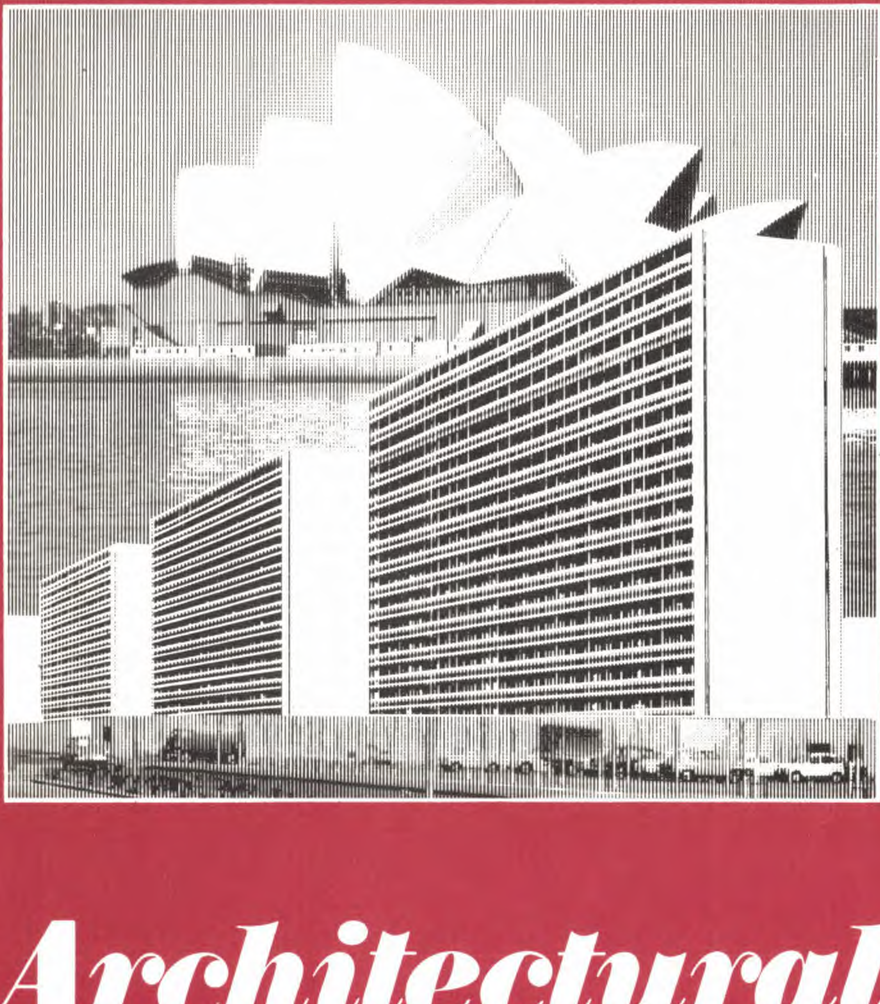

Architectural Acoustics

Brüel & Kjær

ARCHITECTURAL ACOUSTICS

by

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CONTENTS

FOREWORD

This booklet is intended as an introduction to the methods and instrumentation developed by Brüel & Kjær in the sphere of architectural acoustics. With this aim in mind the reader is presented with the basic theory necessary to make the best use of the instruments' capabilities.

As this booklet is an introduction to the subject the amount of mathematics employed has been kept to a minimum. The reader who requires a more thorough rendering of a particular topic should consult the reference literature. The suggested instrument arrangements should be considered as a guide only, for many alternative arrangements are usually possible.

INTRODUCTION

Architectural acoustics can be defined as the study of the generation, propagation and transmission of sound in rooms, dwellings and other buildings.

Although a relatively new science, architectural acoustics permeates every walk of modern life. Correct application of the principles of architectural acoustics can considerably improve the quality of life at work, during leisure time and in the home. Some sounds are desirable and need to be enhanced or emphasized (e.g. music in a concert hall; the speakers voice in a debating chamber etc), other sounds are highly undesirable (known as noise) and need to be reduced or prevented (e.g. noise in a factory workshop; noise from a neighbour's party in the early hours of the morning etc). In many countries minimum limits have been set for the permitted noise levels in a particular environment (e.g. in the home, at the place of work). Regulations have also been drawn up defining the minimum acceptable acoustic properties of building elements (e.g. walls, floors, doors) and the minimum acceptable sound insulation that should exist between adjoining dwellings.

These regulations are sometimes recommendations and sometimes enforceable by law.

Careful thought about the acoustic properties of a proposed building at the design stage, perhaps in conjunction with the results from acoustic measurements on material samples and scale models, can often save much time and effort later on. It is frequently the case, however, that alterations have to be made to improve the acoustic properties of the finished building. To do this effectively, measurements usually have to be performed before a remedy can be proposed.

The first chapter of this booklet gives a brief summary of the more important concepts and definitions necessary for an understanding of architectural acoustics. In chapter two, the acoustics of rooms are examined in detail. Chapter three deals with the acoustics of buildings and the transmission of airborne and impact sound. Chapter four describes some of the more important noise criteria and regulations which are pertinent to architectural acoustics. Chapter five introduces the measuring techniques and chapter six, the instrumentation most commonly used in architectural acoustics.

1. FUNDAMENTALS AND DEFINITIONS

1.1. NATURE OF SOUND

Sound is the sensation perceived by the human ear resulting from rapid fluctuations in air pressure. These fluctuations are usually created by some vibrating object which sets up longitudinal wave motion in the air.

Fig. 1.1. Propagation of ripples on the surface of a pond

Most people have some intuitive idea of what constitutes a wave. Almost everyone has seen ocean waves breaking on the seashore or has noticed the ripples which radiate away from the place where a pebble strikes the surface of a pond (Fig.1.1). Sound waves are a particular type of a general class of waves known as elastic waves. Elastic waves can occur in media which possess the properties of mass and elasticity. If a particle of such a medium is displaced then the elastic forces present will tend to pull the particle back to its original position. The term particle of the medium denotes a volume element large enough to contain millions of molecules so that it may be considered as a continuous fluid, yet small enough so that such acoustic variables as pressure, density and velocity may be thought of as constant throughout the volume element.

The displaced particle possesses inertia and can therefore transfer momentum to a neighbouring particle. The initial disturbance can therefore be propagated throughout the entire medium.

There are several analogies that can be drawn between the propagation of a sound wave and the propagation of the ripples on the surface of the pond. Both disturbances travel away from their respective sources at constant speed. Both disturbances propagate by an exchange of momentum and there is no net transfer of matter away from the sound source just as there is no net fluid flow in the pond.

The important distinction is, however, that the ripples are propagated by transversal waves (i.e. the particle velocity is at right angles to the direction of propagation) whereas sound in air is propagated by longitudinal waves (i.e. the particle velocity is in the direction of propagation.

1.2. WAVE TERMINOLOGY

There are a number of terms in common use used to describe the nature of propagation of a sound wave. Some of the more important terms are defined here.

Plane Wave

When corresponding wavefronts of a sound wave propagate parallel to each other then the sound wave is known as a plane sound wave e.g. the sound wave produced by a piston oscillating in a long cylinder.

Diverging Wave

A diverging sound wave is one where the sound energy is spread over a greater and greater area as the wave propagates away from the sound source i.e. the the sound intensity diminishes with distance from the source.

Spherical Wave

A spherical sound wave is produced by a sound source which radiates sound energy equally in all directions e.g. a monopole source.

Progressive Wave

When there is a transfer of energy in the direction of propagation of the sound wave the wave is designated as progressive.

Standing Wave

A standing wave is produced by the constructive interference of two or more sound waves which gives rise to a pattern of pressure maxima and minima which is stable with time e.g. standing waves can exist in tubes, musical instruments, organ pipes and in larger volumes such as rooms.

1.3. FREQUENCY OF SOUND

Consider the air near to the surface of some vibrating object e.g. piston in a tube, tuning fork, loudspeaker diaphragm. The series of compressions and rarefactions produced by the movement of the object constitute a sound wave, the frequency of which is determined by the rate of oscillation of the object. When the oscillation repeats itself, the motion is said to have completed one cycle. The number of cycles per second is called the frequency, f. The unit of frequency is the Hertz. 1 Hertz = 1 cycle/sec. The time taken for the oscillation to repeat itself is known as the period, T.

$$f = \frac{1}{T} \tag{1.1}$$

Fig. 1.2 shows the relationship between the compressions and rarefactions and the pressure variation produced by a vibrating piston.

It is often useful to express frequency in terms of the angular frequency. For a vibration of frequency, f, the corresponding angular frequency, ω, is

$$\omega = 2\pi f \tag{1.2}$$

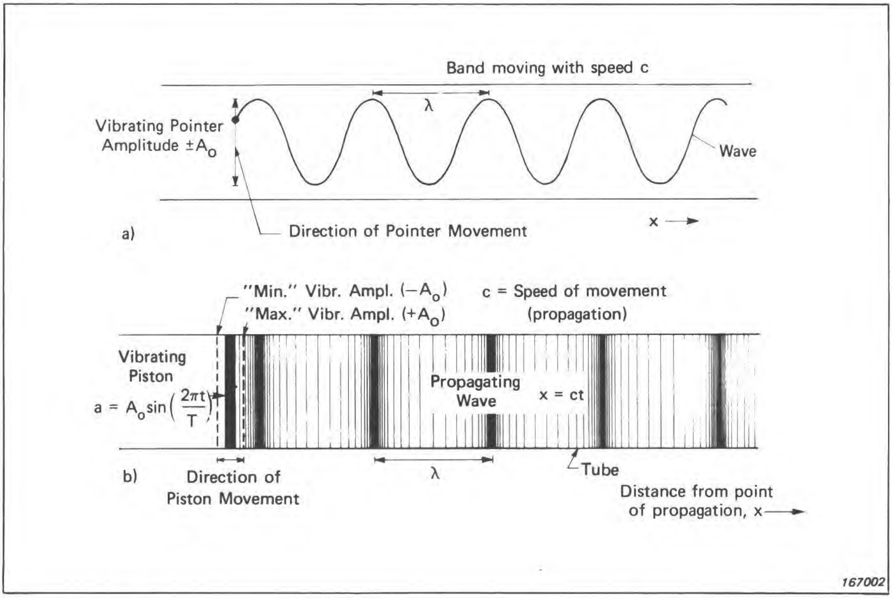

Fig. 1.2. Compressions and rarefactions in a longitudinal plane wave
a) Pressure variation of a sinusoidal wave
b) Propagation of a plane wave in a tube

1.4. SPEED OF SOUND

The speed of propagation or speed of sound is dependent on the mass and elasticity of the medium. The elasticity of air is determined by experiment to be a constant multiplied by the atmospheric pressure. The constant, γ, is found to be the ratio of the specific heat of the air at constant pressure to the specific heat at constant volume. For the temperature range that acousticians usually deal with this ratio is 1,4. The speed of sound, c, in air is given by

$$c = \sqrt{\frac{1{,}4\,P_o}{\rho}} \qquad (1.3)$$

where P_o = atmospheric pressure
ρ = density of air

Assuming that air acts as a ideal gas, it can be shown that the speed of sound depends only on the absolute temperature of the air according to the equation,

$$c = 332\sqrt{1 + \frac{t}{273}} \tag{1.4}$$

where t = air temperature (C°)
c = speed of sound ($m.s^{-1}$)

At ordinary room temperatures, the speed of sound is approximately 340 $m.s^{-1}$.

1.5. WAVELENGTH

The wavelength, λ, is the distance between two successive pressure maxima or between successive pressure minima in a plane wave. The relationship between λ, c and f is

$$c = \lambda f \tag{1.5}$$

The nomogram in Fig. 1.3. gives the wavelength corresponding to a particular frequency.

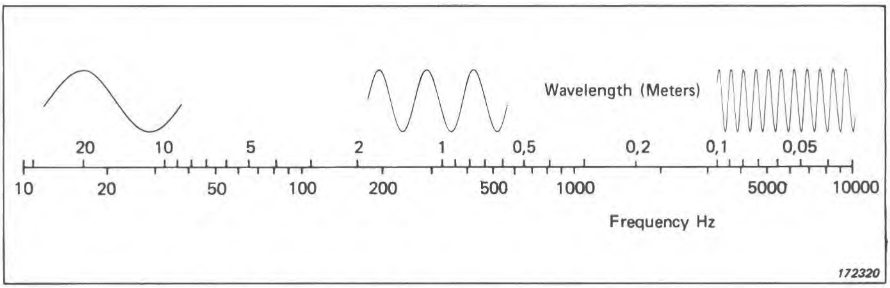

Fig. 1.3. Nomogram relating wavelength and frequency

1.6. DISPLACEMENT

The distance between the instantaneous position of a vibrating particle and its mean position is the displacement of the particle.

1.7. AMPLITUDE

The maximum displacement experienced by a vibrating particle is known as the amplitude of vibration. The amplitudes of vibration of airborne sound waves which occur in practice are very small. The amplitudes range from about 10^{-7} mm up to a few mm. The smaller amplitude corresponds to the sound which is just perceptible by the ear while the greater amplitude is the limit beyond which the ear would suffer damage.

1.8. PARTICLE VELOCITY

The differentiation of the displacement of the particle with respect to time yields the particle velocity. This property of a sound wave can be measured directly using a classical technique developed by Lord Rayleigh known as Rayleigh's disc. The method is, however, tedious and is of little practical importance.

1.9. SOUND PRESSURE

The pressure variations produced when a sound wave propagates through the air are very small compared with the static atmospheric pressure. The slightest sound that an average young adult can detect corresponds to a sound pressure of 0,00002 Pa (N.B. 1Pa = 1N.m^{-2}). This sound pressure is superimposed on the ambient atmospheric pressure which is of the order of 10^5 Pa. The concept of sound pressure is extremely important because of all the quantities which could be used to characterise the "strength" of a sound wave (e.g. particle velocity, sound intensity) it is the sound pressure which is the most amenable to measurement.

1.10. SPECIFIC ACOUSTIC IMPEDANCE

For a sinusoidal wave form, the ratio of the acoustic pressure in a medium to the associated particle velocity is defined as the specific acoustic impedance of the medium for the particular type of wave motion present. The specific acoustic impedance, z, for plane waves is a real quantity (i.e. not complex) of magnitude ρc where ρ is the density of the medium and c is the speed of sound in the medium. The SI unit of specific acoustic impedance is the rayl, expressed in N.s.m^{-3} . As the product ρc is a characteristic property of the medium, this product is also referred to as the characteristic impedance of the medium. At normal temperature and pressure (i.e. 20°C and 10^5 Pa), the specific acoustic impedance of air has the value of 415 rayls.

Although the specific acoustic impedance of a medium is a real quantity for progressive plane waves, this is not true for standing plane waves or for diverging waves. In general, z, will possess both a real part, r, and an imaginary part, jx, so that

$$z = \frac{p}{v} = r + jx \tag{1.6}$$

where r = specific acoustic resistance
x = specific acoustic reactance of the medium for the particular wave motion being considered.

The concept of acoustic impedance is important when dealing with standing waves and absorption of sound energy as for example in the application of the wave equation to certain problems in room acoustics (see Chapter 2).

1.11. DECIBEL

In theorectical investigations of acoustic phenomena, it is convenient to express sound pressure, sound intensity and sound power by Pa, $W.m.^{-2}$ and W respectively. For practical measurements, however, it is usual to express these quantities using logarithmic scales. There are several reasons for using such scales. One is a consequence of the very wide range of sound pressures and intensities encountered e.g. the range of audible intensities is from 10^{-12} to $10W.m^{-2}$. The use of a logarithmic scale compresses the range of numbers required to describe this wide range of intensities. Another reason is that the human ear subjectively judges the relative loudness of two sounds by the ratio of their intensities, which is a logarithmic behaviour.

The most commonly employed logarithmic scale for describing sound levels is the decibel scale. One decibel is the energy or power ratio, r, defined by

$$Log_{10}\, r = 0{,}1 \qquad (1.7)$$

For sound pressure or particle velocity ratios, the definition is $Log_{10}\, r$ = 0,05.

The important point about the decibel is that it is a relative measurement and that each quantity measured in decibels is expressed as a ratio relative to a reference pressure, power or intensity or whatever other quantity is being considered. It is good policy to employ the word "level" whenever decibels are used e.g. sound pressure level, sound power level etc. as a reminder that the measurement is a ratio relative to some reference level.

1.12. PEAK, AVERAGE AND RMS

Whenever measuring a quantity which varies with time, such as sound pressure level, it is necessary to state whether the measurements represent the peak, average or rms value of the signal. The relationship between these three measurements for a sinusoidal signal is shown in Fig. 1.4.

The rms (root mean square) value is the most commonly used because it has a direct relationship to the energy content of the signal. The rms value of a signal is defined as

$$A_{rms} = \sqrt{\frac{1}{T}\int_0^T a^2(t)dt} \tag{1.8}$$

The average value of the signal is

$$A_{|average|} = \frac{1}{T}\int_0^T |a|\, dt \tag{1.9}$$

The peak value, A_{peak}, is the maximum amplitude value that the signal reaches within the period of time T.

For the special case of a pure sinusoid the relationship between these three values is

$$A_{rms} = \frac{\pi}{2\sqrt{2}} A_{|average|} = \frac{1}{\sqrt{2}} A_{peak} \tag{1.10}$$

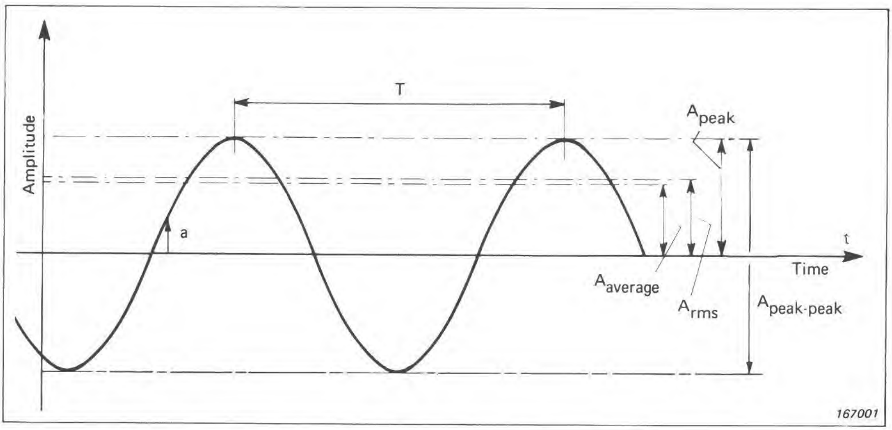

Fig. 1.4. Relationship between rms, peak and average value of a sinusoidal signal

1.13. ENERGY DENSITY

A sound wave contains kinetic energy, as a consequence of the particle velocity, and potential energy as a result of the sound pressure. This energy propagates with the speed of sound. The sound wave, therefore, conveys mechanical energy. The amount of energy per unit volume of a sound wave is measured by a quantity known as the energy density. For a plane sound wave the energy density, E, per unit volume is defined by

$$E = \frac{p^2_{rms}}{\rho c^2} \qquad (1.11)$$

where p^2_{rms} = mean square sound pressure (Pa)
ρ = density of air ($kg.m^{-3}$)
c = speed of sound ($m.s.^{-1}$)
E = energy density ($W.s.m^{-3}$)

1.14. INTENSITY

The intensity, I, of a sound wave is defined as the mean value of the acoustic energy which crosses a unit area perpendicular to the direction of propagation in unit time. Unlike the expression for the energy density of a sound wave, the expression for the intensity is different for different types of sound field.

For any free progressive wave

$$I = \frac{p^2_{rms}}{\rho c} \qquad (1.12)$$

The intensity of a diffuse sound field at the walls of a room is

$$I = \frac{p^2_{rms}}{4\rho c} \qquad (1.13)$$

where p^2_{rms} = mean square sound pressure (Pa)
ρ = density of air ($kg.m^{-3}$)
c = speed of sound ($m.s.^{-1}$)
I = intensity ($W.m.^{-2}$)

1.15. INTENSITY LEVEL

The intensity level, IL, of a sound of intensity, I is defined by

$$IL = 10 \log_{10}\left(\frac{I}{I_0}\right) \qquad (1.14)$$

where IL is expressed in decibels and I_o is the reference intensity usually taken as 10^{-12} $W.m.^{-2}$.

1.16. SOUND PRESSURE LEVEL

The sound pressure level, SPL, of a sound of root mean square pressure P_{rms} is defined by

$$SPL = 20\, log_{10}\left(\frac{p_{rms}}{p_0}\right) \qquad (1.15)$$

where SPL is expressed in decibels and P_0 is the reference sound pressure level of 0,00002Pa. This value is chosen as it approximately corresponds to the sound pressure of the slightest sound an ear can detect in quiet surroundings. Sound pressure level is the quantity that is actually measured when a microphone is placed in a sound field. The portable instrument used to measure SPL is known as a sound level meter. Some typical sound pressure levels are shown in Fig. 1.5.

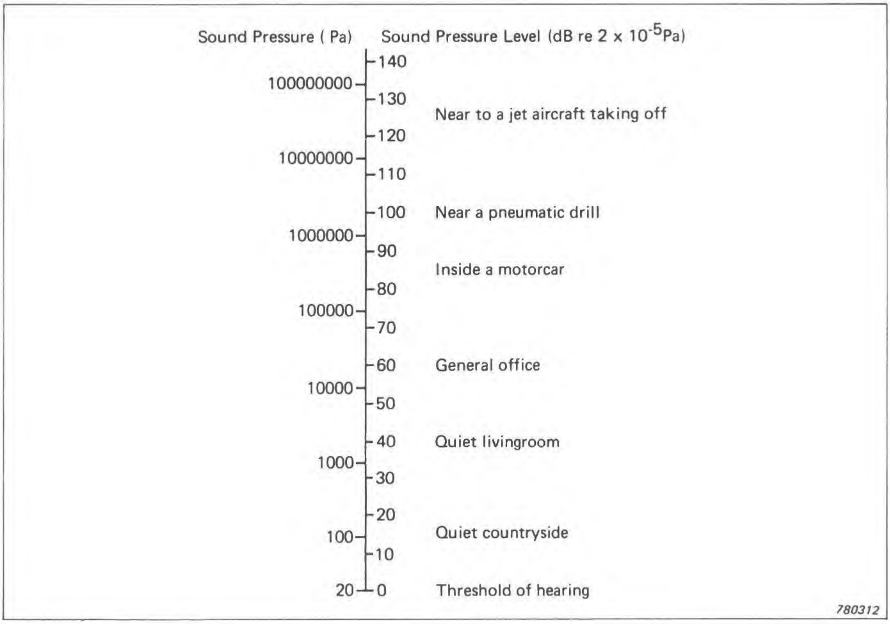

Fig. 1.5. Some typical sound pressure levels

1.17. SOUND POWER LEVEL

The sound power level, SWL or L_W is a measure of the energy output of a sound source. The SWL is defined by

$$SWL = 10\, log_{10}\left(\frac{W}{W_0}\right) \qquad (1.16)$$

where SWL is expressed in decibels, W is the acoustic power of the source and W_0 is the reference acoustic power of 10^{-12} W. When assessing the noisiness of a machine or domestic appliance, it is not sufficient to quote just the sound pressure level measured using the A-weighting network since the level measured is dependent on the environment. This dependence has been known for many years and consequently it has generally been accepted that the sound power emitted by the machine should be determined as this is the fundamental indication of the noise output and is virtually independent of the environment. Some typical sound power outputs of various sources are shown in Fig. 1.6.

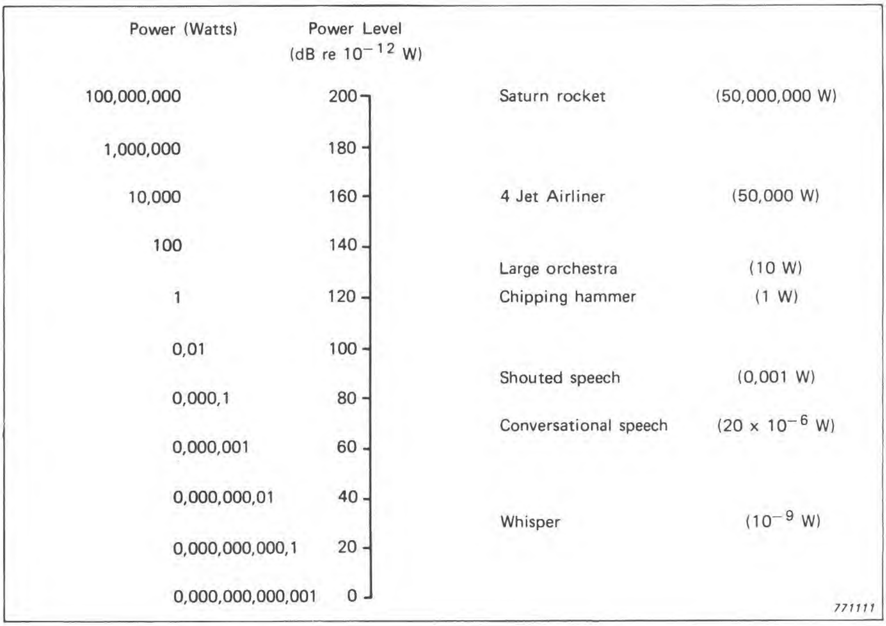

Fig. 1.6. Some typical sound power levels

1.18. SOUND SOURCES — THEORETICAL AND PRACTICAL

The variety of sound sources one has to deal with in architectural acoustics is very large e.g. loudspeakers, machines, the voice, musical instruments. The characteristics and the directivity pattern of the sound generated by each type of source may vary considerably. To describe each type of sound source comprehensively in theoretical terms would be very tedious and involved. It is fortunate, therefore, that many sound sources can, to a good degree of accuracy be approximated by one or a combination of several idealised theoretical sources such as the monopole or the dipole (Ref. 1).

1.19. MONOPOLE OR SIMPLE SOUND SOURCE

The simplest type of sound source for generating spherical acoustical waves is a pulsating sphere. Due to it's symmetry, this simple or monopole source will produce harmonic spherical waves in any surrounding medium that is homogeneous and isotropic. Many practical sources behave, at least to a first approximation, like a monopole providing their dimensions are small compared with the wavelength of the radiated sound (Fig. 1.7).

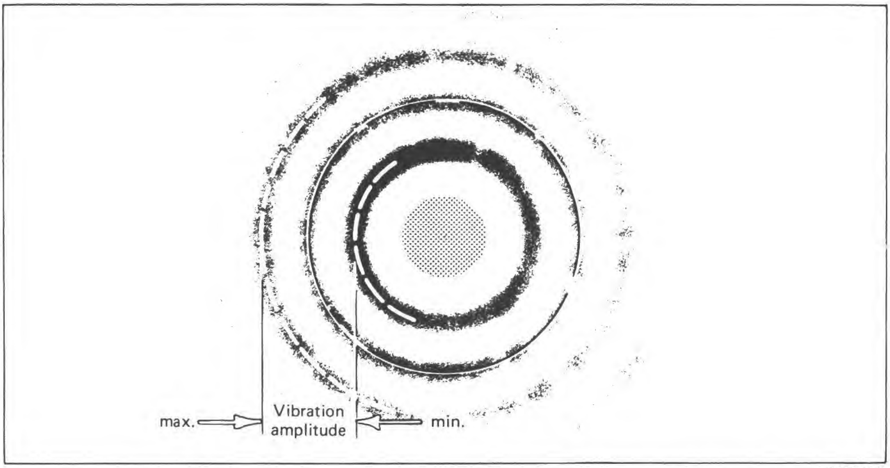

Fig. 1.7. Monopole source — a pulsating sphere

As the distance from the source, increases, the area over which the emitted energy is spread is increased and consequently the resulting sound intensity is reduced.

For a monopole radiating harmonic spherical waves the sound intensity is

$$I = \frac{W_m}{4\pi r^2} \tag{1.17}$$

where I = Sound intensity ($W.m^{-2}$)
W_m = Sound power (W)
r = Distance from monopole (m)

Taking logarithms to the base 10 and using the relationship between sound intensity and sound pressure yields

$$\begin{aligned} SPL &= SWL - 20\log_{10}(r) - 10\log_{10}(4\pi) \\ SPL &= SWL - 20\log_{10}(r) - 11dB \end{aligned} \tag{1.18}$$

where SPL = Sound pressure level re 2×10^{-5} Pa
SWL = Sound power level re 10^{-12} Watt
r = Distance from the source in metres

Therefore, for spherical radiation the SPL decreases by 20 log(2) = 6 dB each time the distance from the source is doubled. This result is known as the inverse square law.

The mean square pressure measured at a distance r from a sphere of mean radius a and pulsating with a frequency f, assuming that ka << 1 rad, is given by

$$p^2 = \frac{1}{16}\left[\frac{\rho c k Q}{\pi r}\right]^2 \tag{1.19}$$

where $k = \dfrac{2\pi f}{c}$ wave number ($rad.m^{-1}$)

c = speed of sound ($m.s^{-1}$)
ρ = density of air ($Kg.m^{-3}$)
Q = strength of the source ($m^3.s^{-1}$)
r = distance from the source (m)

The strength of the source, Q, is defined as the product of the surface area and the rms velocity amplitude of the surface, U, that is

$$Q = 4\pi a^2 U \tag{1.20}$$

N.B. The quantities Q, p, U must all be of the same type i.e. all peak or all rms.

The intensity, I, of the sound waves produced by a monopole source is

$$I = \frac{p^2{}_{rms}}{\rho c} \tag{1.21}$$

From the above equations it is seen that both the mean square pressure and the intensity decrease as one moves away from the source by an amount proportional to the square of the distance from the source.

The sound power, W_m, radiated by the source is

$$W_m = \frac{\rho c k^2 Q^2_{rms}}{4\pi} \tag{1.22}$$

which is independent of the radial distance as required by the principle of the conservation of energy.

Probably the simplest practical source which behaves like a monopole is a loudspeaker mounted in a suitably absorbant enclosure and situated in free space. When the wavelength of the sound produced is much greater than the dimensions of the source i.e. when $ka \ll 1$, the sound pressures measured at distances from the source such that $r \gg a$, are found to be the same for all sources of equal strength regardless of the shape of the radiating surface.

1.20. DIPOLE

Consider a loudspeaker mounted in an open backed cabinet. The radiation pattern produced will be the result of the radiation from both sides of the loudspeaker cone. As the dimensions of the baffle become small compared to the wavelength of the sound emitted then the radiation pattern approaches that of a dipole or acoustic doublet. The situation can be thought of as two monopoles 180° out of phase, of source strength Q separated by a small distance l ($k.l \ll 1$). The sound pressure, p_d, produced by such a system at a point $p(r,\theta)$ is

$$p_d = \frac{\rho c k^2 Q l}{4\pi r} \cdot \cos\theta \tag{1.23}$$

The quantities p_d and Q must be of the same type i.e. all peak or all r.m.s. This formula is only valid for $kr \gg 1$.

The sound pressure pattern radiated by a dipole is shown in Fig.1.8. The sound pressure field is symmetrical about the axis joining the two monopoles which in practice is the axis of the loudspeaker.

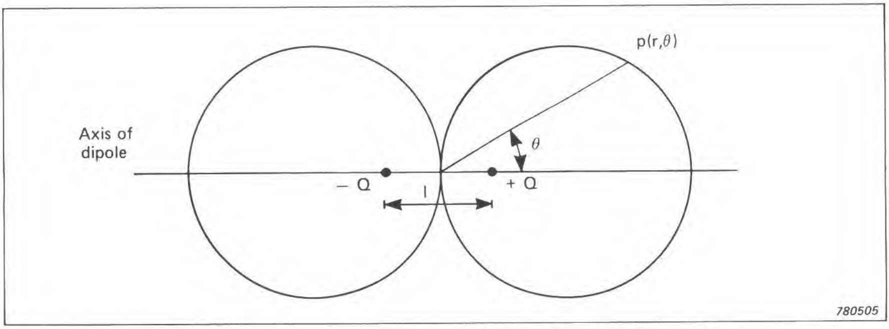

Fig. 1.8. Sound pressure pattern radiated by a dipole

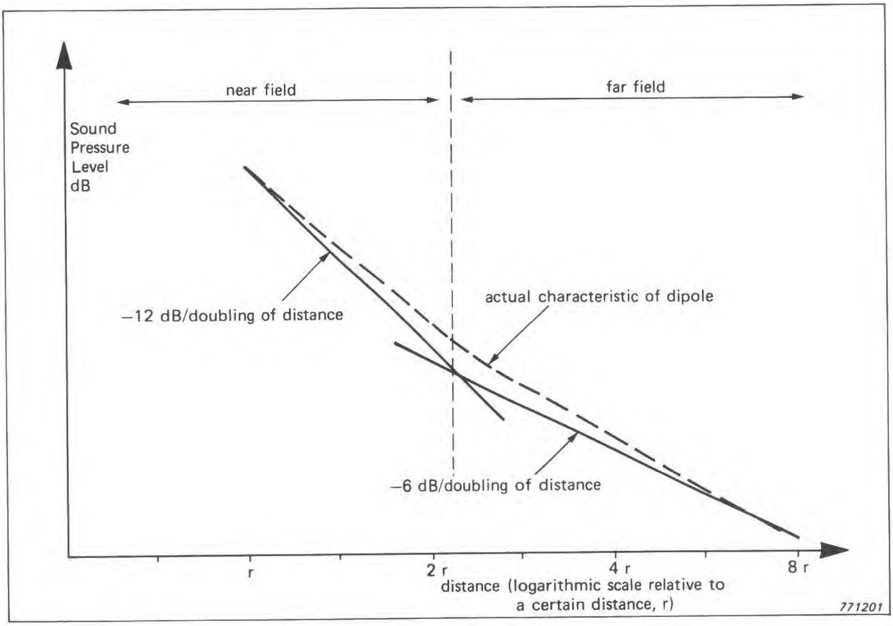

Fig. 1.9. Variation of sound pressure level with distance from a dipole

The variation of sound pressure as a function of distance from the dipole is shown in Fig. 1.9. Close to the dipole (the near field) the sound pressure decays rapidly, $p \propto 1/r^2$, while far from the dipole (the far field) the sound pressure decays more slowly, $p \propto 1/r$. Providing that the sound pressure is measured in the far field, the intensity of the dipole source, I_d, may be written as

$$I_d = \frac{p^2_{rms}}{\rho c} \qquad (1.24)$$

The sound power output of the dipole, W_d, is given by

$$W_d = \frac{\rho c k^4 Q^2 l^2}{12\pi} \qquad (1.25)$$

where Q is an r.m.s. value.

In the derivation of the foregoing equations it is assumed that the distance l is very small i.e. $l \ll 1/k$.

On comparing the sound power of the monopole and the dipole using equations 1.22 and 1.25, one finds that

$$\frac{W_d}{W_m} = \frac{k^2 l^2}{3} \cdot \frac{Q_d{}^2}{Q_m{}^2} \qquad (1.26)$$

When the source strengths are equal (i.e. $Q_m = Q_d$) then since the wave number, k, is proportional to the frequency, it can be seen that at low frequencies the dipole is far less efficient in radiating sound energy than the monopole. This is the reason why a loudspeaker mounted in a cabinet can reproduce low frequencies whereas a loudspeaker without a baffle cannot.

Practical sources which radiate like a dipole include a loudspeaker without a baffle, pure tone fans, a tuning fork and a vibrating beam.

Monopoles and dipoles have been employed to form more complex theoretical sources such as the lateral quadrupole and the longitudinal quadrupole but as these have no direct application in architectural acoustics they will not be discussed here.

1.21. DIRECTIVITY PATTERN

Many sound sources that are met in practice are more complicated than either the monopole or the dipole. Their behaviour has to be determined by measurement rather than by theoretical prediction. Measuring the sound pressure level at a fixed distance from a source, but in different directions, will generally yield different levels. The plot of these levels on a polar diagram is known as the directivity pattern of the source. A surface can therefore be described around a source over which the same sound pressure level exists. However, it is usually sufficient to specify the directivity pattern in the vertical and the horizontal directions Figs. 1.10 and 1.11).

Directivity patterns are extremely useful for supplying a lot of information about the nature of a sound source in a form which is easy to assimulate.

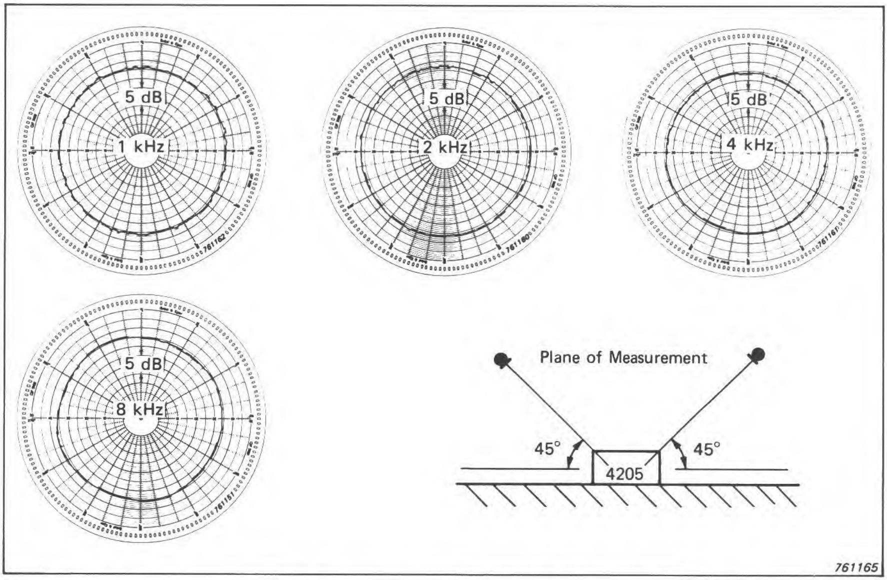

Fig. 1.10. Typical horizontal directivity patterns for the Sound Power Source Type 4205

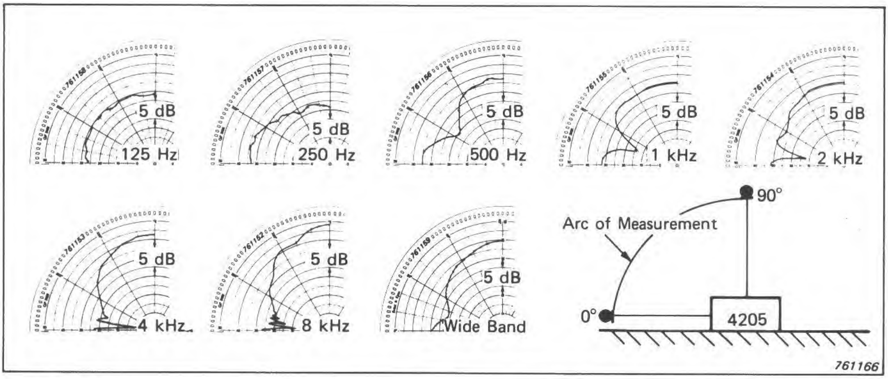

Fig. 1.11. Typical vertical directivity patterns for the Sound Power Source Type 4205

These patterns can and usually do vary with frequency; most sources being highly directive at high frequencies and non-directive or nearly so at low frequencies.

Besides describing the nature of such sources as loudspeakers, reference sound sources, machines etc., the concept of directivity patterns is used for describing the pressure response of microphones and sound level meters.

1.22. DIRECTIVITY FACTOR AND DIRECTIVITY INDEX

The directivity of a source can be specified either by the directivity factor, D, which is a dimensionless quantity defined by

$$D = \frac{I}{I_{ref}} \tag{1.27}$$

or by the directivity index, d, expressed in dB and defined by

$$d = 10 \log_{10} D = 10 \log_{10}\left(\frac{I}{I_{ref}}\right) \tag{1.28}$$

In these equations, I, is the intensity measured at some distance from the source in the direction in which the directivity is to be specified and I_{ref} is a reference intensity defined by

$$I_{ref} = \frac{W}{4\pi r^2} \tag{1.29}$$

where W = sound power output of the source.

Values of the directivity factor range from unity for the case of the monopole to large numbers for highly directive sources. The directivity also depends on the source position. Table 1.1 gives the directivity factor and directivity index for a monopole in various locations.

Source Location	Directivity Factor	Directivity Index, dB
Free field e.g. suspended between floor and ceiling	1	0
On a flat plane e.g. on the floor	2	3
At junction of two perpen. planes e.g. floor and wall	4	6
At junction of three perpen. planes e.g. in a corner	8	9 780118

Table 1.1. Directivity factor and directivity index for a monopole in various locations

From a knowledge of the sound power and the directivity pattern of a source, one can estimate the sound pressure levels generated by the source in almost any given acoustic environment. Such estimations can be used to position noisy machines in workshops and offices etc., where they would cause the least disturbance.

1.23. SOUND FIELD OF A SOUND SOURCE

The nature of the sound field around a sound source in a room consists of two components i.e. the direct field and the reverberant field. (Fig.1.12). The immediate vicinity of the source is known as the near field. In this region the particle velocity is not necessarily in the direction of propagation of the sound wave. Furthermore, the sound pressure may vary considerably with position and the sound intensity is not simply related to the mean square pressure. The extent of the near field is difficult to define as it depends on many factors such as the frequency, dimensions of the source and phases of the radiating surfaces. (Fig.1.13).

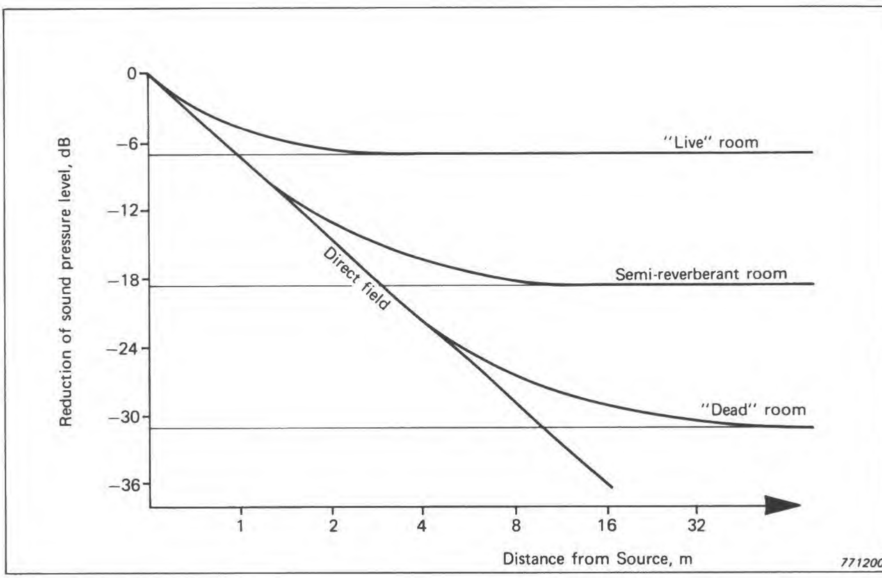

Fig.1.12. Combination of direct and reverberant fields

In the region known as the far field the sound pressure level decreases 6 dB each time the distance between the measuring microphone and the source is doubled providing the source is in free space which can be simu-

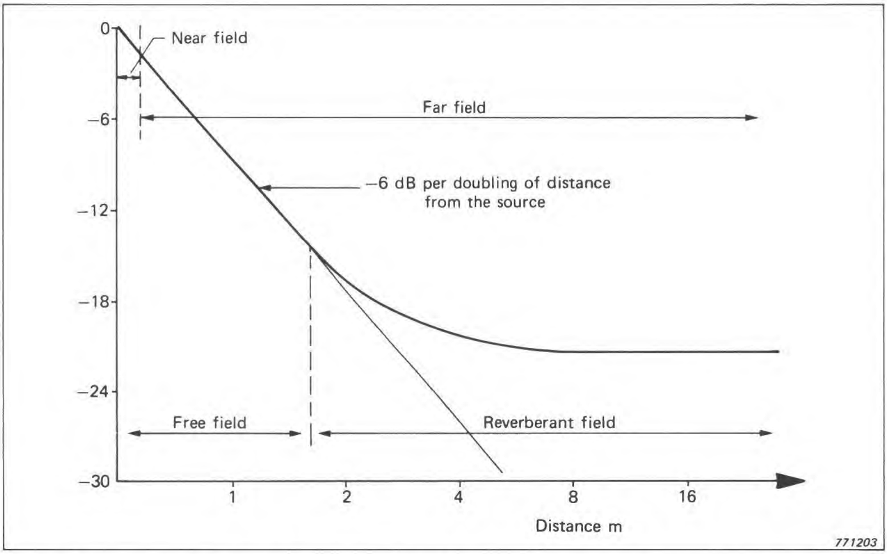

Fig. 1.13. Description of the sound field around a sound source in a reverberant room

lated in an anechoic room. In this region, the particle velocity is principally in the direction of propagation and the intensity is proportional to the mean square pressure.

If the source is radiating into a normal room then reflections of the sound waves from the room's boundaries create a reverberant field which is superimposed on the source's far field. In a highly reverberant or "live" room, the reverberant field may swamp the far field altogether. The reverberant field is referred to as a diffuse field if the sound energy density in this region is very nearly uniform.

For a spherically radiating source it has been shown that

$$I_d = \frac{W}{4\pi r^2} \qquad (1.30)$$

where I_d = sound intensity due to the direct field

The sound intensity in the reverberant field, I_r, can be expressed as

$$I_r = \frac{4W}{R} \qquad (1.31)$$

where R is the room constant defined by

$$R = \frac{S\bar{\alpha}}{1 - \bar{\alpha}} \qquad (1.32)$$

where S = total surface area of room
$\bar{\alpha}$ = average room absorption coefficient

Combining these two expressions yields

$$I = \frac{WD}{4\pi r^2} + \frac{4W}{R} \qquad (1.33)$$

where D = directivity factor

Taking logarithms to the base 10 of this equation and expressing the result in terms of sound power gives

$$SPL = SWL + 10\log_{10}\left(\frac{D}{4\pi r^2} + \frac{4}{R}\right)\ dB \qquad (1.34)$$

where SPL = sound pressure level re 2×10^{-5} Pa
SWL = sound power level re 10^{-12} Watt

Obviously, the closer to the source, the greater the effect of the direct sound on the measured sound pressure level. At a certain distance from the source, however, the contributions to the sound pressure level from both the direct and reverberant field will be equal. This distance, known as the room radius, can be calculated by setting

$$\frac{D}{4\pi r^2} = \frac{4}{R} \qquad (1.35)$$

therefore the room radius, r_r, is given by

$$r_r = \sqrt{\frac{RD}{16\pi}} \qquad (1.36)$$

1.24. THE EAR

Some knowledge of the ear's charactertistics is necessary to fully appreciate the problems which arise in architectural acoustics. The range of sound pressures to which the ear can respond is prodigious. The ear can withstand sounds of pressure amplitudes in excess of 100 Pa and yet can detect sound pressures of 0,00001 Pa. Such small sound pressures, in the ear's most sensitive range which is from 1000 Hz to 5000 Hz, produce a displacement of the eardrum of the order of 10^{-11} m. This minute distance is approximately one tenth of the diameter of a hydrogen molecule. The ear, however, is more than just an extremely sensitive microphone. It is also, together with the brain a frequency analyser capable of fine discrimination between tones.

The minimum intensity level perceptable by the ear at a particular frequency is known as the threshold of hearing or threshold of audibility at that frequency. The threshold of hearing varies from person to person even among people who have "normal" hearing. This threshold is also a function of the age of the listener, the progressive loss in sensitivity at the higher frequencies with age is called presbycusis. From Fig.1.14 one can see that it requires nearly a million times more power to produce an audible sound at 50 Hz than at 3000 Hz.

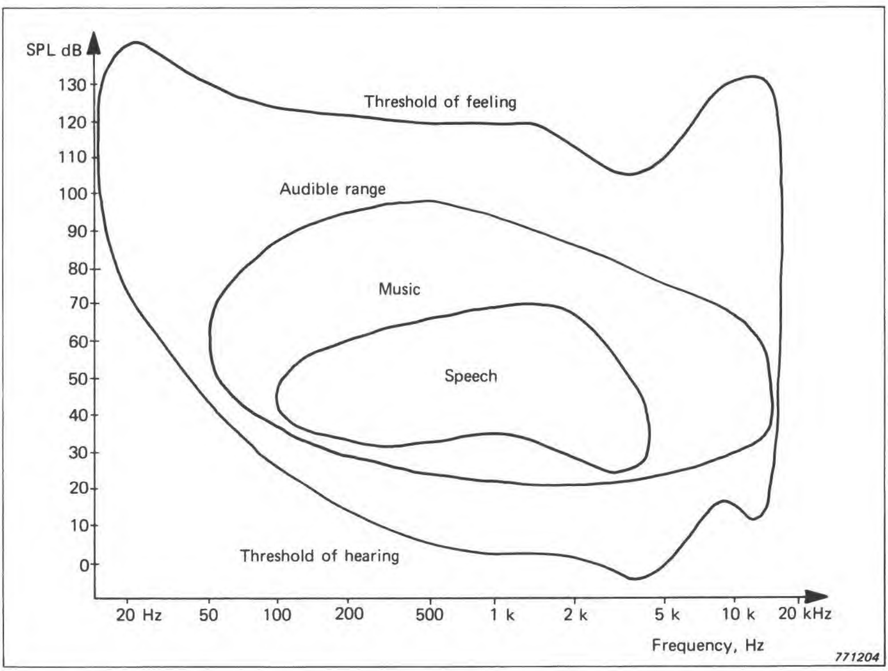

Fig.1.14. Audible range of frequencies and sound pressure levels bounded by threshold of hearing and the threshold of feeling together with approximate regions for speech and music

As the intensity of the acoustic waves incident on the ear is increased, the sound perceived by the ear becomes louder and louder until the sensation ceases to be one of hearing but one of "tickling" or feeling within the ear. This level is known as the threshold of feeling. It is less dependent on frequency than the threshold of hearing and has a value of about 120 dB. The approximate limits of the frequencies and the intensities normally experienced in speech and music are also shown in Fig. 1.14.

1.25. LOUDNESS, PHONS AND SONES

The subjective characteristics of a sound known as the loudness is a function of the sound's intensity and frequency. From Fig. 1.14 for example, one can see that a pure tone having an intensity level of 20 dB and a frequency of 1000 Hz would be clearly audible whereas a tone having the same intensity but a frequency of 100 Hz would not be heard at all as it lies well below the threshold of hearing.

From the results of many subjective experiments, contours of equal loudness can be drawn on an intensity vs frequency diagram as shown in Fig. 1.15.

The unit of loudness level is the phon. The loudness level in phons of any sound is defined as being numerically equal to the intensity level in decibels of a 1000 Hz tone that is judged by the average observer to be equally loud.

As the apparent loudness of a sound is not directly proportional to the sound's loudness level, subjective experiments have been performed in order to establish a scale on which a doubling of the number of loudness units doubles the subjective loudness and trebling of the number of loudness units trebles the subjective loudness and so on.

The unit of loudness is the sone. A sone is defined as the loudness of a 1000 Hz tone having an intensity level of 40 dB, i.e. a sone is equal to the loudness of any sound having a loudness level of 40 phons.

No instrument exists which can actually measure the loudness of a composite sound which contains many frequency components. However, sound level meters can make precise measurements of the sound pressure level of such sources from which reasonably accurate estimates of the loudness can be made.

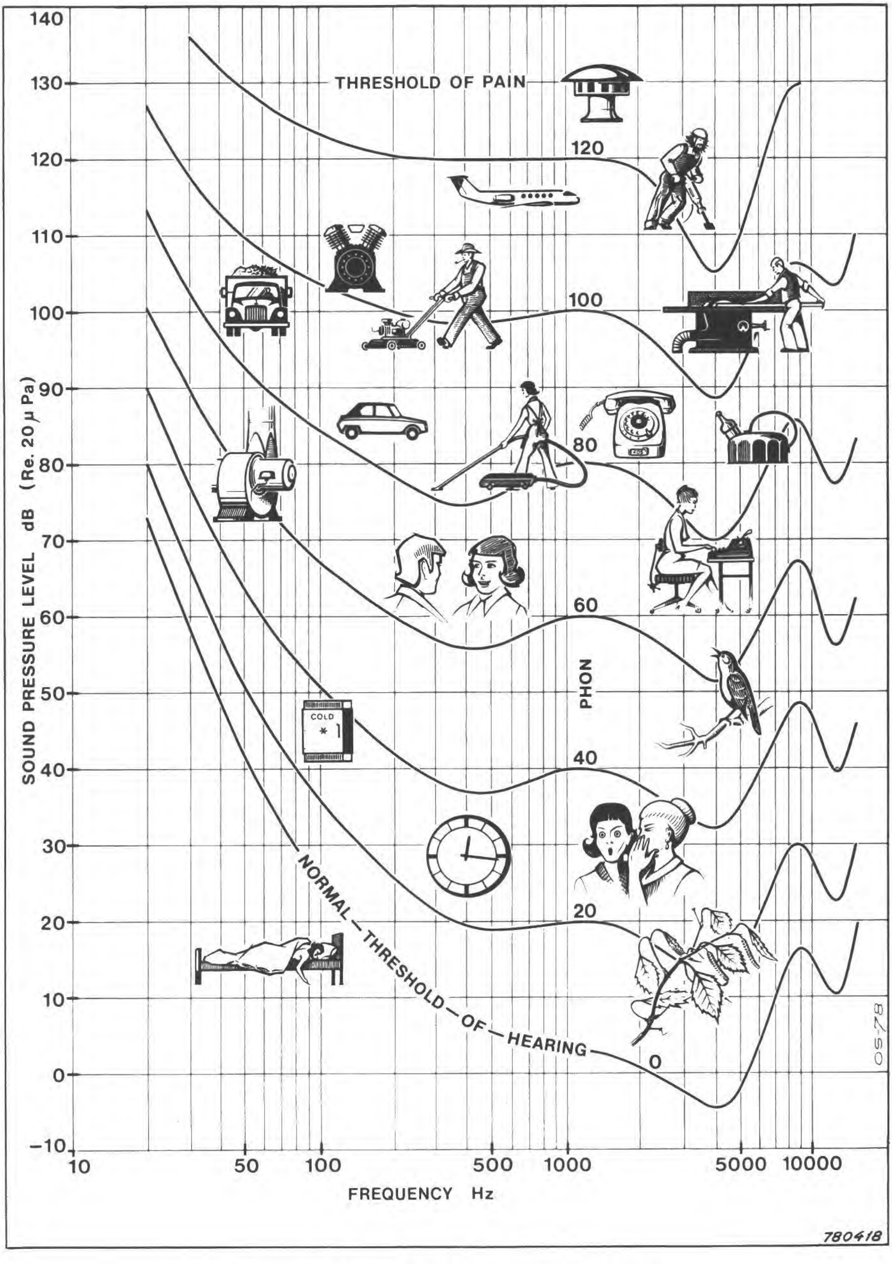

Fig. 1.15. Equal loudness contours

1.26. SOUND LEVEL METERS AND WEIGHTING NETWORKS

A sound level meter consists essentially of a high quality microphone, a linear amplifier, one or more attenuators, a set of frequency weighting networks including a linear response and an indicating meter or an LED (light emitting diode) digital display.

The purpose of the weighting networks is to make the readings on the indicating meter correspond as closely as possible to the perceived loudness levels. The frequency characteristics of these weighting networks are therefore approximately the inverse of the corresponding equal loudness contours shown in Fig.1.16. Ideally the sound level meter would contain a network possessing a frequency response which would be continuously variable over the dynamic range. In practice, however, it is sufficient to use only a few fixed networks. These were originally defined as:

A-weighting:	used for loudness levels below 55 phons.
B-weighting:	used for loudness levels between 55 and 85 phons.
C-weighting:	used for loudness levels over 85 phons.
D-weighting:	used to account for the increase in annoyance produced by the high frequency whine present in the noise produced by certain aircraft.

However, more recent work has not substantiated these historical associations so that the frequency weightings are now largely conventional. Furthermore, the A-weighting is now frequently specified for rating sounds irrespective of level and is no longer restricted to low level sounds.

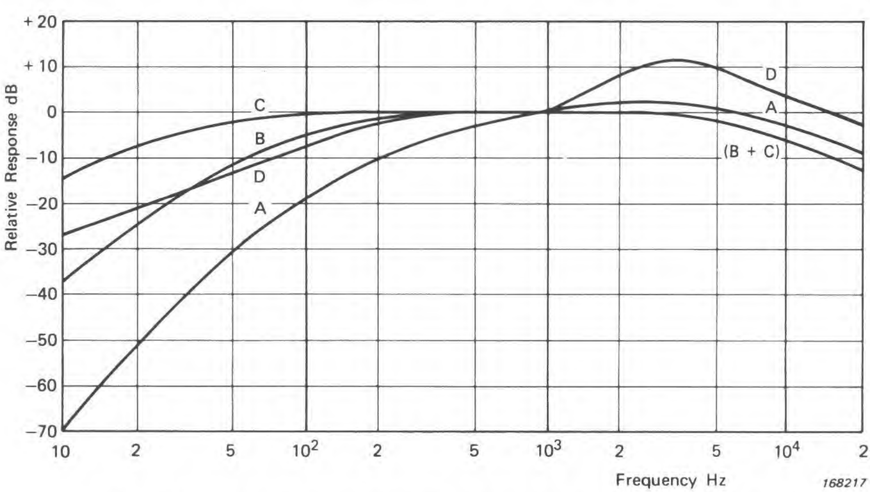

Fig.1.16. Frequency response of the weighting networks

2. ACOUSTICS OF ROOMS

2.1. DEFINITION OF ROOM ACOUSTICS

Consider a sound source which is situated in a room. Sound waves will propagate away from the source until they encounter one of the room's boundaries where, in general, some of the sound energy will be reflected back into the room, some will be absorbed and some will be transmitted through the boundary. The complex sound field produced by the multitude of reflections and the behaviour of this sound field as the sound energy in the room is allowed to build up and decay constitutes the acoustics of the room.

2.2. GEOMETRICAL ROOM ACOUSTICS

If one can assume that the dimensions of a room are large compared to the wavelength of sound then one may treat the sound waves in the room in much the same way as light rays are treated in geometrical optics. This situation frequently occurs in architectural acoustics. In analogy with light rays, sound rays are reflected from hard plane walls in accordance with the laws of reflection i.e. the incident ray, the reflected ray and the normal to the surface at the point of incidence all lie in the same plane; the angle of incidence is equal to the angle of reflection (Fig. 2.1). Therefore sound rays incident on

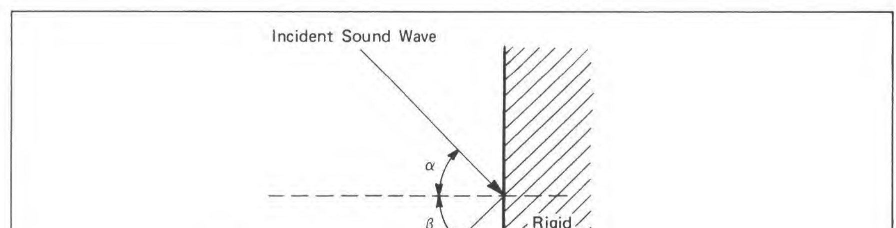

Fig. 2.1. Laws of reflection

a curved surface will either be focused or dispersed depending on whether the surface is concave or convex (Fig.2.2). Diffraction of sound rays can and does occur but the effect is more noticeable for low frequency, long wavelength sounds than with high frequency sounds of short wavelength.

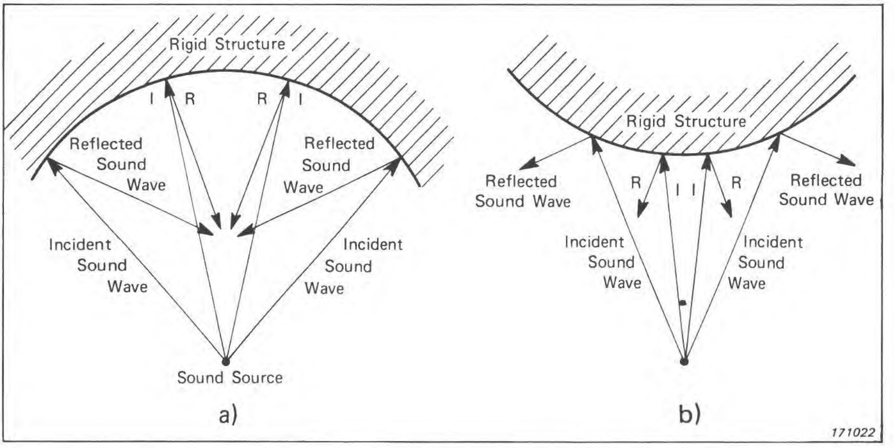

Fig.2.2. Reflections of sound rays

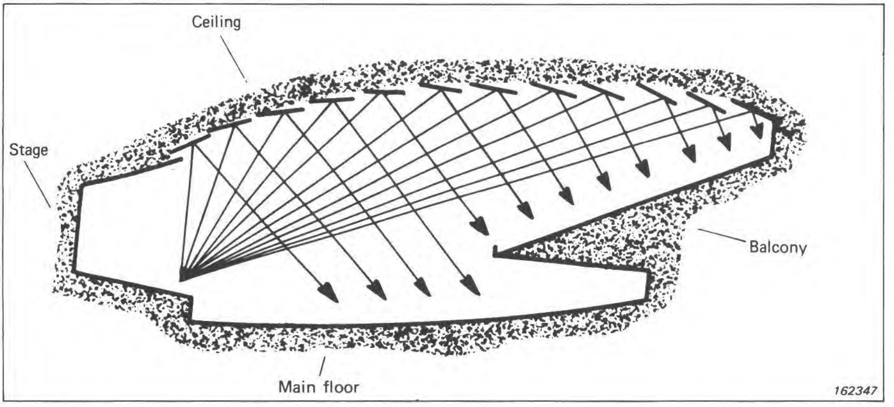

Fig.2.3. Graphical construction of the first reflections of the sound waves in a concert hall

The concept of a sound ray and the geometrical study of sound ray paths plays an important role in the design of large rooms and auditoria, enabling troublesome echoes and flutter effects to be detected and dealt with at the stage of designing the building. Fig.2.3 shows how geometrical constructions can be used to position sound reflectors on the ceiling of a concert hall in order to improve the distribution of sound. A limitation of the geometrical approach is that usually only the primary and possibly the secondary reflections can be studied before the sound ray being followed becomes "lost" in the reverberent sound field.

2.3. GROWTH AND DECAY OF SOUND IN A ROOM

When a sound source is placed in a room, the sound intensity as measured at a particular point will increase in a series of small increments, due to the reflections arriving from the walls, floor and ceiling, until an equilibrium position is attained where the energy absorbed by the room is equal to energy radiated by the sound source. When the sound source is abruptly switched off the sound intensity in the room will not suddenly disappear but will fade away gradually, the rate of decay being prescribed by the amount and position of the absorbing material in the room. This lingering of the sound is known as reverberation. The rate of absorption of sound energy in the room will be, in the main, proportional to the sound intensity so that the growth and decay of sound pressure in the room is an exponential function of time (Fig.2.4.).

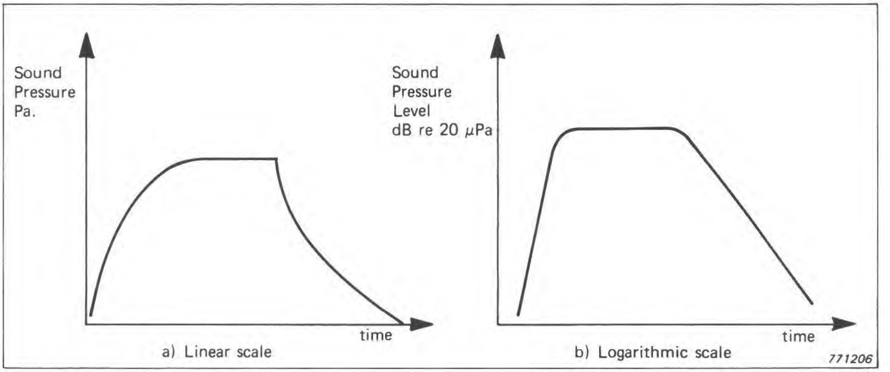

Fig.2.4. Growth and decay of sound in a reverberant room

If one measures the sound pressure levels in dB in a decaying reverberant field as a function of time then one obtains a reverberation curve which is usually a fairly straight line although the exact form depends on many factors including the frequency spectrum of the sound source and the shape of the room.

2.4. REVERBERATION TIME

At the beginning of this century W.C. Sabine carried out a considerable amount of research on the acoustics of auditoria and arrived at an empirical relationship between the volume of the auditorium, the amount of absorptive material within the auditorium and a quantity which he called the reverberation time. This relationship is now known as the Sabine formula:

$$RT = \frac{0{,}161\ V}{A} \tag{2.1}$$

where RT = the reverberation time defined as the time taken for a sound to decay by 60 dB after the sound source is abruptly switched off

V = the volume of the auditorium in m^3

A = the total absorption of the auditorium in m^2-sabins.

The absorption unit of $1\,m^2$-sabin represents a surface capable of absorbing sound at the same rate as $1\,m^2$ of a perfectly absorbing surface e.g. an open window.

2.5. ABSORPTION COEFFICIENT

Material	Frequency, Hz					
	125	250	500	1000	2000	4000
Air, per cu. m.	nil	nil	nil	0,003	0,007	0,02
Acoustic paneling	0,15	0,3	0,75	0,85	0,75	0,4
Plaster	0,03	0,03	0,02	0,03	0,04	0,05
Floor, concrete	0,02	0,02	0,02	0,04	0,05	0,05
Floor, wood	0,15	0,2	0,1	0,1	0,1	0,1
Floor, carpeted	0,1	0,15	0,25	0,3	0,3	0,3
Brickwall	0,05	0,04	0,02	0,04	0,05	0,05
Curtains	0,05	0,12	0,15	0,27	0,37	0,50
Total absorption of one seated person	0,18	0,4	0,46	0,46	0,51	0,46

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Table 2.1. Typical absorption coefficients of $1\,m^2$ of material

The absorption coefficient of a material, as originally defined by Sabine, is the ratio of the sound absorbed by the material to that absorbed by an equivalent area of open window hence the absorption coefficient of a perfectly absorbing surface would be 1.Providing one knows the superficial areas and the absorption coefficients of the various materials to be used, the reverberation time of an auditorium can be determined at the design stage. To facilitate such calculations sets of tables have been published giving the absorption coefficients of the commonly used building materials as a function of frequency. The variation in the reverberation time of specially designed reverberant rooms (also known as live or hard rooms) as one introduces or removes absorptive material is a standard method for determining absorption coefficients (refer to the International Standard ISO 354). (See Chapter 6).

2.6. DERIVATION OF FORMULAE FOR REVERBERATION TIME

Theoretical derivations of Sabine's formula are usually based on geometrical acoustics utilising the assumptions that the sound in the enclosure is diffuse and that all directions of propagation are equally probable. This is a gross simplification of the actual behaviour of sound in enclosures because it neglects such important factors as room modes, the positioning of absorptive material, the influence of the shape of the room and others. For fairly reverberant rooms with a uniform distribution of absorptive material, Sabine's formula gives a good indication of the expected behaviour of sound in the room. As a room becomes more and more "dead" i.e. the boundaries become more and more absorbent, so the results obtained from employing Sabine's formula become more and more inaccurate. In the limiting case of a completely dead or anechoic room where the absorption coefficient of the boundaries is unity then the reverberation time is obviously zero because a reverberant field cannot exist in these conditions. However, for this situation Sabin's formula gives a finite value of 0,161 V/A for the reverberation time. Several different approaches have been used to derive equations which give values of reverberation time which are in better agreement with the measurement results for rooms containing little absorption. As examples, two such formulae are quoted here.

Eyring's formula for reverberation time is

$$RT = \frac{0{,}161\ V}{-\ S\ ln(1-\bar{\alpha})} \tag{2.2}$$

where

$\bar{\alpha} = \dfrac{\alpha_1 s_1 + \alpha_2 s_2 + \cdots + \alpha_n s_n}{s_1 + s_2 + \cdots + s_n}$ the mean absorption coefficient of the room.

$S = s_1 + s_2 + \cdots + s_n$ the areas of the various materials

$\alpha_1, \alpha_2, \ldots \alpha_n$ the respective absorption coefficients.

Eyring's formula gives results which are in much better agreement with the measured reverberation times for dead rooms than Sabine's formula. Also, the Eyring formula gives the correct value of RT = O for an anechoic room i.e. for $\alpha = 1$. One drawback of this improved formula is that it is only strictly valid for rooms which have the same value of α for all boundaries.

The theory of Millington and Sette leads to the formula

$$RT = \frac{0{,}161\ V}{\sum - s_i\ ln(1 - \alpha_i)} \tag{2.3}$$

where s_i = the area of the i^{th} material
α_i = the absorption coefficient of the i^{th} material.

When the materials in a room have a wide variety of absorption coefficients then the best predictions of reverberation times are obtained by employing the Millington and Sette formula. This formula can be obtained by substituting the effective sound absorption coefficient $\alpha_e = -\ln(1-\alpha_i)$ into Sabine's formula.

Millington and Sette's formula indicates that highly absorbing materials are more effective than would be anticipated in influencing the reverberation time. For example, when the absorption coefficient of a material is greater than 0,63 then the effective absorption coefficient is seen to be greater than one.

2.7. WAVE THEORY OF ROOM ACOUSTICS

Although very useful in certain applications, the geometrical approach to room acoustics is not a satisfactory method for attempting to explain the behaviour of sound within an enclosure. A more adequate but more difficult approach is based upon wave acoustics, that is, upon the motion of sound waves within a three dimensional enclosure. This method is characterised by the establishing of boundary conditions which describe mathematically the acoustical properties of the walls, ceiling and the other surfaces in the room. The difficulty involved in determining these boundary conditions for rooms of irregular shape e.g. churches, auditoria, rooms containing furniture etc., means that an analysis employing the wave theory can only be performed exactly for a number of idealised situations. Although the practical application of the wave theory is limited, an understanding of the theory is essential in order to appreciate many of the problems which arise in the acoustics of rooms, e.g. why the frequency response of a loudspeaker varies from room to room; why there are pressure maxima and minima within a room where there is a steady sound source.

When employing the wave theory, a room is considered as a complex resonator possessing many normal modes of vibration and each mode having a characteristic frequency of damped free vibration. These modes can be excited by introducing a sound source into the room. The acoustic energy supplied by the source can be considered as residing in the standing waves established in the room. The characteristic frequencies of vibration of the standing waves depend principally on the room's size and shape whereas the damping of these waves depends mainly on the boundary conditions. Using the simplest possible boundary conditions of rigid walls and no damping, one can derive the characteristic frequencies of a room. The effect of damping can then be taken into account by considering it to be a modification of these simple conditions.

For a rectangular room a simple relationship exists between the room dimensions, l_x, l_y and l_z, and the frequencies corresponding to the normal modes of vibration of the room (See Appendix for the derivation). This relationship is

$$f = \frac{c}{2}\left[\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2\right]^{1/2} \tag{2.4}$$

A normal mode of vibration occurs for every permutation of n_x, n_y and n_z where less indices have the integer values 0, 1, 2, 3...etc. An example of the use of this expression is given in Table 2.2 where the characteristic frequencies below 100 Hz have been calculated from eqn.2.4 taking c as 340 $m.s.^{-1}$, for the large reverberation room at the Danish Technical Highschool. The dimensions of the room are 7,85 × 6,25 × 4,95 m. The room, acting as a resonator, responds strongly at these resonant frequencies. The normal modes of a room are not distributed evenly over the frequency spectrum. At low frequencies there are very few modes but the number of modes increases rapidly with frequency. The modes can occur in bunches around a particular frequency e.g. in Table 2.2 the characteristic frequencies 2,0,0 and the 0,1,1 modes are very close to each other while there is a considerable gap between the 43,5 Hz and the next characteristic frequency which occurs at 48,9 Hz. It is this "preferential" treatment of certain frequencies by rooms which affects the output of loudspeakers. Every room imposes its own characteristics on those of any sound source present, so that the fluctuations in sound pressure which occur as a microphone is moved from one position to another, or as the source frequency is varied, may completely obscure the true output characteristics of the source. If the response curves of loudspeak-

n_x	n_y	n_z	f (Hz)	n_x	n_y	n_z	f (Hz)
1	0	0	21,7	1	1	2	77,0
0	1	0	27,2	2	2	1	77,6
0	0	1	34.3	3	1	1	78,4
1	1	0	34,8	2	0	2	81,2
1	0	1	40,6	0	3	0	81,6
2	0	0	43,3	1	3	0	84,4
0	1	1	43,8	3	2	0	84,7
1	1	1	48,9	2	1	2	85,6
2	1	0	51,1	4	0	0	86,6
0	2	0	54,4	0	2	3	87,6
2	0	1	55,3	0	3	1	88,5
1	2	0	57,6	1	2	2	90,3
2	1	1	61,1	4	1	0	90,8
0	2	1	64,3	1	3	1	91,1
3	0	0	65,0	3	2	1	91,4
1	2	1	64,9	2	3	0	92,4
0	0	2	68,7	4	0	1	93,2
2	2	0	69,5	3	0	2	94,5
3	1	0	70,4	4	1	1	97,1
1	0	2	72,0	2	2	2	97,7
3	0	1	73,5	3	1	2	98,4
0	1	2	73,9	2	3	1	98,6

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Table 2.2. Characteristic frequencies below 100 Hz as calculated for the reverberation room of dimensions 7,85 × 6,25 × 4,95 m at the Danish Technical Highschool

ers have to be measured then the measurements should be performed either in the open air or in an anechoic chamber.

The three types of normal modes which occur in the room are:

1. Axial modes — The component waves move parallel to an axis (one dimensional, two of the indices n_x, n_y and n_z are equal to zero). These are the modes which give rise to flutter echoes. The pressure pattern for an axial mode is shown in Fig.2.6.

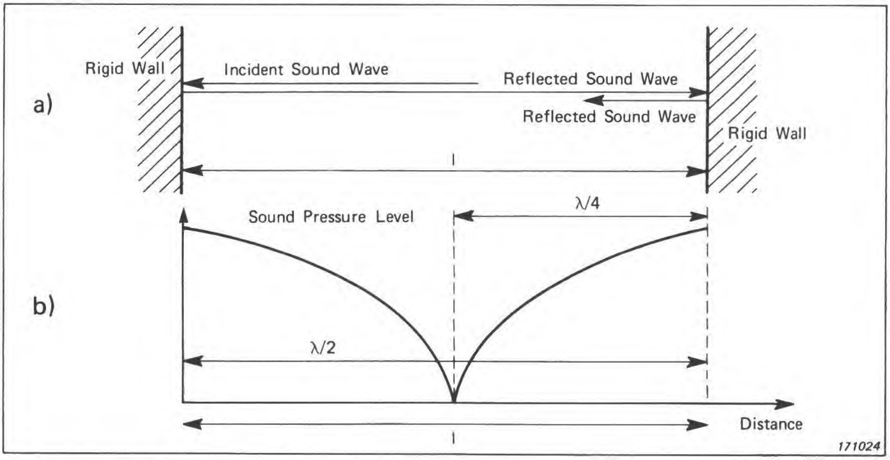

Fig.2.6. Axial mode with corresponding pressure pattern

2. Tangential modes — The component waves are tangential to one pair of surfaces but are reflected from the other two pairs (two dimensional, one of the indices n_x, n_y and n_z is equal to zero).

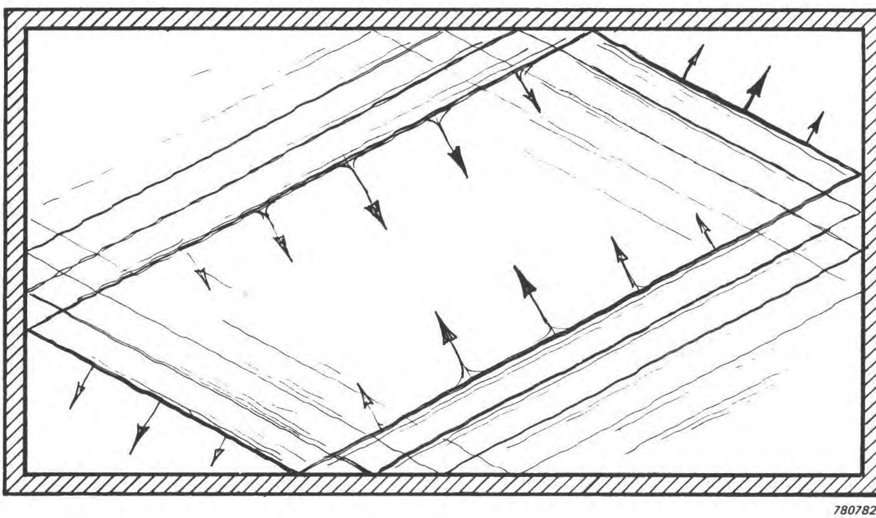

Fig.2.7. Tangential mode

3. Oblique modes — The component waves are oblique and therefore impinge on all six walls (three dimensional, all the indices n_x, n_y and n_z have finite values).

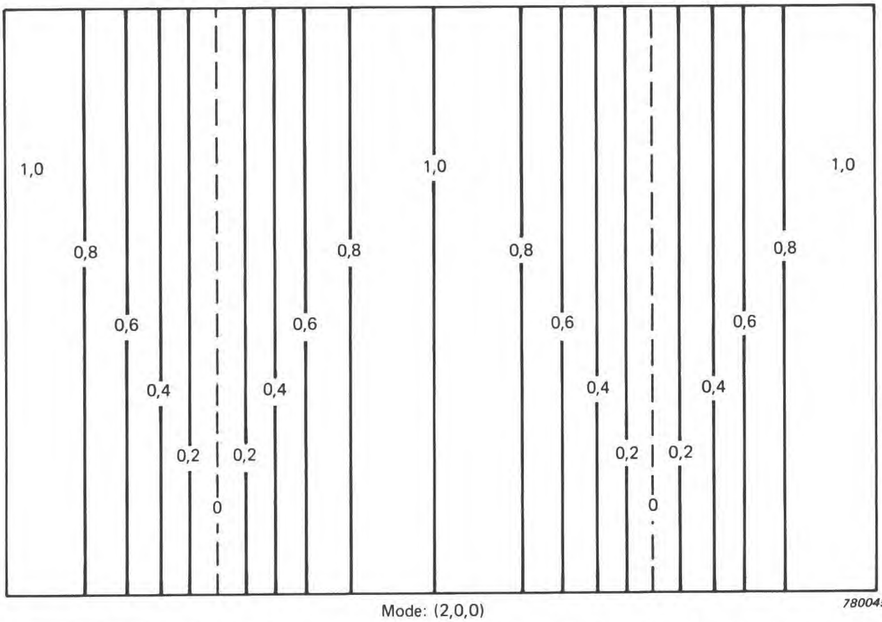

Fig.2.8. Sound pressure contours on a section through a rectangular room for the axial mode (2,0,0)

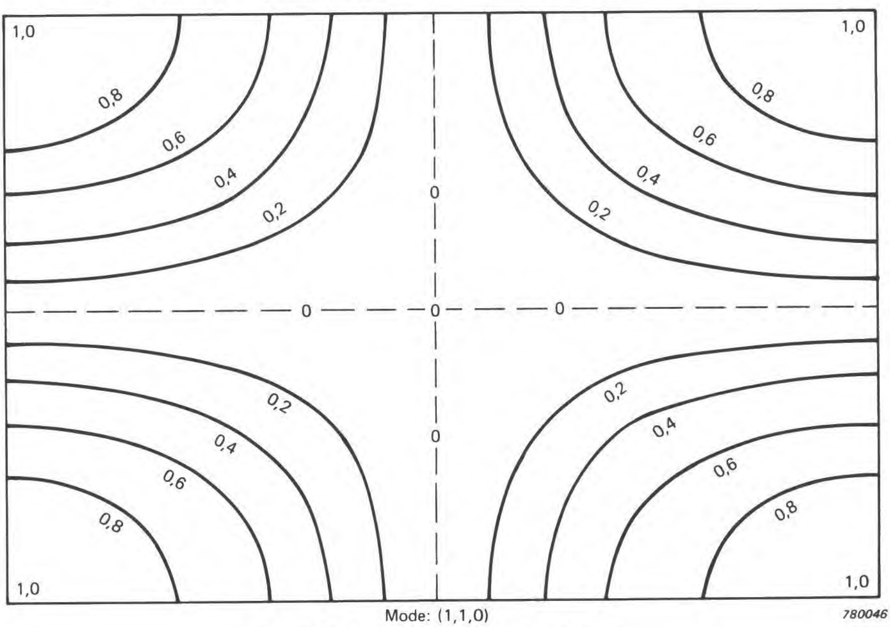

Fig.2.9. Tangential mode (1,1,0)

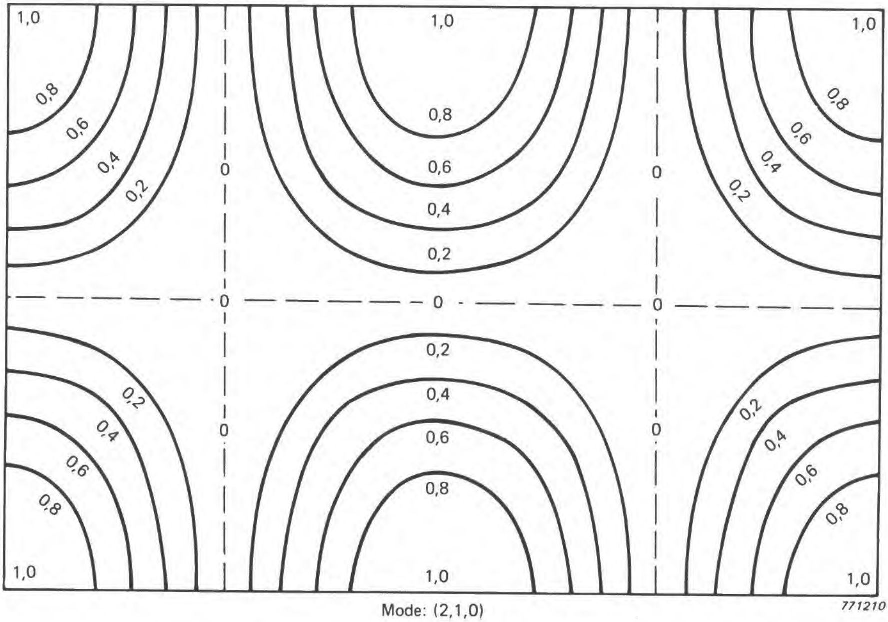

Fig. 2.10. Tangential mode (2,1,0)

If one plots the sound pressure contours for various modes of vibration (Figs.2.8, 2.9 and 2.10) then one can see that for every mode the sound pressure is a maximum in the corners of the room. Furthermore, at the geometric centre of the room, only one eighth of the modes have a finite sound pressure. Therefore, if a monopole sound source be placed in the corner of such a room, it is possible to excite every mode of vibration to its fullest extent. Similarly, if a microphone be placed in a corner of the room, it will measure the peak sound pressure for every mode that has been excited. However, when the monopole source is placed in a position where a particular mode has a pressure node, then this mode of vibration will be excited only weakly if at all.

Each of the characterisitc frequencies, f, given by eqn.2.4 may be considered as a vector in frequency space having the co-ordinates $n_x c/2l_x$, $n_y c/2l_y$ and $n_z c/2l_z$. A normal mode of vibration is represented by a point in this frequency space (Fig.2.11). The length of the line joining the point to the origin is a measure of the characteristic frequency of this particular mode. All the normal modes of a particular charactertistic frequency f and below are included within the octant of frequency space between the positive axes f_x, f_y and f_z and the spherical surface of radius f. The number and types of modes having characteristic frequencies lying within a particular frequency interval can be determined by counting the number of characteristic points on

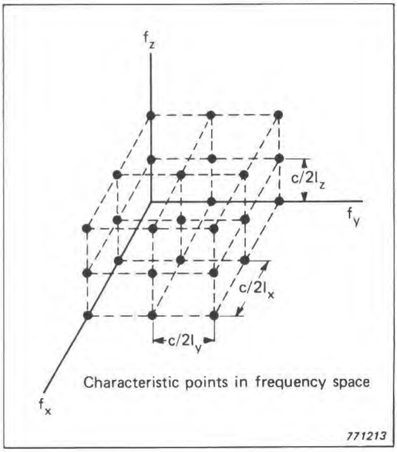

Fig.2.11. Characteristic points in frequency space

a frequency space diagram. As this is a tedious task, a number of equations have been derived which yield approximate values for such counts. The number of normal modes, N, below a particular frequency, f, is given by

$$N \simeq \frac{4\pi V f^3}{3c^3} + \frac{\pi S f^2}{4c^2} + \frac{Lf}{8c} \tag{2.5}$$

where V = volume of the room (m^3)
S = $2(l_x l_y + l_y l_z + l_x l_z)$, the total surface area of the room (m^2)
L = $4(l_x + l_y + l_z)$, the sum of the lengths of all the edges (m)
N = number of normal modes

For the reverberation room of dimensions 7,85 × 6,25 × 4,95m mentioned previously the number of normal modes below the frequency of 100 Hz as calculated from eqn. 2.5 is 44,8 which is in good agreement with the exact value of 44.

On differentiating eqn. 2.5 with respect to f one obtains an expression which yields an approximate value for the number of modes, ΔN, in the band of frequencies, Δf, which is centred on f.

$$\Delta N \simeq \left[4\pi V\left(\frac{f}{c}\right)^3 + \frac{\pi S}{2}\left(\frac{f}{c}\right)^2 + \frac{L}{8}\left(\frac{f}{c}\right)\right]\frac{\Delta f}{f} \tag{2.6}$$

For the reverberation room described above, the number of modes in the third octave band (i.e. Δf is 23% of f) centred on 100 Hz as calculated from eqn.2.6 is 25. Counting the number of modes from Table 2.2 yields the number 28. For the same room, the number of modes predicted by eqn.2.6 in the third octave band centred on 1000 Hz is over 18000. This means that there is a fairly even modal distribution at these higher frequencies and that the spacing between the characteristic frequencies is so close that specific resonances may be neglected.

Fig.6.28 shows the resonances of a room of dimensions 2,5 × 3 × 7 m when excited by a pure tone from a loudspeaker, the electrical power delivered to the loudspeaker being kept constant. The mode numbers are written above the corresponding resonance peaks.

If the room possesses walls which are capable of absorbing sound energy then the effect of this damping on the standing waves in the room can be investigated by introducing a damping term, $e^{-\beta t}$, into the undamped standing wave equation (eqn. A.8) i.e. by replacing $j\omega$ by $j\omega - \beta$.

It is theoretically possible to apply boundary conditions at the various wall surfaces and thus determine the characteristic frequencies of the normal modes. Due to the complexity of the problem, however, at present a solution can only be obtained for particularly simple configurations of sound absorbing material. The mathematical argument for the case of damped vibration is similar to that for the undamped case except that the boundary conditions in the damped case depend on the absorbing characteristics of the walls. These characteristics are best described by using the concept of the normal specific acoustic impedance, z_n, which is the ratio of the pressure to the normal air velocity at the surface of the wall (see definition Ch.1). This quantity is both more fundamental and more readily measured than the absorption coefficient of a surface. The normal specific acoustic impedance of a material relative to air is given by

$$z_n = (r_n + jx_n)\rho c \tag{2.7}$$

For many materials that are used for wall surfaces $r_n \gg x_n$ and $r_n \gg 1$. For these particular conditions one finds that the characteristic frequencies of the damped and the undamped waves are the same.

The damping of the various types of waves depends on the positioning of the absorbing material. Absorbing material is most effective when positioned in the corners of a room because it is here that every normal mode has a pressure maximum.

2.8. PRINCIPLES FOR DESIGN OF ROOMS AND AUDITORIA

The size of a room or auditorium and the amount of money to be spent on the construction are two factors which are usually decided upon before a specialist is consulted to assist in designing the acoustics of the building. The principle acoustical design factor is the reverberation time. Opinions vary considerably as to what is the optimum value of reverberation time for an auditorium intended for a particular use. Generally though, one can say that the reverberation times for speech and recorded music should be as short as possible, as one is only interested in the direct and not the reverberant sound; for light music they should be short and for concert and church music they should be long. By measuring the reverberation times in auditoria which are considered to possess good acoustic qualities, one can arrive at a relationship between the "optimum" reverberation time for a particular use and the volume of the auditorium (Fig.2.12). It should be noted that these curves are intended only as a guide and that the amount of scatter about these mean curves is large.

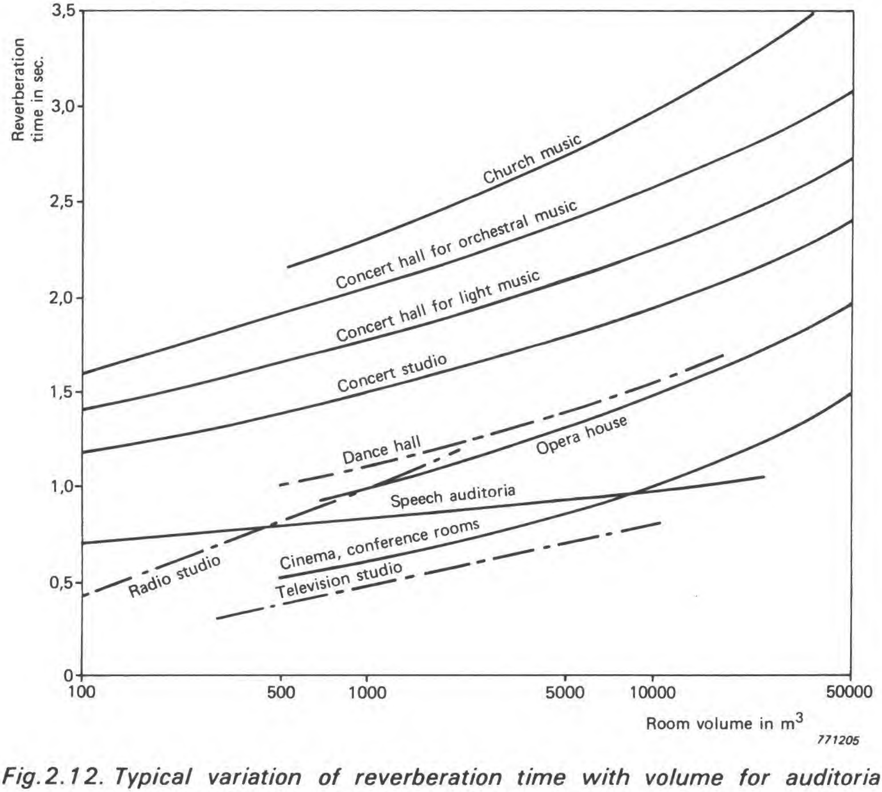

Fig.2.12. Typical variation of reverberation time with volume for auditoria considered to have good acoustical properties

The quality of the acoustics of an auditorium, however, does not only depend on the reverberation time but also on the shape and size of the enclosure, the positioninq of the sound absorbing material and the positioning of the sound source and the audience.

The acoustical defects which can arise due to the size and shape of the room are echoes, dead spots and flutter.

An echo occurs when a strong reflection of an original impulse is heard by a listener after an interval of greater than 0,05 sec since he heard the original impulse. The ear functions in such a way that when the interval between the original and the reflected sound is less than 0,05 sec, then the reflected sound is not recognised as a separate echo but adds to the apparent strength of the original sound. This phenomenon is known as the Haas effect.

Dead spots can occur at positions in auditoria which are far from reflecting surfaces and which receive sound only after it has passed over a particularly absorbing surface. Dead spots can occur, for example, at the back of cinemas and theatres where the sound travels at almost grazing incidence over the audience.

The phenomenon of flutter occurs when both the source and the listener are between a pair of parallel, hard, surfaces and other nearby surfaces are fairly absorbent. Sound emitted by the source will tend to be "trapped" between the reflecting surfaces and will oscillate back and forth and decay relatively slowly. The listener will perceive this oscillating energy as a "fluttering" of the sound.

2.9. DESIGN OF ROOMS FOR SPEECH

When designing a room for speech, the most important criterion is that the speaker should be distinctly and readily heard by all members of the audience. A quantitative measure of the degree of clarity at various positions in the room can be obtained by articulation tests. These tests consist of reading aloud from the speaker's platform a list of monosyllabic nonsensical words. The people in the audience then write down what they think they have heard. An analysis of the percentages of consonants and vowels which were heard correctly is then performed to give "the percentage articulation index" (P.S.A.). Normal, connected speech can be understood even if some of the syllables are unintelligible. This is due to the fact that the listener can deduce the meaning from the context of the sentence. Even under perfect listening conditions, the maximum value of P.S.A. obtained is normally about 95% due to unavoidable errors. A P.S.A. of 80% enables the audience to understand every sentence without undue effort. In a room where the P.S.A. is about 75%, the listener has to concentrate to understand what is said while below 65% the intelligibility is too poor.

Assuming there is no speech amplification system present, there are basically four factors which affect the clarity of speech in a room. Firstly, there is the background noise level which can mask the desired sound. This level should be kept below 30 dBA. Secondly, there is the sound pressure level produced at the listener's ear by the speaker. This depends on the distance between the listener and the speaker, the volume of the room and the nature of the speaker's surroundings. Thirdly, there is the reverberation time. In normal speech the syllables follow each other with rapidity. Unless each syllable decays fairly quickly it will tend to mask those following. Therefore the reverberation time must not be too long. On the other hand, if the reverberation time tends to zero then the absence of reflected components severely restricts the size of a room in which the speech will be heard at a sufficiently high intensity for good intelligibility. Lastly, there is the room shape, although providing this has been designed to avoid echoes and dead spots and each member of the audience has a good view of the speaker then the articulation should not be affected.

A compromise has therefore to be made between a loss of definition due to excessive reflected sound and a loss of sound intensity due to excessive absorption at the room's boundaries.

In practice a speaker may adapt himself to a particular hall by speaking more slowly and loudly when addressing a large audience but there are limits to the amount of accommodation possible and a speech amplification system may have to be employed.

Once one knows the volume that the room is to be and the required reverberation time, one is in a position to decide how much sound absorbent treatment is necessary. The positioning of the absorbing material within the room will depend on the particular circumstances but in general the greater part of the material should be placed at the end of the room opposite to the speaker and mounted on those surfaces which are likely to give rise to troublesome reflections. Absorbent materials of a resilent nature such as perforated boards or wooden slats backed with mineral wool should be used on walls where they are liable to suffer damage or wear. Less robust materials, such as acoustic fibre tiles, should only be used out of hand's reach.

There are certain rooms which are used for speech which require special attention such as rooms used for debate, theatres, lawcourts and multipurpose halls e.g. school halls, community halls etc.

In debating chambers each person must have good sight lines to all the others present and the intelligibility must be acceptable at all positions in the chamber. In theatres it is important that the natural qualities of the performers' voices be maintained. However this requirement of naturalness does not apply to public meeting halls. Rooms used for both speech and music have conflicting requirements (i.e. a short reverberation time to render speech intelligible and a long reverberation time to give depth to music) so that a compromise has to be decided upon. If a sound amplification system is to be used then this compromise is not such a critical decision. The reverberation time can be made long enough for music and the sound amplifying system used to overcome the long reverberation time. The subjective reverberation time of an auditorium can be adjusted to suit the programme by employing electroacoustical techniques, for example, the assisted resonance technique.

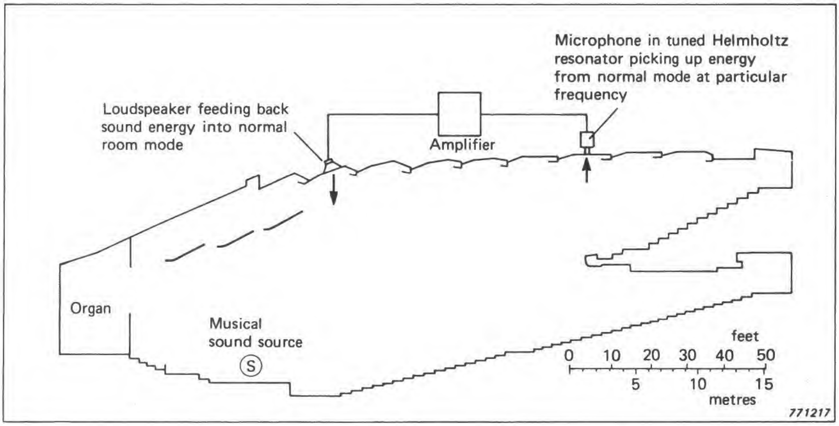

Fig. 2.13. Principle of the assisted resonance technique

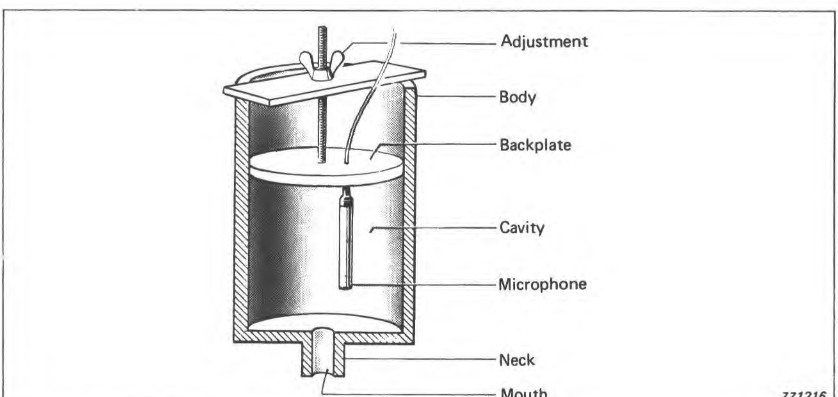

Fig. 2.14. Helmholtz resonator used in the assisted resonance technique

This technique increases the reverberation time by prolonging the dying away of the room modes or resonances in the auditorium. For a particular mode whose reverberation has to be "assisted" a Helmholtz resonator is positioned in the auditorium where there is a pressure maximum for this particular frequency (Fig.2.13). The resonator is then tuned to this frequency by adjusting the volume of the resonator. A microphone placed inside the cavity of the resonator detects the selected frequency and feeds the signal via an amplifier to a loudspeaker which then emits acoustical energy at this frequency back into the auditorium. This "topping up" with acoustical energy, compensates to a controllable extent for the sound energy absorbed by the audience and the room surfaces. The rate of decay of sound is thus slowed down and the reverberation time is therefore lengthened. To deal with a range of frequencies, many microphone-loudspeaker systems have to be employed. The Royal Festival Hall in London, for example, has 172 such channels covering the frequency range from about 60 Hz to 700 Hz. The resonators can be placed anywhere in the auditorium except close to the source of the music because it is the reverberant field, not the direct field, that the resonators must respond to. The loudspeakers can be placed anywhere in the auditorium (except very close to the listeners) because with these tuned circuits it is impossible to tell where the extra sound energy is coming from. This system has been in use in the Royal Festival Hall, London, since 1962.

2.10. DESIGN OF ROOMS FOR MUSIC

It is much more difficult to state criteria for good listening conditions for music than for speech because aesthetic and emotional judgements are involved. The criteria are almost totally subjective making them very difficult to define and often impossible to measure. The design of rooms for music is therefore as much an art as a science.

The notable survey by L.L.Beranek of sixty concert halls situated all over the world enabled 18 acoustical factors to be listed in order of their importance. From the results of his researches, Beranek developed a system of rating concert halls in terms of these 18 factors. These researches are described in the book "Music, acoustics and architecture" by L.L.Beranek published by John Wiley and Sons, New York, 1962.

Some of the more important factors will be discussed here.

Loudness

The music in the room must be sufficiently loud. As the sound energy available from a musical instrument is limited, this therefore sets a limit on the size of the auditorium.

Reverberation

There should be sufficient reverberant sound. The amount of reverberant sound required depends upon the nature of the music. Music from the baroque period or chamber music requires a short reverberation time, whereas music from the classical period such as orchestral works by Tchaikovsky and Wagner demand longer reverberation times.

Definition

The music should possess definition or clarity. This quality is basically the ability of the listener to differentiate between the different instruments in an orchestra and between the different musical sounds. Definition contradicts the requirement for sufficient reverberant sound.

Fullness of tone

This quality describes the blending effect that reverberation has on successive notes and chords when heard in a room. Fullness depends mainly on the reverberation time. The longer the reverberation time (within reason) the better the chance of obtaining adeuate fullness.

Definition and fullness are interelated. As for speech, definition depends on listeners receiving the direct sound and first reflections of it arriving not more than 35 ms later at a strength well above the reverberant sound level. On the other hand fullness appears to depend mainly on having plenty of reverberant sound. Thus it is likely that definition will suffer if the reverberation is made great enough for maximum fullness.

No obvious faults

There should not be any obvious faults in the music room such as noticeable echoes and dead spots.

Intimacy (or presence)

This quality relates to the sense of being enclosed in a space and enveloped in the sound field. Sound must be reflected from many surfaces to the listener from many directions so that he will sense the space in which he is sitting. Intimacy depends upon the time delay between the direct sound and the first reflection. It is found that in narrow halls having a very small initial time delay of about 15 ms the sense of intimacy in the centre of the main floor is extremely high.

Musicians' criteria

The musicians themselves have two important criteria to be satisfied. Firstly, the room should respond to their instruments. This means that some sound should be reflected back to them from the listening area but care is needed to avoid echoes. Secondly, the musicians should be able to hear each other. Reflecting surfaces around the orchestra will help to accomplish this.

The ultimate test for a room or hall intended for music is of course to listen to a live performance. An experienced listener moving about the hall during the test performance should be able to detect the presence of any faults. It is then the job of the acoustical consultant to suggest remedial measures.

2.11. REFLECTORS, ABSORBERS AND RESONATORS

Sound reflectors and sound absorbers are often used to produce the desired acoustical conditions in rooms and auditoria. Some of the properties and applications of sound reflectors and absorbers will be discussed here.

Sound reflectors

The ratio of the amount of reflected to incident sound at the boundary between two media depends on their relative acoustical impedances. For a material to be a reflector of sound its impedance will therefore have to be different to that of air. As impedance is equal to the product of density and the speed of sound in medium, reflectors in an airborne sound field should be massive. Geometrical reflection of the incident sound wave can only be assumed when the reflecting surface is large compared to the wavelength of the incident energy, thus practical reflectors in a room will have a definite low frequency cut off below which they act as diffusers. Generally speaking, a panel with a minimum dimension of about 30 times the wavelength of the incident sound acts as a reflector; when the minimum dimension is about 10 times the wavelength some diffraction occurs, while at less than 5 times the wavelength the incident energy is diffracted.

Many auditoria employ suspended overhead reflectors (e.g. the Albert Hall, London) in an attempt to combine a sufficiently large volume to obtain suitable decay times for music and short path reflections to produce good definition.

Sound absorbers

Every surface of a room and every object within it will absorb sound to some degree. Hard smooth objects such as walls and tiles will absorb far less sound energy than soft, porous materials such as carpets, upholstery and people. The porous sound absorbing materials used for the control of acoustical environments (e.g. mineral wool, glass fibre) are characterised by a number of variables, i.e. porosity, flow resistance and structure factor.

A sound wave impinging on this type of absorber causes the air in the connected pores to vibrate but as the movement of the air particles is restricted by the flow resistance of the material, some of the sound energy is dissipated as heat. The amplitude of vibration of the air particles is progressively damped by friction against the pore walls. This acts as an acoustical resistance depending chiefly on the resistance of the material to direct air flow.

The effective porosity of the material (i.e. the volume of pores connected to external air compared to the total volume of the material) governs the amount of airborne sound energy that may enter and be subjected to attenuation. However the structure factor is also important since air in blind alleys and pores which run parallel to the surface will be little affected by air flow in the main pores and have negligible effect on the absorptive effect.

In practice the actual absorption chracteristics of many of these sound absorbers depends on the method of mounting. Maximum sound absorption will occur in a porous absorber when the particle velocity in the absorber is at a maximum. In a sound wave which is incident on a rigid wall, the maximum

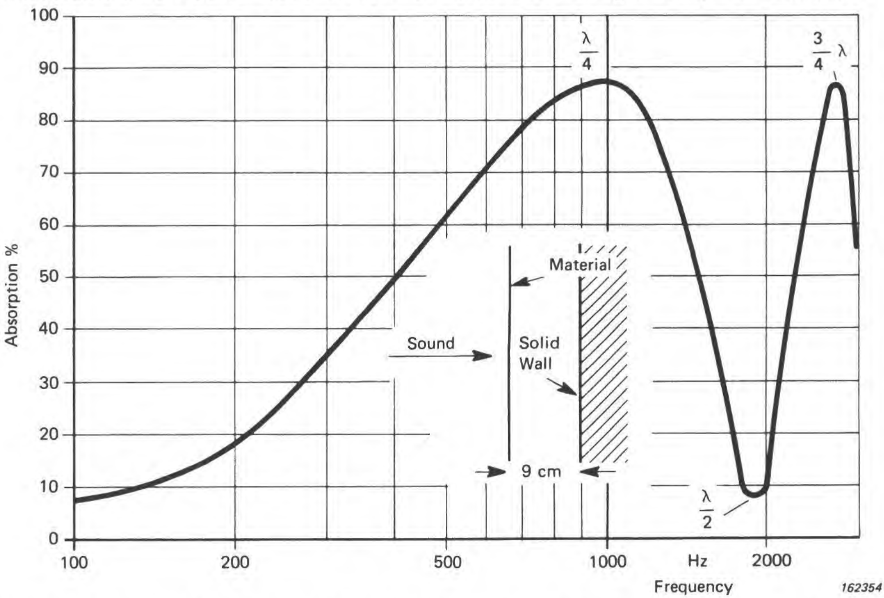

Fig.2.15. Absorption characteristics of an absorptive material mounted 9 cm from a solid wall

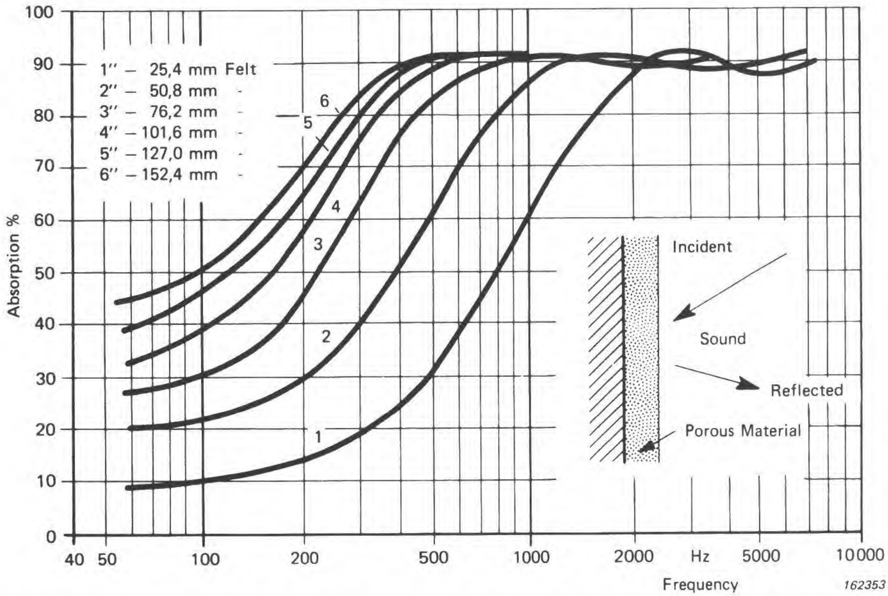

Fig.2.16. Dependence of the absorption of a porous material on the thickness of the material

particle velocity is to be found at 1/4 wavelength from the wall. Thus in order to absorb low frequency energy the absorber must either be very thick or else be mounted some distance in front of the wall. The absorption characteristics of a porous material mounted at a distance of 9 cm from a solid wall is shown in Fig.2.15. The greatest absorption occurs at frequencies for which the distance between the wall and the absorptive material corresponds to a 1/4 and a 3/4 wavelength of these frequencies. Fig.2.16 shows the dependence of the absorption of a porous material on the thickness of the material.

Panel sound absorbers

If a plate or panel is mounted in front of a rigid wall, the arrangement behaves in the same way as a spring-mass system, the plate being the mass and the enclosed air being the spring. When a sound wave impinges on the system, the system will tend to be set into vibration. The maximum transfer of energy occurs when the frequency of the incident sound energy is the same as the resonant frequency of the system.

Since the panel possesses inertia and damping, some of the sound energy is converted into mechanical energy and dissipated as heat, therefore sound absorption occurs. However since the panel itself vibrates it will act as a sound radiator so that it is rare to find such a system with an effective absorption coefficient greater than 0,5. The resonant frequency of such a system can be calculated from

$$f_{res} = \frac{6000}{\sqrt{md}} \tag{2.8}$$

where m = mass of plate in kilograms per square meter
d = distance between plate and wall in metres

Such panel resonators are useful at mid- and low-frequencies (Fig.2.17). The absorption of sound energy falls off rapidly at frequencies above the resonant frequency. Further damping may be obtained by introducing damping material into the air volume between the wall and the panel. This broadens the range of frequencies for which the resonator is active.

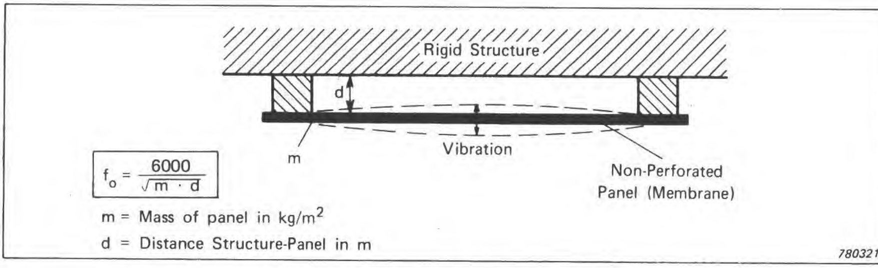

Fig.2.17. Typical panel absorber

Resonator absorbers

The simplest type of resonant absorber is called a Helmholtz resonator. This consists of a volume of air contained within a cavity connected to the air in the room by a small opening known as the neck (Fig.2.18). When a sound wave impinges on the aperture of the neck, the air in the neck will be set into vibration and the air in the cavity will undergo periodic compression and rarefaction. The friction between the resulting amplified motion of the air particles in the neck and the neck itself causes sound energy to be absorbed. The absorption of the undamped resonator falls off very rapidly at frequencies above and below the resonant frequency. Some energy is transferred into the cavity and this energy is *reradiated* after the incident sound has ceased. Thus the resonator may prolong the reverberation time of a room. The resonant frequency of an undamped resonator may be evaluated from

$$f_0 = \frac{c}{2\pi} \sqrt{\frac{S}{lV}} \tag{2.9}$$

where c = velocity of sound ($m.s.^{-1}$)
S = cross sectional area of neck (m^2)
l = length of neck (m)
V = volume of cavity (m^3)

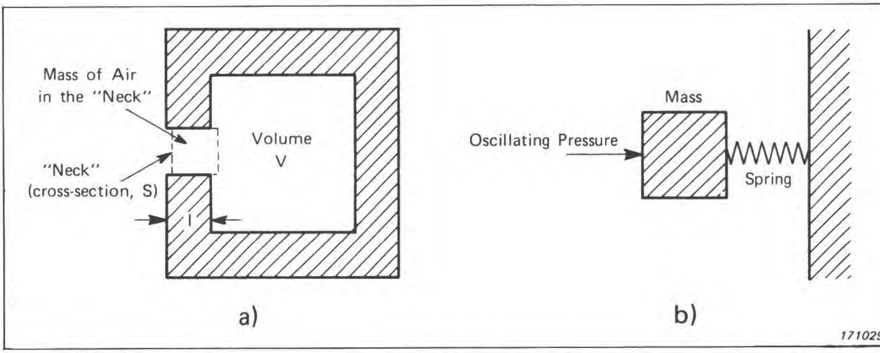

Fig. 2.18. Helmholtz resonator and the equivalent mechanical analogy

If the resonator be damped i.e. by lining the cavity and neck with a porous sound absorbing material then the resonator will be effective over a wider bandwidth although its maximum absorption at resonance will be reduced (Fig. 2.19). Helmholtz resonators can be designed to provide absorption at any point in the frequency scale but owing to their sharp tuning they are not often used for general acoustic treatment where a major change is required but only where a particularly long reverberation is experienced at a single fre-

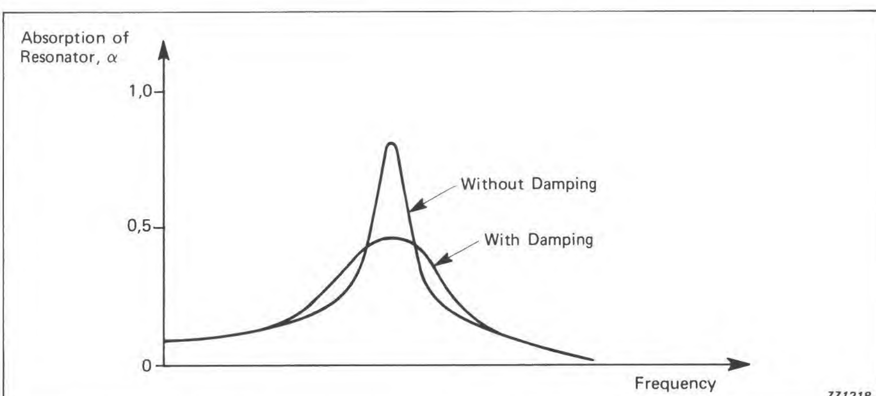

Fig. 2.19. Typical sound absorption characteristics of a Helmholtz resonator with and without damping

quency such as that due to a normal mode and it is desired to reduce this without greatly affecting the average reverberation.

Helmholtz resonators are most efficient at low frequencies.

Perforated panel absorbers

A much commoner application of the Helmholtz resonator principle for sound absorption is to be found in the acoustic panel. This consists of a panel or plate with a drilled or punched pattern of holes, mounted in such a way so as to enclose an air space between itself and the wall (Fig.2.20). The slots or holes form the necks of the resonators and the portion of the air space behind each hole forms the cavity of the resonator. Usually there is no need to divide the separate resonator cavities from one another by partitions. As for the single resonator, the resonant frequency of the multiple resonator is determined by the dimensions of the neck and the cavity but the multiple resonator is not so selective in its absorption. In practical acoustic panels, a damping material such as mineral wool or glass fibre is inserted into the air space thus increasing the effectiveness of the absorption above and below the resonant frequency (Fig.2.21).

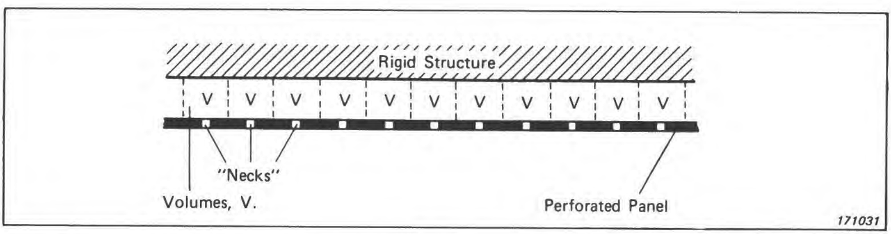

Fig.2.20. Perforated panel absorber seen as an assembly of Helmholtz resonators

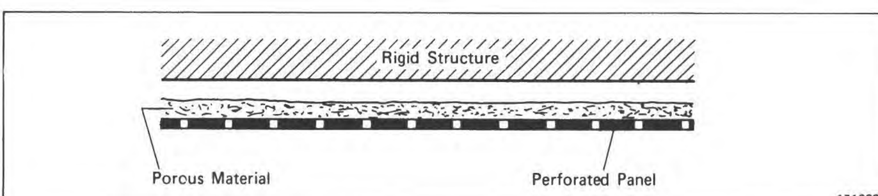

Fig.2.21. Perforated panel absorber lined with mineral wool

For panels with regularly spaced holes of the same diameter such resonator has the same resonant frequency. It is possible to alter the spacing and diameter of the holes to produce a panel with the desired absorption characteristics.

A commercial type of perforated panel absorber is the "Kulihat". This is a conical absorber, resembling a coolie's hat, consisting of two or three perforated aluminium sectors held together by steel clips. The inside of the Kulihat is lined with mineral wool (Fig.2.22 a) and b)). The absorption characteristics for two and three section Kulihats are shown in Fig.2.23.

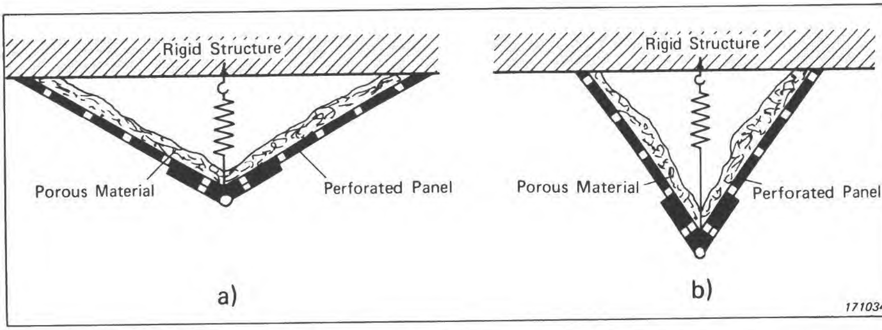

Fig.2.22. The "Kulihat" perforated sound absorber
a) three sectors b) two sectors

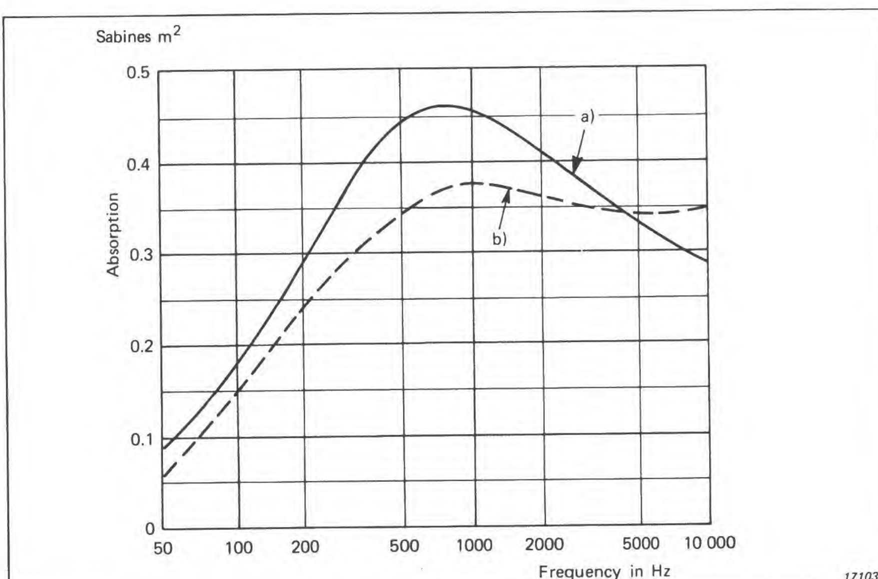

Fig.2.23. Sound absorption characteristics for "Kulihat" absorbers
a) three sectors b) two sectors

Functional absorbers

In some rooms and halls there are insufficient surfaces available for the mounting of the necessary sound absorbing material. In this situation functional absorbers can be employed. These are three dimensional units of sound absorbing material suspended freely in the room some distance from the room boundaries. Since sound energy can impinge on all sides of these units, their efficiency is high. This type of absorber is particularly useful in workshops, gymnasia, swimming pools etc.

3. ACOUSTICS OF BUILDINGS

3.1. INTRODUCTION

So far in this booklet, only the behaviour of sound within rooms and enclosures has been discussed. Now attention will be focussed on the transmission of sound from one room to another and the sound insulation properties of building elements i.e. walls, floors, doors and windows. Sound insulation is especially pertinent when designing multi-roomed buildings such as blocks of flats, hospitals, schools etc., where practically all intrusive noise is considered to be highly undesirable.

3.2. SOUND GENERATION MECHANICS

Sound can propagate throughout a building either via the air or via the building's structure (Fig.3.1.). Sound generation mechanisms can therefore be divided into two general groups.

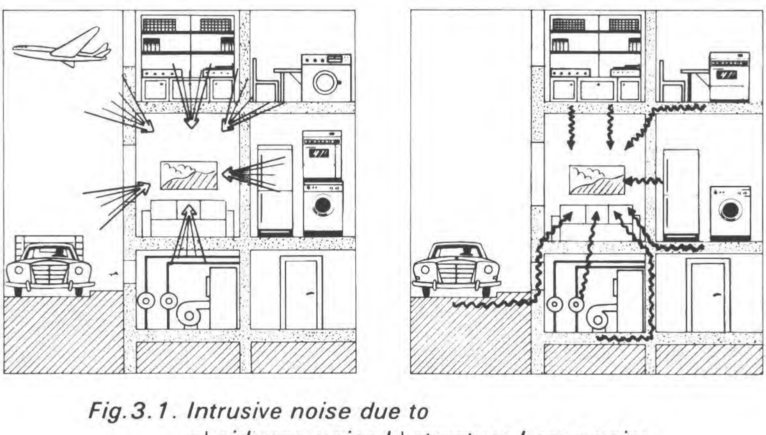

Fig.3.1. Intrusive noise due to
a) airborne noise b) structure borne noise

One group consists of those sources which generate sound directly into the air such as the voice, loudspeakers etc. Insulation against such sound is called airborne sound insulation.

The other group consists of those sources which act directly on the structure of the building usually by means of impact or vibrating equipment. Transmission of the sound is then through and from the structure. Examples are footsteps, noisy plumbing installations and slamming doors. In fact, this type of noise is really a combination of both airborne and impact noise because the impacts will produce airborne noise and this airborne noise will be transmitted. However, in nearly all cases the noise produced in the receiving room by the transmission of the impact noise will predominate. Insulation against such sound is called impact sound insulation.

3.3. SOUND INSULATION

When discussing the insulation that exists between two rooms against airborne or impact sound, it is usual to consider one room as the source room and the other as the receiving room. The basic problem is to determine (and/or reduce) the sound pressure levels produced in the receiving room due to a source acting in the other room. The simplest case to consider is the one where the two rooms have one common dividing element i.e. a wall or a floor/ceiling.

3.4. AIRBORNE SOUND INSULATION

A noise source operating in one room will produce a reverberant sound field which impinges on all surfaces of the room. The sound energy incident upon the dividing wall will depend on the sound power output of the source and the total sound absorption in the room. This incident sound energy will be partly reflected back into the room and partly "absorbed" by the wall, the ratio depending on the wall's acoustical impedance compared to that of air. Of the absorbed energy some will be dissipated as heat, and the rest will be propagated through the wall to the boundary with the receiving room. At this boundary, the relative impedances of the wall material and the air will again determine the percentage of energy transmitted into the receiving room. The overall effect is that the entire wall is forced into vibration by pressure fluctuations of the incident sound waves. (Usually the vibration of the wall can only be detected with the aid of an accelerometer or similar transducer but with particularly intense sound waves one can feel the wall vibrating.) The vibrating wall, acting in exactly the same manner as a loudspeaker, then radiates acoustical energy into the adjoining room.

The amount of radiation from the wall and hence the sound insulation provided by the wall depends on the frequency of the sound, the construction and material of the wall and above all on its weight. As one can imagine, the more massive the wall the more difficult it is for the sound waves to set it into vibration.

The sound insulation characteristic of a wall is usually expressed in terms of sound reduction index, R, (referred to as the transmission loss in the United States of America) expressed by

$$R = 10\,log_{10}\left(\frac{W_1}{W_2}\right)\ \text{dB} \tag{3.1}$$

where W_1 = sound power incident on the wall
W_2 = sound power transmitted through the wall

A typical sound insulation curve for a 15 cm concrete wall is shown in Fig.3.2. For example, if the sound reduction index of a particular wall were 40 dB then the incident sound energy would be reduced by 40 dB in transmission through it. Thus only one ten thousandth of the incident energy would be transmitted. The larger the value for the sound reduction index of a wall, the greater the sound insulation it provides.

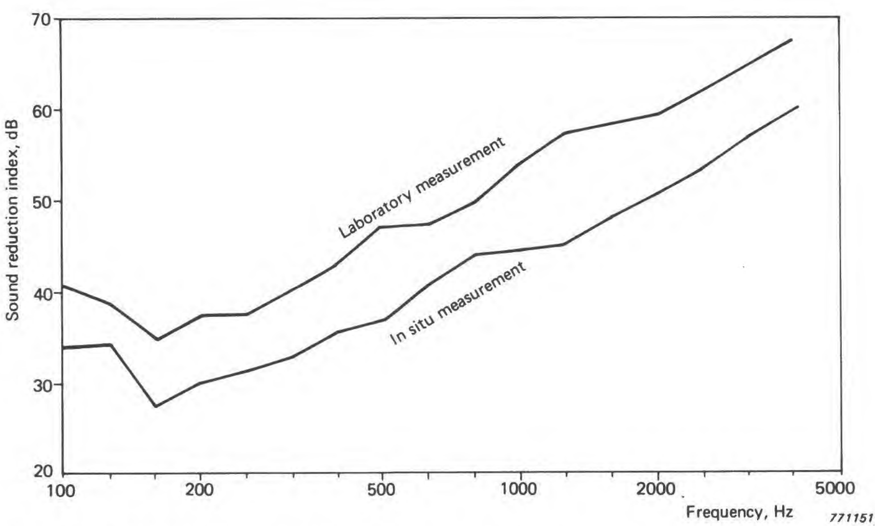

Fig.3.2. Sound reduction index as a function of frequency for a 15 cm concrete wall as measured in a laboratory and as measured in situ.

The sound reduction index depends on the angle of incidence of the impinging sound. If the sound fields in the two rooms are diffuse and providing the sound is transmitted only through the dividing wall then the sound reduction index may be evaluated from

$$R = L_1 - L_2 + 10 \log_{10}\left(\frac{S}{A}\right) \quad \text{dB} \tag{3.2}$$

where L_1 = average sound pressure level in the source room
L_2 = average sound pressure level in the receiving room
S = area of the dividing wall
A = equivalent absorption area of the receiving room determined from reverberation measurements

Except in specially designed laboratory transmission suites, there is practically always a certain amount of indirect or flanking transmission. Various possible transmission paths between adjacent rooms are shown in Fig.3.3. Besides the structure borne paths shown there could also be transmission through airducts, leaks around doors etc.

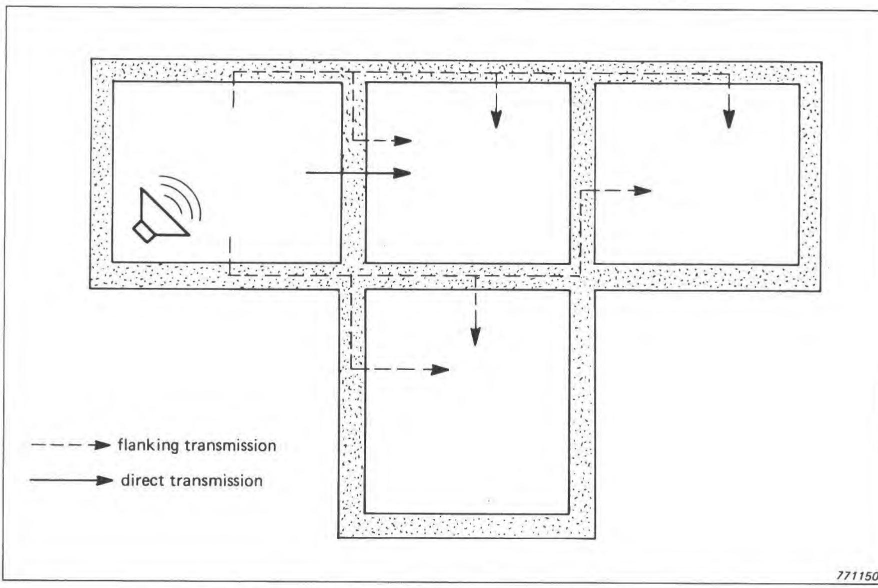

Fig.3.3. Transmission paths between rooms

To account for flanking transmission the quantity known as the apparent sound reduction index, R′, is used. This is defined as

$$R' = 10 \log_{10}\left(\frac{W_1}{W_3}\right) \tag{3.3}$$

where W_1 = sound power incident on the wall
W_3 = total sound power transmitted into the receiving room

The apparent sound reduction index may be evaluated from an expression of the form of eqn.3.2 i.e.

$$R' = L_1 - L_2 + 10\, log_{10}\left(\frac{S}{A}\right) \quad \text{dB} \tag{3.4}$$

In a practical situation (e.g. a block of flats) it is the apparent rather than the absolute sound reduction index that one is interested in.

3.5. IMPACT SOUND INSULATION

Sources of impact sound (e.g. footsteps) act directly on the structure of the building, causing the structure to vibrate and to radiate acoustical energy into the receiving room.

The impact insulation characteristic of a floor is usually expressed in terms of the impact sound pressure level, L_i, which is the average sound pressure in a specified frequency band in the receiving room when the floor under test is excited by a standardized impact source.

To allow for the effect of the absorption of sound which occurs in the receiving room a correction is applied to the impact sound pressure level yielding the quantity known as the normalized impact sound pressure level defined by

$$L_n = L_i - 10\, log_{10}\left(\frac{A_0}{A}\right) \quad \text{dB} \tag{3.5}$$

where L_n = normalized impact sound pressure level
L_i = impact sound pressure level
A_0 = $10\,m^2$ the reference absorption area
A = measured equivalent absorption area of the receiving room

Except in specially designed laboratory transmission suites, flanking transmission of the impact sound will nearly always be present. A typical normalized impact sound pressure level curve for a solid concrete floor is shown in Fig.3.4. Among the ingenious methods devised for providing a standard impact are a person of a specified weight walking at a standard pace and wearing standard shoes for exciting floors; and ball bearings shot from a spring loaded gun for exciting walls and ceilings.

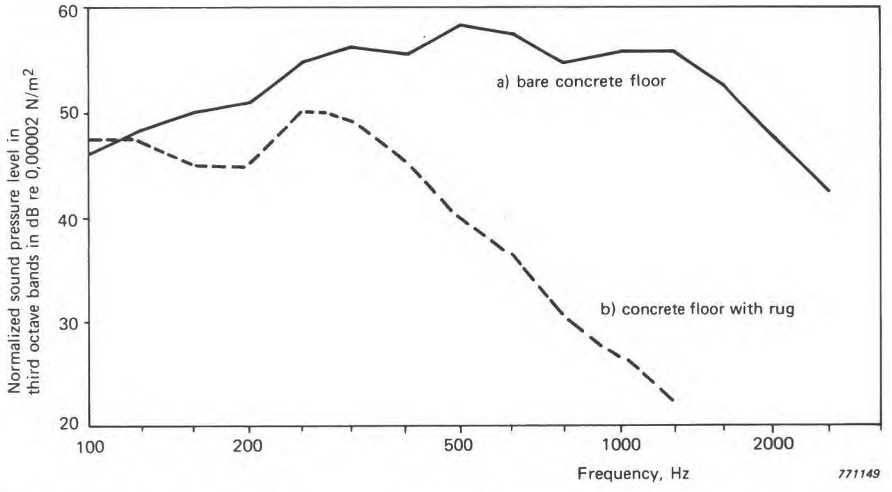

Fig.3.4. Typical normalised impact sound pressure levels measured directly below a concrete slab floor using a standard tapping machine as an impact source
a) bare concrete floor
b) concrete floor covered with a rug

The most common method used at present for the excitation of floors is the standard tapping machine. This machine contains five metal hammers which are lifted by means of an electric motor then allowed to fall and strike the floor one after the other.

The improvement of the impact sound insulation (i.e. the reduction of the impact sound pressure level) is the difference between the average sound pressure levels in the receiving room before and after some treatment of the floor e.g. the installation of a floor covering, a floating floor or a false ceiling.

Material	dB/30 m
Iron	0,3 – 1,0
Brickwork	0,5 – 4,0
Concrete	1,0 – 6,0
Wood	1,5 – 10,0

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Table 3.1. Attenuation of longitudinal waves in building materials

The attenuation of noise borne in building materials is usually quite small (Table 3.1). Therefore, machines and other sources which are likely to produce structure borne noise should be isolated from the main building structure wherever it is possible.

3.6. AIRBORNE SOUND REDUCTION INDEX FOR A SOLID HOMOGENEOUS IMPERVIOUS WALL

The relative importance of the different mechanisms of sound transmission through a solid varies throughout the audio frequency range. A solid wall possesses the qualities of stiffness and mass and can thus exhibit resonance and mode effects. At low frequencies transmission depends mainly on the stiffness of the wall i.e. damping and mass are unimportant. At slightly higher frequencies, the resonances of the wall control its behaviour. At a frequency of about twice the lowest resonance frequency, the walls tends to behave as an assembly of small masses and is said to be mass controlled. From theoretical considerations of the transfer of energy from randomly incident sound waves to the particles of the wall and then into the air on the other side, the relationship known as the Mass Law can be derived. The Mass Law can be expressed as

$$R = 20 \log_{10}(fM) - 47 \text{ dB} \tag{3.6}$$

where f = frequency of the incident sound, Hz
M = surface density of the wall in kg/m^2

This relationship gives the theoretical maximum sound reduction index for random incidence.The Mass Law should only be used to give an approximate guide to the insulation obtainable. In practice, the sound insulation obtained is always a few dB less than the theoretical maximum. In the mass controlled region the sound reduction index increases at a rate of about 6 dB for each doubling of frequency i.e. by 6 dB per octave, and by about 6 dB for each doubling of surface density, which means, for a particular material, for each doubling of thickness. The mass controlled region extends up to the critical frequency. This is the frequency at which the wave length of the bending waves in the wall is the same as the wave length of the radiated sound in the receiving room or in other words, the lowest frequency capable of exciting the coincidence effect (see the following section). Above the critical frequency the stiffness of the wall again plays an important role.

The various regions are shown in the idealized sound reduction index curve in Fig.3.5.

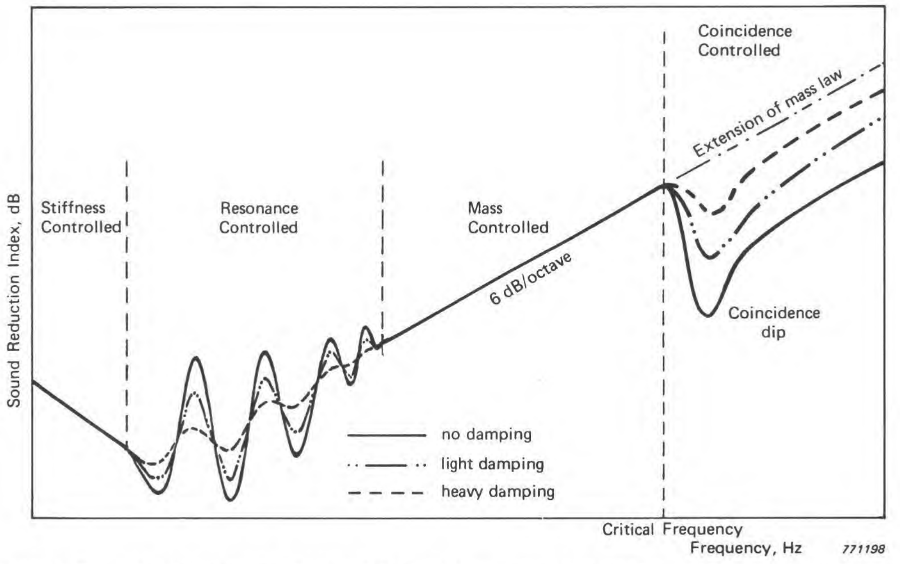

Fig.3.5. Idealised sound reduction index curve for various degrees of damping

3.7. COINCIDENCE EFFECT

In air sound is propagated by longitudinal waves at a velocity which is the same for all frequencies. However in solid structures such as a wall, sound can be propagated by longitudinal, transverse and bending waves. The most important from the point of view of building acoustics are the bending waves. These waves are associated with large transverse displacements which means that they can readily couple to longitudinal waves in the surrounding air.

Bending waves of different frequencies travel at different velocities, the velocity increasing with frequency. This means that for every frequency above a certain critical frequency there is an angle of incidence for which the wavelength of the bending wave can become equal to the wavelength in air projected onto the wall. This is the condition known as coincidence (see Fig.3.6.).

When coincidence occurs it gives rise to a more efficient transfer of energy from the air to the wall to the air on the other side of the wall. Thus the effective insulation of the wall is lowered producing the "coincidence dip" in the insulation curve (Fig.3.5).

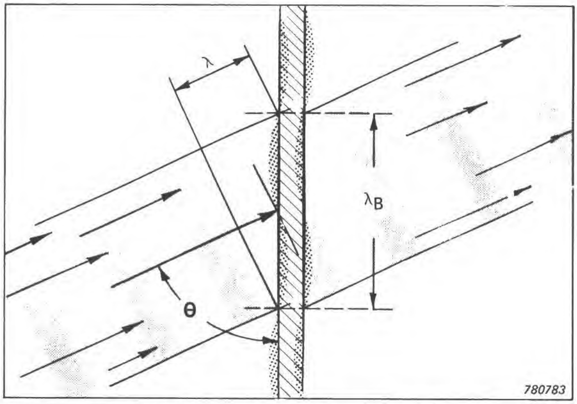

Fig. 3.6. Coincidence effect

The condition for coincidence to occur is

$$\sin\theta = \frac{\lambda}{\lambda_B} \tag{3.7}$$

where λ = wavelength of sound in air
λ_B = wavelength of sound in the wall

Clearly, if the wavelength of the sound in air is greater than the wavelength of the sound in the wall, no coincidence can occur because the sine cannot be greater than 1. When the incident sound has a fixed frequency, the angle at which coincidence occurs is defined as the coincidence angle. When the angle is fixed then the frequency at which coincidence occurs is defined as the coincidence frequency.

The critical frequency is defined as the lowest frequency at which coincidence occurs i.e. that frequency for which $\lambda = \lambda_B$, or $c = c_B$.

where c = velocity of sound in air
c_B = velocity of sound in the wall

In other words the critical frequency is the lowest possible coincidence frequency and occurs when the sound is at grazing incidence to the wall i.e. $\theta = 90°$.

In many cases the coincidence dip occurs in the frequency range 1000 Hz to 4000 Hz, which includes important speech frequencies. When specifying the sound reduction index for a dividing wall it is therefore necessary to define the insulation required over the whole frequency range, because a single figure representing the averaging sound reduction index does not show deficiencies due to resonance or coincidence effects.

3.8. METHODS OF IMPROVING AIRBORNE SOUND INSULATION OF BUILDING ELEMENTS

Damping

For a building element to have a high sound reduction index over a wide frequency range, the element (wall, floor, door, window) should possess a high mass and a low stiffness. An element may have sufficient mass to provide good insulation but its full potential is not realised because high stiffness narrows the frequency range between resonance and coincidence. It is not usually possible to reduce the stiffness of an existing element but the effects of the stiffness can sometimes be reduced by increasing the damping in the element. Damping is only effective in the frequency ranges where resonance and coincidence occurs. There is virtually no effect on the sound reduction index due to damping, in the frequency range where the mass law applies (see Fig.3.5). The most common method of adding damping to an element is to apply a thick layer of a mastic-like material to one side of the element. Providing there is a good bond between the layer and the element a greater proportion of the energy incident in the element will be dissipated in the layer. Multilayered "Sandwich" structures can be built up in this way.

This type of treatment is only effective on elements that have inherently a small amount of damping and low superficial mass. It would be useless for example to apply a damping layer to a 15 cm thick concrete wall although the performance of metal partitions can be greatly improved by this treatment.

Double Leafed Elements

The insulation of a single leaf element can be improved by increasing the mass of the element but this process can only continue up to a certain point. For example if a proposed office partition of surface mass 50 kg/m^2 is replaced by a thick concrete wall of 200 kg/m^2 then the improvement in the insulation will be about 10 dB. The increase in cost and weight may be justifiable in certain cases. If, however, a brick wall exists already and a 10 dB improvement in the insulation is sought then quadrupling the mass of the brick wall would almost certainly not be a practical proposition, quite apart from the fact that at these high values of sound insulation, the flanking walls usually constitute the principal transmission paths. It may at first be thought that one way of increase the insulation would be to construct a second wall just behind the first. Thus, if the walls were identical and the insulation due to one wall alone were 40 dB then the total insulation due to both walls would be 40 + 40 = 80 dB. Unfortunately in practice this is just not so. If the walls were completely separated from each other with no common footings or edge supports and no connecting ties and the air spaces between the walls were larger than about a meter then the total insulation would approach the value of 80 dB. In practice though, walls usually have common supports at the edges and it is rare to find a double leaf wall (or cavity wall) with a cavity wider than a few centimeters. Consider the transmission of sound through a double leaf element such as the one shown in Fig.3.7. There are several possible transmission paths through the cavity, through the edge connections and through the connecting ties. The graph in Fig.3.8 shows the insulation obtained when the leaves are entirely unconnected and the effect of the common connections.

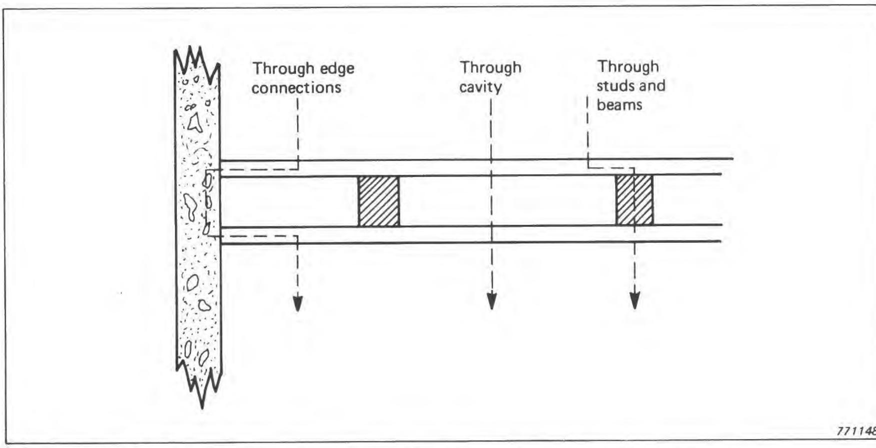

Fig.3.7. Transmission paths through a double leaf wall

At low frequencies the air between the two leaves couples them together rather like a spring. A resonance frequency, f_r, therefore exists which is determined by the mass of the leaves and by the width of the cavity. At the resonance frequency there is a sharp drop in the value of the insulation. In practice f_r should be made less than 100 Hz so that for leaves of low mass a wide cavity is essential. Above the resonance frequency, the insulation increases more rapidly with frequency than it would for a single wall of the same weight but above about 250 Hz the problem of cavity resonances arises which tend to reduce the insulation. These resonances can be considerably reduced by lining the cavity with an absorbent material such as mineral wool or glass fibre. It is not necessary to fill the whole cavity. A layer of about 3 cm of absorbent should suffice. An absorbent lining is most effective in lightweight constructions and has almost no effect on a heavy masonry or concrete wall.

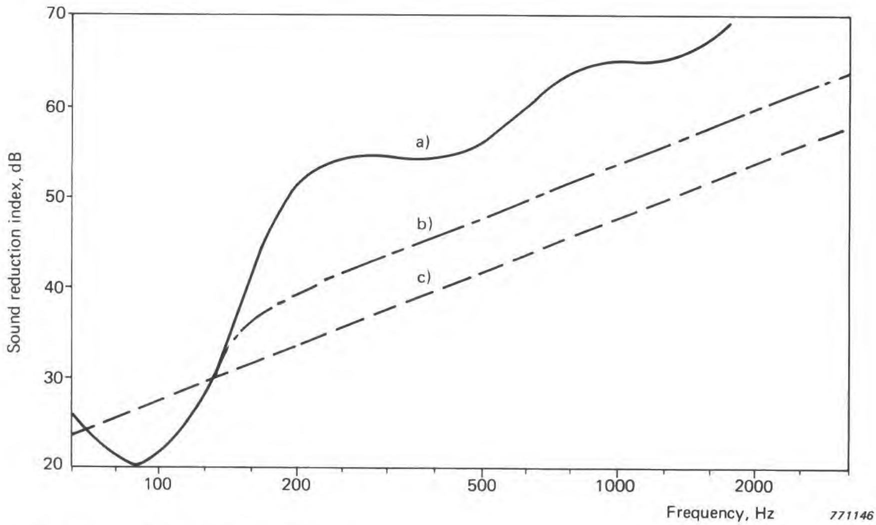

Fig.3.8. Transmission loss
a) theorectical transmission loss of a double wall of mass M
b) typical measured transmission loss for a double wall
c) Mass Law for a single wall of mass M

Transmission through common ties and around the common perimeter becomes important when an insulation of more than 40 dB is required. For insulation in excess of 50 dB the source and the receiving rooms should be vibrationally isolated from each other.

Flanking Transmission

Frequently, the expected insulation of a dividing wall is not realised in practice due to flanking transmission (see Fig.3.3). There are several other transmission paths that sound can follow apart from the direct path through the dividing wall. To obtain the desired insulation none of the flanking paths should be weaker in insulation than the direct path.

The existance of flanking paths means that in some cases it is pointless to construct a high quality insulating wall without making major structural changes in the other building elements of the room.

Providing the critical frequencies of the dividing wall and the flanking elements are low compared to the frequency range of interest then the amount of energy radiated into the receiving room by the various flanking elements can be measured and compared with the amount of energy radiated from the dividing wall by employing the relationship (Ref.2)

$$W_k = \rho c S_k \overline{V}_k^{\,2} \sigma_k \tag{3.8}$$

where W_k = the sound power radiated from element k

S_k = the area of element k

$\overline{V}_k^2$ = the average mean square normal surface velocity

σ_k = the radiation efficiency which has the value of 1 above the critical frequency

ρc = the characteristic impedance of air which has the value of 415 rayls at standard atmospheric temperature and pressure

Doors

The insulation provided by a door does not follow the predictions of the Mass Law for two reasons:

a) there are nearly always small gaps between the door and the door-frame through which sound can be transmitted

b) the size of the door is very much smaller than the wall in which it is placed so that the resonant frequencies of the door occur at much higher frequencies than in a wall made of the same material.

The insulation provided by a door as estimated by the Mass Law will therefore always be somewhat higher than that which can actually be obtained.

When high insulation is required the edges of the door should be sealed very carefully with gaskets of felt or rubber.

Values of insulation greater than those predicted by the Mass Law can be obtained by the use of double doors. If the two doors are separated by a short passageway, sound absorptive material applied to the walls and ceiling in the passageway will further improve the insulation. For double doors separated by at least 8 cm the average insulation is estimated to be at least 5 dB greater than the Mass Law value based on a mass equal to the sum of the masses of the two doors. The seal around the doors must be as carefully made as for the single door. Seals can be provided by compressible gaskets, drop-tongue draught (and noise) excluders and covered key holes.

Outer Walls and Windows

Many of the noises which disturb people within buildings originate out of doors. The noise produced by such sources as air and road traffic and industrial noise enter the building via the outer walls, the windows and the roofs. Many modern buildings have roofs made from concrete or comparable heavy materials so these do not pose such a great problem as the outer walls and windows.

The insulation of the outer facade is usually but not always determined by the insulation of the windows. When a high degree of insulation is required, it is essential that fixed windows be used which means that a mechanical ventilation system is necessary.

The insulation provided by a window is more difficult to estimate than that of a wall because the insulation is more dependent on the window's dimensions and the coincidence effect plays an important role. A rough estimate of the insulation can be obtained from the Mass Law.

The insulation curve of a single glazed window is typically marked by a deep trough in the mid-frequency range; the range in which the ear is most sensitive. This is not so detrimental to the insulation when it is the noise from traffic that has to be excluded because the low frequencies predominate. However, the short comings of a window's insulation are more noticeable when the source of sound is a jet aircraft because of the window's poor insulation at high frequencies.

As with walls and doors,an improvement in the insulation of a window can be obtained by using a double leaf construction i.e. two panes of glass separated by an air gap. The behaviour of such a construction is analogous to a mass-spring-mass system. as the masses of the panes of glass are relatively low, the stiffness of the spring (i.e. the thickness of the air gap) should be large, in order to make the resonant frequency sufficiently low. The double glazing which is used for the purpose of thermal insulation usually has a very narrow cavity (about 1,0 to 1,2 cm) so that the masses are coupled through a stiff spring. Consequently, the resonant frequency is situated at approximately 300 Hz and thus the sound insulation in this frequency region is disappointingly small. Double glazing (i.e. a double window) can be made an efficient sound insulator providing the width of the cavity is at least 3 inches (75 mm). A further improvement in the insulation is obtained by placing sound absorbing material on the sides of the window frame within the cavity.

Floor-Ceiling Elements

The same principles apply for the airborne sound transmission of floor-ceiling elements as for walls. However, special consideration must be given to the control of impact noise to which the floor is subjected because a design that is a good insulator for airborne sound may be unacceptable from the point of view of the transmission of impact noise e.g. a concrete slab floor.

An obvious solution to the problem of impact insulation is to reduce the impact effect on the main structure by covering the floor with a resilient layer such as carpeting or rubber tiling. The action of the resilient layer is to cushion the blow of the impact and thus to reduce the amount of energy transmitted to the structure. Floor coverings are most effective in reducing the higher frequencies of the impact noise. When choosing an appropriate floor finish many non-acoustic factors have to be considered such as durability and resistance to chemical attack. These requirements can sometimes be met by installing a floor composed of a hard upper layer and a resilient lower layer e.g. linoleum on soft fibre board.

Floating Floors

One of the most practical means of obtaining high impact sound insulation in a building is to use a floating floor construction. A floating floor rests on the structural floor but is separated from it by a resilient support such as a mineral wool blanket. The construction can be considered as a mass-spring-damping element system as shown in Fig.3.9. It is vitally important in any floating floor construction that the resilient element is nowhere shorted by a rigid mechanical connection. Such connections which allow sound to be transmitted across the element are sometimes referred to as sound bridges.

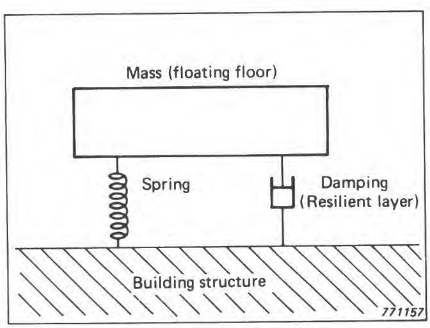

Fig.3.9. Mechanical analogy of a floating floor structure

The resonant frequency of the floor must be chosen to be very low preferably less than 20 Hz otherwise, as seen in Fig.3.10, the insulation at the resonance is less than if the floating floor had not been constructed at all.

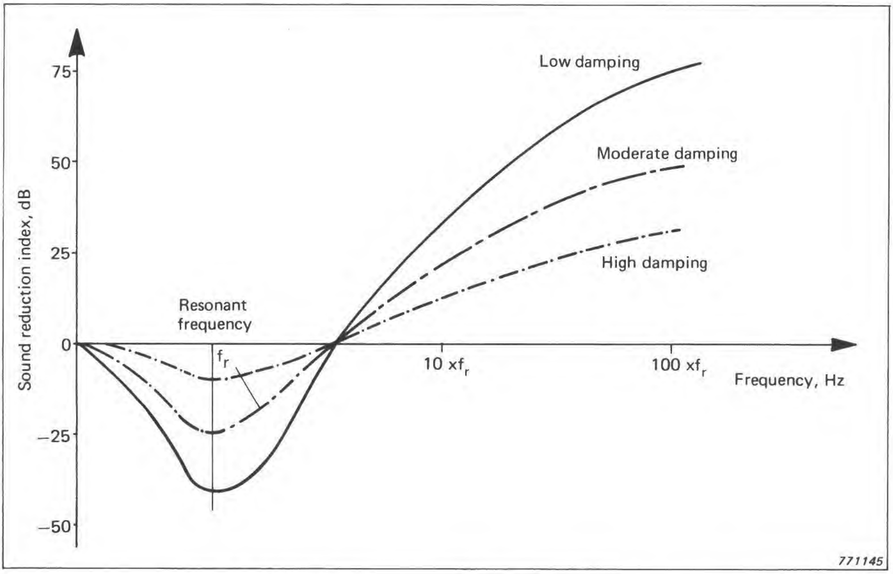

Fig. 3.10. Insulation provided by a floating floor

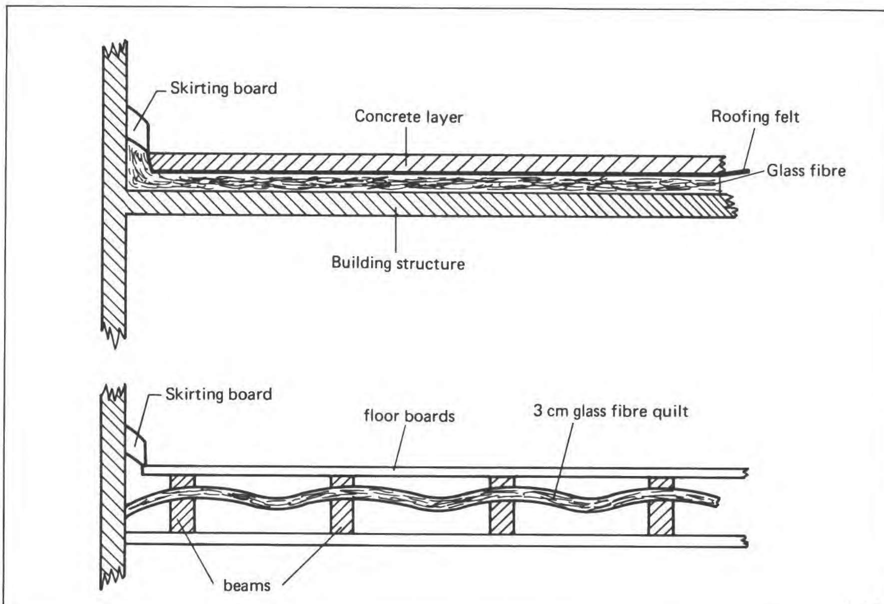

Fig. 3.11. Examples of floating floors

A low resonant frequency is obtained by having a large floating mass. Above the resonant frequency the insulation increases at a maximum of 12 dB/doubling of frequency when the damping is negligible and at a lower rate when the damping is increased.

Some examples of floating floors are shown in Fig.3.11.

Particular attention must be paid to electrical conduits, service pipes from toilets and bathrooms and skirting boards in order to avoid a solid connection between the structural floor and the floating floor. Pipes and conduits should be lagged with a material such as bitumen felt or rubber sheeting where they pass through the floating floor and skirting boards should be insulated from the floating floor by felt or building paper or a similar material.

It should be appreciated that a floor construction designed to reduce the transmission of impact sound to other rooms does not necessarily provide a low noise level in the room in which the impact source is produced. As an example, consider a homogeneous concrete slab structural floor over which is laid a floating wooden floor. The noise in the source room may be greater for the wooden floor than for the bare concrete however the noise produced in other rooms may be reduced because less energy is transmitted to the building structure.

Ceilings

There are two types of ceiling construction which can be used to reduce the radiation of sound from floors set into vibration due to impacts namely, the false ceiling and the suspended ceiling. False ceilings are ceilings which are independent of the main floor ceiling structure. Suspended ceilings are ceilings which are hung from the structural floor by wire or resilient hangers. It should be noted that these ceilings reduce the noise level only in the room where they are installed and that they do not reduce the radiation of sound from the side walls which is due to flanking transmission. They do, however, improve the insulation both for impact and for airborne sounds. Such ceilings are not usually to be recommended as a means of improving sound insulation as they are not very effective. Used alone for improving the insulation of an existing floor, the ceiling would need to be so heavy that it would not be practicable to construct it. If a floating floor is to be built, nothing extra is gained by adding a suspended or a false ceiling. If, however, an existing floor cannot be disturbed and a resilient layer has been laid down to improve the impact sound insulation then the addition of a heavy suspended ceiling may be used to give an improvement in the airborne sound insulation.

Acoustic Leaks

The expected sound insulation of a wall or other building element can be obtained only if sufficient attention is paid to the insulation of the other parts of the construction. For example the transmission loss across a wall, designed to give 45 dB of insulation, will be far below the expected value if the wall contains ordinary doors and windows. Even when doors and windows of special design are used, the measured sound insulation will only approach the expected value providing all the gaps and cracks around the doors and windows are adequately sealed. The higher the sound insulation required, the more important is this attention to detail. Less obvious acoustic leaks can occur via the gaps around service pipes and conduits where they have to pass through walls. Transmission of sound via ventilation ducts can also reduce the effective insulation of a building element. In buildings where suspended ceilings exist it is a mistake from the acoustic point of view for the dividing walls between offices to extend only up to the suspended ceiling as sound can easily propagate from one office to another via the air gap above the suspended ceiling (see Fig.3.12a). A similar type of transmission path can occur when there is a basement or ventilation space below the floor (see Fig.3.12b). These transmission paths can be blocked by continuing the dividing wall through the air gap to join the main structure.

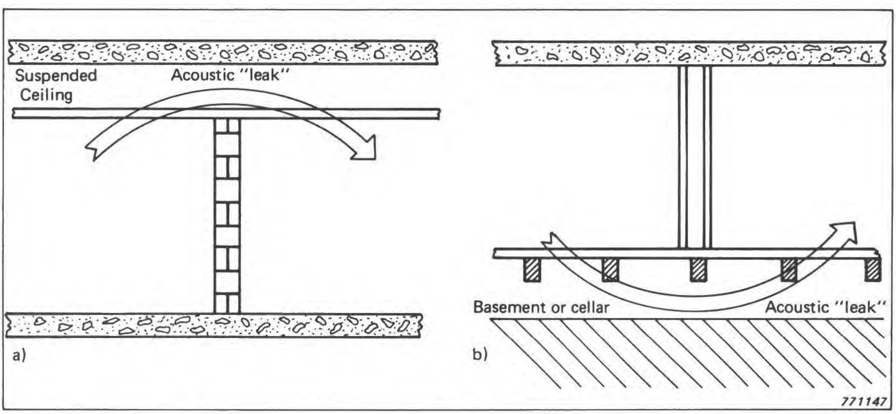

*Fig.3.12. a) transmission path via a suspended ceiling
b) transmission path via a basement*

Discontinuous Construction

When very high sound insulation is required, a discontinuous construction is usually employed. This means that the room (or rooms) to be insulated is completely separated from the main structure of the building supported only by vibration isolation mountings which must be designed for the load of that particular room. It is important to ensure that there are no sound bridges or short circuits by which sound can penetrate into the isolated room. All service pipes, ventilation ducts and conduits must have flexible joints where they traverse the cavity. An example of this "box within a box" construction is shown in Fig.3.13.

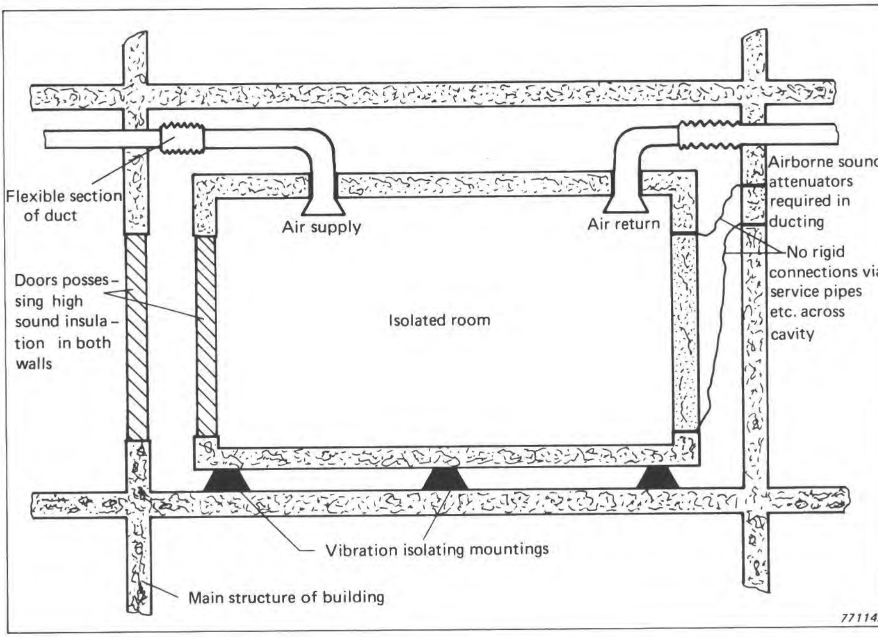

Fig.3.13. Discontinuous construction: vertical section through an isolated room

This type of construction is frequently used for building anechoic rooms used for acoustic research. For anechoic rooms further acoustical treatment is required inside the room to prevent reflections. This is usually accomplished by covering all the surfaces of the room with wedges of mineral or glass wool. The floor can be either a net of steel wire which has a negligible influence on the free field propagation of sound or a retractable metal grid which allows one to enter the room to suspend apparatus but the grid is withdrawn for the duration of the measurements.

3.9. VIBRATION CONTROL

Refrigerators, lifts, air conditioning equipment, heating systems and plumbing are just a few of the many appliances to be found in modern buildings which can give rise to vibrations and structure borne noise.

If a vibrating machine, pipe or other appliance is rigidly connected to the structure of a building then structure borne sound waves will be transmitted with little attenuation. This structure borne sound can "break out" into the various rooms of the building, producing unwanted noise. This problem is best solved at the source by using a vibration isolator to reduce the transmission of energy from the machine to the structure. If a resilient element is inserted between the machine and the structure then little transfer of energy will occur providing the isolator is correctly designed. Using just any resilient elements will not suffice. Resilient elements together with the mass they support have resonant frequencies. If these resonant frequencies correspond to the vibration frequencies of the machine, then energy will not only be transmitted but amplified as well.

The effectiveness of a vibration isolator is measured in terms of the force transmissibility (or the displacement transmissibility) which is defined as the ratio of the transmitted force to the applied force. Transmissibility can be expressed as

$$T = \frac{1}{1 - \left(\frac{f}{f_r}\right)^2} \tag{3.9}$$

where T = transmissibility
f = driving frequency, Hz
f_r = resonant frequency of the isolator, Hz.

The form of this function is shown in Fig.3.14.

Most isolators contain damping as well as resilience. The effect of the damping is to reduce the resonance peak but at the expense of reducing the isolation efficiency at frequencies above the resonant frequency. In many practical cases, damping is necessary to maintain the amplitude of vibration within reasonable limits. This is of particular importance to machines of variable speed which have to be run through the resonant frequency of the supporting isolators.

For the cases where the machine has a particularly annoying resonant frequency or the machine produces vibrations of an impulsive nature then it can be advantageous to place the machine on an inertial block (e.g. a slab of con-

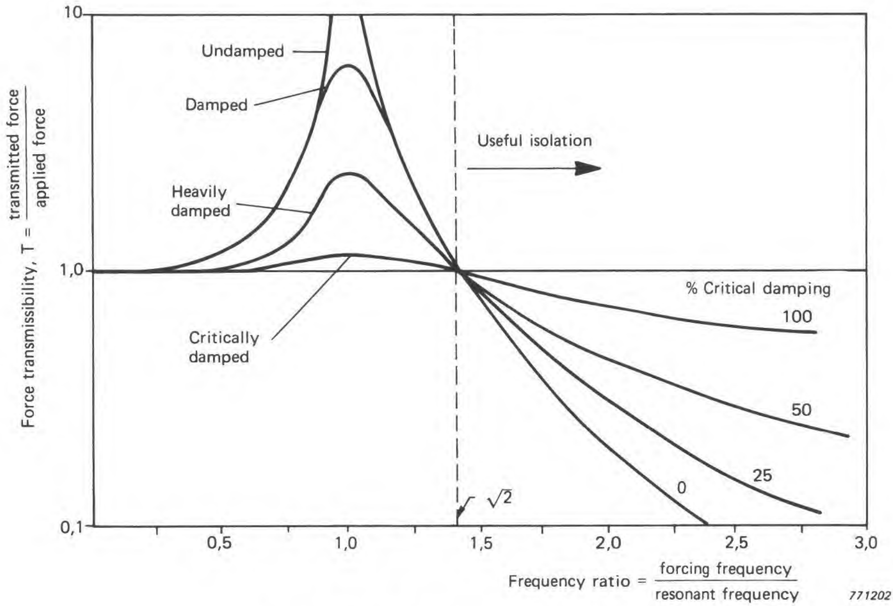

Fig.3.14. Transmissibility function

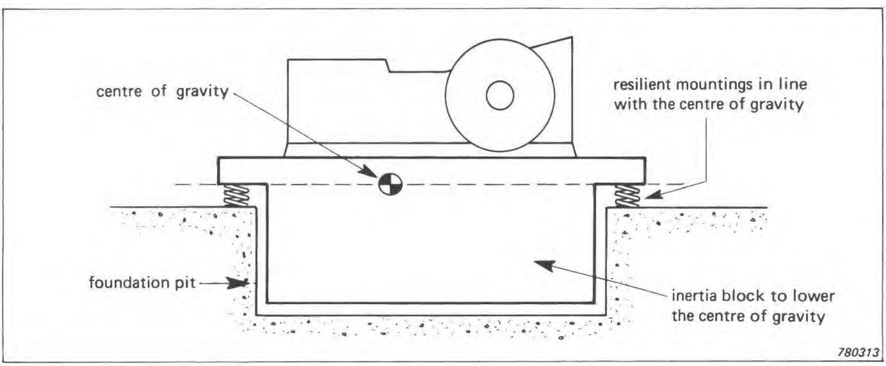

Fig.3.15. Machine mounted on an inertial mass

crete) which in turn is mounted on the vibration isolators. Thus the resonant frequency is reduced and the inertia of the assembly is increased (Fig.3.15).

The type of isolator used depends on the structure and the environment. Some examples of isolators are steel springs in conjunction with oil filled dash pots, rubber in shear and compression held in a steel foot, cork, felt, foam and mineral wool mats.

A useful guide for the selecting and applying resilient devices is the ISO publication Draft ISO 2017.

For methods of measuring the vibration due to machines reference should be made to the ISO publication entitled "Mechanical vibration of large rotating machines with speed range from 10 to 200 rev/s Measurement and evaluation of vibration severity in situ 3945", and also to the B & K publication "Application of B & K equipment to mechanical vibration and shock measurements".

3.10. VENTILATION AND AIR CONDITIONING SYSTEMS

Practically every modern building contains some kind of ventilation or air conditioning system. These systems present a number of noise problems most of which can only be effectively dealt with at the design stage. Structure borne sound is produced by the fan, motor and compressor of the system in the same way as from other machines. Aerodynamic noise is produced by the movement of air in the ducts and through grilles and diffusers. These noises are transmitted via the ducts to all other parts of the building.

Correct planning at an early staqe will save costly modifications later on. The machinery i.e. motor, fan, compressor, should be placed as far as possible from the areas which are liable to be sensitive to noise. The machinery should be isolated from the main structure of the building by means of anti-vibration mounts (see section Vibration Isolation). Short lengths of flexible, resilient hosing should be inserted between the machinery and the ductwork.

To reduce the noise transmitted by the ducts a number of techniques may be employed such as the lining of the ducts with sound absorbing material, the inclusion of a plenum chamber in the system, provision of bends and smooth changes of cross sectional area of the ducts, the insertion of ready made commercially available attenuators and the use of vanes for maintaining a non-turbulent air flow.

Plenum chambers

These chambers are the only effective remedy for low frequency noise. The chambers should be made as large as possible and should be lined with a thick layer of sound absorbing material. Additional absorption can be obtained by installing baffles within the chamber. The volumes which occur naturally in modern buildings e.g. under staircases can be used as plenum chambers.

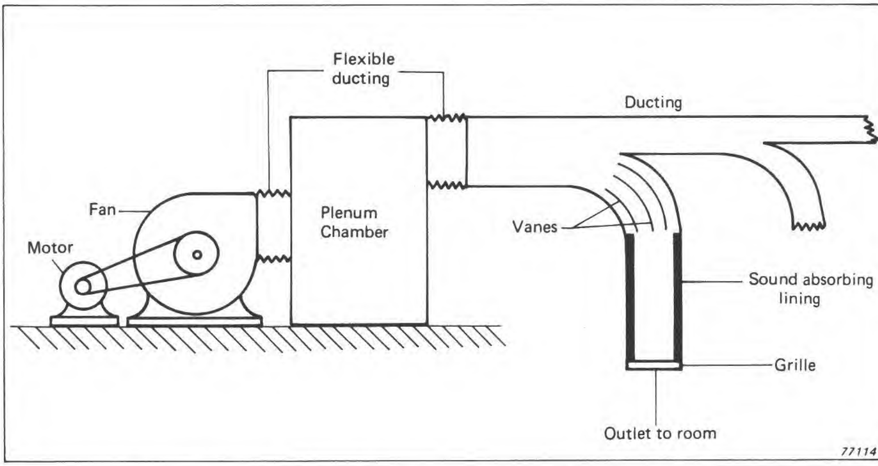

Fig.3.16. Essential elements of a ventilation system

Lining the ducts

One of the simplest ways of reducing the aerodynamic noise present in the air flow is to line the ducts with sound absorbing material. The linings should have

1) high absorption coefficient

2) smooth surface for low air friction

3) adequate strength to resist disintegration due to the air stream

4) odourless, fire, rot and vermin proof

It is advantageous to use lining material in the vicinity of a bend in the duct.

The noise generated by the air flow impinging on a grille (or diffuser) is dependent on the shape of the grille i.e. a badly designed grille will induce turbulence in the air flow. If the grille is at fault then it should be replaced with one of better design.

Companies specialising in ventilation systems are aware of the noise problems involved. It is usually sufficient therefore to specify the acceptable noise levels in the various rooms of the building and to state the required sound insulation between the rooms and to leave the design of the ventilation system up to the suppliers of the ventilation equipment.

4. CRITERIA FOR NOISE CONTROL AND SOUND INSULATION

4.1. INTRODUCTION

In the preceding chapters, the nature and behaviour of sound in rooms and buildings was discussed. Now these concepts will be used to describe the criteria for the acoustic environment for various human activities. Why should criteria be necessary at all? The reason can be seen if some of the more important ways in which noise affects people are considered. Noise can :

1) damage hearing

2) interfere with speech communication

3) disturb concentration thus causing a decrease in efficiency

4) annoy.

At first thought, it would seem that the best environment would be one where there is no noise at all. This is not true, however, for absolute silence can be very disturbing and in any case some noise is necessary in offices, for example, to ensure local privacy. The aim of noise control is to reduce the noise level in a particular environment to an acceptable level and not to remove the noise altogether.

The acceptable noise level depends on the particular situation. The workers in an engine room or a foundry or similar noisy work place would not normally be expected to be annoyed by the noise that they make themselves. There may not even be a need for conversation for the work to be done efficiently. There is a need, however, to protect the worker against the risk of hearing damage. In light industry, factories and the like conversation is usually important. In commercial premises such as offices and shops, the noise levels present should not interfere with conversation and the use of the telephone, nor with the concentration of the staff.

The acceptable noise levels in the home are far more stringent. Although a particular noise may not cause damage or interfere with concentration, it may be extremely annoying.

4.2. HEARING DAMAGE

Noises that are so loud that they cause immediate damage to the ear are fortunately rare. A gradual deterioration in hearing acuity due to exposure to excessive noise is far more common. This deterioration in hearing acuity is usually not apparent to the individual until it is too late and irreparable damage has occurred. The likelihood of permanent damage occurring is a function of length of exposure, noise levels and frequency. This has given rise to the concept of noise dosage.

The acceptable noise dosage according to the ISO criterion is equivalent to exposure to a noise of 90 dB(A) for an 8 hour working day and a 3 dB(A) increase in the sound level requires half the duration to give the same noise dose, e.g. 93 dB(A) for 4 hours is equivalent to 90 dB(A) for 8 hours which is equivalent to 96 dB(A) for 2 hours. The ISO criterion is employed over the greater part of Europe (Fig.4.1.).

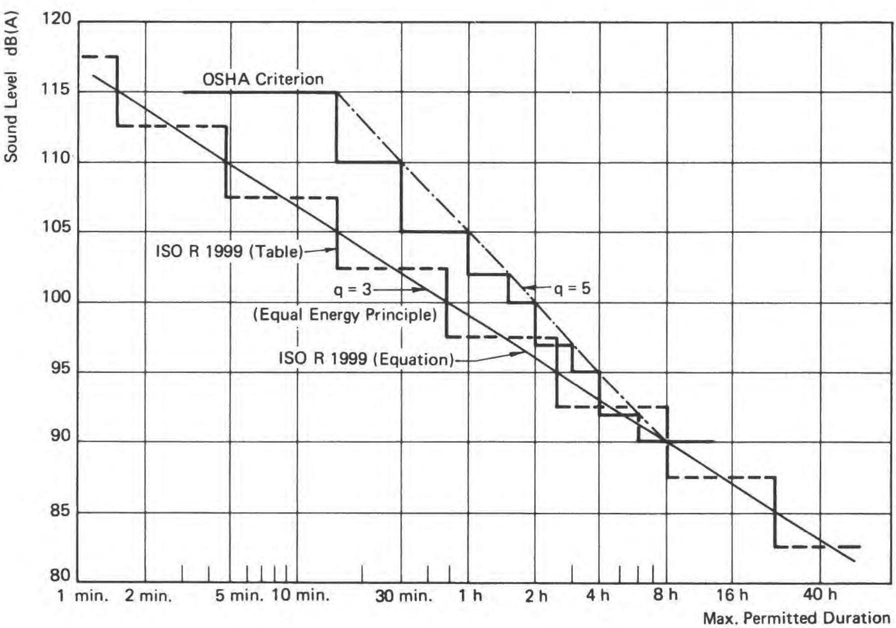

Fig.4.1. Relationship between sound level and duration for the OSHA criterion and ISO R 1999 (for ISO applying the criterion 90 dBA for an 8 hour day and no corrections)

In the United States of America the OSHA criterion is employed (Fig.4.1.). This is again equivalent to exposure to a noise of 90 dB(A) for an 8 hour working day but now a 5 dB(A) increase in sound level for half the duration gives the same noise dose, e.g. 95 dB(A) for 4 hours is equivalent to 90 dB(A) for 8 hours which is equivalent to 100 dB(A) for 2 hours.

The set of Damage Risk Criteria curves (DRC) shown in Fig.4.2. summarises this concept of noise dosage.

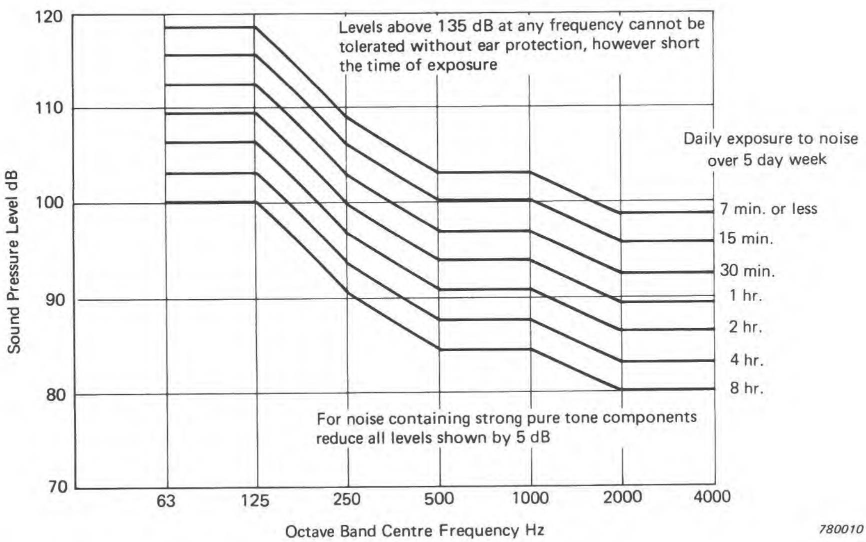

Fig.4.2. Suggested Damage risk criteria curves (DRC) based on recommendation of the U.S. Air Force 1956 (William Burns, NOISE AND MAN, *Murray, 1968)*

4.3. NOISE RATING AND NOISE CRITERIA CURVES

When talking about noise criteria, it is important to bear in mind that the recommended sound pressure levels for a particular situation are intended as a guide to the average acceptability of the noise. Human nature, being what it is, will ensure that there will nearly always be someone who complains no matter what criterion is chosen.

The units most generally used for specifying acceptable noise levels in buildings are the Noise Criteria developed by Beranek in the United States of

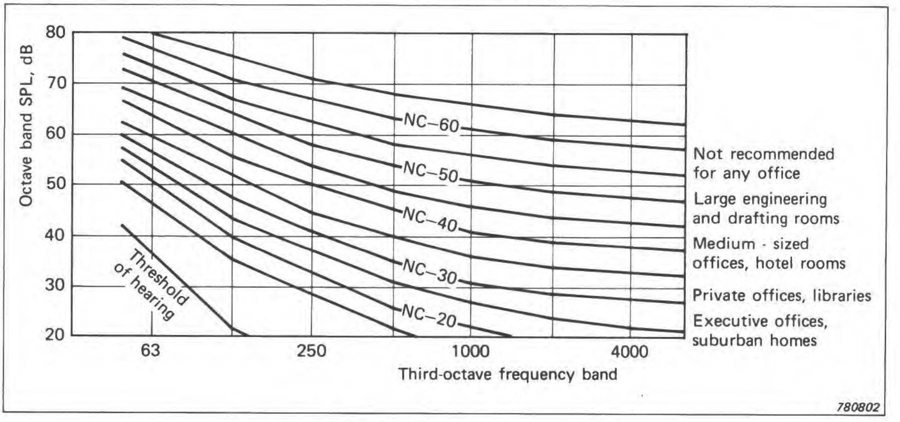

Fig. 4.3. Noise criteria curves (NC)

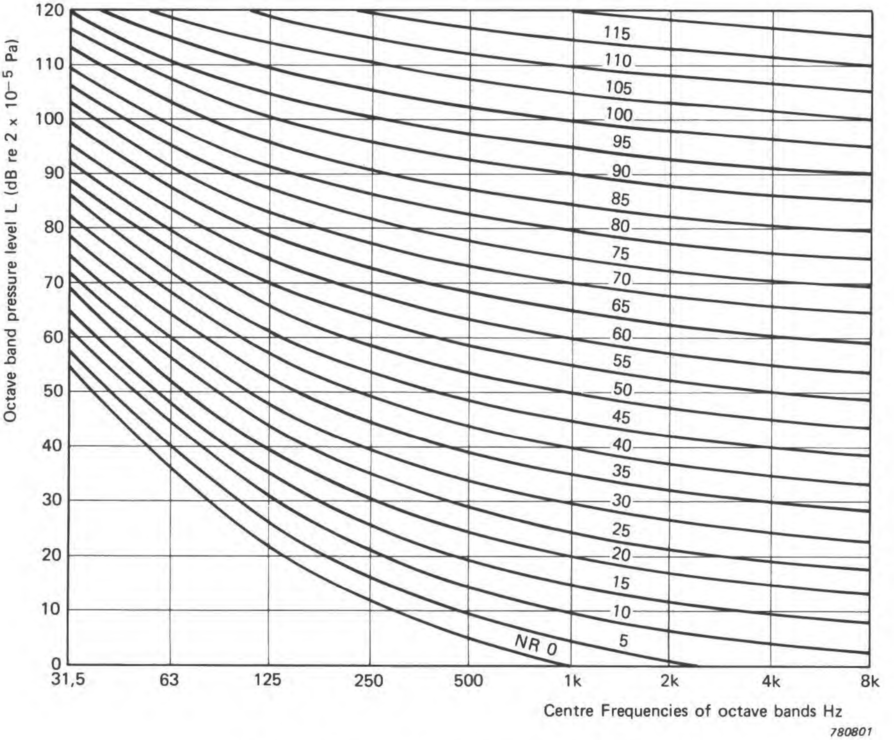

Fig. 4.4. Noise rating curves (NR)

America and the Noise Rating adopted by the ISO and used widely in Europe. The Noise Criteria were developed specifically for commercial buildings and consist of a family of octave band spectra each with its own rating number. The expected or measured noise in the office or shop is compared with the curves and the lowest curve which is nowhere exceeded by the noise gives the Noise Criteria number, (Fig.4.3). The Noise Criteria curves were originally intended to relate the spectrum of a noise to the disturbance it causes to verbal communication, including Speech Interference Levels and Loudness Levels. Nowadays, however, the tendency is to specify an acceptable Noise Criteria number (or a similar criterion) for a particular room instead of talking about Speech Interference Levels and Loudness Levels.

A similar family of curves has been developed by a working group of the ISO and is intended for more general application (Fig.4.4.). Recommended NR and NC values for various environments are given in tables 4.1 and 4.2 respectively. For further details of noise criteria reference should be made to the B & K handbook "Application of B & K equipment to Acoustic Noise Measurement".

Environment	Range of NR Levels likely to be acceptable
Workshops	60–70
Mechanised offices	50–55
Gymnasia, sports halls, swimming baths	40–50
Restaurants, bars, cafeterias	35–45
Private offices, libraries, courtrooms	30–40
Cinemas, hospitals, churches, small conference rooms	25–35
Class rooms, T.V. studios, large conference rooms	20–30
Concert halls, theatres	20–25
Diagnostic clinics, audiometric rooms	10–20

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Table 4.1. Recommended NR values for various environments

Environment	Range of NC Levels likely to be acceptable
Factories (heavy engineering)	55–75
Factories (light engineering)	45–65
Kitchens	40–50
Swimming baths and sports areas	35–50
Department stores and shops	35–45
Restaurants, bars, cafeterias and canteens	35–45
Mechanised offices	40–50
General offices	35–45
Private offices, libraries, courtrooms and schoolrooms	30–35
Homes, bedrooms	25–35
Hospital wards and operating theatres	25–35
Cinemas	30–35
Theatres, assembly halls and churches	25–30
Concert and opera halls	20–25
Broadcasting and recording studios	15–20

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Table 4.2. Recommended NC values for various environments

4.4. NOISE IN THE HOME

However low the loudness level of a noise may be, if it intrudes on peoples' privacy in their own homes then the noise is likely to constitute an annoyance. NR curves and dB(A) measurements give a good indication of how much noise people will tolerate in their own homes but owing to the highly subjective character of the problem several corrections have to be applied befor a particular NR or dB(A) measurement is acceptable. Various environmental factors and the nature of the noise itself have to be taken into account. For example, recognisable audible characteristics of the noise such as pure tones, hissing, impulsiveness etc, will make the noise more annoying than a broad band noise having the same energy. Further, a given noise will be more annoying to a quiet suburb than in the middle of a big city. The time at which a noise occurs (day or night, summer or winter) also plays a role and has to be taken into account.

In Britain, recommendations for the acceptability of noise to householders are given in the British Standard 4142:1967 entitled "Method of rating industrial noise affecting mixed residential and industrial areas". The BS 4142 is based on the work of the Wilson Committee which tentatively suggested that the noise levels shown in Table 4.3. should not be exceeded in living rooms and bedrooms for more than 10% of the time. Similar recommendations have been proposed in many other countries.

Environment	Day	Night
Country areas	40 dB(A)	30 dB(A)
Suburban areas, away from main traffic routes	45 dB(A)	35 dB(A)
Busy urban areas	50 dB(A)	35 dB(A)

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Table 4.3. Noise levels in dwellings which should not be exceeded for more than 10% of the time as recommended by British Standard 4142:1967

4.5. SOUND INSULATION BETWEEN DWELLINGS

To specify the sound insulation between dwellings it is not sufficient to use a single figure index as this could be misleading. The sound insulation is a function of frequency and should be specified over the frequency range of interest. It is therefore usual to specify the insulation in the form of a curve

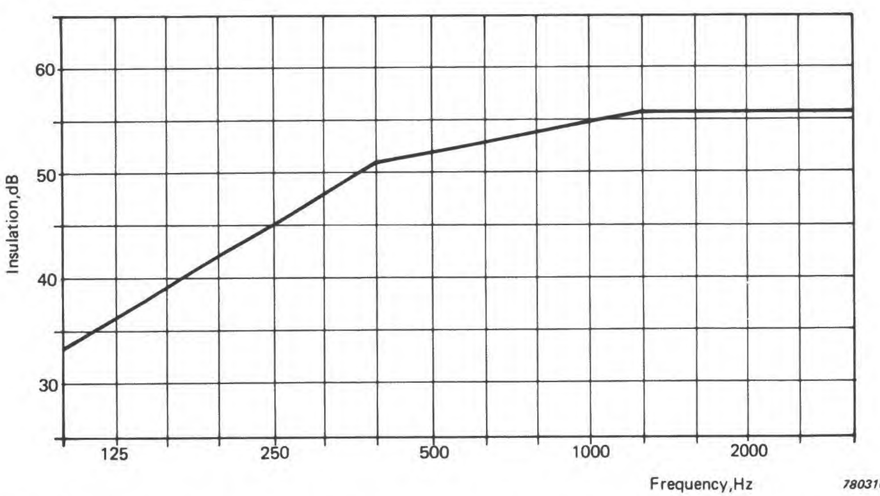

Fig.4.5. Reference values of sound reduction index for airborne sound between dwellings according to ISO/R 717 1968

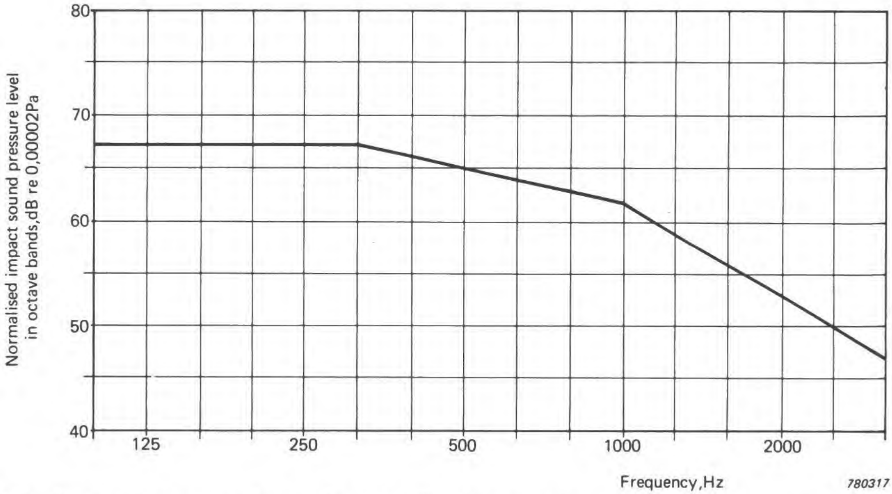

Fig.4.6. Reference values of normalised impact sound level in octave bands in dwellings according to ISO/R 717 1968

and the measured insulation of a building element should not come below this curve by more than a recommended amount.

The International Standards Organisation has published a Recommendation entitled "Rating of sound insulation for dwellings ISO/R 717 1968" in which the measured airborne sound insulation of a building is compared with the reference curve shown in Fig.4.5., and the measured normalized impact sound levels are compared with Fig.4.6.

Examples of national recommendations

Britain

The British system is intended for dealing with the dividing walls (known as party walls) and floors in blocks of flats and terraced houses etc. The party wall insulation is based on the insulation of a 9" brick (0,23 m) wall and the airborne floor insulation is based on the insulation of a concrete floor construction with a floating floor finish. The measured insulation (corrected to a reverberation time of 0,5 sec which is the value of an average living room) is compared with a recommended curve and should fall above this curve although a total adverse deviation of 23 dB over the 16, one third octave bands is permitted. The recommended insulation to be provided by party walls in houses and flats is shown in Fig.4.7.

For impact sound insulation, the British rating system specifies that when

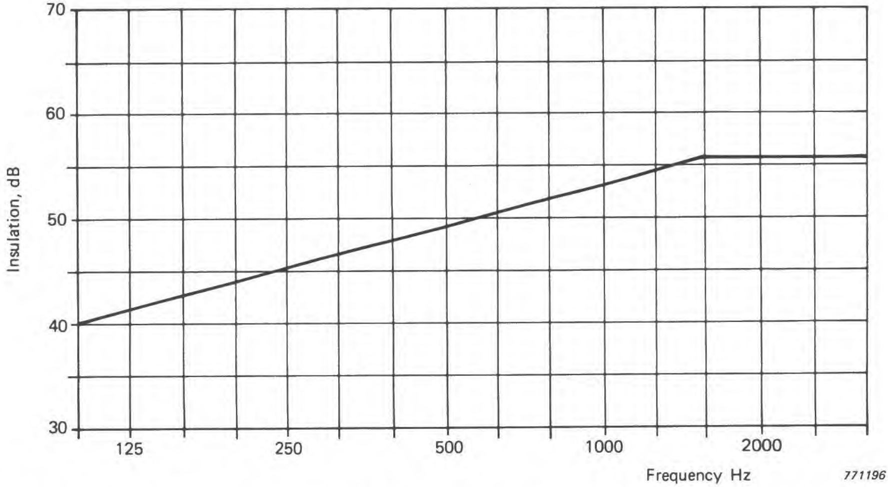

Fig.4.7. Recommended insulation to be provided by party walls in houses and flats (Great Britain)

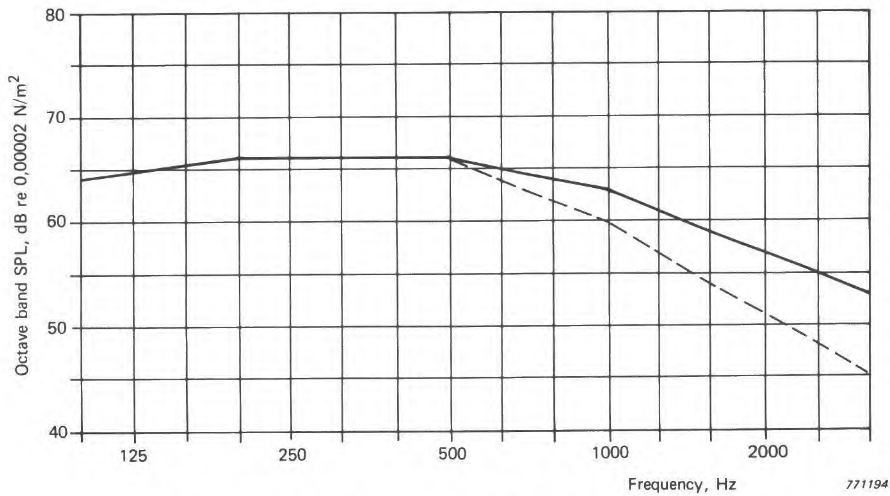

Fig.4.8. Recommended impact sound insulation (Great Britain).
Full line: bare floor
Dashed line: floor covered with linoleum

the sound source is a standard tapping machine (see Chapter 5) then the maximum permitted sound pressure levels in the room below the source room should not exceed the values shown in Fig.4.8. The same conditions for deviations from the recommended curve apply as for airborne sound insulation.

Germany

The German provisional standard DIN 52211 deals with both laboratory and field measurements of sound insulation. The standards used for houses and flats are shown in Figs.4.9 and 4.10.

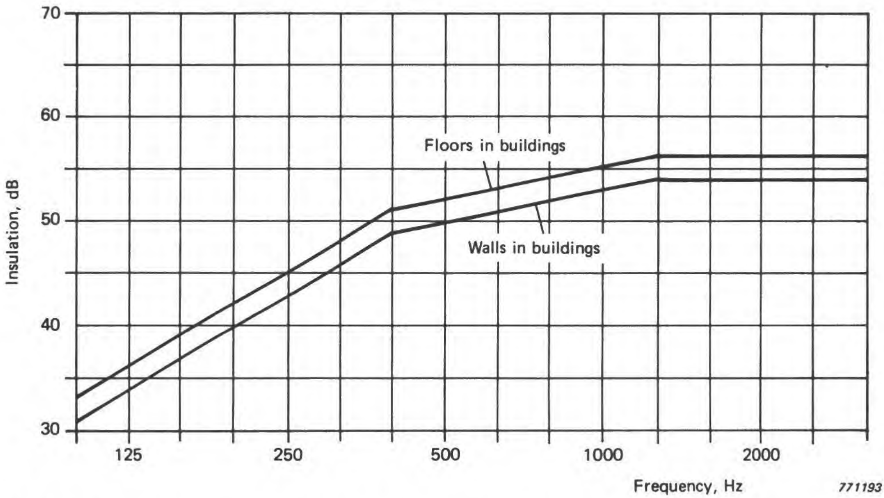

Fig.4.9. Recommended airborne sound insulation to be provided between houses and flats (West Germany)

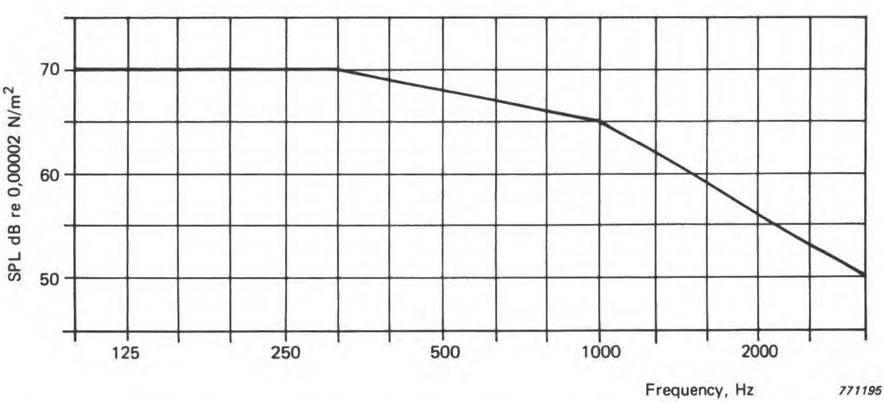

Fig.4.10. Recommended impact sound insulation between flats (West Germany)

Denmark

The Danish Ministry of Housing has published a document entitled "Building regulation for housing in town and country, 1966" which stipulates the amount of sound insulation required in residential and other buildings. The measurement of the sound insulation is performed according to the International Standards Organisation publication ISO/R 140 but a correction has to be applied to make the reverberation time of the receiving room a nominal 0,5 sec.

The document states that the party walls in blocks of flats should have a mean insulation of not less than 49 dB and the insulation in any third octave band should not be less than the values shown in Fig.4.11.

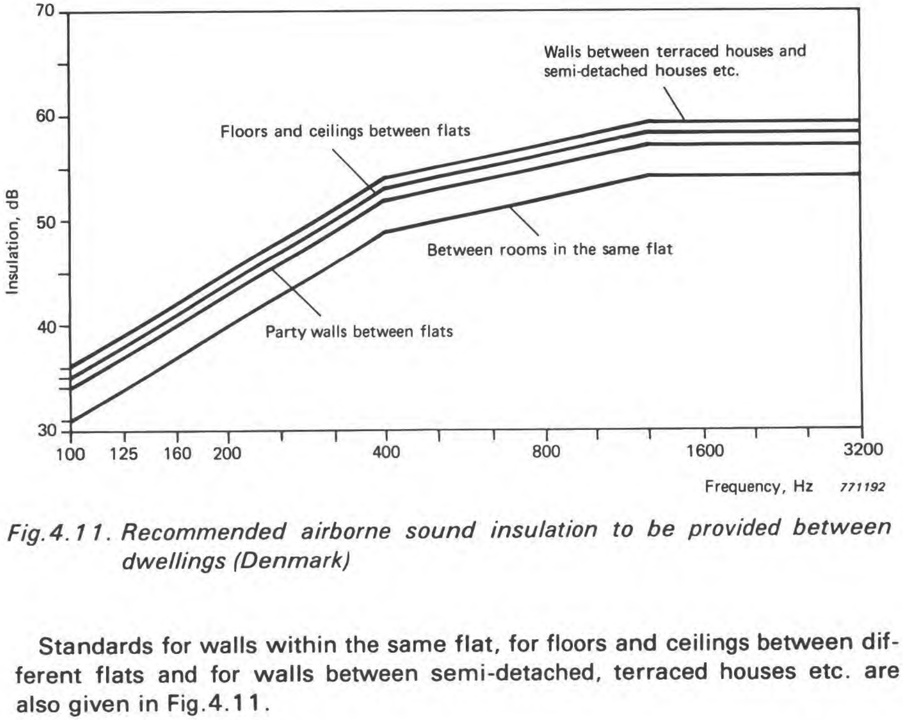

Fig.4.11. Recommended airborne sound insulation to be provided between dwellings (Denmark)

Standards for walls within the same flat, for floors and ceilings between different flats and for walls between semi-detached, terraced houses etc. are also given in Fig.4.11.

Other nations

A list (compiled in 1976) of the building standards in 23 countries is given in the following table.

Standards concerning building acoustic from various countries.

Argentina	
4063	Measurement of sound insulation in dwellings
4065	Measurement of sound absorption coefficients
Australia	Standards House, 80 Arthur Street, North Sydney
AS.1045-1971	Measurement of absorption coefficients in a reverberation room
DR.75060	Method of measurement of normal incidence source absorption coefficient and specific normal acoustic impedances of acoustic materials by the tube method
DR.74163	Code of practice for building siting and construction against aircraft noise intrusion
DR.72090	Standard method for the rating of sound insulation for dwellings
DOC.979	Standard methods for field and laboratory measurements of airborne and impact sound transmission
Austria	Österreichisches Normungsinstitut Leopoldsg. 4 1020 Wien
B 8115	Hochbau, Schallschutz und Hörsamkeit
B 8115 Entwurf	Schallschutz und Raumakustik im Hochbau
	Österreichischer Arbeitsring für Lärmbekämpfung, Regierungsgebäude, 1012 Wien
ÖAL-Richtlinie Nr. 8	Geräuscharme Wohn-, Krankenhaus- und Hotelinstallationen
ÖAL-Richtlinie Nr. 16	Schalltechnische Grundlagen für die Errichtung von Gaststätten und Beherbergungsbetrieben
ÖAL-Richtlinie Nr. 17	Lärmminderung durch Schallschluk-kende Ausstattung
Belgium	Institut Belge de Normalisation, 29 av. de Brabançonne, 1040 Bruxelles
NBN576.05-1963	Mesure en laboratoire de l'indice d'affaiblissement aux sons aériens
NBN576.06-1963	Mesure "in situ" de l'isolement acoustique aux sons aériens
NBN576.07-1964	Mesure en laboratoire de la transmission acoustique des bruits de choc
NBN576.08-1965	Mesure "in situ" de la transmission acoustique des bruits de choc
NBN576.09-1968	Mesure du facteur d'absorption acoustique en salle réverbérante
NBN576.40-1966	Critéres de l'isolation sonique
Brasil	Associação Brasileira de Normas Técnicas — ABNT Av. Almirante Barroso 54 Rio de Janeiro — RJ
P.M.B.432	Medida local e em Laboratório de Transmissão de Sons Aéreos e dos Ruidos de Impacto (Metodo de Ensaio)
NB 101	Norma para Tratamento Acústico em Recintos Fechados

C.S.S.R.	Office for Standards and Measurements, 11347 Praha 1, Václavské Náměsti 19
ČSN 36 8840	Measurement of sound insulating properties of building structures
ČSN 36 8841 1974	Measurement of reverberation time
ČSN 73 0525 1964	Design in the room acoustics. General principles
ČSN 73 0526 1968	Design in the room acoustics. Studios and sound recording rooms
ČSN 73 0527 1973	Room acoustics projects. Rooms for cultural and school purposes. Rooms for public purposes. Administrative rooms
ČSN 73 0531	Protection against noise transmission in building
ČSN 83 0535	Sound absorption coefficient measurement in reverberation room
Denmark	Dansk Standardiseringsråd Aurehøjvej 12 2900 Hellerup
DS/ISO R.140	Felt- og laboratoriemålinger af luftlyds og trinlyds udbredelse
DS/ISO R.354	Måling af absorptionskoefficienter i efterklangsrum
	Statens Trykningskontor
	Landsbyggeloven, kap. 9, lydisolering (Building code, sound isolation)
	Miljøstyrelsen Kampmannsgade 1 1604 København V
Publikation nr.9	Støj, bygge- og anlægsvirksomhed (Noise in building construction)
France	L'Association Francaise de Normalisation (AFNOR), Tour Europe, 92 Courbevoie
NF S 31-002 1956	Mesure, en laboratoire et sur place, de la transmission des sons aériens et des bruits de chocs dans les constructions
NF S 31-003 1951	Mesure du coefficient d'absorption acoustique en salle réverbérante
NF S 31-011 1974	Code d'essai pour la détermination en laboratoire de l'efficacité des revêtements de sol en ce qui concerne la réduction des bruits d'impact
NF S 31-012 1973	Mesure de la durée de réverbération des auditoriums
NF S 31-014 1975	Code d'essai pour la mesure du bruit émis par les équipements hydrauliques des bâtiments
NF S 31-015 1975	Mesure du bruit émis par la robinetterie de puisage (sanitaire et bâtiment)
S 31-016 1971	Mesure du bruit émis par la robinetterie de bâtiment
Germany (B.R.D.)	Beuth Vertrieb GmbH. 1000 Berlin 30, Burggrafenstr. 4 — 7 und 5000 Köln, Kamekestr. 2 — 8
DIN 4109 Bl. 1 — 5	Schallschutz im Hochbau
DIN18041	Hörsamkeit in kleinen bis mittelgroßen Räumen
DIN 52210	Bauakustische Prüfungen. T. 1: Luft- u. Trittschalldämmung, Meßverfahren. T. 2: Prüfstände. T. 3: Prüfungen. T. 4: Einzahl-Angaben

DIN 52212	Bestimmung des Schallabsorptionsgrades im Hallraum
DIN 52214	Bestimmung der dynamischen Steifigkeit von Dämmschichten für schwimmende Estriche
DIN 52215	Bestimmung des Schallabsorptionsgrades und der Impedanz in Rohr
DIN 52216	Messung der Nachhallzeit in Zuhörerräumen
DIN 52217	Flankenübertragung — Begriffe
DIN 52218	Prüfung des Geräuschverhaltens von Armaturen und Geräten der Wasserinstallation im Labor
DIN 52219	Messung von Geräuschen der Wasserinstallation am Bau
VDI 2719	Schalldämmung von Fenstern
Germany (D.D.R.)	Amt für Standardisiering der D.D.R., Mohrenstrasse 37a, 108 Berlin
TGL 10687 Bl. 1 — 8	Bauphysikalische Schutzmaßnahmen, Schallschutz. Siehe S.2. Übersicht der Grundlagenstandards
TGL 10688 Bl. 1 — 12	Messverfahren der Akustik
Great Britain	British Standards Institution, 2 Park Street, London W. 1
BS.2750: 1956	Recommendations for field and laboratory measurement of airborne and impact sound transmission in buildings
BS.CP352: 1958	Mechanical ventilation and air conditioning in buildings (contains a section on sound proofing and anti-vibration devices)
BS.3638: 1963	Method for the measurement of sound absorption coefficients (ISO) in a reverberation room.
BS.CP3: 1972	Part 2: Sound insulation and noise reduction (in buildings)
Hungary	Magyar Szabványügyi Hivatal, Budapest IX, Üllöl út. 25
M.E.-83-65	Technische Vorschriften des Ministeriums für Bauwesen
Italy	Servizio Tecnico Centrale, Ministero dei Lavori Pubblici, Roma
Circolare N. 1769	Criteri di valutazione e collando dei requisiti acustici nelle costruzioni edilizie
Netherlands	Nederlands Normalisatie-Instituut, Polakweg 5, Rijswijk (Z-H)
NEN 1070 1975	Sound insulation measurement in dwellings
NEN 20140	Same as ISO R.140-1960
NEN 20354	Same as ISO R.354
Norway	Norges Standardiseringsforbund, Håkon 7. gt. 2, Oslo 1
NS 3051	Bestemmelse av lydisolering
NS 4804 2.1974	Måling av lydabsorpsjonsfaktorer i klangrom
Poland	Polski Komitet Normalizacji i Miar, ul. Elektoralna 2, 00-139 - Warszawa
PN-70 B-02151	Building acoustics. Soundproof protection for rooms in buildings
PN-61 B-02153	Building acoustics. Terminology
PN-68 B-02154	Building acoustics. Tests on acoustic properties in building partitions
Portugal	Inspecção Geral dos Produtos Agricolas e Industriais (Repartição de Normalização) Avenida de Berna — 1 Lisboa — 1
P-669 1968	Acustica, Ensaio de transmissão dos ruidos aéreos e de percussão (airborne and impact noise transmission)
P-670 1968	Acústica, Determinação em câmara reverberante do coeficiente de absorção e da área sonora equivalente (Determination of sound absorption coefficients)
Roumania	Oficiul de stat pentru Standarde, Str. Edgar Quinet 6, Bucarest 1
STAS 6156-68	Building acoustics. Protection against noise and vibration in buildings. Regulation for design and performance
STAS 6161-60	Methods of measurement of noise in buildings
STAS 8048-67 1967	Measurement of dynamic stiffness of vibration absorbing materials in building acoustics
Sweden	Sveriges Standardiserings-kommision, Box 3295 10366 Stockholm
SIS 025251	Bestämning av ljudisolering i byggnader
SIS 025252	Bestämning av ljudisolering i byggnader, fältmätning
SIS 025253	Metod för värdering av ljudisolering mellan rum i byggnader
SIS 817306	Ljudisolerande dörrar 25 dB, 30 dB och 35 dB
	Liber Förlag, Fack, 103 20 Stockholm
SBN 75 Kap 34	Ljudklimat
	Svensk Byggtjänst, Box 1403, 111 84 Stockholm
KBS 10-1968	Normer för kontorsbyggnader
Switzerland	Schweiz. Ingenieur- und Architekten-Verein
SIA Empflehlung 181-1970	Empfehlung für Schallschutz im Wohnungsbau
U.S.A.	Acoustical and Board Products Association 205 West Touhy Avenue Park Ridge, IL 60068
AMA-1-II 1967	Method of test. Ceiling sound transmission test by two-room method
AM Spec. No. 11 (1972)	Acoustical absorbers
	Air Diffusion Council 435 North Michigan Chicago, IL 60611
AD-63 (1963)	Measurement of room-to-room sound transmission through plenum air systems
FD-72 (1972)	Flexible air duct test code

	American National Standards Institute 1430 Broadway New York, NY 10018
S1.7-1970	Sound absorption of acoustical materials in reverberation rooms
	American Society for Testing and Materials 1916 Race Street Philadelphia, PA 19103
ASTM C384-58 (Reapproved 1972)	Standard method of test for impedance and absorption of acoustical materials by the tube method
ASTM C423-66 (Reapproved 1972)	Standard method of test for sound absorption of acoustical materials in reverberation rooms (ANSI S1.7-1970)
ASTM C634-73	Standard definitions of terms relating to acoustical tests of building constructions and materials
ASTM E90-75	Standard recommended practice for laboratory measurement of airborne sound transmission loss of building partitions
ASTM E336-71	Standard recommended practice for measurement of airborne sound insulation in buildings
ASTM E413-73	Standard classification for determination of sound transmission class
ASTM E477-73	Standard method of testing duct liner materials and prefabricated silencers for acoustical and airflow performance
ASTM E492-73T	Tenative method of laboratory measurement of impact sound transmission through floor ceiling assemblies using the tapping machine (1971)
ASTM E497-73T	Tenative recommended practice for installation of fixed partitions of light frame type for the purpose of conserving their sound insulation efficiency

	Dept. of Housing and Urban Development Washington, D.C.
	A guide to airborne, impact and structure borne noise control in multifamily dwellings
	International Conference of Building Officials 5360 South Workman Mill Road Whittier, CA 90601
UBC 35-1	Laboratory determination of airborne sound transmission class (STC)
UBC 35-2	Impact sound insulation
UBC 35-3	Airborne sound insulation field test
U.S.S.R.	Komitet Standartov, Leninsky Prospekt 9 b, 117049 Moskva M-49
Gost 15116	Sound insulation. Method of measurement. Sound insulation factor
Gost 16297-70	Building wares and materials. The methods of acoustical tests
Yugoslavia	Official Gazette
13 Aug. 1970	Regulation on technical precautions and conditions for sound protection in buildings
International (I.S.O.)	International Organization for Standardization, 1, Rue de Varembé, Geneva, Switzerland
R.140-1960	Field and laboratory measurements of airborne and impact sound transmission
R.354-1963	Measurement of absorption coefficients in a reverberation room
R.717-1968	Rating of sound insulation for dwellings
	Draft Proposal
ISO/DIS 3382	Measurement of reverberation time in auditories

5. MEASURING TECHNIQUES

5.1. INTRODUCTION

This chapter describes some of the many applications of Brüel & Kjær instruments to architectural acoustics by describing a generalised chain of measurement, link by link. The basic theory behind the measurements has been briefly outlined in the previous chapters but those readers who require a deeper insight should refer to one of the many excellent books on architectural acoustics (see Bibliography). Instrument arrangements which can be employed in order to comply with a particular ISO standard, are indicated in the text (Ref.28).

5.2. GENERALIZED CHAIN OF MEASUREMENT

Whatever type of acoustical investigation is being performed in a building, the basic instrumentation required can be considered as a source-transducer-

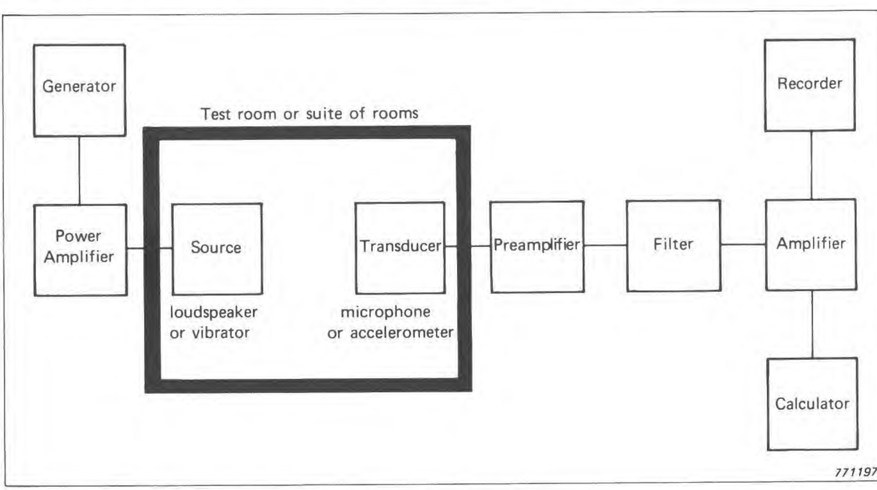

Fig.5.1. Schematic diagram of instrumentation used in architectural acoustics

analyser chain i.e. one requires a source of sound (or vibration) a measuring transducer such as a microphone (or an accelerometer) in conjunction with a filter/amplifier and a recording device. This scheme is shown in a generalised block diagram form in Fig.5.1. Each block does not necessarily represent an individual instrument, of course. The whole of the transmitting side of the diagram, for example, may already be present in the building in the form of a noisy machine. Furthermore, the entire receiving side may be contained in one instrument such as the Precision Sound Level Meter and Octave Analyser Type 2215.

5.3. SOURCES OF SOUND AND VIBRATION

For certain measurements, a suitable sound source may already exist in the room of interest e.g. the organ in a church or the orchestra in a concert hall may be employed as the sound source for reverberation time measurements. Usually, however, to measure such quantities as sound insulation, sound absorption and sound power, a suitable sound source has to be introduced into the room. Three sound sources are produced by B & K for the airborne excitation of rooms and auditoria. These are the Isotropic Sound Source Type 4241, the Sound Power Source Type 4205 and the Reference Sound Source Type 4204 (fulfills ISO 3741). When measuring the impact sound insulation of floors, the standardised Tapping Machine Type 3204 can be employed (fulfills ISO 140) (Fig.5.2).

The Sound Power Source Type 4205, although developed for sound power measurements, is an excellent sound source for most purposes in building acoustics. The 4205 is a portable battery operated instrument capable of supplying white or pink broad band noise. The pink noise can be filtered by any one of the 7 built-in octave filters whose centre frequencies are 125, 250, 500, 1000, 2000, 4000 and 8000 Hz. The sound power output of the 4205 can be varied continuously between 40 dB and 100 dB re 1 pW (100 dB re 1 pW corresponds to a sound pressure level of 92 dB re 2×10^{-5} Pa at 1 m from the 4205). Among the applications of the 4205 are reverberation measurements (as generator can be stopped in less than 30 ms), sound insulation and sound distribution measurements.

The Reference Sound Source Type 4204 is a highly stable, calibrated source fulfilling the demands of ISO 3741. The 4204 is primarily intended for sound power measurements but is also well suited for sound absorption and sound insulation measurements.

Fig. 5.2. Sources of noise: Upper left, Tapping Machine Type 3204; upper right, Isotropic Sound Source Type 4241; Lower left, Reference Sound Source Type 4204; Lower right, Sound Power Source Type 4205

The Isotropic Sound Source Type 4241 is supplied with the required signal from a generator. It is usually necessary to amplify the signal from the generator by employing a power amplifier in order to drive the 4241 or another suitable commercially available loudspeaker.

Fig.5.3 shows three signal generators and a Power Amplifier Type 2706. This power amplifier can be used with any of the three generators to give a power output capacity of 75 VA into a 3 Ω load. The Noise Generator Type 1405 can provide the 4241 with either a white noise or a pink noise signal. The white noise is used mainly in connection with constant bandwidth analysis while the pink noise is used mainly in connection with constant percentage bandwidth analysis. An external filter such as Type 1618 can be used in connection with the generator to produce bands of noise. For frequency response and resonant frequency investigations, a sinusoidal signal is required. This can be supplied by the Sine Generator Type 1023, which can also produce a warbled tone thus avoiding the establishing of standing waves in a room.

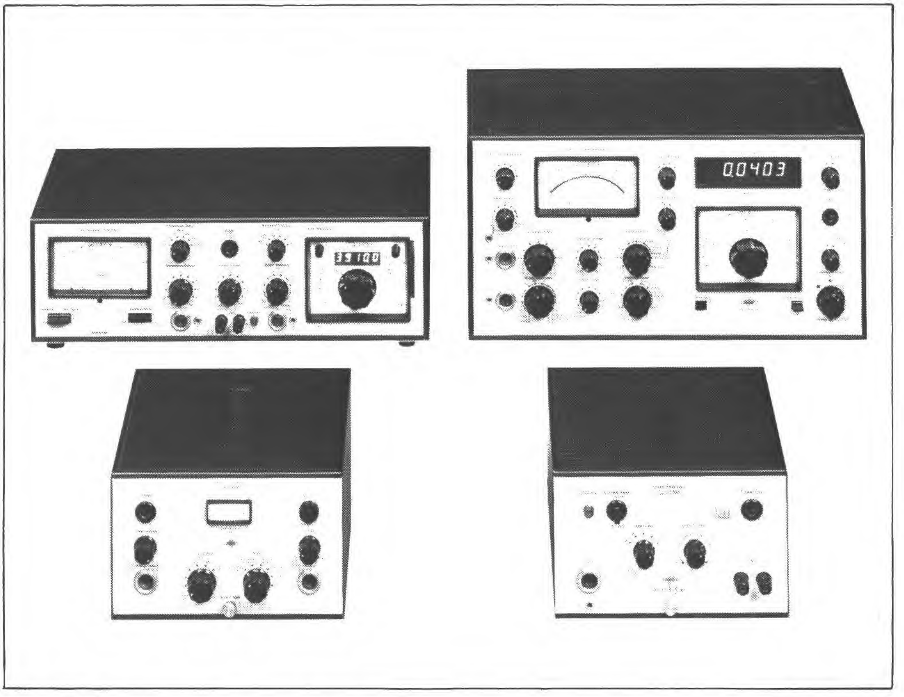

Fig.5.3. Signal generators and a power amplifier : lower left, Noise Generator Type 1405; upper left, Sine Generator Type 1023; upper right, Sine Random Generator 1027; lower right, Power Amplifier Type 2706

A more sophisticated generator than either the 1405 or the 1023, is the Sine Random Generator Type 1027. Four types of output signal are available from this generator. These are white noise, pink noise, narrow band random noise and a sinusoidal signal. Narrow band random noise is used to avoid the build-up of standing waves in a room and also where a swept random noise is considered to be a more relevant test than a swept sinusoidal signal.

All the generators can be remotely controlled allowing the signal output to be stopped abruptly as required for reverberation time measurements. The generators all possess compressor circuits which enable the output voltage of the generators to be controlled by an external voltage. This feature is extremely useful in frequency dependent measurements where a given parameter must be kept constant. The compressor can be used, for example, to maintain a constant sound pressure level in a room, for maintaining the vibration level of a vibration exciter constant, etc. The generators can be synchronised with the B & K level recorders to produce records of frequency analysis on frequency calibrated paper.

Measurement of quantities such as the loss factor of a wall, the magnitude of the flanking transmission etc., require that the structure of the building be excited directly. When investigating the effect of impact, it may be adequate to produce this excitation by a simple hammer blow or the fall of a sack full of sand. Measurement of such quantities as mechanical mobility require a controllable source of vibration as obtained from a vibration exciter from B & K's wide range of exciters and auxillary equipment. As exciters vary considerably in their capabilities, it is important to select the exciter having specifications best suited for a particular application i.e. desired frequency range, sufficient force etc. It is often necessary (e.g. in the investigation of transmission paths) to induce in a wall the same level of vibration as that induced by the incidence of airborne sound. These levels of vibration are usually quite low so that the vibration exciter employed need not be so powerful. For this application, for example a Permanent Magnet Exciter Type 4809 may be used. This exciter has a frequency range of 10 to 20000 Hz and can deliver a force of 44,5 N peak value and up to 60 N with assisted air cooling.

The vibration exciter is driven in the same way as the 4241 described previously, that is, with a signal generator and a power amplifier. The Power Amplifier Type 2712, which has a power output of 180 VA, is designed to drive the Exciter Type 4809 safely to its full rating. Other vibration exciter and power amplifier systems are available which have power handling capacities less than and greater than the Type 4809.

5.4. MICROPHONES AND ACCELEROMETERS

Selection of a Microphone

The sound pressure level produced within the room by the source can be measured by a calibrated microphone. Several factors have to be considered when selecting the most suitable microphone for a particular application e.g. the frequency range, the sensitivity required, the directivity and the size. Brüel & Kjær produce 16 different condenser microphones with 4 different diameters. The principal features of this range of microphones are shown in Table 5.1 together with their main areas of application. (Ref. 38).

By choosing the appropriate microphone from this range, precision sound level measurements can be made in the infrasonic, the audio and the ultrasonic frequency range. B & K microphones have three types of characteristics i.e. free field, pressure and random incidence. The frequency response of the microphone above frequencies of about 2 kHz depends on the type of characteristic the microphone possesses, the differences between the various characteristics becoming more marked as the frequency increases. Standards and recommendations often specify which type of microphone characteristic has to be employed. Free field microphones have uniform frequency response for the sound pressure that existed before the microphone was introduced into the sound field. Any microphone will disturb, to some extent, the sound field in which it is placed but the free field microphone is designed to compensate for its own presence when the microphone is orientated so that the sound arrives perpendicularly to the microphone diaphragm. The pressure microphone, however, is designed to have a uniform frequency response to the actual sound pressure present which of course includes the microphone's own disturbing presence. The random incidence microphone is designed to respond uniformly to signals arriving simultaneously from all angles such as in the case of highly reverberent or diffuse sound fields. When making measurements in a free field e.g. in the open air or in an anechoic room, a free field microphone should be pointed directly at the sound source while a pressure microphone should be orientated at an angle of 90° to the direction of propagation of the sound so that the sound grazes the front of the microphone.

In a diffuse or random sound field, as for example in a fairly reverberant room, the microphone should be as omnidirectional as possible. In general, the smaller a microphone is, the better is its omnidirectional characteristic and frequency response. However, smaller microphones are also less sensitive which may not be acceptable if the measurements are being made under relatively quiet conditions. Under these conditions, it is recommended to use a 1 inch microphone such as the 4144 which combines a high sensitivity with a good random incidence response. By mounting specially designed correctors, such as a random incidence corrector or a nose cone, the response of the 1, 1/2 and 1/4 inch free field microphones can be made practically independent of the angle of incidence over an extended frequency range. Furthermore, the nose cones help to reduce wind induced noise as experienced for example, when the microphone is mounted outside the facade of a building. Another microphone attachment, designed to attenuate noise due to turbulence when measuring the airborne noise in airducts, is the Turbulence Screen UA 0436, which can be fitted to any 1/2 inch microphone and preamplifier assembly.

For such applications as acoustic modelling, measuring the sound pressure in acoustic resonators or near to sound absorbing materials, it can sometimes be advantageous to use the Probe Microphone Type 4170. The 4170 consists of a long narrow probe tube connected to a 1/2 inch condenser microphone and is designed to create the minimum of disturbance to the sound field. The Probe Microphone Kit UA 0040, containing probe tubes of various diameters is designed for use with the 1/2 inch Microphone Type 4134. The necessary tools and material for cutting and inserting damping material to make the probe suitable for a particular application are supplied with the Kit.

When measuring in diffuse fields, it is important to remember that the presence of the instrument case and the operator may obstruct the sound incident from certain directions and spoil the otherwise excellent omnidirectional characteristics of the microphone. Where this problem is likely to occur the microphone should be mounted on a flexible extension rod or, better still, on an extension cable in order to position the microphone as far as possible from the instrumentation and the operator. When microphone cables have to be led through closed doors and windows, a frequent occurence in architectural acoustics, then the Tape Microphone Cable AR 0001 can be employed. The thickness of this cable is only 0,2 mm.

In humid or polluted atmospheres, such as in a boiler room, workshop etc., it may be preferable to use the Miniature Hydrophone Type 8103 instead of one of the microphones mentioned in Table 5.1.

Microphone type	Main area of application	Diameter	Associated preamplifier	Frequency range (± 2 dB)	Response	Sensitivity (mV/Pa)
4145	General sound level measurements including low sound levels	1"	2619 or 2627	2,6 Hz – 18 kHz	Free field	50
4144	Coupler measurements, audiometer calibration, calibration standard	1"		2,6 Hz – 8 kHz	Pressure	50
4146	Sonic boom measurements, acoustic pulse measurement, carrier type (carrier frequency 10 MHz)	1"	2631 (with 2619 same specs. as 4144)	< 0,1 Hz – 8 kHz	Pressure	12–60
4160	Primary calibration standard	1"	2627	2,6 Hz – 8,5 kHz	Pressure	47
4133	Free field general purpose, general sound level measurements, electro-acoustic measurements	1/2"	2619	4 Hz – 40 kHz	Free field	12,5
4134	Pressure, general purpose, general sound level measurements, coupler measurements, probe microphones	1/2"		4 Hz – 20 kHz	Pressure and random	12,5
4149	Best choice for polluted or humid environments, long term outdoor monitoring systems	1/2"		4 Hz – 40 kHz	Free field	12,5
4147	For sonic boom acoustic pulse and infrasonic measurements, carrier type (carrier frequency 10 MHz)	1/2"	2631 (with 2619 same spec. as 4134)	0,01 Hz – 20 kHz	Pressure and random	3,7–18
4148	Free field general purpose, for use with low polarizing voltage	1/2"	2619 with type 2804	2,6 Hz – 16 Hz	Free field	12,5
4165	General free field measurements including low levels	1/2"	2619	2,6 Hz – 20 kHz	Free field	50
4166	General random sound level measurements	1/2"	2619	2,6 Hz – 9 kHz	Pressure and random	50
4125	Low cost applications, low polarizing voltage	1/2"	2642	5 Hz – 12,5 kHz (± 3 dB)	Free field and random	10
4135	Free field sound level measurements model work, high levels	1/4"	2618 or 2619 + UA 0035	4 Hz – 100 kHz	Free field	4
4136	Random incidence, sound level measurements, boundary layer, pulses, coupler, measurements	1/4"		4 Hz – 70 kHz	Pressure and Random	1,6
4138	Very high frequency and very high sound level measurements, models, confined spaces, point source and receiver, sharp pulses	1/8"	2618 + UA 0160 or 2619 + UA 0036	6,5 Hz – 140 kHz	Pressure and random	1

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Table 5.1. Principle features and main areas of application of B & K range of microphones

Generally speaking, for almost all sound pressure measurements in building acoustics one of the following four half inch microphones can be employed, the 4133, 4134, 4165 and the 4166. For diffuse field measurements either the 4134 or the 4166 can be recommended. Both microphones have random incidence responses but although the 4166 possesses a greater sensitivity than the 4134, it has a narrower flat pressure frequency characteristic. For free field measurements, for example in the direct field of a machine or in an anechoic room, either the 4133 or the 4165 can be recommended. A choice has to be made between the wider frequency range of the 4133 and the greater sensitivity of the 4165. The 4165 is the microphone that is usually supplied with the sound level meters.

Preamplifier Selection

Once the appropriate microphone has been chosen, an accompanying preamplifier must be selected. The preamplifier presents the microphone with the correct impedance and also supplies the microphone with a suitable polarisation voltage. All the microphones in Table 5.1 are designed to operate with a DC polarisation voltage of 200 V except for the Type 4125 and the Type 4148 which operate with polarisation voltage of 28 V.

The preamplifier has a very high input impedance and presents virtually no load to the microphone. A low output impedance enables the connecting cable between the measuring instrument and the preamplifier to be of considerable length. The preamplifiers themselves are powered from either the preamplifier input socket of a measuring amplifier or a frequency analyser, or alternatively from special power supplies if it is required to connect the preamplifier to other instrumentation.

For 1 inch and 1/2 inch microphones the Preamplifier Type 2619 is recommended. The 1/2 inch Type 4148 can be used with the Type 2619 providing it is powered from the two channel, battery operated Microphone Power Supply Type 2804.

The 1/2 inch Type 4125 has been especially designed for low cost applications. It is used with the Preamplifier Type 2642 and the two channel, battery operated Microphone Power Supply Type 2810.

The 1/4 inch and the 1/8 inch microphones are designed to operate in conjunction with the Preamplifier Type 2618. The figure overleaf shows some of the microphone equipment already described.

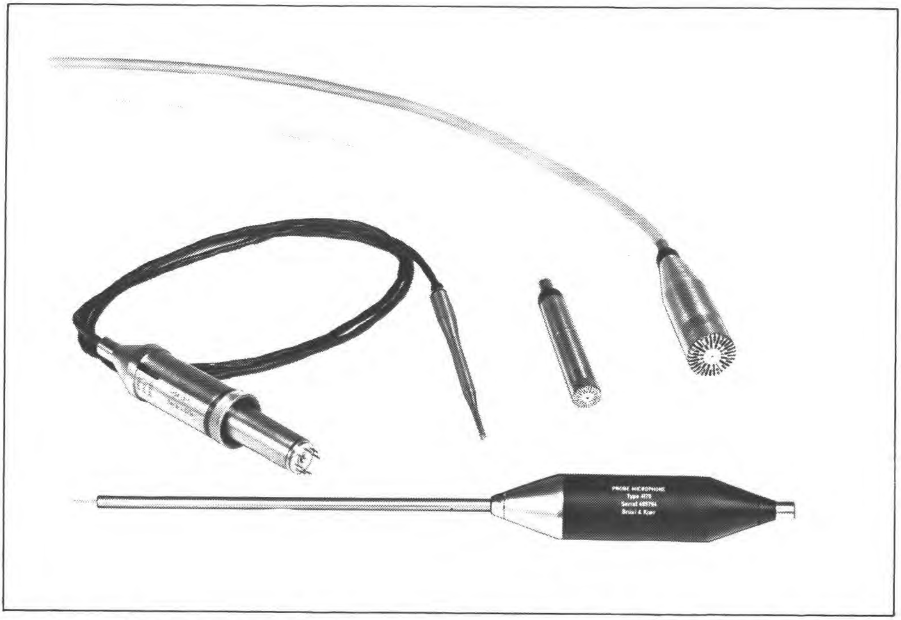

Fig.5.4. Microphones and preamplifiers: top left, 1/8" Microphone + UA 0160 + Preamplifier Type 2618; top centre, 1/2" Microphone + Preamplifier Type 2619; top right, 1" Microphone + Adaptor DB 0375 + Preamplifier Type 2619; lower, Probe Microphone Type 4170 with built-in preamplifier

For powering the preamplifier, one of the three power supplies shown below can be employed.

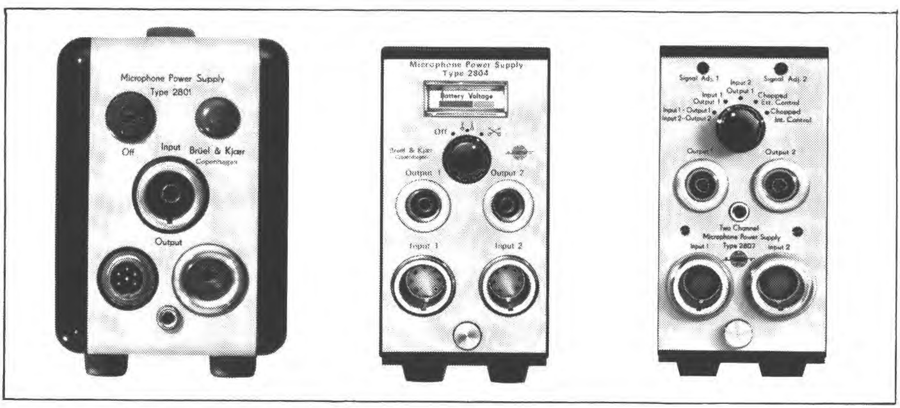

Fig.5.5. Microphone Power Supplies; left to right, Types 2801, 2804 and 2807

The mains driven Power Supply Type 2801 can supply all voltages for the microphone assemblies using 200 V polarisation voltage. The battery driven Power Supply Type 2804 can be used only with the Preamplifier Type 2619. It supplies all the necessary voltages for two microphone assemblies and can be adjusted to give 28 or 200 V polarisation voltage. The mains operated, two channel Power Supply Type 2807 can supply all voltages for two microphone assemblies using 200 V polarisation voltage and allows automatic switching between the measuring points, a very useful feature when measuring, for example, the sound insulation between two rooms.

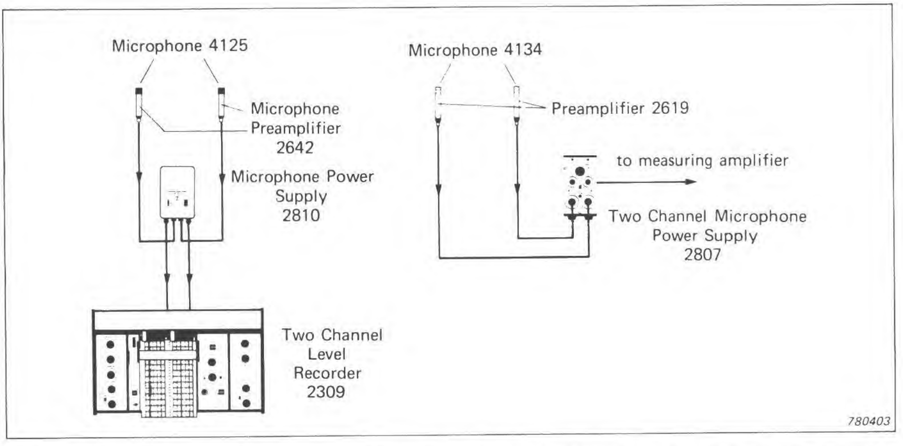

Fig. 5.6. The use of two microphones with a single microphone power supply

Calibration of Microphones and Associated Measuring Systems

Microphones and their associated measuring systems can be rapidly and accurately calibrated by using one of the two following instruments either the Sound Level Calibrator Type 4230 or the Pistonphone Type 4220. These are battery driven instruments which produce a known sound pressure level at a known frequency. The accuracies obtainable from calibrating with these instruments are ± 0,25 dB and ± 0,2 dB respectively. The Miniature Hydrophone Type 8103 can be calibrated as easily as the microphones by employing the Hydrophone Calibrator Type 4223.

It is good policy to calibrate the instrumentation before and after each set of measurements. If a tape recorder is employed, a calibration tone can be recorded on the magnetic tape together with a verbal description of the amplifier settings.

Fig.5.7. Upper, Sound Level Calibrator Type 4230; lower, Pistonphone Type 4220

Selection of an Accelerometer

The vibration of walls, windows, pipes and other elements in a building can be measured by attaching an accelerometer to the vibrating structure. An accelerometer is an electromechanical transducer which gives, at its output terminals, a voltage proportional to the acceleration to which it is subjected. (Ref. 39).

The choice of an accelerometer depends on such factors as the type of vibration to be measured, the level of vibration, the frequency of the vibration and the mass of the vibrating structure. In order not to change the character of the motion of the vibrating structure, the accelerometer must have a mass which is small compared to the mass of the vibrating structure. In architectural acoustics, this is usually the case as one is dealing with massive structures such as walls and floors. Windows can, however, possess a very low mass and therefore their vibration characteristics can be influenced considerably by the addition of a mass in the form of an accelerometer. In these cases a very lightweight accelerometer should be used. When mounting accelerometers on walls, floors etc., the simplest method is to use bees' wax. A thin layer of wax is spread on the surface and the accelerometer pressed firmly onto the wax. The great advantage with this method is that the accelerometers can easily be removed and rapidly placed elsewhere. Another simple method is to attach the accelerometer to the structure with double sided adhesive tape. Other methods of mounting include the use of epoxy based cements and the use of accelerometer with magnetic bases for mounting on metal panels etc. Every B & K accelerometer is individually calibrated and is delivered with an individual calibration chart. The calibration of an accelerometer and the complete measuring system can however be checked by using the Calibrator Type 4291.

In building acoustics, it is often the low frequency vibrations of walls and other massive structures that are of interest. It can therefore be useful to limit the high frequency response of the accelerometer by inserting the Me-

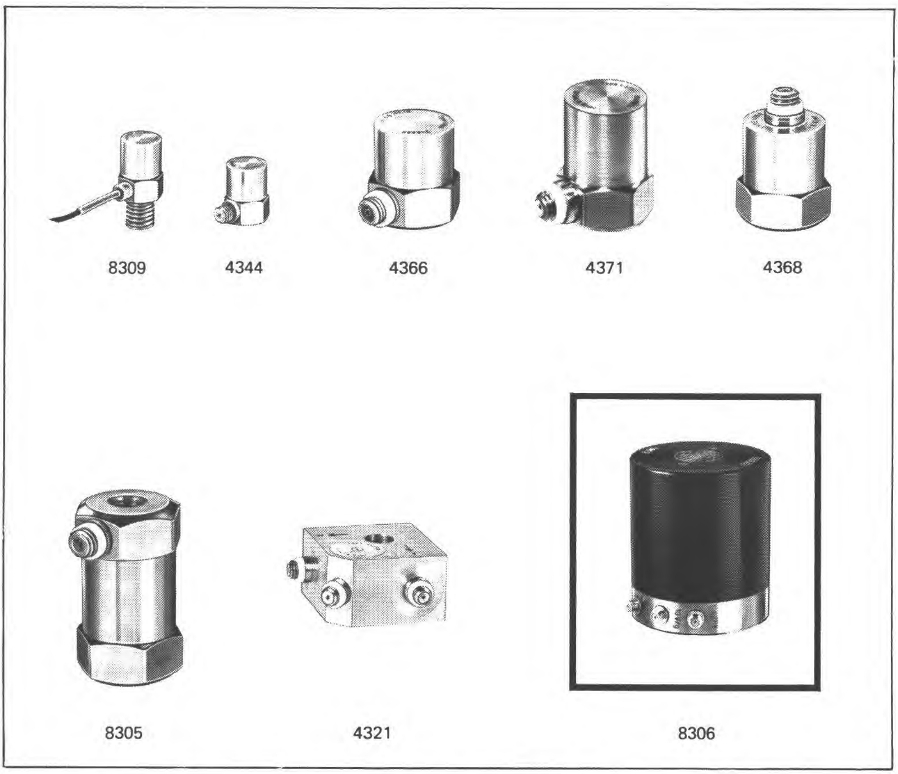

Fig.5.8. A selection of B & K accelerometers

chanical Filter for Accelerometers Type UA 0559 between the accelerometer and the mounting point. The cutoff frequency depends on the accelerometer mass but the —3 dB point is typically 4 kHz with an accelerometer Type 4370 which weighs 54 g. Additional masses can be added to regulate the cut-of frequency. Fig. 5.8 shows a selection of B & K accelerometers.

The principle features of the B & K range of accelerometers are shown in Table 5.2 together with their main areas of application.

Type No.	Charge Sensitivity (pC/ms−2)*	Voltage Sensitivity (mV/ms−2)*	Freq. Range to + 10% limit (Hz) △	Mounted Resonance (kHz)	Weight (grams)	Main area of application
4366	~ 4,5	~ 4	0,2 – 7000	22	28	General purpose
4367	~ 2	~ 1,5	0,2 – 9700	32	13	
4368	~ 4,5	~ 4	0,2 – 7000	22	30	
4369	~ 2	~ 1,5	0,2 – 9700	32	14	
4370●	10 ± 2%	~ 10	0,2 – 6000	18	54	Low level measurements
4371●	1 ± 2%	~ 1	0,2 – 12000	35	11	
4344	~ 0,25	~ 0,25	1 – 21000	70	2	Miniature accelerometers for high level and high frequency measurements. Ideal for delicate structures and confined spaces
8307	~ 0,07	~ 0,22	1 – 25000	75	0,4 excl. cable	
4321●	1 ± 2%	~ 0,8	1 – 12000	40	55	Triaxial. For measurements in three mutually perpendicular directions.
8305	~ 0,12	–	0 – 4400(2%)	30	40	Standard reference accelerometer.
8306●	1000 ± 2%*	1000 ± 2%	0,2 – 1000	4,5	500	Low frequency and low level measurements.
8308●	1 ± 2%	~ 1	1 – 10000	30	100	Permanent vibration monitoring on industrial machinery etc.
8310●	1 ± 2%	~ 1	1 – 10000	30	100 excl.cable	
8309	~ 0,004	~ 0,03	1 – 60000	180	3 excl. cable	High level shocks.

● Uni Gain® types
* 1 $ms^{-2} \cong 0,1$ g
△ Lower limiting frequency dertermined by preamplifier used and environmental conditions

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Table 5.2. Principle features and main areas of application of the B & K range of accelerometers

Preamplifier selection

Accelerometers, just like microphones, have to be followed in the measurement chain by a preamplifier. The preamplifier is necessary for two reasons. Firstly, to convert the high output impedance of the accelerometer to a lower value suitable for input to measuring amplifiers and analysers; secondly, to amplify the relatively weak output signal from the accelerometer if the following instrumentation does not have a sufficiently high sensitivity.

There are basically two kinds of amplifier, namely the voltage amplifier and the charge amplifier. The voltage amplifier is designed to present the highest possible resistance to the accelerometer while the input capacitance is kept low. The voltage amplifier is usually placed near the accelerometer with only a short interconnecting cable in order to avoid loss of sensitivity due to the capacitance of the cable. The voltage amplifier allows the use of long cables between the preamplifier and the measuring amplifier. The charge amplifier is designed to present both a very high input capacitance and resistance to the accelerometer. With the charge amplifier, the variation in input signal due to the varying capacitative loading of the accelerometer is compensated by a capacitance feedback from the preamplifier. An advantage of the charge amplifier therefore, is that very long cables can be used between the accelerometer and the preamplifier without changing the sensitivity of the measuring system. Nowadays, the most widely used accelerometer preamplifier is a charge amplifier. The table below shows the main features of the voltage and charge amplifiers in the B & K range.

It is not necessary for the accelerometer to be followed by a preamplifier in the following cases: when using the high sensitivity accelerometer Type 8306 which has a built in preamplifier and when the signal from the accelerometer is fed into measuring instruments possessing a built in preamplifier such as the vibration meters and the sound level meters.

Type	Voltage or Charge	Adj. Gain	Power Supply	Frequency Range	Meas. Mode	Filters LP	Filters HP	Features	Application
2626	Charge	X	AC mains	0,3 Hz to 100 kHz	Acc.	X	X	3 Digit sensitivity conditioning Wide range of output levels. Two signal level indicator lamps. Direct or transformer output	Multi-purpose, vibration monitoring and measurement
2634	Charge	X	Ext. Batt.	1 Hz to 200 kHz	Acc.	–	X	Small, rugged construction Suitable for single ended and differential output transducers	Multi-channel, vibration monitoring and measurement
2635	Charge	X	Ext. or Int. Batt.	0,1 Hz to 100 kHz	Acc. Vel. Dis.	X	X	3 Digit sensitivity adjustment. Wide range of output levels. Built-in test oscillator	Multi-purpose, vibration monitoring and measurement
2650	Voltage & Charge	X	AC mains	0,3 Hz to 200 kHz	Acc.	X	X	4 Digit sensitivity adjustment Wide range of output levels Two signal level indicator lamps Built-in test oscillator	Calibration and reference work
2651	Charge	X	Ext. Batt.	0,003 Hz to 200 kHz	Acc. Vel.	–	X	Grounded or Floating input. Overload indicator.	Multi-channel, vibration monitoring, vibration severity measurement 780145

Table 5.3. Principle features of the B & K range of accelerometer preamplifiers

Calibration of a vibration measuring system

The accelerometer and its following chain of measuring instruments can be simply calibrated by employing the Accelerometer Calibrator Type 4291. This calibrator is a battery operated instrument with a built in shaker table producing a reference acceleration level of 10 m.s.$^{-2}$ peak at a frequency of 79,6 Hz (= 500 rad/s). The calibration method involves vibrating the accelerometer at 10 m.s.$^{-2}$ peak and adjusting the sensitivity of the indicating instrument for a full scale peak meter indication of 1,0 or for an rms indication of 0,707. The measuring sytem will then give a direct indication of the level of acceleration in m.s.$^{-2}$ peak or m.s.$^{-2}$ rms. (Ref. 40).

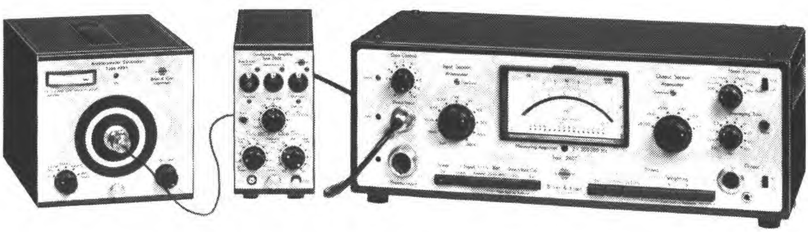

Fig.5.9. Accelerometer calibration : left to right, Accelerometer Calibrator Type 4291 with accelerometer mounted on the vibration table; Conditioning Amplifier Type 2626; Measuring Amplifier Type 2607

5.5. AMPLIFIERS, FILTERS, ANALYSERS AND RECORDERS

Portable instruments for sound measurement

For field measurements, portable instrumentation is an advantage. In many situations a sound level meter can be used in conjunction with a portable filter set thus combining the microphone, amplifier and filter in a single easily manageable instrument.

All the instruments shown in Fig.5.10, except the 2219, have an output for use with chart or tape recorders. The 2210, 2218, 2209 and the 2203 can be used in conjunction with filter sets 1613 and 1616. It is often the case that a straight forward measurement of the sound pressure level within a room, is not sufficient and that a time average of the sound pressure level is required. This time average can be determined by using either the Precision Integrating Sound Level Meter Type 2218 or the Noise Level Analyzer Type 4426.

N.B. If the 4426 be used to determine the mean sound pressure level, L, then the microphone must be powered from a source other than the 4426. The microphone output is then fed into the 4426 via the "Direct" BNC socket on the rear panel of the instrument. This is necessary because the "Preamplifier" input of the 4426 is A-weighted so that if this input were used, the result would be L_{eq} and not L.

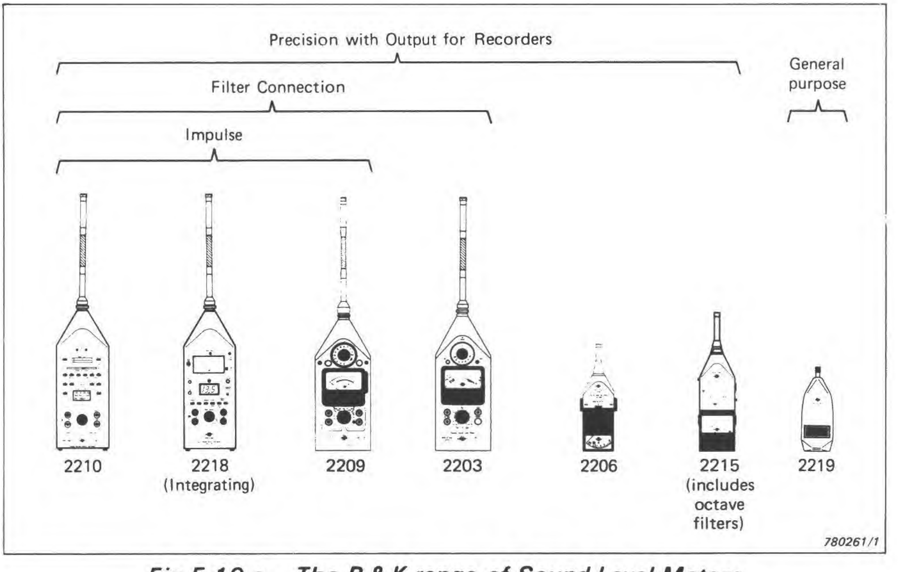

Fig.5.10.a. The B & K range of Sound Level Meters

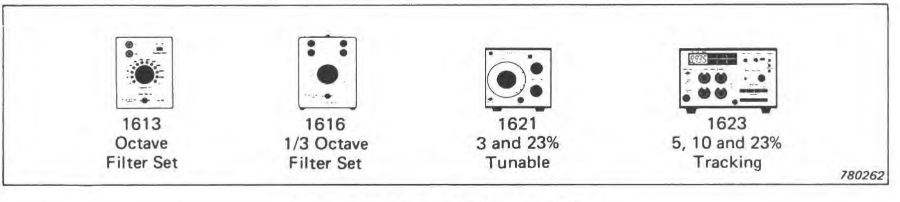

Fig.5.10.b. Portable filters

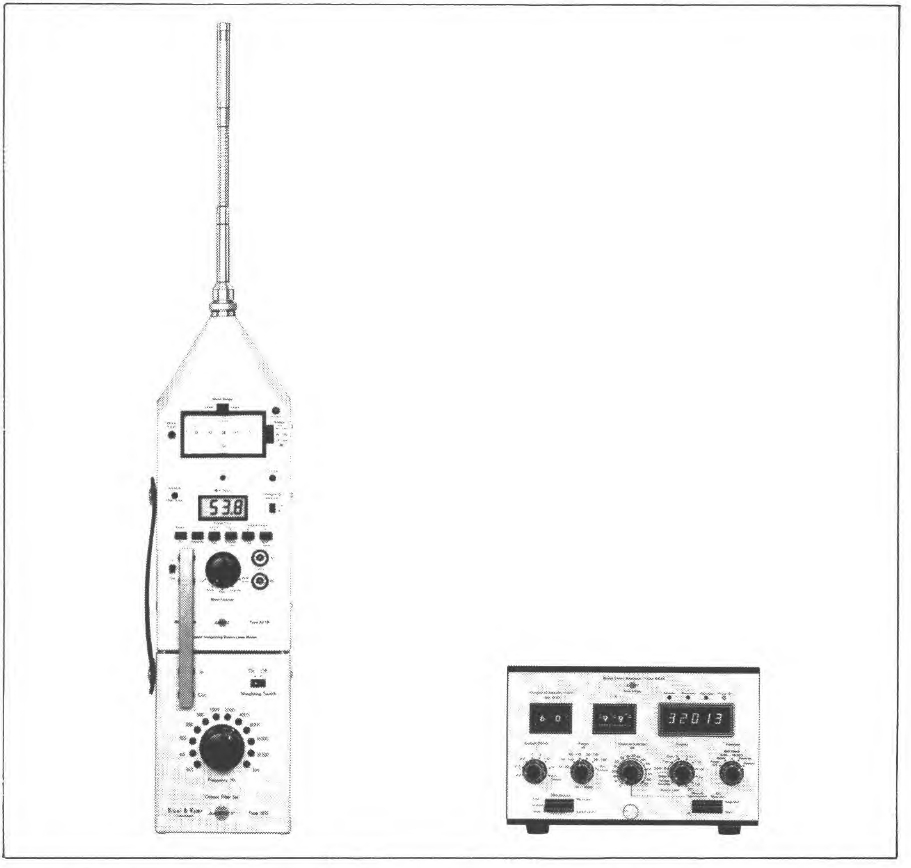

Fig.5.11. Time averaging of sound pressure: left, Integrating Sound Level Meter Type 2218 fitted with Octave Filter Set Type 1613; right, Noise Level Analyzer Type 4426

Portable instruments for vibration measurement

An accelerometer can be used in conjunction with a sound level meter (types 2203, 2209, 2210 or 2218) and an Integrator ZR 0020 to yield a portable system for measuring the acceleration, velocity and displacement due to vibrations. To analyse the vibration signal in frequency bands either the Third Octave or Octave Filter Type 1616 or 1613 can be directly attached to the bottom of the sound level meter (Fig.5.12).

For more detailed on site analysis, the Portable Vibration Analyzer Type 3513 is recommended. This self contained system consists of the Vibration

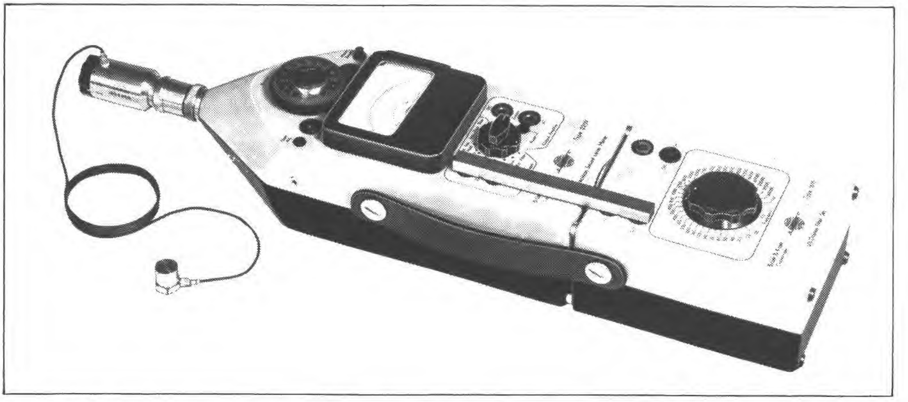

Fig.5.12. Accelerometer used in conjunction with a sound level meter, a filter set and an Integrator Type ZR 0020 to yield a portable system for analysing vibrations in third octave bands

Meter Type 2511 and the Tunable Band Pass Filter Type 1621 mounted in a hard plastic carrying case permitting analysis in 3% and 23% (i.e. third octave) frequency bandwidths.

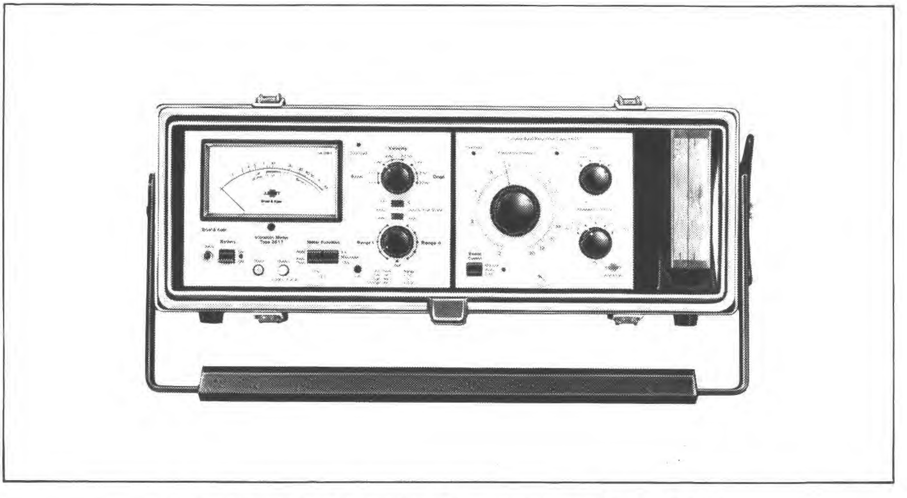

Fig.5.13. Portable Vibration Analyzer Type 3513

Measuring amplifiers and filters

In the laboratory, the signal from the transducer (microphone or accelerometer) can be measured and analysed by mains operated instrumentation such as a combination of a Measuring Amplifier Type 2606, 2607 or 2608 and a Band Pass Filter Set Type 1617 or 1618. These three amplifiers are basically low noise, wide range, calibrated voltmeters and have many common components, the main difference being in their rectifying circuits. Any of these amplifiers may be employed with either of the two filter sets. The various combinations give different measurement and analysis possibilities. The basic system, for example consisting of the Type 1618 and the Type 2608 is intended for rms measurements in the audio frequency range with the option of filter selection being performed by a level recorder. The optimum use of all the measuring modes and control possibilities is obtained with a combination of the Type 1617 and Type 2607 enabling measurements and analysis to be performed in the frequency range from 2 Hz to 160 kHz obtaining rms and peak values from all commonly encountered random, quasi-random, periodic and complex waveforms.

Frequency Analyzers

In the generalised chain of instrumentation shown in Fig.5.1, the filter and amplifier on the receiving side are often contained in a single instrument known as an analyser.

For analysing a signal with a tunable constant percentage band pass filter of bandwidth 1%, 3%, 10% and 23% (i.e. one third octave) the Frequency Analyzer Type 2120 and Type 2121 are recommended. For narrow band frequency sweeps, for example, in determining the frequency response of building elements to an applied force or the modal response of a room to a swept pure tone, the Heterodyne Analyzer Type 2010, consisting of a frequency selective measuring amplifier and a beat frequency oscillator, is recommended.

A much more sophisticated instrument for the analysis of signals is the Digital Frequency Analyzer Type 2131. This instrument enables signals to be analysed in real time, in octave or in third octave bands and the spectra can be displayed on a screen in the form of a bar graph. The spectra are then stored for later recall and comparison. With this facility, for example, the spectra of the noise on either side of a wall in sound insulation measurements, can be displayed simultaneously on the screen.

On coupling the 2131 to a programmeable desk top calculator, the possible applications of the instrument are greatly increased. Measurements of sound insulation and reverberation time for example can be fully automated, the reverberation decay curves being displayed on the screen.

For analysing either sound or vibration signals in narrow band and for the capturing of transient signals, the Narrow Band Spectrum Analyzer Type 2031 is recommended. The 2031 has a screen where either the time history, the instantaneous spectrum or the averaged spectrum can be displayed.

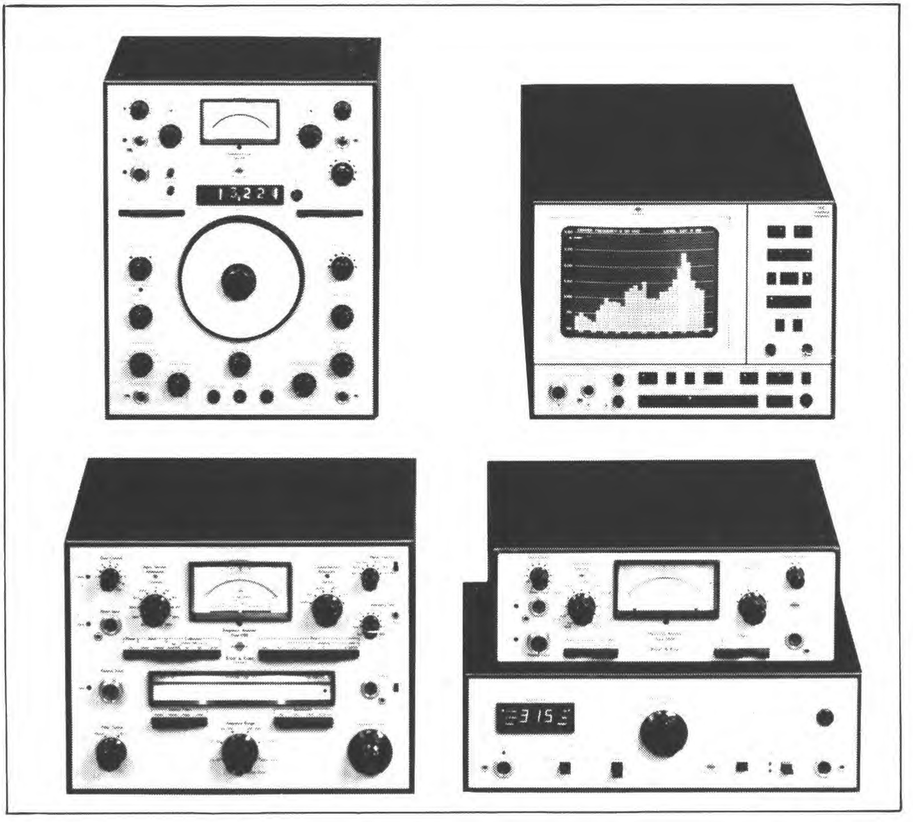

Fig.5.14. Frequency analysers: upper left, Heterodyne Analyzer Type 2010; lower left, Frequency Analyzer Type 2120; upper right, Digital Frequency Analyzer Type 2131; lower right, Measuring Amplifier Type 2608 in combination with the Filter Set Type 1618

Tape Recorders

It is often convenient to record the signals from the transducer on magnetic tape and to perform the analysis later in the laboratory. Two tape recorders are produced by B & K which can be used for this purpose, the Type 7003 and the Type 7004. The 7003 is a four channel frequency modulated recorder intended principally for vibration measurements and has two speeds for frequency transformation, 1,5 ips and 15 ips. Using the 15 ips, the recorder has a linear frequency response up to 12,5 kHz which is sufficient for most measurements in building acoustics.

Chart Recorders

During the analysis of the signal, a permanent recording of the results can be obtained by synchronising a chart recorder with the analyser or the generator in order to produce a frequency scan, a decay curve, a time function etc. The chart recorder can be a level recorder such as the Types 2306, 2307 or the 2309 or the X-Y Recorder Type 2308. The Level Recorders Types 2306 and 2309 are battery operated and possess one and two channels respectively. All three level recorders are capable of recording on both linear and logarithmic scales.

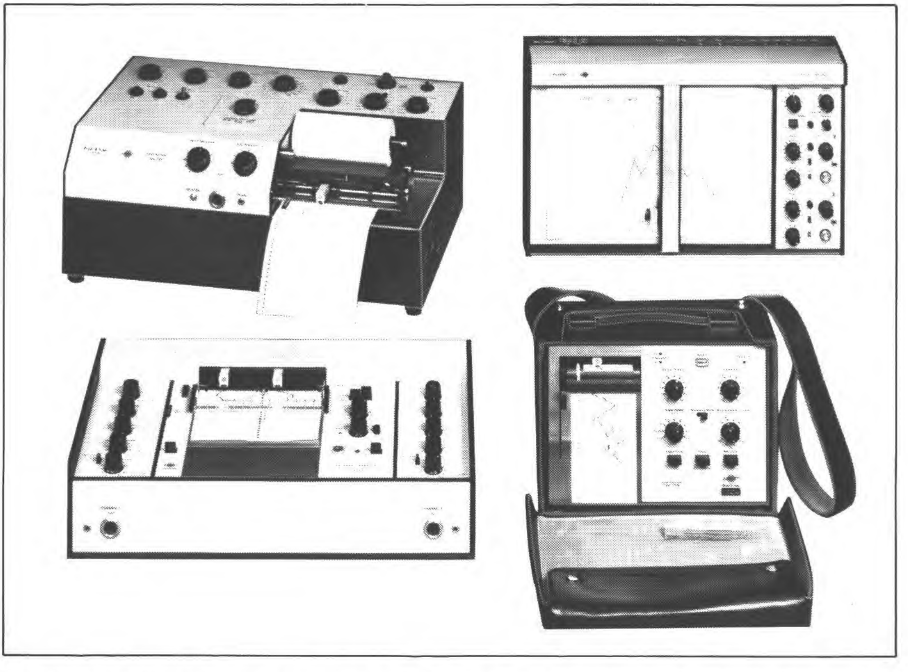

Fig.5.15. Chart Recorders: top left, Level Recorder Type 2307; top right, X-Y Recorder Type 2308; lower left, Two Channel Level Recorder Type 2309; lower right, Level Recorder Type 2306

The X-Y Recorder Type 2308 is mains operated and is intended for the linear recording of DC signals on paper of A4 size. When the analysis is performed by an instrument such as the Statistical Analyzer Type 4426 or the Digital Frequency Analyzer Type 2131, the results can be printed out by the Alphanumeric Printer Type 2312.

Calculators

Many acoustical measurements which would otherwise be repetitive and longwinded can be greatly simplified by employing a Digital Frequency Analyzer Type 2131 in conjunction with a programmable desk top calculator. The calculators recommended by B & K for use with the 2131 are the Tektronix 4051 and the Hewlett-Packard HP 9825A (Ref.30). Two program packages are produced by B & K for performing basic acoustic measurements such as reverberation time, sound power, L_{eq} etc., with these calculators. One program, the BZ 0011, is for use with the 2131/4051 system and the other BZ 0012 is for use with the 2131/9825A system. A listing of the programs available on BZ 0011 is shown in Fig.5.16. The Narrow Band Spectrum Analyzer Type 2031 can also be used in conjunction with a calculator.

B & K

B & K ACOUSTICAL CALCULATIONS

PROGRAM PACKAGE

BZ 0011

B K

Brüel & Kjær

Place the template for the USER DEFINABLE KEYS
and make your choice on these keys:
KEY 1: This program.
2: Input a selectable number of spectra from the 2131 and calculate the average.
3: Difference between the averaged spectra in A & B.
4: Calculate PNDB of each spectrum.
5: Stevens calculation on the averaged spectrum.
6: 3-D plot of the number of spectra in A or B buffer.
7: Reverberation time.
8: Sound power. - Precision method for broad-band sources in reverberation rooms (ISO 3741).
9: L_{eq} calculation.
10: 1/12 octave input from 2131.

PLEASE, MAKE YOUR CHOICE!

770717

Fig.5.16. Listing of the programs available on the Program Package BZ 0011

6. SUGGESTED INSTRUMENTATION

6.1. INTRODUCTION

The foregoing chapter describes the various instruments in a generalised measurement chain. Now various instrument arrangements will be described which are intended for particular architectural acoustic measurements. It should be borne in mind, however, that the instrument arrangements and the measurements which are mentioned in this section constitute by no means an exhaustive survey of all the possible applications of B & K instruments in this field.

6.2. REVERBERATION TIME

One of the most important characteristics of a room or auditorium is its reverberation time. To a large extent it is the reverberation time which determines the suitability of a room for a particular purpose. Reverberation chambers are specially designed rooms found in most acoustic laboratories. Such chambers have hard, non-parallel surfaces in order to have the minimum sound absorption and therefore the longest reverberation time possible, often in excess of 12 seconds. Reverberation chambers are used for the measurement of the absorption coefficients of materials and for the determination of the sound power of noise sources. Long reverberation times are also encountered in stairwells and in churches. Another special type of room used for measurements of directivity and the determination of sound power is the anechoic chamber. All surfaces of such a chamber are covered with sound absorbing material thus preventing the reflection of sound energy which yields, ideally, an enclosure where the reverberation time over the whole audio range is zero. The reverberation times usually met with in architectural acoustics fall between these two extremes (see Fig.2.12).

The technique for measuring reverberation time depends on the particular circumstances but the procedure is basically the same, i.e. the sound source is introduced into the room the source is abruptly stopped and the microphone monitors the decay of the sound pressure level within the room. The decay in sound level is recorded by, for example, a level recorder and the reverberation time is measured from the slope of the decay curve. This procedure can be repeated for each frequency band of interest.

It is sometimes possible to utilise those sound sources which already exist in the room,e.g. the organ in the case of a church, the orchestra in the case of a concert hall. However, the room is usually excited by one of the following methods

1) Pistol shot
2) Wide band noise
3) Narrow band noise

Each method has its advantages and disadvantages. Probably the simplest way of exciting a room is to use a pistol shot. This method has, however, gone out of favour because of its lack of reproducibility and its lack of energy at the lower frequencies.

A better method of excitation is to use electronically generated noise consisting of a wide band of frequencies delivered to the room via an amplifier and a loudspeaker. Two types of wide band noise are in common use; white noise which has constant energy per unit of frequency and pink noise whose energy content is inversely proportional to frequency (i.e. —3 dB per octave or —10 dB per decade). It is usual to filter wide band noise into octave or third octave bands before delivering the signal to the loudspeaker in order that a greater sound pressure level can be obtained in the frequency band of interest for a given loudspeaker with a given power handling capacity. For example, consider a loudspeaker with a flat frequency response over eight octaves. Assume that the maximum sound pressure level obtainable in a particular octave band is x dB when the loudspeaker is fed with wide band pink noise. When this noise is restricted to the octave band of interest by the use of filters then as the bandwidth is reduced from 8 to 1 octave (i.e. $2 \times 2 \times 2$) then the maximum sound pressure level obtainable is increased by 9 dB to $(x + 9)$ dB that is, by 3 dB for each halving of bandwidth.

A band of noise is far more satisfactory than pure tone excitation because a pure tone may well overemphasize the natural resonances of the room. A band of noise is necessary in reverberation measurements in order to obtain an average of the contributions of all resonances of the room to the total sound pressure level in a particular band of frequencies. Another way of overcoming the problem of room resonances is to use the warbled tone facility of the B & K generator Type 1023. Whatever sound source is employed, the sound pressure levels produced in the room should be at least 40 dB above the background noise level in all the frequency bands of interest in order to obtain an adequate decay curve.

When measuring the reverberation time of a room in a laboratory then one may, within reason, take as long a time as necessary and the instruments may remain in the room from one day to the next. Thus making measurements in 15 third octave bands for several positions of the microphone in the room poses no special problem. However, for an occupied living room or in a newly constructed flat on a building site, the measurements have to be performed as quickly as possible. For on site measurements, portable equipment is highly desirable as a suitable mains supply of electricity is not always available. One way of saving valuable time is to record the on site measurements on a tape recorder and perform the necessary analysis in the laboratory.

Problems arise when measurements have to be made in the presence of an audience. One cannot reasonably expect an audience who have paid to listen to a concert, for example, to endure a measuring technique which employs bursts of noise in 15 frequency bands. In such cases a single burst of pink noise or a pistol shot would be suitable alternatives, the sound pressure level decays being recorded on a tape recorder for analysis later.

Having chosen a sound source, one then has to decide where to position it. In some rooms the choice of the position of the source is obvious, e.g. in the pulpit in the case of a church, at the lectern in a lecture theatre and on the stage in a concert hall. Other positions of the source around the room help to locate excessively long RTs due to room modes.

In small rectangular rooms, the best position for the sound source is in a corner of the room, i.e. where all the natural frequencies of the room possess a pressure maximum.

In small rooms a single value of reverberation time is sufficient to characterise the reverberation of the room but in large rooms and auditoria this is not possible as the reverberation time can vary considerably from place to place throughout the volume of the room. In a thorough investigation of the acoustical properties of such rooms therefore, the reverberation decay should be monitored at several positions over the area where the audience are seated.

The following four instruments arrangements all comply with ISO 3382 and ISO/R 354.

Portable Arrangement — Pistol Shot Method

This method has the advantages that it is quick and the necessary instrumentation is kept to a minimum (Fig.6.1). The disadvantages are that the noise spectrum is limited in frequency content and lacks reproducibility. To check that sufficiently high sound pressure levels are produced at both high and low frequencies, it is advisable to frequency analyse the impulse before

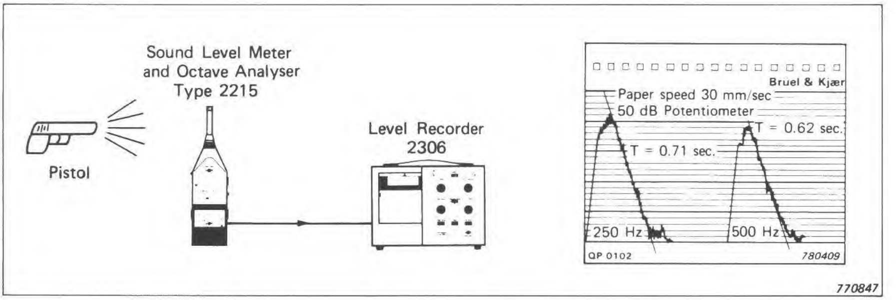

Fig.6.1. Portable arrangement - Pistol shot method

making any reverberation measurements. This analysis should preferably be performed in an anechoic room.

Portable Arrangement — Filtered Noise Method

The instrumentation in Fig.6.2 can be used to measure the reverberation time in the seven octave bands from 125 Hz to 8000 Hz and in wide band. If the reverberation time needs to be determined in third octave bands then the Sound Level Meter Type 2215 can be replaced by a Sound Level Meter Type 2203 and a Third Octave Filter Set Type 1616. The writing speed of the 2306, limits the shortest reverberation time measurable with this arrangement to 0,24 s. For shorter reverberation times either the Level Recorder Type 2307 should be used which is however mains driven, or the reverberation decays should be recorded on tape and analysed later in the laboratory. For measuring the reverberation in several positions in the auditorium, it is often convenient to record the reverberation decays on a tape recorder and to analyse the recordings later.

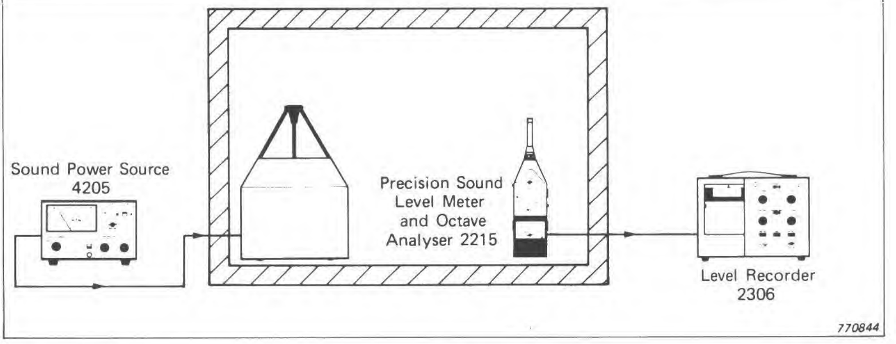

Fig.6.2. Portable arrangement - Filtered wide band noise method

Automatic Arrangement - Paper Loop Method

When many reverberation time measurements are required at various positions in the auditorium and the portability of the instruments is not important e.g. in concert halls, theatres, churches etc., it is advantageous to use an automated system such as the one shown in Fig.6.3.

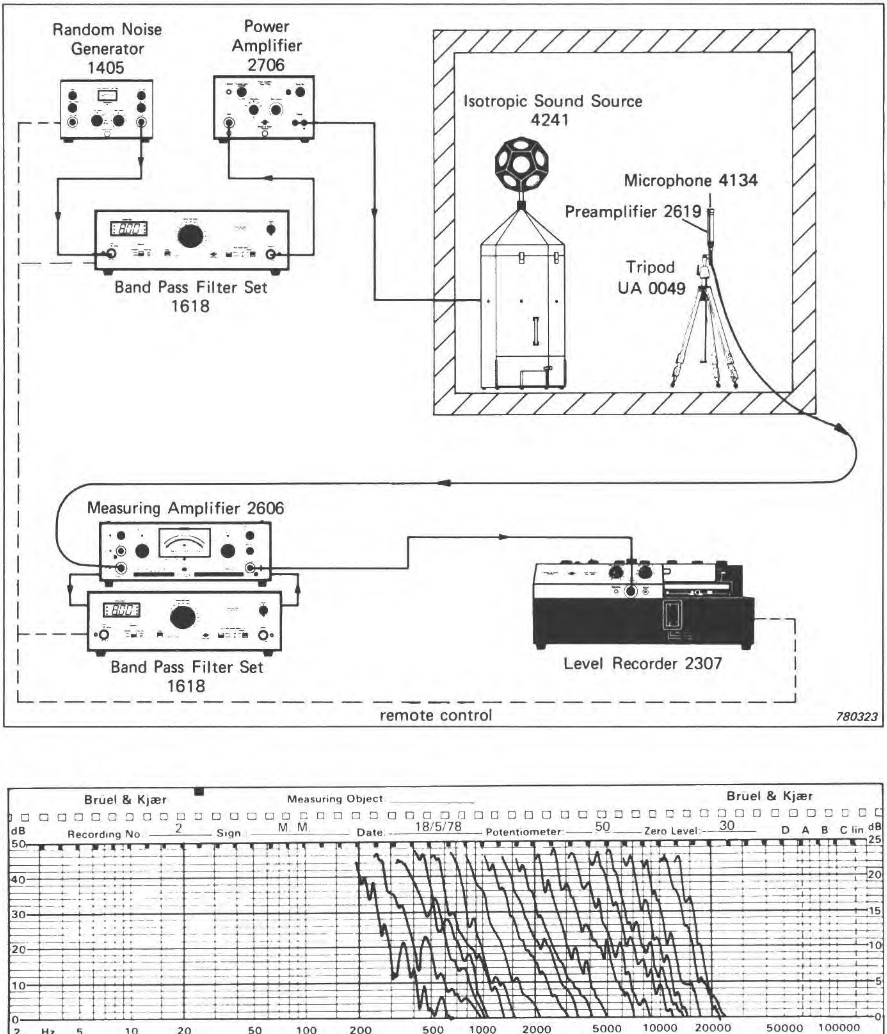

Fig.6.3. Automatic arrangement - Paper loop method

The generating side of the arrangement consists of a random noise generator which sends a white noise signal through the filter and amplifier to the Sound Source Type 4241. The receiving side consists of a microphone, amplifier, filter set and a Level Recorder Type 2307. The filters on both generating and receiving sides are set manually to the same bandwidths and centre frequencies. The level recorder then automatically operates in the correct sequence the starting and stopping of the generator, the stepping of the filters in unison and the lifting and lowering of the recorder pen. When the sound source is switched off, a decay curve is traced by the level recorder. The reverberation time is determined from the slope of the trace which represents the decay in dB per second. A special protractor, SC 2361, is supplied for this purpose. To obtain a series of curves similar to those shown in Fig.6.3, the recording paper should be made into a loop as shown in Fig.6.4 consisting of two chart lengths less one sprocket hole. The start of each decay curve is separated from the start of the next by 5 mm which corresponds with the third octave spacings on the preprinted paper. If third octave filtering be used then each decay curve is representative of a different third octave band. If octave filtering be used then for each octave band there will be three similar decay curves.

Fig.6.4. Automatic arrangement - Loop of recording paper is fitted on the Level Recorder

Automatic Arrangement — Digital Frequency Analyser/Calculator Method

This method employs a Digital Frequency Analyser Type 2131 and a Noise Generator Type 1405 which are controlled via an IEC interface by a desk top calculator containing a B & K Acoustic Program Package Type BZ 0011 or 0012. Once the controls of the 1405 and its associated amplifiers have been set manually, the system may be programmed to automatically repeat a cycle of operation a predetermined number of times. The cycle of operation consists of the noise source being switched on for a long enough time for a steady sound field to be established in the room; the source is switched off and the spectrum of the decay is monitored by the 2131 and transmitted to the calculator; the noise source is then switched on again and the cycle is repeated.

During the decay the 2131 transmits spectra to the calculator at selected read-in intervals, the shortest interval being 44 ms. As the read-in process can always be referred to a definite point in time i.e. the switching off of the noise generator, it is possible to average the spectra for a number of reverberation decays at identical points in time. For example, suppose 40 decays are measured with a read-in interval of 44 ms. All the 40 spectra recorded 44 ms after the generator is switched off are averaged to give one spectrum, all the 40 spectra recorded 88 ms after the switching off of the generator are averaged to give a second spectrum and so on. These 40 averaged spectra can then be plotted on an amplitude-frequency-time landscape (see Fig. 6.6). In this averaged landscape, the points making up the decay curves in each frequency channel are accurately defined. Therefore by displaying

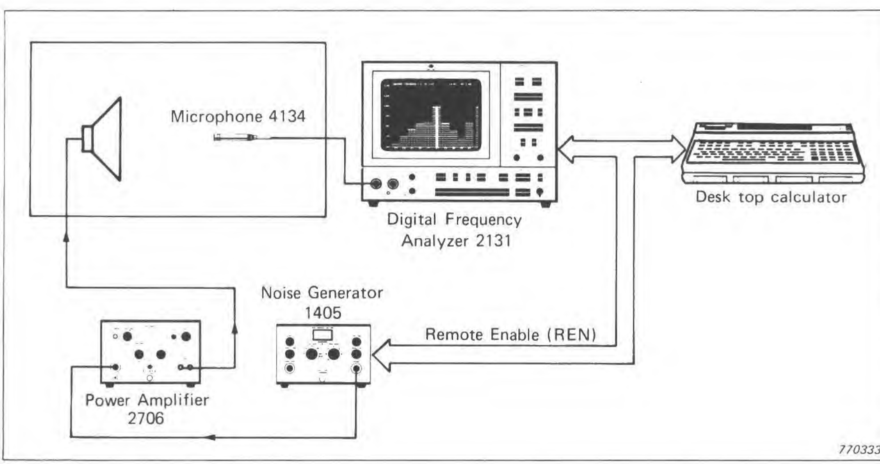

Fig. 6.5. Automatic arrangement - Digital frequency analyser/calculator method

one of these decay curves on the screen of the 2131, the presence of two or more slopes can easily be detected and the corresponding reverberation times calculated. For more detailed information on this method, reference should be made to B & K Technical Review No.2-1977.

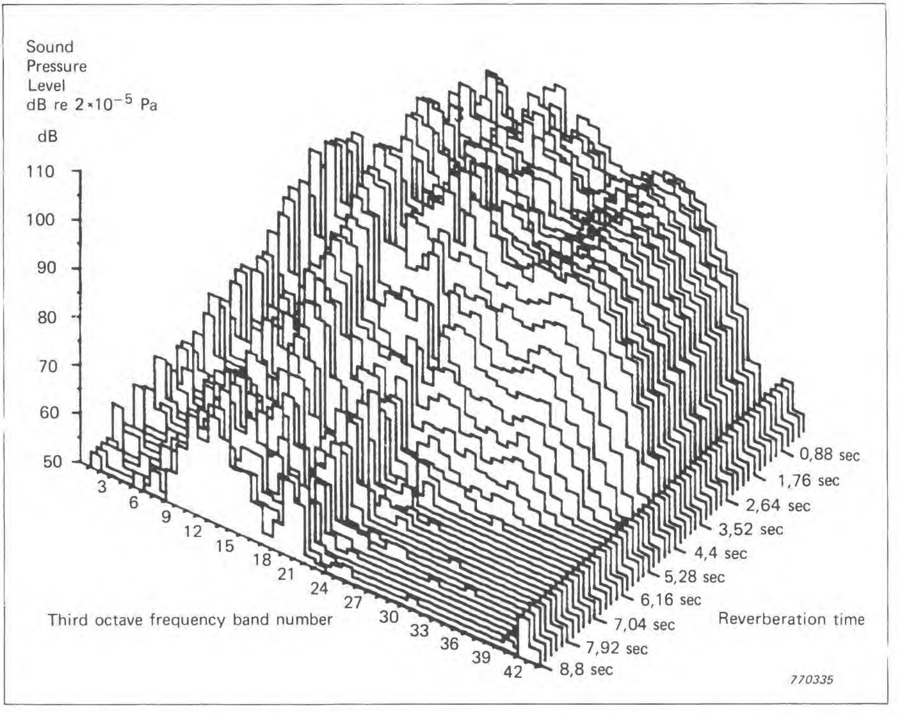

Fig.6.6. Representation of reverberation as a 3-D landscape where the axes are amplitude, frequency and time

6.3. SOUND DISTRIBUTION

Information about the way in which the sound pressure level varies with position within a room is of particular interest to the designers and constructors of theatres, concert halls and other auditoria. In these buildings, it is vitally important that the sound be distributed as uniformly as possible over the area occupied by the audience.

Design Stage — Model Techniques

The effect of a particular design on the sound distribution can be investigated by using a small scale model of the proposed building. It is theoretically possible to perform in an acoustic model all acoustic measurements which are usually made in the full sized buildings e.g. reverberation time, sound distribution etc. However, there are a number of practical problems to be solved before the model can be realised. Three major problems are the selecting of suitable transducers with acceptable sensitivity and frequency response, the modelling of the absorption coefficient for the walls and the modelling of the absorption coefficient for the air. In spite of difficulties acoustic modelling has been used successfully as a design tool on many occasions e.g. Sydney Opera House, Edinburgh Opera House (Ref.8). The frequency of the excitation noise in model experiments should be increased in the same proportion as the model has been scaled down. If, for example, a model of a concert hall is to be made at one tenth of the full scale then all frequencies should be increased by a factor of ten. This frequency transformation can be obtained by recording the excitation noise on a tape recorder then playing the tape back at ten times the recording speed. Alternatively, sound sources rich in high frequency content can be used e.g. the discharging of an electrical spark or an ultrasonic whistle. In certain circumstances the transmitting transducer can be a condenser microphone (Ref.31).

Having produced the required sound field within the model, a small microphone or a probe microphone is used as a receiver. (Small microphones are used for modelling purposes because they have good high frequency response and have little effect on the sound field.) The signal from the receiving microphone is then amplified and can then be analysed in the usual manner either before or after the reverse frequency transformation is performed. By changing the position of the receiving microphone in the model room, the sound distribution within the model can be investigated (Fig.6.7).

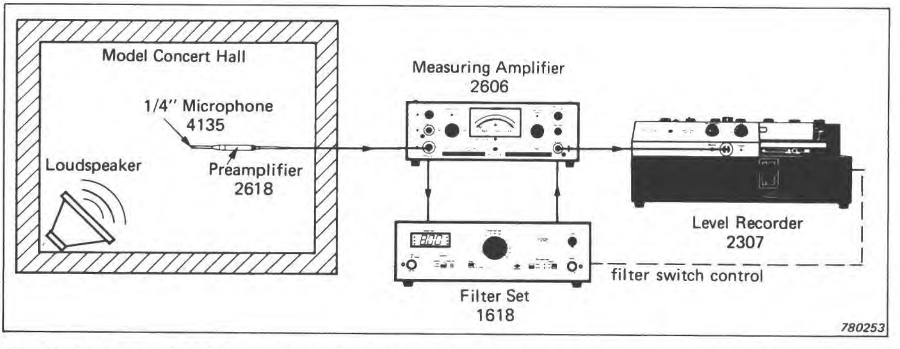

Fig.6.7. Investigating the distribution of sound within a model of a concert hall

Model techniques can also be used to investigate, to a certain extent, such subjective qualities as speech intelligibility, room colouration etc. Speech or music is first recorded in an anechoic room and then played at ten times the normal speed into the one tenth scale model room. The sound within the model room is then recorded and played back after frequency transformation to a listener via headphones. Headphones are generally more linear in their frequency response than loudspeakers therefore the use of headphones is to be preferred.

Existing Room — Measurement Technique

In an existing room, the sound distribution can be measured directly by installing a sound source in the room and employing a sound level meter to measure the sound pressure levels at various positions in the room (Fig.6.8).

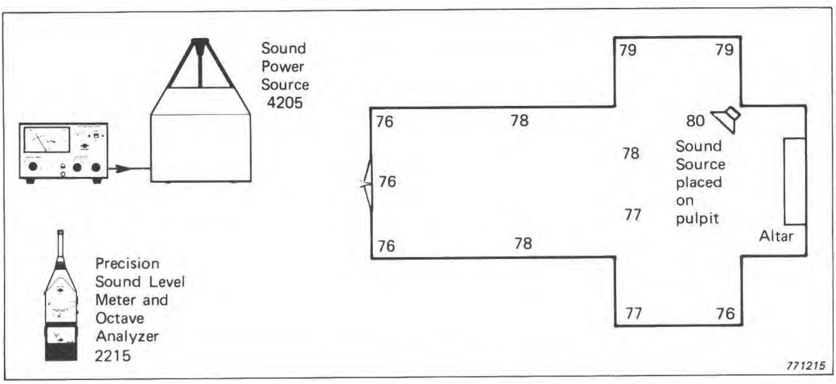

Fig.6.8. Investigating the sound distribution in an existing room: sound pressure levels in dB within a church with a sound source placed on the pulpit

The source should be placed in the most probable positions of the sound sources when the room is in normal use i.e. on the stage of a theatre, on the pulpit of a church, on the speaker's platform in a lecture theatre. From these measurements it can be decided whether further acoustical treatment by means of reflecting surfaces and sound diffusers is necessary to obtain the required uniform distribution. The sound distribution sometimes has to be assisted by means of a loudspeaker system.

An example of the use of sound diffusers in a building already in existance can be seen in the Albert Hall in London. Here, huge "flying saucer" shaped diffusers have been suspended beneath the dome of the concert hall thus preventing sound energy from entering into the dome cavity and returning into the main volume of the hall at a slightly later time and producing an irritating echo.

Loudspeaker systems can be found in most large churches and cathedrals where "columns" of loudspeakers are used to direct the sound of the orator's voice towards the back of the congregation. A column loudspeaker consists of several ordinary loudspeakers placed one above the other. Such a system produces a narrow beam of sound in the vertical plane and a wide beam of sound in the horizontal plane. The main beam of the column is directed at the congregation which absorbs most of the sound energy. Little energy is radiated to the rest of the room so that the direct sound level is maintained at a high level while the reverberant sound level is kept low.

6.4. SOUND ABSORPTION

The sound absorption coefficient, α, of an acoustic material is difficult to measure precisely because α depends on the manner in which the material is installed, whether the tests are carried out on large or small samples, on the angle of incidence of the sound on the sample, upon the characteristics of the room etc. The absorption coefficient claimed by the manufacturer of a particular acoustic material should therefore be looked upon as an average to be used by the acoustic engineer in estimating the amount of sound absorption required.

Several methods have been developed for the determination of sound absorption coefficients each method giving a different value of α for the same test material. As of yet, no reliable method has been devised for relating the results of one method with the results of another. Each method, however, has its use.

Three methods will be described here, namely the reverberation room method, the standing wave method and the tone burst method.

Reverberation Room Method

This method entails measuring the change in reverberation time of a specially constructed reverberation room when samples of sound absorbers are introduced into the room. The equivalent absorption area of separate objects such as furniture, people and functional absorbers may also be measured by this method.

If the reverberation time of the room at a certain frequency (frequency band) before the introduction of the sample was T_1 and after the introduction, T_2, then the equivalent absorption area, A, of the empty reverberation room is increased by ΔA which can be calculated from

$$\Delta A = \frac{55{,}3\,V}{c}\left(\frac{1}{T_2} - \frac{1}{T_1}\right) \tag{6.1}$$

where V = volume of the room
c = velocity of sound under specified measuring conditions (temperature, humidity etc.)

For a plane absorber mounted on the wall, floor or ceiling of the room the absorption coefficient of the specimen may be calculated from

$$\alpha = \frac{\Delta A}{S} \tag{6.2}$$

Due to diffraction which can occur at the edges of the test specimen, the equivalent absorption area determined by this method is not always directly related to the area of the specimen. It is possible that the values for absorption coefficients resulting from this test method are greater than unity.

For the case of a poor absorber (e.g. plaster) it may be necessary to consider that ΔA is actually the difference between the equivalent absorption area of the specimen and that of the portion of the wall or floor covered so that eqn.6.2 has to be corrected to

$$\alpha = \frac{\Delta A}{S} + \alpha_w \tag{6.3}$$

where α_w = absorption coefficient of the wall or floor covered

The ISO has published a Recommendation entitled "ISO/R 354 Measurement of absorption coefficients in a reverberation room" which is intended to promote uniformity in the methods of measuring . In this Recommendation it is stated that a suitable reverberation room should have a minimum volume of 180 m^3 and that the walls should be hard and smooth and no two walls should be parallel, i.e. the room should be of irregular shape (Fig.6.9).

Any of the instrumentation arrangements mentioned under the heading reverberation time may be used to determine α according to this method.

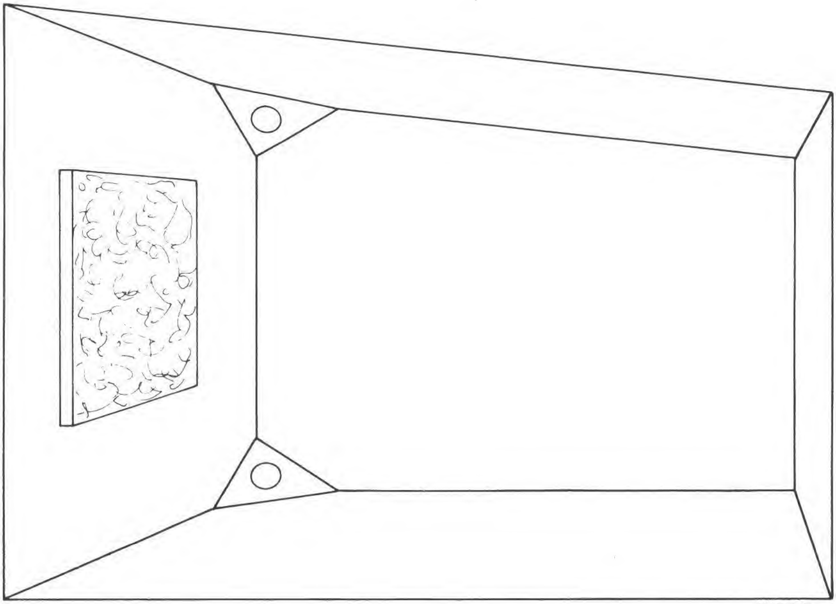

Fig.6.9. A reverberation room suitable for the measurement of absorption coefficients. A sample of the test material of area $10m^2$ is shown mounted on the wall

Standing Wave Method

In this method a loudspeaker is used as a sound source to produce standing acoustic waves in a tube of uniform cross section which is terminated by a sample of the material to be investigated. If the tube were terminated with a nearly perfect reflector, such as a steel plate, then the standing wave pressure pattern established in the tube by the interference of the incident and reflected waves would have the form of that shown to the left in Fig.6.10(a). When the tube is terminated with an absorptive material then the varying phase relationship between the incident and reflected waves causes the acoustic impedance to vary from point to point along the pipe producing a standing wave pressure pattern like that shown in Fig.6.10(b).

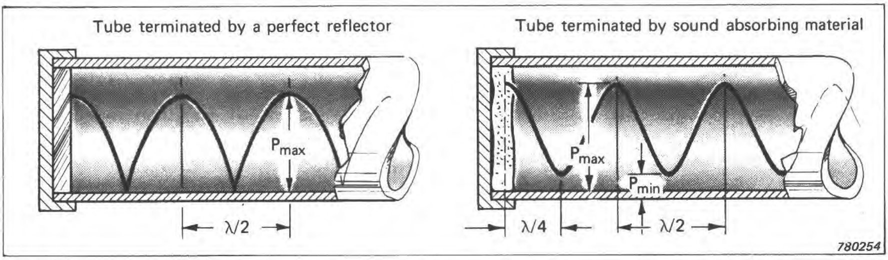

Fig.6.10. Standing wave pressure patterns showing the effect of terminating the wave tube with sound absorbing material

To determine the absorption coefficient of the sample, the relationship between the sound pressure maxima and minima must first be measured. This quantity is known as the standing wave ratio and is defined by

$$SWR = \frac{p_{max}}{p_{min}} = \frac{A+B}{A-B} \tag{6.4}$$

$$\therefore \alpha = 1 - \left(\frac{SWR-1}{SWR+1}\right)^2 \tag{6.5}$$

where SWR = standing wave ratio
p_{max} = maximum sound pressure
p_{min} = minimum sound pressure
A = amplitude of the incident wave
B = amplitude of the reflected wave

As the absorption coefficient of the sample is defined as the ratio between the energy absorbed by the sample to the total energy incident on the sample and as the energy is proportional to the square of the sound pressure then

$$\alpha = 1 - \left(\frac{B}{A}\right)^2 \tag{6.6}$$

It is therefore particularly easy to determine the absorption coefficient for normal incidence using this method. The Standing Wave Apparatus Type 4002 is specially designed for this type of measurement. The 4002 consists of a long tube with a loudspeaker mounted at one end. At the other end is mounted either a sound reflecting metal plate or a circular sample of the material to be tested. The loudspeaker emits a sound wave down the tube and this wave is reflected from either the metal plate termination or the test sample. The pressure maxima and minima within the wave tube can be detected by means of a microphone probe tube which is led through an axial

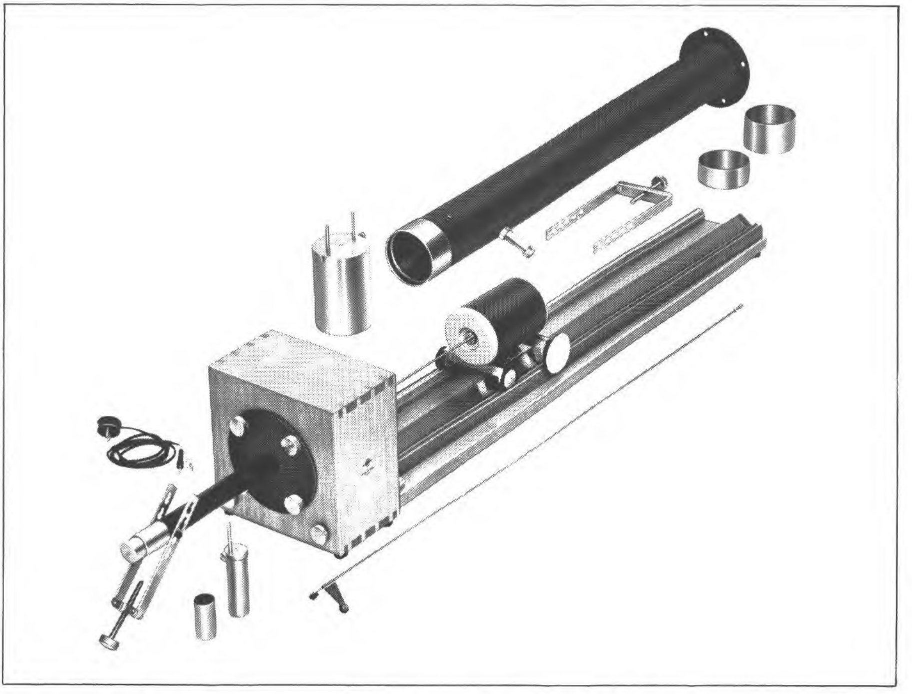

Fig.6.11. The Standing Wave Apparatus Type 4002 and accessories

hole in the loudspeaker. The probe tube is supported within the wave tube by a small gliding carriage. The other end of the probe tube is connected to a microphone car which can be moved back and forth over a graduated track. As the measuring method requires plane waves then the diameter of the sample must not be greater than about half the wavelength at the frequency of interest. To enable measurements to be performed over a relatively wide frequency range, the 4002 is supplied with two wave tubes of different diameters. The larger one (of 100 mm diameter) is intended for the frequency range of 90 Hz to 1800 Hz and the smaller one (of 30 mm diameter) for the range 800 Hz to 6500 Hz. Each tube is supplied with three sample holders of different design; two of fixed depth and one of adjustable depth.

To determine α of a particular material, the 4002 has to be used in conjunction with a generator, amplifier and a filter or with the Heterodyne Analyzer Type 2010 which contains the generator, filter and amplifier in the one instrument. A circular disc of the absorbing material is attached to the end of the wave tube by means of one of the sample holders supplied. Plane waves are produced in the tube by the loudspeaker which is fed by the generator. The positions of the maxima and minima produced within the tube are determined by moving the probe microphone until a maximum (or minimum) deflec-

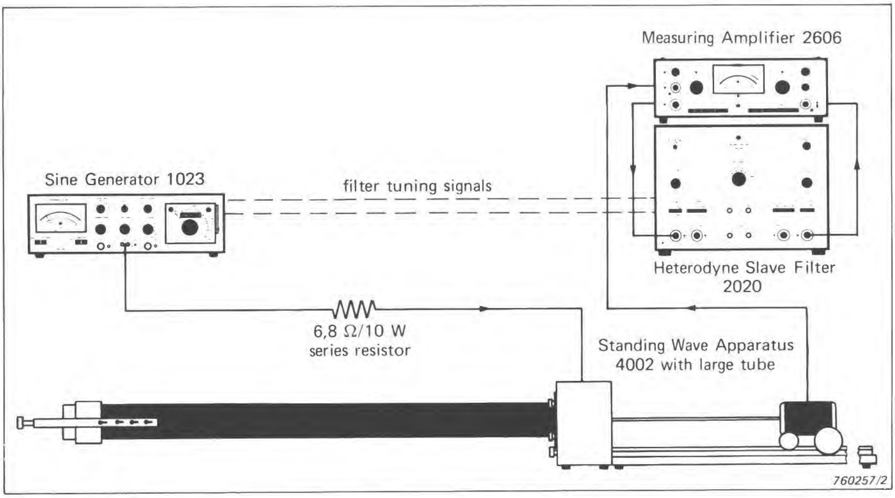

Fig.6.12. Measurement of absorption coefficient employing the Standing Wave Apparatus Type 4002

tion is obtained on the measuring amplifier. The position of the car can then be read from the graduated track. If the special scale Type SA 0054 be fitted to the measuring amplifier then the absorption coefficient may be read directly. To enable the pressure maxima and minima to be determined more accurately, a Measuring Amplifier Type 2606 can be made to selectively amplify the output of the microphone. This is done by connecting the 2606 to a slave filter which is automatically tuned from the 1023 or by using a Heterodyne Analyzer Type 2010 as mentioned previously. This arrangement minimises the effects of noise and harmonic distortion. Such precautions are especially necessary when the material under test has a small coefficient of absorption because in this case the reflected wave is almost of the same magnitude as the incident wave; hence the sound pressures at the minima become very small.

Using the Standing Wave Apparatus Type 4002, it is a straightforward matter to determine the acoustic impedance of an acoustic material. Such information is necessary when the boundary conditions of a room's limiting surfaces have to be inserted into the wave equation to investigate such acoustic problems as flutter echoes and similar phenomena. An acoustic material whose acoustic impedance is real (i.e. where there is no phase difference between the pressure and the particle velocity at the surface of the material) will have a pressure maximum precisely at the surface of the material and a pressure minimum exactly a quarter wavelength away. Hard reflective surfaces have acoustic impedances approaching the value of zero. If the sample possesses a complex acoustic impedance then the reflected wave is out of phase with the incident wave. The phase angle and thus the real and imaginary parts of the complex impedance can be determined by measuring the standing wave ratio and the distance, d, by which the first pressure minimum or node is displaced with respect to the position the node would have occupied if the wave tube had been terminated with a perfectly reflecting surface. The complex impedance can then be obtained by calculation from these two quantities or, a far neater method, these two quantities can be entered on a Smith Chart and the impedance read directly. (N.B. the Smith Chart is a nomographic device widely used in the field of electronics to determine the impedance of transmission lines.)

Tone Burst Method

This method enables the absorption coefficient to be determined for various angles of incidence of the sound energy. An advantage of the method is that no special reverberation room is required for the tests.

The principle of the method is straightforward (Ref.37). A tone burst is produced by gating a sinosoidal signal. The tone burst is emitted from a loudspeaker into the test room and the pressure signal is received by a microphone at a known distance, x, from the loudspeaker (Fig.6.13). The loudspeaker is then aimed at the test specimen at the desired angle of incidence such that the distance between the loudspeaker and the point of reflection is x/2. Almost any type of loudspeaker may be employed as a highly directive loudspeaker is not required for this method. The microphone is positioned to receive the reflected sound, the total path length from loudspeaker, to specimen, to microphone being x.

By comparing the sound pressures (determined from the sound pressure levels measured by the microphone) for the direct and the reflected sound, the reflection coefficient $r_{\theta f}$, of the specimen at that particular frequency and at that particular angle of incidence can be calculated. The absorption coefficient $\alpha_{\theta f}$, can then be determined from

$$\alpha_{\theta f} = 1 - r_{\theta f} \tag{6.7}$$

The reflection coefficient is the ratio between the intensity of the reflected sound to the intensity of the direct sound i.e. the ratio of the square of the reflected sound pressure to the square of the direct sound pressure.

Using the automatic sine sweep facility of the sine generator, the sound pressure level due to the reflected sound from a wall relative to that due to the direct sound from the loudspeaker can be measured as a function of frequency. The absorption coefficient as determined from these results is shown in Fig.6.14.

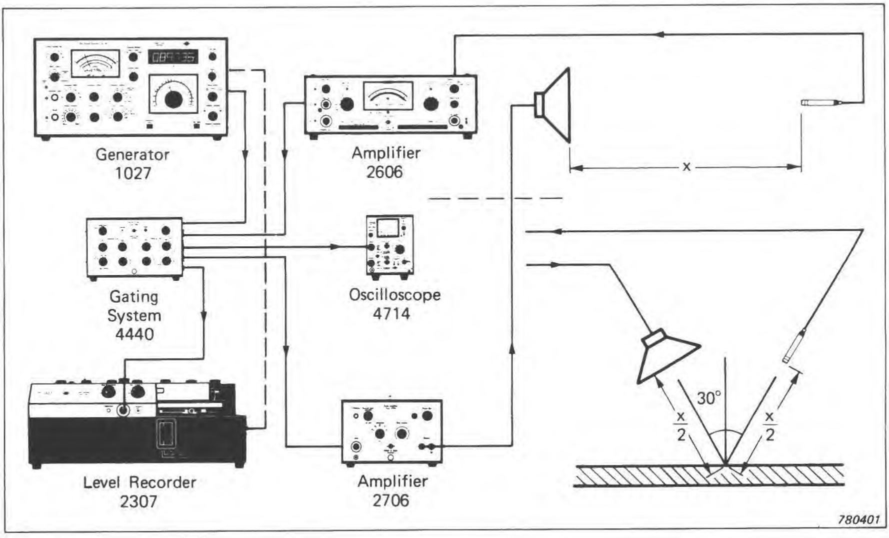

Fig. 6.13. Tone burst method of measuring absorption coefficients

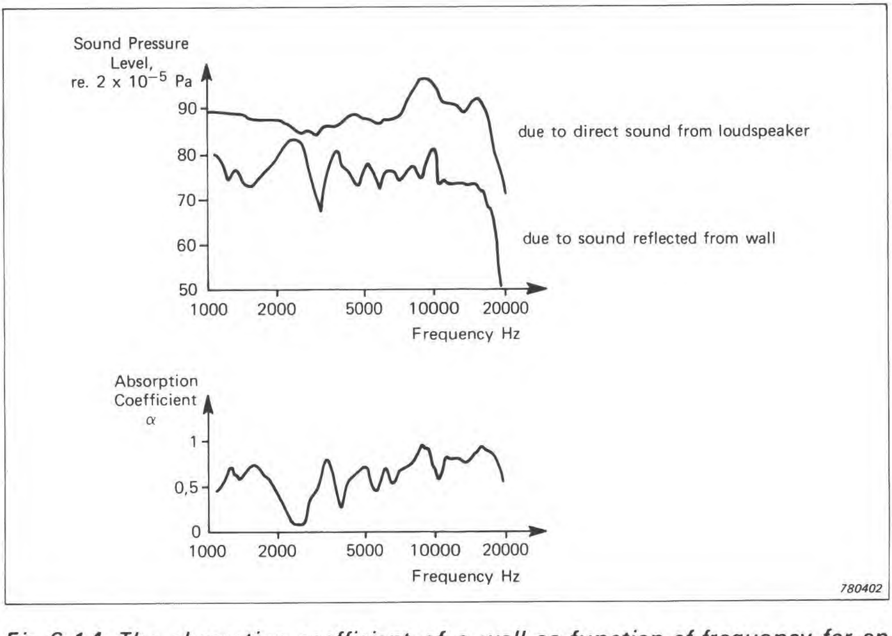

Fig. 6.14. The absorption coefficient of a wall as function of frequency for an angle of incidence of 30° as determined from the differences between the direct sound and the reflected sound

6.5. SOUND INSULATION

Sound measurements can be conveniently divided into airborne and impact sound insulation. These two headings can be subdivided into field and laboratory measurements. The practical acoustician will be mostly involved in field measurements of sound insulation as the laboratory measurements require special facilities which, for economic reasons, are usually the perogative of the large institutions such as government owned research centres. The basic instrumentation in sound insulation measurements requires a sound source (loudspeaker or standard impact machine) placed in the emitting or source room and a microphone in conjunction with filters and an amplifier to measure the sound pressure levels in the receiving and source rooms.

In each frequency band of interest the sound pressure levels produced by the sound source should be at least 5 dB above the background noise level in that band. The average sound pressure level in the source and receiving rooms should be determined either by using a number of fixed microphone positions or a single microphone whose output is integrated while the microphone is swept over a particular path to obtain the spatial average.

Airborne Sound Insulation

Field Measurements

The three battery operated instruments shown in Fig.6.15 form a compact set suitable for octave band insulation measurements on walls, windows, doors etc. and also for reverberation measurements (see section Reverberation Time). For measurements in third octave bands either the Precision Sound Level Meter Type 2203 or 2209 should be employed in conjunction with the Third Octave Filter Set Type 1616. Knowing the sound pressure levels produced in each room, the equivalent absorption area of the receiving room (determined from reverberation measurements) and the area of the dividing wall then the sound reduction index of the dividing wall can be calculated according to Eqn.3.2. In field measurements, however, it is usually the apparent sound reduction index, as calculated from Eqn.3.4, that is of interest. A precision Integrating Sound Level Meter Type 2218 can be used together with the Rotating Microphone Boom Type 3923 to determine the average sound pressure level over a circular path. Fig.6.16 shows such an arrangement used for determining the sound insulation of an external window. In blocks of flats or similar buildings, it may be worthwhile to record on a tape recorder the sound pressure levels in several rooms simultaneously (Fig.6.17). The recordings can then be analysed later in the laboratory to yield the sound reduction index of the walls between the source room and the receiving rooms using, for example the instrumentation shown in Fig.6.18.

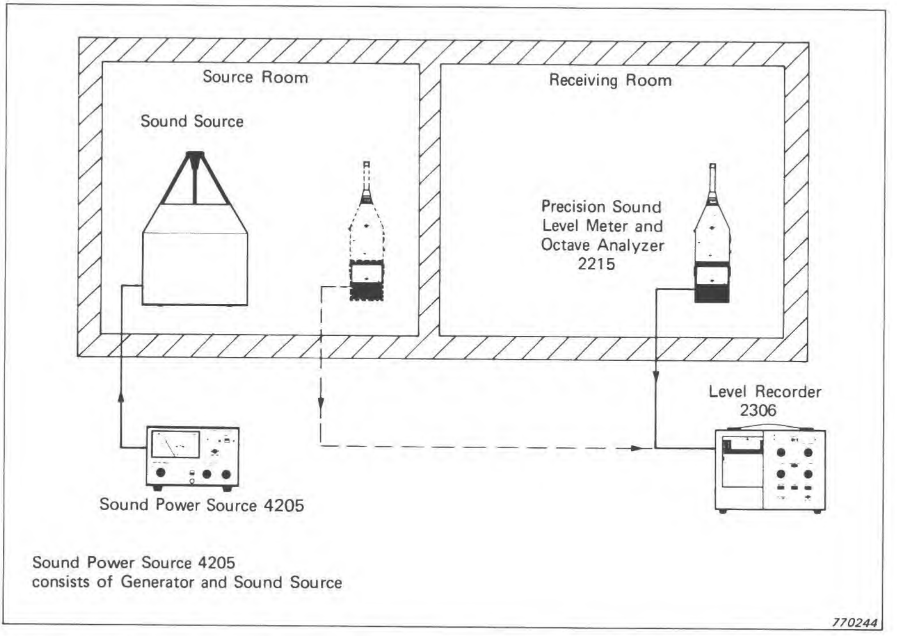

Fig.6.15. Field measurements employing the Sound Power Source Type 4205

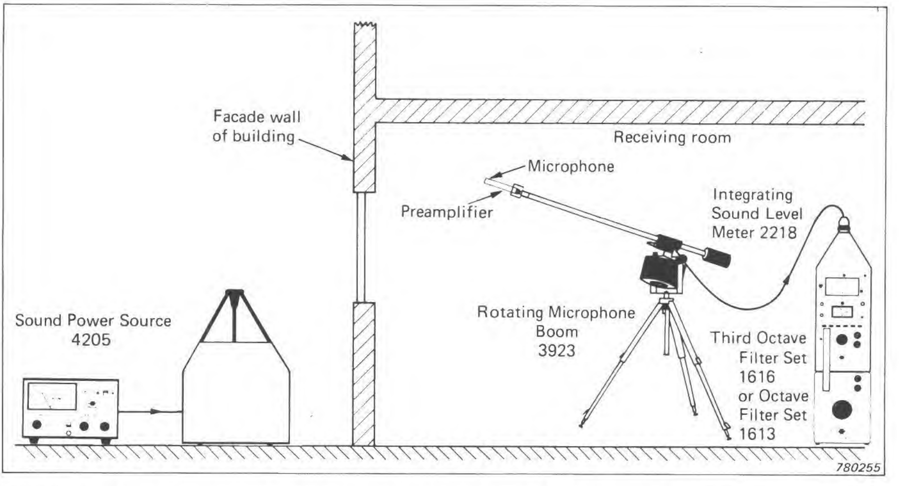

Fig.6.16. Field measurements employing a rotating microphone boom and an integrating sound level meter

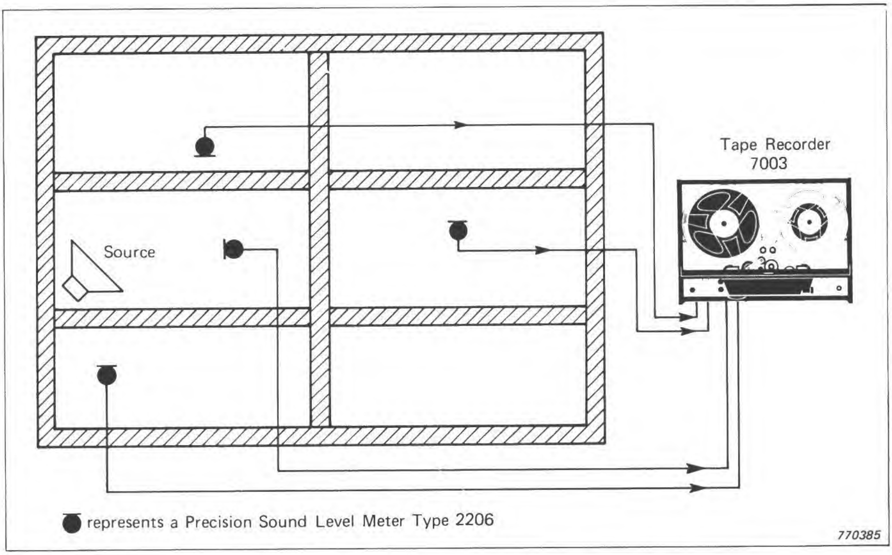

Fig.6.17. Recording the sound pressure levels in several rooms simultaneously

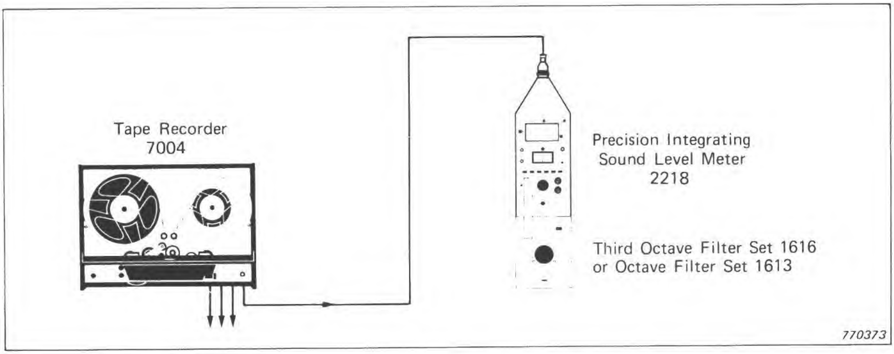

Fig.6.18. Analysis of tape recordings in the laboratory

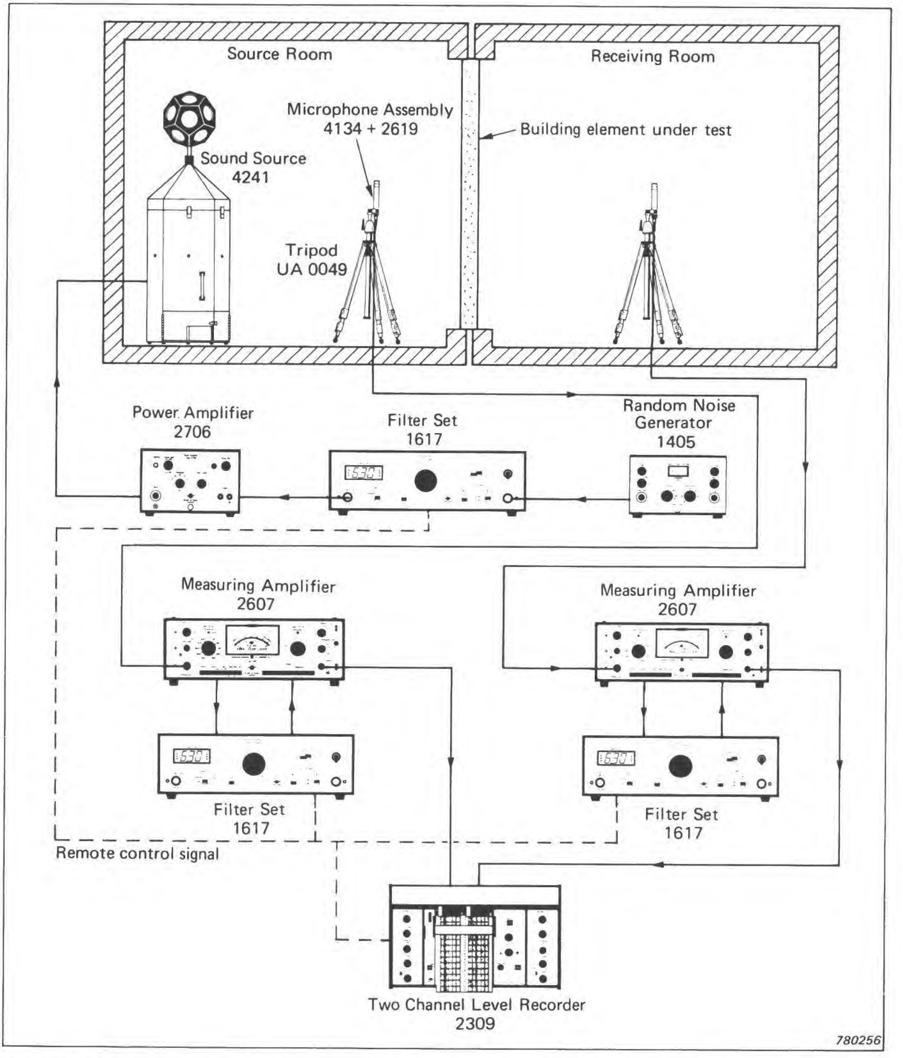

Fig.6.19. Sound insulation measurements in a laboratory sound transmission suite

Laboratory measurements of sound insulation are usually performed in a specially constructed transmission suite where as far as is practically possible the only way for airborne sound (or impact sound when dealing with impact noise) to be transmitted from the source to the receiving room is through the wall or partition (or floor) under test. The most common method of isolating the two test rooms from each other is to build two completely independent structures as shown in Fig.6.19. As the flanking transmission which occurs in this structure is very small relative to the direct transmission, the sound insulation afforded by a wall or partition as determined in this way in the laboratory is always somewhat greater than that experienced in practice.

The arrangement shown in Fig.6.19 can be used to measure the sound insulation in third octave bands employing several microphone positions. Filter sets are used to limit the signals to both the sound source and the measuring amplifier to third octave bands. This means that:

1) a higher sound pressure level in the third octave band of interest can be obtained from a given loudspeaker than if a broad band noise signal were used.

2) the effect of background noise on the measured sound pressure level is reduced.

The filter sets are switched in unison from one filter band to the next by an electrical signal from the level recorder. The curves in Fig.6.20 show typical sound level differences between the two rooms of a transmission suite.

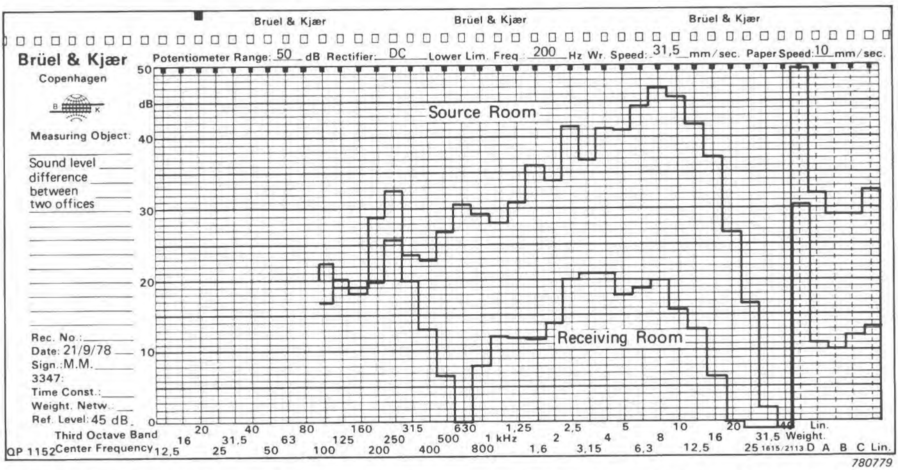

Fig.6.20. Typical sound level differences between two rooms. Note that the D, A, B, C & Lin. levels have been attenuated by 20 dB

A rotating boom can be employed to sweep the microphone around a circular path to obtain the spatial averaged sound pressure level. The sound pressure detected by the microphone can be integrated over a time interval equal to the duration of one complete microphone revolution by means of either the Noise Level Analyser Type 4426 or the Precision Integrating Sound Level Meter Type 2218. When using the 4426 for this application, a separate microphone power supply is required.

The measurement of the sound reduction index of a building element can be automated to a large extent by employing a digital frequency analyser in conjunction with a desk top calculator (Fig.6.21).

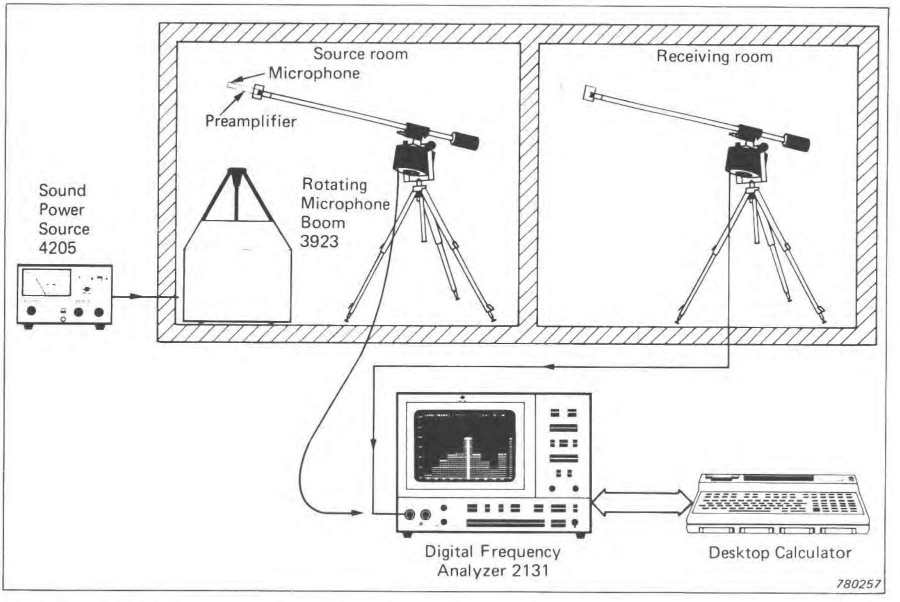

Fig.6.21. Laboratory measurements employing a digital frequency analyser and a desk-top calculator

The signal from the rotating microphone is integrated and analysed into third octave bands by the Digital Frequency Analyser Type 2131. The spectra of the sound fields in both rooms of the transmission suite can be simultaneously displayed in the form of third octave band sound pressure levels on the screen of the 2131. By using a programmable desk top calculator, the required sound reduction index (or apparent reduction index) can be evaluated in a trice from the data stored in the 2131.

The necessary programme is available from Brüel & Kjær in the Acoustic Programme Packages BZ 0011 or BZ 0012.

Impact Sound Insulation

Field measurements

For both field and laboratory measurements of impact sound insulation, a standardized tapping machine is usually employed (i.e. one which complies with ISO 140). The measuring technique for impact sound insulation is similar to that for airborne sound insulation. The Tapping Machine Type 3204 is used to excite the partition (usually the floor) and the average sound pressure levels are measured in octave or in third octave bands in the receiving room. The spatial average of the sound pressure levels can be determined by either using a number of microphone positions (see Fig.6.22) or by using a Precision Integrating Sound Level Meter Type 2218 and integrating the output of a microphone which is continuously swept over a prescribed path (Fig.6.23).

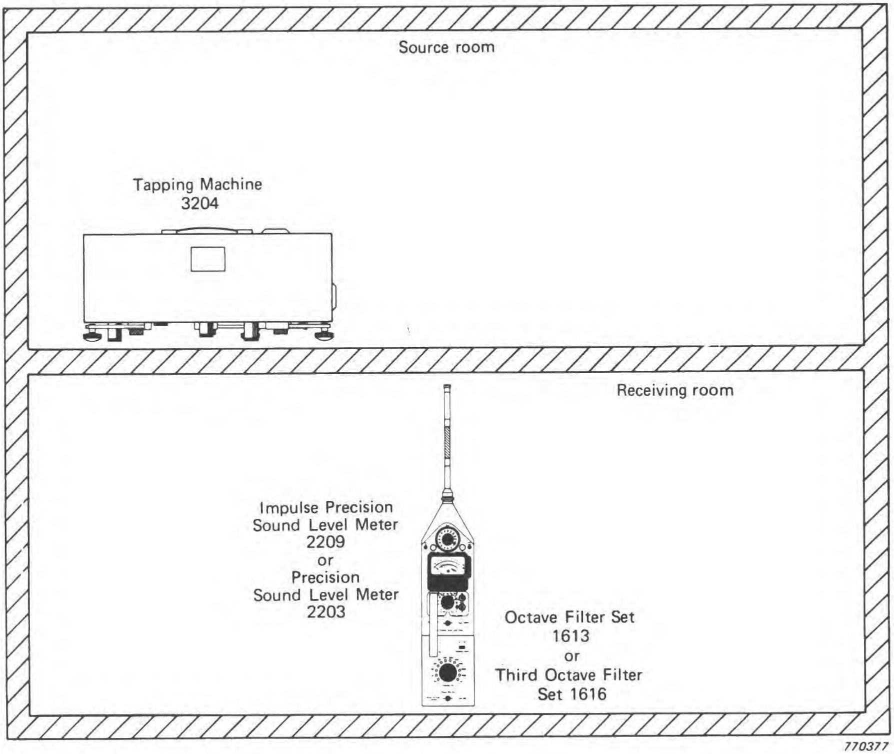

Fig.6.22. Field measurements of impact sound insulation using a number of microphone positions

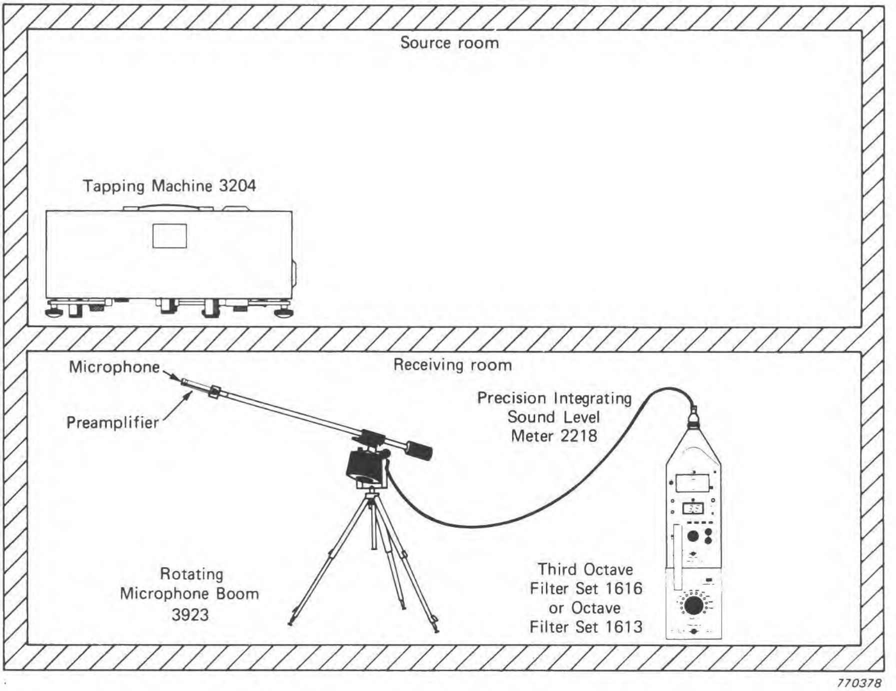

Fig.6.23. Field measurements of impact sound insulation using an integrating sound level meter and a rotating microphone boom

Laboratory Measurements

Two arrangements suitable for the laboratory are shown in Fig.6.24 and 6.25. In Fig.6.24 the level recorder automatically switches the filters from one filter band to the next. A typical spectrogram recorded by this method is illustrated in the diagram.

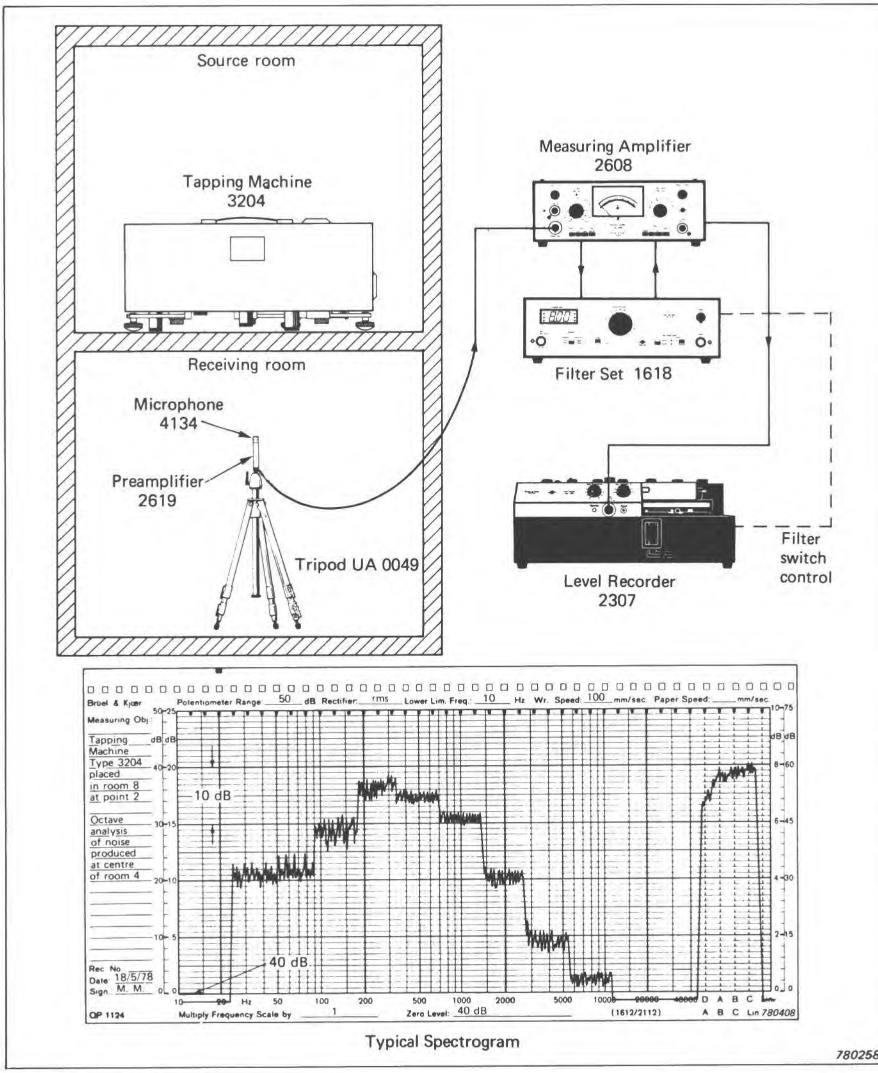

Fig.6.24. Laboratory measurement of impact sound insulation using several microphone positions with a typical octave band spectrogram

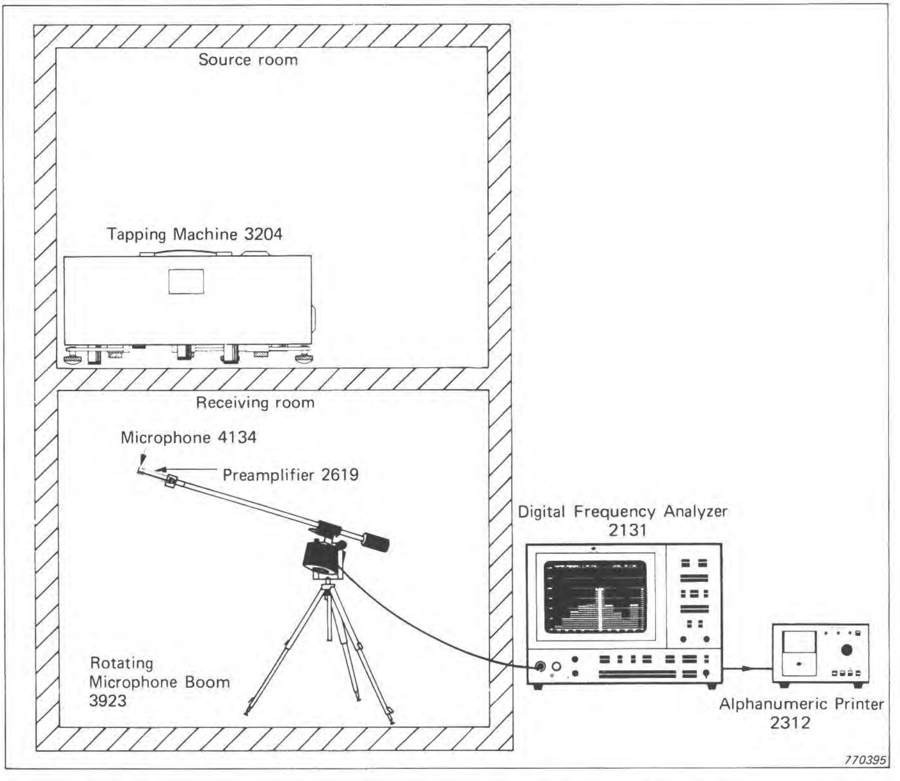

Fig.6.25. Laboratory measuremnet of impact sound insulation employing a rotating microphone and a digital frequency analyser

6.5. SOUND POWER

It is becoming more and more common to rate the sound output of machines and domestic appliances in terms of sound power. The measurement of the sound power output is being introduced more and more as a part of the quality control testing of the machines. It is not, however, a simple matter to measure sound power. This statement is confirmed by a study of the 6 new ISO publications on the subject. Special test facilities are required such as a reverberant room, a semi-anechoic room or an anechoic room and many sound pressure measurements have to be performed before the sound power can be calculated. The rating values are valid for factory tested new machines but after installation or prolonged use the sound output of the machines may well change. It can be important for the acoustician to know whether the machines installed in a particular building fulfill their rated values.

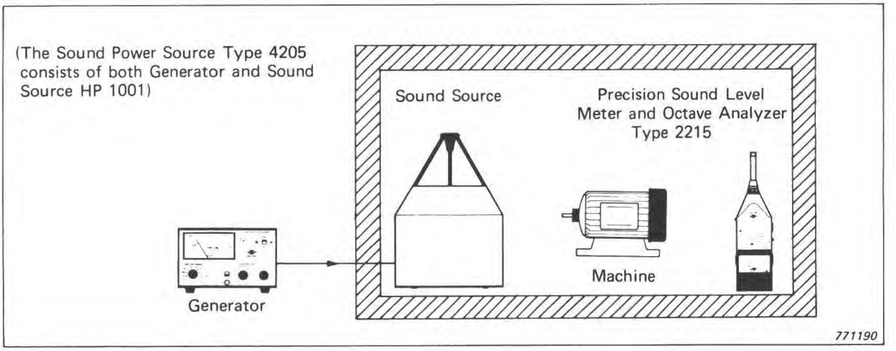

Fig.6.26. Sound power measurements employing the Sound Power Source Type 4205

The rating of a machine in situ can, in many cases, be determined by employing the Sound Power Source Type 4205. This instrument gives a rapid and accurate value of the sound power level which can be read directly from a meter.

Further information about the various measurement methods both in the laboratory and in situ can be found in Ref.28 and 35.

6.6. ROOM MODES

Room modes and their effects e.g. the "colouration" they impose on the sound output of loudspeakers, can be investigated using the instrumentation shown in Fig.6.27. More precisely, the instrumentation measures the irregularity of the transmission of pure tones from a source located at one point in a room to a microphone located at another point. If a loudspeaker is placed in one corner of the room and the microphone is placed in the diagonally opposite corner and a relatively slow sweep rate for the pure tone is used then practically all the room modes will be excited. Fig.6.28 shows a typical example of the transmission of sound for such loudspeaker and microphone positions for a room of 2,5 × 3 × 7 m when the pure tone was slowly increased from 20 Hz to 100 Hz. The normal modes of vibration corresponding to the various resonant frequencies are specified by the numbers in parenthesis above the resonant peaks. Repeating the measurement for the same loudspeaker and microphone in an anechoic room would give a response similar to the dashed curve in Fig.6.28. The difference between these two curves is the colouration that the room places on the free field loudspeaker response.

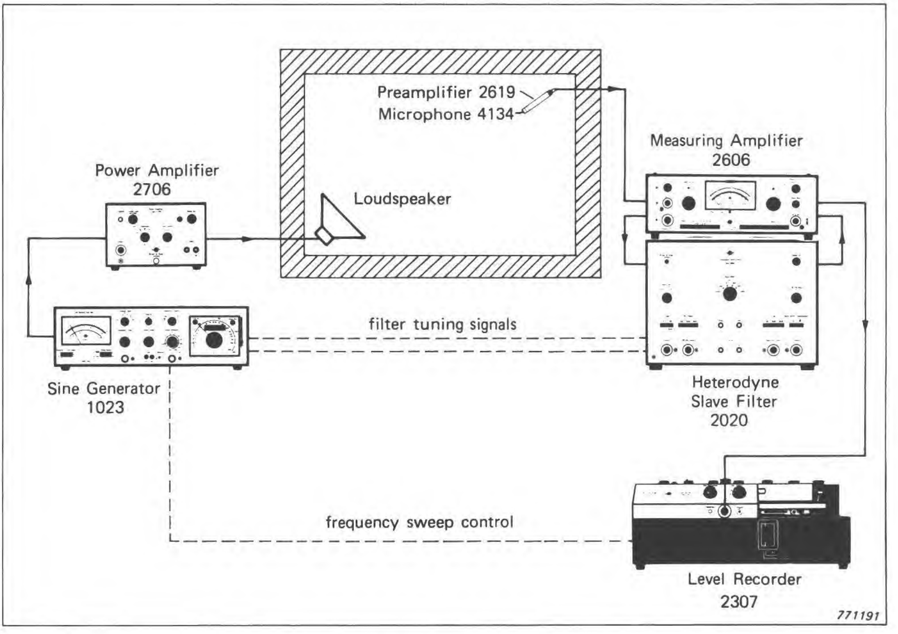

Fig.6.27. Investigation of modes within a room

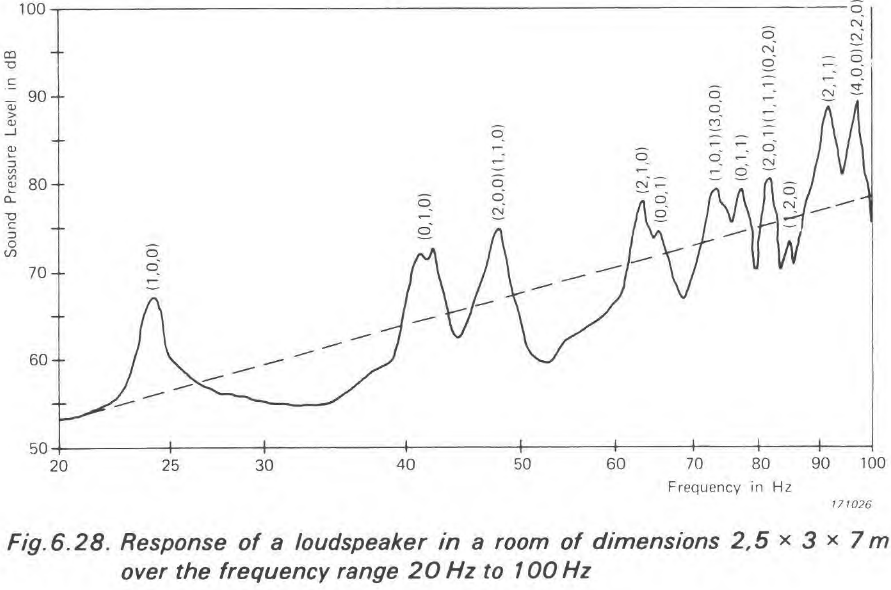

Fig.6.28. Response of a loudspeaker in a room of dimensions 2,5 × 3 × 7 m over the frequency range 20 Hz to 100 Hz

Particular peculiarities of the room e.g. prominent resonances and marked unresponsiveness over a particular frequency range, can easily be seen from a record of the microphone's output.

6.7. DIRECTIVITY OF NOISE SOURCES

A knowledge of the directivity of noises sources is of importance, for example, in the planning of workshops and factory halls, where the correct orientation of the machines and the correct positioning of sound absorbing materials can lead to substantial reduction in the general noise levels.

The directivity of a noise source which is situated in a building can be determined by measuring the sound pressure levels produced at a given distance from the source using a sound level meter. Filters may be used to obtain the directivity for various octave or third octave bands. This method yields reliable results providing the principal contributor to the sound pressure level is the direct and not the reverberant field. If the source can be moved then the directivity can be determined in an anechoic room. Small sources can be mounted on the Turntable Type 3922 and the receiving microphone can be placed in a fixed position while the source is rotated. The results of the measurement can be plotted directly onto polar graph paper by employing the Level Recorder Type 2307.

6.8. VENTILATION AND SERVICE SYSTEM NOISE

The noise produced by ventilation and service systems (gas and oil fired burners, waste disposal chutes, water cisterns etc.) in a building can be measured using a sound level meter. For measurements in octave bands the Precision Sound Level Meter and Octave Analyser Type 2215 can be used. For third octave analysis, the Filter Set Type 1616 can be used with either the Sound Level Meter Type 2209 or Type 2203. For measuring sound pressure levels in regions of high turbulence, for example within airducts, a 1/2 inch microphone can be used fitted with the Turbulence Screen Type UA 0436.

For laboratory methods of measuring the noise emitted by valves, pipes, water heating appliances etc. reference should be made to the ISO publication 3822/1 entitled, "Acoustics - Laboratory tests on noise emission by appliances and equipment used in water supply installations". These measurements may be useful to predict the noise due to plumbing in situ.

6.9. VIBRATION MEASUREMENT

Locating and monitoring vibration

For locating and monitoring vibration, the Vibration Meter Type 2511 is recommended. The transducer can either be attached with wax or a magnet to the structure under test or the transducer can be held in the hand as a probe.

This compact, battery operated instrument can be combined with the Tunable Band Pass Filter Type 1621 in a carrying case to produce a self contained vibration analysis system known as the Portable Vibration Analyser Type 3513. Adding the Portable Level Recorded Type 2306 to the system greatly increases the possibilities e.g. enables automatic recordings of frequency analyses to be made.

Fig.6.29. Measurement of sound and vibration due to running water

An example of vibration monitoring is shown in Fig.6.30. Here a machine has been mounted on anti-vibration mountings. The improvement in vibration isolation can be measured by monitoring the vibration levels on the base of the machine and on the structure of the building. The sound pressure levels produced by the machine are monitored simultenously by a sound level meter. The data can be recorded on a tape recorder and then analysed later in the laboratory.

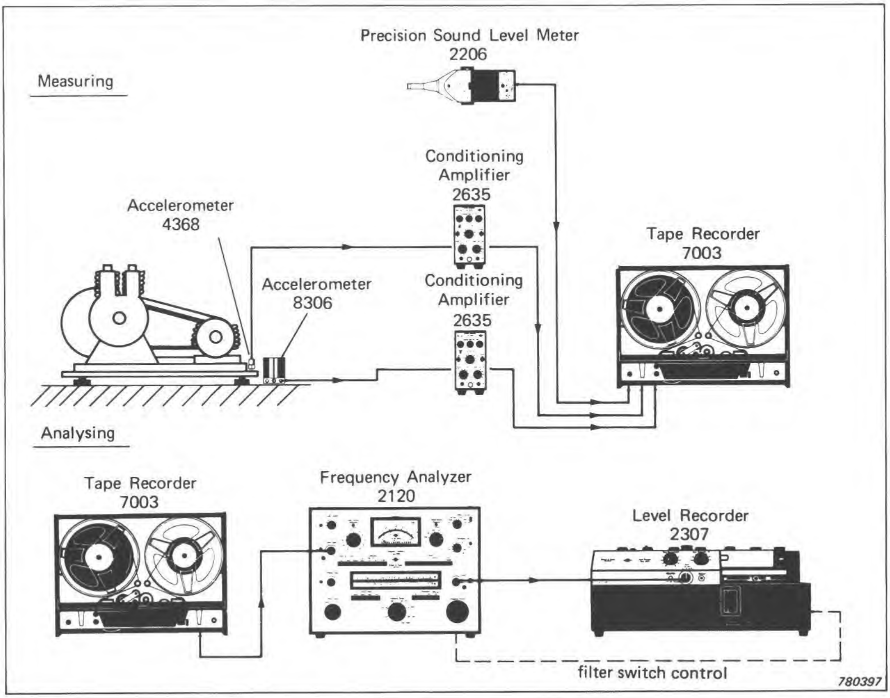

Fig.6.30. Recording and analysing the vibration isolation provided by mounting a machine on resilient supports

Complete sets are available in robust carrying cases which contain all the necessary equipment for making both sound and vibration measurements. Fig.6.31 shows a typical set containing a sound level meter, a filter set, a microphone, accelerometers and all the accessories.

Fig.6.31. The Sound and Vibration Set Type 3507

Two mains operated systems for the detailed narrow band analyses of vibration signals are shown in Figs.6.32 and 6.33. The vibration levels of a surface can be expressed as a velocity, an acceleration or a displacement. When measuring the level of vibration of a surface within a room, it is sometimes useful to determine the average surface velocity levels of the surface and to express the results in decibels using the equation

$$L_v = 10 \log_{10} \left(\frac{v_1{}^2 + v_2{}^2 + \cdots + v_n{}^2}{n v_0{}^2} \right) \text{ dB} \qquad (6.8)$$

where L_v = average surface velocity level
$v_1, v_2, \ldots v$ are the rms normal surface velocities at n different positions on the wall or ceiling
v_o = 10^{-9} m.s.$^{-1}$ the reference velocity

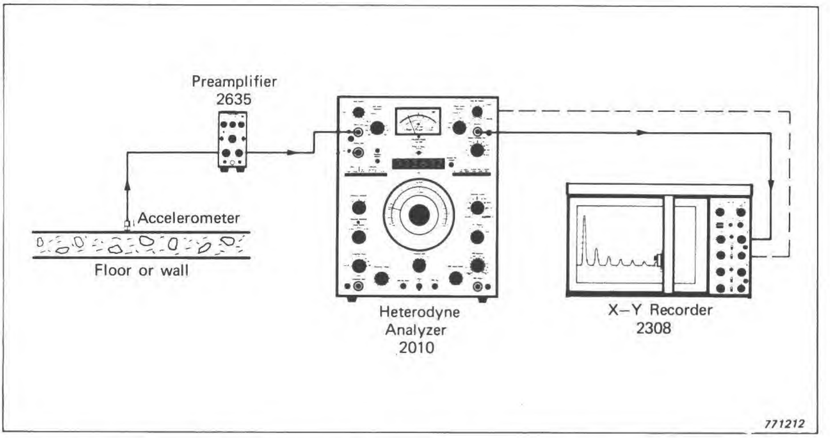

Fig. 6.32. Narrow band vibration analysis employing heterodyne analyser

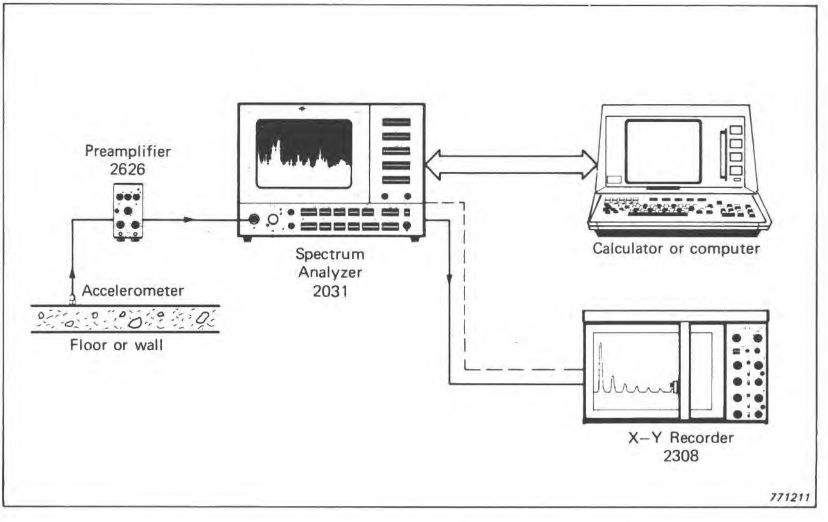

Fig. 6.33. Narrow band analysis employing a spectrum analyser

Flanking Transmission

The instrumentation described in the foregoing section may be used to measure the r.m.s. normal surface velocities of a flanking surface in the receiving room at several different positions (for a medium sized room about six accelerometer positions on each surface are sufficient). The amount of energy radiated from this surface into the room can be estimated from equation 3.8, i.e.

$$W_k = \rho c S_k \overline{V}_k^2 \sigma_k \qquad (6.9)$$

The quantity W_k can then be compared to the amount of energy which is radiated into the receiving room from the dividing room via the common wall thus giving a measure of the flanking transmission via that particular flanking surface (Fig.6.34).

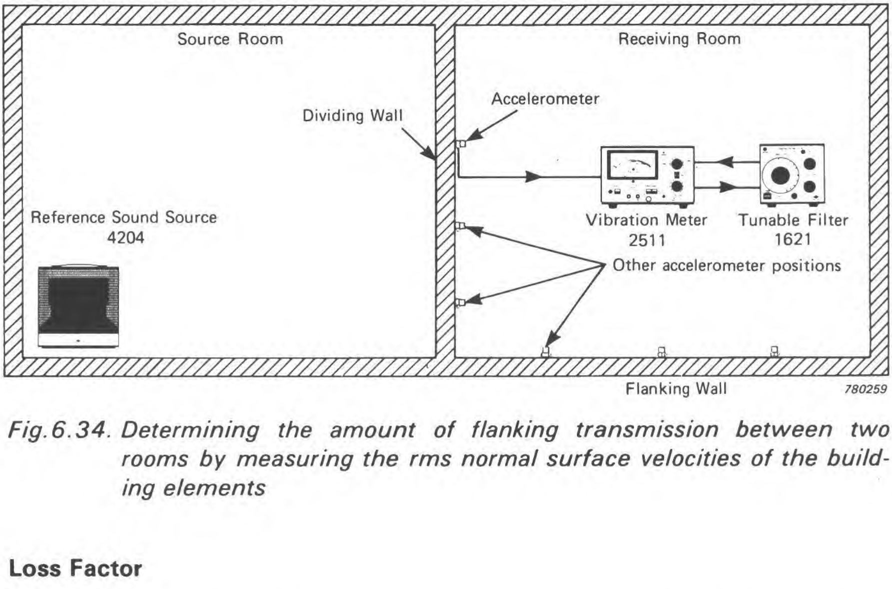

Fig.6.34. Determining the amount of flanking transmission between two rooms by measuring the rms normal surface velocities of the building elements

Loss Factor

The loss factors of the various building elements may be measured by determining the reverberation time of the wall. A vibration exciter forces the wall into vibration; the exciter is abruptly stopped and the decay of the level of acceleration or velocity of the wall is monitored by an accelerometer and the decay in level is drawn by a level recorder (Fig.6.35). From the decay curve a reverberation time can be determined and hence the loss factor by using

$$\eta = \frac{2,2}{fT} \qquad (6.10)$$

where η = loss factor
f = octave band or third octave band centre frequency
T = reverberation time in the octave or third octave band

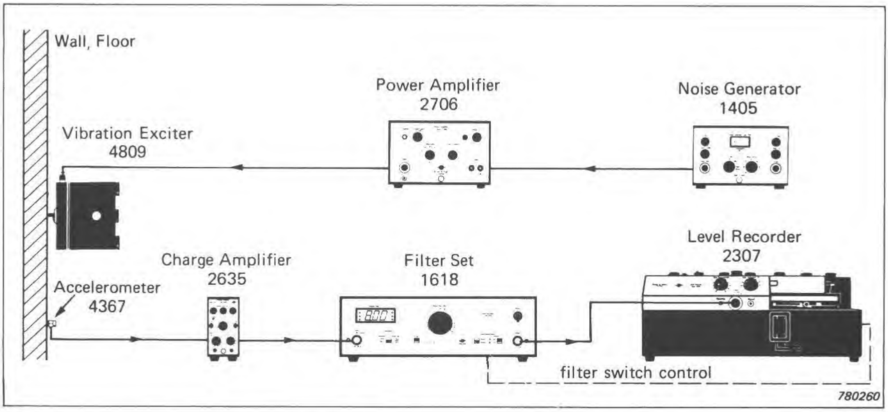

Fig.6.35. Determining the loss factor of a building element

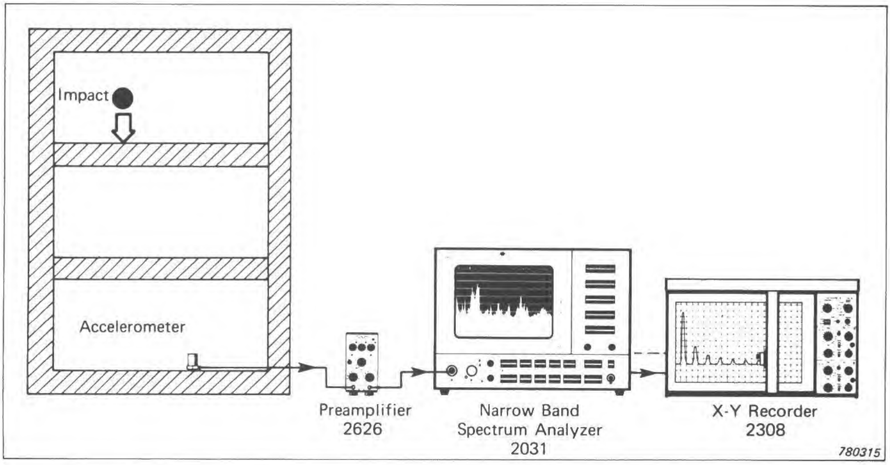

Fig.6.36. Capture and analysis of a shock signal using a Narrow Band Spectrum Analyzer Type 2031 and an X—Y Recorder Type 2308

Transmission of Shock

When investigating the transmission of shocks through a building a simple but very effective way of imparting a shock to the building structure is to let a sack of sand fall from a height onto the floor. The shock produced can be displayed, analysed and recorded using the instrumentation shown in Fig.6.36.

Mechanical Mobility

When a source of structure borne noise has been found in a building and when further damping of the source would be of no avail, there remains the possibility of reducing the transmission sound by damping or closing the transmission paths. The efficacity of a particular transmission path as a function of frequency can be determined by measuring the mechanical mobility of the structure (Ref.23 and 33). The mechanical mobility is defined as the ratio of the vibrating force applied to a certain point on a structure to the resulting velocity of vibration. A maximum in the mobility response curve corresponds to a natural mode of vibration of the structure and therefore to a low transmission loss.

A typical instrumentation arrangement for measuring mechanical mobility is shown in Fig.6.37.

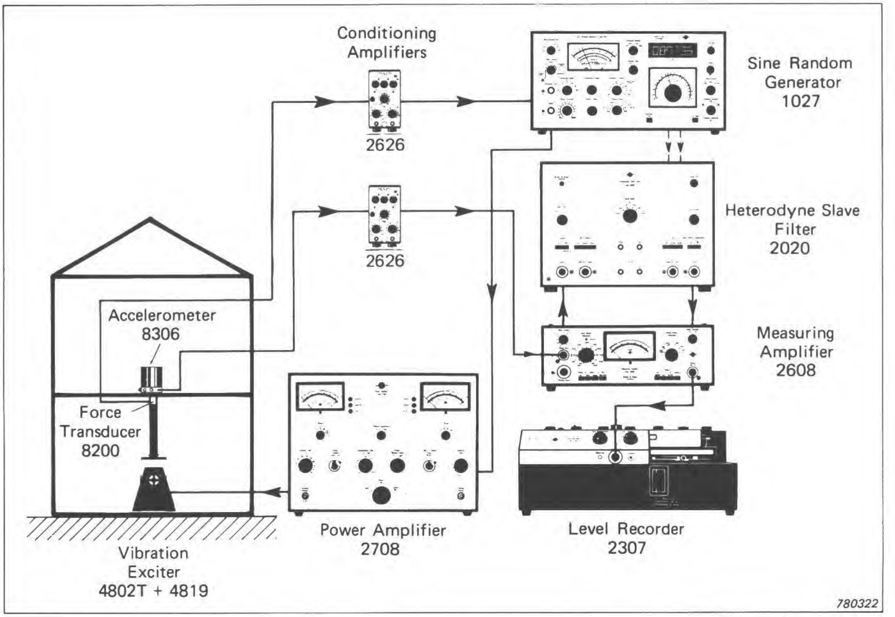

Fig.6.37. Measurement of mechanical mobility

APPENDIX

DERIVATION OF THE NORMAL MODE EQUATION FOR A RECTANGULAR ROOM

Consider a rectangular room with hard, smooth, parallel walls. The room dimensions are l_x, l_y and l_z (Fig.A.1)

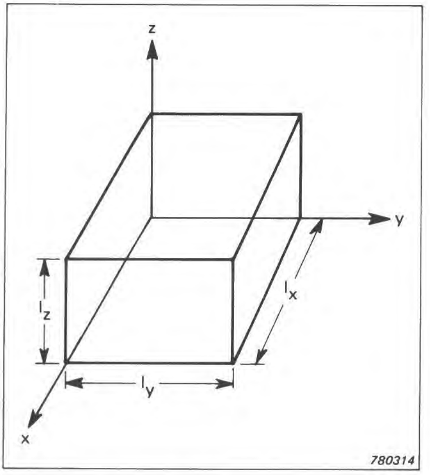

Fig.A.1 Rectangular room of dimensions l_x, l_y and l_z

The general expression for a plane sinusoidal wave may be written in the form

$$p = Ae^{j(\omega t - k_x x - k_y y - k_z z)} \tag{A.1}$$

For this equation to satisfy the general wave equation

$$\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}\right) p = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} \tag{A.2}$$

then the wave numbers k_x, k_y and k_z must satisfy

$$k = \frac{\omega}{c} = \sqrt{k_x^2 + k_y^2 + k_z^2} \qquad (A.3)$$

which can be verified by substituting eqn. A.1 in eqn. A.2. The quantities k_x/k, k_y/k and k_z/k represent the direction cosines made by the propagation of the waves with respect to the x, y and z axes. One can replace one, two or all three negative signs in eqn. A.1 to obtain seven additional equations similar to eqn.A.1 all having identical values of k_x, k_y and k_z. This array of eight wave equations represents the family of waves generated by the original wave as it undergoes reflections at the boundaries of the room.

The particle velocities in the three principal directions u, v, and w, must be determined in terms of the pressure, p, before the boundary conditions can be applied to the problem. For a sinusoidal acoustic wave

$$-\frac{\partial p}{\partial x} = \rho \frac{\partial u}{\partial t} = j\omega\rho u \qquad (A.4)$$

therefore
$$u = \frac{-1}{j\omega\rho} \cdot \frac{\partial p}{\partial x} \qquad (A.5)$$

Similarly
$$v = \frac{-1}{j\omega\rho} \cdot \frac{\partial p}{\partial y} \qquad (A.6)$$

and
$$w = \frac{-1}{j\omega\rho} \cdot \frac{\partial p}{\partial z} \qquad (A.7)$$

As the walls of the room are assumed to be rigid there can be no normal velocity component of the air particles near to any wall. These boundary conditions may be stated as

$u = 0$ for $x = 0$ and $x = l_x$

$v = 0$ for $y = 0$ and $y = l_y$

$w = 0$ for $z = 0$ and $z = l_z$

Applying the boundary conditions at $x = 0$, $y = 0$ and $z = 0$ to the respective equations for the particle velocities in the equations of the form of eqn.A.1. yields the undamped standing wave equation

$$p = P(\cos k_x x \cdot \cos k_y y \cdot \cos k_z z)e^{j\omega t} \qquad (A.8)$$

where P is the magnitude of the pressure. Equation A.8. shows that pressure antinodes exist at the wall surfaces i.e. where x = 0, y = 0 and z = 0. Substituting eqn. A.8. into eqn. A.5., the particle velocity in the x-direction becomes

$$u = \frac{k_x P}{j\omega\rho}(\sin k_x x \cdot \cos k_y y \cdot \cos k_z z)e^{j\omega t} \tag{A.9}$$

Applying to eqn. A.9. the condition

$$\text{for } x = l_x \text{ then } \sin(k_x l_x) = 0 \tag{A.10}$$

or $$k_x = \frac{n_x \pi}{l_x} \quad \text{where} \quad n_x = 0,1,2,3... \tag{A.11}$$

Similarly $$k_y = \frac{n_y \pi}{l_y} \quad \text{where} \quad n_y = 0,1,2,3... \tag{A.12}$$

and $$k_z = \frac{n_z \pi}{l_z} \quad \text{where} \quad n_z = 0,1,2,3... \tag{A.13}$$

From eqns. A.3, A.11, A.12 and A.13 one obtains the important expression

$$f = \frac{\omega}{2\pi} = \frac{c}{2}\left[\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2\right]^{1/2} \tag{A.14}$$

which gives the permissable frequencies corresponding to the normal modes of vibration within the room.

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