Noise and vibration control for HVAC

CIBSE Guide B5

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Department of Trade and Industry

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Note from the publisher

This publication is primarily intended to provide guidance to those responsible for the design, installation, commissioning, operation and maintenance of building services. It is not intended to be exhaustive or definitive and it will be necessary for users of the guidance given to exercise their own professional judgement when deciding whether to abide by or depart from it.

Foreword

It is inevitable that any air conditioning or mechanical ventilation system will produce noise. The hallmark of a well designed HVAC system is that the noise it generates does not cause problems for the occupants. Naturally ventilated buildings are not exempt from problems either, as they require openings in the building envelope to admit external air, and they may also admit unacceptable levels of external noise. Some noise is useful in masking sounds from colleagues and other sources and in multi-occupied offices it is important to provide privacy. Excessive noise from outside or from HVAC systems causes discomfort, unease amongst those working in the building, and may contribute to difficult communications, especially in open spaces. These all contribute to reduced productivity, since those who are not comfortable in their surroundings are not fully effective.

The building services engineer is expected to take responsibility for control of noise, whether from the mechanical plant or external noise transmitted through the system. Where noise is related to the building design, such as natural ventilation openings, the engineer needs to liaise with the architect and agree suitable locations for air inlets. Air inlets for natural ventilation systems are normally placed where they are remote from sources of air pollution and noise sources such as traffic, but they may be exposed to other sources of noise. In some cases the services engineer may be given responsibility for all services noise in a building, including pumps, lifts and escalators.

This new Guide section, *Noise and vibration control for HVAC*, which replaces section B12 of the 1986 edition of Guide B, provides comprehensive guidance on the identification and prediction of sources of noise and vibration in ventilation plant, and of the tools and techniques that are available to control them. It has been prepared by a Steering Committee comprising experts in the application of acoustics and vibration control, and addresses developments in the field since the previous edition was published. The Guide will be a valuable tool both for acousticians and for HVAC engineers working in design.

Dr Geoff Leventhall
Chairman, CIBSE Guide B5 Steering Committee

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1.1 General

Ventilation and air conditioning of buildings are the subject of increasing interest both because of their contribution to effective building performance and occupant satisfaction, and the increasing focus on energy consumption and carbon emissions from buildings. A particular cause of interest is the recently revised Part L[1] of the Building Regulations for England and Wales (and its equivalent Part J of the Building Standards (Scotland) Regulations[2]). The new Part L and its equivalent sets challenging new targets for energy efficiency of buildings in general and of mechanical ventilation systems in particular.

1.2 Overview of Guide B5

CIBSE Guide B5: *Noise and vibration control for HVAC* is the replacement for section B12 of the 1986 edition of CIBSE Guide B[3]. It has been comprehensively rewritten to take account of developments in the subject in the intervening years. It is intended for use by practising designers who hold a basic knowledge of the fundamentals of building physics and building services engineering.

It forms one of five new sections of Guide B, which together cover all the subject matter from the 1986 edition that has not been incorporated elsewhere. The other sections address heating and boilers (B1), ventilation and air conditioning (B2), ductwork (B3) and refrigeration (B4). Although the structure of Guide B5 does not follow the pattern of the other four sections, it is still intended to be used by engineers during the design of ventilation systems. Figure 1.1 sets out the outline design process for the various systems, and indicates those stages at which reference to this section will be most appropriate.

1.3 Noise from HVAC systems

Noise from heating, ventilation and air conditioning (HVAC) systems is one of the problems of air conditioned and mechanically ventilated buildings. Naturally ventilated buildings require convected air currents which, originating in apertures to the exterior, may transmit unacceptable levels of external noise. Excessive noise contributes to discomfort, uneasiness, difficult communication and loss of productivity, since those who are not comfortable in their surroundings are not fully effective. However, some noise is useful in masking the sounds from colleagues and other sources. Masking noise is especially important in multi-occupied offices in order to provide privacy.

For these and other reasons, criteria have been developed for controlling the levels of noise in buildings. Criteria are normally intended as 'levels not to be exceeded', aiming to produce noise levels which are comfortable and provide masking, whilst not being too difficult to achieve. However, they may become downgraded to 'design targets'. An important element of HVAC design is to control the noise to meet the specified criterion, whilst having minimum effect on the cost and aerodynamic performance of the fan/duct installation.

The building services engineer must take responsibility for the control of noise, whether it originates in the mechanical plant, or is external noise transmitted through the system. Where the noise is related to the design of the building, such as apertures for natural ventilation, the engineer must ensure that the architect, or other responsible person, is aware of potential problems and is advised on preferred locations for air inlets. Air inlets for natural ventilation systems are normally chosen to be on a part of the building remote from sources of air pollution and so remote from noise sources such as traffic, but they may be exposed to other noise sources.

The engineer may be required to take responsibility for all the services noise in a building, including pumps, lifts and escalators, as discussed in section 3.

Most services components in a building interact with each other, or with the building through their attachments to it. The HVAC installation should be treated as a complete system, the separate parts of which influence other parts, see Figure 1.2. The system components are duct sections, bends, take-offs, fittings, silencers etc. The termination of the complete system is at the occupants.

In the system shown in Figure 1.2, the air and noise travel from the fan through a number of components of the system, being affected by each one until they finally reach the occupants. During this process, one system may influence the performance of the preceding system. For example, the entry conditions into system 1 may modify the performance and noise generation of the fan. Other noise sources in the services in a building, considered in section 3, include chillers, compressors, pumps etc. Any equipment which is designed to move air or water, or to provide heating or cooling, must be considered as a potential noise source.

The mechanical equipment data should include noise, but the method by which this has been obtained is not always clear. Data may be measured, interpolated or unavailable. There are standardised procedures to be followed[5,6,7] and, unless it is stated that this has been done, the data should be treated with caution. The client, or other person responsible for approving equipment, should be clearly advised to consider the risks in the use of any component for which relevant octave band noise data, obtained under

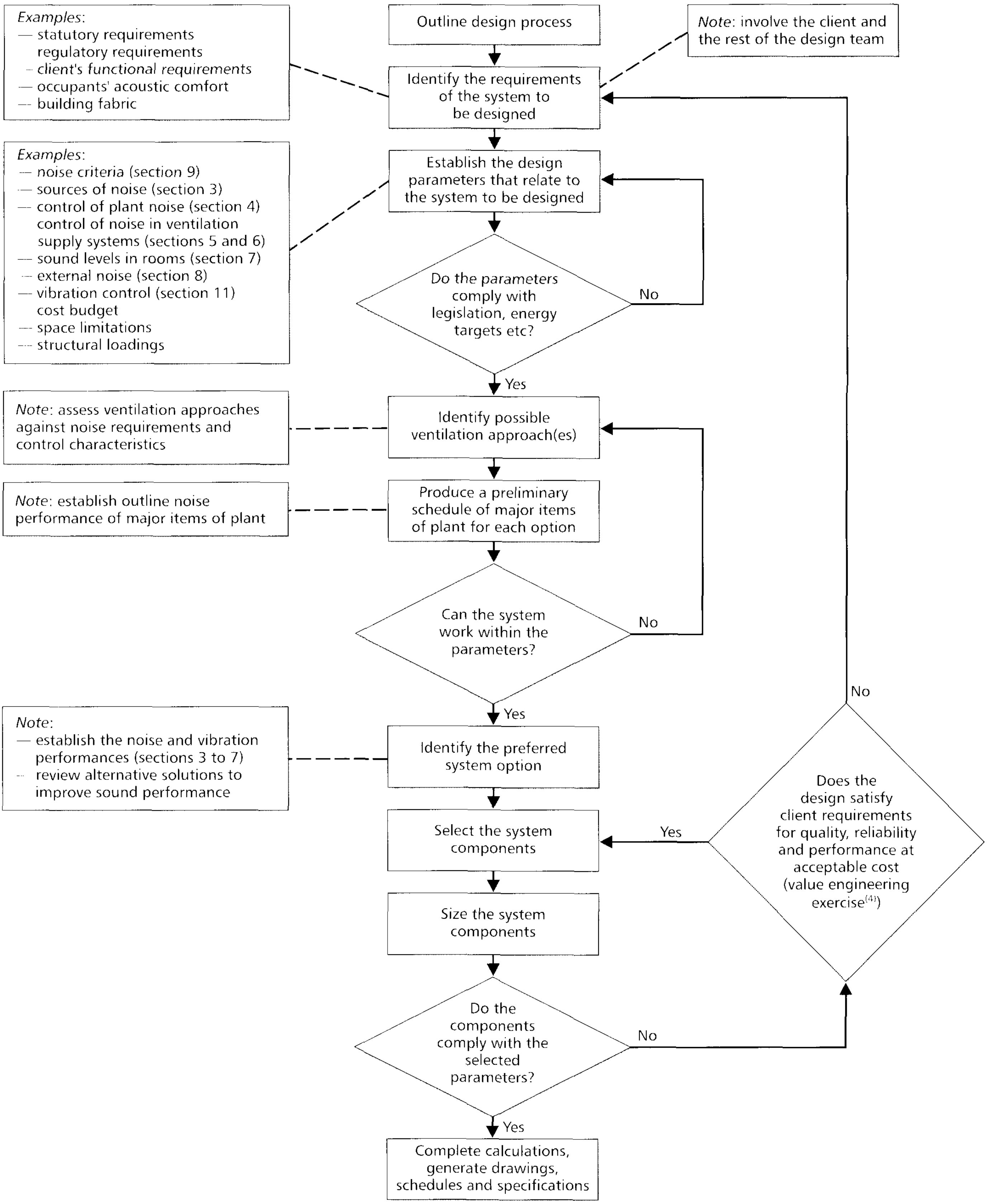

Figure 1.1 Outline design process

British, European, International or equivalent Standards procedures, is not available.

One factor, which the designer should be aware of, is that standardised noise measurement procedures are generally carried out in idealised situations, in which there is an attempt to prevent extraneous factors, e.g. turbulent airflow, affecting the measurements. Thus, the performance on site may be different from the data supplied with the item of equipment and it is important to ensure that the duct design, and other factors, are such as to minimise adverse interactions. This is broadly achieved by limiting air speeds and ensuring good flow conditions, which also contributes to energy conservation.

The complete system, as shown in Figure 1.2, originates in the fan inlet and continues through coil, filter, humidifiers and duct components to the duct termination and

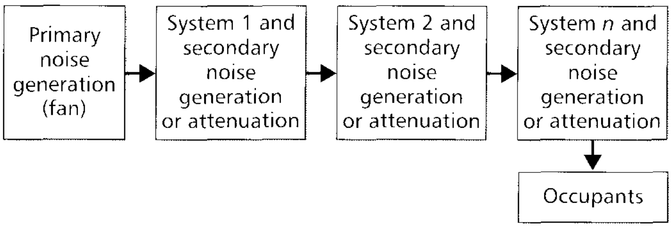

Figure 1.2 HVAC system in which the primary noise source is a fan

from there to the occupants, for whom the system is intended to provide a comfortable environment. Each component in the system either produces or reduces noise. The final noise level in the room is the summation of all these separate effects. The fan is the primary noise source, whilst airflow over duct fittings may generate aerodynamic noise. When the noise level exceeds criterion values additional noise control is required. Poor airflow conditions may cause duct components to have noise characteristics that differ from those given in the manufacturer's data. However, on the positive side, many duct components, such as bends and take-offs, contribute to attenuation of noise. The major noise-reducing component is normally a duct attenuator (silencer), but the operation of this is sensitive to airflow conditions and to its inlet and outlet duct connections. It may act as a source of additional noise, particularly at low frequencies.

Some noise is beneficial as an aid to masking the sounds of colleagues or to assist exclusion of external noise. This is known as 'masking noise', which may occasionally need to be added artificially. Background noise from the HVAC system is a useful masking noise, but should not be so high as to be distracting or to affect speech intelligibility. For a well-designed office, background noise of 35 to 45 dBA (approximating to room noise criterion levels of 30–40) usually permits communication by normal voice with colleagues in close proximity. During early formulation of room noise criteria, it was considered that good speech intelligibility between colleagues was a primary factor in office design. Whilst speech is still a factor to be considered, changes in working practices have modified the manner in which many people work. Most workers now have a local noise source provided by the cooling fan in their computer, and the increasing use of the telephone and e-mail means that speech is, perhaps, becoming less important for communication between colleagues in the same office. An improved understanding of workers' interaction with their environment has focussed attention on general comfort within the environment, of which noise is one factor, and the relation of comfort to productivity.

Noise in HVAC systems can be divided into three frequency ranges:

— low frequencies, characterised by 'rumble' noise, from about 31.5 Hz to 125 Hz on the octave band scale of measurement (see Appendix A5); rumble is typically, but not exclusively, from large central plant fans

— mid frequencies, from about 125 Hz to 500 Hz, lead to 'roar', which might be from small fans located close to the occupied space

— higher frequencies contribute to hiss and whistle, which are often a result of diffuser noise.

An excess in any range leads to an unbalanced noise spectrum and the potential for complaints[8]. A simple A-weighted noise measurement (see Appendix A5) does not give sufficient information on these three frequency ranges and is of only limited use.

The primary path by which HVAC noise reaches occupants of the space being served is directly down the duct and out into the room, but this is not the only path. Other paths include the following:

— *Breakout noise from a duct*: occurs mostly near to the fan and is perceived as a throbbing, rumbly noise, or as a tonal noise if there is tone generation by the fan. Breakout noise often reduces downstream, because the noise has already broken out through the sides of the duct. Breakout can be a problem to occupants when a duct passes over their space.

— *Structure-borne noise*: results from poor vibration isolation of machinery, resulting in fluctuating forces acting directly into the structure and transmitting vibration through the building. The consequent vibration of surfaces radiates 'structure-borne' noise.

It is unlikely that structure-borne vibration will be perceived directly by touching walls or floors (i.e. a 'feelable' vibration) for surfaces outside the plant room, but pipe vibration may still be detectable at a distance from sources such as pumps, compressors etc. However, vibrating surfaces, including those which cannot be felt as vibrating, still radiate noise, which is perceived by the listener as 'machinery hum', which may be fluctuating or steady. Fluctuating noise (throb) is more objectionable than a steady noise.

2 Summary of noise and vibration problems from HVAC

2.1 Typical sources of HVAC noise and their characteristics

Noise is produced by vibrating surfaces and by moving air streams. Sometimes the two interact, as in the case of fan blades. The primary source of the noise normally lies in the rotation of a machine, such as a motor, pump or fan. However, energy imparted to air or water can be converted into noise through interaction of fluid flow with solid objects, e.g. louvres in a duct termination. A very broad generalisation is that the 'noise conversion efficiency' of a machine is around 10^{-7} of its input power, but there are wide variations above and below this figure, whilst aerodynamic noise increases rapidly with air velocity. A fan, which contains both drive motor and fan wheel, is more likely to convert around 10^{-6} of its input power to noise. Sound powers are low in terms of wattage but, because of the sensitivity of the ear, only milliwatts of acoustic power are required to produce a loud noise (see Appendix A1).

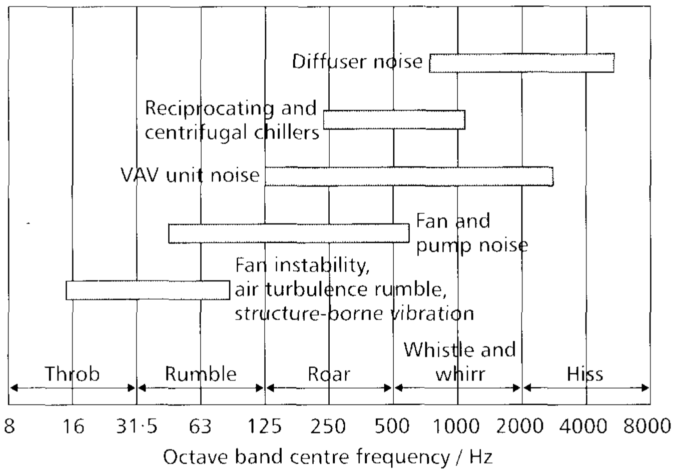

Figure 2.1 Frequencies at which different types of mechanical equipment generally control sound spectra (reproduced from ASHRAE *HVAC Applications Handbook* by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers)

Different types of mechanical equipment produce noise over different frequency ranges. This is illustrated in Figure 2.1, which shows the frequencies most likely to be produced by equipment and gives a typical subjective terminology by which listeners might describe the noises.

Figure 2.1 indicates that central plant (fans and pumps) is likely to cause noise up to about 500 Hz, whilst the very lowest frequencies are a result of defective installation. VAV units lead to noise from about 125 Hz to 3000 Hz, fan powered units being responsible for the lower end of this range. Chillers lead to noise in the 250 Hz to 1000 Hz range whilst higher frequencies are due to diffuser noise. These system components are considered in more detail in section 3.

2.2 Transmission paths

Figure 2.2 shows transmission paths for roof-top and ground level plant rooms and are summarised as follows:

— noise radiates to atmosphere from the air inlet (path 1)

— vibration from the fan transmits to the structure (path 5)

— noise from the plant breaks out of the plant room (path 3)

— noise may break out of the supply duct to adjacent spaces (path 2)

— incorrect duct or pipe anchoring may put vibration into the structure (path 5)

— duct borne noise is emitted from the room units (path 4)

— vibration from ground level plant gets into the structure (path 5)

— noise from plant transmits through walls or windows to adjacent spaces (path 2).

In controlling the noise of the HVAC plant, all transmission paths must be assessed for their contribution to the final noise in occupied spaces and the paths controlled accordingly.

2.3 Control of the transmission paths

This section considers some general principles of good practice in noise and vibration control in HVAC. More details are given in sections 4, 5, 6 and 11. The preferred

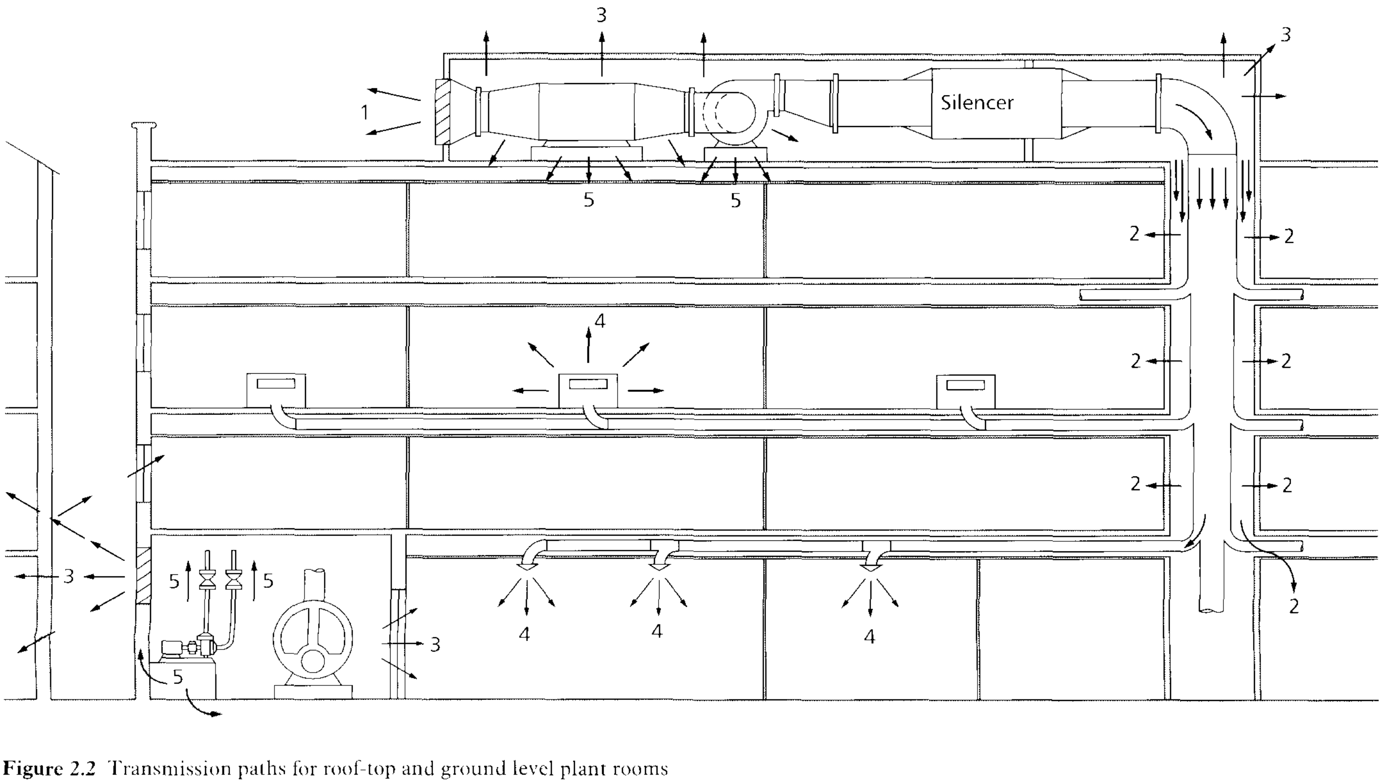

Figure 2.2 Transmission paths for roof-top and ground level plant rooms

way to control noise is to prevent it occurring in the first place, but some noise generation is unavoidable from realistic airflow velocities. In HVAC systems, controlling noise means:

- choosing the operating condition of the fan so that it is at a high efficiency point on its characteristic; this minimises fan noise
- Ensuring good flow conditions for the air stream; the consequent benefits include components behaving more nearly as described in the manufacturer's data and reduced pressure losses, conserving energy and saving operating costs
- isolating vibrating components, including all machinery, ducts and pipework from the structure
- choosing an in-duct silencer or other means to control airborne noise in ducts[9]; a full silencer may not be required, as lining bends with acoustic absorbent may be adequate, but this depends on the results of noise predictions (see section 10).

Noise control relies on attention to detail, both in the design and the implementation. It depends on choosing the correct components and ensuring that they are installed correctly.

There are many instances of problems which have resulted from inadequacies in design and installation, including:

- undersized fans, which could not accept the pressure loss of retrofit silencers
- oversized fans, which were working on an undesirable part of their characteristic
- vibration isolators which were by-passed by solid connections
- unsealed gaps left between spaces.

3 Noise sources in building services

There are a large number of potential noise sources in a building services installation including fans, duct components, grilles and diffusers, plant such as chillers, boilers, compressors, cooling towers, condensers, pumps, stand-by generators, lifts and escalators. The tendency away from central plant to local systems in the ceiling space has brought noise sources closer to occupants and increased the problems of noise reaching occupied rooms. Noise from a plant room, especially large central plant, may break out to the exterior and be a source of annoyance to neighbours. Nuisance to neighbours comes under the responsibility of the local environmental health department, which may require the noise to be abated. Local authorities often apply conditions to planning consents in order to protect neighbours from nuisance caused by building services plant. Such conditions must be complied with.

Prediction formulae have been established for some items of plant by measurements on a sample of the plant. Much of this work was carried out many years ago, when information was not available from manufacturers. Since that time designs have changed. There have been efforts by the larger manufacturers of plant to reduce plant noise, whilst most manufacturers have also become aware of the need to provide data on the noise of their plant. The main source of information on noise is now the manufacturer. Inability, or reluctance, to provide such information might influence the choice of manufacturer.

The measurement conditions for plant noise must be specified along with the relation of the measurement procedure to standardised methods. It should be remembered that the installation conditions may not be the same as the measurement conditions and that there are uncertainties in measurement, especially at low frequencies.

In the very early stages of a project, plant may not have been fully specified and, only under these temporary circumstances, generic information on noise may be used to give an initial overview of the noise of the project and to indicate space requirements for noise control, e.g. how much space to allow for in-duct silencers. Generic prediction information is given in Appendix A2, which must be regarded as for temporary use only, until equipment-specific information is available. The uncertainties of generic information are at least ± 5 dB, and often greater.

3.1 Fans

Control of fan noise depends on:

- choosing an efficient operating point for the fan
- design of good flow conditions
- ensuring that the fan is vibration isolated from the structure
- ensuring that the fan is flexibly connected to the duct.

Where fan noise will be a problem, an in-duct attenuator should be used; these are described in detail in section 6.

3.2 Variable air volume (VAV) systems

Noise from VAV systems depends on the method of control. Where the flow is adjusted by means of a damper or throttle valve, noise is mainly generated by turbulence at the obstruction to flow. Where control is by a fan, either cycled or modulated, the fan is the source of the noise, but modulation may affect the noise by changing the operating point of the fan. Improper air balancing must be avoided to ensure that the fan does not deliver at an unnecessarily high static pressure.

Manufacturers' noise data for a VAV system will not be achieved in practice unless careful attention is given to the support of the box and to the airflow conditions both into and out of the box. There should be straight duct runs at both sides of the box in order to minimise turbulent flows and the resulting potential for enhanced noise. Break-out noise from the box should also be taken into account.

3.3 Grilles and diffusers

Control of air velocity and flow conditions is the key to reducing this noise. Manufacturers' data should be consulted. Grilles and diffusers are the last stage in noise control because, once the sound has escaped into the room, there is no further attenuation other than by room surface absorption. Grilles and diffusers are considered in Appendix A2.

3.4 Roof-top units

Roof-top units have three main noise paths into the building space:

— through the duct

— breakout from the casing of the unit, which then transmits through the roof; this is most likely to occur underneath the unit, where the noise levels will be highest

— vibration transmission from the unit to the roof and consequent re-radiation of noise.

Noise through the duct is treated by absorptive material in the unit or by a silencer in the duct. Both supply and return may require treatment. Breakout, if a problem, is controlled by strengthening the underside of the casing or by adding sound attenuating material underneath the unit. Vibration transmission is reduced by well-designed anti-vibration mounts.

3.5 Fan coil units

These are an example of how noise sources are brought close to the occupants. Room perimeter units must be chosen for their low noise, by reference to manufacturers' information. Ceiling void units must be carefully mounted with inlet and discharge ducts designed to minimise the external resistance and with an adequate return air path ensured. A discharge silencer, or lined duct, may be required. Noise breakout through the casing must not be neglected. The sound power of the units will be provided by the manufacturer.

3.6 Chillers, compressors and condensers

These produce both tonal and broadband noise. The tonal noise is typical of that from rotating or reciprocating machinery, linked to the rotational frequency. The broadband noise is from fluid flows, either liquid or gas. The tonal noise is often dominant, perceived as a whine or whirr, but the frequency range depends on the mode of operation. Reciprocating compressors have a relatively low-frequency fundamental tone, related to the oscillation frequency of the pistons. Screw compressors have strong tones in the octave bands between 250 Hz and 2000 Hz, and may require special attention to noise and vibration control, especially when they are located externally.

3.7 Pumps

Pumps produce external noise from the motor, fluid-borne noise from the impeller and vibration into both the structure and the pipes. Noise problems may arise from the airborne noise, controlled by choosing a non-sensitive location or by an enclosure for the pump. If the pipes make solid contact with a radiating surface, there is the potential for both fluid-borne noise and pipe vibration to reappear as airborne noise at a distance from the pump. It is necessary to:

— use vibration isolators to isolate the pump from the building

— use a flexible connection from pump to pipes

— use resilient mountings for supporting the pipe to the structure.

3.8 Stand-by generators

This noisy plant, which requires to be tested at regular intervals, is often housed in a separate generator room. A flow of fresh air is required both for the engine intake and for cooling. Noise problems arise from:

— the fresh air inlet

— the warm air discharge

— the engine exhaust

— the structure, due to vibration transmission.

The air inlet and discharge may require to be silenced by use of duct silencers, acoustic louvres or equivalent measures. The engine exhaust silencer will need to be selected to satisfy local requirements for environmental noise. Vibration isolation must be discussed with the supplier of the generator. It is common practice to line the generator room with acoustic absorbent in order to reduce the build-up of reverberant sound.

3.9 Boilers

Hot water boilers may vary in size from a few hundred kilowatts, or lower, up to megawatts, depending on the heating requirement. Noise sources within the boiler room are from the air supply fan and the combustion. External noise is from the flue. A small boiler of about 200 kW capacity may have a spectrum peak at around 125 Hz and overall sound power level of 90 dBA. In general, the frequency of the peak drops with increasing boiler capacity so that, in the megawatt range, the spectrum peak is at 63 Hz or below. A large boiler, of several megawatt capacity, may have an overall sound power in excess of 100 dBA. Manufacturers' information should be consulted for octave band data. The presence of low frequencies leads to the total sound power being greater than the A-weighted sound power.

3.10 Cooling towers

Cooling tower noise is mainly noise from the fan, details of which should be available from the manufacturer. See also Appendix A2 for fan noise prediction.

3.11 Lifts

The intermittent operation of lifts, including door opening and closing, motor surges and operation of brakes, may cause disturbance in adjacent occupied spaces. Most of the noise is structure borne, for example impacts on door stops and lift machinery vibration. It is possible to reduce each of the noise sources by design and correct installation. Advice should be sought from the manufacturer.

3.12 Escalators

Escalators are a source of noise and vibration from the motor and drive mechanism. This is not normally a problem provided that the equipment has been installed correctly. However, there is a possibility that vibration input from the motor will couple with a resonance on a surrounding floor or wall to produce a noticeable effect, which will then require correction.

4 Noise control in plant rooms

4.1 Health and safety

Plant rooms may have noise levels which exceed the limits for worker protection. If, for example, a maintenance engineer spends long periods in a very noisy plant room it will be necessary to wear hearing protection as noise exposure may exceed the permitted daily 'noise dose'.

The Health and Safety Executive has defined three 'action levels' in the Noise at Work Regulations[10]. (See Appendices A1 and A5 for definitions of acoustical terms.) These action levels are as follows:

(*a*) *First action level*: daily personal noise exposure ($L_{EP,d}$) of 85 dBA.

(*b*) *Second action level*: daily personal noise exposure ($L_{EP,d}$) of 90 dBA.

(*c*) *Peak action level*: peak sound pressure level of 200 Pa (which is equivalent to a peak sound pressure level of 140 dB re 20 μPa).

The peak action level refers to impulsive noise. A useful guide for steady noise is that, if it is necessary to raise one's voice to shouting level in order to communicate clearly with someone standing about 2 m away, the first action level has been exceeded, but this must be checked by measurement. If measurements show that the first action level has been reached, it is a requirement of the Noise at Work Regulations to make hearing protection available for employees who request it. When the second or peak action levels have been reached, there is an obligation to provide hearing protection for all exposed employees and to ensure that these are worn. Further details are given in the HSE's *Guidance on the Noise at Work Regulations*[11].

It is, of course, advisable to put some effort into design of a plant room in order to prevent breaching the action levels. This is not only healthier for employees, but exceeding the second action level places additional legal obligations on management for enforcement, regular checks, record keeping etc.

(An agreement within the European Union will lead to reduction of the action levels by the year 2004. The first action level will then be a daily exposure of 80 dBA, the second action level will be 85 dBA and the peak action level will be 112 Pa[12]).

4.2 Breakout noise from plant rooms

Figure 2.2 indicates how noise breaks out from plant rooms, either to atmosphere or to adjacent occupied space. In order to reduce breakout noise the following steps must be taken:

— Isolate the equipment from the structural floor. This can be either by individual vibration isolation of each piece of equipment or by using a floating floor (see section 11).

— Ensure that the separating walls give sufficient attenuation (see section 7.4). This requires information on the sound power outputs of each item of plant, so that the overall level and spectrum of the plant room noise can be estimated. The levels are then related to the noise criterion for the adjacent space and the requirements for wall attenuation determined. Many plant rooms have hard walls, which contribute to build-up of reverberant noise. This results in a higher internal level than might be anticipated, but the effect is reduced by lining some of the plant room surfaces with sound absorbent. Reverberation must be included as a factor in predicting plant room noise levels (see section 7.6). It may be necessary to design special noise attenuating double isolating walls to protect sensitive locations adjacent to plant rooms, and to make sure that pipes and other components are not fixed directly to these walls.

— Ensure that all penetrations of the plant room walls, floor or ceiling are carefully sealed.

— Pay proper attention to noise transmission through the plant room external walls and silencing of air inlets and outlets, louvres etc. in order to prevent noise to atmosphere which, as a potential for disturbance in neighbouring buildings, should not be neglected in the design. It may be necessary to include a silencer in the air inlet to the fan.

4.3 Break-in noise in plant rooms

This refers to high levels of plant room noise entering the ducts and then being transmitted to occupied spaces. The problem is controlled by correct location of a duct silencer. The silencer should be placed to penetrate the plant room wall, so that all break-in noise to the duct is reduced along with other duct-borne noise.

4.4 Estimation of noise levels in plant rooms

In a cramped plant room, the direct sound from the nearest item of plant is likely to control the local noise. However in a large uncrowded plant room, where a reverberant sound field may be assumed, the reverberant field is approximated by equation A7.2, see Appendix A7, used with the sound power levels of each item of plant. A reverberation time of 2 seconds should be assumed. The contributions of each item to the reverberant field are added, as in Appendix A1. The direct sound of nearby plant must also be included.

5 Airflow noise — regeneration of noise in ducts

5.1 Airflow generated noise

This noise, also known as regenerated noise, is produced by turbulence in the airflow. It is reduced by ensuring streamline flow and minimising obstructions or abrupt changes in the flow. Airflow noise increases as approximately the sixth power of flow velocity and is generally broad band. Sometimes a tone is perceptible with a frequency of about $f = 0.2\ u/d$, where u is the flow velocity ($m \cdot s^{-1}$) and d the dimension of an obstruction in the flow (m). In general, the regenerated sound power of a duct fitting is given approximately by

$$L_W = C + 10 \lg A + 60 \lg u \qquad (5.1)$$

where L_W is the airflow generated noise power level (dB), C is a constant, which varies with the fitting, A is the minimum flow area of the fitting (m^2) and u is the maximum flow velocity in the fitting ($m \cdot s^{-1}$). (*Note*: lg = $\log_{10}$.)

Equation 5.1 illustrates the importance of limiting the velocity, since a doubling of velocity gives an 18 dB increase in regenerated sound power. Reduction of velocity is achieved by increasing the duct size or, for example, running two parallel ducts. Where it is anticipated that velocity generated noise will be a problem, silencing must be installed after the final in-duct noise source. For typical fittings and flow velocities, the overall regenerated power level is likely to be in the region of 50–70 dB, but the levels vary with frequency. The maximum level is at $f = 0.2\ u/d$, as above. A more detailed determination of regenerated noise is given in Appendix A2.

5.2 System effects on regeneration of noise

Figure 5.1[13] illustrates good principles of duct design in order to avoid turbulence and its associated pressure loss and noise. Some obstructions, such as dampers, are necessary, but multiple dampers are preferred to single dampers in a noise-sensitive system. Dampers should be fitted at least 1.5 to 2.0 meters back from a duct termination in order to reduce damper noise escaping into occupied space. Manufacturers' literature on damper noise should be consulted and the levels assessed in relation to other noise in the duct, in order to determine whether secondary silencing is required after the damper.

A detailed prediction method for duct termination regenerated sound power is given in *Sound and vibration design and analysis*[14] and is considered further in Appendix A2.5. An estimate of regenerated noise is also given by equation 5.1 and Table 5.1, which is for well designed systems as illustrated in Figure 5.1, where regenerated noise is unlikely to be a problem.

Damper manufacturers can supply information from which the regenerated noise of their products may be estimated. The information is often provided in terms of air velocity, resulting pressure drop and a reference overall regenerated sound power level. Spectrum corrections are applied to the reference level. The spectrum corrections will be different for different blade settings.

The aim should be for system design and construction which ensures that regenerated noise in the duct is not a problem. A guide to maximum permitted air velocities is given in Table 5.2 for different types of space. A 'critical' space is, for example, a private office or similar. A 'normal' space is a general office, whilst a 'non-critical' space might be for circulation or storage. These spaces may typically have rating criteria of 25, 35 and 45. Note that the

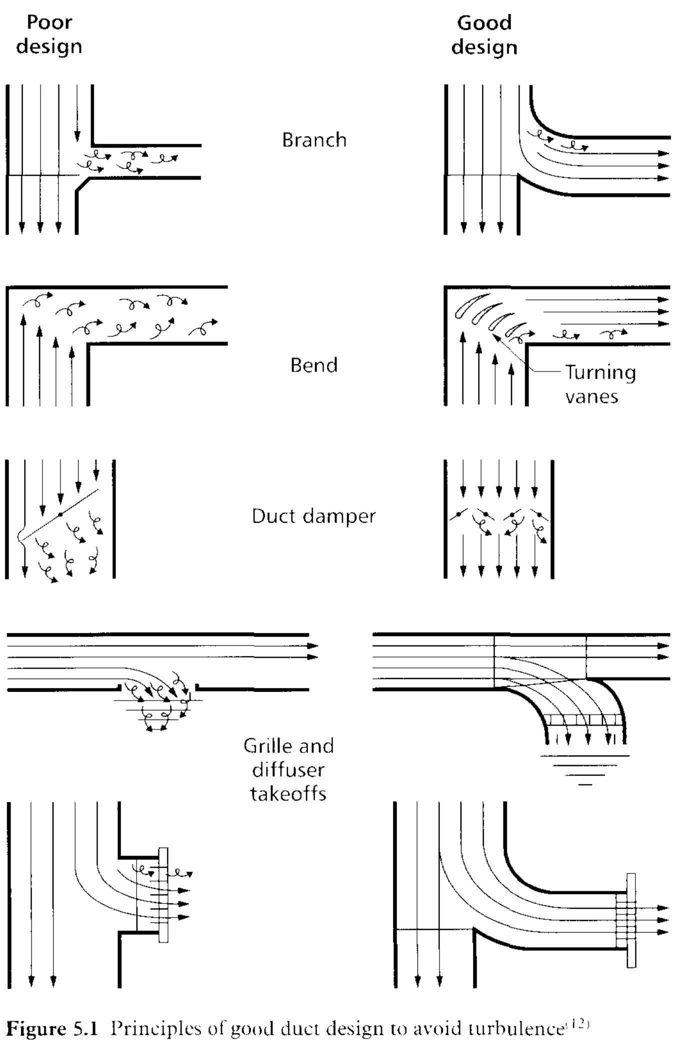

Figure 5.1 Principles of good duct design to avoid turbulence[12] (reproduced from *Control of Noise in Ventilation Systems* by M A Iqbal, T K Willson and R J Thomas, by permission of E & F N Spon)

Table 5.1 Corrections to equation 5.1 for low turbulence duct fittings

Duct fitting	Value of C / dB	Notes	Octave band power level correction / dB for stated octave band / Hz							
			63	125	250	500	1000	2000	4000	8000
Straight duct	−10	No internal projections	0	−2	−7	−8	−10	−12	−15	−19
90° radiused bend	0	Aspect ratio 2:1, throat radius $w/2$	0	−2	−7	−8	−10	−12	−15	−19
90° square bend with turning vanes	+10	Close spaced, short radius single skin vanes	0	−2	−7	−8	−10	−12	−15	−19
Gradual contraction	+1	Area ratio 3:1, A and u as for smaller duct	0	0	−10	−16	−20	−22	−25	−30
Sudden contraction	+4	Area ratio 3:1, A and u as for smaller duct	+3	0	−10	−16	−20	−22	−25	−30
Butterfly damper	−5	A and u apply to minimum area free damper	0	−3	−9	−9	−10	−17	−20	−24

doubling of air velocity between critical and non-critical spaces represents a sound level difference of up to 20 dB.

Similar velocity limits for grilles and diffusers are given in Table 5.3 for general guidance, but manufacturers' information should be consulted. Note that multiple grilles in a room may require a reduction in the velocity through each grille

5.3 Silencers

Silencers are considered in section 6.7. They are an obstruction in the flow and therefore generate turbulence noise. This noise, which is dependent on flow velocity, is sometimes referred to as the 'self-noise sound power level' of the silencer. Self-noise is likely to be from 50 to 80 dB overall sound power level. Manufacturers' data must be consulted.

Table 5.2 Guide to maximum duct velocities for final runs to outlets*

Duct location	Duct type	Maximum air velocity for stated type of space / $m \cdot s^{-1}$		
		Critical	Normal	Non-critical
Riser or above plasterboard ceiling	Rectangular	5	7.5	10
	Circular	7	10	15
Above suspended ceiling	Rectangular	3	5	6
	Circular	5	7	10

* Velocities can be increased by about 50% in main ducts

Table 5.3 Maximum free air velocity for supply and return air openings

Type of opening	Permitted air velocity for stated type of space / $m \cdot s^{-1}$		
	Critical	Normal	Non-critical
Supply	1.5	2.5	3
Return	2	3	4

6 Techniques for control of noise transmission in ducts

6.1 Duct components

Duct components may include:

- straight ducts of various lengths, rectangular or circular in cross-section
- silencers
- bends (elbows), right angled or curved
- branches, which may have one or more take-offs
- distribution boxes (plenums)
- terminal units, grilles, diffusers, registers.

Additionally, transition pieces connect the fan or silencer to the duct. Most ducts are of sheet metal construction but short flexible lengths of other material may be used, e.g. to connect a ceiling void fan coil unit to a terminal. There are specialist duct systems for on-site duct manufacture from sheets of stiff resin-bonded fibreglass or similar.

6.2 Unlined straight ducts

Attenuation of noise in straight unlined ducts is mainly through transfer of energy from the sound wave to the duct wall. This energy then appears as either breakout noise from the duct or as duct vibration. A duct with stiff walls will vibrate less than one with flexible walls, and will therefore have both lower attenuation and lower breakout noise. Duct attenuation is expressed as decibels per metre ($dB \cdot m^{-1}$) and is lower for circular ducts than for rectangular, as circular ducts have greater wall stiffness than rectangular ducts. Circular ducts might require additional attenuation to be added into the system. Duct flexibility varies with the duct dimensions and frequency of excitation, so that attenuation depends on these

Table 6.1 Approximate attenuation of unlined sheet metal ducts at octave frequencies

Duct section	Mean dimension or diameter / mm	Attenuation / dB·m⁻¹ for stated octave band / Hz			
		63	125	250	500 and above
Rectangular	≤300	1.0	0.7	0.3	0.3
	300–450	1.0	0.7	0.3	0.2
	450–900	0.6	0.4	0.3	0.1
	>900	0.5	0.3	0.2	0.1
Circular	<900	0.1	0.1	0.1	0.1
	>900	0.03	0.03	0.03	0.06

Table 6.2 Approximate attenuation of lined circular ducts[14] (reproduced from *Sound and Vibration Design and Analysis* by permission of the National Environmental Balancing Bureau, Gaithersburg, MD)

Duct diameter / mm	Attenuation / dB·m⁻¹ for stated octave band / Hz							
	63	125	250	500	1000	2000	4000	8000
150–300	1	2	3	5	7	6	5	4
300–600	0.5	1	2	4	6	5	4	3
600–900	0	0.5	1	3	4	3	2	2
900–1200	0	0	1	2	2	2	2	2

quantities. The attenuation in straight sheet metal ducts is given in Table 6.1.

6.3 Lined straight ducts

Lined ducts are an effective way of reducing noise, but refer to section 6.9 on the use of fibrous materials in ducts. Published data on absorption coefficients for acoustic lining materials shows a continuous rise with increasing frequency, often up to the maximum of unity (total absorption). Absorption coefficients are measured either at normal incidence or random incidence, following standard procedures. However, these values do not apply to absorbent duct linings, since when the material is used as a duct liner, high frequencies do not interact with it in the same way as for the absorption coefficient measurement, but propagate down the duct with reduced effect from the absorbent lining. This is particularly so at high frequencies, where the attenuation in the lined duct reduces from its maximum at mid frequencies. Physically, at high frequencies the sound propagates down the centre of the duct and has reduced interaction with the lining. For a given lining material and thickness, the smaller the duct widths or diameters, the greater the attenuation.

For a lined plenum, the published absorption properties of a material should be used where the plenum is large enough for the sound to reflect within it; that is, when the plenum dimensions are greater than the wavelength of the sound to be controlled. Prediction formulae for lined ducts are considered in Appendix A2.6. An important parameter is the ratio p_d/A_d, where p_d is the duct perimeter (m) and A_d is the cross sectional area of the duct (m^2).

Results of the prediction formulae are presented in Figures 6.1 and 6.2, where the insertion loss is shown against frequency with p_d/A_d as the variable. Figure 6.1 is for 25 mm lining and Figure 6.2 for 50 mm lining. Thus, from Figure 6.1 for a 600 mm by 700 mm (i.e. 0.6 m by 0.7 m) duct, $p_d/A_d = 6$ and the insertion loss at 500 Hz, for example, is interpolated as about 4 dB·m⁻¹. The basis of

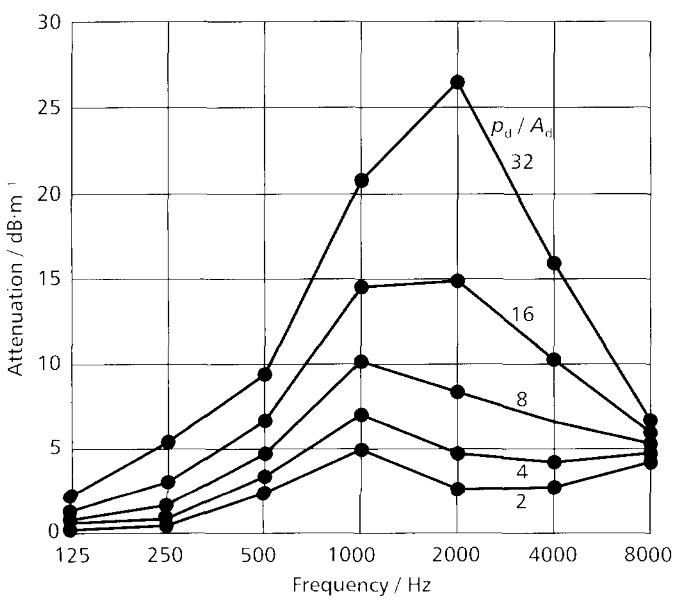

Figure 6.1 Attenuation of lined duct; 25 mm lining

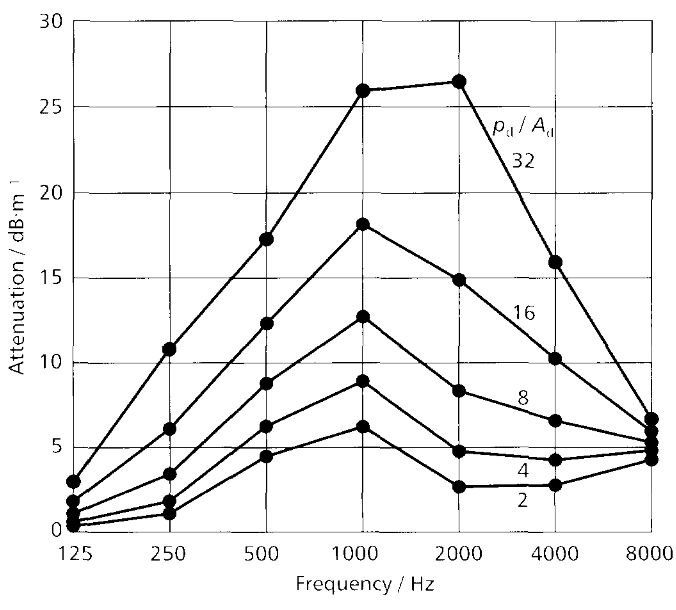

Figure 6.2 Attenuation of lined duct; 50 mm lining

Figures 6.1 and 6.2 is an extensive series of measurements on rectangular ducts, using the substitution method, in which the lined duct is replaced by a similar unlined section.

The attenuation of lined circular ducts is shown in Table 6.2 for a 25 mm lining(14). Increasing the lining thickness to 50 mm increases the attenuation by only a small amount. Short lengths of lined circular ducts, as connections to diffusers, may be the final opportunity for noise control.

6.4 Duct bends

A bend, lined or unlined, has greater attenuation than a similar length of lined or unlined duct, since some of the sound energy impinges at right angles on the facing part of the bend. The attenuation of straight lined ducts is limited at low and high frequencies, but the attenuation of a lined bend increases with frequency, before falling slightly at the highest frequencies. The controlling factors for a particular bend are the duct width and the frequency. To gain maximum benefit from a lined bend, the lining should be installed both before and after the bend for a distance of at least two duct widths or diameters.

6.4.1 Square elbows

Tables 6.3 and 6.4, based on information published by the (US) National Environmental Balancing Bureau(14), compare lined and unlined elbows in terms of the numerical value of the product of frequency f (kHz) and width w (mm). Thus, for a 300 mm duct at 2 kHz, $(f \times w) = 600$.

6.4.2 Round elbows

The insertion loss values for round elbows are not as well known as for square elbows, but an approximation of the attenuation is given in Table 6.5 for unlined round elbows(14).

Table 6.3 Approximate attenuation of unlined and lined square elbows without turning vanes(14)*

Frequency × width / kHz·mm	Attenuation / dB	
	Unlined	Lined
<50	0	0
50–100	1	1
100–200	5	6
200–400	8	11
400–800	4	10
>800	3	10

Table 6.4 Approximate attenuation of unlined and lined square elbows with turning vanes(14)*

Frequency × width / kHz·mm	Attenuation / dB	
	Unlined	Lined
<50	0	0
50–100	1	1
100–200	4	4
200–400	6	7
> 400	4	7

Table 6.5 Approximate attenuation of unlined round elbows(14)*

Frequency × width / kHz·mm	Attenuation / dB
<25	0
25–50	1
50–100	2
>100	3

Lined round elbows have greatest attenuation for smaller ducts, with a gradual decrease in attenuation for a given frequency as the duct dimension increases with constant lining thickness. As an approximation, lined elbows of all sizes with 25 mm lining achieve at least 10 dB reduction at 1000 Hz and above, about 7 dB at 500 Hz, 5 dB at 250 Hz, 2 dB at 125 Hz and zero at 63 Hz. Increasing the lining thickness to 50 mm gives additional attenuation of 3 dB, except at 63 Hz. These figures are very approximate and intended as a qualitative guide. More precise values should be available in a computer prediction program.

6.5 Duct take-offs

When airflow is taken from the main duct by a junction or side branch, it is assumed that the sound power divides as the areas of the ducts as in Figure 6.3. The attenuation is given by equation 6.1.

$$\Delta L = 10 \lg \frac{A_1 + A_2}{A_1} \tag{6.1}$$

where ΔL is the attenuation (dB), A_1 is the cross sectional area of the take-off branch (m^2) and A_2 is the cross sectional area of the main duct after the branch (m^2).

Changes in total cross-section may also cause reflection of sound back up the duct, but equation 6.1 represents the major effect.

* Tables 6.3, 6.4 and 6.5 are reproduced from *Sound and Vibration Design and Analysis* by permission of the National Environmental Balancing Bureau, Gaithersburg, MD)

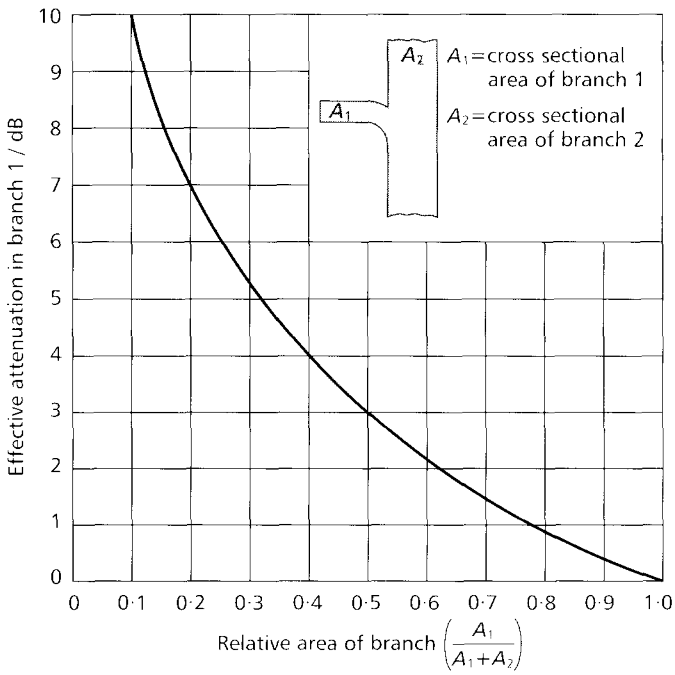

Figure 6.3 Effective attenuation of a duct branch

6.6 End reflection loss

The change in propagation medium, as sound travels from a duct termination into a room, results in reflection of sound back up the duct. The effect is greatest at long wavelengths (i.e. low frequencies) and is a contribution to the control of low frequency noise from the system. When a high level of low frequency noise is anticipated, it can be useful to reduce the sizes of ducts feeding a space and increase their number proportionately.

The end reflection loss of a duct terminated flush with a wall is given as[14]:

$$\Delta L_R = 10 \lg [1 + (0.8\, \lambda / \pi\, d)^{1.88}] \qquad (6.2)$$

where ΔL_R is the reflection loss (dB), λ is the wavelength of the sound (m) and d is the diameter of a circular termination (m). Wavelength is determined from $c = \lambda f$, see Appendix A1. The effective diameter of a rectangular termination is:

$$d = \sqrt{(4 A / \pi)} \qquad (6.3)$$

where A is the area of the termination.

Equation 6.2 shows that the relation between wavelength and duct dimension (λ / d) is the controlling factor. The equation is to be used for end reflection losses for terminations having aspect ratios (i.e. height/width) of the order of unity. Slot diffusers were not investigated in the work which led to equation 6.2 and manufacturers' data should be consulted for these components. Values of end reflection loss are given in Table 6.6.

6.7 Passive silencers and plenums

6.7.1 Passive silencers

A passive silencer, see Figure 6.4, contains localised sound absorbent, normally associated with narrowed air passages. Both rectangular and circular silencers are used. The rectangular silencer is built up from an assembly of absorbent splitter modules. Its acoustic performance is determined largely by that of one of its single assemblies, which approximates to a narrow, lined duct. The parallel assemblies give increased capacity to carry the required air volume without increase of velocity. The cross section of the silencer is often significantly greater than that of the duct in which it is located. Changes in shape or cross section affect the operation of the silencer. Attenuation and pressure loss increase as the airways are narrowed.

Table 6.6 End reflection loss at octave band frequencies[14] (reproduced from *Sound and Vibration Design and Analysis* by permission of the National Environmental Balancing Bureau, Gaithersburg, MD)

Duct dimension, D / mm	End reflection loss / dB at stated octave band / Hz				
	63	125	250	500	1000
150	18	13	8	4	1
300	13	8	4	1	0
450	10	6	2	1	0
600	8	4	1	0	0
750	6	2	1	0	0
1000	5	2	1	0	0
1200	4	1	0	0	0

Another important variable is the length of the silencer. Longer silencers have increased attenuation and some additional pressure loss. Silencer pressure loss is not proportional to the length of the silencer, since significant pressure loss occurs at the entry and exit. A circular silencer is normally either open ('unpodded') or contains an inner assembly, the absorbent 'pod' or 'bullet'. Special systems may have concentric absorbent layers with airways between.

The static insertion loss of a silencer is measured without airflow, using noise from a loudspeaker. The dynamic insertion loss includes effects of airflow and may give lower results.

Care must be taken in the location of silencers in order to prevent interaction with other components. The attenuation values of two silencers placed close together in series are not necessarily additive, since interactions and poor airflows may affect their operation.

Silencers are included in the design when an analysis of the system has shown that the room criteria will not be met. The attenuation of passive silencers is low at low frequencies, rises to a maximum in the middle frequencies (1–2 kHz) and drops at higher frequencies. Manufacturers' silencer data should be backed by a statement of the standards by which it was measured. Factors to be considered in selecting a silencer include its attenuation at different frequencies and its pressure loss. Duct designs leading to poor entry and exit flow conditions increase the pressure loss and may generate additional low frequency noise. Particular attention should be given to the exit conditions.

It is advisable to locate silencers several duct widths or diameters clear of bends, in order to maintain good airflow. Commercial packaged silencers are available in a wide range of configurations giving some control over dimensions, pressure loss and attenuation, in order to optimise the choice for a particular application. In general, higher attenuation is obtained by reducing the distance across the airway and by increasing the length of the silencer. Reduced pressure loss is given by increasing the

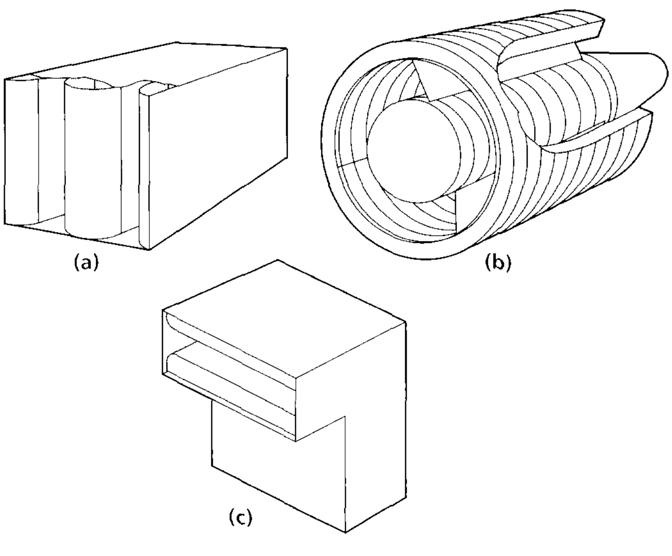

Figure 6.4 Dissipative duct silencers; (a) rectangular, (b) circular, (c) rectangular elbow (reproduced from ASHRAE Handbook: *HVAC Applications*, by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers)

airway area whilst keeping the cross dimension constant. That is, by either increasing the height of the silencer or by adding additional airways to increase its width. Both these measures give a greater area for the airflow and so reduce the velocity in the airways, although too rapid a transition from a duct to a silencer of greater cross-section than the duct, will not produce the full benefit. For a given airway, the pressure loss increases as the square of the air velocity and it is seen that, in specifying a silencer, various requirements of insertion loss, pressure loss, space and cost must be balanced.

Location of a silencer should be between the major noise source and the occupied space, preferably between straight duct runs in order to give good flow conditions at the entrance and exit to the silencer. Often the major noise source is the plant room fan, but fan coil units, for example, introduce noise sources closer to the occupied space. A length of lined duct, between a ceiling space fan coil and the duct termination, may be adequate to deal with fan coils.

6.7.2 Plenums

Plenums are analysed by considering the inlet duct as a source of sound power into the plenum. The sound reflects within the plenum, as in a room, and a proportion of the sound energy passes into the outlet duct. The factors to be considered are then the dimensions, relative positions of the inlet and outlet and the absorption coefficient of the plenum lining.

For the plenum shown in Figure 6.5, it can be shown that the insertion loss is given by:

$$\mathrm{IL} = 10 \lg \left(A_{\mathrm{out}} \left(\frac{Q \cos \theta}{4 \pi r^2} + \frac{1 - \bar{\alpha}}{A \bar{\alpha}} \right) \right) \quad (6.4)$$

where IL is the insertion loss (dB), A_{out} is the outlet area of the plenum (m^2), Q is a directivity factor depending on the location of the inlet (normally taken as 4 for plenums, see section 7.3), θ is the angle between the slant distance (r) and the plane containing the axis of the inlet duct (see Figure 6.5) (degree), r is the slant distance from entry to exit (see Figure 6.5) (m), A is the total inside surface area minus the areas of the inlet and outlet (m^2) and $\bar{\alpha}$ is the average absorption coefficient of the lining.

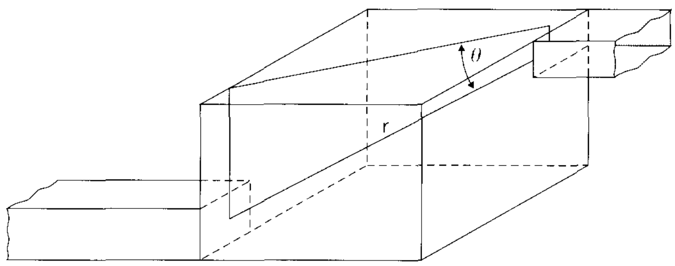

Figure 6.5 Schematic of a plenum chamber

Equation 6.4 assumes that the wavelength of the sound is small compared with the dimensions of the plenum. That is, it assumes that sound in the plenum behaves like sound in a room. (*Note*: there are similarities between equation 6.4 for a plenum and equation A7.1 for a room.)

Equation 6.4 gives best results when the areas of the inlet and outlet are small compared with the total surface area. The positioning of the plenum may affect its performance, as duct lengths into and out of the plenum may resonate with components in the noise.

6.8 Active silencers

Active silencers detect the noise travelling in the duct and generate an opposing noise, which is added in with the travelling noise in order to produce cancellation. They are most effective in the low frequencies, where passive silencers have limited performance.

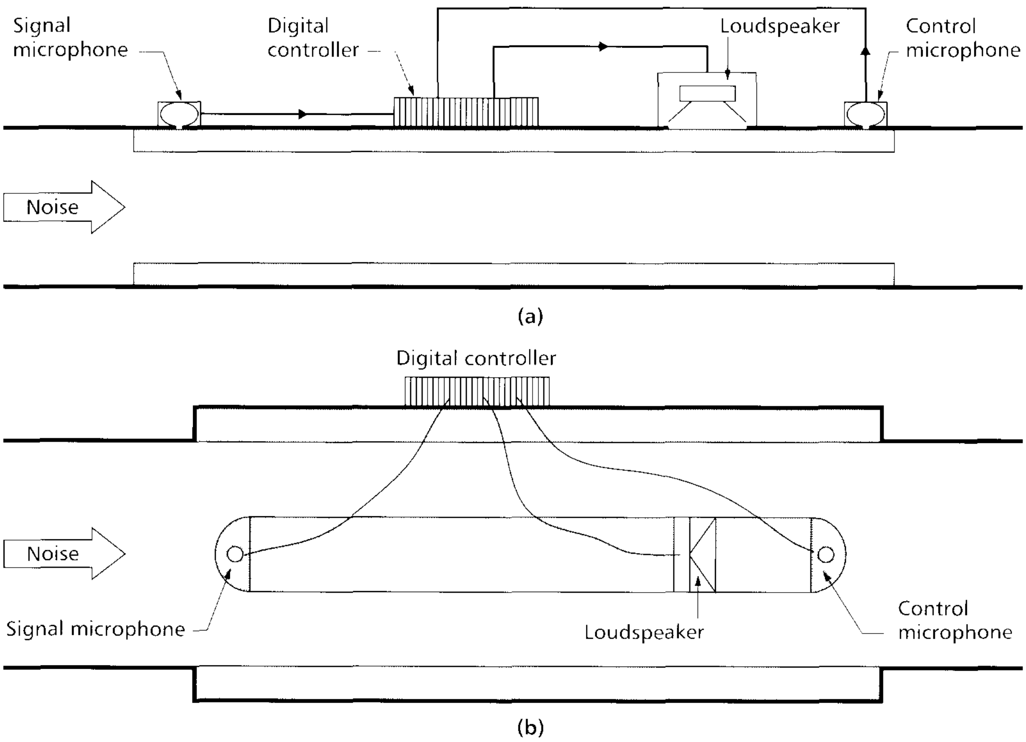

Figure 6.6 Active silencers; (a) mounted externally, (b) in a central pod

There are two main configurations of active silencer in which the active components are mounted either externally on the duct or in a central pod. The first type, illustrated in Figure 6.6(a), has some advantages for ease of retrofitting, whilst the second type, Figure 6.6(b), has technical advantages in the way in which the cancelling sound couples with the travelling sound. It also permits multiple units to be stacked to control noise in a large duct. Multiple units have been used, for example, to control noise breakout to atmosphere from air inlet or outlet openings. Active silencers have an application in natural ventilation, in order to give silenced, low pressure loss penetrations into the building.

Figure 6.6 shows how the upstream signal microphone picks up the travelling noise and sends it to a digital controller which outputs the cancelling noise to the loudspeaker. The downstream control microphone supplies performance information back to the controller to modify its parameters, and minimise the remaining downstream noise. Absorptive material round the perimeter and in the pod control high frequency noise, whilst the active components control lower frequency noise.

Active silencers have developed well beyond the laboratory demonstration and are now produced commercially by several companies, generally in a hybrid configuration incorporating both passive and active absorption, which can give wide-band noise control. The main advantage of an active silencer is that it gives good low frequency attenuation with lower pressure loss and in smaller space than an equivalent low frequency passive silencer. There are a number of factors such as air velocity, frequency range, duct dimensions etc., which need to be considered in the selection of an active silencer. Expert advice should be sought.

6.9 Use of fibrous sound absorbing materials in ducts

Although fibrous materials are excellent and inexpensive sound absorbers, there have been concerns over their use in ducts. Specifically, fibrous duct linings:

— may contribute to mould growth

— degrade with time

— erode from the surface and become carried in the air

— are difficult to clean.

In order to satisfy these concerns some general provisions that may be taken include the following:

— Fibrous linings should be kept at least 3 m away from wet sections as mould growth may occur if they become wet.

— Care should be taken in the installation, especially to seal raw edges.

— Linings are available with toughened surfaces, treated for mould control.

— In sensitive locations, the lining may be covered with a Mylar, or similar, facing although this reduces the absorption at high frequencies.

— If the lining might be damaged it should be protected with a perforated metal sheet having at least 25% open area, in order to maintain the absorption properties of the lining material.

— The material should be kept dry and undamaged prior to installation.

Unprotected fibrous material is sometimes used as a layer above suspended ceilings, for both noise control and thermal insulation. This results in a poor atmosphere for maintenance work above the ceiling and, if the ceiling tiles are moved, may result in fibres entering the space below.

6.10 Duct breakout noise

As duct materials are lightweight and thin, they transmit sound through the duct walls. This is known as breakout noise, which is a particular problem when long runs of duct pass over occupied space. Break out of noise leads to the noise in the duct reducing with distance, but breakout should never be used as a method of in-duct noise reduction unless the duct passes over non-sensitive spaces, such as storage areas. Break-in noise may also occur. For example, if the high noise level in a plant room breaks into a duct, the noise is then transmitted down the duct. The duct silencer for the main fan should be fitted to penetrate the plant room wall, so attenuating any noise which breaks into the duct within the plant room.

In estimating duct breakout, the procedure is as follows:

— determine the sound power in the duct

— determine the acoustic intensity in the duct

— obtain the sound transmission loss of the duct material

— determine the resulting external noise power.

The sound power in a duct is given in Appendix A1 as:

$$P = I A_d \tag{6.5}$$

where P is the sound power (W), I is the in-duct sound intensity (W·m^{-2}) and A_d is the duct cross sectional area (m^2).

However, breakout is associated with a gradual reduction in the internal sound intensity in the duct. This was considered as duct attenuation in section 6.2.

It can be shown[15] that the sound power radiated from a length of duct is given by:

$$P_{(\mathrm{breakout})} = P_{(\mathrm{duct})} - R + 10 \lg (S_d / A_d) \tag{6.6}$$

where $P_{(\mathrm{breakout})}$ is the sound power radiated from the duct (W), $P_{(\mathrm{duct})}$ is the sound power in the duct (W), R is the sound reduction index of the duct wall material, S_d is the surface area of the section of duct wall (i.e. duct perimeter × duct length) (m^2) and A_d is the cross sectional area of the duct (m^2).

Equation 6.6 has limited application for the following reasons:

- The in-duct power level is not constant, but reduces with distance down the duct as the sound breaks out.
- The term (S_d/A_d) does not have a limit and could be so large that the breakout sound power exceeds the in-duct sound power, which is an unrealistic situation. The equation should be limited to ducts lengths of 5–10 m or the maximum breakout sound power limited to 3 dB below the in-duct sound power level.
- The sound reduction index (R) is normally measured under standard reverberant laboratory conditions, which differ from the sound fields in ducts. However, measurements of R have been made in different types of ducts and these should be consulted for further information[16,17]. As an approximation, the R values can be taken as the values for the sheet material from which the duct is constructed.
- The ceiling void and ceiling attenuation influences transmission of casing breakout noise from ceiling units into the occupied space below[17]. Thus, the location of the ducts and units within the void may affect noise radiation into the occupied apace.

7 Room sound levels

A duct system noise calculation gives the noise levels in the duct immediately before the conditioned space. Both the effect of the termination and the propagation from the termination into the room to the occupiers must be taken into account. Duct terminations are considered in Appendix A2.5. This section deals with propagation in the room and with determination of room sound levels when the sound power issuing from the terminations is known.

7.1 Behaviour of sound in rooms

When a source of sound operates in a room, energy travels from the source to the room boundaries, where some is absorbed and some of it is reflected back into the room. There are a number of subsequent reflections before the sound is reduced to such a low level that, in effect, it no longer contributes to the total energy in the space. In a large absorbent and furnished room, particularly where the height is small compared with the other dimensions, the sound may be reflected by the furniture and absorbed by the floor and ceiling materials, so reducing the significance of the perimeter walls.

At any point in a room there are two contributions to the total sound:

- *direct sound*: that which comes directly from the source to the listener
- *reverberant sound*: that which has been reflected before it reaches the listener.

The balance between these contributions depends on the distance from the source and on the reflectiveness of the room perimeter, see Figure 7.1. This is considered further in Appendix A7.

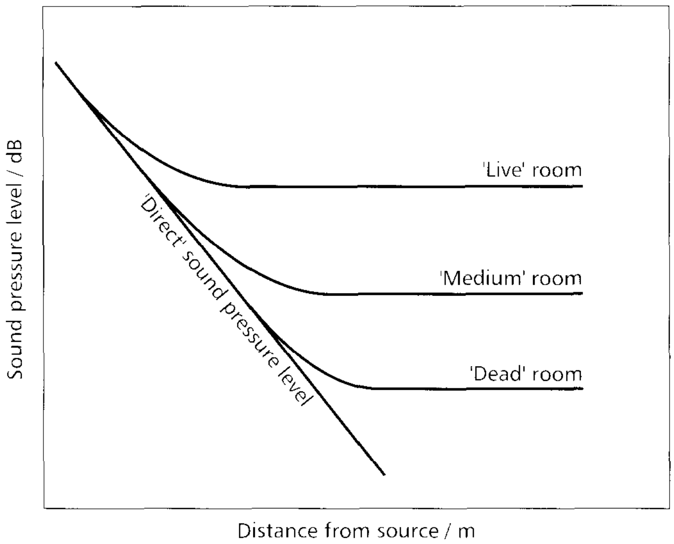

Figure 7.1 Variation of sound pressure level with distance from source

7.2 Determination of sound level at a receiver point

The conclusions from a series of measurements by Shultz[18] were that the relation between sound pressure level and sound power level in real rooms was of the form:

$$L_p = L_W - (10 \lg r) - (5 \lg V) - (3 \lg f) + 12 \qquad (7.1)$$

where L_p is the sound pressure level (dB), L_W is the sound power level (dB), r is the distance from a source (m), V is the volume of the room (m^2) and f is the frequency (Hz)*.

If there are a number of sources, their effects at a point must be added.

Equation 7.1, for a normally furnished room with regular proportions and acoustical characteristics which are between 'average' and 'medium-dead', as defined below, leads to the following equations.

For a room volume less than 430 m^3, for a point source of sound:

$$L_p = L_W + A - B \qquad (7.2)$$

* Although equation 7.1 appears to differ from equation A7.1[17] in Appendix A7, they are connected. The term (10 lg r) means that, in real rooms, the level of the direct sound falls as 3 dB per doubling of distance, rather than 6 dB as in equation A7.3 in Appendix A7. The terms (5 lg V) and (3 lg f) together embody the acoustical characteristics of the space, as does the term ($4/R_R$) in equation A7.2 (Appendix A7), since R_R depends on the surface areas in the room and their absorption coefficients, which vary with frequency. Work following up on Shultz has produced complex empirical equations by curve fitting a range of measurements[14], but other methods use simple equations with tabulated correction factors for both single and multiple sound sources.

where L_p is the sound pressure level at a specified distance (dB), L_W is the sound power level of the source (dB), A is a constant depending on the room volume and the sound frequency (dB) and B is a constant depending on the distance from the source (dB).

Values of A and B are given in Tables 7.1 and 7.2, which are derived from information contained in the ASHRAE *HVAC Applications Handbook*[17]. It will be seen that A incorporates the volume and frequency terms of the Schultz equation (equation 7.1) whilst B incorporates the distance term and falls off at 3 dB per doubling of distance.

For room volumes from 430 m³ to 4250 m³ the influence of room volume is less and the ASHRAE recommendation[17] is:

$$L_p = L_W - C - 5 \tag{7.3}$$

where C is a constant depending on the sound frequency and the distance from the source (dB). Values of C are given in Table 7.3.

In many rooms there is an array of ceiling sources, each one of which has an associated volume for which it is the major noise source. One way of proceeding is to calculate the effect of each source at a reception point and add these levels. However, ASHRAE[17] gives a simplified procedure for determining the noise at a reception height of 1.5 m, incorporating the height of the ceiling and the floor area served by each diffuser:

$$L_{p1.5} = L_{W(s)} - D \tag{7.4}$$

where $L_{p1.5}$ is the sound pressure level 1.5 m above the floor (dB), $L_{W(s)}$ is the sound power level of a single diffuser (dB) and D is a constant depending on the sound frequency, the floor-to-ceiling height and the floor area served by a single diffuser (dB). Values of D are given in Table 7.4.

Equations 7.1, 7.2, 7.3 and 7.4 are valid only for office types of rooms of 'average' to 'medium-dead' (see below) acoustical characteristics. Large reverberant spaces such as sports halls, where there are long unobstructed sound paths, may be analysed using the approach described in Appendix A7.

The accuracy of the equations 7.2, 7.3 and 7.4 is 2 to 5 dB.

For the purposes of the above equations, the characteristics of room spaces are defined as follows:

— *Live*: hard surfaced rooms with no furnishing or absorbent material. These rooms echo or 'ring' when stimulated by a source. Typical reverberation time greater than 2 s.

— *Medium-live*: hard surfaced rooms with no specific attempts at adding absorption other than through the occupants and their furniture. Typical reverberation time around 1.5 s.

— *Average*: rooms with suspended ceilings or soft furnishings, carpeted and with drapes, e.g. typical office spaces. Typical reverberation time 0.7–1 s.

— *Medium-dead*: rooms with suspended ceilings, carpets and soft furnishings, e.g. executive offices. Typical reverberation time around 0.5 s.

— *Dead*: rooms which have been designed to be sound absorbent. Typical reverberation time less than 0.3 s.

Table 7.1 Values of constant A for equation 7.2

Room volume* / m³	Value of A / dB for stated octave band / Hz						
	63	125	250	500	1000	2000	4000
42	4	3	2	1	0	−1	−2
71	3	2	1	0	−1	−2	−3
113	2	1	0	−1	−2	−3	−4
170	1	0	−1	−2	−3	−4	−5
283	0	−1	−2	−3	−4	−5	−6
425	−1	−2	−3	−4	−5	−6	−7

* Irregular values of volume arise from metric conversion from cubic feet

Table 7.2 Values for constant B for equation 7.2

Distance from point source / m	Value of B / dB
0.9	5
1.2	6
1.5	7
1.8	8
2.4	9
3.0	10
4.0	11
4.9	12
6.1	13

Table 7.3 Values for constant C for equation 7.3

Distance from source / m	Value of C / dB for stated octave band / Hz						
	63	125	250	500	1000	2000	4000
0.9	5	5	6	6	6	7	10
1.2	6	7	7	7	8	9	12
1.5	7	8	8	8	9	11	14
1.8	8	9	9	9	10	12	16
2.4	9	10	10	11	12	14	18
3.0	10	11	12	12	13	16	20
4.0	11	12	13	13	15	18	22
4.9	12	13	14	15	16	19	24
6.1	13	15	15	16	17	20	26
7.6	14	16	16	17	19	22	28
9.8	15	17	17	18	20	23	30

Table 7.4 Values of constant D for equation 7.4

Floor-to-ceiling height / m	Floor area / m²	Value of D / dB for stated octave band / Hz						
		63	125	250	500	1000	2000	4000
2.4–2.7	9.3–14	2	3	4	5	6	7	8
	18–23	3	4	5	6	7	8	9
3.0–3.7	14–18.5	4	5	6	7	8	9	10
	23–28	5	6	7	8	9	10	11
4.3–4.9	23–28	7	8	9	10	11	12	13
	32.5–37	8	9	10	11	12	13	14

7.3 Source directivity

When the sound power of the duct termination has been calculated, a further step is required to assess whether the room influences the radiation from the termination. If the adjacent surfaces are acoustically non-absorbing, the influence depends on the location of the termination within the room, and affects only the direct sound. The reverberant sound is not changed by the location. A general consideration for sources is as follows.

If the source is located in free space, a situation which could be approximated by a duct projecting into the centre of a room, the energy from the duct outlet spreads uniformly and, for the direct sound, the relevant relation for intensity is given by (see Appendix A1):

$$I = P / 4\pi r^2 \qquad (7.5)$$

where I is the sound intensity ($\mathrm{W \cdot m^{-2}}$), P is the sound power (W) and r is the distance from the source (m).

Figure 7.2 shows a further three basic locations for an outlet. If the termination is in the centre of a reflecting surface (position A), the sound propagates into half space due to reflections from the wall, so that, at any point, there is twice as much direct energy as before. This is an apparent doubling of the directly radiated source power, caused by reflections from one surface before the sound first reaches a listener. The result is a 3 dB increase in direct sound level.

If the termination is at one edge of the room (Figure 7.2, position B), where two reflecting surfaces meet, propagation is into a quarter space. There is an apparent quadrupling of the directly radiated sound caused by reflections from two surfaces before the sound reaches a listener, leading to a 6 dB increase in the direct sound over the source located in free space.

If the termination is in a corner (Figure 7.2, position C), where three reflecting surfaces meet, propagation is into one-eighth space. This is an apparent eight-fold increase in the directly radiated source power, caused by reflections from three surfaces before the sound reaches a listener. This leads to a 9 dB increase in the direct sound over the source located in space.

These initial reflections from surfaces adjacent to the source do not affect the reverberant sound levels.

A general relation for effective sound power of the source is:

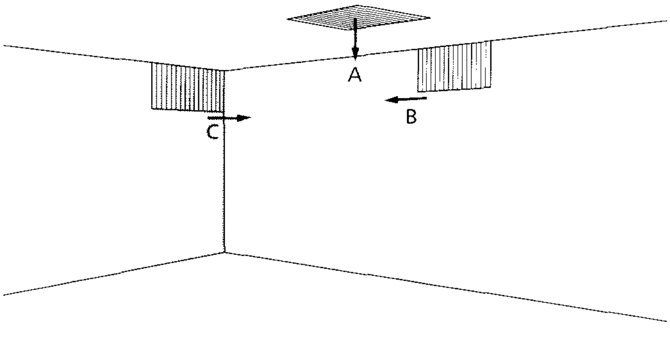

Figure 7.2 Outlet locations

$$L_W = (10 \lg P) + (10 \lg Q) + 120 \qquad (7.6)$$

where L_W is the sound power level (dB), P is the sound power (W) and Q is a directivity factor. The directivity factor indicates how much more energy is received due to reflection of the sound by the adjacent surfaces. Under the conditions of reflective surfaces, as in Figure 7.2, Q takes values of 1 for a source in free space, and 2, 4 or 8 for positions A, B and C respectively.

If the adjacent surface is very absorbing, e.g. a suspended ceiling at higher frequencies, the sound will not be reflected from it and there will not be the simple theoretical interaction described above. Use of these directivity concepts is approximate. (There are parallels with reflection of light from reflective or dull adjacent surfaces).

Directivity is also an inherent property of some noise sources, which radiate preferentially in certain directions, irrespective of their location in a room. If a source is inherently highly directional, it will not be influenced by those adjacent surfaces which do not intercept its radiation. In general, when the dimensions of a source are large compared with the wavelength of the radiated sound, it becomes directional. See *Noise control in building services*[15] for further information.

7.4 Sound transmission between rooms

There are a number of paths by which sound may transmit between rooms, as shown in Figure 7.3. These are as follows:

— *Directly through the wall*: the mechanism is that sound impinging on the wall in the source room causes the wall to vibrate; for example, a sound level of 94 dB gives an oscillating pressure of 1 Pa on the wall. The vibration of the wall then causes it to act as a radiator of sound on the other side, into the receiving room.

— *Through gaps between the rooms*: often due to insufficient sealing of joints or penetrations. Gaps hidden by lightweight components such as skirtings, wall coverings, dry lined walls and electrical sockets can be significant and difficult to locate at

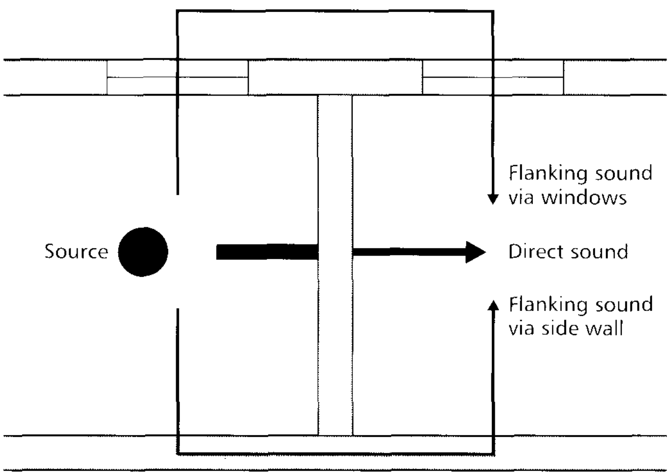

Figure 7.3 Sound transmission paths between rooms

a later stage. These gaps may cause a big reduction in the insulation of the wall.

— *By various flanking paths*: these are indirect paths due to vibration of room surfaces other than the partition wall.

The acoustical conditions in the receiving room influence the sound level within it. If the room is very absorbing, the level is lower than if it is very reverberant.

Problems of sound transmission between rooms may arise from poor design or installation of partitions and ceilings, which is not directly the responsibility of the building services engineer. However, a flanking path between rooms via ducts is the responsibility of the building services engineer. It should be noted that Building Regulations Part E: Resistance to the passage of sound, which previously covered only domestic housing, is to be extended to all residential buildings*. This includes hotels, student halls of residence etc[19].

Inadequate attenuation by plant room walls may also be a problem. The building services engineer must provide expected plant room noise levels for the architect or building designer. The attenuation required depends on the total sound power of the machinery in the plant room, although in a crowded plant room, if the nearest plant to a sensitive wall is noisy, this plant may be the main influence on the noise at the wall. Installation practices are also important, see section 4.

There are a number of ways of expressing the room-to-room airborne sound insulation, as follows.

7.4.1 Level difference

The level difference is simply the difference, as measured on site, of the average levels in the source room and in the receiving room, i.e:

$$D = L_1 - L_2 \qquad (7.7)$$

* At the time of writing (December 2001), the revision is presently at the consultation stage.

where D is level difference (dB), L_1 is the average sound level in the source room (dB) and L_2 is the average sound level in the receiving room (dB).

The level in the receiving room depends on the properties of the partition wall, the flanking paths and on the reverberant build-up of sound in the room. Furnishing a receiving room in heavily absorbent material will tend to increase the measured level difference by decreasing the reverberant build-up of sound in the room.

7.4.2 Sound reduction index

The properties of the wall itself are given by the sound reduction index (R) which is measured by standardised procedures in a test room[20]. The reverberation time in the receiving room is standardised to $T_0 = 0.5$ s, in order to allow for the effects in different rooms. A receiving room reverberation time, T, of 1 second will cause the level difference to be 3 dB higher than in a room having $T = 0.5$ s, i.e. $10 \lg (T/0.5) = 3$ dB when $T = 1$ s.

The sound reduction index is a property of the separating wall material for samples measured in a laboratory according to current standards[20] and this is the quantity which may be quoted in manufacturers' literature. Some examples of sound reduction index are given in Table 7.5, where the values are shown in decibels at octave band frequencies from 63 Hz to 4000 Hz. Although most laboratory measurements have traditionally been made in third octave bands from 100 Hz to 3150 Hz, as required in the older standards, measurement procedures now recommend extending the range from 50 Hz to 5000 Hz third octave bands. This recognises that the limited band of measurements from 100 Hz to 3150 Hz does not provide sufficient information for a full assessment of subjective effects of noise.

Sound reduction index and level difference are related by the reverberation time (see Appendix A7, equation A7.7), giving:

$$R = L_1 - L_2 + 10 \lg (T) + 10 \lg (S / 0.16\, V) \qquad (7.8)$$

Table 7.5 Typical values of sound reduction index

Material	Sound reduction index, R, for stated octave band / Hz						
	63†	125	250	500	1000	2000	4000
6 mm glass	15	18	23	30	35	27	32
Sealed double glazed window (6 mm outer, 12 mm air gap, 6 mm inner)	18	20	18	28	38	34	38
Separate window panes (5 mm outer, 150 mm air gap, 4 mm inner)	20	26	34	44	54	53	51
Acoustic double glazing (10 mm outer, 200 mm air gap, 6 mm inner)	26	37	46	45	47	57	64
Lightweight block (100 mm thick) e.g. 'Thermalite'	20	27	32	37	40	41	45
Galvanised steel sheet:							
— 22 gauge (0.55 mm thick)	3	8	14	20	23	26	27
— 16 gauge (1.6 mm thick)	9	14	21	27	32	37	43
200 mm reinforced concrete	36	42	41	50	57	60	65
Stud partition (9 mm plasterboard on 50 mm × 100 mm studs at 400 mm centres, 12 mm plaster both sides)	20	25	28	34	47	39	50

† *Some of the values at 63 Hz are estimated*

where R is the sound reduction index, L_1 and L_2 are the average sound levels in the source room and receiving room respectively (dB), T is the reverberation time (s), S is the total surface area of the room (m^2) and V is the room volume (m^3).

However, the validity of equation 7.8 reduces for short reverberation times.

7.4.3 Weighted sound reduction index

A single number representation of sound reduction is given by the weighted sound reduction index, R_w. This is obtained by comparing the measured attenuation–frequency curve with standardised curves, moderated by certain conditions. The value of the curve at 500 Hz then gives R_w[21].

There are differences between sound transmission under laboratory conditions and sound transmission in field conditions. The field measurement is often influenced by factors that are controlled in the laboratory, particularly flanking transmission and leakage through gaps. Table 7.5 gives typical values of sound reduction indices. Manufacturers' data should be consulted for standard prefabricated office partitions.

7.5 Privacy and cross talk

Privacy describes the ability to talk within one space without being overheard in another space. In an open plan office, privacy is related to background noise and distance. Privacy between adjacent private offices depends on the level of the sound transmitted between the spaces and the background noise in the 'listening' room. The level should be below the criterion level of the second space. Transmission is determined by the efficiency of the dividing wall and suspended ceiling. Office partition walls should preferably go up to the structural ceiling in order to prevent leakage from one office into the ceiling void and then down into the adjoining office. Where there is an unbroken space in the ceiling void, inclusion of absorption may help to reduce sound transmission.

However, the best acoustical design can be undone by cross-talk between rooms through common ducts. Figure 7.4 illustrates how sound enters a duct and travels to an adjoining room. Prediction is by estimating the sound pressure at the termination leading into the duct. This can be converted to sound power into the duct using equation A1.13, see Appendix A1, which may then be dealt with as described in section 6 and Appendix A4 in order to determine the sound power that enters the second room. The sound level of the intruding speech should be 5–10 dB below the sound level in the second room. Alternatively, the sound power from the first room in the duct should be 5–10 dB below the HVAC sound transmitted into the duct that feeds the second room.

Approximate values for speech sound powers for loud voices are given in Table 7.6[15].

These figures are used to calculate the direct and reverberant sound at the duct termination responsible for the cross-talk. Direct sound is given by (see Appendix A1):

$$L_{p(direct)} = L_W - (20 \lg r) - 11 \tag{7.9}$$

where $L_{p(direct)}$ is the direct sound pressure level (dB), L_W is the sound power level (dB) and r is the distance to the source (m).

Reverberant sound is obtained from equation 7.16 (see section 7.6) by assuming a reverberation time, T, of 0.5 s, typical of offices.

Hence:

$$L_{p(reverb)} = L_W - (10 \lg V) + 11 \tag{7.10}$$

where $L_{p(reverb)}$ is the reverberant sound pressure level (dB), and V is room volume (m^3).

Table 7.6 Sound powers for loud voices

Frequency / Hz	Sound power level, L_W / dB
63	69
125	72
250	77
500	80
1000	80
2000	75
4000	76

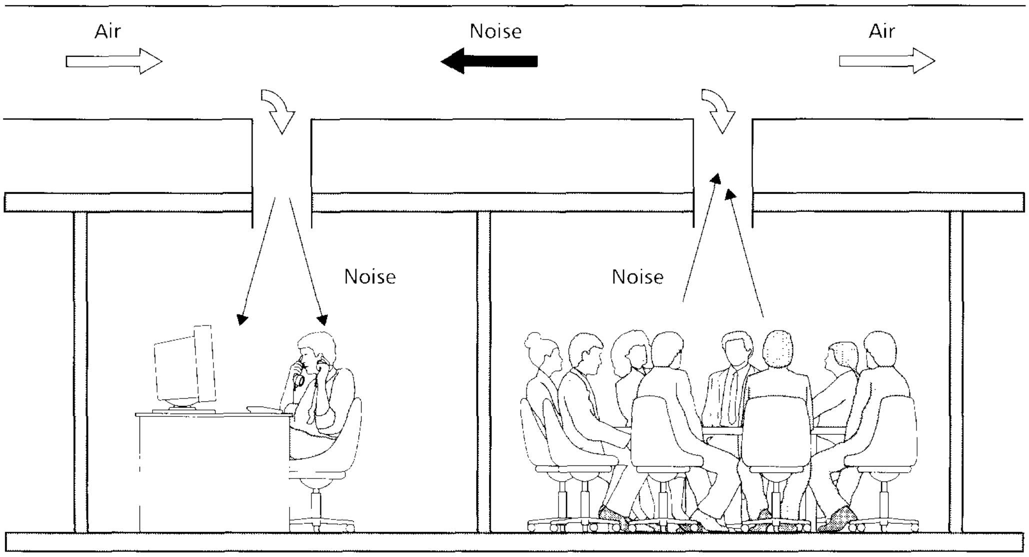

Figure 7.4 Cross talk between rooms

The sound powers into the duct are then given by:

$$L_{W(direct)} = L_{p(direct)} + 10 \lg A_d \quad (7.11)$$

$$L_{W(reverb)} = L_{p(reverb)} + (10 \lg A_d) - 6 \quad (7.12)$$

where $L_{W(direct)}$ is the direct sound power level (dB), $L_{W(reverb)}$ is the reverberant sound power level (dB) and A_d is the cross sectional area of the duct (m^2).

The –6 term in the reverberant power into the duct in equation 7.12 arises because of the random directions of arrival of the reverberant sound.

The total power into the duct is then the decibel summation of the two powers as described in Appendix A1 and the calculation proceeds as in Appendix A4.

Control of cross talk is achieved by lining the duct, as described in section 6, by splitting the duct into two or more runs, so that adjacent rooms are fed from different lines or by using 'cross talk silencers' in the duct between rooms.

7.6 Sound in large reverberant spaces

These spaces are typically assembly halls with hard surfaces, sports halls, swimming pool halls and churches. Multi-purpose halls may be used, as required, for sports, music performances and speech, but the design criteria will be to satisfy the most critical use. There will normally be an amplification system when used for speech. When used for sports or as a swimming pool, the requirement is to keep down the level of reverberant noise by use of absorbing material.

In these large spaces, sound propagation and reverberation is described by equation A7.1, see Appendix A7, in which the direct and reverberant sound can be separated. If the air supply is from a high level in the hall, the reverberant sound is likely to predominate at the level of the occupants and equation A7.2 gives:

$$L_{pR} = L_W + 10 \lg (4 / R_R) \quad (7.13)$$

where L_{pR} is the reverberant sound pressure level (dB), L_W is the sound power level (dB) and R_R is the room constant.

The room constant is defined as (see Appendix A7):

$$R_R = \frac{S \bar{\alpha}}{1 - \bar{\alpha}} \quad (7.14)$$

where R_R is the room constant (m^2), S is the total room surface area (m^2) and $\bar{\alpha}$ is the average absorption coefficient for the room surfaces.

But from Appendix A7, equation A7.8 gives:

$$T = 0.16\, V / R_R \quad (7.15)$$

where T is the reverberation time (s) and V is the room volume (m^3).

Equations 7.13 and 7.15 lead to the reverberant sound level as:

$$L_p = L_W + (10 \lg T) - (10 \lg V) + 14 \quad (7.16)$$

Therefore, the reverberant sound level produced by the ventilation system is estimated either by equation 7.13 if the room constant R_R is first calculated from knowledge of the room surfaces, or by equation 7.16 from the reverberation time. The information for these predictions should be obtained from the acoustics consultant.

8 Transmission of noise to and from the outside

Transmission of noise into or out of buildings has important implications for the building services engineer, whether the noise is from building services within the building or because the building services themselves alter the sound insulation of a building, e.g. by creating a new aperture and thereby admitting external noise. Building services noise may also be transmitted to another area of the same building or to an adjacent building. Some building services equipment is located outside and its noise may affect the building it serves or neighbouring buildings. Reflection of noise from adjacent building surfaces should be taken into account, see section 7.3 and *Noise control in building services*[15].

8.1 Transmission of noise to the outside and to other rooms

Noise from building services travels to the outside in a number of ways. It might be created outside from a roof-mounted air handling unit or condenser, or inside, such as from a fan or boiler plant. It then travels through louvres, ducts or the fabric of the building to the outside. It will be necessary to design the system to ensure that specified criteria are met outside the building. It is usual for the local authority to stipulate the criterion in such situations, to protect public open space or limit noise into nearby buildings.

Estimation of noise passing through the fabric of the building to the outside or to other parts of the building will generally require specialist knowledge of the sound insulation of materials, which is beyond the scope of this Guide, although general principles are given in section 7.4. If the building services engineer concludes that estimation is required, this should be brought to the attention of the project manager.

8.2 Transmission of external noise to the inside

For the building services engineer, the transmission of noise from the outside into a room should be considered from two distinct aspects. The first is that considered in section 8.1 above, where noise from the plant, occurring or created outside a building, travels into the building through windows, the roof or any other element, including the services themselves. Generally, but not always, the building services engineer will have no control over the sound insulation of the building against external noise and it will be necessary to ensure that criteria are met solely by appropriate design of the plant. The second aspect is that of noise from other sources, such as road or rail traffic, aircraft or industrial noise, entering a building via its own services, an inlet or exhaust duct or perhaps an extract fan. The opening for the fan or duct, or any gap around a duct, will have little insulation against noise from the outside and could seriously compromise the sound insulation of a building. Careful thought must be given to the sound insulation of the combined system.

8.3 Naturally ventilated buildings

Natural ventilation generally requires far more apertures in the building fabric with potential for external noise to enter the building. The low pressure drops required for natural ventilation imply the use of large and, probably, unattenuated ducts. These allow noise from the inside to get out and noise from the outside to get in. Generally, natural ventilation is difficult to apply to those buildings that require special levels of sound insulation against external noises, since apertures for intake and exhaust are also sites of acoustical weakness. Care must be taken in:

— *choice of the location of the apertures*: this may also be influenced by concerns over the quality of the incoming air

— *estimation of the noise leakage through the apertures*: a silencer may be required; the relatively low pressure loss of active silencers makes them suitable for noise control in natural ventilation.

9 Criteria for noise in HVAC systems

9.1 Objective

The objective of an acoustical criterion is to guide the design of an occupied space, so that it meets a specified acoustical standard. There are two components to be considered in complying with acoustical criteria:

— limiting the noise emission into the space

— designing the acoustics of the space.

The dimensions, perimeter materials and furnishings determine the acoustics of the space. These are not normally under the control of the building services designer, who has to make assumptions about typical spaces, as in section 7.2. In addition to HVAC noise, an office has activity noise related to the work in progress, e.g. office machines, telephones, conversation. The HVAC designer has responsibility for the HVAC noise, which must comply with the agreed criteria at specified locations. Compliance with acoustical criteria for HVAC is normally considered in the absence of activity noise.

Naturally ventilated buildings must also fulfil the design criteria, although the noise sources may be external to the building and not under the control of the designer.

In developing a criterion, the needs of work efficiency have to be balanced against costs of silencing. The term 'work efficiency' includes worker health, comfort, concentration, absence of errors etc., when working on their own and additionally includes communication with colleagues, when working with others. Communication may be by either telephone or direct speech over a neighbouring area.

9.2 Approaches

At the present time there are conflicting approaches to criteria. The widespread use of A-weighted decibels for environmental noise assessments is influencing room noise criteria, especially in the countries of mainland Europe. This is attractive for its simplicity and the ability it gives to compare noises, although it is difficult to relate, say, a 40 dBA steady services noise with a continuously fluctuating, intruding traffic noise, which averages as 40 dBA, but may have much higher peak levels. Reliance on the A-weighting may be unsafe, as a wide range of different noises could have the same A-weighted value. Some will be subjectively acceptable and some will not be. In particular, it has been found that the A-weighted measure is inadequate for noises which have high levels at low frequencies, as may occur with services noise[22]. Countries which have adopted the A-weighting for general use are now placing additional restrictions on low frequency noise, summarised by Mirowska[23], covering the range from about 10 Hz to 200 Hz. This is mainly to protect people in their homes, although Denmark includes a low frequency restriction for noise in offices at a level about 10 dB higher than for homes[24].

9.3 ASHRAE approach

The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) treats differences in noise spectra by considering the noise 'quality', as determined by the relative values of three frequency ranges (low, medium and high) within the noise[17]. The frequencies considered are as follows:

— *low frequencies*: 16 Hz, 32 Hz and 63 Hz octave bands

— *mid frequencies*: 125 Hz, 250 Hz and 500 Hz octave bands

— *high frequencies*: 1 kHz, 2 kHz and 4 kHz octave bands.

There is normally little problem from duct-borne high frequency noise, although it may be generated by airflow over grilles. Middle-to-low frequencies include fan blade noise from a central fan or noise from a nearby fan coil unit. Low frequency noise often originates in a central fan and is the most difficult to control. It is recognised as an annoying rumble, but is not adequately recorded by an A-weighted measurement.

9.4 Review

The historical difficulty of developing a satisfactory criterion is illustrated by the large number of criteria that have been developed. The ones used most are:

— dBA level: first published in 1936

— NC (noise criterion): first published 1957

— NR (noise rating): first published 1962

— PNC (preferred noise criterion) first published 1971

— RC (room criterion) Mark I: first published 1982

— NCB (balanced noise criterion): first published 1988

— RC (room criterion) Mark II: first published 1995

Descriptions of these criteria are given in Appendix A8.

NR and NC are currently used in the UK. The RC, NC and NCB are used in the USA, but RC Mark II is ASHRAE's current recommendation. PNC is rarely used. The main criteria (NR, NC, RC, NCB) are very similar in the frequency range 125–2000 Hz and there is little to choose between them. There is some divergence at higher frequencies and considerable divergence at lower frequencies. For example, NR is 19 dB more lenient at 31.5 Hz than RC.

The trend in the development of criteria over the past 30 years has been to extend criteria into the lower frequencies, whilst also placing greater limitations on the levels permitted at these low frequencies, as it has become recognised that they may have a particularly disturbing effect[22]. For example, comparing the 35 rating at 31.5 Hz, NR35 is at 79 dB, NC35 (originally defined only down to 63 Hz, but commonly extended to 31.5 Hz) is at 68 dB (11 dB lower than NR), the RC is at 60 dB (19 dB lower than NR). However, NCB35 is at 71 dB, (8 dB lower than NR). The NCB assessment includes noise from all sources in the office, not just HVAC, so that each source, including HVAC systems, office machines, human activity etc., must be lower than the criterion limit.

The NR is an old criterion, which is satisfactory at low frequencies only when those frequencies do not occur. It has the weakness that it will permit high, potentially disturbing, low frequency levels to slip beneath its envelope. The NR is not satisfactory in those circumstances where the HVAC noise contains relatively high levels at low frequencies. ASHRAE approves the use of NC only in non-critical applications, but recommends RC Mark II for offices. Whilst it may be felt that the RC is very stringent in comparison with NR, it must be remembered that:

— a number of criteria for comfort in domestic premises place limitations on the levels of low frequency noise[23,24]

— there is a greater likelihood of disturbance by fluctuating levels at low frequencies[25]

— a sound level meter measurement averages the fluctuations and so does not indicate the peak levels in the fluctuations.

9.5 Criteria for design and commissioning

A design criterion must specify its requirements on a frequency basis, normally at octave bands. However, there is a problem with availability of design data, since it is only recently that 63 Hz data have become more readily available, whilst there are very few at 31.5 Hz and none at 16 Hz. The dimensions of test laboratories in relation to the wavelength determine the frequency limit and it is useful that new laboratories have been built with dimensions that give reliable readings down to 31.5 Hz. Some criteria, although used in design, were developed for troubleshooting, where the full range of octave band measurements are made on site. At the present time, the engineer is constrained to design with data down to 63 Hz and ensure that:

— the fan is used on an efficient part of its characteristic, in order to avoid low frequency instabilities

— the ductwork and airflows are designed to prevent low frequency turbulence

— fans, pumps and similar equipment are vibration isolated from the structure.

Since commissioning checks whether the design criteria have been met, the same criteria must be used for both design and commissioning.

10 Noise prediction

10.1 System noise

System noise prediction follows a simple logical process, but this is sometimes lost in the complexities of real systems. The prediction process for the HVAC system is as follows:

(1) Obtain the noise power of the source in octave frequency bands from manufacturers' data.

(2) Determine the successive effects of system components on the noise as it propagates in the duct, adding the effects, which may be negative (noise reduction) or positive (noise regeneration). Data on component effects should be provided by manufacturers. The end result is the sound powers at the duct terminations, see Appendix A2, section A2.5.

(3) Determine breakout noise from ducts and central devices etc. above a room, see section 6.10 and manufacturers' data.

(4) Determine the total sound power input to the room.

(5) Finally obtain the sound level at the occupant, see section 7.2.

These steps are to be carried out at all frequencies required for the criterion used.

10.2 Noise to atmosphere

This is most likely to occur from:

- a fan intake
- ventilation louvres in a plant room
- breakout through plant room roof or walls.

For the fan intake, the sound power at the opening is determined from the fan sound power and the attenuation that occurs between the fan and intake. Propagation effects to the outside are then included as in Appendix A1, including directional radiation as appropriate.

The louvres in a plant room are treated similarly, but first the noise level in the plant room is estimated and this is assumed to be the level at the inside of the louvres. Louvre attenuation, as defined by the manufacturer, is then subtracted and the resulting propagation predicted.

Breakout through walls is treated similarly to louvres.

Directivity of the sources should be included.

These predictions are approximate and a more detailed analysis is given in BS EN 12354-4[26].

11 Vibration problems and control

11.1 Introduction

Some noise problems, which appear to be of airborne origin, actually originate in structure-borne vibration from poorly isolated machinery and services. Excessive vibration threatens the stability and service life of structures, may interfere with proper functioning of plant and equipment, will shorten (or, in extreme cases, destroy) plant working life, and will also interfere with human comfort. Even allowing for the most ideal design/selection/location conditions likely to be encountered by the building services engineer, vibration control may still be necessary for all of the plant that has been discussed earlier, e.g. boilers, chillers, air handling units, condensers, fans, compressors, generators, lift machinery, cooling towers, pumps, pipes, ducts etc.

The best form of vibration control is avoidance, by careful location of plant rooms and selection and location of low vibration equipment within them such that vibration does not become a concern. Some basic control measures are frequently included within packaged equipment, e.g. rubber bushes, but additional, external vibration control is often required. Vibration isolation, or control, normally refers to the reduction of vibration input through the mounting points of the plant to the building. However, the same mounting system works in reverse and isolates the plant from vibration of the building.

On a large scale, whole buildings are vibration isolated from their foundations in order to reduce problems encountered from building vibration originating in external sources, e.g. underground railways, road traffic, earthquake, ordnance shock etc., and the building services engineer may be involved in these problems, if only to assist in ascertaining the fragility level that services plant and associated control systems can withstand. Here, fragility level is defined as the maximum shock in units of acceleration due to gravity which a piece of equipment can withstand without suffering damage sufficient to cause it to become inoperable. For example, a fragility level of '3 g' means that the equipment can withstand an acceleration of about 30 $m \cdot s^{-2}$. The design for earthquake and ordnance shock is beyond the scope of this Guide but a useful reference is *A Practical Guide to Seismic Restraint*[27].

The two principal divisions of vibration control can be considered as:

- *architectural/structural*: floating floor systems, building isolation bearings, seismic restraints
- *mechanical*: sprung inertia bases, pads, elastomeric mounts, helical spring pedestal mounts, helical spring/elastomeric hangers, pipe/duct flexible connectors.

The building services engineer may be involved in the design of a floating floor for a plant room, but mounting whole buildings on springs is very specialised and should be left to others.

In this Guide, the word 'spring' is generally used in a very wide sense to describe a range of components which compress under load, including pad materials, elastomeric blocks, elastomers-in-shear arrangements, helical steel springs, pneumatic springs, and other arrangements incorporating hydraulic and mechanical damping.

11.2 Fundamentals of vibration and vibration control

11.2.1 Acceleration, velocity, displacement and frequency

Vibratory force has four convenient physical quantities:

- displacement, x (mm)
- velocity, v ($mm \cdot s^{-1}$)
- acceleration, a ($mm \cdot s^{-2}$)
- frequency, f (Hz).

(In practical vibration isolation work, it is usually, more convenient to use millimetres for the unit of length than meters.)

At a given frequency, f, the quantities are related by the following equations:

$$v = 2\,\pi f x \qquad (11.1)$$

$$a = 2\,\pi f v \qquad (11.2)$$

Then:

$$a = 4\,\pi^2 f^2 x \qquad (11.3)$$

Figure 11.1 and the associated Table 11.1 show the relationship, using a simple sine wave for illustration. The symbols X, V and A represent the maximum values of

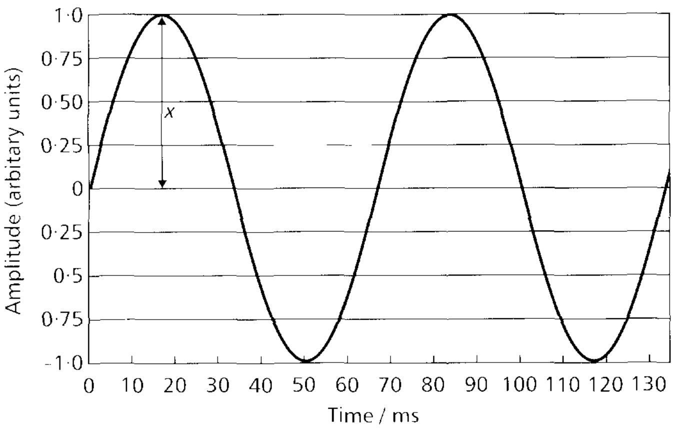

Figure 11.1 Vibration quantities

Table 11.1 Relation between vibration quantities

Quantity	Angular relationship	Maximum value at frequency 14.8 Hz
Displacement, x	$x = X \sin(2\pi f t)$	$X = 1$ mm
Velocity, v	$v = V \cos(2\pi f t)$	$V = 2\pi f X = 93$ mm·s^{-1}
Acceleration, a	$a = -A \sin(2\pi f t)$	$A = 2\pi f V = 8443$ mm·s^{-2}

displacement, velocity and acceleration, respectively. Two complete cycles of the wave occur in 135 ms, giving a frequency of 14.8 Hz. The amplitude scale, which has a maximum of 1.0, could be displacement, velocity or acceleration, but for the purposes of illustration it may be taken to represent a displacement of 1 mm, as in Table 11.1. Given the frequency and any one attribute, values of the remaining ones can be calculated for that frequency, see Appendix A6.1.

The acceleration is nearly 1 g (= 9.81 m·s^{-2}). Excessive vibration has an adverse effect, for which there is some objective data, on structures and on service or process equipment. The frequency is important in assessing such an effect. An acceleration of, say, 1 g at 3 Hz (typical earthquake condition) is a totally different proposition to the same 1 g at 30 Hz, which might be encountered with ordnance or even common mechanical shock problems. The effect of vibrations on human comfort is more subjective, see CIBSE Guide A section 1[28].

11.2.2 Natural frequency, static deflection, disturbing frequency, damping, vibration isolation efficiency

Natural frequency (f_n) is the constant frequency at which an object vibrates when set into motion and left to vibrate freely, e.g. when it is struck. It can be seen physically that, if the mass suspended on a spring is increased, the resonant frequency is lowered, whilst if the stiffness of the suspension is increased, the frequency increases.

Since the deflection under load depends on the stiffness and the mass, a simple prediction formula for resonant frequency is obtained in terms of the length, d, which the loaded spring deflects:

$$f_n \approx \frac{15.8}{\sqrt{d}} \qquad (11.4)$$

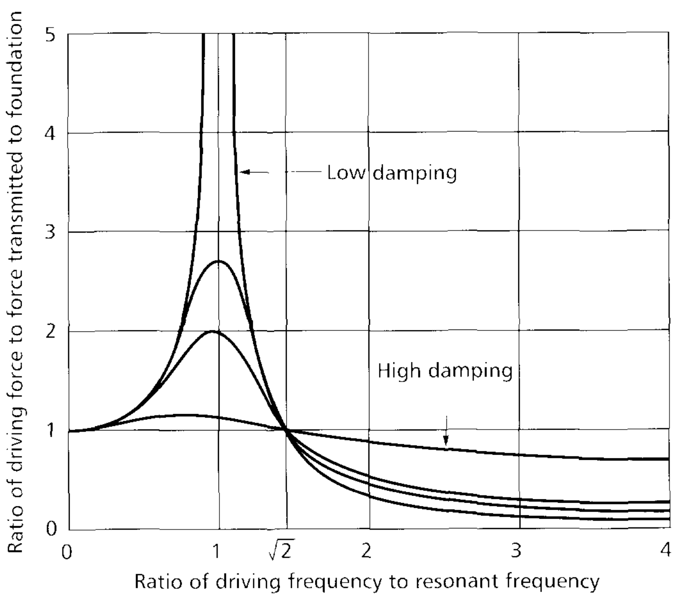

Figure 11.2 Vibration transmissibility

where f_n is the natural frequency (Hz) and d is the static vertical deflection of an isolated load on its springs (mm).

In practical vibration isolation, the spring stiffness is expressed as the spring rate (kg·mm^{-1}), which describes the load in kilogrammes required to deflect a spring by 1 mm.

Figure 11.2 shows how the displacement of the system changes if it is driven by a force that varies over a range of frequencies above and below resonance. There is a peak of response at resonance and a fall-off on either side of this peak. The disturbing frequency, f_d (Hz), is considered to be the frequency of the most probable vibratory force to require attenuation. Usually, satisfactory treatment of the disturbing (or forcing) frequency will produce 'bonus' treatment of all higher frequencies, and some lesser degree of attenuation at lower frequencies. The disturbing frequency may represent the most dangerous or annoying frequency and/or the most difficult frequency to attenuate. For the building services engineer it is usually the lowest speed of rotation/reciprocation present on an item of equipment, but other factors such as amplitude also affect the full assessment. Figure 11.2 also shows that, when the disturbing frequency is in the region of the resonance of the mass/spring system, there is amplification. The attenuation region is for frequencies above about 1.4 f_n for all values of damping, but the attenuation reduces as the damping increases.

Damping relates to the dissipation of energy within the isolator during vibration. Low damping means that vibration continues for a relatively long time after impulsive stimulation. High damping results in more rapid decay. When damping is greater than some 'critical' value, oscillations cannot take place and, after displacement, an object slowly returns to its equilibrium position, e.g. like a pendulum suspended in treacle. Helical spring vibration isolators have very small inherent damping, whilst elastomer materials have a greater level. It may be necessary to add damping to a helical spring isolator. This might affect the resonant frequency of the system if the added stiffness of the damping material is high.

Vibration isolation efficiency (VIE) is the term used to give the amount of the disturbing frequency that is not transferred to the structure. It is sometimes expressed as the opposite, transmissibility (T) where T gives the amount transferred.

Isolation efficiency and transmissibility are related by (VIE+T)=1, although VIE is usually expressed as a percentage.

The equation for VIE when the system has no damping is:

$$\text{VIE} = 100\left(1 - \left(\frac{1}{\left(\frac{f_d}{f_n}\right)^2 - 1}\right)\right) \qquad (11.5)$$

where VIE is the vibration isolation efficiency (%), f_d is the disturbing frequency (Hz) and f_n is the natural frequency (Hz).

Equation 11.5 gives the undamped response in Figure 11.2.

Figure 11.3 shows the relationship between disturbing frequency, the natural frequency and the vibration isolation efficiency. As with equation 11.2, note that a single degree of freedom (vertical vibration) is assumed and that the support structure is infinitely stiff. Mountings on flexible support structures are considered in section 11.4.2. As an example of the use of Figure 11.3, if the supported system has a natural frequency of 2.5 Hz and a forcing frequency of 5 Hz, the VIE is about 70%. In general, a forcing frequency of twice the system resonance frequency always gives 67%. A forcing frequency of three times the resonance frequency gives 87.5% whilst four times gives 93%, five times gives 96% and six times gives 97%. These figures are in the absence of damping.

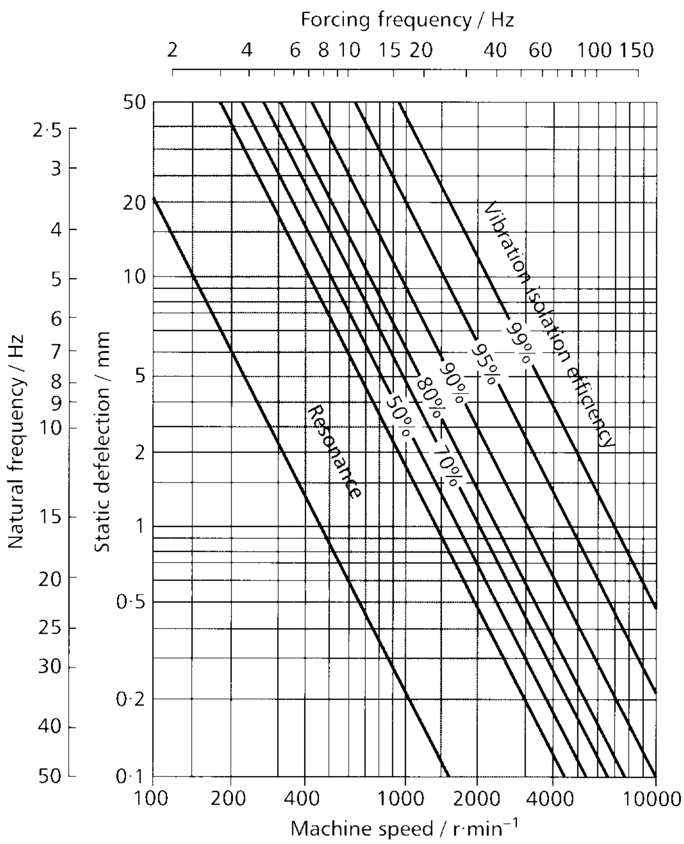

Figure 11.3 Relation between frequency, static deflection and vibration isolation efficiency (reproduced by courtesy of Eurovib (Acoustic Products) Ltd.)

Equation 11.5 applies when there is no damping (energy dissipation) in the system. This is never so in practice. The effects of damping include both limiting the maximum amplitude when the driving frequency is equal to the resonant frequency and reducing the isolation in the attenuation region.

11.3 Rating equipment for vibration emission

The degree of out-of-balance force transmitted by mechanical/electrical equipment in its normal operating mode (i.e. without specific, other than nominal, vibration isolation provision) is obviously critical to the evaluation of the vibration isolation requirement of a given machine in a given location. It is very difficult to obtain manufacturers' data for vibration emission and the conclusion is that very few have such data available. In the same way that it is now a requirement for manufacturers to provide acoustic data, they must be pressed to provide vibration levels. Data may be provided in various formats, but preference should be given to peak velocity, acceleration or displacement (any one or more of these) plus the frequency. The absence of such data has led to the adoption of various rules-of-thumb for treatment, resulting in degrees of under and over provision of isolation, and leading to wide variations of opinion and confidence in specialist advice. There are some areas of activity e.g. fine wine storage, electron microscopy, life science study, semi-conductor production, where such input data is essential, failing which over-design is inevitable.

11.4 Vibration limits

It is rare for clients or building services engineers to specify vibration limits for human comfort, e.g. an upper acceleration level at a specific frequency or frequencies. It is presently customary for engineers to use either the project's acoustical criterion as the limit for structure-borne noise or refer to limiting vibration via 'good practice' or other subjective terms, or perhaps by stating the percentage vibration isolation efficiency (VIE) required from any of the rotating/reciprocating equipment which puts energy into the structure. Any one of these methods might be augmented by specific reference to required treatments, e.g. pumps to be mounted on 25 mm static deflection spring inertia bases (perhaps with a minimum inertia ratio, see section 11.5). However, a distinction must be drawn between the selection/definition of a criterion and the individual isolation product/technique required to achieve it.

A noise criterion is effective only down to its lower frequency, say 31.5 Hz. It follows that a supplementary criterion is required for those frequencies lower than this which have their source in vibration. Exhortation to use good practice in selecting, installing, and isolating equipment is unsatisfactory. Specifying percentage vibration isolation efficiency (VIE) or its opposite, transmissibility, is an attempt to be objective. However the terms are widely misunderstood, in that they are very much frequency and amplitude dependent.

By this is meant that a given VIE level, say 90% (a common criterion), is demanding for high plant power at low frequencies, as the low frequencies call for high deflection of the isolation spring. However, 90% VIE is wholly inadequate for low plant power and high frequencies, since these frequencies easily become audible as radiation from the building surfaces. The satisfaction of the VIE requirement will also vary with the type of machine. Some machines, by their basic nature, emit greater degrees of out-of-balance force than others. Interaction with the structure, e.g. due to resonances in the structure, introduces further variability.

11.4.1 Vibration control to ensure comfort for occupants of buildings

BS 6472[29] gives limiting curves for vibration in buildings (acceleration/frequency and velocity/frequency) in terms of both *x/y* and *z* axes, and makes tentative suggestions as to which curves are likely to cause different degrees of human discomfort, see CIBSE Guide A[28]. However, BS 6472 is intended for diagnosis in existing situations rather than for prediction in the design of new buildings. A different approach is required for design, as described in the following paragraph.

A vibration criterion for general use may be obtained by setting VIE percentage limits in terms of the disturbing frequencies and machine powers. Table 11.3 relates the lowest disturbing frequency of interest to the spring static deflection (d) in order to produce the appropriate mounting resonance frequency which gives the vibration isolation efficiency stated in the table. Thus, for a disturbing frequency of 3.3 Hz and a VIE of about 40%, equation 11.5 shows that a mounting resonance frequency of 2 Hz is required. Equation 11.4 relates this frequency to the static deflection, d, of the mounts, giving 65 mm. To specify a VIE of 90% for a disturbing frequency of 3.3 Hz requires a resonance of about 1.1 Hz and a deflection of over 200 m, which is impracticable. Thus, Table 11.3 sets realistic, experience-based criteria. The required increase in VIE with machine power is linked to the greater force input of larger machines. The VIE values shown are for control of the lowest disturbing frequency given in the table.

It will be noted that this method of setting a criterion has the advantage of stating the required natural frequency and static deflection, as well as the resulting vibration isolation efficiency.

Its disadvantage is that it makes no allowance for variations in structural natural frequency (largely controlled by slab deflection) and it does not discriminate between sensitive buildings, e.g. studios, and less sensitive buildings such as retail stores. It is, nonetheless, a good general purpose guide for most common structures and is relatively easily understood and therefore enforced.

11.4.2 Relation between vibration at source and the associated structure

Simple vibration theory, as used above, assumes that the isolator is mounted on a solid foundation, a condition that might be approached in a basement. However, where plant is at a higher level in the building, the natural vibrations of the support structure, e.g. the plant room floor, might interact with the vibrations of the plant. In order to avoid such interaction, the resonant frequency of the mass on its isolators should be about one tenth of the fundamental resonance frequency of the floor slab. As the resonant frequency of the floor reduces for wider slabs, the mounting resonance of the vibration isolation system must also be reduced for these slabs. That is, in order to lower the mounting frequency, the static deflection should be increased as the slab width between columns increases. The structural engineer will be able to provide information on slab resonances.

Vibration isolation manufacturers have their own recommendations, which might typically be, say, for a

Table 11.3 Illustrating practical expectations for vibration isolation

Machine power / kW	Vibration isolation efficiency (VIE) static deflection (d) and mounting resonance frequency (f_n) for stated lowest disturbing frequency (f_d)								
	f_d = 3.3 Hz			f_d = 7.5 Hz			f_d = 12 Hz		
	VIE / %	d / mm	f_n / Hz	VIE / %	d / mm	f_n / Hz	VIE / %	d / mm	f_n / Hz
0–0.9	42	65	2	72	20	3.5	83	12	4.6
1–9.9	42	65	2	88	40	2.5	83	12	4.6
10–49.9	64	85	1.7	92	65	2	93	25	3
50–99.9	74	105	1.5	95	85	1.7	97	65	2
>100	78	125	1.4	96	105	1.5	98	85	1.7
	f_d = 15 Hz			f_d = 25 Hz			f_d = 33 Hz		
0–0.9	84	8	5.5	96	12	4.6	94	4	7.9
1–9.9	90	12	4.5	96	12	4.6	97	8	5.6
10–49.9	95	25	3.2	98	20	3.5	98	12	4.6
50–99.9	97	40	2.5	98	25	3.2	99	20	3.5
>100	98	65	2	99	40	2.5	99	25	3.2

Notes: (1) the disturbing frequency, f_d, is usually the lowest speed of rotation, (2) the machine power is the absorbed power at the given speed of rotation, not the nominal rating or the absorbed power at a different speed, (3) d is the minimum vibration isolator static deflection under a given static load; where the input data is of doubtful provenance, it is frequently necessary to select, say, a 75 mm static deflection isolator to give a minimum static deflection of, say, 60 mm, (4) f_n is the natural frequency of the isolator/mounted machinery as installed and levelled, and assuming the support structure and the machine base frame are of infinite stiffness and the minimum isolator static deflection is obtained disturbing frequency of 25 Hz and 95% vibration isolation efficiency:

— *installed in basement*: static deflection = 8.6 mm

— *on 10 m floor span*: static deflection = 9.9 mm

— *on 15 m floor span*: static deflection = 11.2 mm.

11.4.3 Criteria for equipment

Criteria for equipment can refer to emission levels which must not be exceeded, in which case an appropriate standard must be selected for that equipment, taking care to ensure that power, velocity/acceleration and frequency are included in the maximum permissible levels.

Criteria for fragility levels is best given as the maximum acceleration that is to be withstood at the critical frequency (usually the lowest), without resulting in machine malfunction.

11.5 Common types of vibration isolator

Figure 11.4 illustrates common types of vibration isolator[30,31].

11.5.1 General

A number of difficulties arise in the selection of vibration isolators. In particular, static deflection depends on vertical load, but this load is not always known accurately. The vertical load from the plant is either given by the equipment manufacturer or calculated from input data supplied by the manufacturer. Either way it is subject to error, often an underestimate. Additionally, anti-vibration mount (AVM) suppliers add safety factors over and above any safety limit built-in to the isolator design. The AVM supplier will not normally produce a bespoke mount, but offer the nearest standard one above the required carrying capacity. It is therefore important to make a distinction between minimum required static deflection and the nominal static deflection that a mount gives when accurately loaded. A common example is the selection of nominal 30 mm deflection helical spring mounts to give not less than 25 mm static deflection.

11.5.2 Pad materials: flat, laminated and contoured

With some exceptions, pad materials are too unpredictable and limited in performance to be regarded as reliable vibration isolators. Their most popular, convenient, and successful role is as cheap and easily fitted sound separators, rather than vibration isolators. Elastomeric materials have inherent variability of elastic properties, which affect their performance. A column of elastomer will not compress uniformly, and will only deflect in proportion to its ability to bulge or distort under load, determined by the 'shape factor', which is given by:

$$\text{Shape factor} = \frac{\text{Area under load}}{\text{Area free to bulge}} \qquad (11.6)$$

Thus, a circular disc of elastomer will deflect somewhat less than a rectangular strip of equal thickness and equal surface area. Generally, a high shape factor produces a stiff mount.

In order to avoid over-stress, a deflection limit on pads is 10–15% of the thickness. Reliable deflection is also compromised by the wide manufacturing tolerance, presently ±5%, allowed for moulded stiffness (related to Shore hardness). Further, the static and dynamic stiffnesses of a given elastomer differ, due to non-linear deflection. The mounted resonance frequency should not be determined from equation 11.4, although an approximation may be obtained by multiplying the result by 1.5. For preference, refer to individual manufacturers for information, but a distinction should be drawn between standard catalogue information and certified data. Much catalogue data will assume equation 11.4 is valid for elastomers, which can be misleading. Trial design exercises will quickly demonstrate the limitations of pads as isolators. They are best regarded as low frequency noise separators, i.e. vibration breaks rather than vibration isolators.

Pad materials are available either as single units or in sheet form, from which the desired size may be cut. The material is often ribbed or similarly profiled, in order to increase its flexibility, or stacked in multiple vertical layers with a stiff diaphragm material between layers to improve 'shape factor' effects. It will be recognised that a doubling of pad thickness will result in approximately twice the deflection at equal load, and a doubling of surface area will result in a doubling of load capacity at constant deflection. However, these approximations ignore the dynamic factors referred to above. Pads must be uniformly loaded to prevent excessive localised compression and, because of their high stiffness, it may be difficult to load pads uniformly with sufficient weight for the required deflection. They cannot be mechanically fastened down to the structure without compromising isolation unless they are incorporated in suitable housings or between plates.

Polychloroprene (neoprene) is considered to have the best all round properties, but a wide range of materials are available, all of which have various advantages and limitations. A distinction should be drawn between a properly specified material and 'commercial grade', where quality can be distinctly variable. Bearings for critical applications are always subject to batch test approval. For example, when elastomeric blocks are used in building bearings, all of the static and dynamic variables are taken into account, as a guaranteed maximum resonant frequency is required. Manufacturing techniques include compression moulding, injection moulding, extrusion and calendering.

11.5.3 Elastomer-in-shear

These mounts suffer from the same fundamental limitations and unpredictable variables as do pads, but to a lesser degree. However, to achieve significant deflections without excessive column height, it has been common practice for some 50 years to install elastomers at least partly in shear. An example is turret mounts, with static deflections up to 10 mm. There is greater flexibility in shear than in compression, although most isolators use a

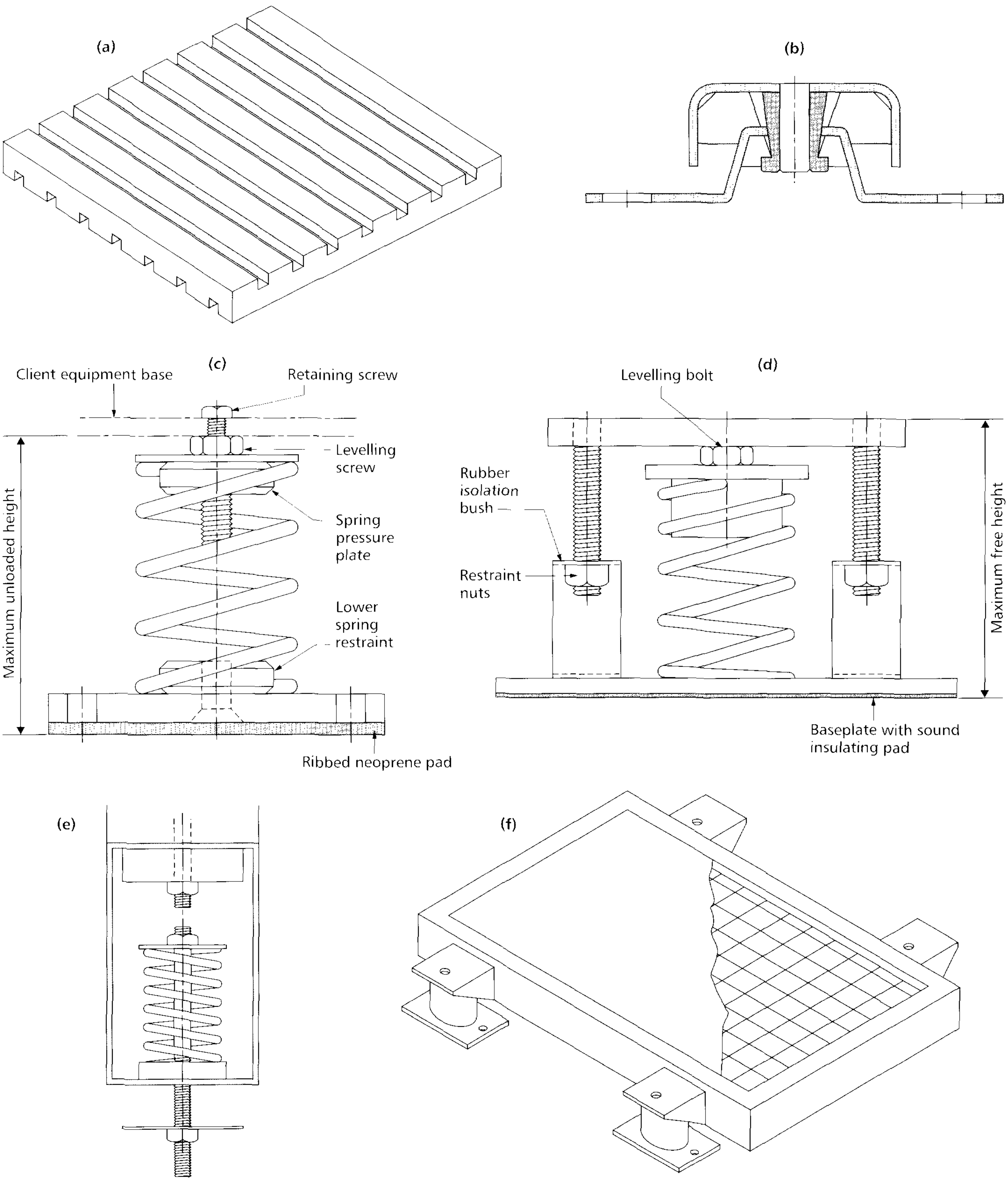

Figure 11.4 Typical vibration isolators; (a) ribbed mat, (b) elastomer in shear, (c) open spring isolator, (d) restrained spring isolator, (e) pipe/duct hanger, (f) formwork for inertia base (reproduced by courtesy of Eurovib (Acoustic Products) Ltd)

combination of shear and compression. The most commonly claimed upper deflection limit is 12 mm, but 10 mm is regarded as the practical maximum.

For the approximate determination of resonant frequency, in the absence of manufacturers' certified data, equation 11.4 may be used and the result multiplied by 1.2. As with pads and blocks, elastomer-in-shear mounts are most commonly associated with lighter loads and higher disturbing frequencies, but there are exceptions. These mounts are also available in hanger form, both on their own and in conjunction with helical springs. Unlike pads, elastomer-in-shear mounts can be fixed down without compromising the isolation. They are very commonly incorporated into fan and pump products by the manufacturer.

11.5.4 Helical springs

The helical spring is the most commonly used, most reliable and most predictable device employed in vibration isolation. Springs differ from elastomer materials in that they deflect uniformly under increasing load and have equal static and dynamic stiffness, as required for equation 11.4. There are various configurations of mounting including open, caged, enclosed and restrained. They are also available as pipe or duct hangers. When supplied in other than 'open' mode they can be damped or undamped. They are available in a wide range of load carrying capacities from kilograms to tonnes per isolator.

Typical deflections are 30 mm, giving a mounting frequency of about 3 Hz and a vibration isolation efficiency (VIE) of over 90% but they are readily available in deflections up to 150 mm. Helical springs occasionally fail at higher machine frequencies due to wave propagation through the spring (especially in hanger mode) and this has led to the incorporation of elastomeric pads as vibration breaks. When in 'open' format it is good practice for the ratio between spring diameter and spring deflected height to be no less than 0.8:1.0 in order to promote spring stability. When enclosed, caged or restrained this relationship is clearly less important, although good spring design should always preclude buckling and other forms of instability.

If springs are to be fixed at their upper or lower ends, methods other than welding, which affects temper, should be sought. Springs are generally designed with up to 50% overload potential, but the overload percentage will diminish as the rated load increases, otherwise the spring will reach unreasonable dimensions. Load range can be increased by ganging springs in parallel, leading to restrained spring mounts of up to 8 tonnes load at up to 150 mm static deflection.

11.5.5 Inertia bases

Any spring mounted base which is used to support mechanical equipment is referred to as an inertia base. A steel base frame without spring support is a known as a skid, and concrete bases without springs underneath are referred to as plinths or 'housekeeping pads'. Displacement forces from the mechanical 'driver' have to overcome the additional inertia provided by the base. This is very effective where there are two or more drivers, perhaps at differing speeds of rotation, as with a motor and a fan or a motor and a pump. The base may be a bolted or welded steel frame, a bolted or welded steel concrete pouring frame, or a cast concrete base on a timber or metal former, the former having first been fitted on the underside with pads or mounts. Inertia bases can be used to provide additional mass per spring, thus increasing static deflection and reducing the resonant frequency of a supported system. However, care should be taken that the additional mass does not cause problems to the structure. The additional mass is useful in lowering the centre of gravity of the supported system and in providing resistance to lateral and axial forces. A typical inertia base consists of a steel form, which is filled with concrete to give the mass.

Support brackets for vibration isolators are usually attached to the base. A larger base is used to provide a platform for a number of items of equipment e.g. run and stand-by pumps, which cannot conveniently be supported individually. Most pumps are very conveniently mountable on inertia bases, which also gives the advantages of improved vibration isolation. Spring mounted concrete pouring frames are available in flat pack form for bolted on-site assembly. These have the advantage of lightweight shipping and site handling and their final location in the plant room can be chosen and adjusted before pouring the concrete. Advice should be sought before finalising the design of an inertia base.

The rule of thumb for the ratio of masses of the base and the supported equipment is 1.5:1.0 for most conventional equipment, including pumps, but could rise to 5:1 for equipment with large lateral out-of-balance forces such as high pressure blowers, where there are high static and velocity pressures.

It has become common for the fan and motor section of air handling units (AHUs) to be internally isolated using a lightweight steel inertia base, spring mounts and one or more flexible duct connectors. Whilst this arrangement can work effectively, it is a generalised optimisation and for more critical applications it is preferable to have the fan/motor section mounted rigidly and to isolate the whole AHU from the structure, if necessary incorporating steel rails to tie the whole assembly together. This has the added advantage of presenting the whole of the unit as inertia to the 'driver'.

11.5.6 Flexible pipe connectors

Although quite short, typically 150 mm flange-to-flange, these give flexibility to prevent anchored pipes by-passing the isolators and also take up strains from minor misalignments. They are not to be confused with expansion joints, which are specialist products employed for different purposes. They are available in a range of reinforced elastomers, and with flanged or screwed connections. For high pressures or risk of displacement they are available with isolated restraint rods. Reliable data for their isolation efficiency are not available but it is generally believed to be good practice to install them, for example, on any pump flow and return. Spring hangers will still be required, especially where pipework is fixed to structures directly adjacent to occupied areas. They are not to be confused with hoses, some of which are properly rated for noise and vibration isolation[32]. A typical pump installation reflecting good vibration isolation practice will consist of a spring inertia base, a flow and return flexible pipe connector, and perhaps four spring hangers.

11.5.7 Flexible duct connectors

Used as a vibration break, particularly between the fan and its duct, they can be of circular or rectangular section. The recommended length is sufficient to allow 150 mm of slack between the two coupled ducts. They are normally made from organic or synthetic canvas impregnated with neoprene or another sealing elastomer, but are also available in a range of alternative materials with varying acoustic and fire ratings. The better models are provided with their own integral fabric flanges and a matching metal flange. They can also be fitted directly to coupling ductwork by jubilee clips. Reliable data for vibration isolation performance are not available but their use signifies good practice, although duct isolation hangers may still be necessary.

11.5.8 Isolation hangers

These are used for ducts and pipes. Where plant rooms directly adjoin occupied areas and especially where 'weak' structures are present, it is essential that pipework and ductwork are isolated from the structure The flexible element may be either helical spring or elastomer or a combination of these. A good design of hanger will have a spring diameter to height ratio of 0.8:1.0, large clearance holes which are grommet and bush protected, an elastomeric element to preclude high frequency bypass, and a restraint cross-bar to enable the load to be taken and pipes to be levelled prior to the spring being actually loaded. This last feature prevents spring overload and obviates erratic pipe levelling. Where detailed pipe and duct co-ordinated services drawings are available, together with adequate architectural and structural detail, hangers can be selected and located in advance. Current practice is for pipework to be installed 'rigid' on standard studding, but leaving enough vertical room for the hangers to be cut-in later, thus ensuring their accurate selection from a site survey. Increasing machine powers, weaker slabs and developing environmental awareness, are putting greater emphasis on the inclusion of spring hangers in plant rooms. Spring hangers are essential where acoustic ceilings are installed.

11.5.9 Air springs

The use of helical steel springs requires increasing amounts of static deflection as the required mounted natural frequency is reduced. Assuming a rigid structure, from equation 11.4, a natural frequency of 1.4 Hz requires a static deflection of 125 mm. To achieve an actual static deflection of 125 mm, helical springs would be designed for 150 mm static deflection, due to the input variables, which probably represents the upper limit for convenient helical spring mount design, especially where very light or very heavy loads are concerned. Air springs operate in a different way and achieve natural frequencies of 1 Hz (equivalent to 250 mm of static deflection). They are basically air-filled reinforced rubber cushions with a valve controlling the internal air pressure, giving variation of the load carrying capacity and natural frequency. They require regular inspection and maintenance, including a constant air supply. Some types of machinery, such as electron microscopes, require a very high degree of isolation. Air springs serve this purpose, taking up much less space than their helical spring counterparts. They are not generally as convenient or competitive as elastomers and helical springs, which are for the general range of equipment encountered by the building services engineer.

11.5.10 Floating floors

The maximum difference in noise level obtainable from a homogenous construction is about 50 dB, particularly at lower frequencies. No matter how much design input and installation care is taken to optimise the mass, the quality of the construction and avoidance of leaks through air gaps, the level difference is controlled by flanking paths. The purpose of floating floors is the minimisation of flanking paths and the introduction of an isolating air space

Whilst there is no reason why floating floors should not be designed for vibration isolation, this can lead to difficulties, particularly from differential plant loading and it is for this reason that the floating floor is more often considered as an acoustical treatment. Floating floors are designed with natural frequencies down to, say, 6 Hz, which is in the range of elastomer in shear isolators, whilst for higher resonant frequencies, say 10–15 Hz, pads and quilts can be used. The simplest floating floor is concrete laid onto a resin bonded mineral wool mat, such that the mat has only a small deflection. More complex systems use an array of pads covered by formwork, onto which the concrete is poured. Some spring systems may be jacked up after the floor is laid.

Design of a floating floor must take the following into consideration:

— the additional mass of floor and plant on the structural floor

— the appropriate load bearing points on the structural floor

— the number and distribution of mounts.

Specialist advice must be sought in the design of a floating floor.

11.5.11 Structural bearings

These are used to isolate whole buildings from earthquake or shock. Construction of the bearings is often elastomeric pads or blocks, sometimes built up as a multiple sandwich construction between steel plates although springs may be used. This is a very specialised area for which expert help must be sought.

11.6 Practical examples of vibration isolation

11.6.1 General observations

11.6.1.1 Degrees of freedom

The simple one-dimensional approach is to consider a spring-mounted body constrained to move in one direction, normally vertical. This is a good approximation to what occurs in most installations, but other movement, generally undesired, may occur. A mounted body has the potential to move in three linear directions, X, Y and Z, whilst also turning on its mounts in three rotational modes about these axes. The non-vertical movements occur when forces acting from the isolators on to the body exercise a turning moment through its centre of gravity. This is avoided when the top supports of the isolators are in the horizontal plane through the centre of gravity but, in practice, this idealised design may not be possible. The design will be to keep the centre of gravity at a low level.

11.6.1.2 Number and location of mounts

Knowledge of the position of the centre of gravity enables moments to be taken in order to determine the load at each mounting point. Where the number and location of mounts is pre-determined by the equipment manufacturer, it is reasonable to expect that the equipment and base design has, as far as possible, centralised the centre of gravity in the horizontal plane and positioned it as low as practicable. Additionally, the mounts should be located for equal loading. Optimally, they will also be located at equal centres and the manufacturers' skid or base will have been designed to be sufficiently stiff for minimal deflection between mounting points. This is the ideal situation and will usually position the mounts at a maximum of 2000 mm centres.

When a beam has its centre of gravity dead centre, and is supported by a number of mounts at equal centres, the actual load on the mounts at the extremities of the beam are half the load on the intermediate mounts. The solution here is to centre the mounts at equal centres such that the span between mounts is twice the span between the end mounts and the ends of the beam. Where this is not possible, the end mounts should be rated at half the load of the intermediate mounts. When equipment is mounted on a very stiff beam and mounts are selected for equal loads, the deflection of the mounts will be equal, despite uneven load distribution along the beam. Bending of the supporting beam could occur where large loads and lighter skids are involved. Many computer programmes for selection of anti-vibration mounts only work in two dimensions and assume that the beam is stiff enough to resist bending. Where the centre of gravity is not geometric dead centre, the same practical approach can be taken, except that, where a load is obviously biased toward one end of the beam, 'tipping' will occur unless the load capacity of the mount is increased at that end.

The rule of thumb for skid design is that the beam should have a depth not less than $^1/_{10}$th. of the span between mounts, whilst beam deflection may be up to $^1/_{250}$th. of this span. It is reasonable for the building services engineer to require equipment manufacturers to mount their equipment on adequately stiff skids, and to give the location of the centre of gravity. Where the manufacturer has pre-determined the mount positions, they should also be able to state the loads for each mount. There will be occasions, however, when the equipment supplier will have only partial information, e.g. the total weight plus individual component weights. A specialist will then have to determine the position of the centre of gravity and calculate individual mounting point loads. Further, there will be occasions when it will be necessary to supplement the manufacturer's skid with an additional base frame, or perhaps an inertia base. For a rectangular base, four or eight mounts are preferable to six, in order to inhibit rocking modes.

Where mount location points are not pre-determined the vibration specialist will endeavour to locate mounts at equal centres and at equal loading. Subject to the maximum span between mounts and to skid stiffness, it is probably better to have mounts at equal loading but different centres, than differential loading at equal centres. Another factor that will affect the number of mounts is load. Most anti-vibration mount manufacturers produce a range of standard, single-spring mounts at 'standard' deflections up to about 1200 kg vertical load. Higher load mounts are produced with load sharing, multiple spring elements arranged in parallel. When selecting a mount for a given application, the specialist will usually choose a standard product with standard load/deflection characteristics. This will be the mount that carries the nearest standard load above the specific design load.

11.6.1.3 Types of mount

Pads and elastomer-in-shear mounts are normally used for light loads and small machine power. Pads and blocks are useful 'sound separators' but are not an effective vibration isolator, other than when specially designed and selected, e.g. bearings, floating floors. Elastomer-in-shear has its lowest dynamic natural frequencies in the region of 6 Hz which will give adequate vibration isolation efficiency (VIE) for disturbing frequencies down to about 1000 r·min^{-1} (17 Hz), depending upon machine power and location. A maximum power of 3 kW and a minimum speed of 1000 r·min^{-1} is a reasonable guide. Although such mounts are available for high loads, the maximum load per mount should be less than 180 kg. These mounts should never be exposed to loads in extension, nor any significant lateral load without the approval of the manufacturer.

Commercial grade neoprene is a good all-round choice of pad material for plant room use. These mounts are sometimes designated 'single deflection', meaning deflections up to 6 mm maximum, and 'double deflection' for up to 12 mm deflection. Elastomer-in-shear mounts, particularly the cheaper ones, do not normally incorporate levelling devices but, because of the limited deflection, any differential loading will not result in undue out-of-level problems. As with all plant mounting operations, however, one should ensure that the initial plant base location is levelled before installation.

For applications involving machinery in excess of 3 kW and/or below 1000 r·min^{-1} (17 Hz), helical steel spring mounts are normally selected. The simplest of these is the open spring, which is typically restricted to operations where there are no significant fluid loads or excessive wind or other lateral loads. They are available in small sizes for light duty, up to 25 mm maximum deflection where lateral stability is not critical, and in larger sizes for 25 mm and higher deflections, where the geometry of the spring design will be much more stable and capable of higher loads. The smaller springs are typically used in the fan and motor section of small air handling units and the larger springs on a wide range of equipment. Springs that might otherwise be unstable are caged or enclosed. A cage will usually offer a means of lateral restraint and guidance, perhaps also being damped by elastomeric side shields.

The principal difference between caged and enclosed springs is that caged mounts allow visual access to the spring, whereas totally enclosed mounts do not. However, the totally enclosed spring is better protected, which is considered essential in some adverse environments, such as off-shore installations. Where significant fluid loads or lateral loads are concerned, restrained spring mounts should always be selected. These mounts are also preferred for high deflection applications above 50 mm. Differential loading or deflection on mounts, whether accidental, or unavoidable, can lead to a variation in mounted height.

Obviously, the minimum height available is that set by the mount which is least deflected and all other mounts must be levelled up (in progressive rotation) at least to that point. (It is always possible to level up, but never to level down.) If it is necessary to predict a levelled height prior to mounting and commissioning a machine, a safety factor should always be added to take account of such variations.

11.6.2 Determination of loads for a mounted system

The following example demonstrates how to calculate loads for a system mounted at four points. It is treated in a step-by-step manner for clarity. Other procedures, generalised formulae and computer programs may also be used. The methods given are applicable to determination of the springs required for a variety of isolated systems such as pumps, fans, air handling units etc. Large systems, e.g. floating floors, cooling towers and whole buildings, require special techniques, some of which are discussed later.

Example 11.1: fan and motor combination

The data for an air handling unit consisting of a single fan and motor mounted on a rigid frame are as follows:

— weight of fan: 21 kg
— weight of motor: 38 kg
— weight of frame: 12 kg
— frame dimensions: 1000 mm × 550 mm
— operating speed: 900 r·min^{-1} (= 15 Hz)
— vibration isolation efficiency: 95%

The positions of the centres of gravity of the fan and motor will have been provided by the manufacturer, or determined separately, and are as shown in Figure 11.5.

The centre of gravity of the combination is on the line joining their separate centres of gravity (CG_1 and CG_2). The length of this line (T) is given by:

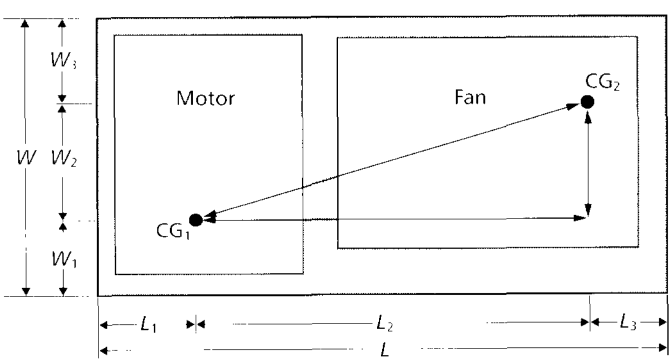

Figure 11.5 Example 11.1: Plan dimensions and location of centres of gravity of fan and motor

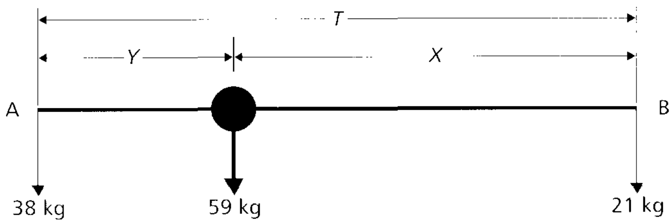

Figure 11.6 Example 11.1: Position of centre of gravity of combination on line joining centres of gravity of fan and motor

$$T = \sqrt{(L_2^2 + W_2^2)} = \sqrt{(600^2 + 220^2)} = 639 \text{ mm}$$

The position of the centre of gravity of the combination on the line joining their separate centres of gravity is then given by moments around B as in Figure 11.6.

Hence:

$$59\,X = 38\,(X + Y) = 38 \times 639$$

Leading to:

$$X = 411.6 \text{ mm and } Y = 227.4 \text{ mm}$$

It is now necessary to determine the weight distribution of the fan and motor at the corners of the frame, where the mountings will be located. The weight of the frame itself is divided equally between the corner supports.

The position of the fan/motor centre of gravity with respect to the edges of the frame is obtained by first determining distances L_2' and W_2' as in Figure 11.7. From consideration of the triangles it follows that $L_2' = 214$ mm and $W_2' = 78.3$ mm.

The centre of gravity of the combination in the horizontal plane is then determined as shown in Figure 11.8, i.e:

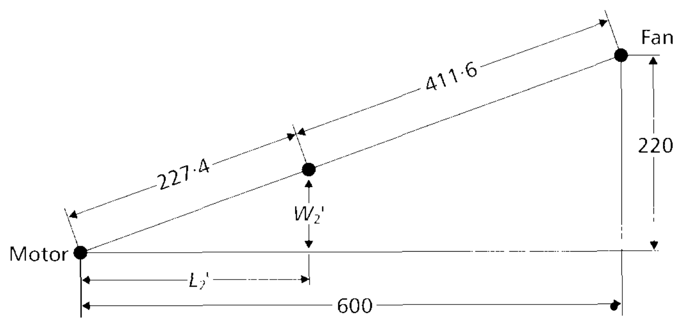

Figure 11.7 Example 1: Location of centre of gravity of combination

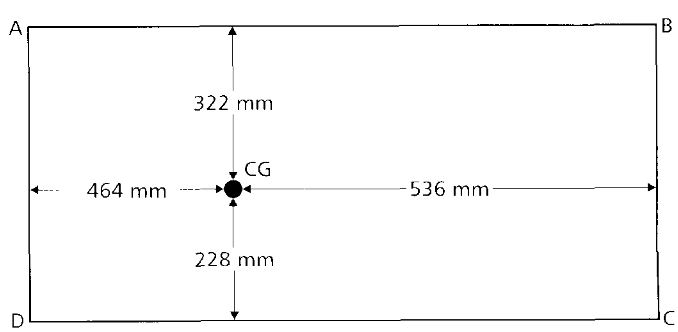

Figure 11.8 Example 1: Position of centre of gravity of combination with respect to frame

— distance from side AD of frame = $L_1 + L_2'$ = 250 + 214 = 464 mm

— distance from side BC of frame = 536 mm

— distance from side DC = $W_1 + W_2'$ = 150 + 78 = 228 mm

— distance from side AB = 322mm

(As vertical motion only is assumed, the height of the centre of gravity is not considered).

The load sharing between sides AD and BC is obtained by a similar process to that shown in Figure 11.6. Hence:

— load carried on side BC = (464/1000) × 59 = 27.4 kg

— giving load carried on side AD = 31.6 kg.

In a similar manner, the distribution of load between points B and C is given by:

— load at B = (228/550) × 27.4 = 11.4 kg

— giving load at C = 16 kg

— load at D = (322/550) × 31.6 = 18.5 kg

— giving load at A = 13.1 kg

In addition to these loads, each support point carries one quarter of the 12 kg weight of the frame.

Therefore, total loads are 16.1 kg at A, 14.4 kg at B, 19 kg at C and 21.5 kg at D. However, it is advisable to add a safety margin of 15–20% to the calculated loads.

The isolation efficiency required is 95%. From Table 11.3, standard 25 mm deflection mounts, chosen for appropriate load carrying, will be suitable.

11.6.3 Mounting specific plant items

The above example is applicable to many items encountered by the building services engineer. Large plant may require additional mounting points which, for very heavy plant, may need to be chosen in relation to positions of support columns in the building. The following sections considers some specific problems.

11.6.3.1 Mounting heavy plant on isolator rails

Cooling towers are considered, by way of example, but the principles are applicable to mounting any large item of plant on isolator rails, e.g. high capacity air handling units.

Traditionally, cooling towers were located either unmounted in non-critical locations or sound-separated by pad/strip isolators. In some cases the fan section alone would be vibration isolated. As requirements became more demanding, isolated rail systems were developed for these and other large plant, whereby a pair of toe-down support channels were fitted along the length of the base of the tower with a nest of helical isolation springs. Isolation for equipment connected to the tower was provided by flexible pipe connectors, flexible connectors on forced draught fan discharges and, perhaps, some on-board isolation for the pump. The deflection limit was generally 25 mm, which worked well and, in many instances, is still effective.

The use of cooling towers has declined but they are still encountered in both closed- and open-circuit form, especially where very large condenser water loads are required.

A large office block may have one or more cooling towers located at roof level, served by induced draught axial flow or centrifugal forced draught fans. Multiple fans with a total absorbed power of 30 kW, and with a pump at perhaps a further 5 kW may be used. The power (thermal) rating for these towers can be as high as 3 MW and the operating weight 15 tonnes. Complications typically arise from:

— inverter speed control, typically 800 r·min^{-1} or less

— a weak roof with a critical space, e.g. directors' suite, immediately beneath

— cooling towers of modern construction that do not allow direct mounting but require some form of base frame with mounts located below.

The static deflection will be determined by the required VIE at the optimum duty (usually the highest speed) plus any additional static deflection to avoid coupling with the natural frequency of a weak roof. The highest speed is the most critical, because at lower speeds there is a reduction in absorbed power due to reduced rotational forces. Table 11.3 indicates isolation on a rigid slab to at least 95% VIE, which requires a static deflection of 30 mm. The spring natural frequency is 2.75 Hz and the forcing frequency is 13.33 Hz (i.e. 800 r·min^{-1}). The importance of the slab stiffness is not so much a function of its deflection under its own load, but rather any additional deflection created by the equipment load.

The floor spring rate is compared with the mount spring rate, where the mount spring rate needs to be not more than 1/10th. of the slab spring rate in order to separate the two resonant frequencies. A short cut to obtain the total mount deflection is to add the additional floor deflection under the load of the machine to the static deflection of the mount. As a final check, ensure that the natural frequency of the mount does not coincide with the natural frequency of the slab (information obtained from the structural engineer) and that a vibrational mode of the slab does not coincide with the disturbing frequency. The most obvious measure, if either should occur, is to further increase the mount static deflection and hence the VIE. For the present example, assuming an additional slab deflection under load of, say, 10 mm, the design static deflection of the mounts is increased from 30 mm to 40 mm, giving a spring natural frequency of 2.5 Hz.

If the floor deflects an additional 10 mm under a load of, say, 15 tonnes, the floor spring rate is 1500 kg/mm and the mount spring rate needs to be not more than 150 kg/mm. In practice, because of the input variables and spring selection and optimisation, the spring selection is preferred as nominal 50 mm static deflection mounts to achieve in situ deflection of not less than 40 mm. Current rooftop applications indicate that static deflections of nominal 75 mm are common and up to 125 mm is not unusual as a result of the combinations of high machine power, low rotational speeds, critical locations and weak roofs.

It is usual to arrange two or three vibration isolated parallel steel rails, laid on the roof or other flat surface, on which to support the plant. The rails should extend to the full dimensions of the base and have small deflection under the load of the plant. The rails are drilled to allow the plant to be mechanically fixed and to accept the requisite number of restrained spring mounts. Because of difficulty in accessing levelling screws and also in providing a clear landing ground on the mounts, internal levelling models of mount are usually provided. Depending upon rail profile the rails may have to be cut out to allow access to the levelling screws. The number of mounts is determined by the total load required and the permitted maximum span between mounts, which in turn will be influenced by structural column centres and the structural engineer's proposals for supporting point loads from the mounts. Typical rail sections are 200 mm × 75 mm parallel channel or 203 mm × 203 mm universal column. The total and sectional static operating loads are generally available from the cooling tower manufacturer and will vary from rail to rail and at various points along the rail. The vibration isolation specialist has the choice to select mounts for individual point loads, or to equalise the loads and allow the beam to accept any stress differentials. The mounts are held-down to the sub-structure and incorporate restraint devices to deter lateral and vertical lift. The mounts are factory-fixed to the upper rail and the upper rail is site-fixed to the tower. Rail-end 'tip-potential' loads should be allowed for in the choice of isolators.

The pump set, if mounted on a tower, is indirectly isolated by the rail system but generally also incorporates some pad or bush sound separation. Flexible connectors are installed on forced draught fan discharges, and pipework connections are isolated via restrained flexible pipe connectors and spring hangers or floor supports. The natural frequency of the pipework isolators is best matched to the springs supporting the rails. Any nearby structural pipe penetrations must be resiliently sealed through the structure to control noise and vibration transmission.

Simple rail isolation systems are available as standard from suppliers. For more complex applications, advice must be sought from a vibration specialist directly, or via the plant manufacturer, and the design criteria and compliance must be clearly understood by all parties.

11.6.3.2 *Vibration isolation of pipework or ductwork*

Vibration transmission occurs along pipes and ducts, despite the use of flexible connectors. In pumped systems, pump vibration is carried by the liquid and may reappear as noise at any location where there is a hard contact between the pipe and the structure. Similarly, duct vibration arises from the fan and from turbulence in the air stream. A vibration isolation pipe hanger is shown in Figure 11.4(e). Rubber bushes are used in pipe clamps but these act in a similar manner to pad materials and are sound breaks, rather than vibration isolators.

The hanger is chosen to compress appropriately under the load being carried. There are considerations of permitted length of unsupported pipe or duct and of correct alignment. If the hanger is misaligned, the lower rod might contact with the hanger assembly, so bypassing the spring. A trapeze support might be used, where the pipe is carried by a length of steel, which is itself supported by a hanger at each end. Trapeze supports are often used to hang ducts.

The hanger itself is fixed back to the structure by a rod, or a wire for small capacity hangers used with light loads. Hangers are available with single or multiple parallel springs to carry loads from about 5 kg up to 1000 kg.

11.6.3.3 *Floating floors*

Floating floors are used both to provide vibration isolation and to increase sound transmission loss. Flanking paths limit noise isolation to about 50 dB, and normally less at lower frequencies. It follows that where high level differences are required, e.g. plant rooms immediately adjacent to conference suites, then floating floors, and possibly floating rooms are a solution. Also, where there is a risk that outside noise and vibration will penetrate a building, floating design for a section might be preferred to the option of isolating a whole building.

Whilst it is possible to design floating floors strengthened locally to accept plant loads, this is not a preferred technique unless disturbing frequencies are high, and loads are low (in which case floating floor requirement is doubtful). The reasons for this include:

— floor fracture during plant installation

— requirement for higher deflection floors than would otherwise be the case

— problem of coupling between vibration characteristics of the plant and those of the floor, leading to the plant driving the floor into vibration.

Therefore there is not a good case for floating plant rooms in their entirety and relying on this measure for total vibration isolation.

The technique is to cast housekeeping pads to the main structural slab to a depth where they will project above the proposed floating floor. The plant can then be mounted on high deflection, high efficiency anti-vibration mounts and, if necessary, on spring inertia bases. This method has the advantage that the local mass increases the efficiency of the vibration isolators and that the floating floor does not cast up to the perimeter of the housekeeping pad using closed cell foam edging strip, or similar, to give resilient separation. In order to complete the isolation of the plant room, walls can be built up on the floating floor and the ceiling hung by springs from the structural slab above. The effect is one of a room within a room with complete separation from the enclosing structure.

Floating floors may be 'wet' or 'dry'. Dry construction is usually used in studios and multiplex cinemas where the anti-vibration mounts, which could be pads, quilts, blocks, or mounts are secured to the bottom of a suitable rigid board (e.g. 25 mm thick marine plywood). Partitions and perimeter walls are in turn erected from the floating floor and various kinds of floor finish are applied in the usual way. The same technique can be applied to wet construction where a polythene membrane is firstly installed above the plywood formers, following which the upper concrete floor is cast. Alternative constructions are available, including 'jack-up' mounts, which are incorporated into a floor cast on polythene sheet and used to raise the floor after it has set. It is unusual for floating floor elements to be installed at greater than 900 mm centres and 600 mm is more usual. Floating floors are better described as sound separators than vibration isolators and when installed with care work extremely well. Bad installations are virtually irretrievable

12 Summary of guidance on noise and vibration control

12.1 Noise in HVAC systems

Noise in HVAC systems is controlled by following the advice given in this Guide. In particular:

- Choose a quiet fan, which is sized to operate at an efficient point on its characteristic.
- Design for good airflow. Aim to minimise turbulence and pressure loss, both of which produce noise.
- Include all sources in predictions, e.g. breakout, in addition to duct borne noise.
- Do not forget that building services systems might affect the sound insulation between neighbouring areas.
- Seal all wall penetration with flexible material. This reduces both noise and vibration.
- Choose the location and selection of external plant and air grilles to avoid noise disturbance to nearby properties.

12.2 Vibration in HVAC systems

Vibration in HVAC systems is controlled by the following the advice given in this Guide. In particular:

- Choose a good location for the plant, remote from sensitive areas. This also helps with noise control.
- Ensure that vibration isolation is properly installed with no bridging material across the flexible mountings.
- Ensure that vibration isolators are loaded to give equal deflections and installed to maintain vertical alignment of their springs and other components.
- Remember that misaligned isolators are a source of many problems.
- Check support bolts for integrity and free movement.
- Do not neglect vibration from pipes and ducts. Use flexible attachments to the structure.

Further advice is given by Schaffer[33].

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Appendix A1: Acoustic terminology

A1.1 Basic parameters

A1.1.1 Frequency, wavelength and velocity

Sound is produced by rapid pressure fluctuations in the air. The fluctuating pressure is about one hundred thousandth of the static atmospheric pressure for what would appear to us to be a 'very loud' noise of 94 dB. The frequency of the fluctuations may be between 20 times a second (20 Hz), or lower for some fan instabilities, and up to 20 000 times a second for audible noise. However, for HVAC, we are not normally concerned with frequencies above 4000 Hz, or occasionally 8000 Hz.

In addition to frequency, the quantities that define a sound wave include:

— wavelength, λ

— velocity, $c = 345$ m·s^{-1} (approx., depending on temperature).

Wavelength, frequency and velocity are related by the following equation:

$$c = \lambda f \qquad \text{(A1.1)}$$

where c is the velocity (m·s^{-1}), λ is the wavelength (m) and f is the frequency (Hz).

Thus we can relate frequency and wavelength by velocity, see Table A1.1.

It is useful to develop an appreciation of frequencies and related wavelengths, since this helps an understanding of the operation of noise control.

Noise frequencies are obtained from a frequency analyser which, in the case of simple analysis requirements, can be incorporated into a sound level meter (see Appendix A4).

A1.1.2 Sound pressure

The sound pressure in a wave is force per unit of area for the wave and has units of pascals (Pa) (i.e. N·m^{-2}). The sound pressure fluctuates above and below atmospheric pressure by a small amount and a time average may be zero. The sound pressure is therefore quantified by the square root of the square of the fluctuations, giving the root mean square (rms) value. (Squaring the pressure fluctuations makes all values positive.)

Table A1.1 Relation between frequency and wavelength of sound

Frequency / Hz	Wavelength / m
63	5.5
125	2.8
250	1.4
500	0.69
1000	0.35
2000	0.17
4000	0.086

A1.1.3 Sound intensity

The sound intensity is a measure of the rate of flow of sound energy in watts per square metre (W·m^{-2}). This is analogous to illumination.

A1.1.4 Sound power

The sound power is a characteristic of the source, expressed in watts (W). The sound power is a fundamental quantity associated with the source alone. Intensity and pressure depend on the transmission path from source to receiver.

A1.1.5 Noise level

Noise levels are generally expressed in decibels (see below, section A1.2) and are measured with a sound level meter (see Appendix A5).

A1.2 Noise levels and the decibel

The decibel is the logarithm of the ratio between two values of some characteristic quantity such as power, pressure or intensity, with a multiplying constant to give convenient numerical factors. Logarithms are useful for compressing a wide range of quantities into a smaller range. For example:

— lg 10 = 1

— lg 100 = 2

— lg 1000 = 3

Hence the ratio 1000:10 is compressed into a ratio of 3:1.

This approach is advantageous for handling sound levels, where the ratio of the highest to the lowest sound likely to be encountered, is as high as 1 000 000:1. A useful development, many years ago, was to take the ratios with respect to the quietest sound which can be heard. This is the threshold of hearing, which is at about 20 μPa (i.e. 2×10^{-5} Pa) of pressure or 10^{-12} W·m^{-2} of intensity for the average person.

When the word 'level' is added to the word for a physical quantity, decibel levels are implied, denoted by L_X, where the subscript 'X' is the symbol for the quantity.

All intensity levels are expressed as follows:

$$L_I = 10 \lg (I / I_0) \qquad \text{(A1.2)}$$

where L_I is the intensity level (dB), I is the measured intensity (W·m^{-2}) and I_0 is the reference intensity (i.e. 10^{-12}) (W·m^{-2}).

However, it can be shown that intensity is proportional to the square of pressure, giving the pressure level, as follows:

$$L_p = 10 \lg (p / p_0)^2 = 20 \lg (p / p_0) \qquad \text{(A1.3)}$$

where L_p is the pressure level (dB), p is the measured pressure (Pa) and p_0 is the reference pressure (i.e. 2×10^{-5}) (Pa).

This is the formulation most commonly used and, by substituting the numerical value of p_0 (i.e. 2×10^{-5} Pa) and taking logarithms, leads to the decibel level as:

$$L_p = (20 \lg p) + 94 \qquad \text{(A1.4)}$$

Thus if $p = 1$ Pa, the sound pressure level is 94 dB.

Similarly, equation A1.2 leads to:

$$L_I = (10 \lg I) + 120 \qquad \text{(A1.5)}$$

Note that if the sound pressure is doubled, that is $p \rightarrow 2\,p$, L_p increases by 6 dB. If the sound intensity is doubled, that is $I \rightarrow 2\,I$, L_I increases by 3 dB. This is because $I \propto p^2$.

Although the sound pressure is the quantity most frequently measured (see Appendix A5) it is not the most fundamental property of the source. Sound represents a flow of energy propagated from the source. The source acts as a reservoir of power (rate of production of energy, i.e. $J{\cdot}s^{-1}$ or watts). At a distance from the source we detect the flow of energy at our location. This flow is the acoustic *intensity*, the energy flow in watts per square meter.

The intensity can be measured directly, but we normally measure sound pressure. The reason for this is that measuring instruments, such as microphones, employ a diaphragm which, deflecting under the fluctuating force of the sound wave, converts its deflection to an electrical signal. Since pressure = force/area, it is the sound pressure which applies a force to the diaphragm.

The reference levels for pressure and intensity are close to normal thresholds of hearing, so that the threshold is approximately 0 dB. A big advantage of these choices of reference level is that the decibel values of sound pressure level and sound intensity level are the same for the same sound, to within about 0.5 dB.

Both intensity and pressure define what is occurring at a point in space. The more fundamental quantity is the sound power of the source, expressed in watts (i.e. joules per second). Acoustic power levels are very low wattage and are given in decibels as:

$$L_W = 10 \lg (P / P_0) \qquad \text{(A1.6)}$$

where L_W is the sound power level (dB), P is the sound power (W) and P_0 is the reference power (i.e. 10^{-12}) (W).

Then:

$$L_W = (10 \lg P) + 120 \qquad \text{(A1.7)}$$

(Some earlier texts used 10^{-13} W as the reference pressure, as it resulted in more convenient formulae when using older systems of units, but this was changed to 10^{-12} W for use with SI units.)

Equation A1.7 shows that an acoustic power of 1 W is a sound power level of 120 dB. Note that if the sound power is doubled, that is $P \rightarrow 2\,P$, L_W increases by 3 dB.

If the source is small compared with the wavelength, it approximates to a point source. The inverse square law of radiation then applies, similar to sources of light, and at a distance r the intensity is:

$$I = P / (4\,\pi\,r^2) \qquad \text{(A1.8)}$$

where I is the intensity ($W{\cdot}m^{-2}$), P is the sound power (W) and r is the distance from the source (m).

By substituting numerical values, the intensity can be expressed as an intensity level in decibels as:

$$L_I = L_W - (20 \lg r) - 11 \qquad \text{(A1.9)}$$

where L_I is the sound intensity level (dB), L_W the sound power level (dB) and r is the distance from the source (m).

But, as the decibel sound pressure levels and sound intensity levels are numerically the same, this can immediately be written in the more familiar pressure terms as:

$$L_p = L_W - (20 \lg r) - 11 \qquad \text{(A1.10)}$$

If the sound is constrained into a hemisphere by reflecting surfaces, the surface area for radiation changes from $(4\,\pi\,r^2)$ to $(2\,\pi\,r^2)$ and the sound pressure level becomes:

$$L_p = L_W - (20 \lg r) - 8 \qquad \text{(A1.11)}$$

The (20 lg r) term for free propagation, either spherical or hemispherical, means that if distance r is doubled or halved the pressure level change is 6 dB (i.e. 20 lg 2 = 6).

These equations illustrate the importance of the sound power of the source as the fundamental quantity. Sound power is in watts (or sound power level, see equation A1.6) whilst the sound pressure at a point is what we perceive after the sound travelling from the source has been modified by propagation effects.

If the sound is constrained in a duct it does not spread out and so does not reduce in intensity as the distance increases. The intensity ($W{\cdot}m^{-2}$) in a duct of constant cross-section will remain constant in the absence of losses. However, losses always occur by breakout, energy transfer to duct wall vibration, absorption etc. The internal noise level in a duct section near to the fan is typically 90–100 dB and proper design reduces it to around 35 dB in the room. By using the equations for sound intensity and sound power, it follows that a sound pressure level/sound intensity level of 96 dB in a duct of 0.5 m^2 cross sectional area, equates to the sound power into the duct of (2×10^{-3}) W (i.e. two milliwatts). A general relation for sound intensity, area and sound power is:

$$L_W = L_I + 10 \lg A_d \qquad \text{(A1.12)}$$

where A_d is the cross sectional area of the duct (m^2).

Then for the example above, where the area is 0.5 m^2, sound power in the duct is:

$$L_W = 96 + 10 \lg 0.5 = 93 \text{ dB}$$

Working backwards and converting sound power level (dB) to sound power (W) using equation A1.6:

$$P = 10^{-12} \text{ antilg } 9.3 = 2 \times 10^{-3} \text{ W}$$

However, as the decibel levels for pressure and intensity are equal, equation A1.12 can also be written as:

$$L_W = L_p + 10 \lg A_d \quad \text{(A1.13)}$$

The sound power level is not numerically the same as the sound pressure or intensity level (except for ducts having a cross sectional area of 1 m^2, since 10 lg A_d is then zero). Pressure and intensity levels are affected by the propagation path.

When calculating duct system noise, the fundamental quantity is the sound power entering or leaving a duct element. This continues up to the duct termination, which acts as a source of power into the room, from which the room sound level is predicted, see section 7.

Equation A1.12 is also useful for estimating the level at a distance from an extended source, such as boiler house louvres or an opening in a wall. If the sound pressure level averaged over the extended source is known, combining this with the area gives the sound power level, which can then be used with equation A1.11 to determine the level at a distance. However, large sources are directional, so that equation A1.11 will underestimate at higher frequencies in directions to the front of the source.

A1.3 Addition and subtraction of decibels

As decibels are logarithmic ratios they do not add arithmetically. It is necessary to convert back to the original physical units, add or subtract values of intensity or $(\text{pressure})^2$ and then re-convert the result back to decibels. This means first converting the decibel levels, N, using antilogs (i.e. $10^{N/10}$), summing (or subtracting, as appropriate), and then taking the logarithm of the sum (or difference) (i.e. 10 lg (sum or difference)).

Example A1.1

(*a*) Add 55 dB and 57 dB sound pressure levels

From equation A1.3:

$$55 \text{ dB} = 20 \lg (p_1 / p_0) = 10 \lg (p_1 / p_0)^2$$

Then:

$$(p_1 / p_0)^2 = \text{antilg } 5.5 = 10^{5.5}$$

Similarly:

$$(p_2 / p_0)^2 = \text{antilg } 5.7 = 10^{5.7}$$

The sum of the sound pressure levels is:

$$10^{5.5} + 10^{5.7} = 316228 + 501187 = 817415$$

Hence, the sum of the two levels, in dB, is:

$$10 \lg (817415) = 59.1 \text{ dB}$$

which would normally be rounded to 59 dB.

(*b*) Subtract 60 dB from 61 dB

The difference is:

$$10^{6.1} - 10^6 = 1\,258\,925 - 1\,000\,000 = 258\,925$$

Hence:

$$10 \lg (258\,925) = 54 \text{ dB}$$

This shows that if a noise of level 60 dB has one of level 54 dB added to it, the resulting level is 61 dB.

It is necessary to work back to the original quantities, e.g. $(\text{sound pressure})^2$, which is a measure of the energy in the sound wave, add them, and then work forward to determine the total decibel level. (A programmable calculator is useful if such calculations need to be undertaken frequently.)

To simplify addition and subtraction Figure A1.1 can be used. For addition, the scales are used as shown. For subtraction, enter the difference between the two levels on the upper scale. The amount to be subtracted from the smaller decibel level is given on the bottom scale.

Example A1.2

The calculations performed in Example A1.1 are carried out using Figure A1.1, as follows.

(*a*) Add 55 dB and 57 dB sound pressure levels:

— bottom scale: difference between levels to be added is (57 – 55) = 2 dB

— top scale: corresponding value to be added to the higher level is 2.1 dB

— sum is (57 + 2.1) = 59.1 dB.

(*b*) Subtract 60 dB from 61 dB:

— top scale: difference between levels to be subtracted is (61 – 60) = 1 dB

— bottom scale: corresponding value to be subtracted from the smaller level is 6 dB

— difference is (60 – 6) = 54 dB.

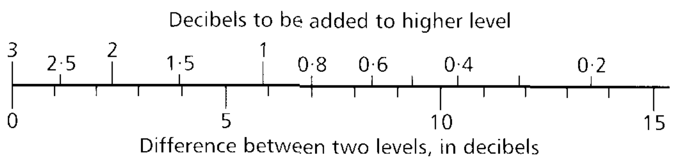

Figure A1.1 Line chart for addition of sound pressure levels in dB

Appendix A2: Generic formulae for predicting noise from building services plant

A2.1 Fans

There have been many investigations of fan noise, but a generally satisfactory prediction formula has not been found. It is known that fan noise depends on the air volume flow rate, fan pressure and fan operating point. An approximate prediction formula which may be used for initial work is:

$$L_W = K_W + 10 \lg Q + 10 \lg p_f + C \qquad \text{(A2.1)}$$

where L_W is the estimated fan sound power level (dB re. 10^{-12} W), K_W is a constant (see Table A2.1) (dB), Q is the volume flow rate ($m^3 \cdot s^{-1}$), p_f is the fan pressure (Pa), C is a correction factor for the fan operating point (see Table A2.2) (dB).

Although equation A2.1 was formerly recommended by ASHRAE, it has since been withdrawn because of its inaccuracies. It is provided here for initial guidance only. Once the fan has been selected, the manufacturer's noise data should be used.

A2.2 Chillers and compressors

The primary sources of noise are the compressors and drive motors. The following relations give the overall A-weighted sound pressure level (see Appendix A5, section A5.2) at 1 m[A2.1].

For centrifugal compressors:

$$L_{pA} = 54 + 11 \lg P_c \qquad \text{(A2.2)}$$

For reciprocating compressors:

$$L_{pA} = 66 + 9 \lg P_c \qquad \text{(A2.3)}$$

where L_{pA} is the A-weighted sound pressure level at 1 m (dB) and P_c is the electrical power input to the compressor (kW).

A2.3 Cooling towers

The overall sound power level is given by:

$$L_W \approx 12 + 10 \lg P \qquad \text{(A2.3)}$$

where L_W is the sound power level (dB) and P is the fan sound power (W).

Octave band levels are given approximately by allocating the 63 Hz band a level 4 dB below the overall level and reducing each successive octave band by 2 dB.

A2.4 Regeneration of noise by duct components

This method, which is based on recent work at Liverpool University[A2.2], is applicable to in-duct components such as bends, transition pieces and take-offs, but is not applicable to termination devices. It is based upon a theoretical model for rectangular ductwork but is also applicable to circular section ductwork.

For a particular element, the required information is pressure loss factor, ζ, obtained from section 4 of CIBSE Guide C[A2.3] or other data source, duct cross sectional dimensions and air velocity. It is necessary to determine an approximate value of the clear area ratio (σ) and a characteristic dimension (d).

These can be estimated from pressure loss factor as follows.

The clear area ratio is given by:

$$\sigma = \frac{\zeta^{1/2} - 1}{\zeta - 1} \qquad \text{(A2.5)}$$

where σ is the clear air ratio and ζ is the pressure loss factor.

Table A2.1 Values of K_W in equation A2.1

Fan type	Correction factor K_W / dB for stated octave band / Hz							Blade frequency increment* / dB
	63	125	250	500	1000	2000	4000	
Centrifugal:								
— forward curved	38	38	28	30	20	15	10	3
— all other types	30	30	28	24	19	13	10	3
Propeller	33	36	43	41	40	37	30	5
Vane axial for stated hub ratio:								
— 0.3 to 0.4	34	28	28	33	32	30	23	6
— 0.4 to 0.6	34	28	31	28	26	21	15	6
— 0.6 to 0.8	38	37	36	36	34	32	28	6
Tube axial for stated wheel diameter:								
— over 1000 mm	36	31	32	34	32	31	24	7
— under 1000 mm	33	32	34	38	37	36	28	7

* The blade frequency increment is added into the octave band that contains the blade passage frequency, which is given by multiplying the number of blades by fan speed.

Table A2.2 Values of C in equation A2.1

Fan efficiency / %	C / dB
Peak	0
80	6
70	9
50	15

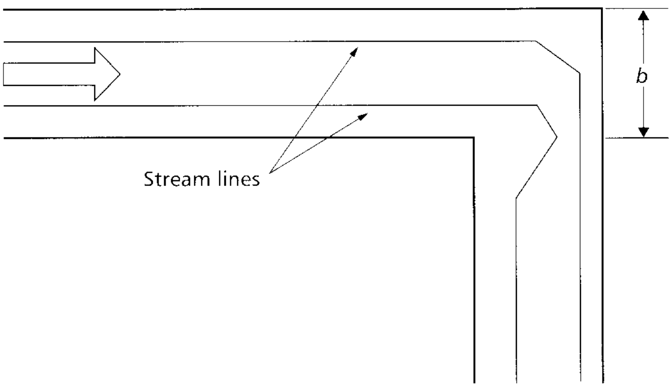

Figure A2.1 Constriction in a duct

The characteristic dimension is given by:

$$d = b\,(1 - \sigma) \qquad \text{(A2.6)}$$

where d is the characteristic dimension (m) and b is the duct dimension in the direction of flow constriction (see Figure A2.1) (m).

Figure A2.1 illustrates how the flow of air is constrained in one direction when it encounters a mitred bend. Similar effects are observed for other in-duct elements.

A graph giving σ in terms of ζ is shown in Figure A2.2. There are different expressions for noise regeneration below and above the duct 'cut-on' frequency, which is the frequency above which complex acoustic modes are propagated in the duct; propagation is as a plane wave below the cut-on frequency.

The cut-on frequency is given by:

$$f_c = \frac{c}{2\,l} \qquad \text{(A2.7)}$$

where f_c is the cut-on frequency (Hz), c is the velocity of sound (m·s^{-1}) and l is the longest duct cross sectional dimension (m).

Where the required octave band frequency, f_0, is below cut-on (i.e. $f_c > f_0$), the sound power generated by the fitting is:

$$L_W = -37 + 20\lg(K(St)) + 20\lg\zeta + 10\lg A + 40\lg u \qquad \text{(A2.8)}$$

where L_W is the sound power level (dB), $K(St)$ is an experimentally determined factor related to the Strouhal number, ζ is the pressure loss factor, A is the cross sectional area of the duct and u is the air velocity in the duct (m·s^{-1}).

For $f_c < f_0$, where the required octave band frequency, f_0, is above the cut-on frequency:

$$L_W = -84 + 20\lg(K(St)) + 20\lg(St) + 10\lg\zeta - 40\lg\sigma + 10\lg A + 60\lg u \qquad \text{(A2.9)}$$

where (St) is the Strouhal number.

The term 20 lg ($K(St)$) is determined as follows. $K(St)$ is an experimentally determined factor, see Figure A2.3, where the vertical axis on the curve is 20 lg ($K(St)$) and the horizontal axis is the Strouhal number, (St).

So, a value for 20 lg ($K(St)$) may be obtained if the Strouhal number is known. (*Note*: Figure A2.3 is based on data for a variety of air velocities in two different duct sizes and has an accuracy of around ± 2 dB. Similar curves have been obtained for other elements.)

The Strouhal number is given by the equation:

$$(St) = f\,d\,/\,v_c \qquad \text{(A2.10)}$$

where (St) is the Strouhal number, f is the frequency, d is the characteristic dimension (m) (see equation A2.6) and v_c is the constriction velocity (m·s^{-1}).

The constriction velocity is given by:

$$v_c = u\,/\,\sigma \qquad \text{(A2.11)}$$

where u is the air velocity in the duct (m·s^{-1}) and σ is the clear air ratio (see equation A2.5).

The procedure for determining the regenerated noise level for a given octave band frequency is as follows:

— calculate (St) from d and σ (both functions of ζ) and air velocity u

— read off appropriate value of 20 lg ($K(St)$) from Figure A2.3

— insert values in the above equations depending on whether the octave band frequency, f_0, is below or above duct higher mode cut-on, $f_c = c/(2\,l)$ (equation A2.7).

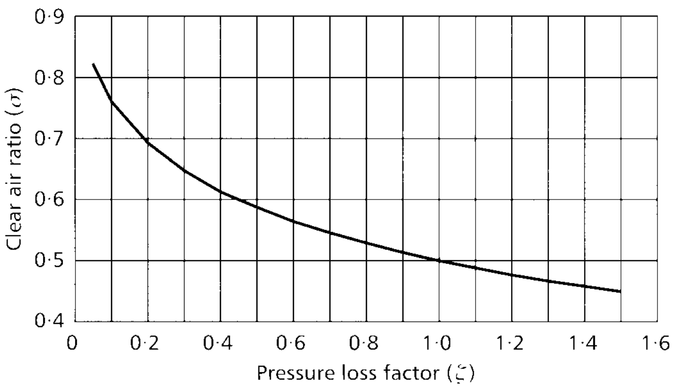

Figure A2.2 Determination of clear air ratio from pressure loss factor

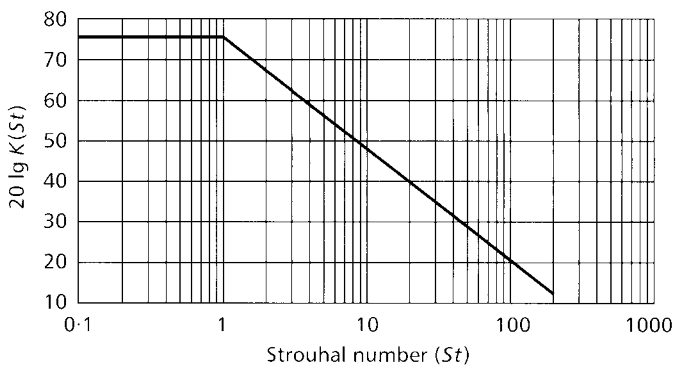

Figure A2.3 Determination of term 20 lg ($K(St)$) from Strouhal number (accuracy approx. ± 2 dB)

Other terms in the equations are physical constants or duct dimensions.

Thus, from knowledge of pressure loss coefficient, duct dimensions and air velocity one can calculate the sound power level for any chosen octave band centre frequency.

Example A2.1

A duct element of 600 mm by 400 mm, where the 400 mm dimension is in the direction of the constricted flow, has pressure loss factor (ζ) of 1.25, and a clear air ratio (σ) of 0.47.

Given the duct dimension in the direction of constricted flow as 400 mm, from equation A2.6 the characteristic dimension is:

$$d = 0.4\,(1 - 0.47) = 0.21 \text{ m}$$

For an air velocity (u) of 15 m·s^{-1}, the constriction velocity is obtained from equation A2.11:

$$v_c = 15 / 0.47 = 31.9 \text{ m·s}^{-1}$$

From equation A2.7, the duct cut-on frequency for higher modes is:

$$f_c = 345 / (2 \times 0.6) = 288 \text{ Hz}$$

Then, from equation A2.10, for f_o = 125 Hz, the Strouhal number is:

$$(St) = 125 \times 0.21 / 31.9 = 0.82$$

Estimating from Figure A2.3, a Strouhal number of 0.82 gives 20 lg ($K(St)$) = 75. Then, as the frequency is below cut-on, equation A2.8 applies, i.e:

$$L_W = -37 + 75 + 20 \lg 1.25 + 10 \lg 0.24 + 40 \lg 15$$

$$= -37 + 75 + 2 - 6 + 47 = 81 \text{ dB}$$

The octave band sound pressure level for the 125 Hz octave band is thus 81 dB.

At 1000 Hz, which is above the duct cut-on frequency, the Strouhal number is 6.6, leading to 20 lg ($K(St)$) ≈ 53.

Equation A2.9 then applies, i.e:

$$L_W = -84 + 53 + 20 \lg 6.6 + 10 \lg 1.25$$

$$- 40 \lg 0.47 + 10 \lg 0.24 + 60 \lg 15$$

$$= -84 + 53 + 16 + 1 + 13 - 6 + 71 = 64 \text{ dB}$$

The octave band sound pressure level for the 1000 Hz octave band is thus 64 dB.

Similar calculations may be carried out at other frequencies. A spreadsheet can also be developed.

A2.5 Duct terminations

These are airflow noise sources situated on the room boundary and could result in an audible hissing noise.

The preferred source of noise data is the manufacturer, relating the air velocity and diffuser configuration to the sound power generated by the flow. In the absence of such data an initial estimate of the noise may be obtained from a detailed study of noise from diffusers, summarised by Beranek and Ver[A2.4], and calculation algorithms in *Sound and vibration design and analysis*[A2.1].

Overall sound power is determined as:

$$L_W = 10 + 10 \lg A_d + 30 \lg \xi + 60 \lg u \qquad \text{(A2.12)}$$

where L_W is the sound power level (dB), A_d is the area of the duct cross-section prior to the diffuser, ξ is a normalised pressure drop coefficient (see equation A2.13) and u is the air velocity upstream of the diffuser (m·s^{-1}).

The normalised pressure drop coefficient (ξ) is given by:

$$\xi = 2\,\Delta p / \rho\, u^2 \qquad \text{(A2.13)}$$

where Δp is the pressure drop across the diffuser (Pa).

Pressure drop increases with constriction in the diffuser, such that pressure drop coefficients from about 3 to 20 cover a wide range of diffusers.

The diffuser spectrum is typically a broad band with a peak which spans an octave band and falls off at about 3 dB/octave at lower frequencies than the peak and 5 dB/octave at higher frequencies. The frequency of the peak is the air velocity in the duct multiplied by 160 and is shown in Table A2.2 for the octave band in which the peak will be located.

Expanding the pressure drop coefficient and inserting numerical data enables equation A2.12 to be written as:

$$L_W = 10 + 10 \lg A_d + (30 \lg \Delta p) + 5 \qquad \text{(A2.14)}$$

where the pressure loss is obtained from manufacturers' information for the air velocity.

The octave band level at the peak frequency is lower than the total level and given approximately by:

$$L_{W(\text{peak})} \approx 10 + 10 \lg A_d + 30 \lg \Delta p \qquad \text{(A2.15)}$$

Table A2.2 Peak frequencies for diffuser noise

Velocity u / m·s^{-1}	Peak octave band / Hz
10	2000
9	2000
8	1000
7	1000
6	1000
5	1000
4	500
3	500
2	250
1	125

Diffuser noise is then obtained by determining the level of the peak using equation A2.15 whilst Table A2.2 gives the octave band in which this occurs. Octave band levels at frequencies higher and lower than the peak frequency are given approximately by deducting 3 dB/octave at lower frequencies and deducting 5 dB/octave at higher frequencies.

This is a simplified approach to the fuller treatment(A2.1) and Beranek and Ver(A2.4) and is less accurate than these, which claim to give the level to within 5 dB of measurement.

Excessive noise is controlled by increasing the diffuser area or by using a greater number of diffusers, which also reduces the velocity and pressure drop.

A2.6 Lined ducts

Prediction formulae for attenuation in lined rectangular ducts include the ratio p_d/A_d, where p_d is the duct perimeter and A_d is its cross sectional area. Other factors are the frequency of the sound and the thickness of the lining. Detailed studies have been made of insertion loss of lined ducts(A2.5). An empirical prediction equation(A2.1,A2.6) is:

$$\text{IL} = 3.28\ 10^{A}\ (0.305\ p_d / A_d)^{B}\ (0.039\ t)^{C} \qquad \text{(A2.16)}$$

where IL is the insertion loss (dB), A, B and C are coefficients (see Table A2.3), p_d is the duct perimeter (m), A_d is the duct cross-sectional area (m^2) and t is the lining thickness (mm).

The information on which this equation is based was for a limited range of variables. For example, the ratio p_d/A_d varied between about 2 and 30 and t may be either 25 mm or 50 mm, with a density between 24 and 48 kg·m^{-3}. (It was found that insertion loss is not sensitive to lining density.)

Table A2.3 Coefficients for equation A2.16 (reproduced from ARI Standard 885-1998(A2.6) by permission of the Air-Conditioning and Refrigeration Institute)

Octave band centre frequency / Hz	A	B	C
125	–0.865	0.723	0.375
250	–0.582	0.826	0.975
500	–0.0121	0.487	0.868
1000	0.298	0.513	0.317
2000	0.089	0.862	0
4000	0.0649	0.629	0
8000	0.150	0.166	0

References (Appendix A2)

A2.1 *Sound and vibration design and analysis* (Gaithersburg MD: National Environmental Balancing Bureau) (1994)

A2.2 Waddington D C and Oldham D J Generalized flow noise prediction curves for air duct elements *J. Sound and Vibration* 222 163–169 (1999)

A2.3 *Flow of fluids in pipes and ducts* Section 4 in *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

A2.4 Beranek L L and Ver I L (eds.) Noise and Vibration Control Engineering (Chichester: Wiley Interscience) (1992)

A2.5 Kuntz H L and Hoover R M The interrelationships between the physical properties of fibrous duct lining materials and the lined duct sound attenuation *ASHRAE Trans.* **93** (2) 449-470 (1987)

A2.6 *Procedure for estimating occupied space sound levels in the application of air terminal units* ARI Standard 885: 1998 (Arlington VA: Air-Conditioning and Refrigeration Institute) (1998)

Appendix A3: Interpreting manufacturers' noise data

Manufacturers' noise data should be available for fans, silencers, fan coil units, dampers, VAV boxes etc. Interpretation of the data requires close reading of accompanying information, in order to discover which standards were used in the measurements, and to understand the symbols and subscripts employed. Noise data are required from the 63 Hz octave band to the 4000 Hz band. However, some data do not yet go below 125 Hz. Generally, if a manufacturer cannot supply noise data, certified as measured according to accepted standardised procedures, an alternative supplier should be considered.

The data available may be expressed in a number of forms, e.g. as 'A-weighted' (dBA), as 'unweighted octave band' or as 'A-weighted octave band'. Additionally, a single decibel level might be allocated, e.g. to a fan, and corrections in tabular or graphical form given in order to derive the octave band levels.

Each statement of noise might be expressed as either 'sound pressure' or 'sound power' (see Appendix A1). The distinction between pressure and power is often shown as a subscript. For example, L_p is sound pressure, whilst L_W is sound power. Some manufacturers give A-weighted levels, shown as L_{pA} or L_{WA}, which might be either the total level or A-weighted octave band levels. Propeller fans, which could be located on the perimeter of an occupied space, may be given as an A-weighted sound pressure level at 1 m, 3 m or 4 m, and shown, for example, as L_{pA4m}. Some American manufacturers express the noise as a loudness level in 'sones', but this is limited to fans which discharge directly into occupied areas.

The inlet and discharge noise should be identified separately. Formerly it was assumed that the noise split equally between inlet and discharge but it is now known that the discharge sound power may be several decibels greater than the inlet. The casing sound power breakout should also be known.

Practices differ between countries. UK manufacturers are likely to give unweighted octave band levels, whilst some Continental companies give A-weighted octave band levels, which is permitted in their national standards. Silencer attenuation may be given as an overall A-weighted reduction. This must be viewed cautiously, since the frequency dependence of silencer attenuation leads to its A-weighted performance being input spectrum dependent. Therefore, do not make, or accept, assumptions, but read the data carefully.

Standard measurement methods attempt to remove extraneous factors, of which poor airflow is the most likely to occur in an installation. Consequently fans and silencers are measured under ideal conditions[A3.1,A3.2], with care taken to ensure good airflow. Different conditions in field installations may affect the noise attenuation of a silencer and the noise generation of a fan and duct fittings.

Despite careful measurements, there are uncertainties in the published data of about ±5 dB at 63 Hz for fans and less than this for silencers. Uncertainties reduce at higher frequencies. Not all published data are directly measured; some are extrapolated from measurements on representatives of a family of equipment types.

A full discussion of manufacturers' data is given in *Application of Manufacturers' Sound Data*[A3.3].

References (Appendix A3)

A3.1 BS EN 25136: 1994: *Determination of sound power radiated into a duct by fans — In-duct method* (London: British Standards Institution) (1994)

A3.2 BS EN ISO 7325: 1996: *Acoustics — Measurement procedures for ducted silencers — Insertion loss, flow noise and total pressure loss* (London: British Standards Institution) (1996)

A3.3 Ebbing C and Blazier W E *Application of Manufacturers' Sound Data* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1998)

Appendix A4: Basic technique for prediction of room noise levels from HVAC systems

A4.1 Prediction

In the prediction, each component of the HVAC system is considered separately. Components of interest might be the fan, plenum, duct, branch, elbow etc., finally leading to the duct termination at the room. Additionally, there may be breakout and regenerated noise. Figure A4.1 is used to illustrate the prediction, where it is required to predict the noise in the room at a distance of 2 m from the duct termination.

The inlet (A) goes to the fan (B), which discharges into a 900 mm by 750 mm duct, 10 m long, between (B) and the first branch at (C), supplying a 600 mm by 600 mm branch serving other parts of the building. The main duct continues as 600 mm by 600 mm for 5 m to a second branch at (D). The branch at (D) is a 3 m run of 600 mm by 450 mm duct to elbow (E), whilst the main duct continues at 600 mm by 600 mm to (G) and beyond. At (E), a further 600 mm by 450 mm run, 3 m long, leads to a 600 mm by 450 mm duct termination at (F), to supply a room 10 m × 8 m × 3 m. All ducts, branches and the elbow are unlined.

This hand calculation has been given to illustrate the procedures, but use of a computer program is recommended. The results from different programs may differ by one or two decibels, depending on the data and calculation processes built into the program. In selecting a program, make sure that it has been rigorously validated and is guaranteed to conform to a recognised calculation procedure

Table A4.1 shows the calculation, in which breakout noise is not included because there is no breakout path into the room.

The following points should be noted with respect to Table A4.1:

(1) Fan (Table A4.1, row 1): the fan sound power is not given at 31.5 Hz. It is rare for sound power at this frequency to be known. However, as 31.5 Hz is included in criteria, the duct design must avoid turbulence and pressure losses which might lead to generation of low frequencies.

(2) Straight duct between (B) and (C) (Table A4.1, row 2): see Table 6.1

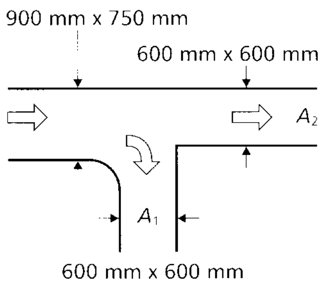

Figure A4.2 Example A4.1: branch dimensions for noise calculation

(3) Branch at (C) (Table A4.1, row 3): see Figure A4.2.

The attenuation of noise power into the continuing main duct is then given by equation 6.1:

$$\Delta L = 10 \lg (0.72 / 0.36) = 3 \text{ dB}$$

Since $A_1 = A_2$, the attenuation of noise power into the branch will also be 3 dB. (See also Figure 6.3.) These branch attenuations are constant across the frequency range.

(4) Straight duct between (C) and (D) (Table A4.1, row 4): see Table 6.1.

(5) Branch at (D) (Table A4.1, row 5): the areas of the ducts after the branch are 0.36 m^2 and 0.27 m^2, giving an area ratio of (0.63 / 0.27) = 2.33; with the attenuation in the smaller duct required, equation 6.1 gives 3.7 dB attenuation, rounded to 4 dB (as in Table A4.1, row 5).

(6) Straight duct between (D) and (E) (Table A4.1, row 6): see Table 6.1.

(7) Elbow at E (Table A4.1, row 7): see Table 6.3.

(8) Straight duct between (E) and (F) (Table A4.1, row 8): see Table 6.1.

(9) Termination (Table A4.1, row 9): for a rectangular duct 600 mm × 450 mm, equation 6.3 (see section 6.6) gives an effective diameter of 586 mm, rounded to 600 mm. Table 6.6 gives end reflection losses for various effective diameters.

(10) Room effect (Table A4.1, row 10): room volume is 240 m^3; Table 7.1 (see section 7.2) gives values of attenuation for rooms of various volumes.

The final row in Table A4.1 is the sound level at the occupier, which must be compared with the specified criterion.

Section 1 of CIBSE Guide A[(A4.1)] suggests a criterion of NR35 for a general office, but this rating will not be satisfactory for noises with a high content of low frequencies.

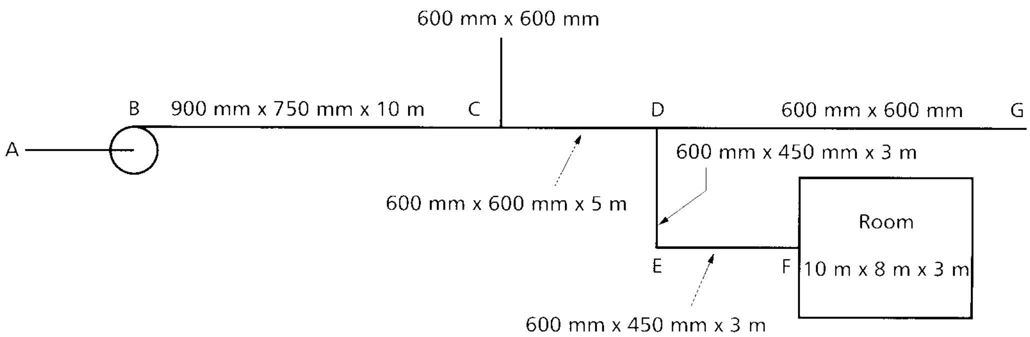

Figure A4.1 Example A4.1: ductwork system schematic

Table A4.1 Room noise prediction for example calculation

Item		Sound power level / dB for stated octave band / Hz							
		63	125	250	500	1000	2000	4000	8000
1	Fan	97	94	99	93	85	74	68	60
2	Rectangular duct (900 mm × 750 mm × 10 m)	–6	–4	–3	–1	–1	–1	–1	–1
3	Branch duct (50% continuation)	–3	–3	–3	–3	–3	–3	–3	–3
4	Rectangular duct (600 mm × 600 mm × 5 m)	–3	–2	–1	–1	–1	–1	–1	–1
5	Branch duct (42% continuation)	–4	–4	–4	–4	–4	–4	–4	–4
6	Rectangular duct (450 mm × 600 mm × 3 m)	–2	–1	–1	0	0	0	0	0
7	Rectangular mitred elbow (600 mm)	0	–1	–5	–8	–4	–3	–3	–3
8	Rectangular duct (450 mm × 600 mm × 3 m)	–2	–1	–1	0	0	0	0	0
9	End reflection (600 mm × 450 mm)	–8	–4	–1	0	0	0	0	0
10	Room (10 m × 8 m × 3 m); occupant 2 m from duct termination	–8	–9	–10	–11	– 12	–13	–14	–15
Sound level at occupier / dB		61	65	70	65	60	49	42	33

Table A4.2 Room levels compared with various criteria

Criterion	Room level and attenuation / dB for stated octave band / Hz							
	63	125	250	500	1000	2000	4000	8000
Room (unsilenced)	61	65	70	65	60	49	42	33
NR35	63	52	45	39	35	32	30	28
NC35	60	52	45	40	36	34	33	32
RC35	55	50	45	40	35	30	25	—
Silencer attenuation	8	12	25	38	39	23	16	10
Silenced room level	53	53	45	27	21	26	26	23

If the duct passes across the ventilated space, breakout noise should be calculated and the resulting levels in the room compared with those from the duct borne noise, in order to estimate whether extra attention has to be given to the breakout noise. Similarly, breakout noise from units above the ceiling must be considered as additional sources of noise into the room (see sections A4.2 and 6.10).

Table A4.2 compares the room levels obtained above with several criteria, where the attenuation required is the difference between the room level and the criterion. The criteria are described in Appendix A8. Clearly, the RC criteria (both Mark I and Mark II) are the most stringent at low frequencies, but there are only small differences between all the criteria in the range from 125 Hz to 2000 Hz. Although lower frequencies than 63 Hz occur in the criteria, calculation data for these frequencies are not normally available.

The attenuation of a commercial silencer to generally meet the requirements is also shown in Table A4.2. The resulting room levels are given in the final row of Table A4.2. The silencer is a 2.1 m long absorptive silencer, with cross-sectional dimensions of 900 mm × 750 mm, which should be fitted into a straight section of the main duct, not too close to the fan discharge and preferably where the duct penetrates the plant room wall. The pressure loss is 70 Pa for a fan delivery of 5 $m^3 \cdot s^{-1}$. A silencer is often chosen to meet requirements at the most critical frequency, typically 125 Hz or 250 Hz. Satisfying the attenuation at this frequency may lead to over-attenuation at higher frequencies.

In addition to its attenuation, factors to be considered in the selection of a silencer include airflow, pressure loss and size. For example, when the fan in the example gives 5 $m^3 \cdot s^{-1}$, the air velocity in the main duct is 7.4 $m \cdot s^{-1}$, which is an acceptable velocity, resulting in 70 Pa pressure loss, as above. However, if the same silencer is used on a 10 $m^3 \cdot s^{-1}$ fan, the doubling of velocity leads to a quadrupling of pressure loss.

The main attenuation at low frequencies is from duct losses, including breakout, and end reflection loss. At 63 Hz these total 21 dB, which is greater than the silencer attenuation at this frequency. If breakout noise was a problem, requiring stiffening of the duct or change to a circular duct, the duct breakout attenuations might not be obtained, giving a potential room level of 66 dB. For example, compare Tables 6.1 and 6.2.

Table A4.2 shows that the room levels comply closely with the NR35 and NC35 criterion levels. The A-weighted room level is 40 dBA. However, this is an example of how a criterion can be met but the room noise may not give an acceptable acoustic environment. The rapid fall in room level between 250 Hz and 500 Hz, due largely to the silencer attenuation at 500 Hz, unbalances the spectrum. In practice, office activity noise may fill in the levels at mid frequencies and help to balance the spectrum, but this cannot be relied upon, and will not be applicable to commissioning measurements. A problem with commissioning measurements is that, in practice, the noise level will vary over the room space. The measurement positions for commissioning should be specified in the contract.

Noise that is judged to be satisfactory by NR and NC criteria may have inherent problems exposed by the more detailed RC criterion and its application to quality assessment. See ASHRAE Handbook: *HVAC Applications*(A4.2) for further details.

A4.2 Breakout travelling through the ceiling

For breakout from ducts, the duct sound power is determined as in section 6.10, and the effect in the room is estimated as described in section 7, taking attenuation through the ceiling into account.

For casing breakout, the manufacturer of boxed equipment, such as a fan coil unit or similar, will supply information on casing breakout sound power. This is used to determine room noise levels as above.

A4.3 Regenerated noise

All duct sections, straight or with bends, are potential sources for regenerated noise, depending on the velocity. Manufacturers of in-duct components provide data to enable regenerated noise to be estimated. The data are often stated in terms of the product of velocity and pressure loss, leading to an overall sound power figure, which is then corrected in a specified way in order to give octave band sound power levels. The regenerated sound power levels must be compared with the in-duct sound power levels from the fan at the point of origin of the regenerated noise, in order to estimate their significance. Table 5.1 may also be used to give an estimate of regenerated noise from low turbulence duct fittings. The calculations in Appendix A2, sections A2.4 and A2.5, may also be used.

A4.4 Multiple sources

A room will normally have more than one noise source from the HVAC systems, e.g. multiple duct outlets, breakout from a box or a duct in the ceiling void etc. It is the summation of the noise from these sources which must meet the criterion, where summation is carried out as in Appendix A1.3. Consequently, where there are multiple sources, the noise from each must be lower than the criterion and, for example, the silencer attenuation derived above, may need to be increased. Return air ducts must be included

A4.5 Computer predictions

Many organisations have HVAC noise prediction software, either developed in-house or obtained from an outside supplier. Use of a proven system is recommended and it is advisable to carry out some checks on the software against manual calculations.

References (Appendix A4)

A4.1 *Environmental criteria for design* Section 1 in *Environmental design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999)

A4.2 *Sound and vibration control* Chapter 46 in ASHRAE Handbook: *HVAC Applications* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1999)

Appendix A5: Noise instrumentation

A5.1 Sound level meter

This is the most widely used method of measuring noise. Whilst hand-held instruments appear to be easy to use, lack of understanding of their operation and limitations and of the meaning of the varied measurements which they can give, may result in misleading readings. For serious measurements, a sound level meter should only be used by those who have studied the instrument manual, become familiar with the meter and, preferably, had a course of instruction on the use and calibration of the instrument.

The operation of the sound level meter is indicated in Figure A5.1. The microphone output is amplified and passed to weighting networks or electrical filters before being sent on to an indicating instrument.

The electrical filters are an important part of the sound level meter, as they give an indication of the frequency components of the sound. The filters are as follows:

— *A-weighting*: on all meters
— *C-weighting*: on most meters
— *linear (L-weighting)*: on some meters
— *octave filters*: on some meters
— *third octave filters*: on some meters.

The main classes of sound level meter are Type 1 (precision) or Type 2 (general purpose), in which Type 1 is the more accurate. (There is also Type 3 for survey work and preliminary investigations and Type 0 (high precision) mainly for laboratory measurements.)

A5.2 Sound level meter weighting networks

Weighting networks are shown in Figure A5.2.

Originally, the A-weighting was intended for low levels of noise. B-weighting was intended for medium levels of noise. C-weighting was intended for higher levels of noise. The weighting networks were based on human hearing contours at low, medium and high levels and it was hoped that their use would mimic the response of the ear. This concept, which did not work out in practice, has now been lost. A- and C-weighting are used at all levels whilst B-weighting is rarely used. Linear weighting is used to detect low frequencies. A specialist G-weighting is used for infra-sound below 20 Hz.

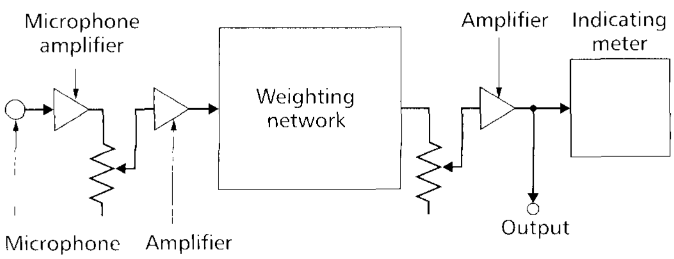

Figure A5.1 Components of a typical sound level meter

The A-weighting depresses the levels of the low frequencies, as the ear is less sensitive to these. This is acceptable if there is largely middle and high frequency noise present, but if the noise is high at low frequencies, the A-weighting does not give a valid measure(A5.1). Low frequencies are often the residual problem in HVAC. A-weighting is adequate for placing noises of similar spectrum (frequency components) in order from worst to best. But, if the spectra are very different, the A-weighting is not reliable as an indicator of subjective response.

A5.3 Equivalent level (L_{eq})

This is the steady level over a period of time that has the same energy as that of the fluctuating level actually occurring during that time. A-weighted equivalent level, designated L_{Aeq}, is used for many legislative purposes.

Mathematically,

$$L_{Aeq} = 10\lg\left[\frac{1}{T_2 - T_1}\int_{T_1}^{T_2}\left(\frac{p_A(t)}{p_0}\right)^2 \mathrm{d}t\right] \qquad \text{(A5.1)}$$

where T_1 is the start time of the noise, T_2 is the end time of the noise, p_0 is the reference pressure of 20 μPa and $p_A(t)$ is the A-weighted instantaneous sound pressure (Pa).

Here we are averaging the fluctuations in the noise and converting to decibels.

Daily noise exposure, $L_{EP,d}$, is a form of equivalent level which is used to assess noise exposure with respect to hearing loss(A5.2). It is given by:

$$L_{EP,d} = 10\lg\left[\frac{1}{T_0}\int_0^{T_e}\left(\frac{p_A(t)}{p_0}\right)^2 \mathrm{d}t\right] \qquad \text{(A5.2)}$$

where T_e is the duration of the exposure to the noise (hours or seconds), T_0 is 8 hours (= 28 800 s), p_0 is the reference pressure of 20 μPa, $p_A(t)$ is the time weighted

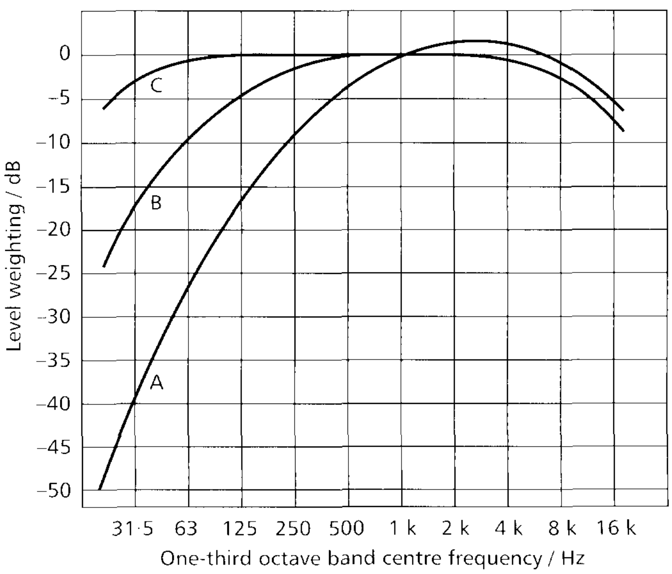

Figure A5.2 Sound level meter weighting networks variation of instantaneous sound pressure in the undisturbed field in air at atmospheric pressure to which the person is exposed, or the pressure in the disturbed field adjacent to the person's head, adjusted to provide the notional equivalent undisturbed field (Pa).

A5.4 Percentiles (L_N)

These are a statistical measure of the fluctuations in noise level, i.e. in the envelope of the noise, which is usually sampled a number of times per second. The most used percentiles are L_{90} and L_{10}. The L_{90} is the level exceeded for 90% of the time and represents a low level in the noise. It is often used to assess background noise. The L_{10} is the level exceeded for 10% of the time and is a measure of the higher levels in a noise. It is often used for assessment of traffic noise. Modern computing sound level meters give a range of percentiles. Note that the percentile is a statistical measure over a specified time interval. Percentiles cannot be combined like decibel levels without knowledge of the statistics of the noise.

A5.5 Frequency analysis

This gives more detail of the frequency components of a noise. Frequency analysis normally uses one of three approaches: octave band, one-third octave band or narrow band.

Octave and one-third octave band filters can be incorporated in sound level meters or be externally connected components. For an octave band filter, the higher limit is twice the lower limit. For a 1/3-octave band filter the higher limit is about 1.28 times the lower limit. A narrow band analysis can be in fractional octaves, such as 1/12-octave or in constant bandwidth analysis using, say, filters of constant bandwidth (e.g. 4 Hz) over the whole frequency range. Thus, for octave band centre frequency, f_c:

— lower limit $\approx 0.7 f_c$

— upper limit $\approx 1.4 f_c$.

For 1/3-octave band centre frequencies, f_c:

— lower limit $\approx 0.88 f_c$

— upper limit $\approx 1.12 f_c$.

Commonly used centre frequencies for octave band filters are 31.5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1 kHz, 2 kHz, 4 kHz and 8 kHz but lower and higher bands are also defined.

Narrow band analysis is most useful for complex tonal noises. It could be used, for example, to give a precise numerical value of a fan tone frequency, to determine the frequencies of vibration transmission from machinery or to detect system resonances.

Criteria for assessment of noise are based on dBA, octave bands or 1/3-octave band measurements. These measures clearly give increasingly detailed information about the noise.

References (Appendix A5)

A5.1 *Guidelines for community noise* (Copenhagen: World Health Organisation) (2000)

A5.2 *Reducing Noise at Work. Guidance on the Noise at Work Regulations 1989* HSE L108 (London: Health and Safety Executive) (1998)

Appendix A6: Vibration instrumentation

A6.1 Vibration quantities

Vibration may be measured as acceleration (a), velocity (v) or displacement (x).

Velocity is the rate of change of displacement, i.e:

$$v = \frac{\mathrm{d}\,x}{\mathrm{d}\,t} \tag{A6.1}$$

Acceleration is the rate of change of velocity, i.e:

$$a = \frac{\mathrm{d}\,v}{\mathrm{d}\,t} = \frac{\mathrm{d}^2 x}{\mathrm{d}\,t^2} \tag{A6.2}$$

For the special case of a sinusoidal vibration, where:

$$x(t) = X \sin \omega t$$

the relationship is:

$$v = \omega X \cos \omega t \tag{A6.3}$$

$$a = -\omega^2 X \sin \omega t \tag{A6.4}$$

where ωt is the angular frequency (i.e. $2\,\pi f$) ($\mathrm{rad{\cdot}s^{-1}}$) and X is the amplitude of the displacement

Acceleration is normally measured using a piezoelectric accelerometer and suitable preamplifier.

Acceleration levels in decibels are given by:

$$N = 20 \lg (a / a_0) \tag{A6.5}$$

where N is the acceleration level (dB) and a_0 is a reference acceleration of $10^{-6}\ \mathrm{m{\cdot}s^{-2}}$.

The reference levels for velocity and displacement are $10^{-9}\ \mathrm{m{\cdot}s^{-1}}$ and 10^{-12} m. The decibel levels for acceleration, velocity and displacement are numerically equal at $\omega = 1000\ \mathrm{rad{\cdot}s^{-1}}$ (i.e. $f = 159$ Hz).

Vibration measurements are also expressed in the measured physical quantities, $\mathrm{m{\cdot}s^{-2}}$, $\mathrm{m{\cdot}s^{-1}}$ and m, but sometimes millimetres are used as the unit of length, rather than metres. This depends on the magnitude of the vibration.

A6.2 Piezoelectric accelerometer

This is widely used, although other types of accelerometer are available. Operation is similar to a mass spring system, in which the piezoelectric element acts as the spring and carries a seismic mass. Vibration of the surface to which the accelerometer is attached then results in distortion of the piezoelectric element, leading to production of an electrical charge. Accelerometers have resonant frequencies, but because of the stiffness of the element, this is usually at a much higher frequency than the range of measurement. The charge sensitivity of an accelerometer is expressed in picocoulombs per unit of acceleration, or $\mathrm{pC/(m{\cdot}s^{-2})}$. The charge developed across a piezoelectric accelerometer can also be sensed as a voltage. The voltage sensitivity is expressed in $\mathrm{mV/(m{\cdot}s^{-2})}$.

A6.3 Amplifiers

The output of an accelerometer must first go to a preamplifier, which is designed to detect the output and convert the high electrical impedance of the accelerometer to a low output impedance, suitable for connection to a range of analysing instruments. The amplifier may also contain circuits to carry out the mathematical functions that convert acceleration into velocity and displacement by integration of acceleration. The final level indication is either the root mean square (rms) or peak value of the acceleration. Accelerometers may be connected directly into the microphone socket of some sound level meters, which then perform the measurement and analysis functions.

A6.4 Accelerometer fixings

Attachment of the accelerometer to a vibrating surface must be done with care. Magnetic fittings are a convenient method of fixing on flat magnetic surfaces. Alternatively, beeswax or similar temporary adhesive can be used. The accelerometer must be attached so that it is in intimate and secure contact with the surface, in order to ensure that the two vibrate together.

In general, vibration measurement is more specialised than sound measurement and fewer engineers are familiar with it, except in certain industries. Vibration measurement is one way of condition monitoring of equipment and it may be used for this purpose on rotating systems. Machinery vibration normally consists of a large number of discrete harmonic frequencies, which should be investigated with a narrow band analysing system.

Uses of vibration analysis in building services include:

- checking the efficiency of vibration isolators
- investigating pipe vibration
- investigating complaints of vibration of building surfaces
- looking for correlation between vibration and noise in a building.
- general diagnostic purposes.

These are successively more complex applications. The engineer should be aware of when to call on expert help.

Appendix A7: Direct and reverberant sound in a room

Considering a point source in a room, the simple approach is that, as one moves away from the source, the direct sound pressure level, due to energy which travels straight from the source to the ear, falls at 6 dB per doubling of distance (see Appendix A1). After a certain distance the summation of all the reflected sound in the room exceeds the direct level from the source. This is a region which is controlled by reverberant sound (see Figure 7.1). Thus, close to a source the total level is controlled by the source. Distant from a source the total level is controlled by the reverberant sound and, in this region, is constant over the room. This is expressed in equations A7.1 to A7.3:

$$L_p = L_W + 10\lg\left(\frac{4}{R_R} + \frac{Q}{4\pi r^2}\right) \tag{A7.1}$$

The reverberant sound is given by:

$$L_{pR} = L_W + 10\lg\left(\frac{4}{R_R}\right) \tag{A7.2}$$

The direct sound is given by:

$$L_{pD} = L_W + 10\lg\left(\frac{Q}{4\pi r^2}\right) \tag{A7.3}$$

where L_p is the total sound level at the receiver point (dB), L_W is the sound power level (dB), L_{pR} is the reverberant sound level at the receiver point (dB), L_{pD} is the direct sound level at the receiver point (dB), R_R is the room constant (m^2), Q is the directivity factor for the source and r is the distance from source to receiver (m).

The term $(4/R_R)$ relates to the reverberant field and $(Q/4\pi r^2)$ to the direct sound field.

The room constant is defined as:

$$R_R = \frac{S\bar{\alpha}}{1 - S\bar{\alpha}} \tag{A7.4}$$

and:

$$S = S_1 + S_2 + S_3 + \ldots\ldots + S_n \tag{A7.5}$$

where S is the total room surface area (m^2), S_1 etc. are the surface areas of the room surfaces (m^2) and $\bar{\alpha}$ is the average absorption coefficient of the room surfaces.

The average absorption coefficient, $\bar{\alpha}$, is given by:

$$\bar{\alpha} = \frac{S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \ldots .\ S_n\alpha_n}{S} \tag{A7.6}$$

The α terms are the random incidence absorption coefficients of the materials of the room surfaces. These coefficients vary with frequency and are usually smallest at low frequencies.

The room constant may also be obtained through measurement of the reverberation time, T, as follows:

$$T = \frac{0.16\,V}{S\bar{\alpha}} \tag{A7.7}$$

If the average absorption coefficient, $\bar{\alpha}$, is small, equation A7.4 reduces to $R_R \approx S\bar{\alpha}$.

Hence :

$$T \approx \frac{0.16\,V}{R_R} \tag{A7.8}$$

Referring to equation A7.1, the first term inside the brackets is controlled by the absorption in the room and represents the reverberant level (equation A7.2). The second term represents the direct sound from the source according to the inverse square law, as if the room were not present (equation A7.3). The directivity factor, Q, indicates how the noise is radiated preferentially in a direction of interest.

Equations A7.2 and A7.3 show that the direct and reverberant sound levels are equal when:

$$\frac{4}{R_R} = \frac{Q}{4\pi r_r^2} \tag{A7.9}$$

or:

$$r_r = (Q\,R_R / 16\,\pi)^{1/2} \tag{A7.10}$$

where r_r is the reverberation radius (m).

In reverberant rooms, the reverberation radius indicates distances at which either the direct or reverberant sound predominates. The direct sound is predominant at distances less than the reverberation radius, whilst the reverberant sound predominates at greater distances. The greater the value of room constant, R_R, the less reverberant the room and the greater the reverberation radius. However, most furnished offices do not fulfil the assumptions on which equation A7.1 to A7.3 are based.

This simple approach is reasonable for large rooms in which all three dimensions are of the same order, say 3:2:1, which are considered further in section 7. For many furnished rooms, and especially offices where the floor area is large compared to the height, the simple approach does not always hold. Furnished offices have reflections from furniture whilst the HVAC terminals are multiple sources. Although equation A7.1 is useful due to its simplicity, it may give misleading results in real rooms. Recognition of this led to research sponsored by ASHRAE, summarised in the section 7, in order to determine what happens in real rooms.

Equations A7.1 to A7.3 may be used in large reverberant spaces such as sports halls.

Appendix A8: Noise criteria

A8.1 Noise rating (NR)

Kosten and van Os[A8.1], developed octave band criteria for assessing the effects of external industrial and other noises on people in their homes. Whilst the resulting NR curves and criteria[A8.2] have become well known and widely used, it should be noted that the work was aimed at domestic premises, especially for night-time disturbance. Corrections were made for characteristics of both the noise and the residential district, although these were not included in the subsequent use for HVAC noise. NR is a tangent assessment, where the rating is given by the highest NR curve which is tangential to an octave band analysis of the noise. (A tangent method should also state the octave band that determines the highest criterion level, but this is not normally given.) NR criteria were included in a draft ISO Standard on environmental noise[A8.3] and are sometimes referred to as the 'ISO criteria', but they were removed from the final version of the standard and have no status within International Standards.

An examination of the evaluation spectra used by Kosten and van Os in their validation of the NR curves, shows that, for 'acceptable spectra' related to NR25–30, the low frequency levels of the test spectra are, on average, 10–15 dB below the 63 Hz criterion level. 31.5 Hz was not tested, as the initial definition of the NR curves did not go below 63 Hz. Thus, Kosten and van Os's data did not test the NR curves at the lower frequencies for which they are used. As a consequence, circumstances arise in HVAC applications in which an objectionable low frequency noise satisfies the NR design criterion.

Whilst it would be possible to modify NR in order to increase its stringency at low frequencies, there are other criteria, described below, which already do this and which should be used as an alternative to NR.

A8.2 Noise criterion (NC) curves

Beranek developed criteria to ensure good speech intelligibility. Much of the original work was carried out in engineering offices where speech communication between colleagues was required, but where the background noise sometimes interfered. Beranek determined the highest levels, with respect to the mid frequency speech interference bands, which could occur at both higher and lower frequencies and still give 'acceptable' conditions. This work led successively to the NC[A8.4], the PNC[A8.5], and the NCB[A8.6,A8.7] criteria, as problems with the each version became apparent.

Beranek[A8.4] describes the development of NC curves, the forerunner of both PNC and NCB. The work was aimed at determining the maximum noise level under which office workers maintain efficiency.

A relationship for acceptability was obtained, as follows:

$$\mathrm{LL} - \mathrm{SIL} \leq 22 \text{ units}$$

where LL is the Stevens loudness level[A8.8] and SIL is the three-band speech interference level, i.e. the arithmetic average of levels at 500 Hz, 1000 Hz and 2000 Hz. The result is based on the responses of 300 office workers in executive offices/small conference rooms and stenographic/engineering drafting rooms, during actual and contrived noise exposures.

The NC curves are a representation of the SIL/LL criteria on an octave band basis, i.e. the number attributed to the curve is equal to its speech interference level, whilst the loudness level (phons) for a spectrum following the curve is 22 units greater than the SIL, in accord with the relationship above. This permits maximum low frequency levels, whilst still satisfying the criterion adopted. Beranek was also aware of the complaint potential of beats and fluctuations at low frequencies, but did not include these. Bradley[A8.9] has studied fluctuations in HVAC noise.

NC is described in section 1 of CIBSE Guide A[A8.2]. It has been widely used in the assessment of noise in buildings. NC is a 'tangent' approach similar to NR. It was found that a noise spectrum closely following an NC curve, and therefore satisfying the basis for the curves, does not itself give a pleasant sound, but rather has both throb (rumble) and hiss. In a similar manner, an environmental noise, which falls off rapidly with frequency, can be annoying even though it does not exhibit tonal characteristics. NC should not be used for assessment in spaces that are sensitive to noise, but could have an application in non-sensitive areas, such as busy lobbies, where there is considerable activity noise.

A8.3 Room criterion (RC) Mark I and II

Blazier[A8.10] developed the RC Mark I curves on the spectrum of noise from 68 offices, which were known to have good acoustics, and later developed a method for determining the 'noise quality' of a spectrum[A8.11].

It was found that a spectrum falling at 5 dB/octave represented the noise in the offices. The levels in the lowest octave bands (16 Hz and 31.5 Hz) indicate the possibility of noise-induced vibration of lightweight building components. The RC Mark I curves are described in section 1 of CIBSE Guide A[A8.2]. Recent work, leading to RC Mark II, has modified the 16 Hz band to lower it to the same level as the 31.5 Hz band[A8.12]. Large, low frequency fluctuations, which can occur from poor running of fans, are controlled more effectively by this lower 16 Hz limit.

The use of RC Mk II is to derive a noise quality assessment for the HVAC noise. This is the method currently recommended by ASHRAE[A8.12].

A8.4 Balanced noise criterion (NCB)

Beranek[A8.6,A8.7] further modified the NC curves to produce the NCB, which extends down to the 16 Hz octave band. Beranek again starts with the proposition that the most important acoustical requirement of working spaces is satisfactory speech communication. Thus each curve is based around, and has the rating number of, its (four-band) speech interference level (SIL). The four band SIL is the average of the 500 Hz, 1000 Hz, 2000 Hz and 4000 Hz bands.

The second most important requirement is spectrum balance, which is a determinant of the 'quality' of the noise. Balance is obtained by equalising the octave band loudness levels of each band. This assumes that 'balance' is a function of octave band loudness.

The NCB curves are intended for both occupied and unoccupied space, including activity noise, which may be 10–15 dB above air handling noise at mid-frequencies and above, but similar to air handling noise at the lowest frequencies. The procedure for using NCB curves is described by Beranek(A8.6,A8.7), although neither the PNC or NCB criteria have gained widespread acceptance.

A8.5 Comparison of criteria

Rating curves at the levels of NC35, PNC35, RC35 Mark I and II, NCB35, NR35 are compared in Table A8.1.

All criterion curves are very similar from 125 to 1000 Hz and fairly similar at higher frequencies, but diverge at lower frequencies. The divergence is such that the difference between NR35 and RC35 is 19 dB at 31.5 Hz. This means that a noise, which just met the NR35 criterion at 31.5 Hz, would exceed RC35 by 19 dB. The difference between the criteria is of no consequence if there is no low frequency noise present, but the difference does mean that a low frequency HVAC rumble, which would exceed the RC criterion, could still pass the NR criterion whilst being subjectively objectionable.

A8.6 The dBA

The dBA is a single number measure of a noise (see Appendix A5). Its effectiveness in preventing disturbance depends, however, on the spectrum of the HVAC noise, as a wide range of spectra will give the same dBA reading. Reduction of dBA will not necessarily lead to a more acceptable noise. For example, a reduction of, say, 5 dBA for a noise with an audible tone will be most effective if the tone is controlling the overall value of the dBA. 125 Hz is attenuated by 16 dB in the dBA weighting, so that a tone at this frequency will have to be more than 16 dB above the levels of the mid frequencies for it to influence the dBA strongly. If an attempt at reducing the dBA did this by reducing higher frequencies, a lower frequency tone would become more prominent and disturbing.

Table A8.1 Comparison of noise criteria

Octave band/ Hz	Sound pressure level obtained for stated criterion / dB					
	NC35	PNC35	RC35 Mk I	RC35 Mk II	NCB35	NR35
16	—	—	65	60	84	—
31.5	—	62	60	60	71	79
63	60	55	55	55	58	63
125	52	50	50	50	50	52
250	45	45	45	45	44	45
500	40	40	40	40	40	39
1000	36	35	35	35	37	35
2000	34	30	30	30	32	32
4000	33	28	25	25	30	30
8000	32	28	20	20	27	28

A8.7 Assessment of criteria

The two decades from 1980 to 2000 have seen considerable developments in North America, which have made new criteria available for high quality acoustic design of ventilated buildings. The newer criteria, in particular RC, are not yet well known in the UK, but should be given serious consideration for use(A8.2). Compliance with them ensures a subjectively neutral and acceptable sound in the room.

The NR criterion (noise rating) is not safe to use under all circumstances, since it permits unacceptable levels of low frequency noise. It is recommended that use of the NR be phased out.

The NC criterion is an improvement on the NR, although it is no longer recommended by ASHRAE except for non-sensitive locations. Developments from the NC, that is, PNC and NCB have not been widely accepted. RC and NCB are both given in ANSI Standard S12.2: *Criteria for evaluating room noise*(A8.13) as alternative methods for assessing room noise.

RC Mark II(A8.11) is a recent criterion, which has been developed specifically for assessment of occupant satisfaction in the presence of HVAC noise. It is used to give an indication of the quality of a noise, based on relative levels at low, mid and high frequencies, so that any correction required is focussed on the appropriate frequency range.

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