfaces. When the face dimension of the projections are equal to half a wavelength the diffusion is very efficient. Random placing of areas of absorbing material on a sound reflecting wall or ceiling will also create diffuse conditions.

Acoustic devices
Both sound reflectors and diffusers find use in auditoria acoustics, the former to direct sound to distant seats to reinforce direct sound, the latter to help mix the sound from a number of sources, such as in an orchestra, to give a good balance to the sound.
Sound reflectors are usually large sheets of impervious material, such as plywood or perspex (either flat or curved), which are positioned to reflect the sound in a given direction. Often they will be found suspended above a stage. To reflect successfully, their smallest dimension must be greater than half a wavelength of the sound or diffraction occurs at the edges and reflection is reduced. Thus smaller sized panels actually act as diffusers (i.e. scatterers) rather than reflectors.

3.2.1 Reverberant sound level

Definition
When a sound source is operated in an enclosure the sound level throughout the body of the enclosure will reach a reasonably steady value. This steady value is referred to as the reverberant sound level and, as will be shown, its value depends upon the power of the source and the total absorption within the enclosure.

Energy balance
When the steady sound level is established the energy supplied per second by the source, which is equal to its sound power, is balanced by the removal of energy by absorption.

Thus $W = I \times A$

where W is the power of the source and I is the sound intensity falling on to the surfaces of the absorbers whose total absorption is A. The absorption is provided not only by the treatments on the room surfaces but also by the fittings within the room. The intensity incident upon a surface is related to the mean square reverberent sound pressure, $p^2{}_r$, by

$$I = p^2{}_r/4\rho c$$

where ρ is the density of air and c is the velocity of sound in air

$$\text{so } p^2{}_r = \frac{4W\,\rho c}{A}$$

and the reverberant sound pressure level L_r is given by

$$L_r = 10 \log \frac{p_r{}^2}{p_0{}^2} = 10 \log \frac{4W\,\rho c}{A\,p_0{}^2} \quad \text{dB}$$

where p_0 is the reference pressure, i.e. 2×10^{-5}Pa. We find by expanding the logarithm that

$$L_r = 10 \log W - 10 \log A + 10 \log 4 + 10 \log \frac{\rho c}{p_0{}^2}$$

$$\text{and as } \frac{\rho c}{p_0{}^2} \simeq 10^{12}$$

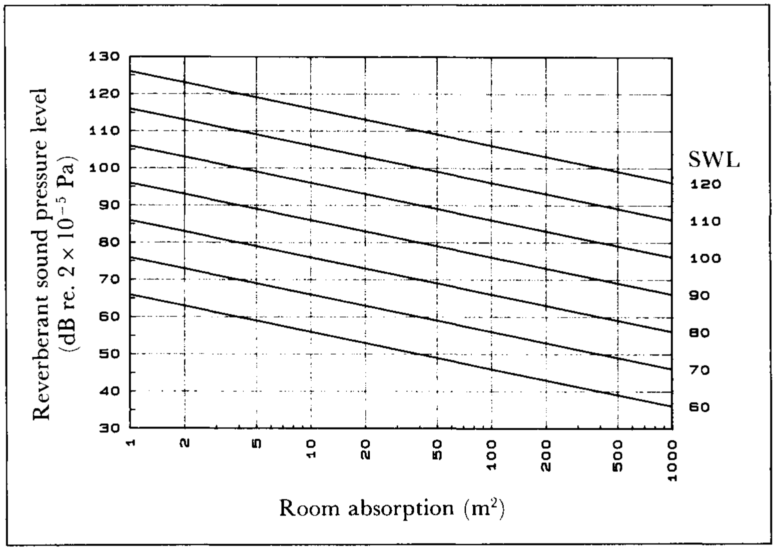
$$L_r = 10 \log W - 10 \log A + 6 + 120 \quad \text{dB}$$

or $$L_r = \text{SWL} - 10 \log A + 6 \quad \text{dB} \qquad (3.1)$$

where $\text{SWL} = 10 \log \dfrac{W}{10^{-12}}$

is the sound power level of the source.
Equation 3.1 is presented graphically in Figure 3.2.

Figure 3.2 Graph relating reverberant sound pressure level to source sound power level and room absorption.



Effect of absorption on reverberant SPL

Equation 3.1 shows that if the total absorption is doubled from A to 2A then the reverberant sound pressure level decreases by 3dB and similarly an increase from 2A to 4A produces a further 3dB decrease. Hence, doubling the amount of absorption reduces the reverberant sound pressure level by 3dB. Thus, area for area, the absorbent becomes less effective as the total amount is increased.

Number of sources

If there are N similar sources in an enclosure the reverberant sound pressure level is given by

$$L_r = SWL - 10 \log A + 6 + 10 \log N$$

Thus, if there are two sources, the level increases by $10 \log 2 = 3$dB, while four sources give an increase of $10 \log 4 = 6$dB.

When the sources are not similar equation 3.1 can still be used but now the sound power level must be the total sound power level of all the sources. This is found by adding up the sound power levels of the individual sources in the same way as the sound pressure levels from incoherent noise sources are added. Thus equation 2.12 or Figure 2.15 can be used to find the total sound power level.

3.2.2 Direct sound field

Definition

The sound reaching a point in an enclosure directly from the source without being reflected from any of the surfaces constitutes the direct sound field.

The mean square direct sound pressure p_d^2 is given by equation 2.17

$$p_d^2 = \frac{QW\rho c}{4\pi r^2}$$

where Q = directivity factor.

Some directivity factors for simple situations are given in Figure 2.17. The total mean square pressure at any position is the sum of direct and reverberant pressures.

So $\quad p_T^2 = p_r^2 + p_d^2$

$$p_T^2 = \frac{4w\rho c}{A} + \frac{Qw\rho c}{4\pi r^2}$$

and the resultant sound level $L = 10 \log \frac{p_T^2}{p_0^2}$ becomes

$$L = 10 \log \frac{W\rho c}{P_0^2} \left\{ \frac{4}{A} + \frac{Q}{4\pi r^2} \right\}$$

or $$L = SWL + 10 \log \left\{ \frac{4}{A} + \frac{Q}{4\pi r^2} \right\} \qquad (3.2)$$

Room radius

At points close to the source the direct field will dominate, but as one moves away from the source the reverberant field will take over. When the direct and reverberant sound pressure levels are equal

$$p_d^2 = p_r^2$$

and $$\frac{QW\rho c}{4\pi r^2} = \frac{4w\rho c}{A}$$

or $$r = \sqrt{\frac{QA}{16\pi}}$$

This distance r is often referred to as the room radius.

Effect of absorption on direct field

Addition of absorption to the enclosure only affects the reverberant field and as this is reduced the range of influence of the direct field is increased. To reduce the noise level of the direct field other control measures must be used.

Example

In an office the ventilation grilles are positioned in the ceiling and are a source of fan noise. The total absorption in the unfurnished office at mid frequencies is 150 m^2 so that the room radius is given by

$$r = \sqrt{\frac{2 \times 150}{16\pi}}$$

where Q = 2 as the grilles are situated in a large flat plane

$$\therefore \quad r = 2.4 \text{ m}$$

Thus anyone working within this distance of the source will be in the direct sound field and will not benefit from any room treatments when the office is furnished.

Correction

In the above it has been assumed that the power supplied by the source all goes into the reverberant field. This is not really correct if the reverberant energy is considered to be that which remains after the first reflection. With this modification the power supplied to the reverberant field becomes $W(1-\bar{\alpha})$, where $\bar{\alpha}$ is the average absorption coefficient of all the absorbers, and all the above equations are accordingly modified. As the analysis implicitly assumes that the average absorption coefficient is low, less than 0.1, the change makes only a small difference to the reverberant level and can in most cases be ignored.

3.2.3 Reverberation time

Sabine equation

When a sound source is switched on in an enclosure it takes a finite time for the energy to reach its equilibrium value. Similarly, if the source is switched off the energy will take a finite time to decay. The response of the enclosure is usually characterised by its reverberation time which is defined as the time taken for the sound level to decay by 60dB from its equilibrium value. For enclosures in which a diffuse sound field exists and where the average absorption coefficient is less than about 0.1 the reverberation time can be found using

$$T = \frac{0.16V}{A} \text{ sec} \tag{3.3}$$

Where V is the enclosure volume in m^3 and A is the total absorption in m^2. This equation is known as the Sabine equation.

Norris-Eyring equation
When the average absorption coefficient is greater than 0.1 the reverberation time can be found from

$$T = \frac{0.16V}{-2.3 \times S \log_{10}(1-\bar{\alpha})} \text{ sec} \qquad (3.4)$$

where S is the total surface area of the enclosure and $\bar{\alpha}$ is the mean absorption coefficient of all the surfaces given by

$$\bar{\alpha} = \frac{A}{S} = \frac{S_1\bar{\alpha}_1 + S_2\bar{\alpha}_2 + \ldots}{S_1 + S_2 + \ldots .}$$

This is known as the Norris-Eyring equation. In practice the Sabine equation is used in most cases unless the absorption throughout the enclosure is very non uniform.

Air absorption
In large enclosures the absorption of sound by the air can become important and should be included in the estimation of the total absorption. In an enclosure of volume V the absorption of the air is equal to 4mV, where m is the energy attenuation constant, and is dependent on humidity and frequency. Including the effect of air absorption the equations for reverberation time become

$$T = \frac{0.16V}{A + 4mV} \text{ sec}$$

$$T = \frac{0.16V}{-2.3 \times S \log(1-\bar{\alpha}) + 4mV} \text{ sec} \qquad (3.5)$$

The parameter m is equal to 2α where α is the sound attenuation coefficient. The sound attenuation coefficient has units of Nepers per metre (Npm^{-1}) and should not be confused with the acoustical energy absorption coefficient which is represented by the same symbol. Values of α can be found in the American National Standard SI 26-1978. However, some air absorption data at 20°C in a form suitable for reverberation time calculations are given in Table 3.2.

Frequency dependence
As the absorption of sound depends upon frequency it is clear that the reverberation time of an enclosure will also be frequency dependent; it is usual to estimate or measure reverberation times in octave or third-octave bands covering the frequency range 100–4000Hz.

Example

A lecture hall is of length 30 m, width 15 m and height 5 m. One side wall is of textured concrete up to a height of 1.5 m and above this it is glazed with 6 mm glass. The other walls are all of textured concrete. The floor is constructed of wood laid over an air space, while the roof is of structural concrete with a suspended ceiling of fissured mineral tiles placed 1 m from the structural roof. The hall is designed to seat 300 on lightly upholstered seats which occupy 70 per cent of the total floor space.

We need to calculate the reverberation times of a fully occupied hall in each of the octave bands from 125 to 4000Hz using the information given in Table 3.1 and also to estimate the reverberant sound pressure level in dB(A) for a source with acoustic power of 1mW in each octave band.

		Absorption $A = S \times \bar{\alpha}$ m² Octave band centre frequency (Hz)					
Item	Area S.m²	125	250	500	1K	2K	4K
Textured concrete	285	3	6	11	17	23	29
Suspended ceiling	450	135	158	180	248	360	315
Windows	75	23	23	15	8	4	4
Uncovered wooden floor	135	20	14	14	10	8	8
Floor covered by seating*	315	38	22	19	11	8	4
300 occupied seats		99	120	132	135	135	135
Air absorption†	4mV	1	2	6	11	20	51
Total absorption		319	345	377	440	558	546
Reverberation time $T = \frac{0.16\,V}{A}$ sec		1.13	1.04	0.95	0.82	0.65	0.66

* The shading of the floor caused by the audience has been accounted for as indicated in Table 3.1.

† The air absorption has been estimated assuming a temperature of 20°C and a humidity of 60 per cent.

Example (cont.)

To calculate the A weighted reverberant sound pressure level

	(Frequency Hz) 125	250	500	1K	2K	4K
Sound power, W, watts	0.001	0.001	0.001	0.001	0.001	0.001
10 Log W	−30	−30	−30	−30	−30	−30
Total absorption, A, m^2	319	345	377	440	558	546
10 log A	25	25.4	25.8	26.4	27.5	27.4
$L_R = 10\log W - 10\log A + 126\text{dB}$	71	70.6	70.2	69.6	68.5	68.6
A weighting dB	−16.1	−8.6	−3.2	0	1.2	1
A weighted band level dB	54.9	62.0	67.0	69.6	69.7	69.6

A weighted reverberant sound pressure level = 75.4 dB is found by adding the weighted octave band sound levels together using equation 2.12 or Figure 2.15.

Using the Norris-Eyring equation

To use the Norris-Eyring equation the mean absorption coefficient of all the absorbing surfaces has to be calculated which means that the effective surface area of the audience must be known. As this quantity is not easily defined use of this equation cannot be guaranteed to increase the accuracy of the estimates of reverberation time. However, to illustrate the use of this formula an absorbing area of 0.75 m^2 has been assigned to each occupied seat. Thus when 300 seats are occupied the total absorbing area is 225 m^2. The total area of absorbing surfaces in the hall is thus 1485 m^2.

Hence

	Octave band centre frequency (Hz) 125	250	500	1K	2K	4K
Total absorption excluding air absorption	317	343	371	429	538	495
Mean absorption coefficient $\bar{\alpha}$ = total absorption ÷ 1485	0.21	0.23	0.25	0.29	0.36	0.33
$-2.3\log_{10}(1-\bar{\alpha}) \times S$	350	388	427	508	662	594
Air absorption 4mV	1	2	6	11	20	21
$-2.3\log_{10}(1-\bar{\alpha}) \times S + 4mV$	351	390	433	519	682	615
Reverberation time $T = \dfrac{0.16V}{-2.3\log_{10}(1-\bar{\alpha}) \times S + 4mV}$	1.03	0.92	0.83	0.69	0.53	0.59

Example (cont.)

To calculate the A weighted reverberant sound pressure level

	Frequency (Hz) 125	250	500	1K	2K	4K
Sound power, W, watts	0.001	0.001	0.001	0.001	0.001	0.001
120 Log W	– 30	– 30	– 30	– 30	– 30	– 30
Total absorption, A, m^2	351	390	433	519	682	594
10 Log A	25.5	25.9	26.4	27.2	28.3	27.7
$L_R = 10\log W - 10\log A + 126\text{dB}$	70.5	70.1	69.6	68.8	67.7	68.3
A weighting dB	– 16.1	– 8.6	– 3.2	0	1.2	1
A weighted band level dB	54.4	61.5	66.4	68.8	68.9	69.3

The A weighted reverberant sound pressure level = 74.8 dB is found by adding the weighted octave band sound levels together using equation 2.12 or Figure 2.15. An alternative approach for calculating the absorption in a hall has been proposed by Kosten. If the seating/audience provides the majority of the absorption, the absorption, A, is obtained using the equation $A = S \times \alpha_{eq}$ where S is the area of audience including aisles up to 1 m wide and α_{eq} is an equivalent absorption coefficient. In unoccupied halls α_{eq} has a value of 0.81 while in occupied halls its value is taken to be 1.07.

Table 3.2 Absorption values

Frequency	Relative humidity % 20	30	40	50	60	70	80
125	0.06	0.05	0.04	0.04	0.03	0.03	0.02
250	0.14	0.13	0.12	0.11	0.1	0.09	0.08
500	0.25	0.25	0.26	0.26	0.26	0.25	0.25
1000	0.57	0.47	0.45	0.46	0.48	0.5	0.51
2000	1.78	1.21	1	0.9	0.88	0.88	0.88
4000	6.21	4.09	3.1	2.6	2.27	2.08	1.95
8000	19.0	14.29	11.0	8.95	7.61	6.69	6.04

Air absorption, 4 mV, in m^2 units for a volume of $100m^3$ at 20°C.

3.2.4 Sound in large enclosures

The use of the Sabine or Norris-Eyring equations in large enclosures with unusual aspect ratios, such as industrial halls, tends to give results which are in poor agreement with measurement. In these large halls there is not really a reverberant field, nor can the conditions be considered to approximate to those found in the free field. In practice, the sound field from a source lies somewhere between the two extremes with the sound level decreasing by something less than 6dB for each doubling of distance from the source.

The behaviour of sound in large irregularly shaped enclosures has been studied using scale models and mathematically analysed using the concept of image sources and ray tracing. However, a useful currently available method for determining the sound field in such enclosures is based on a survey of industrial halls undertaken by Rockwool AB of Sweden. The data collected on the survey has been analysed to produce empirical equations for estimating reverberation times and sound level decay rates from which sound level distributions can be obtained. To use the method the room and its contents must first be assigned to a particular category, as detailed below.

Classification of the room

Narrow rooms: defined as those in which the width, b, is less than four times the ceiling height, h. These rooms are denoted by the letter N.

Broad rooms: defined as those in which the width is greater than six times the ceiling height. They are denoted by the letter B.

For rooms with dimensions between four and six, interpolation has to be made when using Tables 3.3 and 3.4.

Classification of the room contents

The furniture – meaning machines, screens, product stock or anything which may absorb or scatter sound – is classified into three groups:

Low furnishing: furniture whose average height is less than one-eighth of the ceiling height. Included in this category are empty and sparsely furnished rooms. This category is denoted by the letter L.

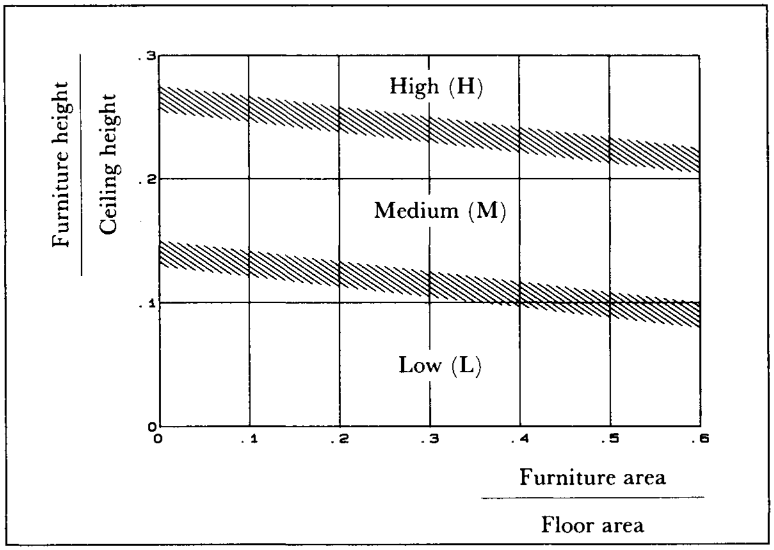
Medium furnishing: dense furnishing with average height between one eighth and one-quarter of the ceiling height, or sparser, higher furniture. This category is denoted by the letter M.

High furnishing: dense furnishing with an average height greater than a quarter of the ceiling height. This category is denoted by the letter H.

These categories are summarised in Figure 3.3 where the density of furniture is defined as furniture area to total floor area.

From the classification of the room and furniture size, six categories of room are obtained: BH, BM, BL, NH, NM and NL.

Figure 3.3 Classification of furniture height in connection with predicting sound levels in large enclosures.



Estimation of reverberation time

The reverberation time is estimated using the equation

$$T = 0.15h - 1.8\,\alpha + 1.8 + K_T$$

T = reverberation time at 1000Hz (sec)

h = ceiling height (m)

α = average value of random incidence absorption coefficient of the ceiling in the 500, 1000 and 2000Hz octave bands.

K_T is a room constant which is obtained from Table 3.3:

Table 3.3 Room constant K_T

Room/furniture classification	Ceiling absorption coefficient 0.1			1.0		
Ceiling height m	5	10	15	5	10	15
BH	−0.5	−0.3	0	0	0	0
BM	0	0	0	0	0	0
BL	0.5	0.3	0.3	0	0	0
NH	−0.5	−0.3	0	0	0.5	1.0
NM	0	0	0	0	0	0
NL	0	0.5	1.0	—	0.3	0.5

For other values of α and h interpolation has to be made.
At frequencies other than 1000Hz the reverberation time will generally be lower. At high frequencies the reduction is due to air absorption, while at low frequencies it is the absorption of the roof, especially if it is lightweight, that lowers the reverberation time.

Sound level reduction with distance
The reduction in sound level for each doubling of distance from a source is given by

$$\Delta L = a \times \alpha + b$$

where ΔL is the sound level reduction in dB(A)
α = random incidence absorption coefficient of the roof (average of values in 500, 1000 and 2000Hz octave bands)
a and b are room constants obtained from Table 3.4:

Table 3.4 Room constants a, b

Room furniture classification	Room constants a	b
BH	3	4
BM	2.5	3.75
BL	2	3.5
NH	3	3
NM	2.75	2.75
NL	2.5	2.5

Table 3.5 Room constant A

Source density	Value of constant A
Sparse (d = 10m)	5
Middle (d = 5m)	4
Dense (d = 2.5m)	3

d = the average distance between noise sources.

Estimation of sound level in an enclosure
At any point in the enclosure the sound level will be the sum of the sound levels from all the sources operating in the enclosure. To make this calculation either the sound power of each source will be required or the sound pressure level at a known distance from each source. This information combined with the estimated decay rate will allow the sound levels to be derived for any location in the room.
However, if the effect of altering the enclosures acoustics (for example by introducing an absorbing ceiling) is required, the following empirical estimates can be used.
The reduction in the A weighted sound level at any location L_R is given by

$$L_R = A + \Delta - 1$$

where $\Delta = \Delta L_a - \Delta L_b$
and ΔL_a = decay rate after treatment
ΔL_b = decay rate before treatment
and A = a constant given in Table 3.5 (above).

Example

An industrial hall 7.5 m high × 32 m wide × 55 m long has 20 per cent of its floor area covered by machines of average height 3 m which are, on average, a distance of 7 m apart. The roof is constructed from profiled steel decking. It is required to check the effect of hanging sound absorbers to the roof underside.

$$\frac{\text{Room width}}{\text{Room height}} = \frac{32}{7.5} = 4.3$$

Thus the room falls between narrow (N) and broad (B).

$$\frac{\text{Furniture area}}{\text{Floor area}} = 0.2$$

$$\frac{\text{Furniture height}}{\text{Ceiling height}} = \frac{3}{7.5} = 0.4$$

Thus, from Figure 3.3, the furniture is classified as high (H).

The reverberation time at 1000Hz is calculated from

$$T = 0.15h - 1.8\alpha + 1.8 + K_T.$$

For a profile steel roof α may be taken as 0.1 at 1000Hz and so from Table 3.5 K_T is interpolated as -0.4.

$$\therefore \quad T = 0.15 \times 7.5 - 1.8 \times 0.1 + 1.8 - 0.4 \text{ sec}$$
$$T = 2.3 \text{ sec}$$

When fitted with hanging sound absorbers the absorption coefficient of the roof is about 0.9, and from Table 3.5 the value of K_T is interpolated to be 0.2.

$$\therefore \quad T = 0.15 + 7.5 - 1.8 \times 0.9 + 1.8 - 0.2 \text{ sec}$$
$$T = 1.1 \text{ sec.}$$

Thus, introducing the absorptive roof has reduced the reverberation time at 1000Hz from 2.3 to 1.1 sec.
The sound level reduction for each doubling of distance from a source ΔL, is given by

$$\Delta L_b = a \times \alpha + b \quad \text{dB}$$

For the steel roof $\alpha = 0.1$ and from Table 3.4 $a = 3$ and $b = 3.1$

So that $\Delta L_b = 3 \times 0.1 + 3.1$
$= 3.4 \quad \text{dB}$

With the absorptive roof $\alpha = 0.9$

So that $\Delta L_a = 3 \times 0.9 + 3.1$
$= 5.8$ dB

The introduction of the absorbing roof has increased the noise level decay from 3.4 to 5.8dB per doubling of distance from each machine.
The approximate reduction in dB(A) at any location in the hall is given by

$$L_R = A \times \Delta - 1 \text{ dB(A)}$$

where $\Delta = \Delta L_a - \Delta L_b$ and

from Table 3.5 A is interpolated to be 4.5

$$\therefore \quad L_R = 4.5 \times 2.4 - 1$$
$$L_R = 9.8\text{dB(A)}$$

This estimate of L_R assumes the machines are uniformly distributed.

3.3 Sound Insulation

3.3.1 Sound reduction index R

Definition
Air-borne sound insulation involves separating by a physical barrier the space to be protected from the space containing the noise source. Sound waves in the air on the source side impinge on the partition, causing it to vibrate and so radiate sound into the receiving space. The proportion of the incident sound intensity which is transmitted through the partition by this mechanism is known as the transmission coefficient τ and the sound reduction index (SRI),R, is defined as

$$R = 10 \log \left[\frac{1}{\tau}\right] \text{ dB}$$

i.e. $$R = 10 \log \left[\frac{\text{intensity incident upon partition}}{\text{intensity transmitted by partition}}\right]$$

So that if $\tau = 0.01$ (1 per cent of the incident sound intensity is transmitted) the sound reduction index is 20dB; while if $\tau = 0.001$ (0.1 per cent of the incident sound intensity is transmitted) the sound reduction index is 30dB, and so on.
A sound reduction index of 20dB would in fact be considered quite modest, while one of 50dB is typical of that provided between dwellings. The value of R will vary with frequency and it is usual to measure the performance of a partition of each of the sixteen one-third octave frequencies ranging from 100 to 3150Hz. Although this does

not cover the whole audible frequency range it is sufficient for specifying the performance of a partition for most purposes. Some useful performance figures of various constructions are given in Table 3.6.

Table 3.6 Sound reduction indices for elements of structure dB

Material or construction	Average	Octave band centre frequency (Hz) 125	250	500	1000	2000	4000
Walls and partitions							
Single leaf fairfaced brick 102 mm	45	36	37	40	46	54	56
Single leaf brickwork plastered on both sides 13/102/13	47	34	36	41	51	58	60
Double leaf brickwork plastered on both sides 13/204/13	51	41	45	48	56	58	60
Cavity brickwork with ties 102/50/102 plastered on both sides	52	34	34	40	56	73	76
Fairfaced 115 mm lightweight concrete blockwork	38	32	32	33	41	49	57
Fairfaced 115 mm light concrete blockwork + 13 mm plaster on both sides	41	32	34	37	45	52	57
Fairfaced 115 mm light concrete blockwork + 12.7 mm plasterboard on plaster dabs on both sides	44	28	34	45	53	55	52
Fairfaced 215 mm light concrete blockwork	44	35	38	43	49	54	58
Fairfaced 215 mm light concrete blockwork + 13 mm plaster on both sides	47	37	39	46	53	57	61
Fairfaced 215 mm light concrete blockwork + 12.7 mm plasterboard on dabs on both sides	47	33	39	50	55	56	50

Table 3.6 (cont.)

Material or construction	Average	Octave band centre frequency (Hz) 125	250	500	1000	2000	4000
Double wall of two 100 mm dense concrete blocks with 50 mm cavity + 13 mm plaster on both sides	52	35	41	49	58	67	75
Double partition of two 12.5 mm plasterboard skins with 50 mm cavity completely filled with glass fibre quilt	40	21	35	45	47	47	43
Double partition of two 12.5 mm plasterboard skins with 75 mm cavity – glass fibre blanket suspended within	39	24	37	44	42	43	44
Double partition of two 12.5 mm plasterboard sheets on each side of 50 mm cavity completely filled with glass fibre quilt	47	33	45	51	52	52	52
Double partition of two 12.5 mm plasterboard skins on 75 × 50 mm staggered timber studding	37	24	28	37	46	46	38
50 mm woodwool cement slabs sealed on one side only	30	26	28	30	32	33	36
Sealed on both sides	33	25	31	36	35	35	37
100 mm woodwool cement slabs sealed on one side only	31	28	28	32	34	33	38
Sealed on both sides	35	29	30	32	36	39	46
50 mm woodwool cement slabs with 100 mm concrete on one face	42	36	36	42	44	46	53
50 mm standard woodwool cement/200 mm airspace/ 2 × 9.7 mm plasterboard attached to legs of units	51	34	44	51	57	61	60

Table 3.6 (cont.)

Material or construction	Average	Octave band centre frequency (Hz) 125	250	500	1000	2000	4000
150 mm concrete supporting on resilient mounts 100 mm prescreeded woodwool cement slabs – large airspace	71	60	66	77	83	not measurable	
38 mm × 22 SWG fluted steel cladding sheet/60 mm glass fibre/9.5 mm plasterboard. Mounted on 50 × 50 mm timber spacers	39	18	30	41	46	49	50
38 mm × 22 SWG fluted steel cladding sheet/146 mm space containing 60 mm glass fibre/ 2 × 12.7 mm plasterboard. Mounted on 146 mm steel studs at 600 mm centres	47	32	41	47	49	53	58
Glazing							
Single – non-openable							
4 mm	28	20	22	28	34	34	29
6 mm	29	18	26	31	36	30	38
6.4 mm laminated	30	22	24	30	36	33	38
12 mm	34	28	31	35	34	39	37
4 mm glass in aluminium frame 100 mm opening	11	10	10	11	12	12	13
Sealed units – non-openable							
4/12/4 mm	29	22	17	24	38	42	38
6/12/6 mm	30	20	20	29	30	36	46
4/12/10 mm	34	25	22	33	41	44	44
4/12/10 mm + SP6	36	22	19	43	47	47	47
6/12/10 mm	34	26	27	35	41	39	47
6/150/4 mm	44	29	35	45	56	52	51
Sealed units – openable							
3/6/3 mm weather stripped	26	25	22	25	28	27	31

Table 3.6 (cont.)

Material or construction	Average	125	250	500	1000	2000	4000
		Octave band centre frequency (Hz)					
Dual units – non-openable							
4/200/4 mm with absorbent reveals and separate heavy frames	42	37	37	44	53	47	36
6/200/6 mm with absorbent reveals in separate heavy frames	46	37	41	48	54	47	47
Dual units – openable							
4/200/4 mm with absorbent reveals in aluminium frames	39	27	33	39	42	46	44
4/200/4 mm with absorbent reveals in aluminium frames – opposite ends open 25 mm	27	15	23	34	32	28	32
opposite ends open 100 mm	22	10	16	27	25	27	27
Sheet materials							
3 mm sheet lead	34	30	31	27	38	44	33
6 mm steel plate	38	27	35	41	39	39	46
20 g profiled steel sheet	18	8	14	29	26	32	36
Floors							
T & G Boards (or chipboard) to joists, plasterboard and skim soffit	35	18	29	37	49	44	46
As above with fibreglass under chipboard, 50 mm sound pugging on meshbacked plasterboard	51	37	42	47	52	60	64
50 mm screed on 125 mm reinforced concrete	47	35	37	42	49	58	63
As above with 13 mm fibreglass under screed	51	38	43	48	54	60	64
50 mm screed on 200 mm reinforced concrete	51	38	45	47	52	60	64

Table 3.6 (cont.)

Material or construction	Average	125	250	500	1000	2000	4000
		Octave band centre frequency (Hz)					
Doors							
Flush hollow doors, normal edge gaps	16	12	13	14	16	18	24
Solid door, normal edge cracks	26	17	21	26	29	31	34
Acoustic metal doorset, double seals	47	36	39	44	49	54	57
Folding steel door	25	16	23	26	27	28	27

These are typical values measured in the laboratory. Field tests may give lower values.

Effect of frequency

In general, sound reduction index values are greater for higher frequencies although some irregularities often occur, a partition perhaps exhibiting a weakness at one or two particular frequencies. Such weaknesses are effectively hidden by quoting the average value of the sound reduction index which is defined as the arithmetical average of the values measured in the sixteen different third-octave bands. Alternatively, the information may be combined to give the values in octave bands. If the sound reduction indices in the three third-octave bands are R_1, R_2 and R_3 respectively, the octave band value is given by

$$R_{octave} = 10 \log \left[\frac{3}{10^{-R_1/10} + 10^{-R_2/10} + 10^{-R_3/10}} \right]$$

$$R_{octave} = 10 \log 3 - 10 \log [10^{-R_1/10} + 10^{-R_2/10} + 10^{-R_3/10}]$$

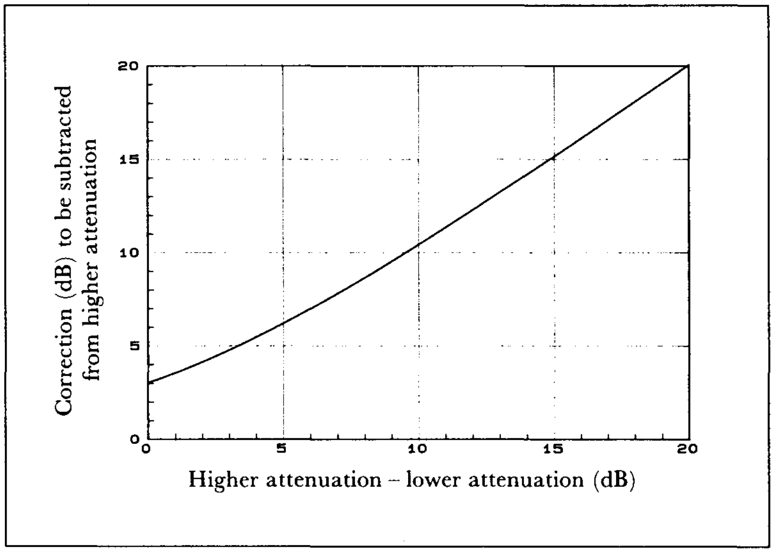
$$R_{octave} = 5 + C$$

To evaluate C we can use the chart in Figure 3.4 which enables two attenuations to be combined. To find the value of the sound reduction index in the octave the one-third octave values are combined in pairs, as shown below.

Example

If the values of the SRIs in the 800, 1000 and 1250Hz one-third octave bands are 25, 28 and 30dB, the 1000Hz octave band value is obtained in the following way:
1. Combine $R_2 = 28$dB and $R_1 = 25$dB. As $(R_2 - R_1) = 3$dB correction from graph is -4.8dB. Hence combined value is $28 - 4.8 = 23.2$dB.
2. Combine $R_2 = 30$dB and $R_1 = 23.2$dB. As $R_2 - R_1 = 6.8$dB correction from graph is -7.6. Hence combined value is $30 - 7.6 = 22.4$dB.
3. The value of the sound reduction index in the 1000Hz octave band is $5 + 22.4 = 27.4$dB.

Figure 3.4 Graph to be used in the combination of two attenuations.



Air-borne sound insulation index rating, R_w

This is a weighted single figure descriptor of a panel's performance obtained by comparing the sound reduction indices measured in the third-octave bands from 100 to 3150Hz with a set of standard curves. The standard curves are drawn and listed in BS 5821. The value of R_w for a given panel is that value for which the deviations of the measured values from the standard curve are as close to -32dB as possible.

The R_w rating is in some ways similar to the A weighting, giving more significance to mid and high frequencies than to low frequencies. It is thus a better unit with which to compare the performance of panels than the arithmetic average of the sound reduction indices.

In pre-1982 literature reference may be found to the air-borne sound insulation index I_a which was the former name of the sound insulation index rating R_w.

Sound transmission class, STC

This is the American equivalent of the R_w rating, the main differences being that the frequency range covered is 125–4000Hz and that the sound reduction index curve must not fall more than 8dB below the STC contour at any one frequency. The STC and R_w ratings of practical partitions will be very similar.

3.3.2 Sound level difference between two spaces

The sound reduction index is an intrinsic property of a partition. The sound level difference between two spaces separated by the partition depends upon the value of the sound reduction index, the area of the partition and the acoustic properties of the two spaces concerned. We shall deal with three specific cases.

Two reverberant spaces – room to room

Because the source room is reverberant the sound field is likely to be uniform and the sound energy will travel in all directions. However, we need to know the sound intensity incident upon the partition. It can be shown that this intensity is equivalent to an incident sound level 6dB less than the reverberant level, L_1, in the source room. The incident sound level, L_i, is therefore

$$L_i = L_1 - 6 \quad \text{dB}$$

If the sound reduction index of the partition is R, the transmitted sound level is given by

$$L_t = L_i - 6 - R\text{dB}$$

This is the sound level equivalent of the sound intensity being transmitted through the partition and relates to unit area. The total power transmitted by the partition will depend on its area, i.e.,

$$SWL_2 = L_1 - 6 - R + 10\log S \quad \text{dB}$$

where SWL_2 is the sound power level equivalent of the power passing through the partition of area S.

As the receiving room is reverberant the average sound level, L_2, in this room is related to the acoustic power entering the room, SWL_2, and the absorption, A, in the room, as given in equation 3.1.

So that $\quad L_2 = SWL_2 - 10\log A + 6 \quad \text{dB}$

or $\quad L_2 = L_1 - 6 - R + 10\log S - 10\log A + 6 \quad \text{dB}$

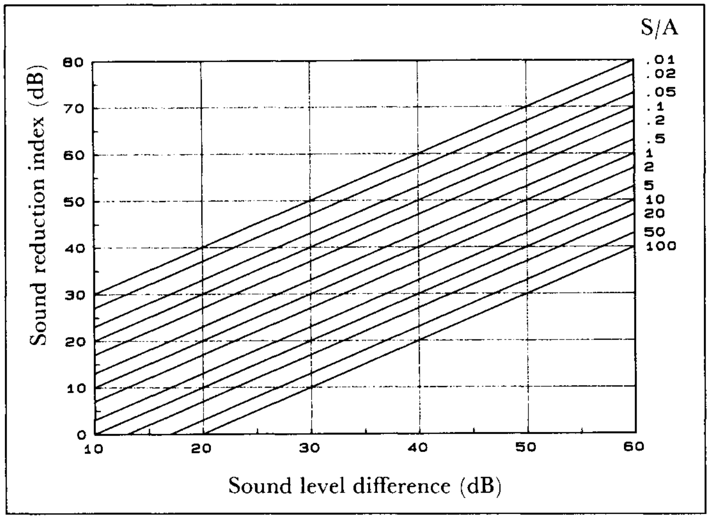
or $\quad L_2 = L_1 - R + 10\log\frac{S}{A} \quad \text{dB}$

The sound level difference $D = L_1 - L_2$ is

$$D = R - 10\log\frac{S}{A} \quad \text{dB}$$

This relationship, given in graphical form in Figure 3.5, is most useful for calculating sound levels in adjacent rooms.

Figure 3.5 Graph giving the sound level difference between two rooms.



Example

Two rooms are separated by a partition of area 10 m^2 with sound reduction index 28dB. A sound source in one room gives an average sound level of 80 dB. If the second room has a volume of 50 m^3 and a reverberation time of 0.5 sec what will be the sound level in this room?

Use Sabine's equation to find total absorption in second room.

$$T = \frac{0.16V}{A} \text{ sec}$$

$$\therefore \quad A = \frac{0.16 \times 50}{0.5} \text{ m}^2$$

$$A = 160 \text{ m}^2$$

Area of partition $S = 10$ m^2

$$\therefore \quad \frac{S}{A} = \frac{10}{160} = 0.0625$$

As the sound reduction index is 28dB, Figure 3.5 gives a level difference $D = 40$dB

$$\therefore \quad L_2 = L_1 - D = 80 - 40 = 40\text{dB}$$

If the equation for the level difference is rearranged, we see that

$$R = D + 10 \log \frac{S}{A}$$

which indicates how the sound reduction index can be obtained from a measure of the level difference between two rooms. This is the normal procedure employed under controlled laboratory conditions. When field measurements are made it is the sound level difference that is given as a measure of the performance of the partition.

Standardised level difference

As the sound level difference will depend upon the absorption in the receiving room it is recommended (BS 5821 : 1984) that the measured level difference be corrected to give the standardised level difference, D_{nT}, which is defined as

$$D_{nT} = D + 10 \log \frac{T}{0.5} \text{ dB}$$

where T is the reverberation time in the receiving room.
The level difference, D, and the standardised level difference D_{nT} will normally be measured in each of the third-octave bands from 100 to 3150Hz. Single figure values can be obtained by weighting the third-octave values using the same procedure for obtaining the weighted sound reduction index R_w.

The weighted level difference

The weighted level difference is denoted by D_w while the weighted standardised level difference is denoted by $D_{nT,w}$. They are defined in BS 5821 : Part 1 : 1984 and referred to in the Building Regulations, Part E, Approved Document (section 3.3).

Reverberation space to anechoic space – inside to outside

As the source room is reverberant the sound power transmitted through the partition will be the same as in the room-to-room case so that the total acoustic power transmitted by the partition is

$$SWL_2 = L_1 - 6 - R + 10 \log S \text{ dB}$$

This can be used in conjunction with equation 2.18 for calculating the level at some distance from the partition

i.e., $L_2 = SWL_2 - 11 - 20 \log r + DI$
$L_2 = L_1 - 6 - R + 10 \log S - 11 - 20 \log r + DI$

In this equation it is assumed that r is measured along a line normal to the plane of the

partition and that it is large compared to the dimensions of the partition. If this is so, the directivity index can be obtained by deciding into what fraction of a spherical surface the sound is radiating. For example, if the partition is one façade of a building, it can be assumed that the sound is radiating into a quarter sphere and so the directivity index will be +6dB. Little, if any, usable information exists about the directivity indices of walls or roofs at angles to the normal. Hence, if the equation is used to calculate sound levels at positions not on the normal to the radiating area the results must be considered to be approximate.
For very large façades or positions close to façades it is sometimes best to divide the area into a number of smaller areas and apply the above equation to each, and then calculate the resultant sound level by adding the sound levels from each area as though they came from incoherent sources.

Anechoic space to reverberant space – outside to inside
In this case we are concerned with estimating how much external sound energy enters a building. The presence of the partition, which is likely to be part of the building façade, reflects the sound field and causes a local increase in the external level. The intensity incident upon the façade is equivalent to a sound level L_i where

$$L_i = L_l - K_l \text{ dB}$$

where L_l is the level measured external to the façade
and $K_l = 6$ if the measuring microphone is very close to the partition
$K_l = 2.5$ if the measuring microphone is 1 m from the façade
$K_l = 0$ if the measuring microphone is far from the façade or if the external level is calculated without taking a façade reflection into account.

If the sound reduction index of the façade is RdB, the transmitted level, L_t, is

$$L_t = L_l - K_l - R \text{ dB}$$

and the transmitted power is given by

$$SWL_2 = L_l - K_l - R + 10 \log S \text{ dB}$$

As before, the sound level, L_2, in the receiving room is obtained from

$$L_2 = SWL_2 - 10 \log A + 6 \text{ dB}$$
$$L_2 = L_l - K_l - R + 10 \log S - 10 \log A + 6 \text{ dB}$$

Hence the level difference $D = L_1 - L_2$ is given by

$$D = R + K_1 - 10 \log \frac{S}{A} - 6 \text{ dB}$$

which can be evaluated using Figure 3.5 if $R + K_1 - 6$ is used instead of R.

3.3.3 Single skin partitions

Mass law

These are constructions without an air space, filled or otherwise, in the middle. They need not be made from a homogeneous material but must vibrate as an entity. The sound reduction index of this type of partition is determined largely by its superficial mass per unit area. For sound falling at an angle θ on to a partition of surface mass, M, theory predicts that the sound reduction index will be

$$R_\theta = 10 \log \left[1 + \left(\frac{\pi f M \cos\theta}{\rho c} \right)^2 \right]$$

For $\theta = 0°$ $R_o = 10 \log \left[1 + \left(\frac{\pi f M}{\rho c} \right)^2 \right]$

which can be written for most frequencies with little error as

$$R_0 = 20 \log \left[\frac{\pi f M}{\rho c} \right]$$

In practice sound will fall on the partition over a wide range of incident angles and the field sound reduction index is taken as the average of the sound reduction indices for angles of incidence from 0 to 78° and it is found that

$$R_{\text{field}} \approx R_0 - 5\text{dB}$$

Theoretical average v field, R

The average theoretical field sound reduction over the frequency range 100–3150Hz is the same as the value calculated at 561Hz.

$$\therefore \quad \text{average } R_{\text{field}} = 20 \log \left[\frac{\pi 561 M}{\rho c} \right] - 5 \text{ dB}$$

or average $R_{\text{field}} = 7.6 + 20 \log M$ dB where M is in kgm^{-2}

In theory the average sound reduction index will increase by 6dB/doubling of surface mass. However, in practice the increase is not as great as this and the empirical dependence derived from a large number of tests on different single skin partitions is shown in Figure 3.6. It should not be assumed that the performance of a particular partition will fall on this line – it might be better or it might be worse, because the line is the best fit straight line to the available data. Nevertheless it is a guide, especially if it is borne in mind that a high density material such as steel is likely to be superior while a low density material such as aerated concrete is likely to be inferior.

If we refer back to the equation for the normal sound reduction index we see that it, and hence the field value, will increase by 6dB for each doubling of frequency. In practice this is true over part of the frequency range but the performance will drop below that predicted by the simple equation because of two effects – coincidence and partition resonances.

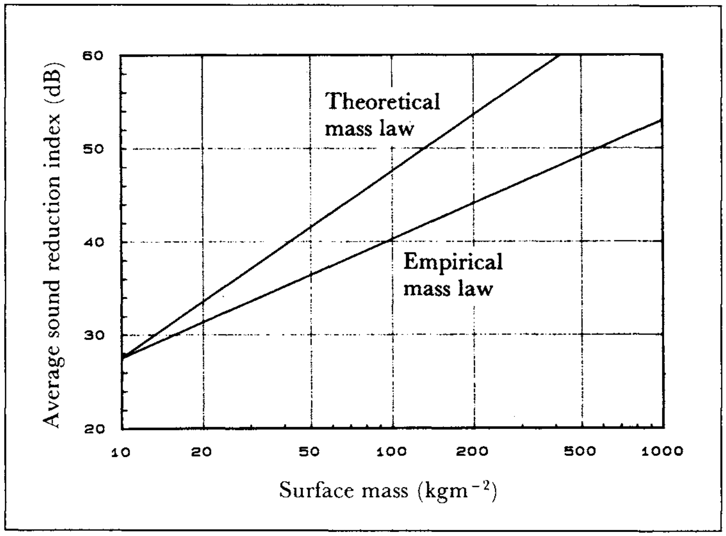


Figure 3.6 Theoretical and empirical variation of the average sound reduction index of a single leaf partition with surface mass.

Coincidence

The sound waves falling on to the panel excite bending waves in it, the velocity of which depends upon frequency. As the frequency increases the bending wave velocity increases and at some frequency, known as the critical frequency, f_c, it is equal to the velocity of sound in air. As the frequency of the air wave and bending wave is equal, the wavelengths are also equal and conditions are correct for resonant excitation of the panel. This leads to increased sound transmission and thus a reduced value of R. Matching at the critical frequency occurs for sound striking the panel at grazing incidence and little energy is actually transferred to the panel. However, as the frequency increases there will be an angle of incidence for which matching occurs at each frequency, and above the critical frequency the sound reduction index falls below that predicted by the mass law. As much higher frequencies R again approaches the value predicted by the mass law. This type of behaviour is indicated in Figure 3.7.

Figure 3.7 Idealised variation of the sound insulation of a single leaf partition with frequency.

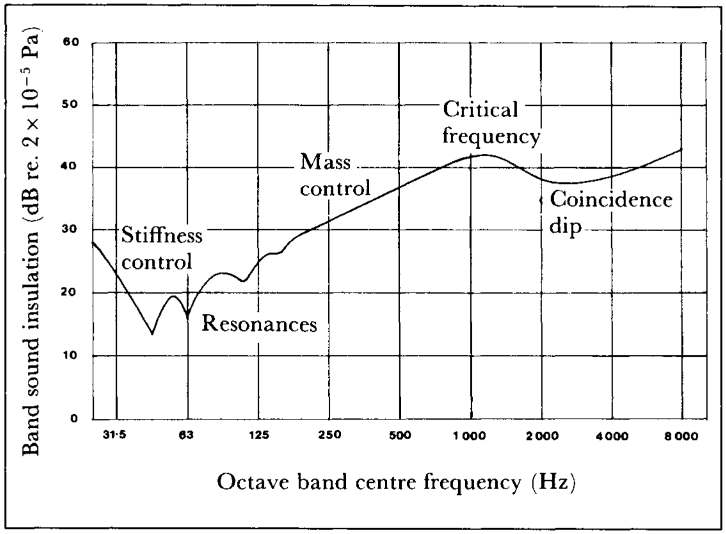
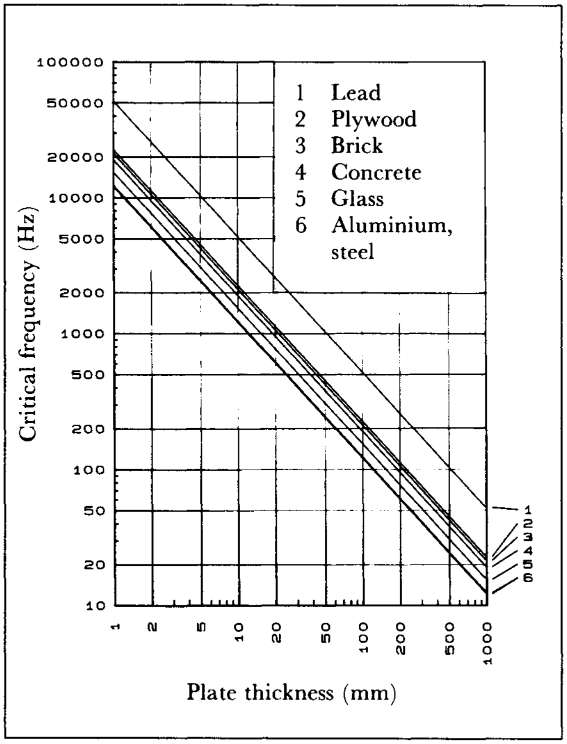


Figure 3.8 Variation in critical frequency of some common materials with thickness.



Coincidence frequency

The coincidence frequency depends upon the type of material and also on the thickness of the panel. If the thickness is doubled the critical frequency is halved so that although increasing the thickness of a material increases its surface mass it also reduces the critical frequency. Thus one cannot always expect the gain in the sound reduction index that the increase in mass would indicate. Figure 3.8 shows how the critical frequency varies with thickness for some common materials.

In general with a partition the idea is to push the critical frequency above the frequency range of interest. However, this is not always possible. Take, for example, a 110 mm brick wall. This has a coincidence frequency of about 200Hz which means that over the usual frequency range of interest its performance will be below that predicted by the theoretical mass law. Its surface mass of 185 kgm^{-2}, however, provides sufficient mass to compensate to some extent for the coincidence effect. The empirical mass law equation in Figure 3.6 will allow the influence of the coincidence effect.

Partition resonances

Because of the finite size of the partition, standing waves will occur in it and at the frequencies that this occurs the resonances will reduce its performance. The frequency of the resonances depends upon the size and the way the partition is fixed at the edges. For most partitions the resonances occur at low frequencies outside the range of usual interest and except in special cases can be ignored. The fundamental resonant frequency, which is the most important, can be calculated from

$$f_{11} = \frac{\pi t}{2} \sqrt{\frac{E}{12\rho}} \left[\frac{1}{a^2} + \frac{1}{b^2}\right]$$

where a and b are the partition dimensions (m), t is its thickness (m) and E and ρ are its Young's Modulus (Pa) and density (kgm^{-3}). In this case we see that the resonance frequency increases with increasing thickness which again reduces the frequency range at which the mass law will apply. Values of E and ρ for a few common materials are given in Table 3.7.

Table 3.7 Values of E and ρ

Material	Young's Modulus (Pa) E	Density ρ (kgm^{-3})
Steel	2×10^{11}	8000
Glass	4×10^{10}	2500
Aluminium	7×10^{10}	2700
Lead	1.6×10^{10}	11300
Brick	1.6×10^{10}	1900
Concrete	2.4×10^{10}	2300
Plywood	4.3×10^{9}	580
Plasterboard	1.9×10^{9}	750
Chipboard	3.4×10^{9}	550

3.3.4 Double skin partition

This means a construction containing a cavity or air space between two separate leaves. In practice the leaves are unlikely to be totally separated; a cavity brick wall, for example, has wall ties, while stud partitions often have two leaves fixed to common studs. For maximum insulation, however, the ties should be kept to a minimum and any connections should, if possible, be made flexible rather than rigid.
Double partitions are used either when very high reduction is required or when moderately high reduction (around 40dB average) is required without incurring the weight penalty associated with a single skin. The sound reduction index of the double skin construction is not, unfortunately, the arithmetic sum of the reductions provided by the individual leaves because the separation is not sufficiently great for them to be uncoupled. Nevertheless, provided a few simple rules are observed it is possible to get a higher SRI from a double skin construction than from an equivalent weight single skin.

Measures for high SRI

1. Make the space between the two leaves as large as possible. The two leaves and the air space resonate at a frequency f_r given by

$$f_r = 60\sqrt{\frac{m_1 + m_2}{m_1 m_2 d}} \quad \text{Hz}$$

   where m_1 and m_2 are the mass per unit area of the leaves (kgm^{-2}) and d is the separation of the leaves in metres
   If $m_1 = m_2$ then

$$f_r = \frac{120}{\sqrt{md}} \quad \text{Hz}$$

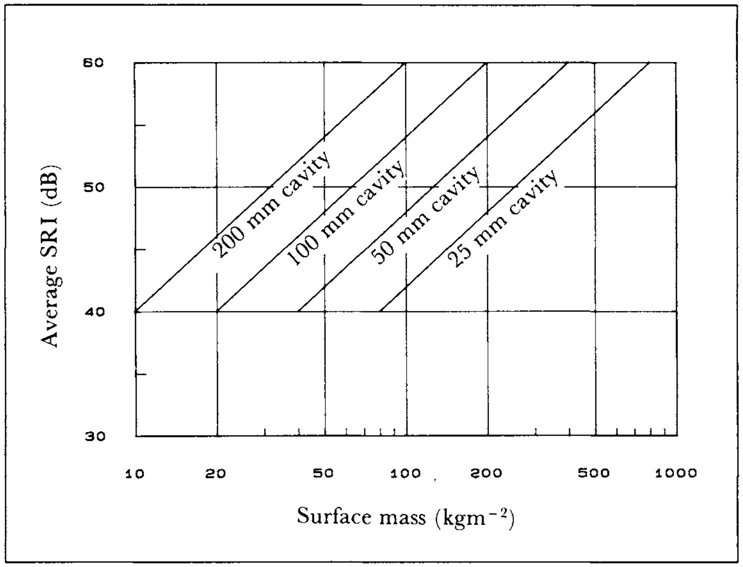
   where m is the total mass per unit area in kgm^{-2} of both leaves.
   Although the resonance will always be present, the resonant frequency can be arranged to fall below the frequency range of interest, usually below 50Hz, by choosing a sufficiently large value for d in relation to m.
2. Keep the number of physical ties between the leaves to a minimum. For maximum insulation there should be no ties and each leaf should be isolated at the edges from floors, walls or ceilings which could link the leaves.
3. Introduce fibrous sound absorbent material into the cavity between the leaves. This prevents the build up of the sound in the cavity and helps improve the insulation especially in the lower frequency range 200–800Hz. The material should not fill the cavity. Lining the reveals is often sufficient. The improvement gained in insulation is best for partitions of low total weight.

There is little useful information available to help the prediction of the performance of double skin partitions. An empirical equation based upon data gathered using woodwool cement slabs suggests that an estimate of the average field sound reduction index can be obtained using the equation.

$$R = 34 + 20 \log md$$

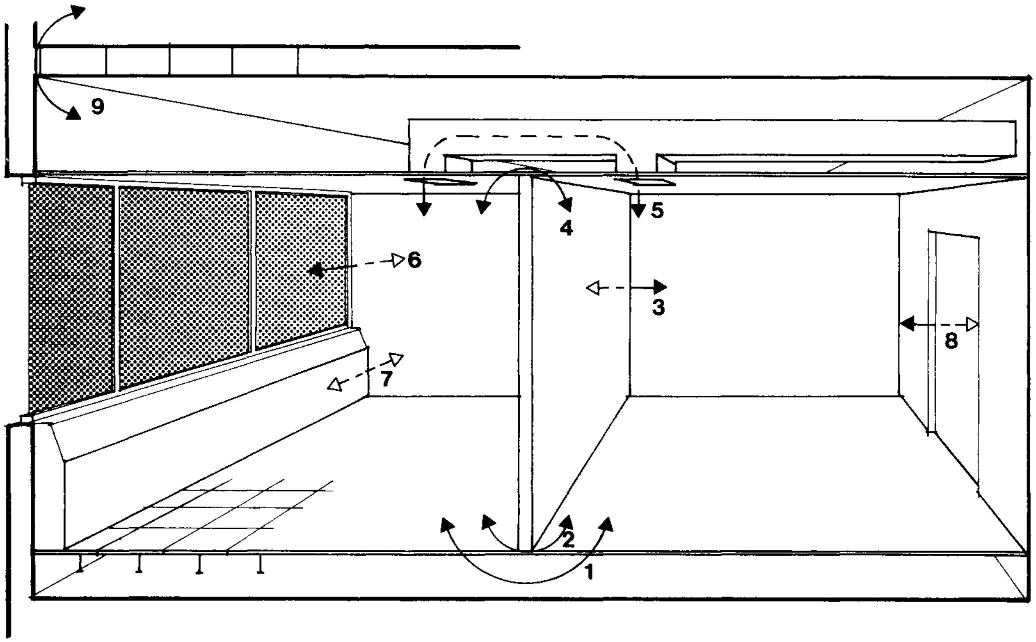
where m is the total mass of both leaves in kgm^{-2} and d is the separation of the leaves in metres. This relationship is shown in Figure 3.9 from which it is clear that a minimum value of about 40dB is assumed.
In practice, when considering the use of double skin partitions it is suggested that the performance of a system should be obtained from laboratory tests.

Figure 3.9 Variation of average sound reduction index of double leaf partition with total surface mass. (Based on measurements using woodwool slabs.)



Flanking transmission
The transmission of sound energy via paths which bypass the partition is known as flanking transmission. It can occur because the sealing of the panel around its perimeter is inadequate or small gaps have been left where services pass through the partition or because energy which has got into the main structure travels through the structure to be reradiated on the other side of the partition. Flanking transmission means that the partition fails to give the performance that laboratory measurements would indicate it is capable of. In practice, flanking transmission is always a possibility and every effort should be made to guard against the most obvious means by which energy can bypass the partition. Flanking routes are illustrated in Figure 3.10.

Figure 3.10 Some common noise flanking paths.



1 Floor void
2 Partition base
3 Partition edge
4 Ceiling void
5 Common ductwork
6 Curtain wall mullions
7 Continuous convector units
8 Door edges
9 Floor edge

3.3.5 Composite construction

A composite construction is one having areas with different sound reduction indices, for example, a brick wall containing a door and window. The total sound power transmitted by such a structure can be obtained by calculating the power transmitted by each component separately and then adding them together. Alternatively, the effective sound reduction index of the partition can be derived and used to estimate the total power transmitted.

For a partition consisting of areas S_1, S_2, S_3, etc, with sound reduction indices R_1, R_2, R_3, etc, the value of the sound reduction index for the composite partition is obtained using the equation

$$R_{comp} = 10 \log_{10}\left[\frac{S_1 + S_2 + S_3 + \ldots}{10^{-R_1/10} \times S_1 + 10^{-R_2/10} \times S_2 + 10^{-R_3/10} \times S_3 \ldots}\right]$$

Alternatively, the value of R_{comp} can be found using Figure 3.11. If the partition contains more than two areas with different R values the calculation must be performed taking two areas at a time.

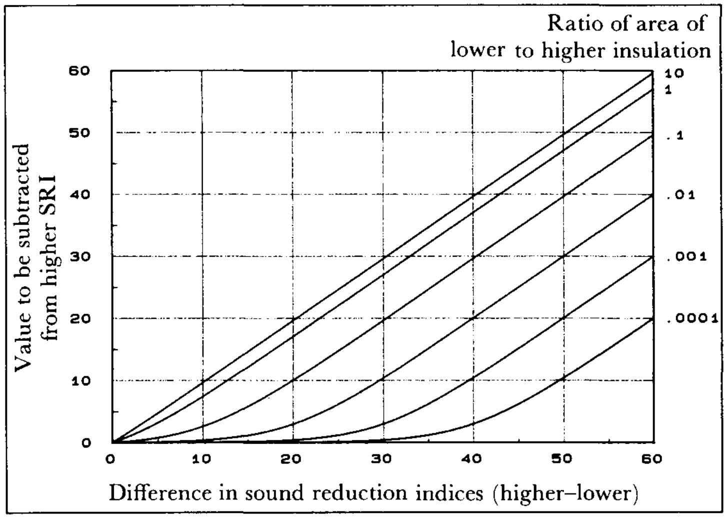


Figure 3.11 Graph showing correction to be made to the sound reduction index of a partition due to the presence of an area of lower insulation.

Example

A 275 mm brick wall with an average sound reduction index of 49dB and area 10 m^2 contains a window of area 2 m^2 and a door of area 2.5 m^2. If the sound reduction indices of the window and door are 28 and 26dB respectively, what is the sound reduction of the façade?

$$\frac{\text{area of window}}{\text{area of brickwork}} = \frac{2}{5.5} = 1:2.25$$

$$R_{\text{brickwork}} - R_{\text{window}} = 49 - 28 = 21\text{dB}$$

Hence from Figure 3.11 loss of insulation = 15.5dB

$$\therefore \quad R_{\text{brickwork/window}} = 49–15.5 = 33.5 \text{ dB}$$

$$\frac{\text{area of door}}{\text{area brickwork/window}} = \frac{2}{7.5} = 1:3.75$$

$$R_{\text{brickwork/window}} - R_{\text{door}} = 33.5 - 26 = 7.5\text{dB}$$

Hence from Figure 3.10 loss of insulation = 3dB

$$\therefore \quad R_{\text{façade}} = 33.5 - 3 = 30.5\text{dB}$$

Another example shows the disastrous effect small leaks can have on the performance of a partition. The effect is usually more marked at high frequencies because the partition's sound reduction index is higher at these frequencies. The performance of an acoustic door is to a large degree governed by its seals. Small gaps between door and frame will seriously reduce the effectiveness of the door. Similarly, enclosures constructed from pre-fabricated units need to be effectively sealed at joints.

Example

A 275 mm brick wall of area 10 m^2 with average sound reduction index 49dB has a small hole of area 0.01 m^2 in it. What is the resultant sound reduction index?

$$\text{Area of wall} = 10 - 0.01 = 9.99\ m^2$$

$$\therefore \quad \frac{\text{area of hole}}{\text{area of wall}} = \frac{0.01}{9.99} \simeq 1:1000$$

$$R_{\text{brickwork}} - R_{\text{hole}} = 49 - 0 = 49\text{dB}$$

Hence from Figure 3.11 loss of insulation = 20dB

$$\therefore \quad R_{\text{wall with hole}} = 49 - 20 = 29\text{dB}$$

Noise control

4.1 Prediction and Prescription

4.1.1 Noise sources

An essential part of noise control is the assessment of the problems that might occur from given noise sources. It may be that the sources are external to a building and the noise penetration into the building has to be controlled, or the source may be within the building and the noise break out must be controlled. At other times the internal environment from sources within a building will be the concern. In all cases a knowledge of the sound field will be required and for detailed design it may be essential to measure the specific noise. However, it is often helpful in the planning stage to acquire a feel for the magnitude of the problem; in this section noise levels of some typical sources are presented.

It is far better to ensure that there is no problem at an early stage of a programme than to discover a problem at a later stage. If a problem is identified at an early stage, it can be designed out or control measures can be taken. Dependence on a retrofit can be embarrassing and give a less than ideal solution.

Road traffic

The commonest external noise is that of road traffic. It is generally accepted that it is the sound level exceeded for 10 per cent of the time, i.e., the L_{10} level, that correlates best with dissatisfaction caused by traffic noise. Values of L_{10} can be estimated and procedures for making the calculation exist in a number of countries. In the UK the method is given in the Department of the Environment publication *Calculation of Road Traffic Noise.* The calculations usually only apply to freely moving traffic and require detail of both traffic and site layout. Indicative noise levels to be expected from different types of road are given in Table 4.1.

The level of noise, while depending on traffic volume, composition and speed, will also be influenced by the road gradient and surface finish. An increase of around 3dB(A) will be caused by a 15 per cent gradient and increases of up to 4dB(A) have been observed with 15 mm deep textured ribbing compared with an asphalt surface. While an increase in heavy vehicle content does cause an increase in the overall dB(A) level, the important part to bear in mind is that the largest increase in noise levels occurs at low frequencies where façade insulation is usually poorest.

Table 4.1 Typical road noise levels

Type of road	18 hour L_{10} level dB(A) at a façade 10 m from nearside lane
Motorway ~ 40,000 vehicles/18 hour day	80
Class A dual/3 lane, 64 km/hour speed limit 41,000 vehicles/18 hour day	78
Class B 48 km/hour speed limit 21,000 vehicles/18 hour day	71
Class C 48 km/hour speed limit 4,800 vehicles/18 hour day	63

As distance, *d*, from the road increases, the noise level falls, the rate of fall depending on the nature of the intervening ground. For hard surfaces the rate is approximately 3dB for each doubling of distance while for propagation over grassland about 4.5dB per doubling of distance may be assumed. A 'hard ground' correction for distance quoted in BRE Digest 186: 1976 is $-10 \log \frac{d}{3.5}$. When roads run through towns and cities and there are continuous rows of buildings close to the road on both sides, reflections cause sound levels to increase. In addition it is observed that there is little fall off in sound level with height.

Table 4.2 Corrections to be applied to A weighted sound level of transport noise sources to give approximate unweighted octave band sound levels

Source	Octave band centre frequency 63	125	250	500	1K	2K	4K	8K
Traffic	+3	+3	+1	−2	−6	−10	−15	−20
Diesel locomotive hauled passenger and freight	+4	+3	−4	−5	−5	−8	−12	−16
Diesel multiple unit (dmu) passenger	−2	−2	−5	−6	−3	−7	−15	−23
Electric locomotive hauled passenger	−9	−11	−15	−9	−4	−4	−14	−28
Electric locomotive hauled freight. Electrical multiple unit (emu) passenger	−4	−5	−8	−4	−3	−6	−13	−26
Inter-City 125	−2	0	−9	−6	−4	−8	−10	−17
Jet aircraft take-off	−3	−1	+1	0	−3	−11	−27	−39
Jet aircraft landing	−9	−3	−2	−4	−6	−5	−8	−17

If a value of L_{10} has been established and a typical sound spectrum is required, the corrections given in Table 4.2 can be applied to the L_{10} dB(A) level to give the unweighted octave band levels.

Train noise

The assessment of train noise is usually made in terms of the 24 hour equivalent continuous sound level. However, from the point of view of estimating the required façade performance to meet a given internal criterion, the peak bypass level in dB(A) is of more use. Train noise results in part from wheel–rail interaction and in part from locomotive noise, so noise level and spectrum shape will vary with both track condition and the power output of the locomotive. In general, the noise level will increase with train speed and as the condition of the track deteriorates. For trains running on track of average to good quality at typical operating speeds, representative noise levels are given in Table 4.3.

Table 4.3 Typical train noise levels

Type of train	Typical speed mph	Maximum bypass level in dB(A) at 50 m
Diesel locomotive hauled passenger	100	84
Diesel locomotive hauled freight	70	80
Diesel multiple unit (dmu) passenger	70	77
Electric locomotive hauled passenger	100	88
Electric locomotive hauled freight	70	85
Electrical multiple unit (emu) passenger	90	81
Inter-City 125	125	83

The rate of fall in level as the distance from the track is increased is generally taken to be 3dB for each doubling of distance from the line out to a distance equal to one train length. Beyond this point the rate of decay is 6dB per doubling of distance.

If a representative spectrum of the noise is required, the corrections given in Table 4.2 can be applied to the maximum bypass level to given the maximum level in the octave bands. In practice, the peak level in each octave band will not necessarily occur at the same time and quite large variations in the band levels are possible, especially at frequencies below 250Hz where ground interference effects can occur.

Underground noise

For cities with an underground system, for example London, Paris and New York, many redevelopment or refurbishment sites will be over or near tunnels. The ground vibration effects resulting from the passage of underground trains will depend on many variables – soil strata, speed of train, train length, depth and distance, for instance. Vibration levels typically peak at 63Hz and may affect sensitive buildings up to 70 m or more either side of tunnel routes.

Special isolated foundation details may be used for the most badly affected sites, but generally the use of heavy rather than lightweight masonry structures will reduce the pick up of ground vibration by an extra 10–20dB.

Aircraft noise

Frequent disturbance from aircraft noise is confined to relatively small areas around airports and under flight paths. For those unfortunate enough to live under take-off and landing flight paths the noise levels can be very high and have a considerable impact on residential areas.

There are many objective measures for assessing the disturbance. One example is the noise and number index (NNI), defined in section 2.2.1.

For design purposes, an idea of the maximum noise level that will occur is likely to be more useful, so in Table 4.4 some typical noise levels in dB(A) are given for commercial aircraft during both take-off and landing. The levels are given for a position directly under the flight path at a distance of 5 km from start of take-off or touch down. For distances other than 5 km the height of the aircraft can be estimated by assuming constant slope flight paths. The noise level is then obtained from the value given in the table by applying a correction of $-26.7 \log_{10} (h_1/h)$, where h_1 is the new height and h is the height given in the table. This correction is equivalent to a reduction of 8dB for each doubling of the distance from the aircraft. At positions other than under the flight path the minimum slant distance to the aircraft has to be estimated.

On take-off, beyond 5 km, the engine power is generally reduced and the ascent rate is less so that typically at 25 km the height is 1500 m. During this phase noise levels are also lower and in addition to the decrease due to distance a reduction of about 5dB(A) can also be assumed.

In Table 4.2 corrections are given which enable representative unweighted octave band spectra to be obtained from the A weighted noise level for both take-off and landing.

Although no information for military aircraft or helicopters has been included in Table 4.4 it should not be forgotten that these can be a cause of disturbance, especially close to military bases and in areas where low flying is practised.

Industrial noise

There are many different industrial noise sources and it is not possible to generalise about the noise produced by them. It is hoped that a few examples will illustrate the types of situation that have to be dealt with and the range of noise levels that can occur in practice.

Although there are many obvious noisy items of plant and industrial processes the less obvious ones must not be overlooked. In many cases disturbance is caused by small items of plant often ignored or not considered as sources of noise – inlet and extract fans, small compressors, standby generators and transformers, for example.

Table 4.4 Typical aircraft noise levels in dB(A) at a distance of 5 km from start of roll or touchdown

Aircraft type	Take-off Height m	Take-off Noise level dBA	Landing* Height m	Landing* Noise level dBA
BAe Concorde	250	120	260	101
Trident†	380	105	260	93
HS125	500	92	260	86
HS748	358	90	260	83
111	450	97	260	90
Douglas DC8	500	99	260	93
DC9	450	94	260	88
DC10	450	92	260	86
Boeing 707	450	101	260	96
747	280	102	260	90
747SP	380	97	260	85
727	450	99	260	88
737	500	93	260	84
767	600	85	260	84
757	650	82	260	84
Lockheed Tristar L101	350	93	260	85
European Airbus A300	600	86	260	85
Fokker F28	600	87	260	81
Twin Piston	400	83	260	71
DHC Twin Otter series 300	550	68	530‡	70
DHC Dash 7	500‖	67	260§	68

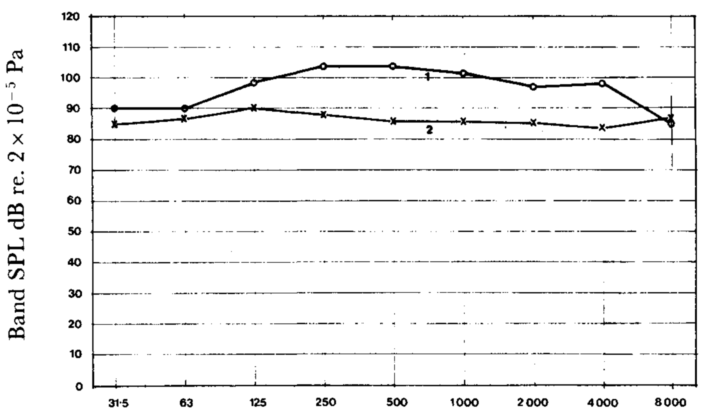
* A 3° approach is assumed.
† Taken out of service in the UK 1 January 1985.
‡ 3 km from touchdown 6° approach.
§ 2 km from touchdown 75° approach.
‖ 4 km from start of take-off.

Generating houses

An example of an extremely noisy internal environment is a generating house which may contain as many as ten diesel or gas turbine generators, each of large megawatt capacity. The reverberant noise level within such a room is usually well over 100dB(A), the actual level depending on the number of generators and the operating load. Figure 4.1 shows the noise levels measured in a hall of approximate volume 6500 m^3 when two 2MW diesel generators were operating near full load. Also shown in the figure are the estimated levels from two 2MW gas turbine generators operating in the same building.

In both cases it should be realised that, as well as possible noise breakout from the generating house, noise from the engine air inlets and exhausts will also have to be controlled.

Figure 4.1 Representative octave band reverberant sound pressure levels in a generating hall.



1 Diesel engines (106dB(A))
2 Gas turbines (92dB(A))

Metal fabrication
The hammering and forming of large steel plates can generate very high levels of impulsive noise and this has long been recognised as a potential hearing hazard for the worker. This type of noise may also create hostility in a local community if it is not properly contained. The average reverberant noise levels measured in a workshop of volume 12,500 m^3 are given in Figure 4.2. Peak levels can be as much as 15–20dB higher than the average reverberant levels.

Manufacturing and assembly areas
Not all manufacturing and assembly processes are intrinsically noisy, but noise sources such as motors, fans, air jets, steam vents and metal-to-metal contacts can give surprisingly high noise levels. To illustrate this, Figure 4.2 gives average noise levels measured in a large industrial hall, volume 45,000 m^3, containing eleven production lines on which electric light bulbs were made.
Also given in this figure are the noise levels recorded in a bottling hall of volume 10,000 m^3. The major noise sources in this case were bottle-to-bottle contacts and air jets.

Entertainment noise
High levels of noise can result from both indoor and outdoor activities. Noisy outdoor pursuits include motor racing, scrambling, rifle or pistol shooting and model cars or aircraft. In these cases noise control has usually to be either at source and/or through legislative restrictions on times and places of operation. The noisiest indoor entertainments are generally associated with music performances of one kind or another.

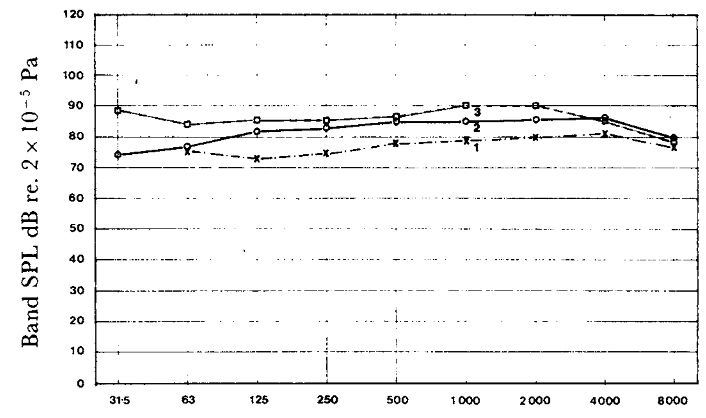


Figure 4.2 Representative octave band reverberant sound pressure levels in industrial halls.

Discotheques

In discotheques and night-clubs amplified music creates extremely high noise levels and published data have recorded values as high as 113dB(A). On average, levels range between 90 and 110dB(A) and the spectrum is rich in low frequencies, making its containment that much more difficult. A typical spectrum for discotheque noise is given in Figure 4.3.

An example of the disturbance that the low frequencies can cause was encountered in a cinema located over a discotheque where the cinema audience was all too aware of a low frequency beat from the discotheque, clearly audible even during noisy parts of the film.

Live rock and pop music

At this type of venue large banks of speakers produce amplified sound which creates sound levels as high as 120dB(A) in the area of the stage, and levels of between 100 and 105dB(A) near the centre of the auditorium. Again, the sound spectrum has considerable low frequency content, as is shown in Figure 4.3. In practice rooms and recording studios, levels of over 110dB(A) are to be expected.

Classical music

Noise levels from classical music are considerably lower than those from rock and pop music, with the loudest levels being created by large orchestras employing both percussion and brass. During loud passages the average sound level in the auditorium is around 80dB(A) although peaks up to 20dB higher can occur. Classical music does not generally have as much energy at low frequencies as rock and pop music. A representative spectrum is shown in Figure 4.3.

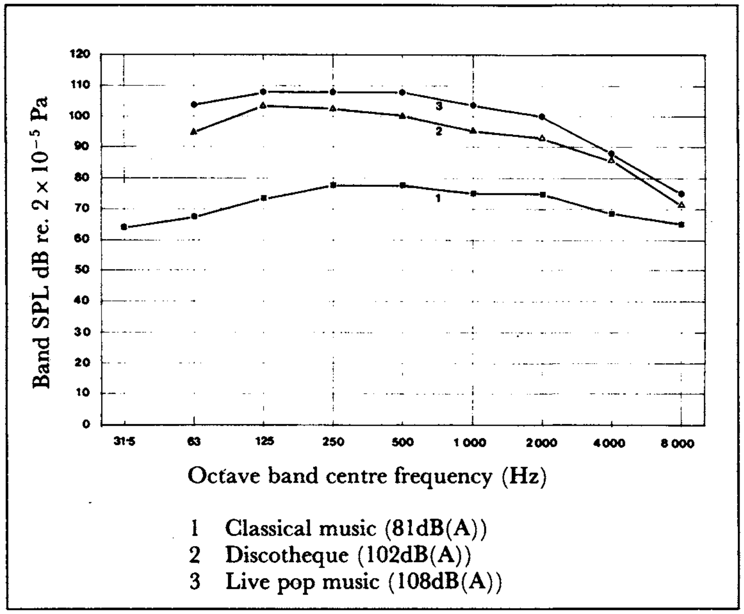


Figure 4.3 Representative octave band reverberant sound pressure levels for some types of music.

Gymnasia and swimming-pools

These spaces generally tend to be quite reverberant which raises the noise levels and may also make communication difficult, giving problems to anyone teaching or coaching. The actual level will depend upon occupancy and activity. Generally, the maximum levels can be expected to be in the 70–80dB(A) region.

4.1.2 Effects of meteorology on sound propagation

Meteorological effects on the propagation of noise may have to be considered in the case, say, of meeting boundary noise control standards on a very large site given a noisy industrial process plant remote from the boundary.

While measuring noise in the open it can be observed that the level continuously varies. If variation in the source power is discounted the variation can be attributed to either noise generated at the microphone due to wind passing over it or to atmospheric turbulence along the path between source and microphone.

Wind noise

Wind generated noise can cause very large fluctuations and to minimise its effect the microphone is generally covered by a wind shield usually made from an open-celled polyurethane foam. Wind noise is predominantly low frequency and by using the A weighting its effects can also be offset.

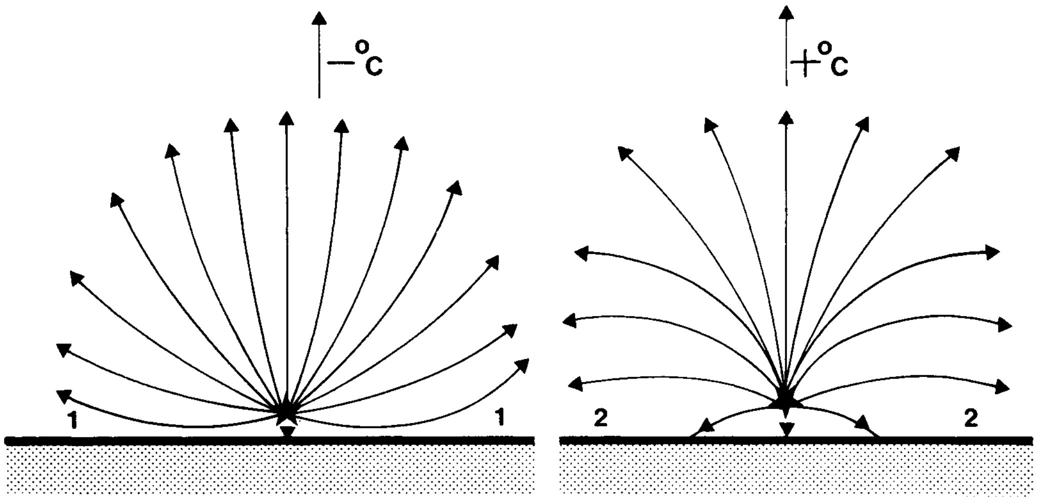
Turbulence

The effects of turbulence cannot be reduced and to obtain a value of the noise level the fluctuations have to be averaged over a period of time. The longer term changes that different climatic conditions can produce are important. In the atmosphere, temperature and wind velocities will vary from the ground upwards and it is the presence of these gradients which influences the propagation of sound. Due to the gradients the effective velocity of the sound varies with height and this variation causes the sound to be refracted either towards or away from the ground. If the sound is refracted down towards the ground an increase in sound levels over those observed in a calm isothermal atmosphere can occur, while if the sound is refracted away from the ground a reduction in sound levels can be obtained.

Temperature gradient

First consider a vertical temperature gradient in the absence of any wind gradient. If the temperature decreases with height, that is, there is a temperature lapse (typically $-1°C/100$ m), the velocity of sound decreases with height and sound is bent upwards away from the ground, giving a shadow zone or reduction in sound levels at positions all around the source. However, if the temperature increases with height, so that there is a temperature inversion (typical values are between 1°C and 10°C/100 m), the velocity of sound increases with height and sound is bent down towards the ground, giving an increase in sound levels all around the source. These conditions are illustrated in Figure 4.4.

Figure 4.4 Refraction of sound rays by temperature gradients.



Effects of temperature variation with height on external sound propagation

1 Sound shadow region
2 Sound enhancement region

Wind gradient

Now consider a vertical wind gradient in the absence of any temperature gradient. Wind speed always increases from the ground upwards, hence, in a direction with the wind, the effective sound velocity will be increased with height and the sound will be returned to the ground, giving enhanced sound levels. Going against the wind the effective sound velocity will decrease with height and the sound will be refracted away from the ground, giving a shadow zone or area of reduced sound levels. This is illustrated in Figure 4.5

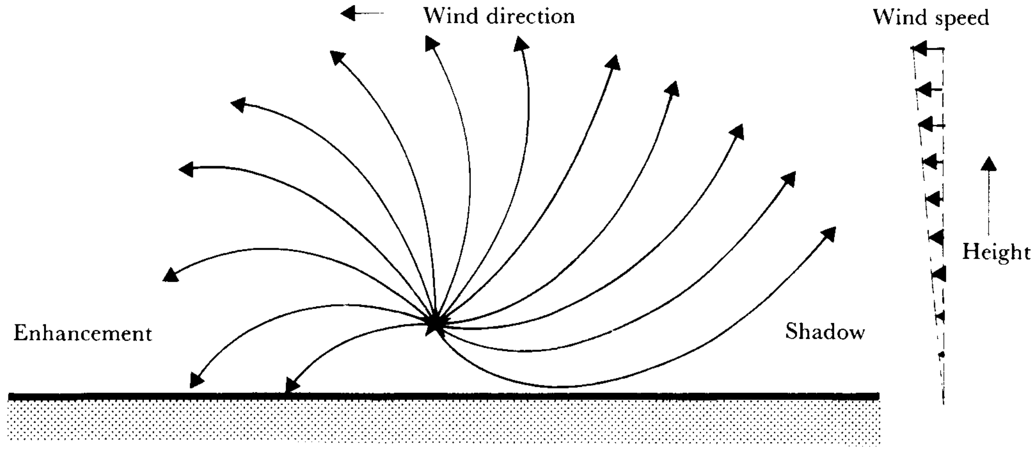


Figure 4.5 Refraction of sound rays by a wind speed gradient.

Combination effects

In the past the importance of temperature inversions in measuring sound levels has been stressed, but it is quite rare for temperature gradients to exist in the absence of wind gradients and generally wind gradients have a much greater effect on sound propagation. The temperature gradients determine the wind gradients which then have more effect than the temperature gradients themselves. As has been emphasized, it is the wind gradient or wind shear that is important and this may not follow the same pattern as the surface wind velocity measured, say, at 2 m or 10 m above the ground. The wind shear is related more to the stability of the atmosphere. For example, on a hot sunny day there will usually be a temperature lapse and vigorous mixing of the atmosphere takes place because of thermals. The surface flow is thus mixed with that at, say, 600 m and the wind speeds and directions vary little from the surface to this height. The wind shear is thus small and, although there may be a significant ground level wind, the effect on noise levels is small.

Night-time propagation

On a clear night the surface temperature falls faster than at 600 m and eventually a temperature inversion will develop. Vertical mixing of air is inhibited, producing laminar flow. The surface air is slowed by friction and backs in direction relative to the 600 m wind. Thus, wind shear is much increased and strong enhancements are obtained downwind and strong shadows upwind. Thus, for the same surface wind component these stable conditions have a much greater effect on noise levels than the unstable daytime conditions.

Warm and cold fronts

Surface wind direction can also be misleading in the presence of warm or cold fronts. Ahead of a warm front the winds will increase and veer with height, while to the rear of a cold front they will increase and back with height. The change in direction can be as great as 180° from the surface to a height of 3000 m. This change in wind direction, coupled with an increase in wind speed, has a marked effect on the velocity gradient. Sound initially bent upwards when projected upwind can be bent back down by the upper winds giving large increases, up to 20dB, in level several kilometres from the source in a direction effectively upwind of the surface wind direction.

The quantitative estimation of climatic effects on noise levels is extremely difficult and there are few reliable procedures that can be followed. However, one method, to be found in CONCAWE Report 4/81, is worth considering.

The basic meteorological data required for the method are the wind velocity (speed and direction) at a height of about 10 m and a measure of the stability of the lower atmosphere. This latter quantity is expressed in terms of Pasquill Stability Categories. These categories define the state of the lower atmosphere in terms of wind, cloud cover and solar radiation. These are summarised in Table 4.5. The definitions of meteorological categories are given in Table 4.6.

The atmospheric attenuations corresponding to each meteorological category are given in Table 4.7.

Table 4.5 Definition of Pasquill Stability Categories

Wind speed m/sec*	Daytime incoming solar radiation mW/cm^2				1 hour before sunset or after sunrise	Night-time cloud cover		
	>60	30–60	<30	overcast		0–3	4–7	8
$\leqslant 1.5$	A	A–B	B	C	D	F or G†	F	D
2.0–2.5	A–B	B	C	C	D	F	E	D
3.0–4.5	B	B–C	C	C	D	E	D	D
5.0–6.0	C	C–D	D	D	D	D	D	D
>6.0	D	D	D	D	D	D	D	D

* Wind speed is measured to the nearest 0.5 $msec^{-1}$.

† Category G is restricted to night-time with less than 1 octa of cloud and a wind speed of less than 0.5 $msec^{-1}$.

Data in this form can be obtained from meteorological stations.

From the meteorological data six meteorological categories for estimating atmospheric attenuation are defined as in Table 4.6.

Table 4.6 Meteorological categories

Meteorological category	Pasquill Stability Category A,B	C,D,E	F,G
1	$V < -3.0$	—	—
2	$-3.0 < V < -0.5$	$V < -3.0$	—
3	$-0.5 < V < +0.5$	$-3.0 < V < -0.5$	$V < -3.0$
4	$+0.5 < V < +3.0$	$-0.5 < V < +0.5$	$-3.0 < V < -0.5$
5	$V > +3.0$	$+0.5 < V < +3.0$	$-0.5 < V < +0.5$
6	—	$V > +3.0$	$+0.5 < V < +3.0$

A positive wind velocity is from source to receiver.

Table 4.7 Atmosphere attenuations corresponding to meteorological category

Meteorological category	Attenuation in dB Octave band frequency (Hz) 63	125	250	500	1K	2K	4K
1	8.0	5.0	6.0	8.0	10.0	6.0	8.0
2	3.0	2.0	5.0	7.0	11.5	7.5	8.0
3	2.0	1.5	4.0	3.5	6.0	5.0	4.5
4	0	0	0.0	0.0	0.0	0.0	0.0
5	–1.0	–2.0	–4.0	–4.0	–4.5	–3.0	–4.5
6	–2.0	–4.0	–5.0	–6.0	–5.0	–4.5	–7.0

Negative attenuations indicate an increase in levels.

Qualification

A fault with the method is that the stability categories were originally designed to deal with the calculation of material dispersed in the atmosphere. Consequently, certain combinations of wind and temperature gradients, which behaved similarly in terms of the dispersion of air-borne material and were therefore combined in one category, have a markedly different effect on sound propagation. Thus, the attenuations given in Table 4.7 should be treated as broad indicators of behaviour rather than precise estimates.

4.1.3 Legal control

The law relating to noise is a field of study in its own right. When a noise causes a problem to people, they can take legal action to get the noise abated. The commonest redress for private nuisance is in equity by an injunction – a court order to restrain the continuation of the wrongful act or omission. Failure to comply could mean imprisonment. The injunction can be granted at county court level, without recourse to the high court.

Categories

There are three categories of noise nuisance: the first two are criminal matters, the third entails the law of tort (a tort is a civil wrong which is not bad enough to be treated as a crime).

- public nuisance – a danger or inconvenience to people
- statutory nuisance – one defined by statute
- private nuisance – interference of an individual's rights to enjoy his property and maintain reasonable conditions to live in.

Rights

The individual's rights are likely to be put above the nuisance causer's rights for commercial gain arising from his activities. According to Lord Denning, a private nuisance becomes a public one when its effects are so far reaching that it would no longer be reasonable for one individual to take action or proceedings. Often the first line of action in complaint is approach by members of the public to the Environmental Health Officer (EHO) at the local authority who may then investigate the noise source. In planning matters, noise and other environmental issues may be the subject of public inquiry.

In the case of an existing noise nuisance, a prescriptive right to continue can be assumed if the nuisance has already been committed for twenty years without interruption and without dispute.

Context is relevant to noise nuisance – a judgment from the 1879 *Sturges* v *Bridgeman* case often quoted is 'what would be a nuisance in Belgrave Square would not necessarily be so in Bermondsey'.

Health and Safety Act

Employers have a particular responsibility to protect employees' hearing. In industry, some areas have to be marked off as 'ear defender zones' when the exposure is likely to cause insidious hearing loss. Allowable levels are related to exposure, but such zones are essential where employees experience eight hours or more at a level of 85 Leq dB(A) (80 Leq dB(A) for pure tones).

Certain vibrations can be hazardous; for example, pneumatic drill users can experience pain, numbness and swelling caused by the trauma of the hands having to control vibrating tools. Transmission of vibration through the body can destabilise body balance and coordination mechanisms. Explosive reports from guns and the shot-firing tools used in the construction industry make it necessary for protective measures to be taken.

Planning control

When someone applies for planning permission to build a new building or change the use of an existing building, the local planning authority may

- impose conditions regarding noise levels, or
- refuse permission if the site is too noisy for the intended use, or if the proposed use is likely to constitute a noise nuisance.

Reference by the local planning authority may be made to either DoE Circular 10/73 'Planning and Noise' or 1/85 'The Use of Conditions in Planning Permissions'.
The setting of noise control standards at planning stage and their subsequent observance by a developer or owner does not preclude an individual taking legal action under common law to attempt to abate noise nuisance arising from the development. The owner as applicant for planning permission in some instances may have a residual responsibility to comply with a planning control condition, even if a tenant, say, is causing that nuisance. An outline of noise legislation is given in Kerse's *The Law Relating to Noise.*

4.2 Control at Source

4.2.1 Impact sound

Impact sound refers to sound produced when a short duration impulse, such as a footfall, acts directly on a structure. The force of the impulse causes the structure to vibrate and thus radiate sound (Figure 4.6). The frequency content of the sound depends largely on the duration of the impact; a short, sharp event giving a broad band frequency content while a longer duration event caused, for example, by having a resilient layer over the structure, will contain mainly low frequency sounds and be subjectively less disturbing.

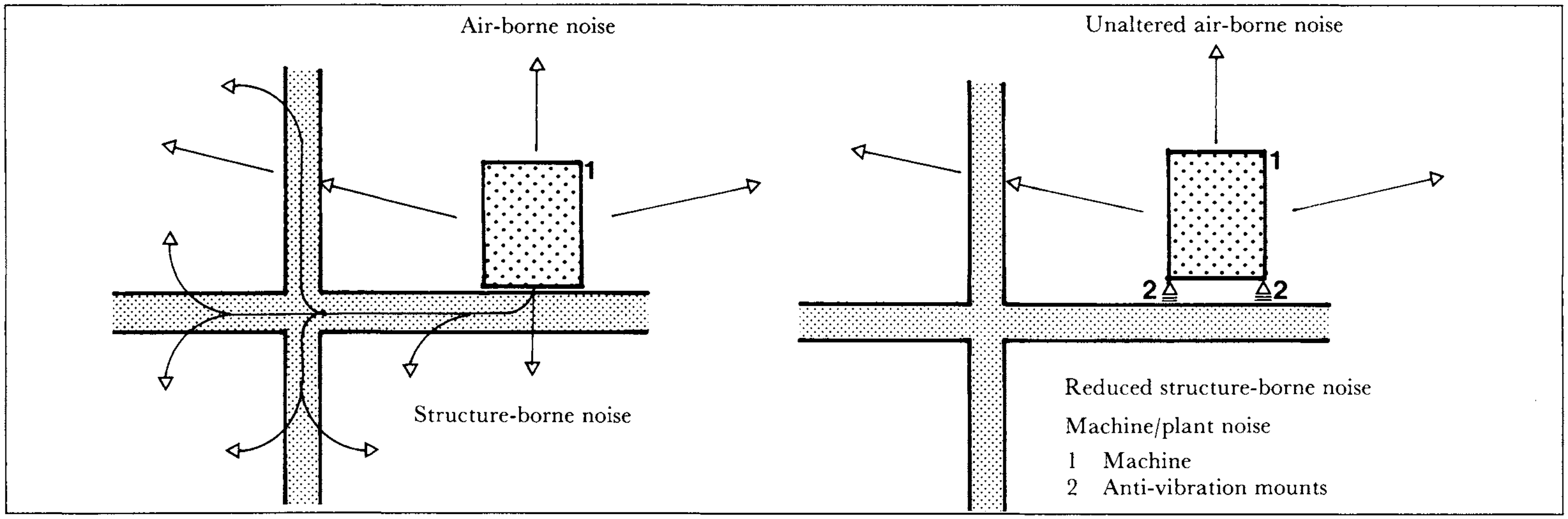


Figure 4.6 Reduction of structure-borne noise by use of vibration isolation.

4.2.2 Structure-borne sound

Vibration which gets into a structure, especially homogeneous structures, can travel a long way with little attenuation because the internal damping of the structure is usually low. If the vibration eventually appears as sound because of the vibration of some large surface such as a wall or partition, the sound is referred to as structure-borne sound. The original vibration of the structure can be caused by air-borne sound, mechanical vibration or impacts.

Control of impact and structure-borne sound

To reduce both impact and structure-borne noise the energy entering the structure should be minimised and the attenuation of energy in the structure should be increased. This is achieved by the use of floating floors, vibration isolators and resilient surface layers, and the use of structural discontinuities.

4.2.3 Floating floors

A floating floor provides a practical means of obtaining high impact sound insulation in a building. It usually takes the form of a slab of reinforced screed or boarded battens set on a resilient layer which isolates it from the structural floor, Figure 4.7. The floating floor and resilient layer form a mass-spring system which has a resonant frequency, f_r, given by

$$f_r = \frac{1}{2\pi}\sqrt{\frac{E}{Md}}$$

where E is Young's Modulus for the resilient material, M is the effective mass per loaded unit area and d is the thickness of the loaded resilient layer,

$$\text{or } f_r = 5.03\sqrt{\frac{E}{Md}} \qquad \text{where}$$

M has units of kgm^{-2}, E units of newtons m^{-2} and d is in mm.

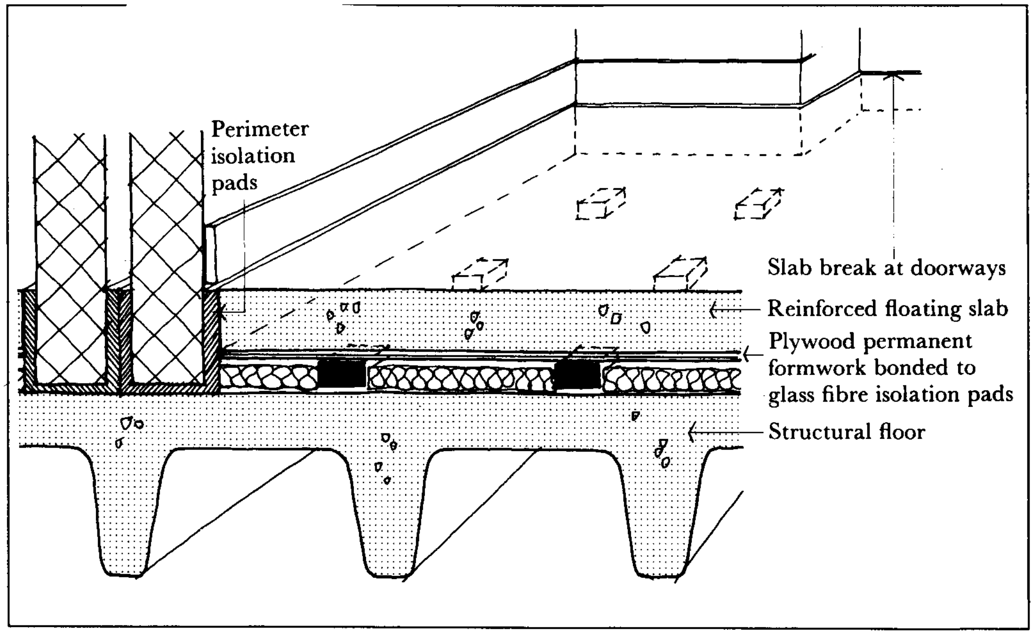


Figure 4.7 An example of a floating floor construction.

Example

1. A floating floor is constructed of 22 mm tongue-and-groove boards laid on a 50 mm glass fibre quilt. The boards are supported on 50 mm wide battens at 500 mm centres. To estimate the frequency above which an improvement in impact isolation might be expected we proceed as follows.

   The flooring has a surface mass of 15 kgm^{-2} but as it is strip supported the effective loading on the glass fibre quilt is

$$15 \times \frac{\text{total area of floor}}{\text{total area of battens in contact with quilt}}$$

$$\simeq 15 \times \frac{\text{batten spacings}}{\text{batten width}}$$

$$= 15 \times \frac{500}{50}$$

$$= 150 \text{ kgm}^{-2}$$

   The resonant frequency of the floating floor is given by

$$fr = 5.03 \sqrt{\frac{E}{Md}} \quad \text{Hz}$$

   where E is Young's Modulus of the glass fibre quilt

   $= 10^5 Nm^{-2}$, M is the mass loading per unit area of the quilt $= 150$ kgm^{-2} and d is the compressed thickness of the quilt in mm.

   For a typical impact grade glass fibre quilt the compression will be about 30 per cent so that compressed thickness, d, will be

$$d = 50 \times 0.7 \text{ mm}$$
$$= 35 \text{ mm}$$

   Hence

$$fr = 5.032\sqrt{\frac{10^5}{150 \times 35}} \quad \text{Hz}$$

$$fr = 22\text{Hz}$$

   So that for frequencies above 31Hz (i.e. $22 \times \sqrt{2}$) an improvement in impact insulation should be achieved.

Resonant frequency

The resonant frequency is a most important parameter as effective isolation only occurs at frequencies about two or three times higher than it. For frequencies lower than the resonant frequency, the floating and structural floors vibrate approximately in phase. The three layers react almost as an entity and there is no improvement in insulation. In fact at the resonant frequency there can be an increase in the transmitted energy. For higher frequencies, above the resonant frequency, the separated layers move out of phase and the energy transmitted across the system is reduced as the frequency increases. Thus, careful choice of resilient layer in relation to the load is required and it must be remembered that it is the load per unit area that is important. A point or strip loading will require a much stiffer resilient layer than slab loading for the same resonant frequency.

A well-designed floating floor that is structurally isolated from walls and floor can provide improved air-borne sound insulation as well as improved impact performance.

Resilient surface layers

Elastic surface layers covering a surface will cushion an impact and reduce the energy entering the structure. The softer the covering in relation to the impacting mass the greater will be the isolation.

In homes and offices the use of floor carpet, especially if used with a good underlay, is an extremely effective way of reducing the noise from footsteps, chair scrapes, etc. Resilient vinyl or cork can also be effective but it should be in the form of a reasonably thick layer, say 6 mm.

4.2.4 Vibration isolators

A common way of reducing structure-borne noise caused by mechanical vibration is to isolate the source of the vibration from the main building structure by the use of flexible mounts or springs, as shown in Figures 4.8 and 4.9. The source of the vibration might be a large airhandling fan, a compressor, a pump, or a duct or pipe, the walls of which are set into vibration by turbulent fluid flow or noise within them. The use of vibration isolators, although they may reduce the amount of vibrational energy entering the structure, will not reduce the actual noise radiated by the source itself, and other noise control methods will have to be applied if this is a problem.

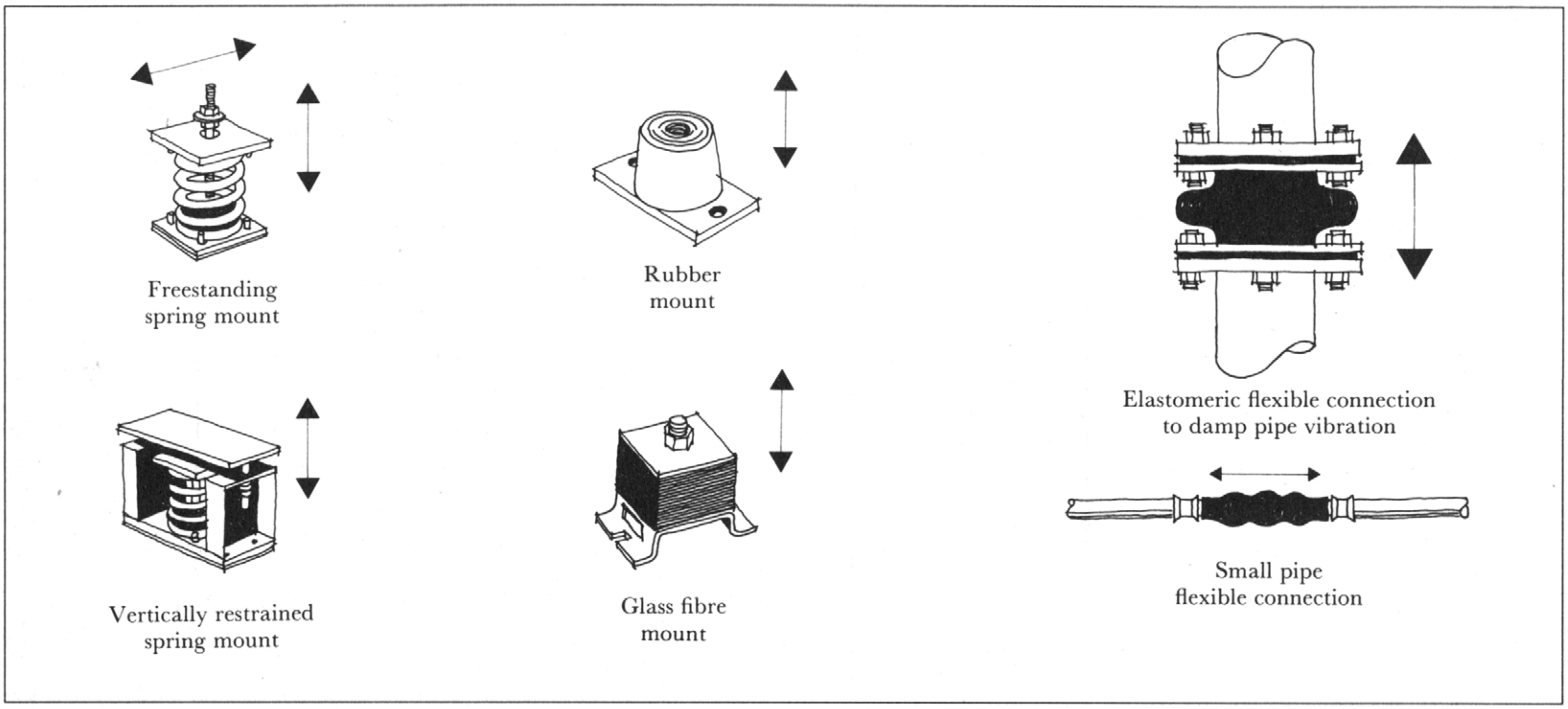


Figure 4.8 Examples of vibration isolators.

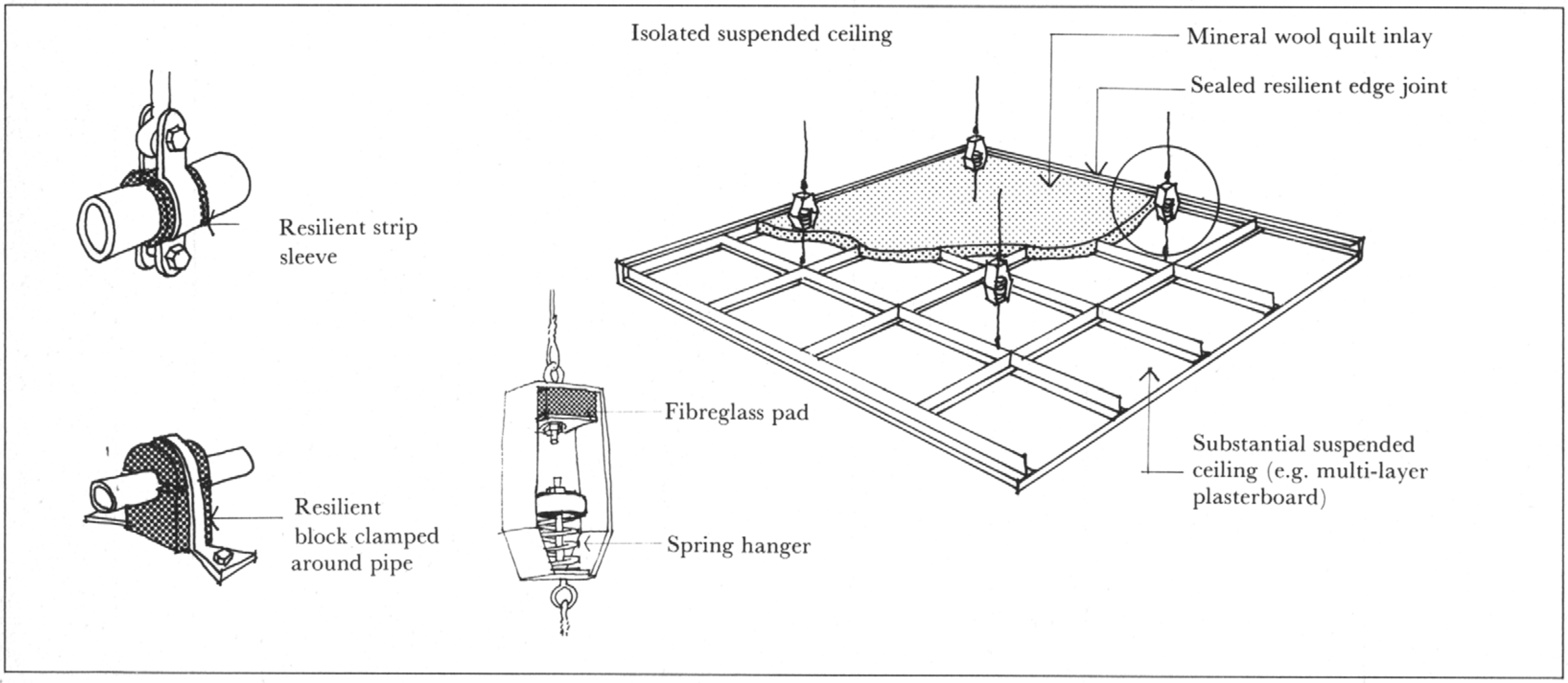


Figure 4.9 Examples of vibration isolators.

Force transmissibility

A measure of the performance of a spring or isolator is the force transmissibility defined as

$$\text{force transmissibility} = T_F = \frac{\text{force transmitted to the structure}}{\text{force applied by source}}$$

If the system is lightly damped, it can be shown that the force transmissibility is given by

$$T_F = \left| \frac{1}{1 - (f/f_r)^2} \right| \tag{4.1}$$

where f is the frequency of the driving force and f_r is the resonant frequency of the spring-machine system. This equation is shown graphically in Figure 4.10 from which it can be seen that isolation does not begin until the driving frequency is $\sqrt{2}$ times the resonance frequency. Also shown are some curves when damping is present and it can be seen that the degree of isolation for a given forcing frequency decreases as the damping in the system increases. However, some damping is usually necessary to prevent excessive vibration as the system passes through the resonance frequency as it starts up or slows down.

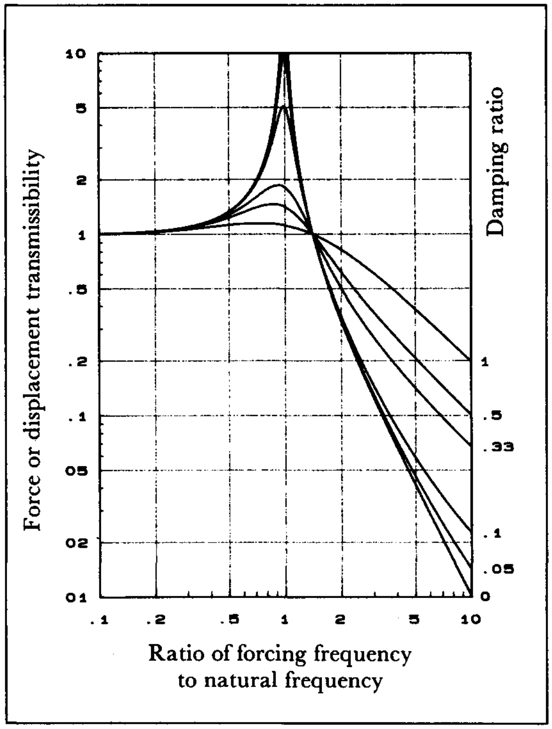
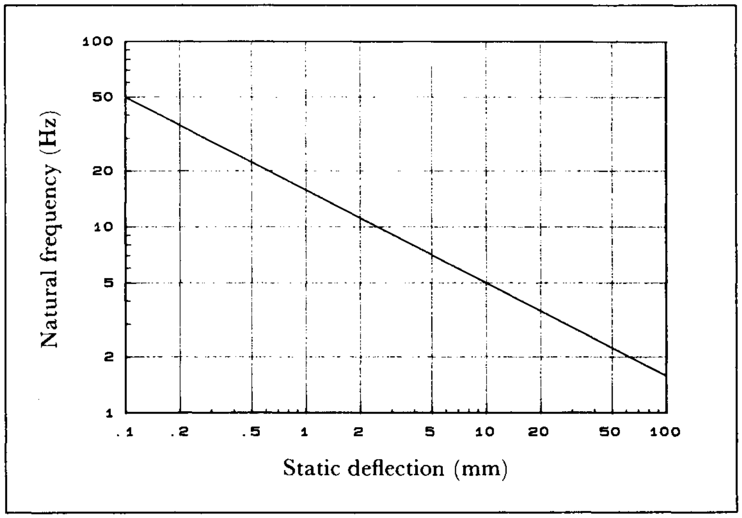


Figure 4.10 Transmissibility of a single degree of freedom vibration isolator as a function of frequency.

Figure 4.11 Resonant frequency of a single degree of freedom system as a function of its static deflection.



Mass-spring system

The resonant frequency of the mass-spring system is given by

$$f_r = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \quad \text{Hz}$$

where K is the stiffness of the spring and M is the mass of the machine. Alternatively f_r can be found using

$$f_r = \frac{15.8}{\sqrt{d}} \quad \text{Hz} \tag{4.2}$$

where d is the static deflection, in mm, of the spring under the weight of the machine. This equation is given graphically in Figure 4.11. For isolators made from rubber the equation should be modified to

$$f_r = \frac{19.5}{\sqrt{d}} \quad \text{Hz}$$

as they exhibit an apparent stiffening under vibration.

Combining equations 4.1 and 4.2 we can, for lightly damped systems, estimate the static deflection required to give a specific transmissibility. The result is represented graphically in Figure 4.12. For example, if an isolation of 80 per cent (transmissibility of 20 per cent) is required at all frequencies above 10Hz, from the graph we see that a natural frequency of 4Hz is required or that the isolator selected should undergo a displacement of about 15 mm under its share of the total load.

Example

A water pump weighing 100 kg operates at a rotational speed of 1440 rpm. To isolate it from the main structure it is mounted on a frame which is supported on four vibration isolators. Assuming each isolator is equally loaded, specify the performance required from each spring so that 90 per cent isolation is achieved.
As the rotational frequency is 1440 rpm the fundamental forcing frequency is $1440 \div 60 = 24\text{Hz}$.
Using Figure 4.9 a transmissivity of 10 per cent is achieved, with zero damping when

$$\frac{\text{forcing frequency}}{\text{natural frequency}} = 3.3$$

So $$\text{natural frequency} = \frac{24}{3.3} = 7.3\text{Hz}$$

Thus, as each isolator carries a load of 25 kg it must under this load give a static deflection of 4.4 mm if an isolation of 90 per cent is to be achieved.
This result could have been found directly from Figure 4.11 or by using equations 4.1 and 4.2.

Displacement transmissibility

In some situations isolation of delicate instruments from structure vibrations is required. In this case it is the displacement transmissibility, T_d, defined as

$$T_d = \frac{\text{displacement of isolated instrument}}{\text{displacement of supporting structure}}$$

which is important.
Fortunately the displacement transmissibility is identical with the force transmissibility and all the above equations and figures apply.
It is not always obvious when to use vibration isolators or which ones to choose. With the isolation of delicate instruments, measurement of the structural vibration will indicate whether or not it is above the limits required by the instrument and hence whether or not isolation is required. However, deciding whether or not to isolate a vibrating machine from a structure is not so simple. In an existing situation the noise within the room containing the machine can be measured, and also in adjacent rooms. If the difference in noise levels is less than might by expected from the airborne insulation properties of the intervening walls and floors, it can perhaps be assumed that vibration isolation is needed although flanking transmission must first be discounted. In the planning stage it is unlikely that there will be sufficient information to make calculations and hence the use of isolators on all sources that might impart vibrational energy to a structure should be considered.

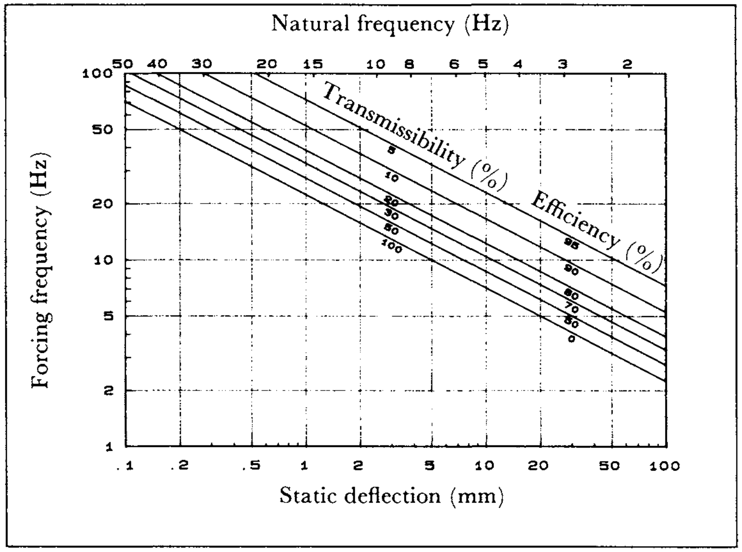
The primary concern when selecting an isolator is that it will give adequate transmissibility. The information required to estimate the required transmissibility is usually unavailable and as a general rule 90 per cent efficiency should be aimed for, i.e. a transmissibility of 0.1 or 10 per cent.

From the curves in Figure 4.10 it is clear that to achieve this degree of isolation the natural frequency of the system must be equal to or less than one-third of the lowest forcing frequency. With rotating machines the lowest frequency is usually the shaft rotational frequency, but with reciprocating machines there may be lower frequency components associated with power strokes.

Isolator specification

Once the natural frequency has been fixed using, for example, Figure 4.10 or equation 4.1, the required static deflection is found from equation 4.2 or Figure 4.11. An isolator which gives this static deflection under its share of the machine weight is then selected.

Figure 4.12 Performance of lightly damped isolators.



If the isolator requires little damping, Figure 4.12 can be used directly to obtain the static deflection in terms of efficiency and lowest forcing frequency.

To see the difference that damping can make, consider again an isolator which is required to give an isolation of 80 per cent at all frequencies above 10Hz. From Figure 4.11 we find that a system natural frequency of 4Hz is required. If the isolator is damped and the damping ratio is 0.5, it can be seen from Figure 4.10 that for this natural frequency and a forcing frequency of 10Hz (ratio 2.5) the transmissibility is 43 per cent, or the isolation is only 57 per cent. With a damping ratio of 0.5 we see, using Figure 4.10, that to obtain an isolation of 80 per cent the ratio of forcing frequency to natural frequency should be about 5 so that a natural frequency of 2

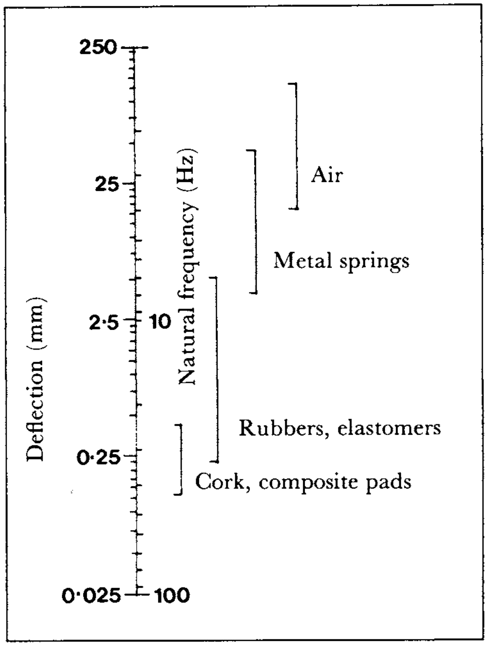


Figure 4.13 Operational ranges of vibration isolators.

rather than 4Hz, as would have been the case with no damping, is required.
An isolator mount will need inherent or added damping if

- the machine has a variable speed over a range that includes the natural frequency of the isolator and machine
- the machine has a long run up and run down time
- the machine is impact loaded.

A variety of materials and methods are used to provide vibration isolation but the details are probably better obtained from the various manufacturers. Because of the differing properties of the different materials, however, each one tends to be best suited to a particular range of natural frequency and deflection. This is illustrated in Figure 4.13. Table 4.8 also gives some guide towards application.

Table 4.8 Use of vibration isolators

Application	Type of isolator
Air compressors	Passive-air, damped metal springs, rubber/neoprene in shear
Airhandling units	Undamped metal springs, rubber/neoprene in shear shear
Blowers	Rubber/neoprene in shear or compression or in pad form
Diesel engines	Damped metal springs, rubber/neoprene in shear. Pads of cork or composite material
Fans	Undamped metal springs, rubber/neoprene in shear
Microscopes	Active-air, rubber/neoprene in shear or compression
Punch pressers	Passive-air, rubber/neoprene in shear or compression
Transformers	Rubber/neoprene in compression pads of composite material
Typewriters	Rubber/neoprene or felt pads.

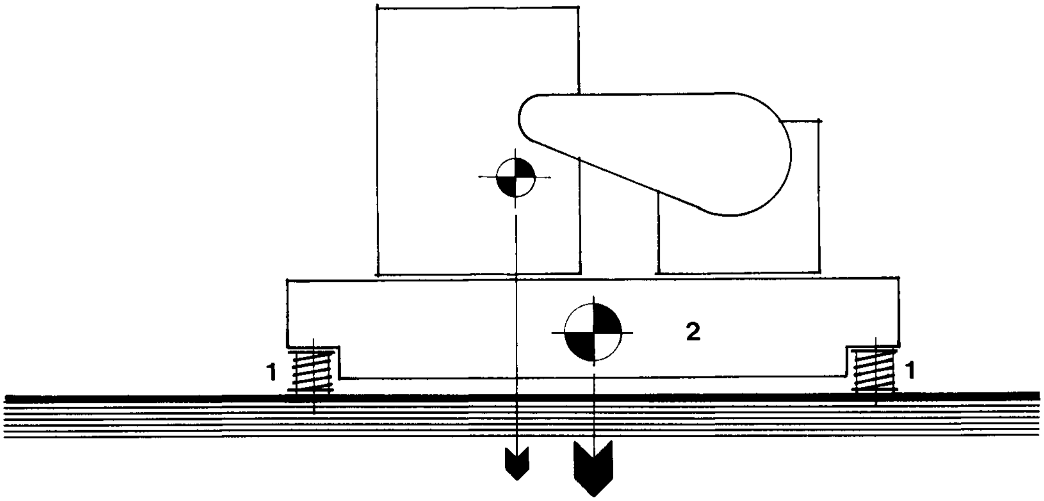
Finally, several other points should be borne in mind when using isolators.

Equality of loading

It is essential that each mount be equally loaded to reduce the possibility of rocking or pitching motion. Some ways of achieving this are shown in Figures 4.14 and 4.15. The alternative is to evaluate exactly the actual load on each mount and select mounts of different stiffnesses so that the individual deflections are equal.

Flexibility of supporting structure

If the supporting structure is flexible it will act like a spring and any forces passed by



Figures 4.14 Use of inertia block to give equal loading of all isolators.

Equal loading

1 av mountings
2 Concrete inertia block, heavy compared with machine

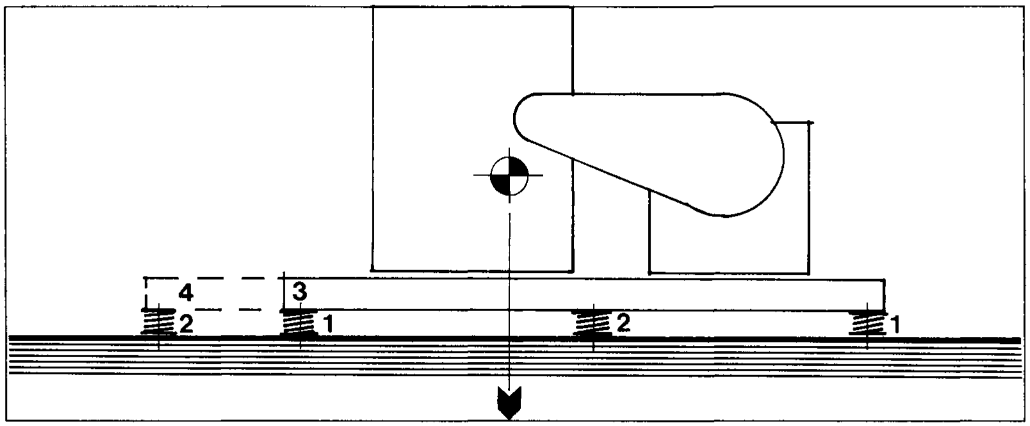


Figure 4.15 Use of modified base frame and extra isolators to ensure equal loading of all isolators.

Equal loading

1 Original av mountings
2 New av mountings
3 Original base frame
4 Extended base frame

the isolator can excite the floor into resonance if they have frequencies close to the resonant frequency of the floor. For installation on flexible structures the natural frequency of the isolator should be, at most, half the natural frequency of the floor. A building and each element of structure, for example floor or beam, will have a characteristic resonant frequency. The structure should not only be stiff enough and not deflect unduly for structural reasons, but also should avoid being easily 'driven'. Resonant frequencies should be kept well above 5Hz.

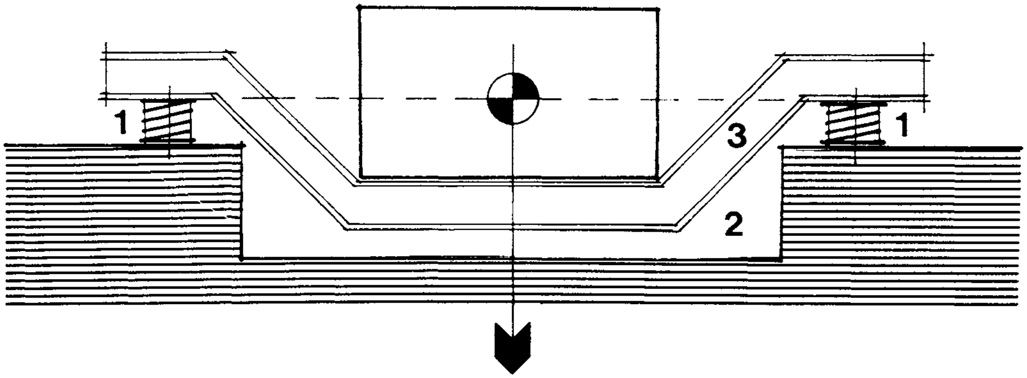
Lateral stiffness of mounts

Obviously a mount will have to provide lateral stability and this generally means that the springs will have a greater lateral than vertical stiffness. Hence, the system will have a higher lateral natural frequency and any sideways forces could cause serious resonances. If the centre of gravity of the system is above the top of the isolators, pitching motions can also result. To minimise the possibility of this, the centre of gravity should be in line with the top of the mounts. A way of achieving this is shown

Figure 4.16 Lowering of the centre of gravity to reduce rocking or pitching motions.

Low centre of gravity

1 av mounts
2 Machine pit
3 Rigid support frame



in Figure 4.16.

Bridging of mounts
Any mechanical bridging of mounts negates their usefulness. Electrical conduits should be flexibly joined to motor terminal boxes or switch-gear, water pipes joined to pumps, radiators or airhandling units should have a length of flexible hose immediately before the sprung equipment, and air ducts should be connected to the fan by means of a flexible duct connector.

Durability of mounts
The mount used must be capable of withstanding the environmental conditions in which it will be fixed.

Inertia blocks
These are usually large concrete blocks with masses much greater than the mass of the vibration source mounted on it. They are used when

- the mass of the system to be isolated is of such a value that the stiffness of the springs needed to give the required isolation would make the system too unstable to lateral movements
- several mechanical components, e.g. motor and gearbox, which need good mechanical alignment, need isolating
- pitching and rocking modes need to be eliminated. By bringing the centre of gravity of the whole system in line with the isolator mounts, the inertia block can minimise the possibility of these modes occurring
- isolation of impact sources is required.

Structural discontinuities
The extent to which energy can travel throughout a structure can be minimised by including structural breaks consisting of elastic layers or changes in cross-sectional areas. However, these tend to be frequency dependent and should be used only in conjunction with other noise control methods.

Damping

Damping really does not play a major part in the control of noise in architectural acoustics. The most common way of adding damping to an element is to add to it a layer of synthetic material which has high internal energy losses. This is most effective when the damping layer is as thick as the structure to which it is applied and when the structure is vibrating at resonance. Thus, this type of treatment would be quite impracticable for applying to masonry walls or concrete floors to minimise the effect of low frequency and coincidence resonances. However, it can be effective when applied to thin metal partitions such as found in fan casings or ventilation ducts. Often, damping around a duct is supplemented by external lagging, the damping control to duct wall excitation and the lagging to control duct noise break-out.

4.3 Control in the Sound Path

4.3.1 Barriers and enclosures

Barriers

A barrier placed between a noise source and a receiver can reduce the received noise. The amount of reduction depends upon the size of the barrier and the frequency content of the noise source. As low frequencies are diffracted or bent around the corners more easily than high frequencies, the higher the frequency the more effective the barriers.

The shielding, R_B, provided by an infinitely long barrier of finite height in the open, can be estimated using

$$R_B = 10\log_{10}(3 + 20N)$$

Where N is the Fresnel number, defined as,

$$N = \frac{2\,f\,d}{c}$$

f is the frequency and c the velocity of the sound. To estimate the attenuation in an octave or third-octave band f is taken as the band centre frequency.

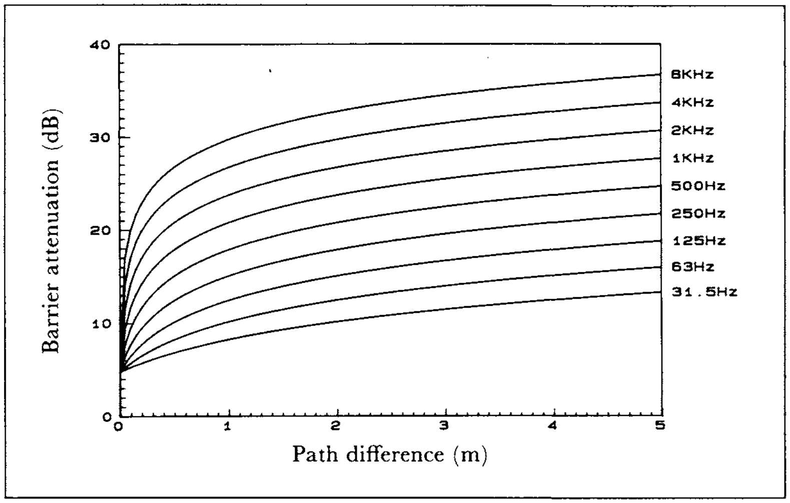
The quantity d is the difference between the distance from source to receiver over the barrier and the direct distance from source to receiver in the absence of the barrier. The equation for barrier shielding is given in graphical form in Figure 4.17.

If the barrier is not infinitely long, the sound getting around the ends must be taken into account when estimating the total shielding. The shielding around the end of the barrier is estimated in exactly the same way as the shielding over the top of the barrier and the resultant overall shielding, R_T, is given by

$$R_T = -10\log_{10}\left[10^{\frac{-R_{B1}}{10}} + 10^{\frac{-R_{B2}}{10}} + 10^{\frac{-R_{B3}}{10}}\right]$$

where R_{B1}, R_{B2}, and R_{B3} are the shieldings over the top and around the ends of the barrier. Alternatively the chart shown in Figure 3.4 can be used. In this case the attenuations must be combined two at a time.

Figure 4.17 Noise reduction by a barrier.



Buildings as shields

The barrier formula is really only applicable to thin barriers. However, for thick barriers, such as buildings, the formula will give an approximate value for shielding if an effective height and position of an equivalent thin barrier is used. These are obtained by finding the intersection of two straight lines both just grazing the top edges of the thick barrier, one drawn from the receiver and one drawn from the source, as shown in Figure 4.18.

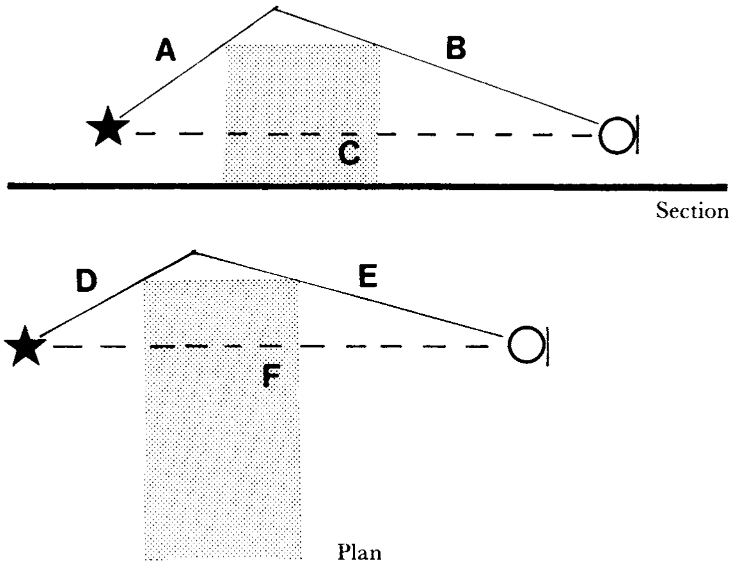


Figure 4.18 Estimation of an equivalent thin barrier.

Sound path difference $d_1 = A + B - C$ (section)
$d_2 = D + E - F$ (plan)

Example

A barrier 4 m high × 20 m long is used to screen an office window from a small extract fan situated in a wall facing the office. The window and fan are at an effective height of 2 m. Without the barrier the noise level in the 500Hz octave band is 65dB. Estimate the level with the barrier in position.

Path difference over top of barrier = 0.2 m.

Hence, using Figure 4.16, the shielding is 12dB.

Path difference around the ends of barrier is 4.65 m.

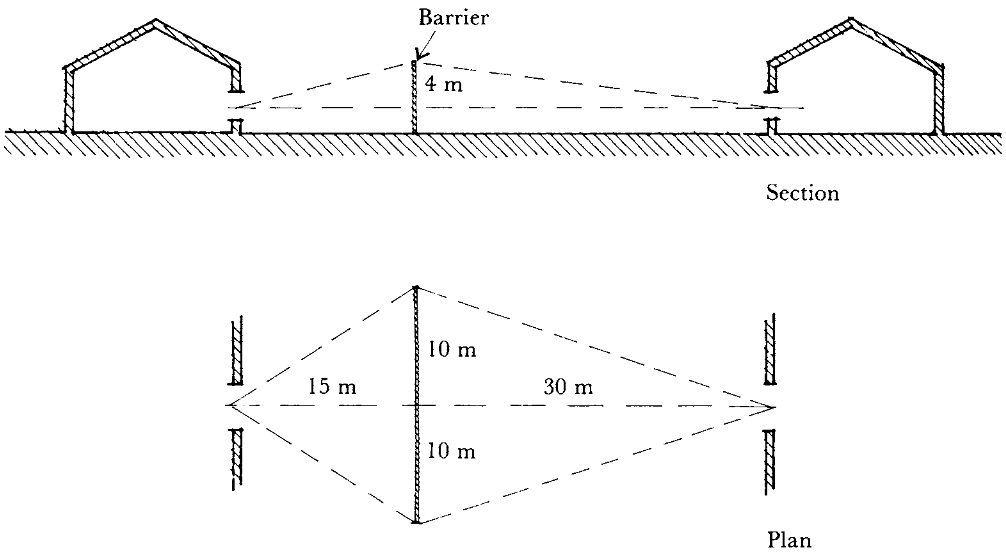
Hence, again using Figure 4.16, the shielding around each end of the barrier is 24dB.

Thus the total attenuation is found by combining the attenuations 12, 24 and 24dB.

Using Figure 3.3. Combining 24dB and 24dB gives 21dB

Combining 21dB and 12dB gives 11.5dB

Hence the level with the barrier is 65 – 11.5 = 53.5dB.



Barrier mass

For any barrier the surface mass of the material from which it is constructed must be sufficient to prevent more noise going through the barrier than over the top. As the maximum reduction is limited to about 25dB because of scattering effects, a surface mass of between 20 and 50 kgm^{-2} will usually be sufficient.

Barrier materials

Barriers for external use can be constructed from a variety of materials, for example:

- brick or concrete blocks
- screeded woodwool slabs
- cement sheets or pre-cast concrete slabs
- wood panels
- high density rigid polyethelene
- earth mounds.

In practice, the closer the barrier can be positioned to either source or receiver the more effective it is likely to be for a given height. It must be remembered that barriers will not generally absorb energy but merely redirect it and it is quite probable that noise levels on the source side of the barrier will be increased. Hence, it may be necessary in some cases to cover the side of the barrier adjacent to the source with an absorbing material.

Barriers in enclosures

Barriers or screens are also of use in enclosures although their effectiveness will be less than would be obtained in the open because of reflections from walls and ceilings which bypass the barrier. Reduction of these reflections can be obtained by placing absorbing materials on the walls and ceiling. Screens are most effective near the centre of large open areas well away from the walls, and the more complete the screening the greater will be the sound reduction. Screens should be faced with absorbent materials to reduce reflections.

There are no simple formulae or curves for determining the performance of a barrier in an enclosure. As stated above, they will not give the sound reduction predicted for the same barrier in the open and in practice one should not really expect more than 5–10dB reduction from any screen.

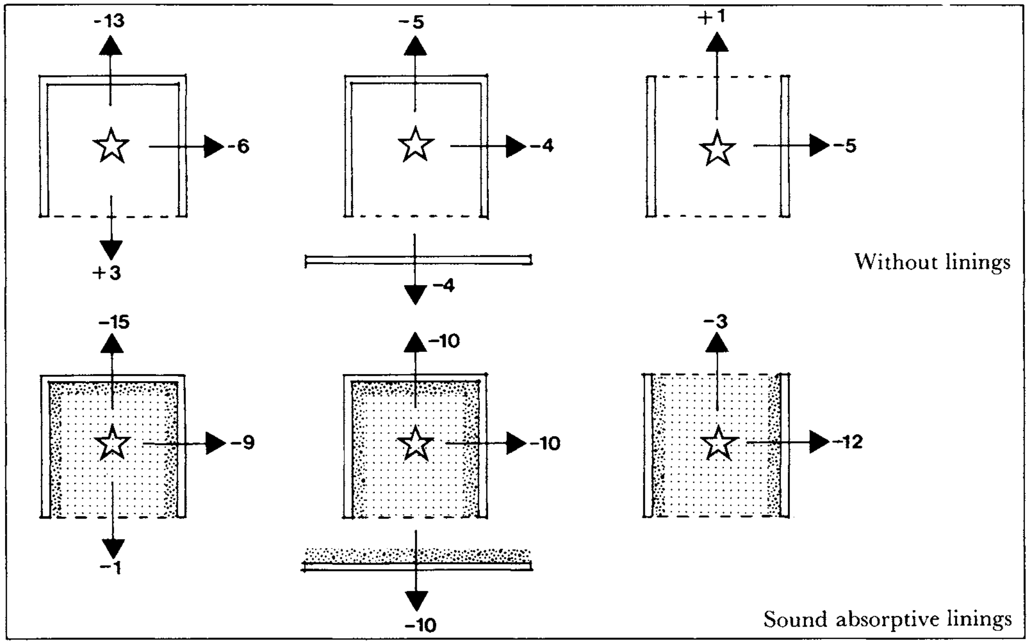
4.3.2 Partial and full enclosures

A useful reduction in noise levels can often be obtained by partially enclosing a source. Partial enclosures are often better than barriers but less effective than properly designed full enclosures.

Partial enclosure

This is used when total enclosure would be impossibly restricting for machine use, or for temporary noise sources such as might be used on construction sites. Some simple partial enclosures are illustrated in Figure 4.19. Partial enclosures can be constructed from material with a fairly small surface density (20 kgm^{-2}) but they should be covered with absorptive material on the faces adjacent to the noise source.

Figure 4.19 Examples of partial noise enclosures.



All values dB in range 500–4 kHz for near field only

Full enclosure
A complete enclosure is the logical extension to the barrier and partial enclosure type of protection. The enclosure can completely enclose either the noise source or the receiver. By careful design, reductions of up to 30dB or more can be achieved.
The insertion loss or reduction in noise level at a given location by enclosing a machine is essentially governed by the sound reduction index of the enclosure walls and the absorption within the enclosures. If no absorption is used the energy inside the enclosure builds up so that the insertion loss produced is negligible. Hence, any machine enclosure must be lined on the inside with absorptive material.

Sound haven
An alternative to enclosing the noise source is to enclose the receiver within a 'sound haven' which must also have some absorptive treatment in it to keep the reverberant sound levels down. Such an enclosure should have its own attenuated air supply.
An enclosure should not have holes or cracks through which sound may leak; lack of attention to detail can drastically reduce its effectiveness.

Materials
The traditional building materials, such as bricks and mortar, present no problems because they are so heavy. The only point to consider is that the ceiling should be as effective as the walls. Lighter constructions can easily be made to give a moderate noise reduction, and with extra care double leaf constructions with 75 or 100 mm

cavities partially filled with an absorbent can be made to give a reduction of 40dB or more. Individuals might prefer to use commercially available lightweight demountable partitions with a specified noise reduction rather than construct their own enclosures, but with either method the crucial part is the sealing of the gaps and joints. Access doors and hatches are, of course, weak points in this context.

Machine enclosures

Machine enclosures are either of the close-fitting type or they are much larger than the machine, with room for the operator as well. In the close-fitting type, care must be taken to see that there is no direct contact between the machine and the enclosure. Pipes or other services running to the machine should never make a rigid contact with the enclosure, but soft gaskets of rubber or some similar material should be provided for sealing.

Complete enclosure of a machine frequently gives rise to cooling and ventilation problems. Cooling may be achieved by means of radiators and water circulation, as in a conventional car engine, and this method is unlikely to give rise to any noise problems. Air cooling and ventilation can be more difficult because holes must be cut in the enclosure which will also permit the passage of noise. The only satisfactory solution to this problem is to fit properly designed silencers which will attenuate the noise by the same amount as the rest of the enclosure. If fans are being fitted to provide forced ventilation they should, in general, be fitted at the noisy end of the silencer to prevent the fans from becoming noise sources via ductwork.

4.3.3 Active noise control

Introduction

The concept of active noise control, that is the introduction of a secondary acoustical signal which is out of phase with the primary signal (i.e. an acoustic 'mirror image') and therefore provides cancellation, has been around for fifty years but only quite recently has any practical success been achieved. The earlier attempts to use such a system were too ambitious and this, combined with the technical limitations of the equipment, led to disappointing results and the feeling that the whole idea was somehow fallacious. More modest requirements, however, coupled with more sophisticated technology, have shown that noise reduction can be realised in certain well-defined situations. Computer controlled signals are enabling development in this field.

Methodology

There are two distinct approaches. The first is based on pre-sampling, i.e. sampling the signal prior to adding the cancellation signal. This is supposedly suitable for any type of noise. The second is based on post-sampling, i.e. sampling the cancelled signal, which ideally should be zero. This is only suitable for noises that are cyclic in nature. These methods have been applied to noise in ducts, essentially a one-dimensional problem; noise in small spaces, where the dimensions are less than a wavelength and wave motion does not really exist; and to noise from a well-defined source radiating into free space, such as a chimney or a transformer. 'Antisound' works best to cancel out air-borne low frequency noise because bass sounds produce pressure waves that change slowly enough for microprocessors to produce the 'antidote' with negligible time delay.

Pre-sampling concept

Figure 4.20 shows the basic idea underlying the pre-sampling method. The noise to be cancelled is sampled using, for example, a directional microphone; a time-delay is inserted which is equivalent to the distance between the microphone and the cancelling loudspeaker and then the signal is inverted and amplified before being used to drive the directional loudspeaker.

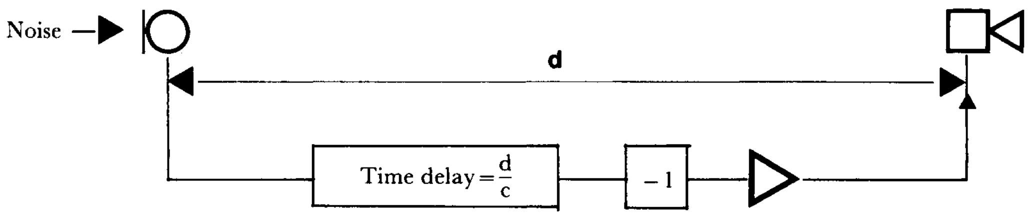


Figure 4.20 Illustration of pre-sampling active noise control concept.

Post-sampling concept

This relies on the primary noise being cyclic in nature and therefore repetitive. The microphone measures the residual sound which, ideally, should be reduced to zero. Initially there is no cancellation signal and the microphone measures the primary noise. The waveform of one complete cycle is stored in a memory, inverted, delayed to synchronise it with a later cycle and then used to drive the cancelling loudspeaker. For convenience, digital signal processing is employed. If the residual signal is non-zero then the waveform stored in the memory is corrected until the residual reaches a minimum value. If possible, a synchronisation signal is taken directly from the noise source; not difficult if the source is, say, an engine. Figure 4.21 illustrates the idea.

The practical use of anti-noise systems depends largely upon the electrical control systems which must be capable of rapidly reading the noise. A 'learning network' enables prediction and hence 'antidote' action by analysis of the past pattern, checking whether the noise was properly abated as it goes along.

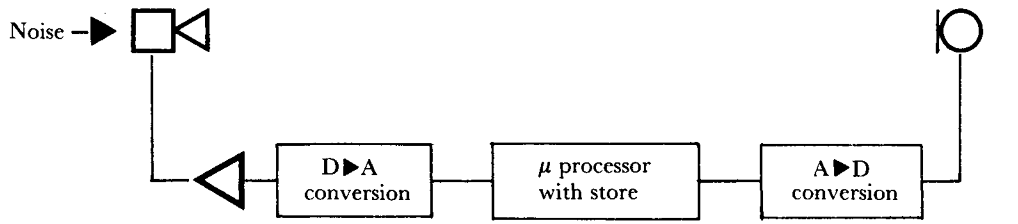


Figure 4.21 Illustration of post-sampling active noise control concept.

Duct noise

Figure 4.22 shows the direct application of the pre-sampling technique to duct noise. The working range is restricted to frequencies at which the duct cross-sectional dimensions are small compared with the wavelength of sound. The pair of loudspeakers act as a uni-directional source and are used to provide a cancellation signal equal and opposite to the primary noise signal. In this case the loudspeakers 'absorb' the noise coming along the duct.

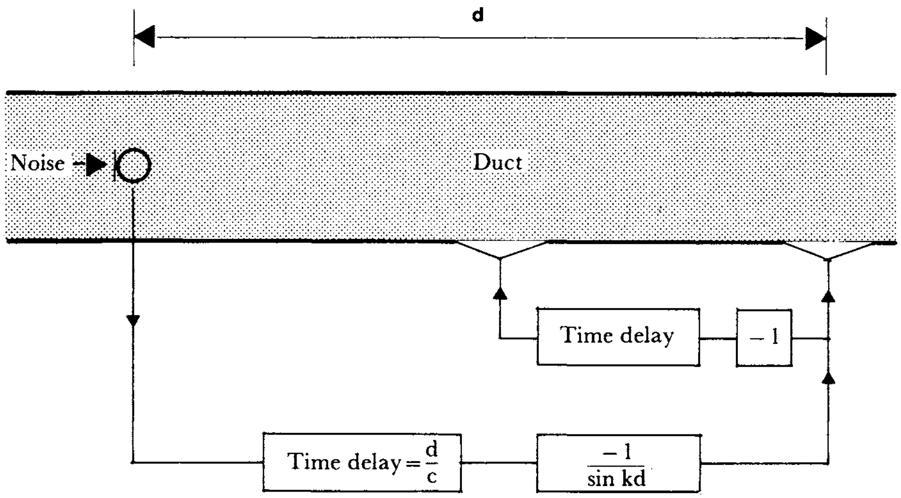


Figure 4.22 Illustration of duct noise cancellation using the pre-sampling concept and a uni-directional loudspeaker cancelling system.

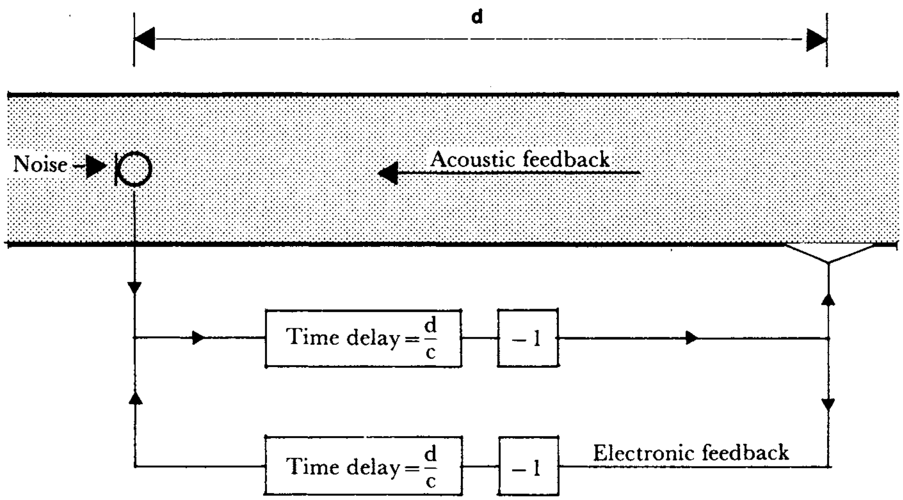


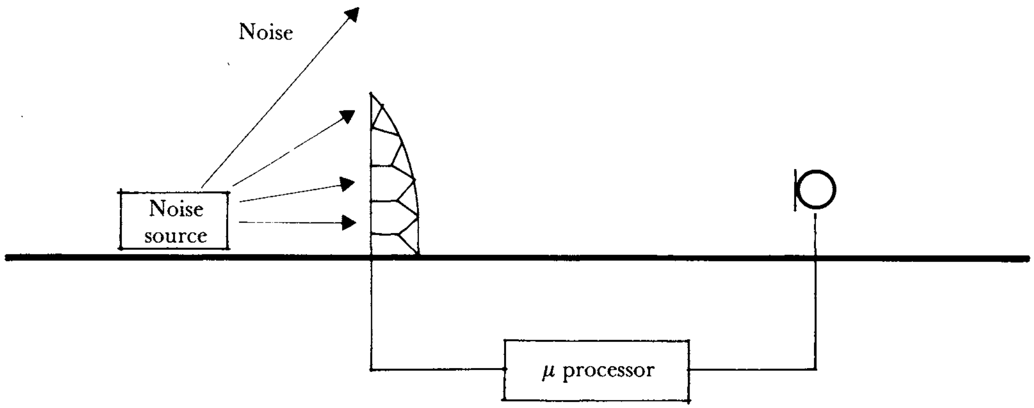
Figure 4.23 Illustration of duct noise cancellation using the pre-sampling concept and an omni-directional loudspeaker cancelling system.

Figure 4.23 shows an alternative application of pre-sampling, in which a single loudspeaker is used for stopping the noise. In this case the loudspeaker creates a plane of zero impedance in its immediate vicinity and reflects the noise back to the primary source, in just the same way as a quarter wave tube or Helmholtz resonator sidebranch operates, although the loudspeaker works over a much greater frequency range. The microphone is now subjected to a reflected wave as well as an incident sound wave. If the microphone were truly uni-directional this would not matter, but it has been found in practice that some acoustic feedback is picked up by the microphone. This cannot be tolerated, so an omni-directional microphone is used in conjunction with negative electronic feedback which exactly cancels the positive acoustic feedback.

Transformer noise

Figure 4.24 shows a transformer with a number of directional loudspeakers located on a constant phase surface not too far from the source. The source does not have to be surrounded, although diffraction will occur at the edges of the cancelling area. The loudspeaker matrix acts rather like an absorbent barrier. The post-sampling technique is illustrated in the figure, but pre-sampling could be used just as easily.

Figure 4.24 Illustration of the use of active noise control in the open.



4.4 Services Noise

Building services equipment as a noise source has become the subject of growing concern within the last twenty-five years, because of the more widespread use of air-conditioning and airhandling units, the move towards more compact, space-efficient buildings using lightweight construction methods, and because of more exacting environmental standards. An example of the latter is the statutory minimum temperature permitted for workers in office premises.

Noise sources

Building services noise can result from items of plant, for example boilers, pumps, fans, compressors, refrigeration plant and cooling towers, or from distribution of

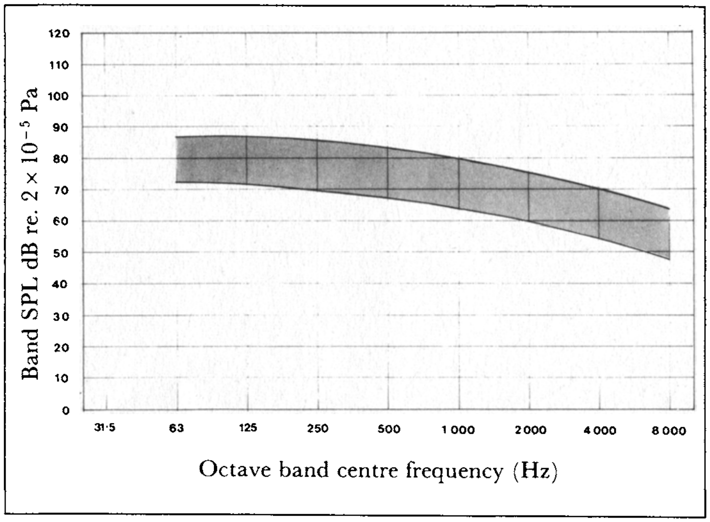
miscellaneous systems – lift motors, water pipework, and heating systems.
Mechanical engineers have differing views on how to heat and ventilate large, complex buildings. Sometimes, central plant will be favoured, with large airhandling units and boilers grouped in a plant room and pipes running out in vertical service shafts and along suspended ceiling voids. For other schemes, decentralised plant 'packages' may be preferred for flexibility and reduced ductwork and pipework, plant being distributed in various locations near the spaces served. Either approach can be made to work for most situations, but may involve a different noise control strategy.

4.4.1 Primary plant

Primary plant, such as boilers and refrigeration plant, is likely to be centralised, often in a ground floor or basement location. Pumps and compressors within a plant room containing one or several boilers can produce a high reverberant noise level. In order to give some idea of the levels produced by particular items of plant, empirical formulae are sometimes used. A check of such formulae published shows a wide range of predicted values, even given the same duty data. Care should therefore be taken over such predictions. While noise control hardware for mechanical systems may only amount to 2–5 per cent of services cost, a retrofit rather than correct initial design may entail a cost many times greater because of the complications involved.

Boilers
The sound power level produced by boilers depends to some degree on the type of burner rather than the fuel source. Additional noise within the boiler house may be generated by combustion air fans and their drive motors. Ancillary equipment associated with coal-fired boilers, particularly pneumatic coal transport systems, can be particularly noisy.

Figure 4.25 Representative octave band reverberant sound pressure levels in a boiler room.



While the dB(A) level is of some help in assessing the noise control measures required, octave band spectra (63Hz–8kHz) should be used for design calculation checked against NR curves. This is because there may be pure tones and other aggravating characteristics to the noise. A typical range of boiler room noise spectra is shown in Figure 4.25.

Flues

The prediction of sound level in a chimney is even more empirical than for boiler noise prediction, as further variables apply.

The noise level at a point at some distance from the flue outlet depends on the flue's height, any directivity factors, its cross-sectional area, and any lining absorption present. Calculation is a complex matter, beyond the scope of basic prediction formulae. Noise control measures include special linings inside flues, or attenuated discharge openings.

Cooling towers

This item of plant is frequently placed on roof tops or together with other plant in a central location, radiating noise external to the building. Often a number of units are involved. The noise results primarily from the tower fan, although water noise can be a significant factor in towers. Noise levels can be reduced by using lower speed fans, by the use of inlet and discharge attenuators, or by screening. A typical noise spectrum is shown in Figure 4.26.

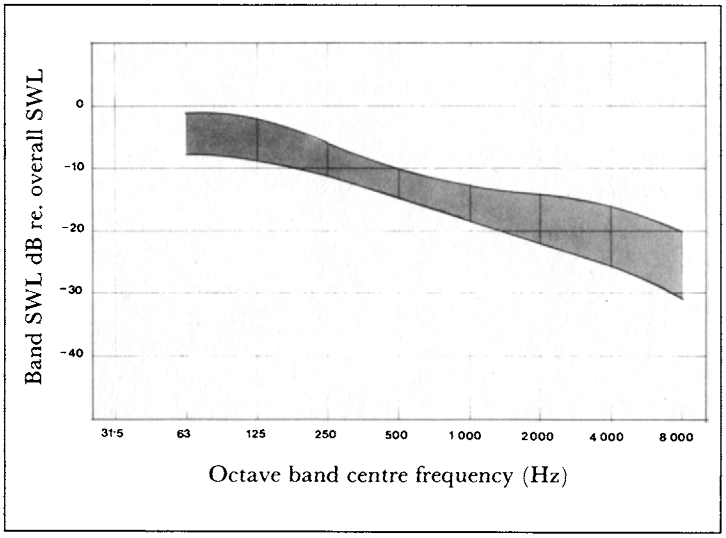


Figure 4.26 Representative octave band sound power levels of cooling towers.

Pumps

In general the rating of the pump determines the sound power level.

4.4.2 Ventilation systems

Modern ventilation systems entail movements of large quantities of air through compact ductwork, activated by fan. In order to keep duct runs modest, airhandling plant may be located in one of the following locations:

- within space. Room units can only be acceptable for small-duty requirements in tolerant spaces, NR 35 or more
- roof. Roof plant should not be mounted in mid span of large spaces below, otherwise even low transmissibility anti-vibration mounts will not prevent the support structure being 'driven'. Break out of fan noise at high level may upset neighbouring properties
- separate plant room. A remote location may be desirable for low noise installations.

A typical ventilation or air-conditioning system comprises supply and extract fans, connected to diffusers or grilles in the rooms served by a system of sheet metal ductwork. The supply fan is usually packaged with heating and cooling coils and filters to form an 'airhandling unit'. The ductwork incorporates dampers to regulate airflow, fire shut-off dampers, and attenuators to reduce noise transmission. Sometimes, secondary heater batteries and automatic volume regulatory devices (terminal units) are fitted in ductwork branches to particular rooms. All the components of the system can provide some reduction of noise from the fans reaching the rooms served, but each will also generate noise in the presence of air flow. Analysis of noise from ventilation systems requires the noise attenuation and generation characteristics of each element to be determined. The main noise sources and possible control components in a ventilation system are shown in Figure 4.27. The main items will be considered in turn.

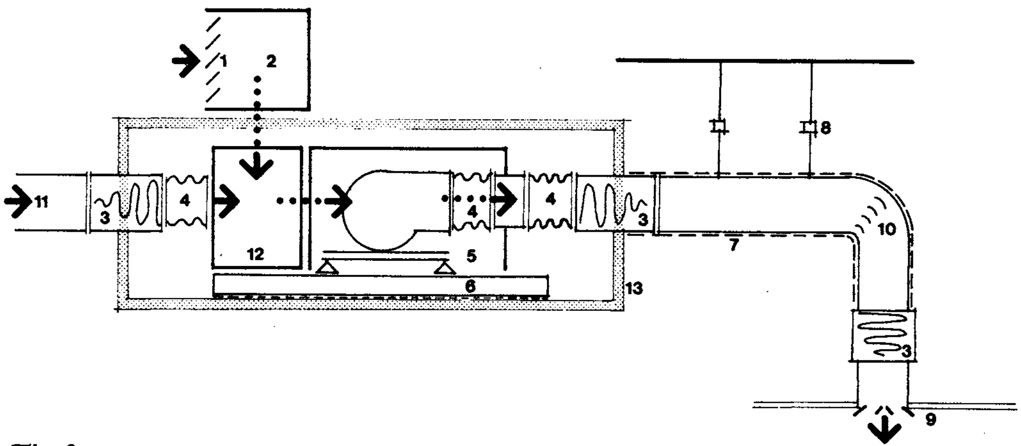


Figure 4.27 Noise sources and control measures in a typical ventilation system.

1 Acoustic louvre to fresh air inlet
2 Plenum chamber
3 Silencers
4 Flexible connections
5 Fan chamber (may be part of a package air handling unit – casing, fan, heater, batteries, dampers, av mounts to chassis)
6 Inertia slab
7 Double skin ductwork (where break in or break out of in-duct noise is a problem)
8 Resilient ductwork hangers
9 Aerodynamic air grille terminal
10 Air turns to ductwork bends
11 Return air
12 Mixing chamber
13 Plant room enclosure

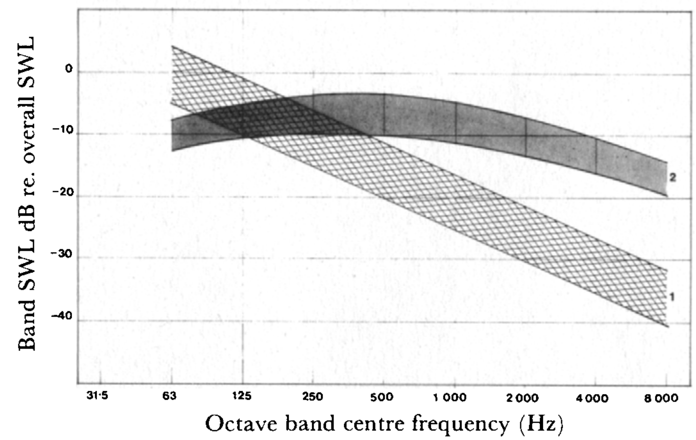
The fan

In low pressure, low velocity systems, the fan is normally the significant noise source. In badly designed systems, excessive noise may also be generated by turbulent air flow through duct fittings because of incorrect sizing of ductwork, grilles or diffusers or poor duct layout. High pressure, high velocity systems cause more complex noise problems. Airflow generated noise and noise breakout from duct walls and from the casings of terminal units above suspended ceilings are persistent problems in such systems.

The parameters affecting fan noise output are the power (kW), airflow speed ($m^3\,sec^{-1}$), fan static pressure ($N\,m^{-2}$), speed of fan rotation, blade diameter, number of blades, backward or forward curve of blades, and the fan type – axial or centrifugal.

Characteristic noise spectra for axial and centrifugal fans are shown in Figure 4.28.

Figure 4.28 Representative octave band sound power levels for centrifugal and axial fans.



Ductwork

Air in a ventilation system is distributed via rectangular, round or oval thin-walled metal ducts, usually in a ceiling void. Round ductwork is sometimes preferred for low noise systems because the airflow is smoother and the greater rigidity of the duct reduces the break-out of duct-borne noise. The flat walls of rectangular ducts are prone to excitation by airflow and this can result in radiated noise from the duct to the space served. On the other hand, noise control measures can more conveniently be applied to rectangular duct systems – internal linings, air turns within radiused bends, and silencers with many rectangular absorber slab splitters inside.

In very low noise, low velocity situations (studios and more critical auditoria) 'builder's work ducts' may be used because the cross-sectional area is so great that masonry or dry partitioning are more practical and cost effective for the duct walls rather than sheet metal.

An estimate of the sound power level of noise escaping from a metal duct can be obtained using:

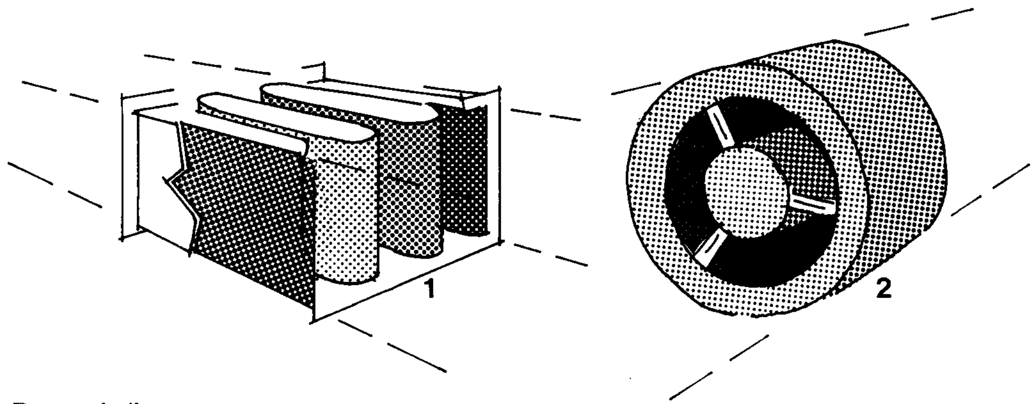
$$SWL = SWL_I - R + 10 \log \frac{S_w}{S_D} \qquad dB$$

where SWL_I is the average sound power level of the noise inside the duct, R is the sound reduction index of the duct wall, S_w is the total surface area of ducting and S_D is the cross-sectional area of the duct.

The silencer

The commercial attenuator or silencer shown in Figure 4.29 has become the most common air-borne sound control item. It is commonly a box set in a length of ductwork, or where the ductwork penetrates a plant room wall, with acoustically absorbent slabs set within. These are spaced around airways and absorb the ductwork fan noise as the air travels through the silencer. The variables of splitter thickness and airway width will determine the frequency range where silencing will be most effective and the silencer length will govern the degree of silencing, often referred to as the 'insertion loss' due to the silencer's presence in the duct.

Figure 4.29 Examples of absorptive silencers.



Ductwork silencers

1 Rectangular
2 Circular

The absorbent material with splitters has to be covered in perforated metal or other containment to prevent erosion by airflow and resultant contamination of the air supplied. Larger silencers with many splitters give better results, with mid frequency attenuation of duct-borne noise by as much as 50dB.

Use of computers

Mini-computer programs have come to be used for analysis of basic ventilation noise data, allowing noise control specialists to offer cost effective solutions to silencing. Once the details of fan, duct runs and cross-section, velocities, room conditions and other parameters are fed in, the program produces a choice of silencers to achieve target figures. Some silencers cost much more than others, so it is important to identify economic sizes and sections for the task.

Duct lagging and lining

To prevent noise break-out, ducts can be lagged externally using proprietary materials developed for the purpose. The material may be a dense board, which also serves to damp the walls of rectangular ducts from being excited by the airflow, or a sandwich of mineral wool with lead foil interlayer which can be wrapped around rectangular or round ductwork. Double-skinned ductwork can also be assembled, i.e. the duct walls consist of outer and inner metal sheet layers separated by a quilt of glass fibre or mineral wool. Uprating of duct walls can also serve to prevent break-in of noise into the duct path on the room side of attenuators.

Attenuation of duct-borne fan noise can be achieved by internal linings to ductwork, an approach particularly favoured in the USA. Limited attenuation of 0.05–5dB/m, dependent on the duct's cross-sectional area and whether the duct is circular or rectangular, occurs in unlined ducts. This increases at mid frequencies to 3–13dB/m if an internal lining of, say, 100 mm mineral wool is used. Linings are less effective for large section ductwork and should be added to the system to supplement, not replace, commercial attenuators for low-noise applications.

Bends in ducts

Duct-borne noise can be attenuated at bends – the mid-frequency reduction can be up to 8dB, or as little as 0dB for each bend. Duct lining around the bend increases this range to 2–10dB. Sound will be attenuated better at bends in large rather than small cross-sectional area ductwork because the short wavelength energy will not be guided by the duct sides and impinge on the duct wall at the bend, suffering attenuation by reflection.

Radiused bends with turning vanes rather than right angle or T-junctions will avoid undue regenerated noise through air turbulence. The air vanes smooth and guide the air flow around the bend.

Branches also have an attenuating effect, which can be estimated from the expression:

Attenuation of each branch =

$$10 \log_{10}\left[\frac{\text{volume flow of air in branch}}{\text{total volume flow of air up to branch}}\right]\text{dB}$$

Terminals

Duct terminals within rooms produce noise. Diffusers are noisier than grilles, particularly linear diffusers. The roomside sound power will depend on the open area and velocity as well as grille pattern.

Terminal noise adds to the duct-borne noise. The noise level within a room caused by duct-borne noise entering via a supply or extract grille can be a problem at near field locations, due to direct sound, or through its contribution along with other grilles to raising the reverberant level at the room centre. Compared to the in-duct sound power level, the noise entering the room is attenuated by 0–6dB due to the fact that not all the duct noise finds its way out into the room. A small terminal has greater attenuation but this tends to be cancelled out by the regenerated airflow rate through the restricted openings.

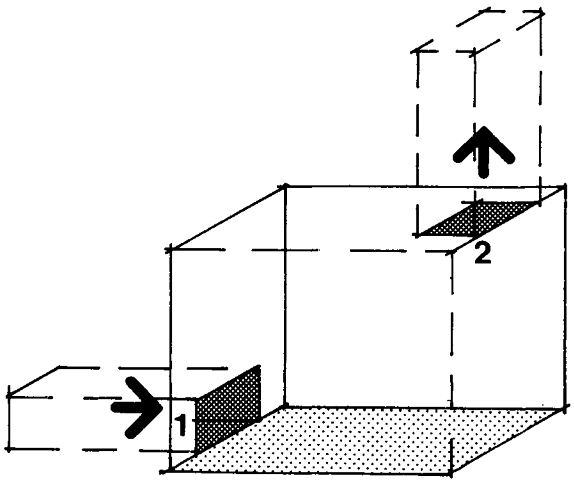
Plenum chamber

A plenum chamber consists of a box lined with fibrous sound absorptive material. The cross-section is much greater than the duct, and entry and exit points into the chamber should be offset, as shown in Figure 4.30. Plenum chambers are useful where acoustic louvres alone will not adequately reduce plant room noise break-out at openings. The approximate reduction in sound power level due to the plenum chamber, SWL_R, is given by:

$$SWL_R = 10 \log S_0 \left[\frac{\cos\theta}{2\pi d^2} + \frac{1-\alpha}{S\alpha} \right] \text{ dB}$$

where S_0 is the outlet area, d is the distance from inlet centre to outlet centre, θ is the angle the line d makes with the inlet axis, S is the total internal surface area of the plenum which has an average random incidence absorption coefficient $\bar{\alpha}$.

Figure 4.30 Layout of a plenum chamber.

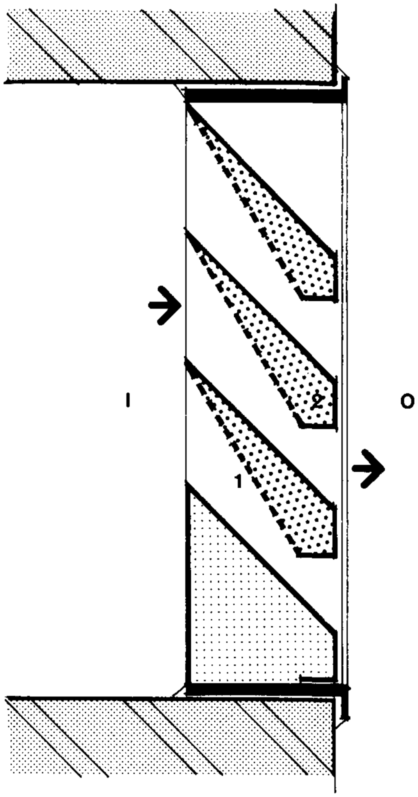


1 Air inlet
2 Offset air outlet

Acoustic louvres

Outside openings to plenum chambers, fresh air supply ducts and boiler or plant rooms are usually faced by louvre blades angled to keep out the weather but offering a large percentage of free area for air to circulate. If some noise control (either break-out or break-in) is required, acoustic louvres can be used. These will provide absorption on the blades, underside (Figure 4.31) and so act as low grade silencers. The insertion loss possible is restricted to about 15dB at 500 Hz for a 300 mm deep louvre. Low frequency attenuation is limited – even a double-banked 600 mm deep acoustic louvre will give less than 10dB at 63Hz.

Figure 4.31 An acoustic louvre.



1 Perforated rear face
2 Mineral wool fill to louvre blades

Noise control standards

The implications of achieving a particular noise rating standard of noise caused by ventilation systems can be seen in Figure 2.8 or Table 2.5. The method of use of the curves is described in section 2.2.1. In many installations, ventilation noise can be allowed in moderation: for example, in offices (NR 35–40) ducts are of modest size, velocities relatively high and only single-stage attenuation is included. For more critical spaces, like auditoria and studios, large-section ductwork, two- or even three-stage attenuation, and other measures, will all be required.

If a standard is exceeded in practice, it may be difficult to pin down the cause. Some general causes can, however, be noted:

- low frequency noise excess – lack of basic silencing to fan noise
- broad-band noise excess – the fan manufacturers' fan noise spectra will have been measured in more ideal conditions, showing the product in the best light. The fan as installed will usually be noisier to a lesser or greater degree
- mid and high frequency excess – regenerated airflow noise. The system may be delivering over volume or the terminal dampers may be set too closed.

Design

5.1 Building Types

Different buildings demand different acoustical conditions within, and greater or less control of intrusive noise. Some notes for the various building types are set out in this chapter; reference to BS 8233 (Code of Practice for Sound Insulation and Noise Reduction for Buildings), shortly to replace withdrawn CP3: Ch.111: 1972, will be useful for outline guidance. In the case of multi-use buildings or a novel combination of different facilities in a building complex, particular care is required so that conditions for one setting do not compromise another.

The most exacting standards of internal acoustics and sound insulation apply for studios and auditoria. The low occupancy, small room size and cellular planning in studios make it practical to introduce special construction techniques like multi-leaf walls, structural discontinuity, and absorber panel linings. Auditoria present a different challenge, with their high occupancy, large volume spaces, more complex circulation patterns and the requirement of low noise, large-duty ventilation systems.

Public and civic buildings also need good acoustics but will have other design considerations besides. For example, in hospital design hygiene and medical efficiency take priority over acoustics. Nevertheless, privacy in consulting rooms, conditions recognising the importance of rest and sleep, and noise control of complex engineering installations are essential design ingredients.

Offices and commercial buildings appear at first sight to offer little acoustic design challenge. However, they generate a disproportionately large number of complaints about noise problems. These result often from the misapplication of open planning principles, noisy office equipment, and the frequent internal rearrangements that occur in any changing organisation.

Industrial buildings have diverse problems, including the creating of noise nuisance from lightweight buildings with many openings (for example, delivery doors and process plant extracts), and over-reverberant internal conditions.

5.1.1 Churches

Church buildings have been put to a wider variety of uses in recent years. New buildings may involve communal and educational as well as ecclesiastical accommodation.

Traditionally, the congregation was little involved in the church service, but now there is greater participation in the Eucharist or Mass. This has led to reorganisation in existing churches, or greater design attention to matters such as speech articulation in new ones. Smaller churches have better speech intelligibility characteristics than larger, allowing a more intimate and comprehensible church service.

Roman Catholic churches traditionally require greater volume per person than Anglican or Methodist churches or Friends' Meeting Houses. Loudspeaker systems comprising directional column speakers along the aisles can alleviate poor natural acoustics in large old buildings. If sound absorptive finishes can be incorporated, they are best located at the back of the church.

5.1.2 Cinemas

Modern cinema auditoria do not have the same problems of scale as older centres. Current practice is to plan low capacity studio cinemas which are obtained by the sub-division of large older ones or are purpose built spaces, having 6, 8 or even 10 screens, each holding 200–250 people.

The natural acoustics of the auditorium itself should not interfere with the soundtrack effects, so reverberation times should be kept short – usually less than a second. When a multi-use hall is also used for showing films, the need for soundtrack clarity demands reduction of reverberation time by 25 per cent. Adjustment of graphic equaliser controls in the sound system can compensate for some inevitable lengthening of reverberation time at bass frequencies.

The width of studio cinemas is influenced by the seating layout restrictions. Using the maximum fourteen seats per row and side aisles, a working width of 10 m is obtained. Seats should be plush to provide both comfort and sound absorption. The narrow configuration of room arising from the central seating block demands the corrective addition of sound absorptive, side wall finishes to prevent cross reflections, particularly if side loudspeakers supplement the screen array. Concealed sound absorption behind the screen is one way of increasing the total absorption, besides carpet to floor and acoustic tile ceiling.

As speech levels delivered over film soundtrack are high compared with sound levels in, say, live theatres, a slightly lower ambient noise control standard is usually adequate. A good standard of isolation to outside, or to adjacent cinemas in a multi-screen complex, is essential. In addition, entrance sound lobbies and thick glass to projection ports are recommended.

Decentralised airhandling units serving individual studio cinemas in a multi-screen arrangement is the standard practice in the USA. The units should not be set directly on the auditorium roof structure, otherwise it will be difficult to control vibration. Supply and exhaust terminals are frequently overhead, connected to ductwork in the suspended ceiling.

5.1.3 Concert halls

Concert halls are generally sited in urban centres and could therefore be subject to high external noise levels from traffic and adjacent buildings. The performance areas should be buffered by auxiliary accommodation and have substantial roof construction, often multi-leaf, to give isolation to outside of 50–55dB average.

The shape of traditional halls is rectangular, with a necked stage area to house the orchestra. The ratio of length to width should be between 2:1 and 1.2:1. The ratio of height to width should be similarly off square, width dimension to exceed height. A rectangular format ensures that the audience is grouped in front of the orchestra, receiving well-blended sound enhanced by strong diffused sound reflections off side walls. The volume of the hall is advisedly 8–9 m^3 per occupant, although many existing well-loved concert halls have very different values.

The reverberation time should be between 1.5 and 2.5 sec, with the longer reverberation times suited to larger halls. Values should not vary greatly through the frequency range, although an increase of up to 50 per cent in the lower frequencies (below 250Hz) is considered desirable for response to bass notes. Recent design developments have placed emphasis on the pattern of sound decay in different parts of the hall, rather than just the basic reverberation time characteristics.

The number of seats in a hall has to be related to the size of orchestra. A chamber orchestra of, say, 30–40 musicians suits a modest hall holding 700–1200 people. A full orchestra of ninety would be overpowering in a small hall and, in any case, needs a large audience for economic return on performance costs.

Flat seating at the front of a hall should be backed by raked rows, so that a 100 mm viewline difference is maintained if the eyeline of successive rows to the stage is plotted. Theoretically this would entail a gradually increasing rake away from the front. For most halls, the seating should be plush so that an empty seat has similar sound absorption properties to an occupied seat. By this means, an empty hall does not sound dramatically different from a filled hall.

Generally, finishes within a hall should be sound reflecting, for example hard wall plaster, but mounted on sound diffusing surfaces. To be effective in diffusing incident sound throughout the frequency range, projecting wall elements need to be at least 2000 × 2000 × 300 mm.

A noise control standard for a new hall where professional musicians may be recorded during concerts may be an exacting NR 15, while a realistic standard for converted existing spaces or for use by amateur groups may be NR 25. The ventilation plant has to cope with the presence of large numbers in the audience, without discernible noise or draughts. Plant rooms should be remotely located, lined plenum chambers should be used as part of air supply and extract systems (i.e. 'builder's work ducts'), primary and secondary attenuation incorporated, and a self-regulating air distribution system used. Large section ducts running within the acoustic shell of a concert hall are to be avoided as the hall's internal acoustics are affected. Large slow-running fans are necessary to move the air.

5.1.4 Conference rooms and lecture theatres

Conference halls are for group discussion, and this is the first priority in designing their acoustics. Lecture theatres place greater emphasis on the individual speaker. For both facilities the recommended ambient noise control standard is NR 35 for small rooms of 10 or so, NR 30 for rooms holding 20, and NR 25 for large auditoria for 50 or more.

In large meeting rooms, a raked floor and dais are required, with the reverberation time controlled to about 0.75 sec. The ceiling should be kept low and be sound reflective to help speech carry from the front to the rear, or 'across the table' for a conference or banquet setting. Carpeted floors and sound absorptive wall finishes towards the rear (i.e. to the opposite end from the dais), are also recommended. Where the distance between the speaker and the farthest listener approaches or exceeds 10 m, a speech reinforcement sound system may be used.

5.1.5 Council chambers and courtrooms

Good acoustics in council chambers and courtrooms are essential for a clear understanding of speech. As the speaker may be anywhere in the room there is a limit to the chamber size for speech from one corner to be audible in the opposite corner. The increase in the size of councils has frequently meant the introduction of electro-acoustics to larger chambers, often as a result of introducing tape recording facilities for proceedings.

Recent council chambers and courts have been built to specification 1.5–2.5 m^3/occupant and from 70 to 200 occupants. The shape varies, but is often near square with irregular wall profiles which help in diffusing reflected sound.

Courts in particular require privacy for hearings, and judges have been known to adjourn cases if they find conditions unsuitable. The courtroom shell should be insulated to 50dB average or more, with sound lobby entrances and segregated planning to different ancillary accommodation for judge, jury and public. Ceilings are typically 4–5 m high with sound reflective angled surfaces to project speech. Some sound absorption to surfaces near the bench will help keep confidentiality in that area.

5.1.6 Hospitals

Noise control is the principal consideration in hospitals and in the UK DHSS procedures should be followed. These are defined in Hospital Design Note 4 'Noise Control' (1966 amended by HN(76)126) and 'Hospital ENT Services–A Design Guide' (1974), both obtainable from HMSO. Ambient noise control standards vary from NR 25, for 'quiet' wards, operating theatres and nurses' residences, to NR 40 for kitchens, dayrooms and treatment rooms. Greater noise levels, e.g., NR 55, may have to be accepted for specially controlled areas, for example, laminar flow rooms.

Overhearing by flanking transmission of sound over partitions is a frequent complaint, for example between waiting areas and consultants' offices. At one newly built hospital, applause across an entire department greets successful births in the delivery rooms.

Hygienic, hard-wearing surfaces tend to give hard internal acoustics. However, some recent suspended ceiling types can be hosed down and yet retain reasonable sound absorptive qualities. Cushioned vinyl or carpet floor finishes will damp footstep noise. Many health buildings include audiology suites. These comprise small rooms where hearing is tested and hearing-aids fitted. The rooms must be well isolated from air-borne and structural noise, have ambient noise control standard NR 25, and be tested to reverberation time within in the order of 0.2–0.25 sec. Such rooms are available as a 'package' from specialist suppliers.

5.1.7 Housing

Privacy at home is the basic right of everybody, at least in theory. The recommended ambient noise control levels are NR 25 in bedrooms and NR 35 in living rooms, kitchens, etc. In fact many houses are so close to main roads that, unless dual glazing is used, higher internal NR values are experienced. Dual glazing may improve internal conditions but will be of negligible benefit in summer when windows are opened.
For proposed residential developments, care should be taken that sites experience at present, or even, say, in ten or fifteen years, long-term average levels less than 65 dB. London guidelines (*GLC Guidelines for Environmental Noise and Vibration*, 1976) have referred to recommended maximum average levels for parks and gardens, of 55 dB.
For existing residences, grants have been made available (Noise Insulation Regulations 1975) in situations where new roads are constructed or existing roads widened, and residents suffer from road traffic noise of 68 dBA or more (L10 18-hour average value). Local authorities can set up noise abatement zones (empowered by the Control of Pollution Act 1974). If there is a loss of amenity to residents through a new noise source, compensation may be due under the Land Compensation Act 1973.
In England, Part E of Schedule I of the Building Regulations 1985 require party walls and floors separating dwellings to have reasonable resistance to air-borne sound. In Scotland, the Building Standards (Scotland) Regulations (1981) have mandatory standards for party walls and floors. In Northern Ireland, the Building Regulations (Northern Ireland) 1977 (as amended) apply.
Partitions within houses as well as party walls deserve attention; those around WCs should have an average sound insulation of 38dB or more. Reference to CIRIA Report 114, 1986, 'Sound Control for Homes', is recommended for detailed design.

5.1.8 Libraries, museums and art galleries

Libraries require reasonable conditions, particularly in reference areas which tend to be 10–15dB quieter than check-out or popular fiction areas. Intrusive noise should be controlled to within NR 30. Sound absorptive finishes and quiet floor finishes are advisable.
Museums and art galleries should be in settings that allow visitors to discuss exhibits at reasonable speech levels without disturbing others. Designing in a variety of aural as well as visual environments will add interest and variety.

5.1.9 Offices

It is arguable that the main cause of acoustic difficulty and staff complaint is lack of privacy at the place of work. People requiring a high degree of confidentiality need cellular offices, even within an overall open plan arrangement.

Light partition assemblies are frequently used to divide a large floor space into separate offices. Proprietary systems are typically metal or ply skinned panels, 50 mm or so thick, with mineral wool filling. The mass is typically less than 40 kgm^{-2} and in a typical installation 30–35dB may be expected. This provision may be compared with the implication of different degrees of separation (see Tables 5.1, 5.2).

Some well-assembled dry partitioning can be uprated to give 40–45dB which will ensure that conversations in the next office are unintelligible if not inaudible. Table 5.2 shows some specifications for plasterboard-panelled partitioning. Ceiling void flanking paths will downrate the partitions' performance unless the partition is carried up through the suspended ceiling to floor soffit above. Alternatively, a jointless ceiling of mass greater than 10 kgm^{-2} may be used.

Table 5.1 Speech privacy

Whether conversation overheard other side of division	SRI of dividing element dB
Normal speech easily overheard	20
Loud speech clearly heard	25
Loud speech distinguished under normal circumstances	30
Loud speech heard but not intelligible	35
Loud speech can be heard faintly but not understood	40
Loud speech or shouting heard with great difficulty	45

Table 5.2 Sound insulation from typical partitions

Specification	Typical SRI (average 100–3150Hz) dB
13 mm plasterboard either side of 48 mm metal studs	34
As above with 25 mm glass fibre quilt within studding	41
Two layers 13 mm plasterboard either side of 48 mm metal studs	43
As above with 25 mm glass fibre quilt within studding	48

Open planning

Open planning should be used only for layouts where great privacy is not required between workstations. Folding or sliding partitions may provide only 10–25dB average separation, and space divider screens contribute only about 5dB attentuation between adjacent workstations.

Some distance between workstations (2.5–3 m) and sound absorbing surfaces as well as screening can give at best modest separation. Overcrowding in some facilities (for example, dealing rooms at 5–6 m^2/workstation) makes for tiring conditions, as speech or telephone conversation is automatically raised to compensate for intrusive background noise from adjacent workplaces, once general levels are about 57dBA. At least 12 m^2 per workstation and immediate circulation is advisable, with screening of local problematic noise sources like Xerox machines, coffee machines and printers.

In the relocation of personnel from traditional premises consisting of small cellular offices to a large open plan space, suitable new screen-based office furniture must be introduced as the transposition of plain desks, metal filing cabinets and storage cupboards into any old open area is a recipe for disaster.

Perimeters are distant to most workstations in large open plan offices, so the local absorbing surfaces are all important in controlling acoustic conditions. Acoustic screens should be solid cored and at least 1.5 m high, with absorbing surfaces either side. Acoustic ceilings should be at height less than 3 m, with absorption coefficient characteristics of average value at least 0.8 for the octave band centre frequencies important to speech: 500, 2000 and 4000Hz.

5.1.10 Public houses and clubs

Pubs are often the cause of complaints by local residents, because of the slamming of car doors and the starting up of engines at closing time by patrons. Little can be done except to screen the car park. The use of function rooms within pubs for live music or discotheque sessions adds to the disturbance – groups can produce 100–110dBA compared with taped music played at 75–90dBA. Adequate isolation from other premises is essential, but even so the low frequency beat component will carry over long distances.

For a relaxed atmosphere in restaurant areas, ventilation noise should not be excessive. Social club areas may benefit from a slightly higher standard, especially if they are to be used for meetings and bingo sessions. In all areas, soft furnishings and sound absorptive finishes will contribute to aural comfort.

5.1.11 Schools

Basic planning of the various facilities can eliminate some of the acoustic problems. Definitions of requirements are given in DES Design Note 17 (1981): 'Guidelines for Environmental Design and Fuel Conservation in Educational Buildings' and Building Bulletin 51: 'Acoustics in Educational Buildings'. BS 8233 classifies accommodation into four groups, with sound insulation requirements from 25 to 48dB average minima.

Classrooms and assembly halls are key areas where speech intelligibility is critical and where disturbance must be minimised. To this end, the classroom reverberation time should be controlled to 0.75 sec at mid-frequencies and 40dB average separation between adjacent teaching areas aimed for; the reverberation time in the assembly hall may be 1.0–1.25 sec. In the past, the ubiquitous use of glazed tiles and gloss-painted wall plaster was acoustically notorious. Cushioned vinyl or carpet floor finish and sound absorbing ceilings reduce school noise most successfully. Local screening and absorption are particularly important in open plan areas. Ventilation noise control in teaching areas should be to NR 35, except for music teaching where NR 25 should be applied.

Particular attention should be given to special schools for use by deaf, blind or other handicapped children. Deaf children rely on hearing aids which will not function satisfactorily in noisy or reverberant background conditions. Blind children depend on clearly hearing speech and other sounds not supported by visual messages. Guidance for such schools is given in DES Design Note 25 (1981) 'Lighting and Acoustic Criteria for the Visually Handicapped and Hearing Impaired in Schools'. Similar principles of design apply for teaching rooms within universities and colleges as for schools. The smaller groups and greater pupil/student participation involved should be borne in mind in considering the acoustics of each space.

5.1.12 Sports buildings

Sports halls may be for participatory or spectator sports. In either case a moderate noise control standard of NR 40 will suffice, although a higher standard might be expected for halls where sports events such as snooker are to be televised. Site location should allow for isolating the noise of those attending particularly for late night or even 24 hour operation (for example, ice rinks). Insulation from outside noise is not usually a problem, unless the sports hall doubles as an auditorium. Separation of facilities should be carefully considered; for example, squash courts and projectile halls add noise to any complex.

Hard surfaces in sports halls or swimming-pools make teaching difficult by creating over-reverberant conditions. In the case of ice rinks, skaters struggle to coordinate their movements to accompanying music. The Sports Council recommends a reverberation time range of 1.8–3 sec (mid frequency values for hall empty) as acceptable, but the use of built-in absorptive finishes – for example, perforated metal roofing sheets or suspended banners with back-up quilting – can improve further on these values. The sound reception from loudspeakers will be considerably improved by such measures.

Ventilation systems in sports buildings tend to be smaller and simpler than for auditoria, given their ability to deliver air at greater velocity and the relatively low occupancy. One area where silencing deserves greater attention is in the health suites often included in sports centres – a quiet, relaxed atmosphere has to be created whilst satisfying heavy airhandling demands.

5.1.13 Studios

In the UK, local radio stations have to comply with IBA standards involving pass–fail tests. Studios have to be 'dead' acoustically, i.e. with very low reverberation time characteristics (0.16–0.3 sec). Double or triple leaf walls separate adjacent technical spaces. Ventilation has to be very quiet (within NR 15 or 20).

Reference is increasingly made to the European Broadcasting Union recommendations, summarised as follows:

- preferred volume for control rooms 80 m^3
- preferred volume for listening rooms 100 m^3
- rooms should be symmetrical around the listening axis, avoiding single integral ratios between length, width and height
- the principal monitoring positions should be the third vertex of a horizontal equilateral triangle where the sides are at least 2 m, the other two points being loudspeaker positions. The two loudspeakers should be at least 1 m from the walls
- over the one-third octave bands 200–2500Hz, the average reverberation time should be 0.3 ± 0.1 sec. At low frequency one-third octave band 50Hz, the reverberation time should be less than 0.45 sec. The difference in RT between adjacent one-third octave bands should be less than 0.04 sec between 200 and 10,000Hz
- ventilation noise should not have a pronounced tonal quality or cyclic variation. The sound level at any monitoring position should not exceed NR 15.

Recording studios are much larger and may be 1000 m^3 to house a large orchestra, or 150 m^3 for a pop group and its equipment. Room proportions of 1.6 : 1.25 : 1 have been found suitable. Modern devices like graphic equalisers can offset less than ideal studio conditions to some degree.

Television studios tend to compromise aural conditions to meet visual standards. A general purpose studio may be large and high, 11 m in height to include a lighting grid, and length to breadth ratio of 1.25 : 1. A high degree of isolation is used for sound control rooms – 75dB average achieved by wide-spaced layers of masonry. The reverberation time has also to be closely controlled, to within a third of a second throughout the frequency range. This is achieved by low frequency wall panel absorbers. Studios should also be 50 dB separated from circulation areas.

5.1.14 Theatres

Theatres need quiet conditions, good viewlines, and reasonably 'dead' conditions for good sight and sound from the stage. Large theatres need particularly high ambient noise control standard for external and services noise.

Like concert halls, theatres are often in the centre of towns, so entrance foyers, lobbies, bars and ancillary accommodation should be planned around the auditorium itself to provide a buffer to outside noise.

The traditional building form is two coupled spaces, one high and bare except for the 'sets' hung in it, the other packed with seating, ornament and furnishings. The proscenium arch between controls the scene presented to the audience. More recent theatres are single space, with stage at one end or in the centre. For small theatres (500 seats) a volume of 3 m^3/person will give suitable conditions. Theatres should have reverberation time kept within a second or so at mid frequencies. Finishes generally should be sound absorbing except in the immediate vicinity of the stage, and perimeter surfaces non-parallel to avoid cross reflections.

For seating, a 20° rake should be used with no more than 25 rows from the stage, as facial expressions can hardly be seen at 20 m and beyond – seeing lip movements and gestures helps our comprehension of the spoken word. Modern theatres have rediscovered the stacking of audiences in the form of shallow balconies or boxes. The recent revival of the musical means that the opera house form, with orchestra pit fronting the stage, may still be used on specific projects.

For ventilation, similar comments apply as for concert halls, although the problem is slightly easier as the hall is deader and smaller in volume. A particular concern is mid and high frequency ventilation noise components which could mask speech from the stage towards the rear of the house.

5.2 Building Elements

5.2.1 Roofs

As an element of structure, roofs inherently tend to be more lightly constructed than walls or floors. This does not necessarily downrate the total building fabric for external noises like traffic or industry, but can mean that the building is more vulnerable to an overhead noise source like aircraft.

Traditional slated or tiled roofs are substantial, but because of all the gaps between individual slates and the need to ventilate under it, the roof is acoustically poor on its own. Combined with a loft void, insulation, and plastered ceiling, the total sound insulation performance is reasonable (38dB).

High performance felts on metal decking, or composite metal sheet/insulation/metal sheet construction, are frequently used on industrial and office developments. Their average SRI rating is limited to 30–35dB.

Lightweight concrete pre-cast units with asphalt roof finish, 'upside down roofs' with pebble ballast to hold down insulation boards, and built-up roofing on pre-screeded woodwool slabs, all form good mid-weight systems, achieving over 45dB.

For the highest degree of isolation (over 50dB), asphalt and thermal insulation on in situ concrete, combined with ceiling void and resilient hangers to substantial ceiling, should be used.

5.2.2 Ceilings

Ceiling constructions are mainly used to modify the noise field within the enclosure in which they are installed. They consist of sound absorbing tiles or planks which are suspended from the main structure or are mounted independently of the main structure off, for example, internal partitioning. Such ceilings do little to improve the air-borne or impact insulation of the main structural floor.

Sound insulation

For good sound insulation the ceiling will have to be reasonably heavy and such construction may not be practical. One method that has been used to improve the performance of structural ceilings in the conversion of old houses to flats is described as follows: a 25 mm thick double layer of plasterboard is supported on joists at about 200–300 mm below the existing ceiling with a 30 mm glass fibre mat laid on top of the plasterboard. This type of construction is most effective when the original structure shows poor sound reduction and in these cases typical improvements of about 10dB can be obtained.

Isolated facilities like studios, control rooms or practice rooms sometimes comprise heavy ceiling on resilient hangers as part of 'boxes-in-box' arrangements.

Sound absorption

Nearly all proprietary suspended ceilings are at least moderately sound absorbing, by textured and fissured surface (mineral tile), gaps with quilt over (metal plank) or perforated with quilt over (metal tiles). Inclusion of a ceiling modifies the noise field not only by its absorbent surface but also by its reduction of the room's effective volume. The height of suspended ceiling as well as its specification should therefore be considered.

Open waffle assemblies of sound absorbing panels add large areas of absorption to rooms, but do not modify the room volume.

5.2.3 Walls

The most basic sound insulation is afforded by industrial buildings, with their lightweight profiled metal cladding and linings. Roller shuttered delivery doors further reduce the isolation of inside to out to not much over 25–30dB. Blockwork rather than lightweight internal linings and gasketed folding-panel industrial doors can improve this.

Traditional masonry and cavity wall construction has good inherent qualities, achieving 48–50dB. Windows and doors are the weak links in such walls.

Many office buildings use curtain walling techniques, full-height framing holding glazed, or insulated lightweight, panels. 1950s and 1960s office blocks are susceptible to noise intrusion because of their use of single glazing, opening lights, and narrow plan format. Double curtain walling units with wide spacing between panels and air conditioning can raise a separation of, typically 25dB for older systems, to 35–40dB.

For buildings where a high degree of isolation is required, for example, studio or auditoria, perimeter accommodation can be used to buffer critical spaces to external noise.

Party walls

Under the Building Regulations 1985, walls of adequate sound insulation standard are mandatory to

- separate a dwelling from another building or from another dwelling, or
- separate a habitable room within a dwelling from another part of the same building which is not used exclusively within the building.

The Regulations recognise that clear construction parameters have to be given to ensure that a reasonable standard is consistently met. The relevant provisions are contained in section E: Sound, E1 Air-borne Sound (Walls).

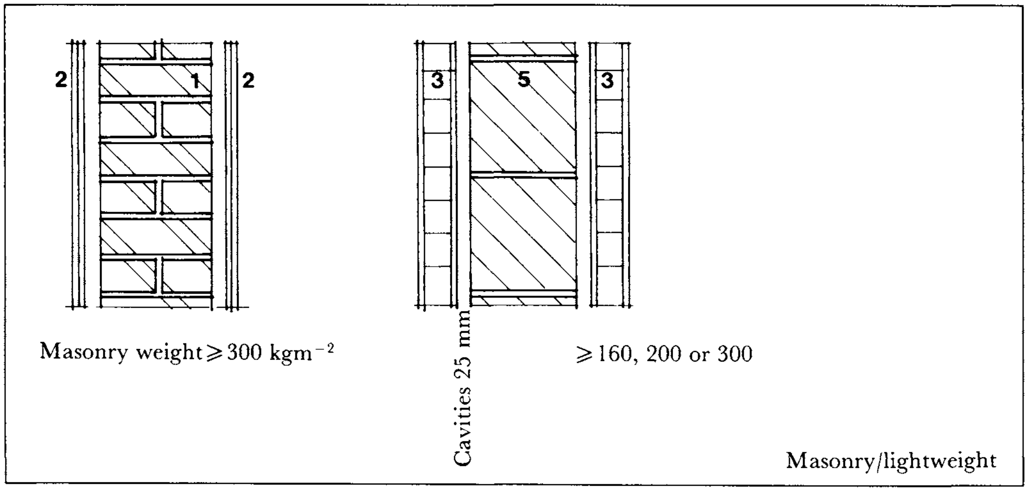
For direct transmission, timber framed walls are favoured to give the best isolation involving less weighty structure (Figure 5.2). The timber frame alternative differs in its use of two isolated frames with sound absorbing quilt in the space between, rather than relying on mass alone. Substantial single walls (374 kgm^{-2} in plastered brick, 415 kgm^{-2} in concrete blockwork or concrete) are acceptable forms of party wall (Figure 5.3). Cavity masonry walls will not necessarily give improved performance over that of single leaf walls of equivalent weight, at all frequencies.

Some of the alternative constructions recommended are interpreted in Figures 5.1–5.4. However, the Approved Document guidance should be examined in detail. Construction junctions should be designed to avoid the deleterious effect of flanking sound. For example, BRE research in the UK has highlighted that a high proportion of party walls failed to work properly as sound barriers so construction standards are also important.

Figures 5.1 – 5.4 UK Building Regulations Approved Document wall assemblies.

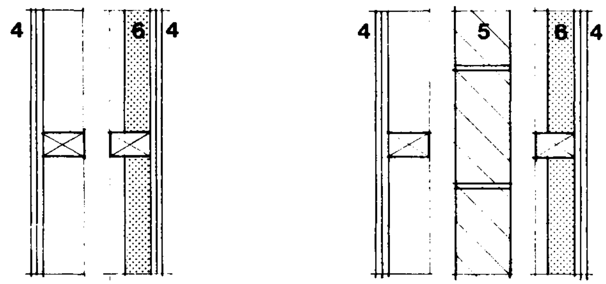
Typical specifications

1 Brickwork $\geqslant$ 300 kgm^{-2}
2 2 × 12.5 mm plasterboard
3 As above, either side of cellular core
4 As 2, with ply sheathing
5 Autoclaved aerated concrete blockwork or concrete blockwork (block density < 1500 kgm^{-3}) or concrete blockwork ($\geqslant$ 1500 kgm^{-3})
6 50 mm mineral wool $\geqslant$ 12 kgm^{-3}

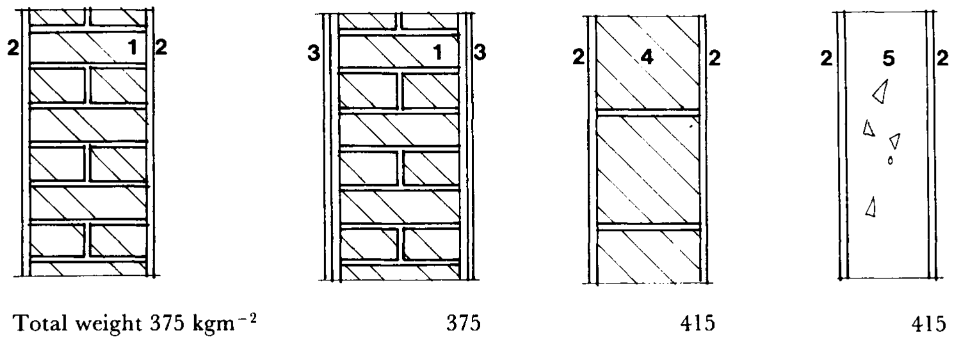


Typical specifications

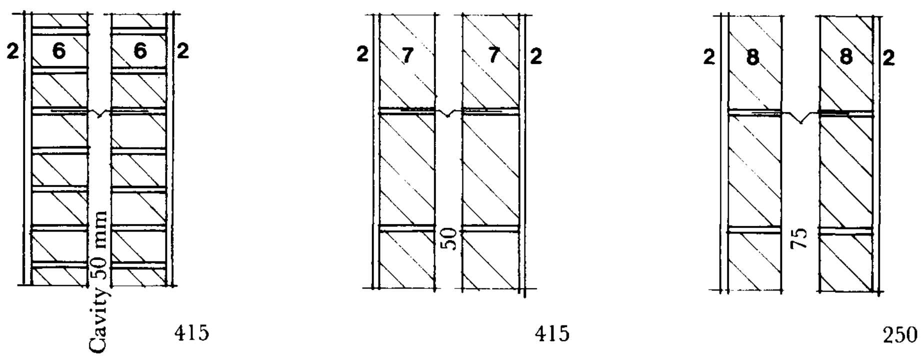
1 225 mm brickwork $\geqslant$ 355 kgm^{-2}
2 12.5 mm plaster
3 12.5 mm plasterboard
4 200 mm blockwork
5 Dense concrete
6 112 mm brickwork
7 100 mm blockwork
8 100 mm lightweight blockwork



Timber frame



Solid masonry



Cavity masonry

Other wall types can be used provided that the weighted standardised level difference meets certain values. An individual wall must give a minimum value of 49dB while if four walls are tested the minimum mean value must be 53dB and if eight walls are tested the minimum mean value must be 52dB. The corresponding impact requirements are that the weighted standardised sound pressure levels have values of 65, 61 and 62dB respectively.

Dry linings

A typical dry lining will consist of plasterboard joined to 50 mm of glass fibre fixed to a structural wall by means of plaster dabs (see Figure 5.1). Dry linings are used to improve the sound reduction of existing constructions. They tend to be effective at frequencies above 250Hz and maximum increases of no more than 10–15dB can be expected at any frequency. Obviously they are more effective when applied to low performance construction than when applied to constructions which are already giving reasonable insulation when their effectiveness is limited by flanking transmission.

It has been found that dry linings can marginally improve the performance of separating floor, if used at both levels. This is because edge flanking transmission is reduced.

5.2.4 Floors

Basic planning can alleviate some noise source problems: the lowest floor location for noisy plant or processing plant will not offend anyone on a floor below and there are fewer problems in 'driving' the structure.

Timber floors offer the poorest separation, typically 33dB for chipboard flooring on joists and plasterboard and skim ceiling below. Such measures as isolating the floor deck, placing quilt in the void between joists, and using thicker multi-layer plasterboard linings can lift the average SRI value to 45dB (Figure 5.5).

Lightweight 'pot and plank' pre-cast concrete units as a floor structure tend to produce disappointing results unless a screed topping seals the top jointing. This will achieve 40–42dB. In situ concrete slab with isolated screed or deck over can be rated 48–50dB (Figure 5.6).

For floors impact as well as air-borne sound transmission must be considered; isolation of the floor finish base to the structure floor improves both impact and air-borne characteristics, but particular care at floor edges is required to maintain separation.

Party floors

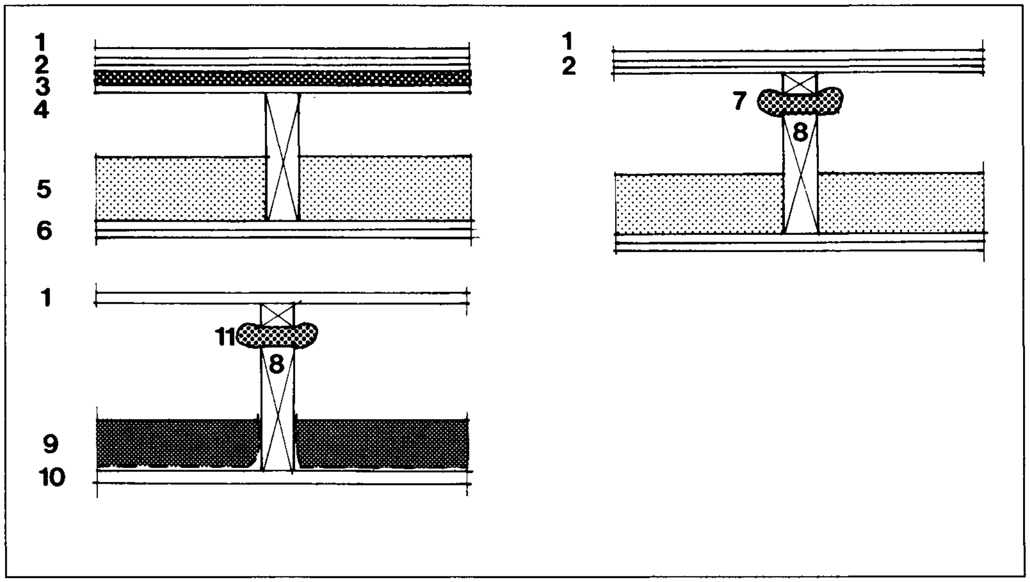
Under the Regulations, Sound: E2 Air-borne Sound (Floors) and E3 Impact Sound (Floors), sound resisting floors have to be provided, either to the dwelling to keep out noise from accommodation below, in which case only air-borne sound is a consideration, or to the floor above a dwelling, in which case the floor must resist both air-borne and impact sound.

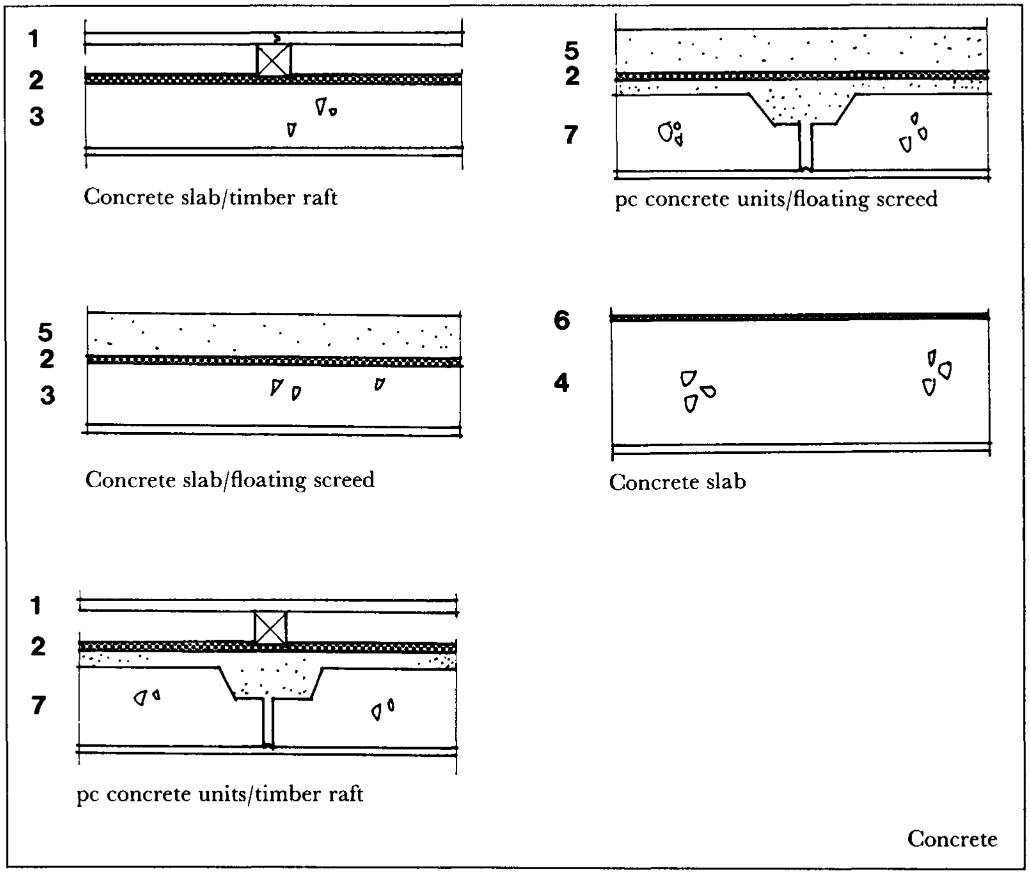
Figures 5.5 and 5.6 UK Building Regulations Approved Document floor assemblies.

Typical specifications

1 $\geqslant$18 mm timber t & g boarding
2 $\geqslant$19 mm plasterboard
3 $\geqslant$25 mm mineral fibre (60–80 kgm^{-3})
4 $\geqslant$12 mm ply deck
5 $\geqslant$100 mm mineral fibre ($\geqslant$12 kgm^{-3})
6 $\geqslant$30 mm plasterboard
7 $\geqslant$25 mm mineral fibre (90–140 kgm^{-3})
8 $\geqslant$50 mm wide timber joists
9 dry sand or fine gravel (80 kgm^{-2}) on plastic sheeting
10 $\geqslant$19 mm dense plaster on expanded metal
11 $\geqslant$25 mm mineral fibre (70–140 kgm^{-3})

Authors' note: use with masonry walls only





Typical specifications

1 $\geqslant$18 mm timber t & g boarding
2 $\geqslant$13 mm mineral fibre (36 kgm^{-2}) or ISD expanded polystyrene
3 Concrete ($\geqslant$220 kgm^{-2})
4 Concrete ($\geqslant$365 kgm^{-2})
5 $\geqslant$65 mm reinforced cement/sand screed
6 $\geqslant$4.5 mm resilient floor finish
7 pc concrete units ($\geqslant$220 kgm^{-2})

Three acceptable floor types are considered:

- o concrete base with soft covering
- o concrete base with floating layer
- o timber base with floating layer.

These are shown in Figures 5.5 and 5.6.

For concrete bases, any of the bases can carry either of the resilient layers, or either of the floating layers. Resistance to air-borne sound depends mainly on the weight of the concrete base and to a limited extent on the weight of the floating layer.

For timber bases, reduction of air-borne sound depends both on the structural floor/absorption or pugging between, and the floating layer which also limits the transmission of impact sound.

Timber floors need less weight than concrete floors for a specific acoustic performance because the materials radiate sound less efficiently.

Care over the application of such floors is advised: there have been problems in practice for the use of pugging (floor type 3/specification C). It is recommended that a 'ribbed' floor with heavy pugging only be used with masonry walls. An important departure from previous deemed-to-satisfy provisions is the use of thick (30 mm) plasterboard layers which impose self-weight loads to the floor.

Both concrete and timber floor systems must have full edge isolation to the floating layer.

Other forms of construction can be used provided acoustic tests show that air-borne sound is sufficiently reduced. The weighted standardised level difference must have a minimum value of 48dB if only one floor is measured, while the minimum mean values if 4 and 8 floors are measured must be 52 and 51dB respectively. The impact pressure level should have values of 65, 61 and 62dB.

Outside the UK, the ISO Standard 717: 1982 is used to give an air-borne sound insulation rating for any enclosing construction; for example, a partition might be 35dB, party wall 53dB and a party floor 52dB.

5.2.5 Windows

To obtain good sound reduction from windows they must be sealed with no opening lights. The average sound reduction index of a single pane increases with increasing surface mass, which generally means with increasing thickness. However, the coincidence frequency, which is approximately given by 1200/t, where t is the glass thickness in millimetres, tends to occur around the frequency range 2000Hz, reducing the performance below that predicted by the mass law. For single glazed windows it should be possible to obtain average SRIs of 25–30dB while using sealed double units values of between 30 and 38dB should be possible. Use of laminated glass (1.5 mm interlayer) gives a slight advantage over plain glass. With double units there tends to be a significant dip in performance at around 200Hz because of the mass–air–mass resonance. This can be eliminated to some extent by replacing the air in the gap between the panes with a different gas such as sulphur hexafluoride or argon. To obtain the maximum performance from a sealed unit it is also best to have two panes of differing thicknesses, so reducing the coincidence effects.

High-performance glazing

When sound reductions greater than 35dB are required, dual or triple glazing will have to be used. With dual glazing the larger the gap between the two panes the better the performance. A minimum air space of 150 mm is recommended and the reveals should be lined with absorbent material to reduce cavity resonances. For maximum performance the panes should be of different thicknesses, mounted in separate frames and held in rubber or neoprene mouldings. Triple glazed units are used if very high reductions of 45–50dB are required. The third pane is inserted between the two outer panes, often at an angle to the vertical, to reduce standing wave resonances. The larger the air gaps between the panes the more successful the insulation. The performance of this type of unit can be reduced by flanking transmission through the reveal and a break should be incorporated in the reveals. As a cavity wall construction will normally be employed, one half of the reveal can be included in each wall. Features in window design for good sound insulation are shown in Figure 5.7.

Corrections to glazing performance data will be necessary if the sound is not arriving at the glazing at normal or random incidence, e.g., road traffic noise on to a tower block, or if reveals are lined with acoustically absorbent material. The effects of these vary from case to case and, ideally, laboratory test results should be consulted if available. If not, the data shown in Table 5.3 will provide a reasonable estimate for the correction to the SRI figures.

Figure 5.7 Features in window design for good sound insulation.

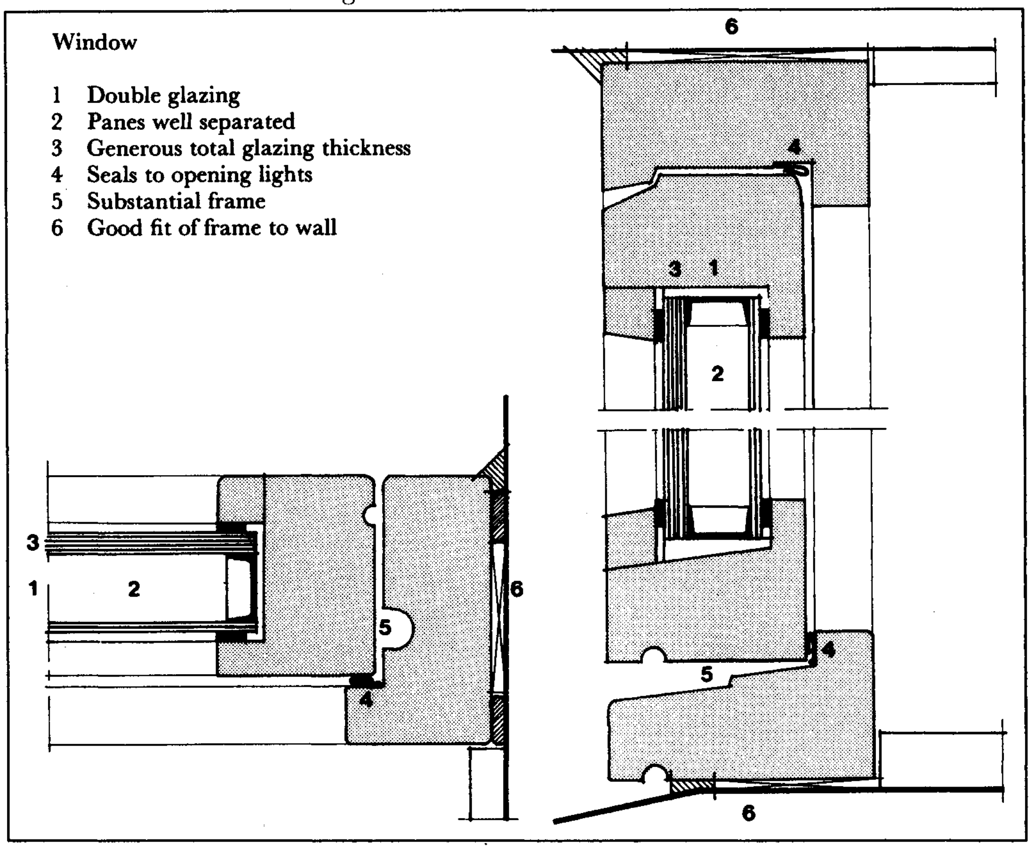


Table 5.3 Effect of noise directivity and reveals lining on transmission through window systems

	63	125	250	500	1K	2K	4K	Hz
(a) Angle of incidence (indicative only):								
45° from normal	0	−1	−2	−3	−5	−5	−5	dB
75° from normal	0	−4	−6	−8	−8	−8	−8	dB
(b) Lined reveals (indicative only):								
typical 12 mm lining	0	0	+1	+1	+2	+3	+3	dB
typical 25 mm lining	0	+1	+2	+3	+4	+5	+5	dB

5.2.6 Doors

The acoustic effectiveness of a door depends greatly on the quality of the seal around the door. However, it is still necessary to construct the door so that it is capable, ignoring sealing effects, of providing the required sound reduction. As with walls this means having the correct surface mass. Additional factors to be considered to obtain its potential sound reduction are:

- an airtight seal of magnetic strips or rubber gaskets must be provided in a rebated frame all round the door, including a threshold. If a threshold is not possible, for high insulation some form of retractable seal must be provided at the bottom of the door
- door closers and catches should cause the door to compress on to the seals as it closes; keyholes should also be covered with escutcheons
- the frame must be flat so that the seal is compressed evenly all round and it must be well sealed into the wall structure. For high insulation doors it is advisable to use a proprietary door which is supplied complete with frame, preferably metal (Figure 5.8). Frames in heavy acoustic doors need rigid, substantial anchorage into masonry. If acoustic doors have to be fitted into high-performance dry partitioning (e.g. in studios), structural steel sub-frames must be built in.

Sound lobby

If space is available it is often easier to obtain a high sound reduction of 40–50dB, by using two medium performance doors in the form of an air-lock rather than one high performance door. The two doors, which must be well sealed are separated by a short passage way, the walls and ceiling of which are lined with absorptive material. The longer the passageway the better.

Larger doors

Double doors can be obtained with sound insulation rating compatible with single

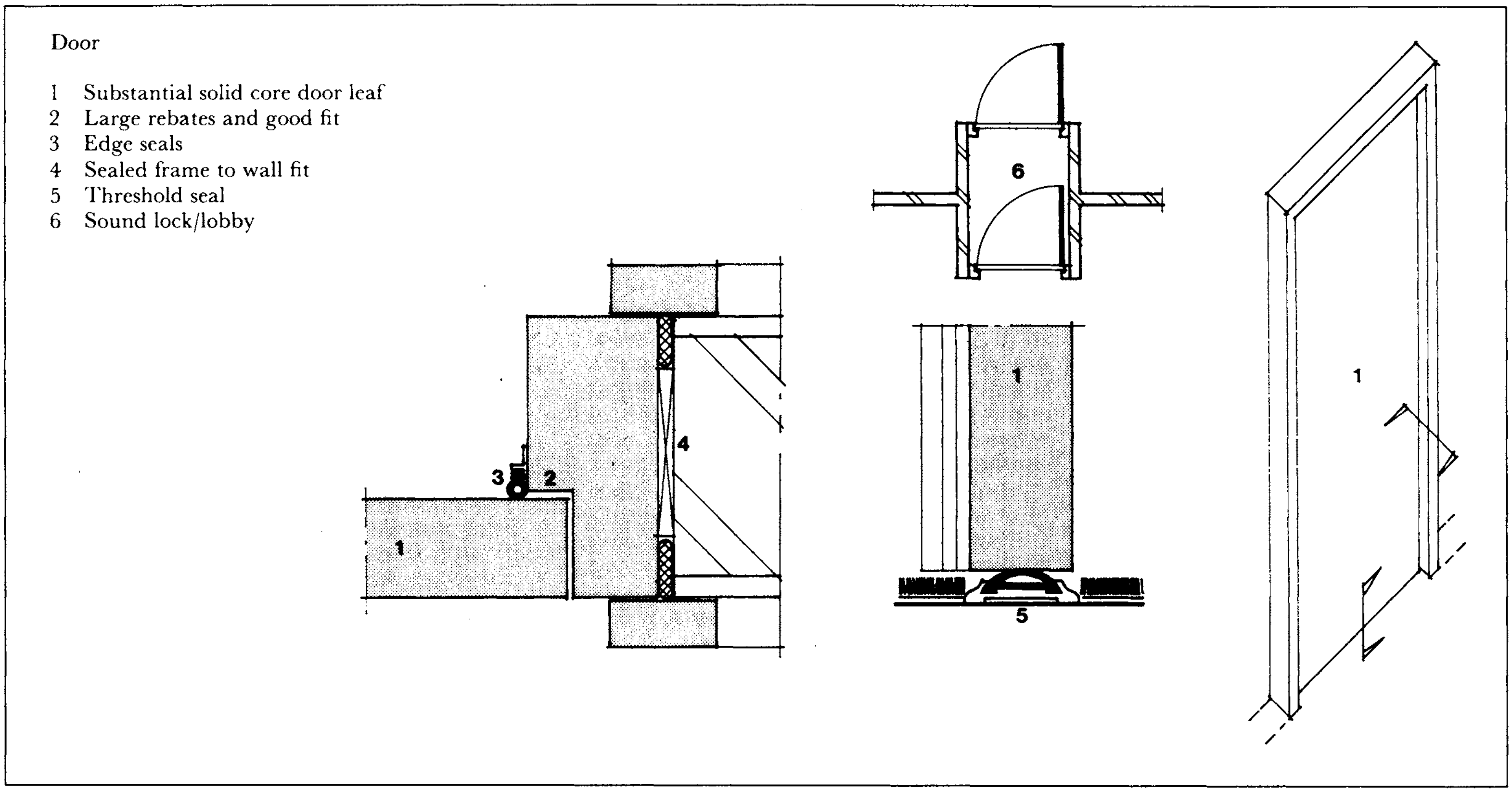


Figure 5.8 Features in door design for good sound insulation.

doors, but rebated centre closure to seals is essential. As with single doors, care has to be taken with door closers where these are required for fire doors. The closing face must be strong enough to close the doors against seal resistance; equally, opening and closing of doors should be easy. A selector mechanism is required for double doors with rebated edges.

Industrial door openings will have very poor performance if standard roller shutters are used, because these have gaps at edges and joints in shutter sections. Thermally insulated vertical rise or horizontal folding panel doorsets are slightly better (average SRI 30–35dB).

A high degree of separation can be afforded by special 'sliding wall' arrangements for studios – average SRI can exceed 50dB.

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