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Acoustics and Ventilation

Matthew K. Ling

AIVC

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Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty four IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems (ECBCS)

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but also identifies new areas where collaborative effort may be beneficial. To date, the following projects have been initiated by the Executive Committee (completed projects are identified by *):

1. Load Energy Determination of Buildings *
2. Ekistics and Advanced Community Energy Systems *
3. Energy Conservation in Residential Buildings *
4. Glasgow Commercial Building Monitoring *
5. Air Infiltration and Ventilation Centre
6. Energy Systems and Design of Communities *
7. Local Government Energy Planning *
8. Inhabitant Behaviour with Regard to Ventilation *
9. Minimum Ventilation Rates *
10. Building HVAC Systems Simulation *
11. Energy Auditing *
12. Windows and Fenestration *
13. Energy Management in Hospitals *
14. Condensation *
15. Energy Efficiency in Schools *
16. BEMS - 1: Energy Management Procedures
17. BEMS - 2: Evaluation and Emulation Techniques *
18. Demand Controlled Ventilating Systems *
19. Low Slope Roof Systems *
20. Air Flow Patterns within Buildings *

21. Calculation of Energy and Environmental Performance of Buildings*
22. Energy Efficient Communities *
23. Multizone Air Flow Modelling (COMIS) *
24. Heat Air and Moisture Transfer in Envelopes *
25. Real Time HEVAC Simulation *
26. Energy Efficient Ventilation of Large Enclosures *
27. Evaluation and Demonstration of Domestic Ventilation Systems
28. Low Energy Cooling Systems*
29. Daylight in Buildings*
30. Bringing Simulation to Application*
31. Energy Related Environmental Impact of Buildings
32. Integral Building Envelope Performance Assessment*
33. Advanced Local Energy Planning
34. Computer-aided Evaluation of HVAC System Performance
35. Design of Energy Efficient Hybrid Ventilation (HYBVENT)
36. Retrofitting in Educational Buildings - Energy Concept Adviser for Technical Retrofit Measures
37. Low Energy Systems for Heating and Cooling of Buildings
38. Solar Sustainable Housing.
39. High Performance Thermal Insulation (HiPTI)
40. Commissioning of Building HVAC Systems for Improved Energy Performance

"Ordinarily there is not a close connection between the flow of air in a room and its acoustical properties, although it has been frequently suggested that thus the sound may be carried effectively to different parts. On the other hand, while the motion of air is of minor importance, it is on reliable record that serious acoustical difficulty has arisen from abrupt differences of temperature in an auditorium. Finally, transmission of disturbing noises through the ventilation ducts, perhaps theoretically a side issue, is practically a legitimate and necessary part of the subject."

W. C. Sabine
Collected Papers on Acoustics
Dover, 1964

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Contents

1 The Basics of Acoustics

This Chapter aims to provide the reader with an overview of the basics of acoustics, which are required as part of the ventilation systems design process. With this knowledge the designer can, with the remaining chapters, apply these principles to providing quiet and effective ventilation in buildings.

1.1 Sound and Noise

Noise can be described as **unwanted sound**. In the context of the built environment, noise is normally sound which causes annoyance or disturbs activities. We can generalise by saying that sound becomes unwanted when it:

- hinders speech communication;
- impedes thinking processes;
- interferes with concentration;
- obstructs activities (work or leisure) ;
- presents a health risk due to hearing damage.

Obviously, not all of these factors occur together, and within living or working spaces levels that would cause hearing damage are rare. However, even at this stage we must be careful not to make the assumption that the presence of noise is necessarily a bad thing. We will see later (Chapter 4) that background noise can improve the acoustic environment of an office space.

1.2 How are Sounds Described?

Sound is a form of energy that is transmitted through the air. In transmitting sound, the air particles vibrate and cause rapid cyclic pressure changes that are sensed by the ear. Sounds are characterised by their *frequency* and *amplitude (level).*

The most common terms that are used to describe a wave are defined in Table 1.1.

1.3 Frequency

Frequency is the rate at which the air particles vibrate. The more rapid the vibrations, the higher the frequency and the higher the musical pitch. 'Hum', 'drone' and 'throb' are words applied to sounds that contain mainly low frequencies. 'Whistle', 'squeal' and 'hiss' describe sounds containing mainly high frequencies.

Frequency is measured in Hertz (Hz). Older books use the units 'cycles per second' (cps). The two are equivalent.

Table 1.1 Wave Terminology

Term	Definition	Symbol	Unit
Amplitude	The maximum displacement of a layer from its rest position. Amplitude is related to the loudness of a sound.	Various	Metres, m
Wavelength	The distance between two successive layers which have the same displacement and are moving in the same direction	λ	Metres, m
Frequency	The number of complete cycles in one second. Frequency is related to the pitch of a sound.	f	Hertz, Hz, kHz, cycles per second
Velocity	The speed of a wave in a specified direction.	c or v	Metres per second, ms^{-1}, m/s

The human ear can detect sounds in the range 20 Hz to 20 000 Hz (20 kHz) approximately. The scale is not linear because the ear responds to proportional changes. For example, a doubling of frequency represents the same musical interval wherever it occurs on the scale. Thus, the 'equal' steps of a piano keyboard do not correspond to equal increments on the frequency scale.

Each frequency value has an associated wavelength. The higher the frequency, the shorter the wavelength.

1.4 Relationship Between Velocity, Frequency and Wavelength

The three wave properties velocity, frequency and wavelength are related by the expression:

$$v = f\lambda$$

where v is the velocity of sound (m/s)
f is the frequency of the sound (Hz)
λ is the wavelength of the sound (m)

This expression shows that for a constant wave velocity;

- a *high frequency* wave has a *short wavelength*
- a *low frequency* wave has a *long wavelength*.

1.5 Level

In transmitting sound, the air particles vibrate back and forth about a mean position. The further they move, the greater the energy of the sound and the higher the level. It is difficult to measure energy directly. It is more convenient to measure the magnitude of the fluctuating air pressure caused by the sound wave.

1.6 Decibels and Sound Levels

Acoustical quantities are normally measured in decibels - a logarithmic unit. Decibels are used for a number of reasons. Firstly, because sound pressures vary from very small values to very large, and secondly because the ear’s response to sound levels is non-linear. If decibels were not used a scale would be needed consisting of 10^{13} divisions to cover the range of minimum detectable sound level (20 µPa) to pain level (60 Pa). Table 1.2 shows the sound pressure levels typical of commonly encountered noise sources.

The **sound pressure level (SPL)** is defined as;

$$L_p = 10\lg\frac{p^2}{p_o^2}dB$$

$$or$$

$$L_p = 20\lg\frac{p}{p_0}dB$$

where the reference level p_0 is 2×10^{-5} Pa
and p is the measured r.m.s. sound pressure in Pascals

Table 1.2 Typical SPLs for Common Sources

Sound Source	**Sound Pressure Level (dB)**
threshold of hearing	≥0
rustle of leaves	10
soft whisper	30
mosquito buzzing	40
average townhouse	50
ordinary conversation	60
busy street	70
power mower	100
threshold of pain	120
rock concert	130
jet engine at 30m	150
rocket engine at 30m	180

Whilst this gives rise to a convenient scale for measurement purposes, confusion can exist when interpreting dB levels and level changes into subjective response. The human ear responds to continuous sound sources in broadly the ways identified in the table below. We will return to other measures of subjective response to sound later in this chapter.

Apart from SPL there are other measures of noise that are commonly used, depending on the application and measurement techniques being used.

Sound intensity level is defined as:

$$L_I = 10\lg\frac{I}{I_o}dB$$

where I_0 is the reference sound intensity level = 1×10^{-12} Wm^{-2}
I is the sound intensity measured in Wm^{-2}

Sound power level (PWL) is defined as:

$$L_w = 10\lg\frac{W}{W_0}dB$$

where W_0 is the reference power level = 1×10^{-12} W
W is the measured sound power level in W

Table 1.3 Subjective Effect of Noise Levels

Change in Level (dB)	Subjective Effect
1	smallest audible change in level. It would be noticed only if the two sounds were presented in quick succession.
3	just perceptible – the smallest audible change which can be detected over a period of time
5	easily perceptible
10	twice as loud

1.7 Sources of Noise and Impedance

Any vibrating system in a medium (fluid or solid) will generate noise. The amplitude of vibration, and the degree of coupling with the medium determines the level of noise radiated. The ability of sound to travel from one medium to another is characterised by its impedance, and the ability of a surface to emanate sound into a medium is often described by the radiation efficiency, σ. For the discussion in this Note we do not need to have an in-depth understanding of these properties.

When we are considering ventilation systems the majority of the noise sources are either mechanical (vibrational) or aerodynamical in origin. Mechanical plant noise might be generated by pumps, motors, air handling units, compressors, condensers etc. Aerodynamic noise is produced by airflow through fans, ducts, bends, diffusers and grilles.
In addition to these sources, ventilation systems can also contribute to the internal noise levels by providing a flanking path through which noise can travel. For example windows might allow traffic noise to enter a room, or a ventilation stack allow aircraft noise to enter.

1.8 Simple Decibel Arithmetic

It was mentioned previously that the dB scale is logarithmic which means combining sound pressure levels is not achieved by simply adding together the decibel values. A conversational voice at 1m distance results in a sound-pressure level of about 65 dB. Two voices together would not cause a level of 130 dB; that would be louder than a pneumatic drill. To calculate the effect of two sources acting together actual energy values have to be added which means converting back to pressure, intensity or power.

A simple method for combining levels, accurate to ±1 dB can be used with the following table.

Difference between levels to be combined (dB)	Add to higher level (dB)
0 or 1	3
2 or 3	2
4 to 9	1
10 or more	0

A nomogram is given below in Figure 1.1 to simplify this procedure.

Example

If one source alone gives 65 dB, two identical sources together give:

65 and 65 (difference 0, add 3) = **68** dB

Adding a third source gives:

68 and 65 (difference 3, add 2) = **70** dB

Adding a fourth source gives:

70 and 65 (difference 5 add 1) = **71** dB
or
68 and 68 (difference 0, add 3) = **71** dB

To achieve the most accurate result when using this approximate method of combining more than two levels, start with the lowest number and work upwards.

If we are summing the levels by converting the sound pressure levels back into energy to add together, we get the equation:

$$L = 10\lg \sum_{i=1}^{n} 10^{\frac{Li}{10}}$$

Note that this expression assumes that there is no relationship between the sources i.e. they are not related harmonically.

Example

Two fan units separately have noise levels (SPL) of **82** dB and **88** dB, measured at 5m. What is their combined noise level?

L= 10 lg (antilog 82/10 + antilog 88/10)

L = **89** dB

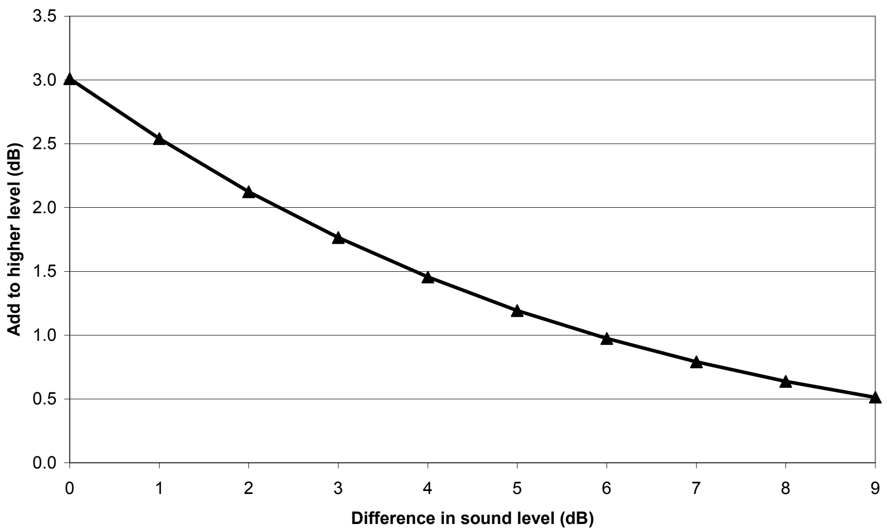


Figure 1.1 Addition of decibels

1.9 Frequency Analysis – Octaves, One-Third Octaves and Narrow Bands

For most building purposes, a restricted frequency range is used rather than the complete spectrum. The range is divided into octave bands and one-third octave bands. The term 'octave' comes from musical notation. It is the interval between the first and the eighth note in a scale and represents a doubling in frequency. Table 1.4 highlights the range of octave and one-third octave bands normally used.

A sound can be described by reporting its sound pressure level in each of a number of frequency bands. Information of this kind is normally obtained by measurement using a sound-level meter fitted with an octave or a one-third octave band filter set.

Often it is important to analyse a sound source and obtain a measure of the relative contributions of the different frequencies. The information about frequencies and intensities provide a way of characterising the noise and inform decisions made about relevant noise control measures. Frequency analysis is normally done either for discrete frequencies using a mathematical algorithm called a fast Fourier transform (FFT) or in bands. These bands are normally either the octave or third octave bands. The centre frequencies of these bands are given in Table 1.4.

Table 1.4 Octave and One Third Octave Band Centre Frequencies

Octave (Hz)	1/3 Octave (Hz)
31.5	25 **31.5** 40
63	50 **63** 80
125	100 **125** 160
250	200 **250** 315
500	400 **500** 630
1000	800 **1000** 1250
2000	1600 **2000** 2500
4000	3150 **4000** 5000
8000	6300 **8000** 10000

1.10 Response of the Ear to Sound

The operation of the ear is not completely understood. To overcome this lack of knowledge many psycho-acoustic experiments have been carried out on the response of the human ear to sound. The following important facts have been discovered:

- The ear does not perceive sounds of equal intensities, but different frequencies, as being of equal loudness;
- The loudness of a sound depends upon the intensity;
- Short sounds (impulses) appear quieter than continuous levels of the same sound intensity.

The experiments have enabled curves of equal loudness to be constructed. These are reproduced below. If you were listening to a pure tone, which had its level and frequency changed according to one of the curves, it would be perceived as having the same loudness throughout.

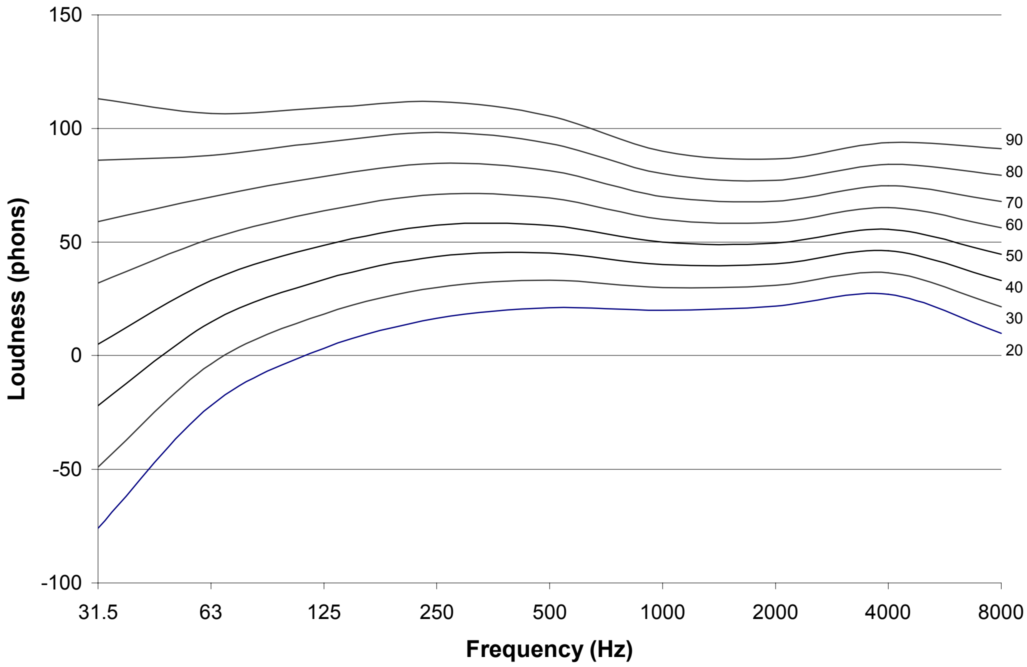


Figure 1.2 Equal Loudness Curves (from ISO R 226-1961)

1.11 A-weighted Levels, dB(A)

The curves of equal loudness above show that the ear notices high frequencies more easily than low frequencies i.e. it is more sensitive to high frequencies than low frequencies, due to the morphology and physiology of the ear. This characteristic has been incorporated into sound-level meters using the A-weighting network. Figure 1.3 is a graph showing the adjustments made at each frequency. This shape correlates well with human response to many types of noise.

A-weighted sound-pressure levels can be measured directly on a sound-level meter or obtained from octave-band sound levels by applying the weightings at each frequency and then combining the resulting levels using the rules for addition of decibels.

The 'A' values in the (3rd column in the right hand table) are obtained by adding the correction from the graph to the source level

Table 1.5 Frequency and Sound Pressure Level

Frequency (Hz)	**Sound Pressure Level (dB)**		
		A	**Combine**
125	79	63	63+64=67
250	73	64	66+67=70
500	72	69	69+70=73
1000	72	72	71+72=75
2000	70	71	73+75=77
4000	65	66	Answer=**77 dB(A)**

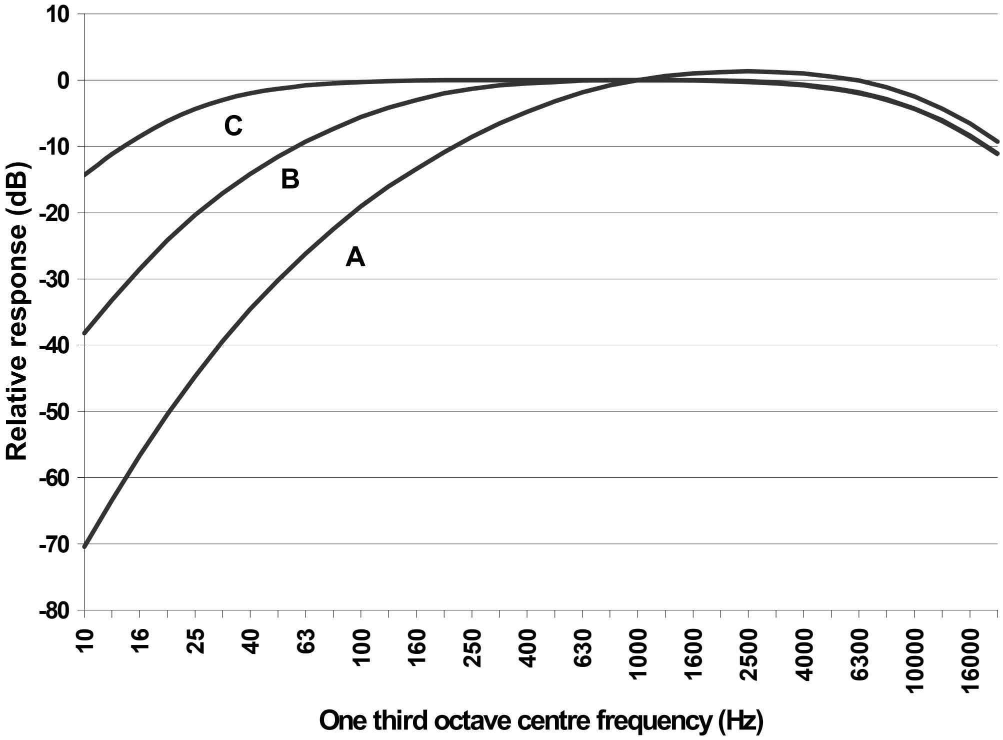


Figure 1.3 Weighting Curves

Table 1.6 Weighting Values for A, B and C

One third octave centre frequency (Hz)	A	B	C
10	-70.4	-38.2	-14.3
12.5	-63.4	-33.2	-11.2
16	-56.7	-28.5	-8.5
20	-50.5	-24.2	-6.2
25	-44.7	-20.4	-4.4
31.5	-39.4	-17.1	-3
40	-34.6	-14.2	-2
50	-30.2	-11.6	-1.3
63	-26.2	-9.3	-0.8
80	-22.5	-7.4	-0.5
100	-19.1	-5.6	-0.3
125	-16.1	-4.2	-0.2
160	-13.4	-3	-0.1
200	-10.9	-2	0
250	-8.6	-1.3	0
315	-6.6	-0.8	0
400	-4.8	-0.5	0
500	-3.2	-0.3	0
630	-1.9	-0.1	0
800	-0.8	0	0
1000	0	0	0
1250	0.6	0	0
1600	1	0	-0.1
2000	1.2	-0.1	-0.2
2500	1.3	-0.2	-0.3
3150	1.2	-0.4	-0.5
4000	1	-0.7	-0.8
5000	0.5	-1.2	-1.3
6300	-0.1	-1.9	-2
8000	-1.1	-2.9	-3
10000	-2.5	-4.3	-4.4
12500	-4.3	-6.1	-6.2
16000	-6.6	-8.4	-8.5
20000	-9.3	-11.1	-11.2

1.12 Sound Levels that Vary with Time

One further factor must be considered in describing sounds; the way in which they vary with time.

For example, free flowing traffic on a major road may be continuously audible, with fluctuations in level and frequency content related to individual passing vehicles. In contrast, aircraft may be inaudible for long periods between loud events. Research into the way humans are affected by noise has led to a variety of different measurement units for different noise sources. Two widely used noise descriptors are defined here.

1.13 Equivalent Continuous A-weighted Sound Pressure Level - $L_{Aeq,T}$

This is also called the *time interval average sound level*. It is the A-weighted energy mean of a noise, averaged over a time period T. By considering the energy content of incident sound it converts a fluctuating sound into an equivalent continuous sound level for the same period of time. It is one of the most common indices that is encountered. Mathematically it is defined as:

$$L_{Aeq,T} = 10\lg\frac{1}{T}\int_0^T\left(\frac{p_A(t)}{p_0}\right)^2 dt \quad \text{dB(A)}$$

where $p_A(t)$ is the A-weighted instantaneous acoustic pressure
p_0 is the reference pressure = 2 x 10^{-5} Pa
T is the total measurement time.

This equation is sometimes expressed as:

$$L_{Aeq,T} = 10\lg\frac{1}{t_2 - t_1}\int_{t_1}^{t_2}\left(\frac{p_A(t)}{p_0}\right)^2 dt \quad \text{dB(A)}.$$

1.14 Percentile Levels, L_N

Noise is often described statistically. This leads to noise indices of L_N where N indicates the % of time where the value L_N is exceeded. For example an $L_{A,10,T}$ of 65 dB indicates that during the measurement period the noise levels exceeded 65 dB for 10% of the time. Common values of N are 1, 10, 50, 90, and 95, and are usually A weighted. The $L_{A,90,T}$ and $L_{A,95,T}$ are often used as measures of background noise level. L_{A10} is often used when measuring traffic noise, (e.g. in the UK and Australasia).

1.15 Attenuation with Distance

For a point noise source radiating equally in all directions, i.e. spherically, the SPL reduces by 6dB for each doubling of distance from the source in a free field condition (i.e. not a diffuse field). This can be calculated by the expression:

$$\begin{aligned} L_p &= L_W + 10\lg\left(\frac{1}{4\pi r^2}\right) \\ &= L_W - 20\lg r - 11 \qquad \text{dB} \end{aligned}$$

where L_p is the sound pressure level (dB)
L_w is the source sound power level (dB)
r is the distance from the source (m).

For a point source on a flat reflecting plane the SPL is increased by 3dB i.e.:

$$L_p = L_W - 20\lg r - 8 \qquad \text{dB.}$$

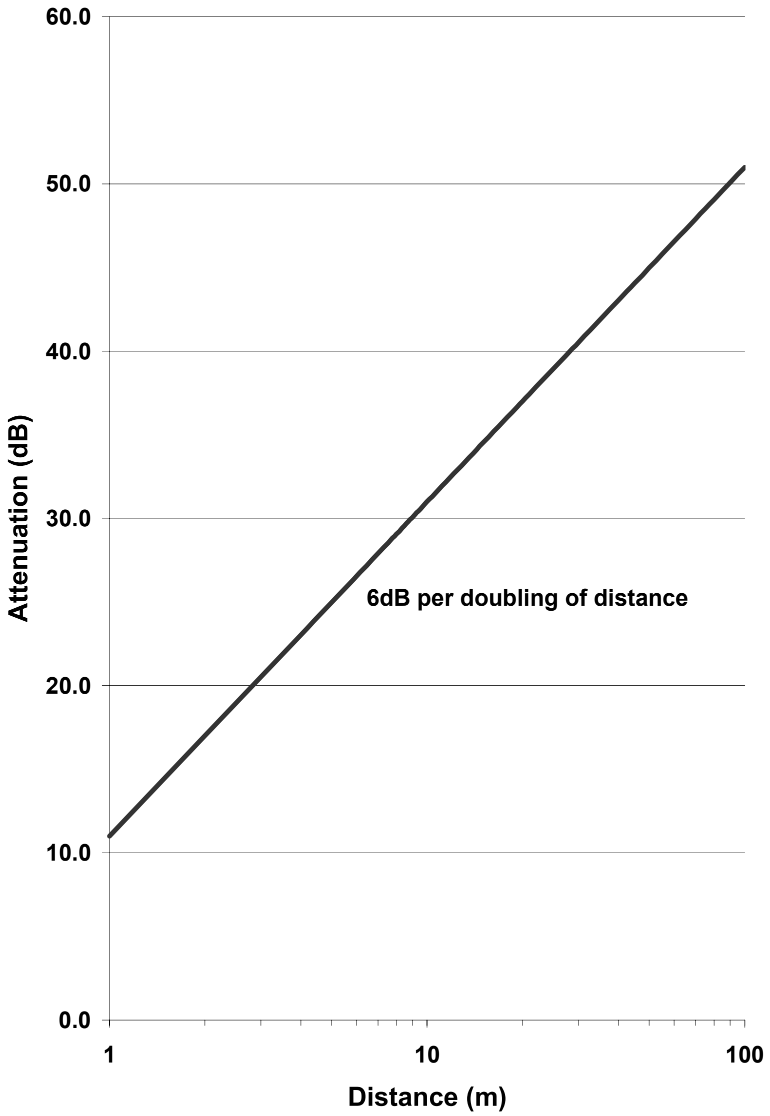


Figure 1.4 Attenuation with Distance for a Point Source in Free Space

For a line source the attenuation with distance is 3dB per doubling of distance in a free field condition. Note that close to sources, in the near field, the sound field does not behave as described by the above equations. When estimating sound pressure levels in the near field, measurements should be used to verify any predictions.

1.16 Directivity

Many sources such as louvres, diffusers and grilles exhibit directional characteristics and do not radiate sound equally in all directions. Additionally, the positioning of a ventilation element and its proximity to reflecting surfaces may influence its radiation pattern into the environment. The SPL of the source is given by:

$$L_p = L_W + 10\lg\left(\frac{Q_\theta}{4\pi r^2}\right) \qquad \text{dB}$$

where Q_θ is the directivity factor at the angle θ,
L_p is the sound pressure level (dB),
L_w is the source sound power level (dB),
r is the distance from the source (m).

The directivity of a source is sometimes described by its directivity index (DI) which is defined as:

$$DI_\vartheta = 10\lg Q_\vartheta \quad \text{dB.}$$

A source which is non-directional has a $DI = 0$ (ie. $Q_\theta = 1$).

1.17 Noise in Rooms

The sound field at any point in a room is the result of the direct sound from a source, and the reverberant sound from the source, combined with the influence of the room. The direct sound level is dependent upon:

- source (fans, mounting, ductwork, control valves, grilles etc.);
- distance from the source.

The reverberant sound level is largely independent of position within a room. This is the case where the sound field is said to be *diffuse*, such as in a reverberation chamber or transmission loss suite.

The total sound pressure at any point is the combined effect of the direct and reverberant sound field and is given by:

$$L_p = L_W + 10\lg\left(\underbrace{\frac{Q_\theta}{4\pi r^2}}_{direct} + \underbrace{\frac{4}{A}}_{reverberant} \right) \quad \text{dB}$$

where A is the equivalent sound absorption area (m^2), also known as the room constant R where $A = S\alpha$
α is average absorption coefficient for the room,
S is room surface area (m^2),
Q_θ is directivity factor of source in direction θ,
r is distance from source (m),
L_W is the sound power (dB).

The reverberant sound also has components added from other sound sources such as domestic appliances, transportation noise etc. The best way of considering these is to separate their contribution from that of the ventilation system, and then sum the levels at the last stage of a calculation procedure.

The directivity factor Q_θ is a function of the position of the source in a room and can also be influenced by the geometry of the source. These factors are given in Table 1.6.

Table 1.7 Directivity Factors

Position in room	Q_θ
In or near room centre	1
Centre of wall/floor/ceiling	2
Junction of 2 adjacent surfaces	4
Corner (i.e. 3 adjacent surfaces)	8

1.17.1 Reverberation Time, RT

The reverberation time of a room is a measure of the quantity of sound absorption that is present. Thus a room with smooth, hard, surfaces will have a higher RT than a room with soft furnishings. A typical furnished room will have an RT of between 0.3 and 0.5 seconds. RTs are measured in each frequency band (one octave or one-third octave) and are used when level differences from sound transmission measurements need to be standardised, or corrections to sound pressure predictions in rooms need to be made. A room with too high an RT will cause speech communication difficulties, due to all the additional echoes and reflections.

1.18 Sound Absorption

Sound absorption is a measure of how much an incident wave is dissipated when it hits a surface and is converted into other forms of energy (normally heat). The sound absorption in a room controls the reverberation time. The more absorption there is in a room the lower the RT.

1.19 Room Constant, Absorption and RT

The room constant, R, is often encountered in the literature as a variable that characterises the room absorption. As mentioned previously it is same as the equivalent sound absorption (A). It is defined as:

$$R = A = \frac{S\overline{\alpha}}{1-\overline{\alpha}} \approx S\overline{\alpha} \quad where\, \overline{\alpha} \le 0.15$$

where S is the total surface area of the room (m^2),
α is average statistical sound absorption coefficient,
R is measured in Sabines (m^2).

The relationship between the room constant (R or A) and reverberation time is given by:

$$A = R = \frac{0.161V}{T}$$

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

1.20 Transmission Loss

Transmission loss (TL) is the term used to describe the quantity of sound energy that is reduced by a panel or partition. It is essential to consider the implications of the transmission loss of building elements, whether plant room walls or windows, as they provide routes noise transfer from one part of a building to another. The TL is given by the expression:

$$TL = 10\lg(\frac{1}{\tau})\ \text{dB}$$

where τ is the total transmissibility of the wall and is defined as the ratio of transmitted acoustic intensity to the incident intensity on the panel. It is given by the expression $\tau A = \sum_{i=1}^{n} \tau_i A_i$ where τ_i and A_i are the transmission coefficient and area of element i.

The simplest calculation method of transmission loss assumes that the panel or wall is limp i.e. that it has no stiffness, and the sound transmission is only controlled by its mass. For normally incident[1] sound the 'mass law' transmission loss can be calculated from:

$$TL = 20\lg[mf] - 42 \quad \text{dB}.$$

This is illustrated in Figure 1.5. It can be seen that doubling the surface mass (kgm^{-2}) or the frequency increases the TL by 6 dB.

In practice walls and panels rarely behave according to the mass law. Sound is incident at a variety of angles, and a phenomenon called coincidence occurs. This is where the incident wave number in the air is equal to the bending wave number in the wall. It occurs for one 'critical' frequency for every angle of incidence, and causes a dip to occur in the TL curve. At the critical frequency damping also becomes important.

Above the critical frequency the TL is no longer solely controlled by the mass of the panel, but rather by its stiffness. If the frequency is doubled, above the critical frequency, the TL is increased by 9dB for single panels or walls.

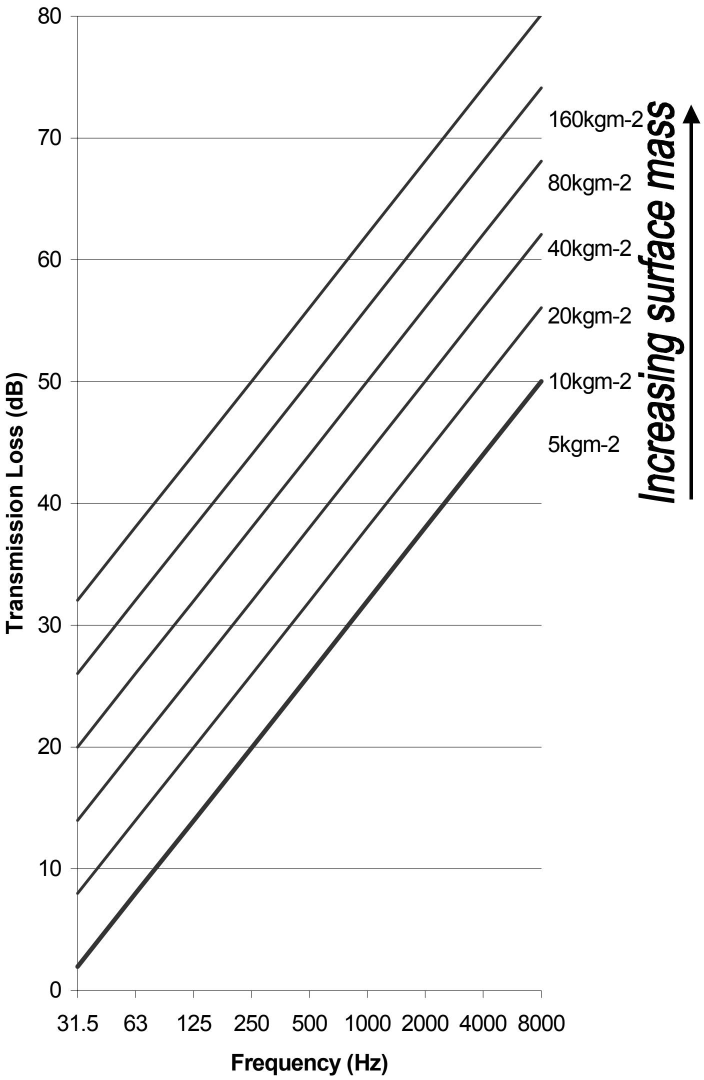


Figure 1.5 Generalised transmission loss curve

[1] This equation assumes that the sound is normally incident to the wall. The 'field' incidence TL is the -5 dB below the normal incidence value.

To attain good transmission loss the following factors should be considered:

- adequate mass;
- low stiffness, where low frequency performance is important;
- internal damping;
- uniform surface mass;
- no air leakages.

1.20.1 Measurement of TL

Procedures to measure the transmission loss (or sound reduction index, SRI) are well documented and should be carried out to the relevant national and international standards (see EN ISO 140 in Chapter 4 for further details). Only the general principles are described here. The standards set down separate conditions for measurements in the field and laboratory. The most common type of measurement required is probably that of two rooms separated by a partition as illustrated in Figure 1.6.

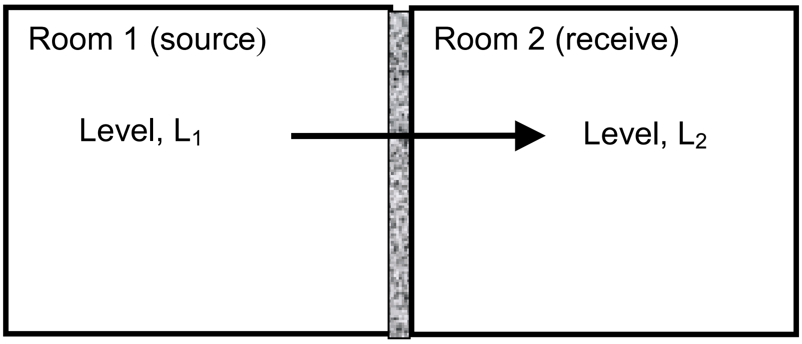


Figure 1.6 Schematic of TL measurement

A noise source is placed in room 1. The noise should be steady with a continuous spectra in the frequency range of interest. (Note that both pink or white noise spectra can be used.) Space averaged sound pressure levels are measured in both source and receive rooms. Normally a number of source and receive positions are measured, to ensure that the sound fields are amply sampled. Multiple static microphone positions or a rotating boom and microphone can be used.

The level difference, D is given by:

$$D = L_1 - L_2 \text{ dB.}$$

The level difference can also be standardised using the reverberation time in the receiving room to give:

$$D_{nT} = L_1 - L_2 + 10\lg(\frac{T}{T_0}) \text{ dB.}$$

As with the sound levels the RT is normally measured for a number of source and receive positions.

The transmission loss (TL) can also be calculated from:

$$R = L_1 - L_2 + 10\lg(\frac{S}{A}) \text{ dB},$$

where S = area of partition (m^2),

A is the equivalent absorption area ($\frac{0.16V}{T}$),

and V is the volume (m^3) of the receive room,

T is the reverberation time of the receive room (s).

Once values have been calculated for each third octave of interest, single figure rating methods (see Chapter 4) can be used to calculate the Sound Transmission Class (STC) or $R_{w.}$ These can be useful when specifying the performance of a partition or sound insulation element. Typical values of TL are given in Table 1.7.

Table 1.8 Typical Transmission Loss Values

	Transmission Loss (dB)							
	Octave Band Centre Frequency (Hz)							
Partition	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**STC**
Gypsum board 12.5 mm		18	22	26	29	27	26	
Gypsum board 15mm		16	22	28	32	29	31	30
6mm glass	17	11	24	28	32	27	35	39
9mm glass	18	22	26	31	30	32	39	43
6mm glass, 100mm cavity, 6mm glass	20	28	30	38	45	45	53	50
Hollow core door, no seals	13	12	11	14	17	16	16	15
190mm concrete block (plastered)		45	46	50	54	61	62	54

1.21 Measurement of Building Elements Performance

Acoustic performance of small building elements can also be measured using EN ISO 140-10: 1991 in the laboratory. The standard applies to the following ventilation elements:

- transfer air devices;
- 'airing panels' (ventilators);
- outdoor air intakes.

The element under test is mounted in a partition of sufficiently high sound insulation (at least 10dB higher than element under test), and built into the test opening between two reverberation chambers. The level difference is measured then at each third octave of interest. The normalised level difference of the element $D_{n,e}$ is given by:

$$D_{\mathrm{n},e} = L_1 - L_2 + 10\lg\left[\frac{mA_0}{A}\right] \text{ dB},$$

where m is the number of units installed,
A_0 is reference area (m^2) = 10 m^2,
A is equivalent absorption area in the receiving room (m^2).

Results may also be expressed as a single number quantity for airborne sound insulation by comparison with a given reference curve which is the value at 500 Hz of the reference curve after shifting it in accordance with specified methods. This is known as the weighted element-normalised level difference, $D_{n,e,w}$. In terms of sound insulation the higher the value of $D_{n,e,w}$ the greater the sound insulation.
The spectrum adaptation term C_{tr} is a value in decibels added to the single number rating to account for the characteristics of traffic noise spectra and human response.

1.22 Façade Insulation

The façade insulation is given by:

$$G_A = R_A + 10\lg\frac{A}{S} - 3 \quad \text{dB(A)},$$

where R_A is the overall sound reduction of the façade (dB(A)),
A is the room absorption (m^2 sabines),
S is the total surface area of façade (m^2),
-3 is the façade correction.

The characteristic façade insulation is the façade insulation standardised to the volume and RT of the room. It is defined as:

$$G_{A,c} = G_A - 10\lg\left(\frac{0.161V}{ST}\right) \quad \text{dB(A)},$$

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

1.23 Combinations of Elements

In the field situation, (rather than in the laboratory situation), flanking paths exist. Here not all the energy from one room will enter an adjacent via a separating partition. This is important to remember when considering ventilation systems, as they can often provide an easy route for sound energy to be transferred from one place to another. Obvious examples are ventilation ductwork in over ceiling voids, and heating ductwork along perimeters of rooms.

In these practical situations, where a wall or ceiling is comprised of more than one element, the transmission loss of wall is reduced. The expression that relates the two is relatively complex as we saw in Section 1.20. However, for most purposes Figure 1.7 below can be used. Similarly, pipes or ductwork through walls that have air gaps around can severely reduce the TL of the composite structure. The graph illustrates how small areas, acting as airgaps, can greatly reduce the composite TL. For example where a wall with TL of 30 dB has a hole through it equivalent to 5% of the total wall area, the wall TL is reduced by 17 dB!

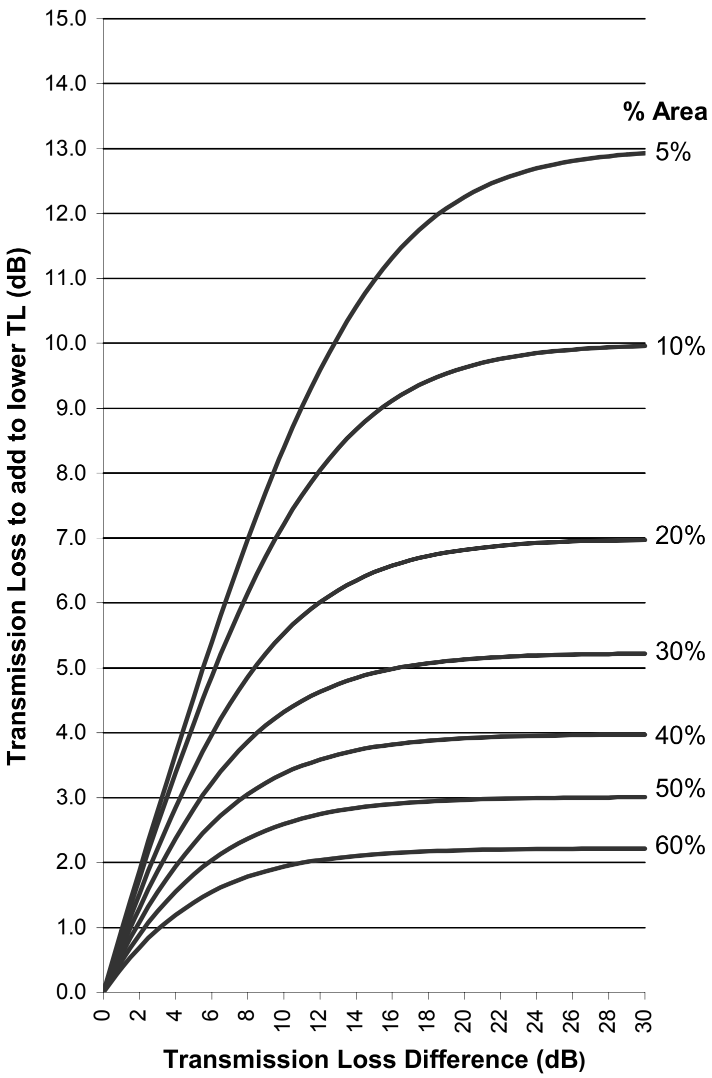


Figure 1.7 Calculation Chart for Combining Transmission Loss Values

Example

Consider a wall of area 8 m^2 with an average insulation of 45 dB. A window of area 2 m^2, and with average sound insulation of 15 dB is inserted into the wall.

The difference between the insulation values is 45 - 15 = 30 dB.

The window occupies 2/10 x 100 % = 20% of the total wall area.

Using the 20% curve in the Figure a value of 30 on the horizontal (x) axis gives an additional TL of 7 dB on the vertical (y) axis. Thus the composite TL of the wall with window is 15 + 7 = 22 dB.

The method, whilst simple, is an effective and quick way of calculating composite transmission loss values. Normally it would be applied for values at all frequencies of interest. Some authors (e.g. Op 't Veld) have related this to a simple table indicating the maximum noise reduction values (G_A) that can be obtained with ventilation openings in a façade, for a range of cross sectional area of the ventilation (Table 1.8).

Table 1.9 Maximum Sound Insulation
(Brick façade, room volume = 80 m^3, façade surface area 10 m^2)

Cross-section of the ventilation opening (cm^2)	Maximum noise reduction (dB(A))
10	40
50	34
100	31
200	28
300	26
400	25
500	24

While this approach is valid, the use of the figure above is probably of wider application, unless a more detailed analysis is to be conducted and multiple tables of maximum facade levels are calculated for all the different building elements (e.g. windows, louvres, doors etc.).

1.24 General Noise Control Principles

There are three distinct stages to the noise control process:

1. source selection;
2. transmission path identification and attenuation;
3. receiver criteria.

If we are to attempt to design a system that is quiet and also to achieve this in a cost effective way, attention to each of these areas is important. However, a pragmatic solution will often require us to concentrate more on one of these elements e.g. selection of the quietest fan, rather than recommend that a worker should be wearing hearing defenders!

For ventilation purposes we often have most control over the source i.e. which plant is selected, and the transmission path i.e. the route the noise (or vibration) travels to the receiver.

Choosing the assessment criteria for the receiver is crucial to the whole design process and is discussed in Chapter 4. We need to be designing a ventilation system which does not create a noise level that is inappropriate to the situation, occupant or activity it is intended for.

A generalised strategy, which can be followed for good acoustic design, is given in Figure 1.8.

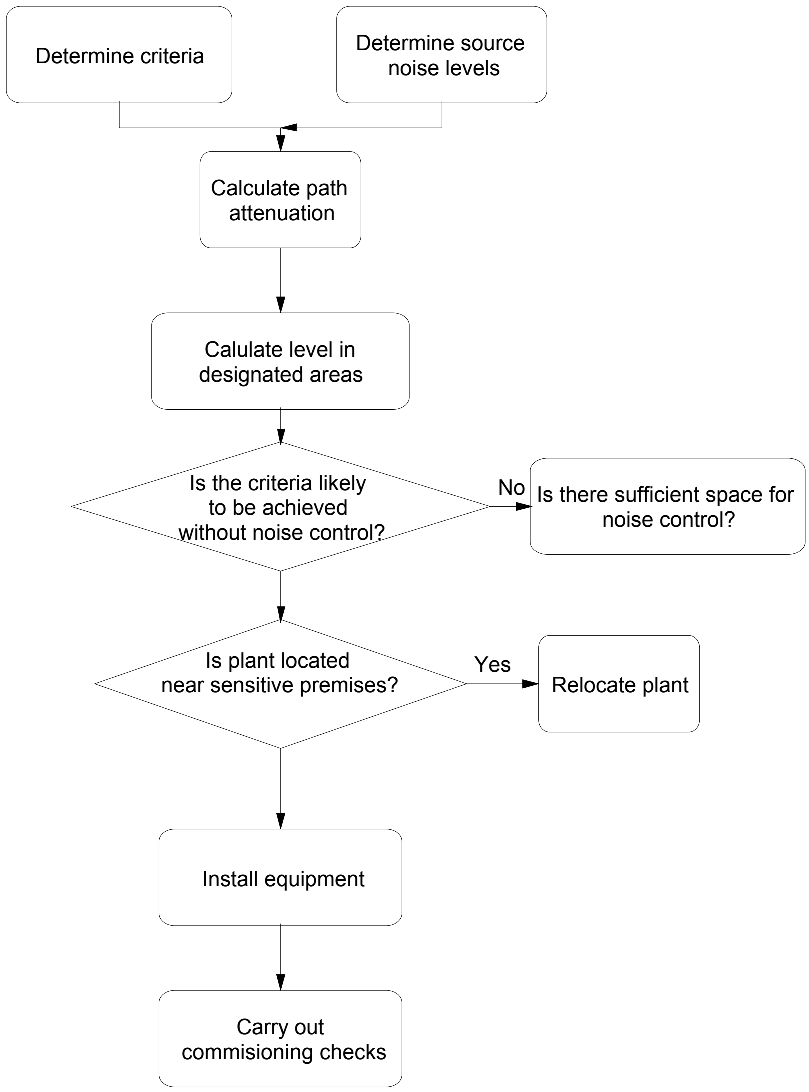


Figure 1.8 The Noise Control Process

2 Acoustics and Mechanical Ventilation

2.1 Introduction to Ventilation System Selection

Generally, an architect or building services engineer knows at the concept stage which type of ventilation system is to be used. Selection is based upon the various factors:

- building type;
- amount of cooling required;
- climate;
- end user;
- energy use;
- location.

There are three main types of mechanical ventilation systems (Shaw, 1987):

- **Balanced**, where air is drawn into a house from outside, and an equal amount is expelled to outside;
- **Supply only**, where a fan brings air into the house from outside. This raises the internal pressure, causing return air to leave the house via openings in the building envelope;
- **Exhaust only**, where a fan extracts air from inside a house. This lowers the internal air pressure, causing supply air to enter the house via openings in the building envelope.

From a noise perspective each of these systems potentially presents problems. Introduction of mechanical equipment into a dwelling immediately brings a potential noise source into areas and conflicts between noise levels and activity disturbance can occur. Any system which involves air moving at speed, and also air moving non-uniformly also induces regenerated noise. Reliance on openings in buildings to act as supply or return, immediately provides paths through which external noise can enter a building. In some cases the converse is also true, noise from within a building can escape and cause community annoyance. This emphasises the need to take real are needs when mechanical systems are being designed and installed.

Thus noise within a building is due to one (or a combination) of three causes:

1. external noise (entering a dwelling through ventilation openings, or building envelope defects);
2. internal noise (due to ventilation system noise);
3. path effects (where sound is transported through a building, either via a ventilation system, or via the building voids).

Op 't Veld has rated the importance of these three areas, Table 2.1.

Table 2.1 Importance of Noise Sources (after Op 't Veld)

	Natural ventilation	Natural supply/mechanical exhaust	Balanced ventilation
Outdoor noise	✓	✓	○
Internal (system) noise	×	✓	✓
Path effects	○	○	✓

Key:
× irrelevant/not applicable
○ in general of minor importance
✓ Important

This table helps in the selection of a ventilation system, by emphasizing the need for noise control in particular areas. However, the decision is often made based on non-acoustic considerations. To factor in the acoustic elements into the overall system performance the matrix below gives a useful aid to the process.

Table 2.2 Ventilation System Selection Matrix (after Seppänen, 1995)

	Natural	Passive stack	Passive stack with kitchen fan	Mech-anical exhaust	Mech-anical exhaust with kitchen fan	Mech-anical supply and exhaust
Acoustical environment	× (✓✓✓)	✓✓✓	✓✓	✓✓	✓✓	✓✓
Removal of pollutants	×	✓	✓✓	✓✓	✓✓✓	✓✓✓
Supply of fresh air where needed	×	✓	✓	✓	✓	✓✓✓
Independency of weather	×	×	×	✓✓✓	✓✓✓	✓✓✓
Thermal comfort	×	✓	✓	✓✓	✓✓	✓✓✓
Control by demand	✓	✓	✓✓	✓	✓✓	✓✓(✓)
Heat recovery	×	×	×	✓	✓	✓✓✓
Low first cost	✓✓✓	✓✓	✓✓	✓	✓	✓
Small space requirements	✓✓✓	×	×	✓✓	✓✓	✓

Key:
× poor
✓ reasonable
✓✓ good
✓✓✓ excellent

The diversity of equipment used for heating ventilation and air conditioning (HVAC) purposes is huge. However, the strategy to reducing noise can be stated by addressing four areas:

- selection of low noise equipment;
- use of low air volumes for the size of opening;
- reduction of fan speeds;
- correct installation of equipment.

As with any potential noise problem, the selection of the quietest piece of equipment can minimise subsequent noise problems. Most, if not all, manufacturers publish data for their products. Normally this will be stated in a dB(A) value, or octave values of sound power. An example of the data is given in the Table 2.3. Always study the data carefully, and ensure that when you are comparing different manufacturers data that you are comparing like with like. Additionally, remember that manufacturers' published data are for specific (hopefully standardised) test conditions and may not be representative of the application the equipment is being put to. If in doubt, measure a piece of equipment in-situ, in a similar building and application to the one you are concerned with, where you know the operating conditions.

Table 2.3 Typical Manufacturers' Fan Sound Power Data

Type	Fan diameter (mm)	Fan speed (rev/s)	Power (W)	Sound power (dB) at Octave band centre frequency (Hz)						
				63	125	250	500	1K	2K	4K
Axial	350	22	370	67	68	67	63	66	65	59
Axial	400	22	370	70	69	70	69	71	69	63
Axial	450	22	370	72	71	72	71	72	71	65
Axial	500	22	550	73	74	74	76	79	76	74

Correct installation of plant is all to do with placing equipment in the optimum position. From the noise perspective the optimum position is away from sensitive rooms, with non-sensitive rooms in between wherever possible. Thus in a domestic situation the location might be within a kitchen where low noise levels are not essential. In commercial premises, plant might be located in a basement or in a plant room on the roof. Note that this latter option will require a separate noise analysis if there are noise sensitive properties in the vicinity. Care needs to be taken in locating ductwork in positions where noise breakout will not interfere with normal activities.

An indication of the frequency content of different mechanical components is given in Figure 2.1.

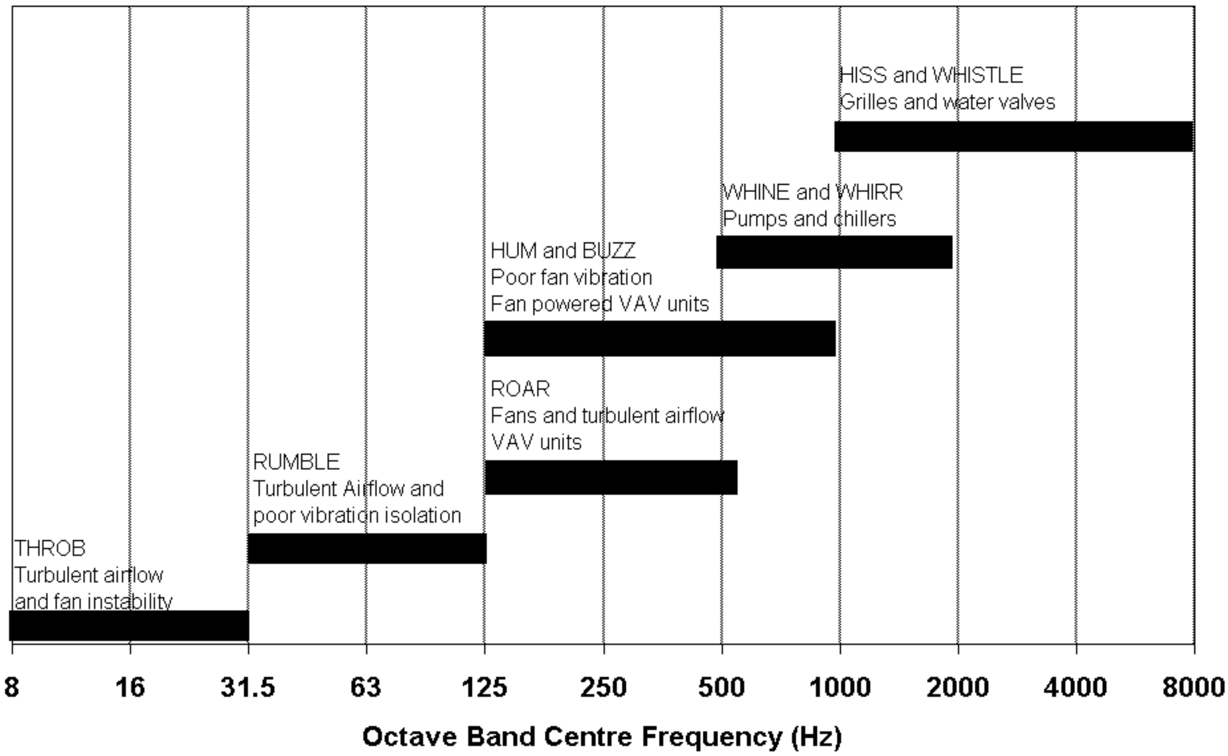


Figure 2.1(a) Frequency Ranges of Likely Sources of Sound-Related Complaints
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)

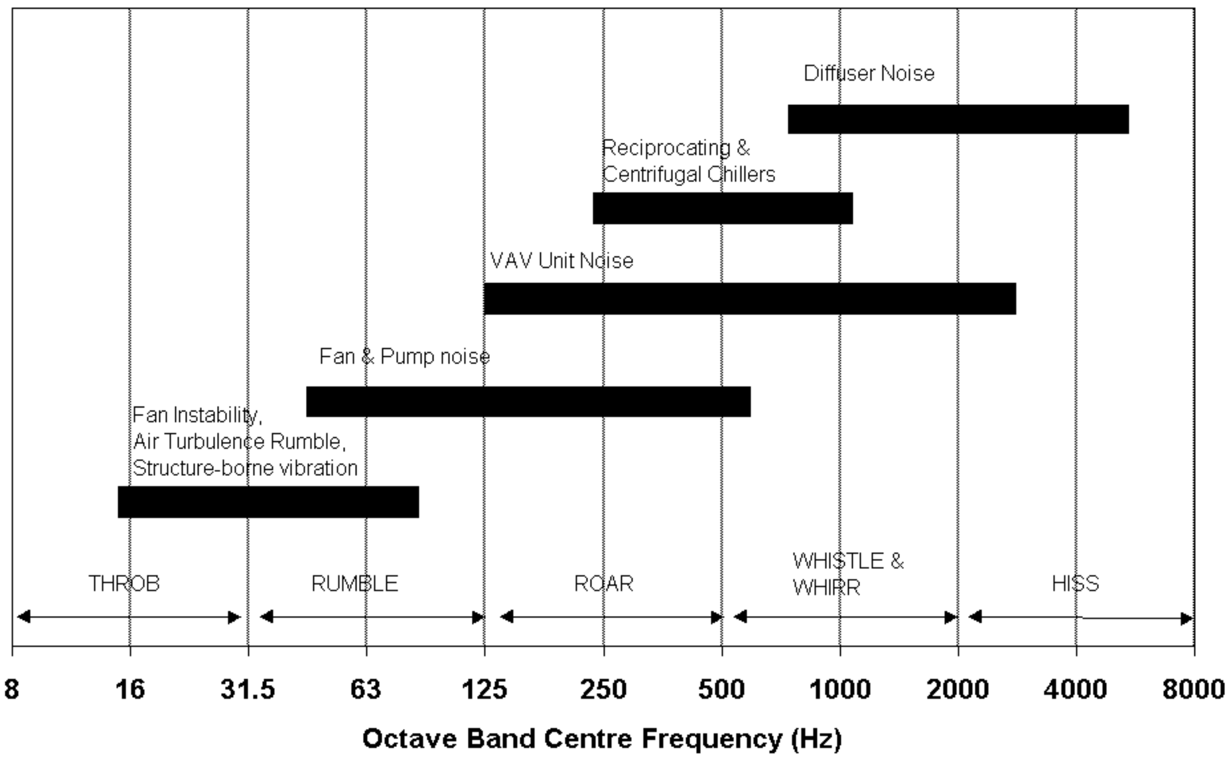


Figure 2.1 (b) Frequencies at which Different Types of Mechanical Equipment Generally Control Sound Spectra *(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)*

Schaffer (1993) relates subjective and objective characteristics of ventilation components which, if present, are likely to cause complaints:

Table 2.4 Sources and Characteristics of Annoying HVAC Noises

Frequency range (Hz)	Noise description	Noise source
8 - 32 15 - 125 63 - 250 125 - 500 250 - 1000 1000 - 8000	throb rumble roar hum and buzz whine hiss and whistle	turbulent airflow and fan instability turbulent airflow and poor vibration isolation fans and turbulent airflow, VAV fans poor vibration isolation, fan powered VAV pumps and chillers grilles and water valves

The use of air-conditioning that does not operate continually can be problematic. This is due to the positive application of air-conditioning noise as an aid to office privacy (see Chapter 4 for further details). Where this type of zoning does occur, it is often necessary to introduce additional noise into spaces. This is provided by broadband noise fed into loudspeakers in from the ceiling, and is called masking noise.

The most comprehensive guides to noise control of HVAC systems are those produced by ASHRAE. Their Handbooks and other publications should be considered indispensable, if detailed octave band analysis of ventilation systems is to be undertaken.

2.2 Major Boundary Conditions

A generalised diagram of mechanical ventilation systems is shown in Figure 2.2. The primary noise source is the fan, with the other elements contributing to the potential problems! The noise produced by the fan is essentially aerodynamic noise. This noise is thus dependent upon the type of fan and the speed of its operation. As might be expected linear (laminar) flow creates least noise, whilst system designs which introduce protuberances and discontinuities cause more turbulence to occur and resultant increases regenerated noise levels. The main parts of the system that cause this turbulence are:

- branching elements;
- duct work – e.g. turning vanes;
- dampers;
- diffusers.

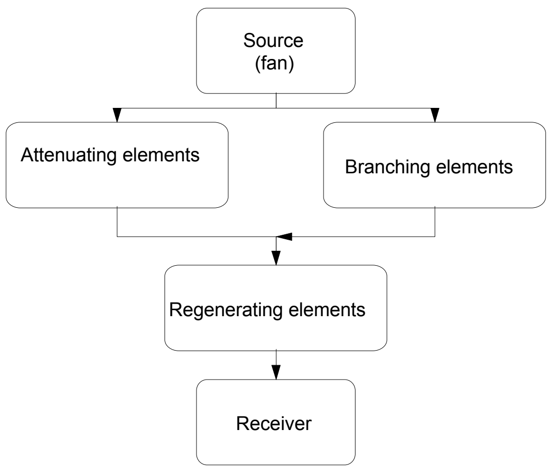


Figure 2.2 Generalised Mechanical Ventilation

Many manufacturers give guidelines in their catalogues for the reduction of excess turbulence. Often these are given for reasons of ensuring the best airflow. The secondary effect of lower noise levels is thus a bonus!

2.3 Fans

Fans will always operate at their quietest when the duty corresponds to its maximum efficiency. Their radiated sound power can be simply calculated from the following equation:

$$L_W = C_F + 10\lg Q + 20\lg P + \frac{E}{3} - 48 + BFI \quad dB(re.10^{-12}W)$$

where C_F is a correction (see Table 2.5) (dB),
E is percentage of peak efficiency,
P is static pressure (Pa),
Q is the flow capacity (m^3/h).

Table 2.5 Corrections for Fan Power Prediction[2]

	Octave Band Centre Frequency (Hz)								
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**BFI**
Propeller (wheel diameter < 3.5 m)	48	51	58	56	55	52	46	44	5
Centrifugal (< 0.9 m)	36	38	36	34	33	28	20	12	3
Axial (< 1 m)	40	41	47	46	44	43	37	35	5

The blade frequency increment (BFI) correction is only applied to the octave containing the blade passing frequency (BPF). The BPF is given by:

$$BPF = \frac{N\,x\,RPM}{60} \quad Hz\cdot$$

2.3.1 Axial

Axial fans, whilst being more immune from low frequency rumble, will often have a characteristic tone associated with them. Control of this is important to prevent annoyance.

Axial fans come in three main forms:

- propeller (3-6 blades);
- tubeaxial (6-9 blades);
- vaneaxial. (8-16 blades).

[2] A more complete list of corrections (including for BFI) can be found in the ASHRAE Handbook 1999 HVAC Applications

Figure 2.3(a) Bifurcated Axial Flow Fan (Fantech BFA Series) *(Reproduced with permission of Fantech Pty. Ltd.)*

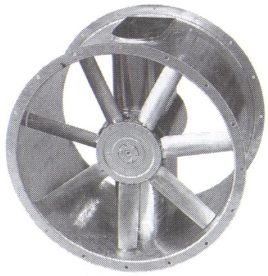


Figure 2.3(a) Direct Drive Axial Flow Fan (Fantech AP Series) *(Reproduced with permission of Fantech Pty. Ltd.)*

2.3.1.1 Propeller Fans

These fans, typically found in cooling towers, are amongst the most difficult to control the radiated noise, without causing a large decrease in overall performance. Lined ducts or hoods can be effective, but losses in static airflow needs to closely checked against the design specification. Additionally, isolation from the ductwork is important to reduce vibration. These fans tend to have few blades (3 to 6) and the blade passing frequency is often exhibited as a strong pure tone.

2.3.2 Centrifugal

Centrifugal fans generate low-frequency noise which is characterised as a rumble or roar.

They can come in the form of:

- backward curved;
- forwards curved;
- radial bladed;
- in-line (tubular).

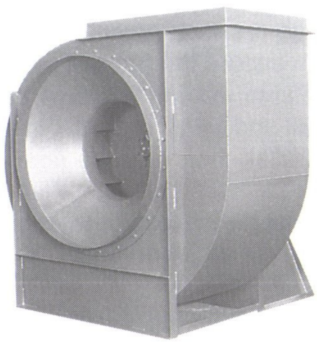


Figure 2.4 Centrifugal Flow Fan *(Reproduced with permission of Fantech Pty. Ltd.)*

2.4 Air Supply

2.4.1 Variable Air Volume Units (VAV)

The majority of commercial air conditioning systems run at partial load most of the time. This is to accommodate varying demands of heating and cooling that is required in normal commercial or residential buildings. Local thermostatic sensors control airflow through terminal units, inlet guide vanes (backwards fans), speed variation (i.e. frequency control for forward and backwards fans) and blade pitch control (axial fans). All these controls can give different results in the system acoustics and resulting noise levels. Speed control and blade pitch control can be recommended because they change the operating point. Where inlet guides, used to control the fan, disturb the inlet airflow and can typically increase operating noise levels by 5 dB. In all cases, fan selection must be done for the full range of operating conditions.

In VAV systems airflow is continuously changing. In some cases, the fan's operating point can reach some part of the operating curve where efficiency is quite bad, and the induced sound power level may be higher than for the nominal airflow.

Note that the problem of low background noise levels is not exclusive to VAV systems, but it is here that the problem is most prevalent, and noise-masking systems may need to be introduced to improve privacy.

2.4.2 Air Handling Units (AHU)

These are normally driven using a centrifugal fan. Thus all the guidelines in the later section about these fans should be observed. However, it is advisable that the distance from the intake and nearest wall is at least one unit height.

In situations where low noise systems are important blow through units are preferable to draw air through units, although radiated noise levels on the inlet side are comparable.

2.5 Air Exhaust

2.5.1 Extractor Fans

Extractor fans are used as a means of local air exhaust. They typically comprise of a fan in a wall or window, or an axial fan in a short length of ductwork. Some guidance is available relating flow rates to sound pressures (Table 2.6 below).

Table 2.6 SPL for Different Flow Conditions
(Source: Op ‘t Veld)

Fan type	Flow capacity (m^3/h)	SPL at 1m (dB(A))
Axial	200	51
	300	55
	750	57
	1400	65
Axial in duct	100	54

Break in of external noise is possible with extractor fans. However, in general they are placed in non-noise sensitive area such as bathrooms and kitchens. More problematic is the need to resilient mount the fan units from the ceiling, to prevent structure-borne noise transmission.

Quirt (1991) reported a series of tests on residential bathroom and kitchen extract fans. Table 2.7 summarises his findings.

Table 2.7 Sound Power Levels for Extract Fans

Location	**Flow (l/s)**		**PWL (dB(A))**	
	Field	**Laboratory @ 25Pa**	**Field**	**Laboratory**
Bathroom	30	48	55	57
Kitchen	42-78	91	68	67

Quirt noted that the SPL measured in the room decreased with distance and that the perceived loudness was also dependent upon distance. Differences between the field and laboratory measurements were attributed to room characteristics (eg. absorption). An expression was developed to predict noise levels from these fan types:

$$L_p = L_w - 10\lg\left(\frac{6}{4\pi r^2} + \frac{4}{R}\right) \qquad \text{dB}$$

where the variables are as defined in Chapter 1.

2.6 Cooker Hoods

Differing practice appears to exist for specification of cooker hoods between Australasia and Europe. In Australasia, air speeds are specified to be > 10 ms^{-1} to minimise debris deposits within ductwork. Resultant pressure drops need to be kept as small as possible. A 2D cylindrical attenuator is normally the best solution. (2 pole fans are often used, giving speeds of 2880 rpm. The resulting noise can be difficult to contain.)

However, in Europe silencers are rarely used for these installations because oil deposits quickly decrease any attenuation of course, silencers can be washed regularly but, in France (for example) fire protection regulation reduces the number of possible solutions. Fans are usually selected with 1000 rpm - 1500 rpm, and ducts are dimensioned with less than 7 ms^{-1} and less when ducts go through occupied rooms.

2.7 Fan Operation Point

Sound power levels from fans are not constant over the range of operating condition. Manufacturers provide graphs of efficiency, pressure and flow rate. Lowest levels of noise will occur when fans are operating at their peak efficiency. Blazier (1993) discussed this with particular regard to the trend towards packaged systems rather than centralised fan rooms. He identifies a number of potential problems:

- air handling equipment which is placed in noise sensitive locations;
- packages are often noisier due to space and economic considerations;
- fans are operated below peak efficiency causing an increase of low frequency noise ('rumble');
- lower air flows result in lower diffuser noise, which acts as masking noise. Fan noise then becomes subjectively more annoying.

A number of system design options allow reductions in noise levels to be achieved, the principal one being:

- the use of variable frequency drives (VFD) rather than variable inlet vanes (VIV) to change flow capacity.

This type of system change can reduce noise levels by approximately 5 dB, and are typically 17 dB lower than a system using inlet vanes. An additional benefit is that of reduced power consumption.

2.8 Regenerated Noise

Correct design of an air-conditioning system is the best tool to prevent noise problems at a later stage. It is the layout, coupled with the airflow velocities, that determine the final noise levels in a room. Regenerated noise is where the aerodynamics of a system actually increases the level within a duct. The areas that are of principle concern are:

- terminal devices (grilles, diffusers) and control valves (eg. dampers);
- transitions (where different area ductwork is joined);
- bends;
- junctions where ductwork splits into multiple ducts.

Dampers are generally installed for two reasons; as control dampers and as fire dampers. In either case they should be designed so that they minimise airflow disruption through a duct. Systems should be designed so that optimum airflow is provided, without the need to rely on dampers to correct large imbalances of airflow. They should only be used as a method of making small changes in system balancing.

2.9 Connecting Ductwork

Connecting ductwork can offer useful amounts of attenuation. Ductwork is normally obtainable in rectangular or circular cross-section, and as lined or unlined. Even straight, unlined sections can give useful noise attenuation. However, it should be noted that circular ductwork is much stiffer than rectangular ductwork and does not attenuate noise as well. Note that this property can be used to good effect where ductwork is to run through noise-sensitive areas, as the noise breakout is less as a result of this property. Note also that rectangular section ductwork has more problems with air-tightness than does circular section.

The two main factors, which need to be considered when connecting fans to ductwork, are:

- turbulence;
- breakout.

Adhering to the following recommendations can reduce noise:

- ensure duct transitions are smooth;
- use rectangular ducts (Note that (i) where ducts need to be rectangular these should be square in cross-section if possible – i.e. low aspect ratio (ii) lined rectangular ducts give higher attenuation along the duct path.);
- breakout can be reduced by using round ducts or increasing duct wall thickness (density);
- maximum air velocity guidelines should be followed, where more detailed analysis is not to carried out (see Tables below).

Table 2.8 Maximum Ductwork Airflow
(Source: Op 't Veld)

Duct location	**Maximum airflow velocity (m/s) for specified internal room level**	
	30 dB(A)	**35 dB(A)**
main duct	5.0	6.0
branch	4.0	5.0
branch to grilles	2.0	3.0

Table 2.9 Maximum Ductwork Airflow

Noise criteria (NR or NC)	**In-duct air velocity (m/s)**	
	Main	**Branch**
20	4.5	3.5
25	5.0	4.5
30	6.5	5.5
35	7.5	6.0
40	9.0	7.0

The values in the above tables are those for low-flow situations, e.g. residential environment, where ductwork is likely to be within the rooms of concern. Where larger, commercial, installations are in place ductwork is often concealed behind walls or ceilings. Higher flow rates are acceptable in these situations (Table 2.10).

Table 2.10 Recommended Maximum Airflow Velocities for Various Installations
(Source: Schaffer, 1993)

Duct location	**Rating (RC or NC) in adjacent occupancy**	**Maximum airflow velocity (m/s)**	
		Rectangular	***Circular***
In shaft, or above solid drywall ceiling	45	17.5	25.0
	35	12.5	22.5
	<25	8.5	15.0
Above suspended acoustical ceiling	45	12.5	22.5
	35	9.0	15.0
	<25	6.0	10.0

If analysis of the total path attenuation is to be made then more precise data and guidance available from ASHRAE should be used. A selection of these values is reproduced below.

Table 2.11 Rectangular Duct Insertion Loss Values
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)

Internal cross-section (mm)	**Insertion loss (dB/m) Octave band centre frequency (Hz)**					
	125	250	500	1000	2000	4000
Rectangular duct lined 25mm fibreglass						
150x150	0.6	1.5	2.7	5.8	7.4	4.3
300x300	0.4	0.8	1.9	4.0	4.1	2.8
460x460	0.3	0.6	1.6	3.3	2.9	2.2
Rectangular duct lined 50mm fibreglass						
150x150	0.8	2.9	4.9	7.2	7.4	4.3
300x300	0.5	1.6	3.5	5.0	4.1	2.8
460x460	0.4	1.2	2.9	4.1	2.9	2.2

Table 2.12 Circular Duct Insertion Loss Values
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)

Internal diameter (mm)	**Insertion loss (dB/m) Octave band centre frequency (Hz)**							
	63	125	250	500	1000	2000	4000	8000
Circular duct lined 25mm fibreglass								
150	0.38	0.59	0.93	1.53	2.17	2.31	2.04	1.26
305	0.23	0.46	0.81	1.45	2.18	1.91	1.48	1.05
610	0.07	0.25	0.57	1.28	1.71	1.24	0.85	0.80
1015								
Circular duct lined 50 mm fibreglass								
150	0.56	0.80	1.37	2.25	2.17	2.31	2.04	1.26
305	0.42	0.67	1.25	2.18	2.18	1.91	1.48	1.05
610	0.25	0.46	1.01	2.00	1.71	1.24	0.85	0.80
1015	0.16	0.24	0.73	1.63	0.68	0.57	0.55	0.58

2.10 Terminal Devices

Terminal devices are those parts of the air-conditioning system which interface with the air in a room i.e. grilles, diffusers etc. They are used for controlling and distributing heat or cooling in to spaces. They act as noise sources in two capacities:

1. allowing noise to escape from the air distribution system;
2. airflow through the terminal unit generates noise.

Where terminal devices are installed without fans, they do *not* contribute greatly to the low frequency noise level. The terminal device can have a very beneficial property when it comes to considering escape of the noise. The impedance mismatch from the duct to the room volume reflects sound energy back into the ductwork. This is especially evident at low frequencies, such that a large number of small area outlets will result in a reduced level of low frequency noise being transmitted compared with a larger terminal device.

The end reflection loss is frequency dependent and can be obtained from Table 2.13. The data in the table is only valid where the duct is 3 - 5 duct diameters from the primary ductwork, without diffusers attached.

Table 2.13 End Reflection Loss - Duct Terminated Flush with Wall
Note: Do not apply for linear diffusers or diffusers tapped directly into primary ductwork. If duct terminates in a diffuser, deduct at least 6dB.
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)

Mean duct width (mm)	End reflection loss (dB) at Octave band centre frequency (Hz)				
	63	125	250	500	1000
150	18	13	8	4	1
200	16	11	6	2	0
250	14	9	5	2	0
300	13	8	4	1	0
400	10	6	2	1	0
510	9	5	2	1	0
610	8	4	1	0	0
710	7	3	1	0	0
810	6	2	1	0	0
910	5	2	1	0	0
1220	4	1	0	0	0
1830	2	1	0	0	0

The other factors that are important are:

- length of branch to terminal device from main ductwork (minimum 2 to 5 diameters of duct is best);
- whether the termination is flush with the surface or in free space;
- poor installation design.

The ASHRAE handbook suggests that design of the system to enable maximum end reflection losses is often worthwhile with losses equivalent to 15 m - 25 m of lined ductwork being achieved. Equations for end reflection losses are also given in ASHRAE. However, the values in the table reproduced above are a useful guide.

2.10.1 Prediction of Levels

The sound power level from a grille can be estimated as:

$$L_W = -4 + 70\lg v + 30\lg\xi + 10\lg S \quad dB(A)$$

where v is the air velocity of the grille (m/s),
ξ is the airflow resistance factor,
S is the surface area of the grille (m^2).

To aid selection of grilles, Op 't Veld has given a table, based on an acceptable indoor level of 35 dB(A).

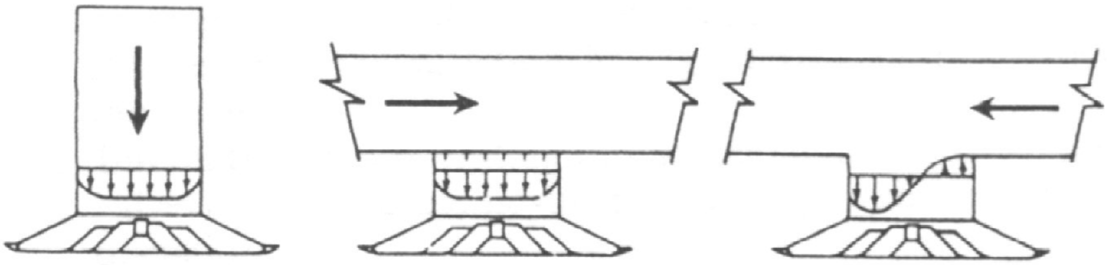
Table 2.14 Sound Power of Grilles

No. of grilles	Maximum individual grille sound power (dB(A))
1	35
2	32
3	30

Testing of air diffuser devices should be to ASHRAE 70 *Rating of Air Diffusers and Air Diffuser Assemblies* and for air terminals to ASHRAE 130P *Air Terminals*

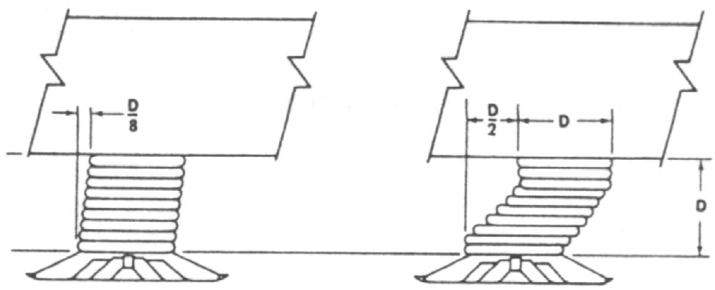
2.10.2 Installation

The way the air travels through a grille or diffuser is crucial to good acoustics. Poor installation can easily increase sound levels by 12 dB to 15 dB. Typical configurations are given in the figures below from ASHRAE.



Basis of manufacturer's rating | *Sound levels same as manufacturer's rating with equalising grid* | *Sound levels up to 12 dB higher with no equalising grid*

Figure 2.5 Proper and Improper Airflow Conditions to an Outlet
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)



Sound levels same as manufacturer's ratings | *Sound levels 12 to 15 dB higher than manufacturer's ratings*

Figure 2.6 Effect of Proper and Improper Alignment of Flexible Duct Connector
(Reprinted with permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers from the 1999 ASHRAE Handbook)

These figures illustrate some of many factors that need to be considered (Op 't Veld, Schaffer 1993):

- Airflow must be distributed evenly over surface of device (see Figure 2.7), using turning vanes if necessary;
- Use sound power data from manufacturer. Select one that is rated at 5 points below the required value of NC, RC, or dB(A) rating. (This is important because the airflow is rarely uniform, and turbulence increases noise levels);
- Ductwork to a supply device should be straight at least 3 equivalent duct diameters upstream of the device;
- Use several smaller outlets rather than one large one, thus reducing the air volume per diffuser;
- Airflow controls (e.g. dampers) should be positioned at least 3 equivalent duct diameters upstream from the device;
- Suspend a deflector plaque beneath duct outlet (at least 1 duct diameter below outlet);
- Maximum air velocity guidelines should be observed, where more detailed analysis is not to be carried out (see Tables 2.15 and 2.16);
- Rigid ducts should be installed, rather than flexible ductwork, at the diffuser connection.

Table 2.15 Maximum Airflow at Grilles (after Op 't Veld)

Position of grille	**Maximum airflow velocity (m/s) for specified internal room level**	
	30 dB(A)	*35 dB(A)*
wall or floor	3.0	3.5
ceiling	2.3	2.5

Table 2.16 Maximum Recommended Air Velocities (after Egan, 1988)

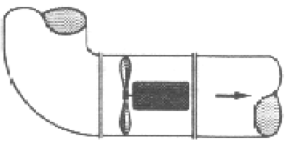
Noise criteria (NC)	**Airflow velocity (supply) (m/s)**	**Airflow velocity (return) (m/s)**
15 - 20	1.27 - 1.52	1.52 - 1.83
20 - 25	1.52 - 1.78	1.83 - 2.13
25 - 30	1.78 - 2.16	2.13 - 2.59
30 - 35	2.16 - 2.54	2.59 - 3.05
35 - 40	2.54 - 2.92	3.05 - 3.51
40 - 45	2.92 - 3.30	3.51 - 3.96

These values assume lined ducts, and grilles with a minimum of 12mm slot openings. Other authors (e.g. Templeton, 1997) recommend values 20% higher than these.

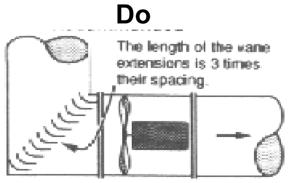
2.11 General Installation Guidelines

Most fan manufacturers give guidance as to the correct connection of bends and ductwork to their products. The figures below (Fantech, Schaffer 1993) give a useful set of rules for connection of fans to duct work.

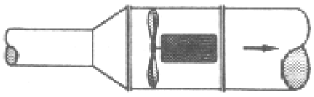
Don't | **Do**



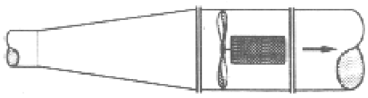
Upstream radius elbow creates imbalance at elbow.



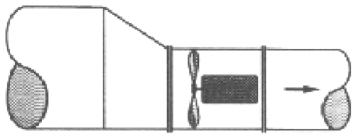
Square inlet elbow with extended trailing edge vanes deliver less turbulent airflow to fan inlet.



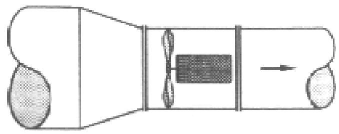
Abrupt inlet transition causes turbulence.



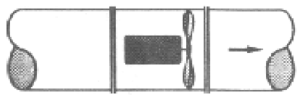
Gradual (1:7) exapnsion of inlet duct avoids impeller turbulence.



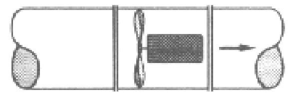
Assymetrical transition creates imbalanced load on fan, with excess tubulence and noise.



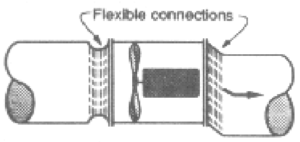
Symmetrical transition balances load on fan, which minimises turbulence and noise.



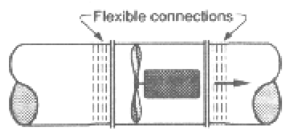
Motor upstream of impeller increases turbulence and noise



Motor downstream from impeller minimises turbulence and noise



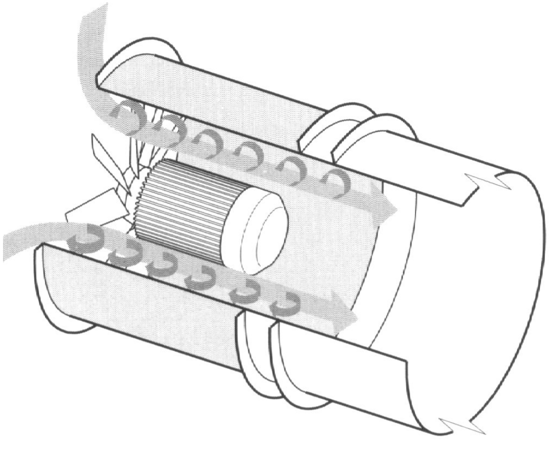
Slack or offset flexible connection causes turbulent air flow



Taut, in-line flexible connections provide optimal vibration isolation without creating turbulence

Figure 2.7 Guidelines for Ducted Axial Flow Fan Installations
(Source: Schaffer, 1993, reproduced with permission of Sound Attenuators Limited)

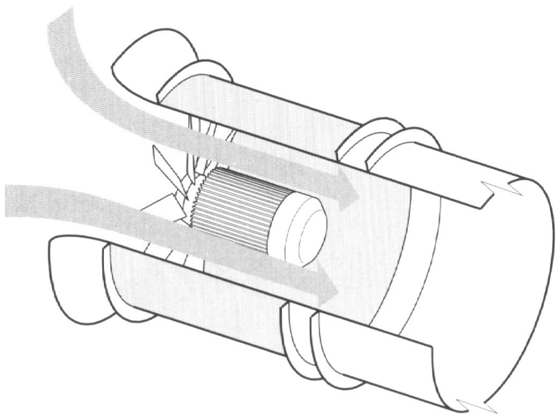
Don't



(Reproduced with permission of Fantech Pty. Ltd.)

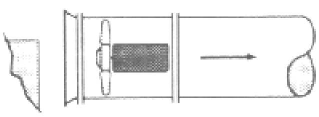
The abrupt entry into the fan will generate turbulence at the impeller, increasing the noise level. It will also degrade performance.

Do

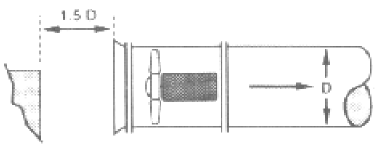


(Reproduced with permission of Fantech Pty. Ltd.)

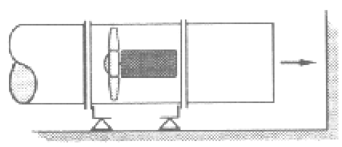
The inlet cone (or bell mouth) causes the airflow to be uniform, reducing turbulence and thus reducing the noise level.



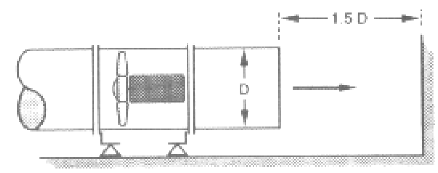
Avoid obstructions close to fan inlets. Part of the impeller is starved for air.



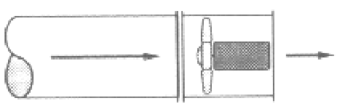
Allow a space of at least 1.5D at the fan inlet.



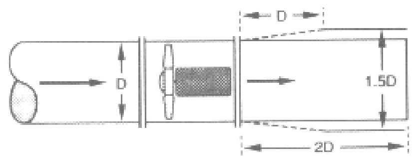
Avoid obstruction close to the fan outlet.



Allow a space of at least 1.5D at the fan outlet.



Avoid having the fan terminate the discharge end of the system.



Add a duct length of 2D, or an outlet cone of 1.5D at the discharge end of the fan. Extra ducting after the cone is better.

Figure 2.8 Guidelines for Unducted Axial Flow Fan Installations
(Source: Schaffer, 1993, reproduced with permission of Sound Attenuators Limited)

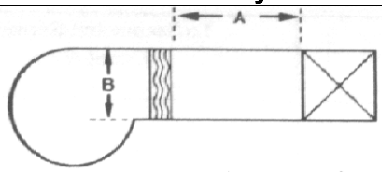
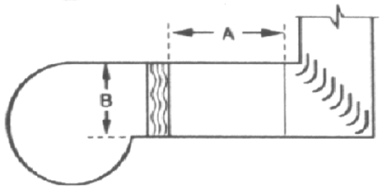
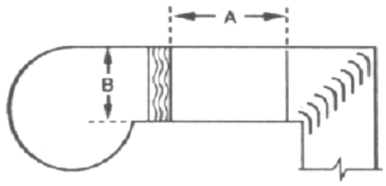
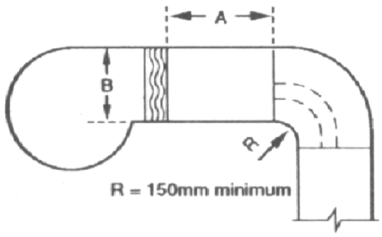
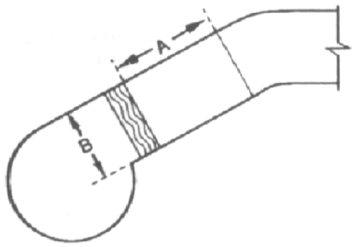
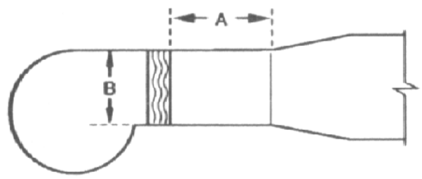
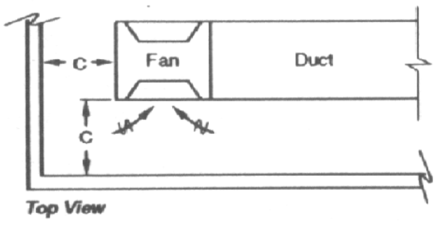
Fan/duct layout	Comment
	*Bad if A < 3B*
	*Bad if A < 3B* *Very bad if turning vanes are deleted*
	*Fair if A < 3B* *Bad if A < 1.5B*
	*Good if A > 3B* *Fair if A > 1.5B* *Bad if A < 1.5B*
	*Very good if A > 3B*
	*Best if A > 1.5B* *Transition wall slopes of 1:7 preferred.* *Slopes of 1:4 permitted if inlet velocity is less than 10 m/s.*
	*Best if C > 1.5 x fan wheel diameter* *Good if C = fan wheel diameter* *Poor if C < fan wheel diameter*

Figure 2.9 Guidelines for Centrifugal Fan Installations
(Copyright 1991 by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. Reprinted with permission from A Practical Guide to Noise and Vibration Control for HVAC Systems)

2.12 Flanking Due to Ductwork

Where a duct services a number of rooms the layout of the ducts can be critical. Noise from one room can be transmitted to another, causing privacy or annoyance problems. This is called flanking or cross talk. The figure shows how ducting is best located outside rooms, with branches going to the individual spaces. Lined ductwork will reduce the activity noise from one room being transferred to another. In some cases a silencer will also be necessary.

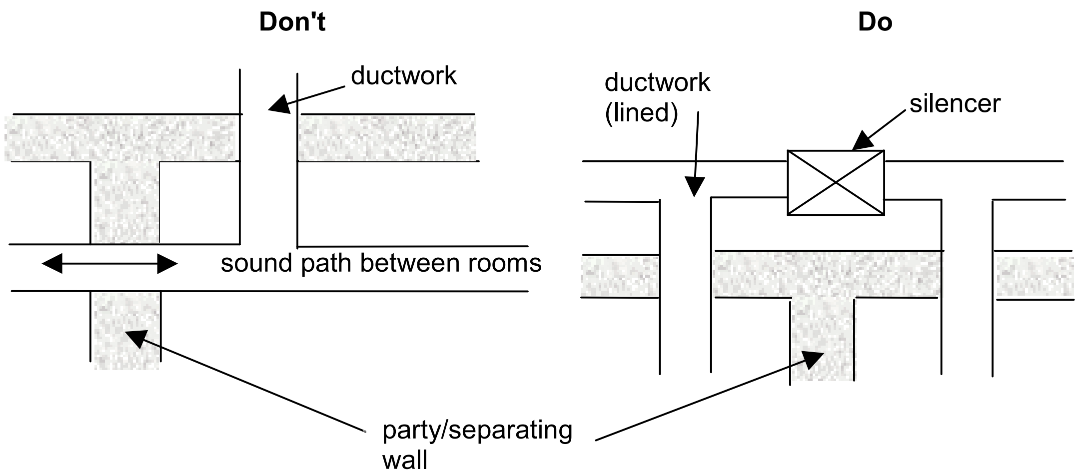


Figure 2.10 Ductwork Location to Prevent Cross Talk

2.13 Packaged Domestic Units

Packaged room air conditioners are a convenient form of local air conditioning for domestic or small commercial installations. They are easily installed in walls or window mounted. They also require little in the way of ductwork and thus eliminate potential cross-talk problems. Sound levels vary between manufacturers, the table below giving typical values.

Table 2.17 Typical Manufacturer's Data for Domestic Packaged Room Air Conditioners
[1] levels measured in a 100m^3 volume room with RT of 0.5 s
(Source: Carrier US)

Cooling capacity (kW)	Heating capacity (kW)	Internal [1] SPL at 2 m (dB(A))	External SPL at 2 m (dB(A))	Airflow indoor (l/s)	Size (height x width x depth) (mm)	Weight (kg)	Model
2.43	2.40	49-51	58	116	314 x 470 x 528	33	VF125H
4.4	4.25	53-59	64	160-205	395 x 620 x 716	65	VF175H
6.44	6.15	51-57	67	182-236	445 x 660 x 735	74	VF250H

Portable coolers are a convenient of providing localised cooling on an occasional nature.

Table 2.18 Typical Manufacturer's Data for Portable Coolers
(Source: Carrier US)

Cooling capacity (kW)	Heating capacity (kW)	SWL (dB(A))	Airflow indoor (l/s)	Size (height x width x depth) (mm)	Weight (kg)	Model
1.83	2.0	52 - 56	89 - 72	850 x 440 x 350	27.5	51AKM006G
3.27	1.6	53 - 57	97 - 133	850 x 330 x 430 (required outdoor unit 430 x 413 x 230)	30 11.5 (outdoor)	51AKM012

Both the above room units produce low noise levels which are unlikely to cause annoyance or speech interference in normal room use. Lower levels can be achieved by the use of smaller reverse cycle units mounted near ceiling height. Levels for these are typically 32 dB(A) - 42 dB(A) at 2 m. However their cooling capacity is also reduced to around 2.5 kW.

One of the issues with small packaged belt driven units (Blazier, 1993) is the tendency to use the fan housing as a structural support for the bearings and motor. The drive system is thus directly coupled to the housing and radiated noise results, normally as low frequency, rumbly, noise. The nature of low frequency noise is that this is often perceived as beats, or periodic changes in perceived loudness. Solutions to this are to ensure that coupling of components are isolated from each other. Additionally, the correct ratio of belt length to sheave can reduce the vibration.

When choosing a packaged system the best solution to reducing noise is to purchase one with a direct drive system and variable frequency power.

2.14 Passive Attenuation

Passive attenuation of noise is achieved by the use of attenuators or silencers. The range of attenuators available is wide but they mostly are constructed in the same way; a duct lined or split by sound absorption material. There are three types of silencer generally available:

1. cylindrical;
2. cylindrical pod;
3. rectangular.

Selection of the appropriate silencer requires optimisation of the most economic selection for the job. The selection is in the order of:

- length;
- acoustical performance;
- pressure drop;
- cross-section area.

The performance of a silencer is characterised by its insertion loss (IL). This is measured in dB and indicates the acoustic ability of a silencer to reduce the sound pressure level. The values given below are the reduction in sound power level of a system connected to a fan, with and without the attenuator connected. The insertion loss is affected by airflow direction as well as air velocity. Attenuation is provided by absorptive material enclosed by a perforated metal sheet. Often this is protected from other environmental factors by a protective layer such as Melinex. The lining can cause a slight reduction in performance at higher frequencies.

2.14.1 Cylindrical Silencer

A typical silencer is shown in the figure. The outer material can be varied according to its application. Circular silencers are convenient when attenuating noise from axial fans, and when breakout needs to be minimised. However, they are more expensive than rectangular attenuators. They are often specified as multiples of the duct diameter D, i.e. 1D or 2D silencer.

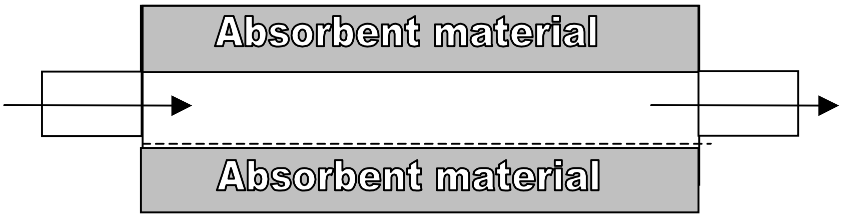


Figure 2.11 Idealised cylindrical attenuator

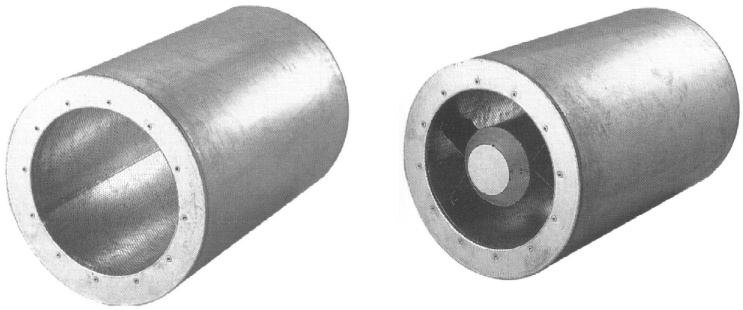


Figure 2.12 Cylindrical Silencers Non-Pod and Pod
(Reproduced with permission of Fantech Pty. Ltd.)

Typical insertion loss figures are given in the Table below.

Table 2.19 Insertion Loss Figures for Open Type Circular Silencer
(Nominal 1 diameter length)

Length (mm)	Static insertion loss (dB) at octave band centre frequencies (Hz)							
	63	125	250	500	1000	2000	4000	8000
300	1	3	5	9	13	10	8	7
600	2	3	5	9	13	10	8	7
1150	3	4	9	14	13	8	7	6
1800	4	6	11	13	10	7	5	5

Table 2.20 Insertion Loss Figures for Pod Type Circular Silencer
(Nominal 1 diameter length)

Length (mm)	Static insertion loss (dB) at octave band centre frequencies (Hz)							
	63	125	250	500	1000	2000	4000	8000
300	3	6	8	13	20	19	16	14
600	4	6	9	14	21	19	16	13
1150	5	7	11	20	20	16	13	11
1800	5	7	12	18	17	12	10	9

2.14.2 Rectangular Silencers

Rectangular units are provided in a variety of standard module widths, with the height being specified by the application. The cross-sectional area and the % open area are determined by the requirements of velocity and allowable pressure drop.

Selection of attenuators is similar to the procedure described in Chapter 3 for louvres and is represented in the flow diagram below. Four parameters are required for correct selection of a silencer:

- required insertion loss;
- air flow requirements;
- maximum pressure drop across attenuator;
- physical space available.

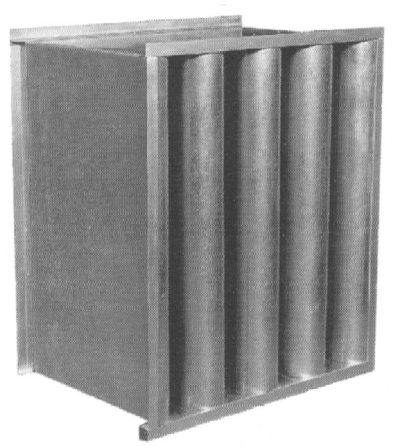


Figure 2.13 Rectangular duct silencer *(Reproduced with permission of Fantech Pty. Ltd.)*

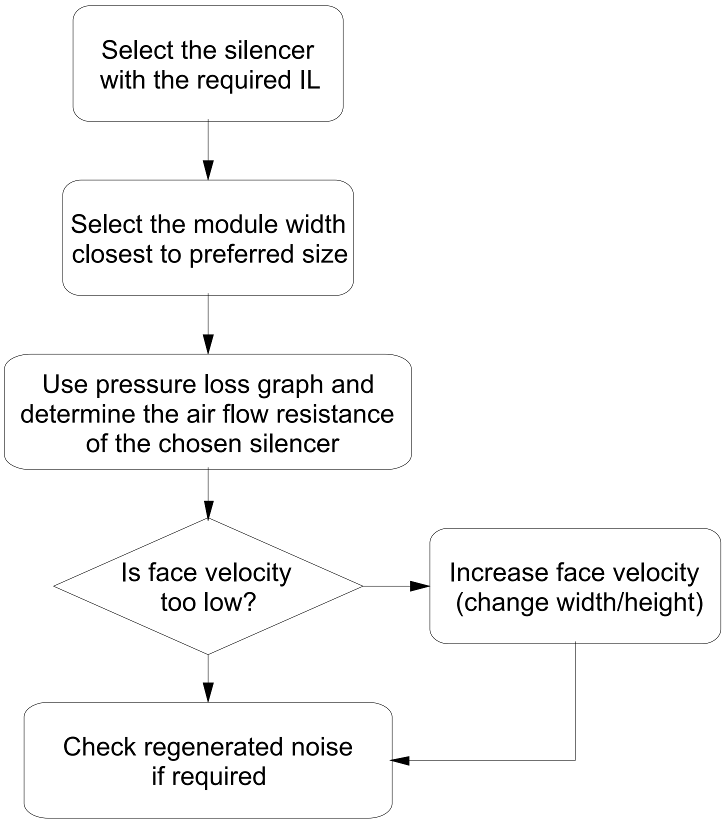


Figure 2.14 Attenuator Selection Procedure

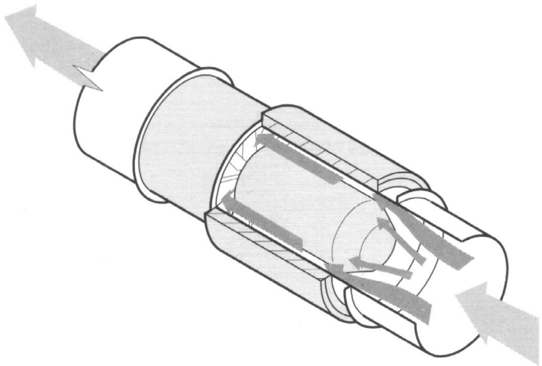
In the above flow diagram regenerated noise is mentioned. Manufacturers routinely provide this information which may be required if a comprehensive system noise analysis is being carried out. In general regenerated noise will not need to be considered if the following conditions are met (Jones, 1997):

- face velocity < 7 m/s
- velocity of air through pathways < 15 m/s
- air pressure drop < 75 Pa.

2.14.3 Connecting Attenuators to Ductwork

The same general rules for fans apply in connecting ductwork to attenuators. (See diagrams below.) The aim is to reduce turbulence and maintain laminar flow wherever possible. Sizing of ducts should be for the lowest velocities that a system allows. Lined ducts can be used where higher levels of attenuation are required.

Don't



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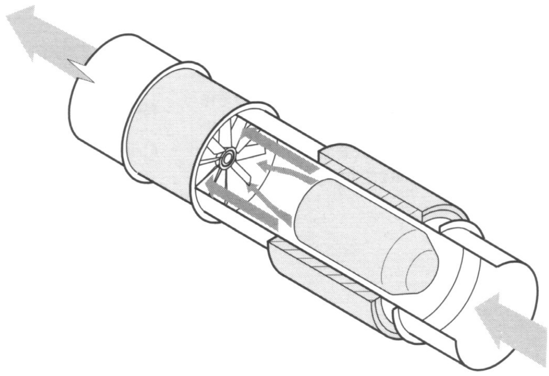
Pod attenuators sited close to a fan will block the airflow causing turbulence.

Do



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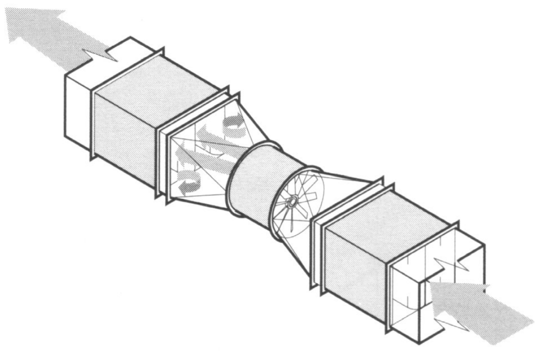
To ensure even flow allow a straight duct between the pod attenuator and the fan impeller should be inserted. For axial fans this should be one diameter (1D), and for in-line centrifugals fans this should be 2D.



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Siting a rectangular attenuator close to a fan intake or discharge causes air to be accelerated through the fan, becoming turbulent.

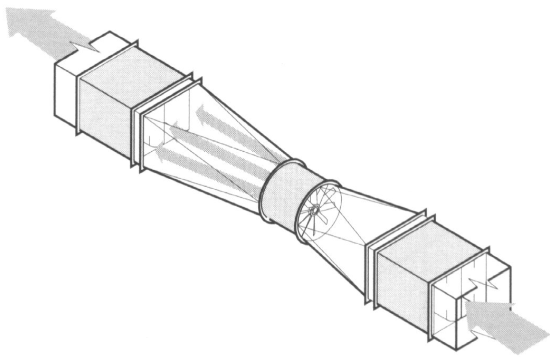

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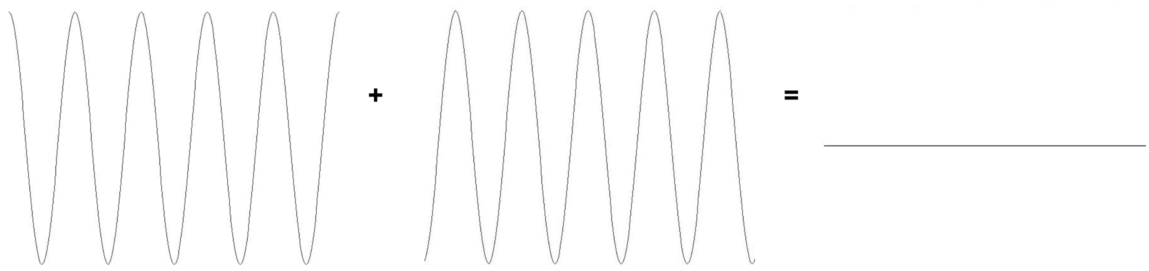
Figure 2.15 Connecting Ductwork

2.15 Active Attenuation

Active attenuation (or 'anti-noise') is the process whereby sound energy is put into a system e.g. duct, to eliminate the unwanted noise that is present. In simple terms a sound wave is generated that is in anti-phase to the noise. The sum of the two signals is thus zero. This process is illustrated in Figure 2.16.



(a) 'noise' signal *(b) 'anti-noise' signal* *(c) resultant cancelled noise signal*

Figure 2.16 Idealised Active Noise Cancellation

A schematic for the system is given in Figure 2.17.

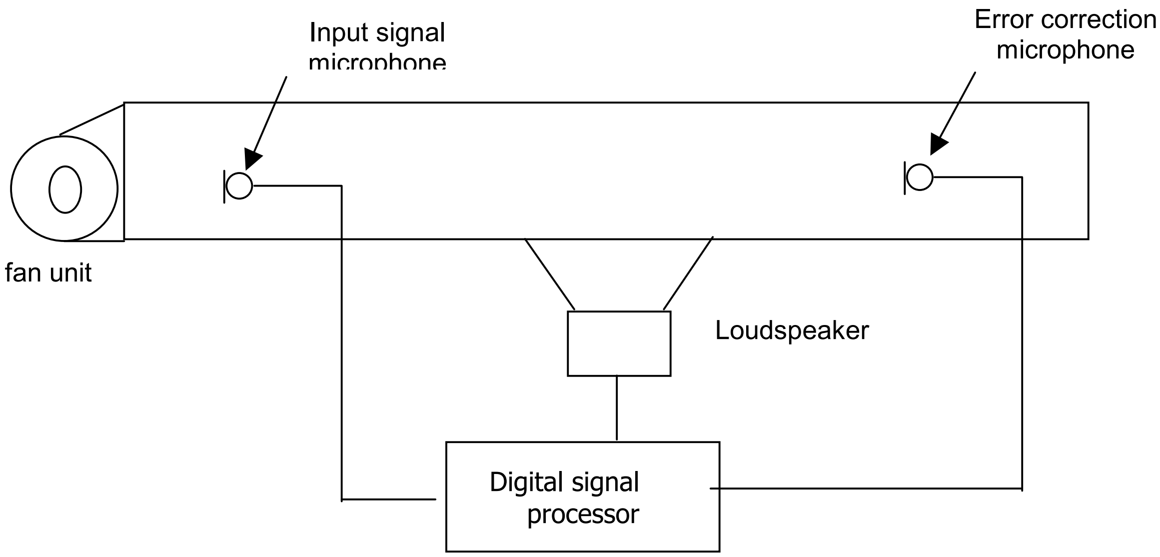


Figure 2.17 Active Noise System for Duct Noise Reduction

Although the system illustrated has only a single channel (i.e. one microphone pair and loudspeaker) complex systems exist which using multi-channel systems. Cancellation of the noise is generally at the lower frequencies, typically 10 dB - 20 dB below 500 Hz. Gelin (1997), in his tutorial on active noise control gives typical values for noise reduction (see Table 2.21).

Table 2.21 Typical Broadband Insertion Loss of Active Noise Control System for 'Small' Fan Under 2360 l/s (Gelin, 1997)

	Insertion Loss (dB)							
	Octave band Centre Frequency (Hz)							
	31	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
Active Noise Control	-	5 - 8	10 - 15	10 - 15	5 - 8	-	-	-
25 mm lined, 914 mm long duct	-	1	1	2	4	9	7	6
914 mm prefabricated silencer	-	3	7	11	22	19	14	12

Table 2.22 Typical Broadband Insertion Loss of Active Noise Control System for 'Small' Fan Over 2360 l/s (Gelin, 1997)

	Insertion Loss (dB)							
	Octave band Centre Frequency (Hz)							
	31	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
Active Noise Control	5 - 8	10 - 15	10 - 15	2 - 4	-	-	-	-
25 mm lined, 2134 mm long duct	n/a	2	2	4	10	20	17	15
2133 mm prefabricated silencer	n/a	8	17	26	43	43	26	19

Situations where active control works most effectively are those where tonal components are present, and where noises are typically described as 'rumble', 'roar' or 'throb'. They should only be used where air-flow velocities are less than 8 m/s and there is minimal turbulence. Because active noise control is most effective at low frequencies, it is best employed as part of an hybrid system using conventional passive silencers.

2.16 Calculations

Complete analysis of an air-conditioning system in octave bands is recommended where noise criteria are critical. Many manufacturers, such as GEC Woods, provide so-called 'downduct' computer programs which enable a whole system to be modelled from air inlet to room supply. These programs allow the user to select attenuators and alter system parameters to attain the required noise levels.

2.17 Summary of Good Practice

Within this chapter there have been a number of solutions identified that enable a system to be designed to operate more quietly. The main ones are summarised here:

- Select the quietest fans possible;
- Minimise system pressure to reduce fan sound power;
- Limit air velocities to within guidelines;
- Maintain good airflow by careful attention to transitions and connections;
- Avoid installing dampers at duct terminals;
- Install dampers at branches if balancing is necessary, but better to make systems self balancing to reduce regeneration noise problems;
- Use circular ducts where breakout needs to be limited;
- Route ductwork through non-noise sensitive areas.

3 Acoustics and Natural Ventilation

3.1 Air Supply

In this chapter, we concentrate on rooms that are naturally ventilated. These may be naturally exhausted or exhausted via mechanical means.

3.2 Air Transfer Between Rooms

Noise can enter a building through all parts of the envelope but windows and doors are usually the most acoustically weak areas in traditional brick/block buildings. Since doors usually open into non-critical areas (such as corridors, lobbies etc), windows which open into rooms warrant the most attention.

3.2.1 Windows

A major part of the insulation loss caused by windows and roof-lights is due to gaps or air-paths that permit direct leakage of sound. Even small gaps, such as cracks around openable windows, can reduce sound insulation significantly, whilst open windows which may be required for normal ventilation during summer or for the control of solar heat gain, virtually preclude anything more than minimal sound reduction. If double windows are to be installed for sound insulation, it is important to remember that when they are open to fulfil ventilation needs they do not function well acoustically. It is, therefore, essential to consider noise exclusion and ventilation together, because the two functions will often conflict. The primary function of the window is light transmission, but traditionally it has been adapted for ventilation as well. If noise control is a necessary part of the rooms, separate provision should be made for ventilation.

Solar heat gain depends on a number of factors including:

- aspect
- weight of structure
- size of windows
- openings for ventilation.

Heavy structure and small window size help to reduce both solar heating and noise transmission, but free natural ventilation to control solar heating seriously undermines noise protection.

Considerable sound insulation data, both field and laboratory measured, exists for windows (see for example Quirt, 1988). The tables below give an indication of the general performance values that can be expected. More detailed octave band data was given in Chapter 1. In general openable windows reduce the performance by 3 dB - 5 dB.

Table 3.1 Typical Window Sound Insulation (Tinsdeall, 1994)

Description	Sound Insulation, R_w dB
Window when open about 45º	10 - 15
4mm single glazed window	22 - 30
6 mm - 12 mm - 6 mm thermal insulating unit	33 - 35
4 mm - 200 mm - 4 mm secondary glazed window	40 - 45

The usual method of improving the sound insulation provided by a window is to install a second pane of glass separated from the primary pane by using either:

- a secondary pane with a spacing of more than 50 mm, or;
- thermal glazing with a pane spacing which is usually less than 25 mm.

The pane spacing produces different insulation characteristics and is the main factor in controlling insulation. However, it can be seen from the above table that where openable windows are being used, for ventilation purposes, that the maximum expected insulation is in the region of 10 - 15 dB.

Additional factors which affect the sound insulation (Tinsdeall, 1994) include:

- sealing;
- frame type;
- window pane size;
- reveal lining;
- ventilation openings.

Tinsdeall measured a wide range of windows, in a number of configurations to investigate these parameters. His results are summarised in Table 3.2.

Table 3.2 Single Figure Ratings (R_w) for Various Glazing Combinations

Frame and window details			Glazing details		
Material	Pane size	Seal type	Primary pane (mm)	Secondary pane (4 mm glass)	R_w
Wood	Large	Normal	4	None	30
Wood	Large	Normal	4 - 6 - 4	None	32
Wood	Large	Normal	6 - 12 - 6	None	34
Wood	Large	Normal	4	150 mm gap	42
Wood	Large	Normal	4	300 mm gap	47
Wood	Large	Normal	6 - 12 - 6	150 mm gap	46
Wood	Large	Normal	6 - 12 - 6	300 mm gap	52
Wood	Large	Normal	4	150 mm + liner	44
Wood	Large	Normal	4	300 mm + liner	51
Wood	Large	Good	6 - 12 - 6	None	34
PVC-U	Large	Normal	6 - 12 - 6	None	35
PVC-U	Large	Poor	6 - 12 - 6	150 mm gap	33
Wood	Small	Normal	4	None	29

3.2.1.1 Positioning of Windows

When using windows to assist in the ventilation system, the possible noise control measures include:

- reduce number;
- reduce size;
- locate windows on the quiet side of the building;
- awareness of flanking paths from room to room via windows.

This last point is illustrated in the figure below:

Don't

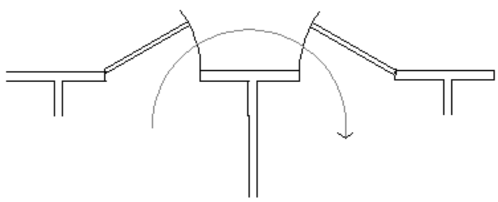


Figure 3.1 (a) Window Positioning
Noise travels through open windows or doors into adjacent apartment.

Do

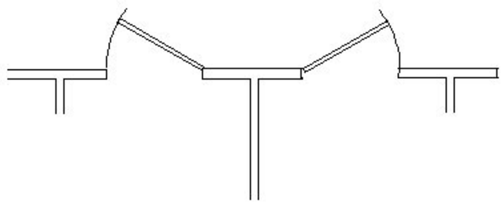


Figure 3.1 (b) Window Positioning
Arrange windows so that they deflect sound away from each other, or in the case of apartments, so that they all open in the same direction.

3.2.1.2 Alternative Window Designs

Where openable windows need to be used, one alternative is specially designed units which introduce a sound trap. The sound trap operates as an absorptive lined duct when the window is opened. Their effectiveness is good at higher frequencies, with low frequencies being more poorly attenuated. Two types are found in the literature (Figures 3.2 and 3.3):

The staggering of the openings of the windows, with the inner and outer panes opening on different sides can be beneficial to sound insulation whilst fulfilling airflow requirements. BRE (Utley, Sargent, 1989) reported that one study suggested it may be possible to obtain a ventilation rate of two to three air changes per hour under certain conditions while maintain a sound insulation of about 27 dB(A). Unfortunately, the ventilation obtained with such an arrangement relies on natural pressures, which in turn depend upon:

- wind conditions;
- inside/outside temperature differences;
- whether cross ventilation within the building is possible.

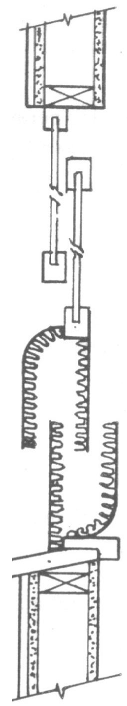


Figure 3.2 Acoustic Sash Window
(Source: Harris, 1997)

It is generally considered unlikely, that double windows with staggered openings can produce sufficient ventilation to maintain thermal comfort consistently, but it may be an acceptable option for rooms with low ventilation requirements.

The application of resonators in window frames to reduce noise levels have been considered for some time (Burgess, 1985) but are not widely used. More recently Field et al (1998) proposed a system of tuned resonators in conjunction with an activated window opening system, similar to that proposed by BRE in the 1980s. The window (or vent) is opened in response to approaching or receding transportation noise sources. Improvements of up to 7 dB(A) with just the resonators in place and up to 17 dB(A) improvement with the complete system in place were reported. At present only an experimental system the design has the potential for being used where high amplitude intermittent noise sources are present, for example at airports.

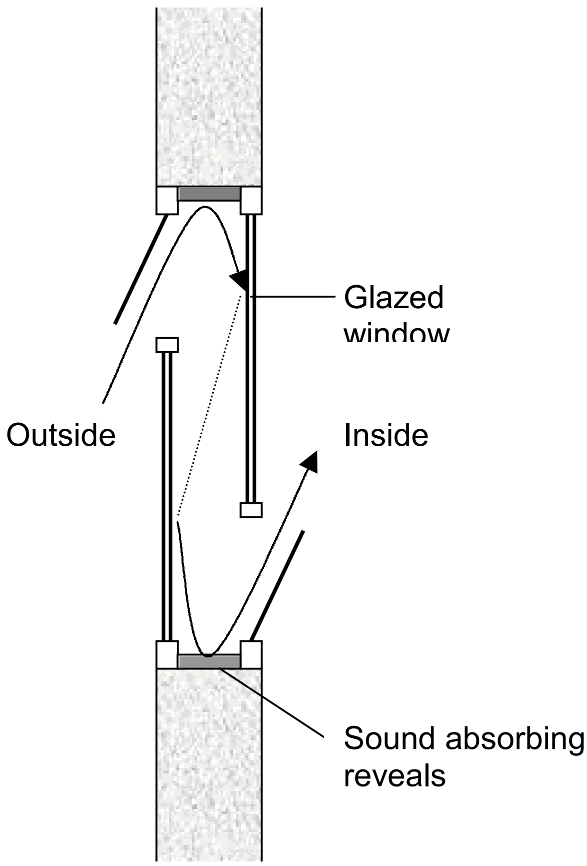


Figure 3.3 Detail of Double Glazed, Ventilating Window for Offices

3.2.1.2 Façade Calculations

In Chapter 1 the relative importance of the building elements was highlighted, and it was shown how small air-gaps could have significant effects on insulation values. A design example by Op 't Veld gives a useful comparison of differing window performance, and is reproduced here:

A room (volume 40 m^2, RT = 0.5 s) has a 10 m^2 brick façade with a 3 m^2 window installed. Supply of ventilation air is from a soundproofed, or non-sound-proofed, opening with a cross section of 150 cm^2. Five glazing alternatives were considered, and the relative façade performance compared. The results are given in Table 3.3.

Table 3.3 Maximum Attainable Façade Reduction for Different Ventilation Systems

Description	Noise reduction (dB(A))			
	A	B	C	D
Standard glass, (4 mm - 12 mm air gap - 6 mm glass) no weatherstripping	21	22	22	22
standard glass,(4 mm - 12 mm air gap - 6 mm glass) minor weather stripping	23	25	26	26
standard glass, (4 mm - 12 mm air gap - 6 mm glass) good single weather stripping	25	28	29	29
'high-quality' acoustic glazing (8 mm - 20 mm gas filled cavity - 10 mm laminated glass), double weather stripping	26	33	36	38
'luxury quality' glass, (42 mm laminated), very good weather stripping	26	34	40	44

Key:
- A ventilation opening without soundproofing
- B ventilation opening with adequate sound proofing (10 dB(A))
- C ventilation opening with excellent soundproofing (15 dB(A))
- D no ventilation opening

3.2.2 Passive Ventilators

3.2.2.1 Trickle Ventilators

Rather than opening a window, trickle ventilators are often used for background ventilation. This will have little adverse effect on the insulation provided by a single glazed window. However, the use of such ventilators will limit the sound insulation provided by a thermal double window. The reduction in the insulation occurs mainly above 315 Hz (depending on the size of the vent) with little or no effect at lower frequencies, so a properly designed system should be used where ventilation and high sound insulation are required.

The performance of this type of ventilator is consistent with its physical dimensions: i.e. large ventilators let more sound through than small ones. For a given ventilator size, those with holes through the window frame attenuate sound more than those with a slot - because the slot has a greater open area. It is this open area that determines the attenuation. When closed the ventilator attenuation increases by at least 5dB compared with the value achieved when open. A comparison of performance is given in Figure 3.4.

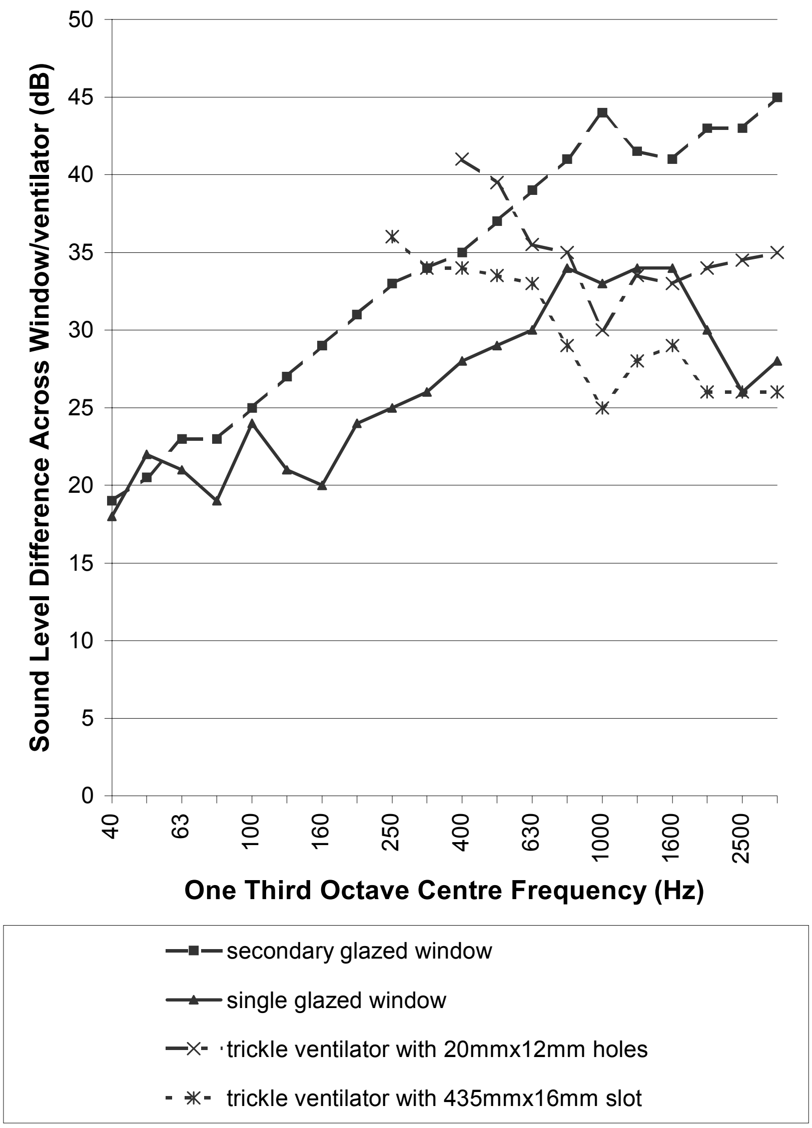


Figure 3.4 Effect of Trickle Ventilation on Acoustic Performance

For a ventilator in the open position, some acoustic impedance to noise is evident in the frequency range up to a resonance dip caused by standing waves within the openings. Above this frequency, the sound passes unimpeded through the ventilators. Consequently, in noisy external environments, the use of trickle ventilators could increase the indoor noise level above 630 Hz when used with single pane windows, and from 315 Hz when used with more effective secondary pane glazing.

Laboratory tests by BRE (Jorro, 1991) showed no significant difference in the air flow rates through openings of given area made by a slot or holes; rather the flow depends upon the pressure difference across the opening and this is determined by such factors as location, internal/external temperature difference and weather conditions.

The most recent work by BRE (White *et al*, 1999) compared performance of trickle vents from eight manufacturers. Table 3.4 identifies the vents, indicates whether manufacturers specifically described them as acoustic vents, and gives their performance.

The acoustic performance of vents is affected by adjacent reflecting surfaces such as walls and ceilings. To simulate the effect of these each ventilator was tested in three configurations, one in the centre of a plain wall, one to simulate the effect of an adjacent wall and one to simulate location near a corner. Each ventilator was tested in both the open and closed position with noise passing from ‘outside’ to ‘inside’ when installed.

Table 3.4 Description of Trickle Vents Tested by White *et al* (1999)

Vent Ref	Open (Free) Area (mm^2)	Acoustic Vent?	open	closed
			$D_{n,e,w}$	$D_{n,e,w}$
A	2800	No	32	36
B	24000	No	23	34
C	3000	No	34	44
D	3000	No	30	31
E	10000	No	27	41
F	3000	No	34	39
G	5500	No	29	40
H	2900	No	34	41
I	1600	Yes	35	N/a
J	2000	No	35	41
K	6600	Yes	47	49
L	11100	Yes	49	50
M	9500	Yes	39	46
N	6500	Yes	42	46

The results presented in Table 3.4 show the variation in performance between the open and closed tests for the individual trickle vents. The performance of the open vents is dictated mainly by the open area and by any acoustic lining to the aperture. It is noted that the results for Vent I indicate similar open performance to Vent J, a non-acoustic vent with (according to manufacturer data) a larger open area. However, the airflow results indicate that Vent J has a smaller equivalent area.

There were a wide variety of designs of trickle vent tested. These ranged from simple open apertures with thin plastic shutters (e.g. Vent B) to purpose built acoustic units (e.g. Vents K, L, M, and N) which exhibit a much better performance although this tends to be at the cost of size and complexity of construction.

In terms of single figure values the performance of the vents in the closed position ranges from 31 to 50 dB $D_{n,e,w}$. The performance with vents open is in the range from 23 to 49 dB $D_{n,e,w}$.

3.2.2.1 Passive Stack Ventilators

Passive stack ventilation (PSV) is often used as a means of providing simple extract ventilation in the "wet" rooms of dwellings. Their operation is by a combination of the stack effect (i.e. movement of air due to temperature difference in side and outside the house) and the wind moving over the roof. For them to be effective a source of air, to enable flow through the system must be available. This can be achieved by a vent or grille elsewhere in the house or by a fan to pressurise the system.

In whichever system type that is used attention must be given to noise break-in through the inlet vent and outlet vents. These vents in the building envelope are likely to have an adverse effect on the sound level in dwellings, especially those exposed to high levels of external noise.

Experiments by Jorro (1991) have shown that more sound attenuation was given by flexible, spiral wound tubing to the ridge of the roof than for rigid smooth plastic tube with no bends. The test house configuration used is given in Figure 3.5.

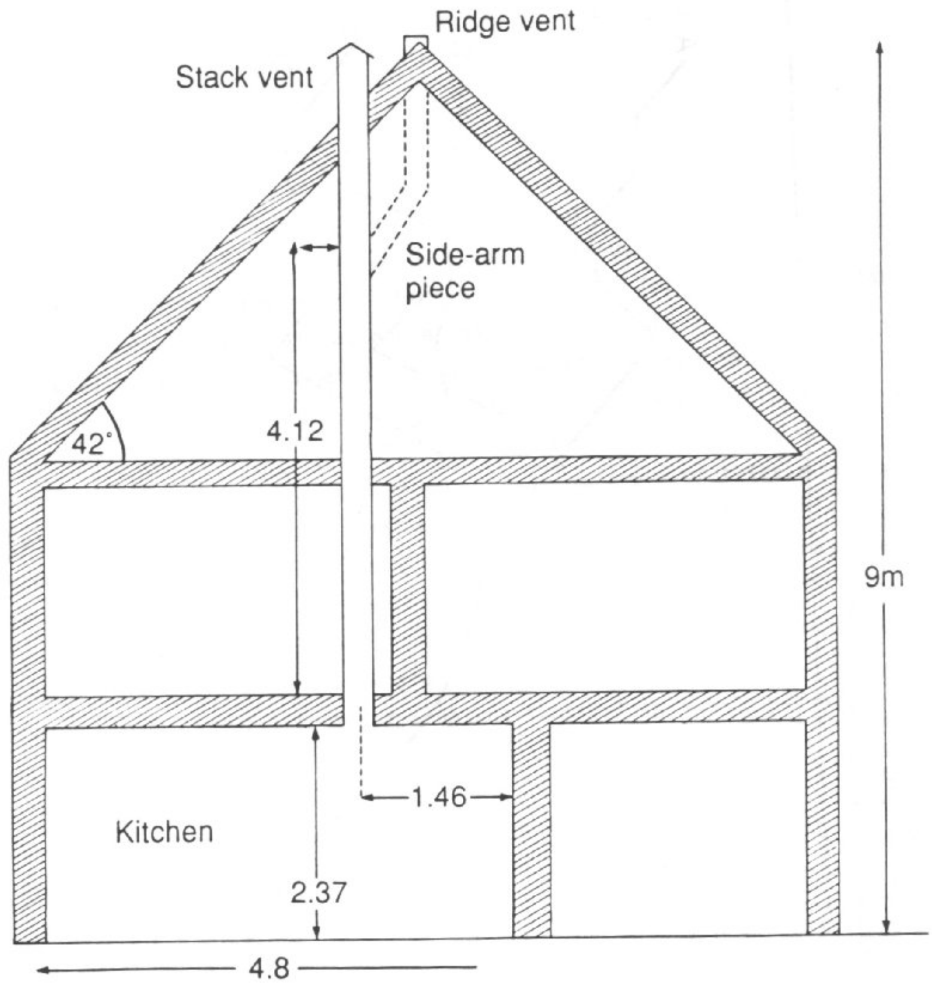


Figure 3.5 Experimental House with Passive Stacks (Jorro, 1991)

The dimensions of the stack pipe also appears to be significant, with 100mm diameter pipe giving lower performance that a 155mm diameter equivalent. The air flow rates in the larger pipe are of the order of twice that in the smaller pipe if straight and 50% higher if bends occur. Note that, in practice, the spiral tubing and the stack can have a small diameter. Whilst this will increase the acoustic attenuation, it will also increase the pressure drop and could cause air movement problems.

3.3 Doors

Doors are often used as a method of providing airflow between rooms. This is typically achieved by leaving gaps at the sill, or by means of louvres in the door itself. It should be clear from the previous discussion in Section 1, that any airgaps that are introduced will reduce the sound insulation. Rouseau carried out work, looking at specifying door options for acoustic performance. He reported the trend towards pressurising corridors in multiple occupancy buildings with the intention of:

- confining odours inside dwellings;
- supplying fresh air to the apartments;
- compensating for exhaust air (from bathroom fans, kitchen fans, and clothes driers).

From the acoustic perspective leaving a gap at the top of the door is preferable to one at the bottom. If the gap needs to be of large cross-section then a lined duct is necessary. Op 't Veld (1994) recommends that a 3mm gap allows for a flow of 10 m^3/h of air to be supplied or taken from the room. The composite sound insulation of the wall / door / ventilation gap *must* be calculated to ensure that an appropriate level of inter-tenancy sound insulation is achieved. In general it is better to supply air via a branch duct, with appropriate attenuation, than to install air gaps in entrance doors.

3.4 Louvres and Wall Ventilators

Whilst louvres are often used where mechanical ventilation is in place, they often function as a passive device. Louvres are essentially a series of horizontal blades and, for the purposes of noise attenuation, are assumed to have 0 dB attenuation. Where louvres are open to noise from external environments an acoustic louvre can be used. The selection procedure is similar to that for attenuators and is illustrated in the flow-chart in Figure 3.7.

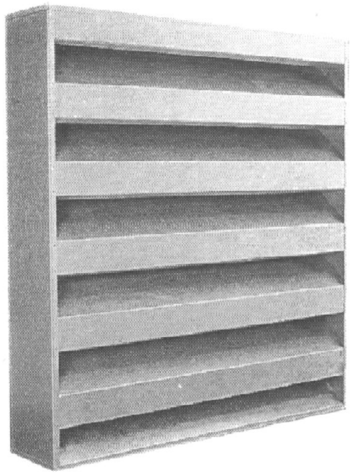


Figure 3.6 Louvre
(Reproduced with permission of Fantech Pty. Ltd.)

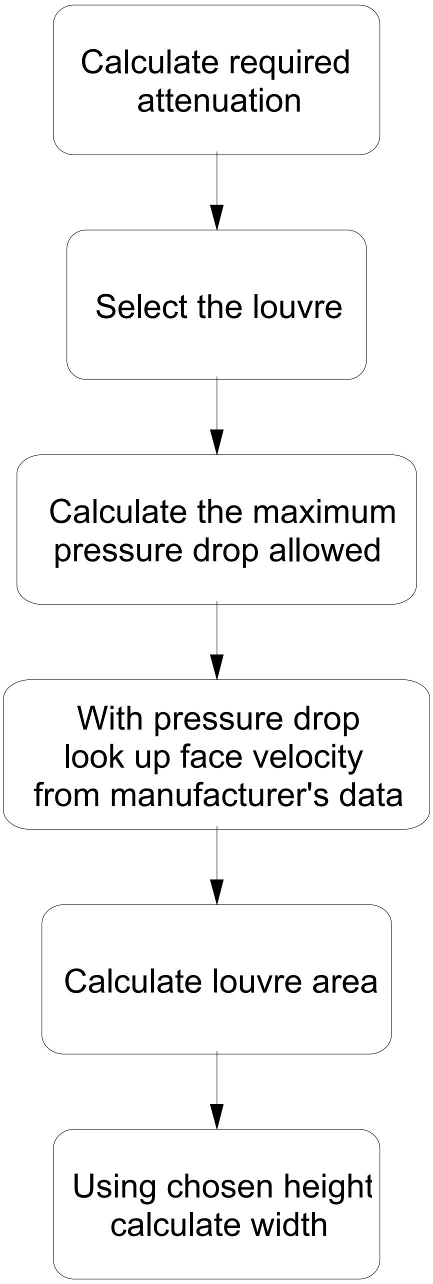


Figure 3.7 Selection Procedure for Acoustic Louvre

Example

An existing louvre is located near to a noise source that is creating an unacceptable noise level within a dwelling.

A calculation indicates that the required attenuation in dB at each octave band is:

63	125	250	500	1k	2k	4k	8k
5	6	8	12	15	13	12	8

Using the manufacturer's data a suitable attenuator is chosen. This must have attenuation at **all** frequencies higher or equal than required.

The manufacturer will supply a graph of pressure drop vs. face velocity. An example is given in Figure 3.8.

Knowing the face velocity the cross-sectional area (CSA) can be calculated from the expression:

$$CSA = \frac{airflow\ \ volume\,(m^3/s)}{face\ \ velocity\,(m/s^2)} \quad m^2$$

The manufacturer's data can then be used to calculate the louvre dimensions.

Generally you'll find that one of the dimensions (normally the height) has a range of fixed manufactured values. The width can then be specified as required.

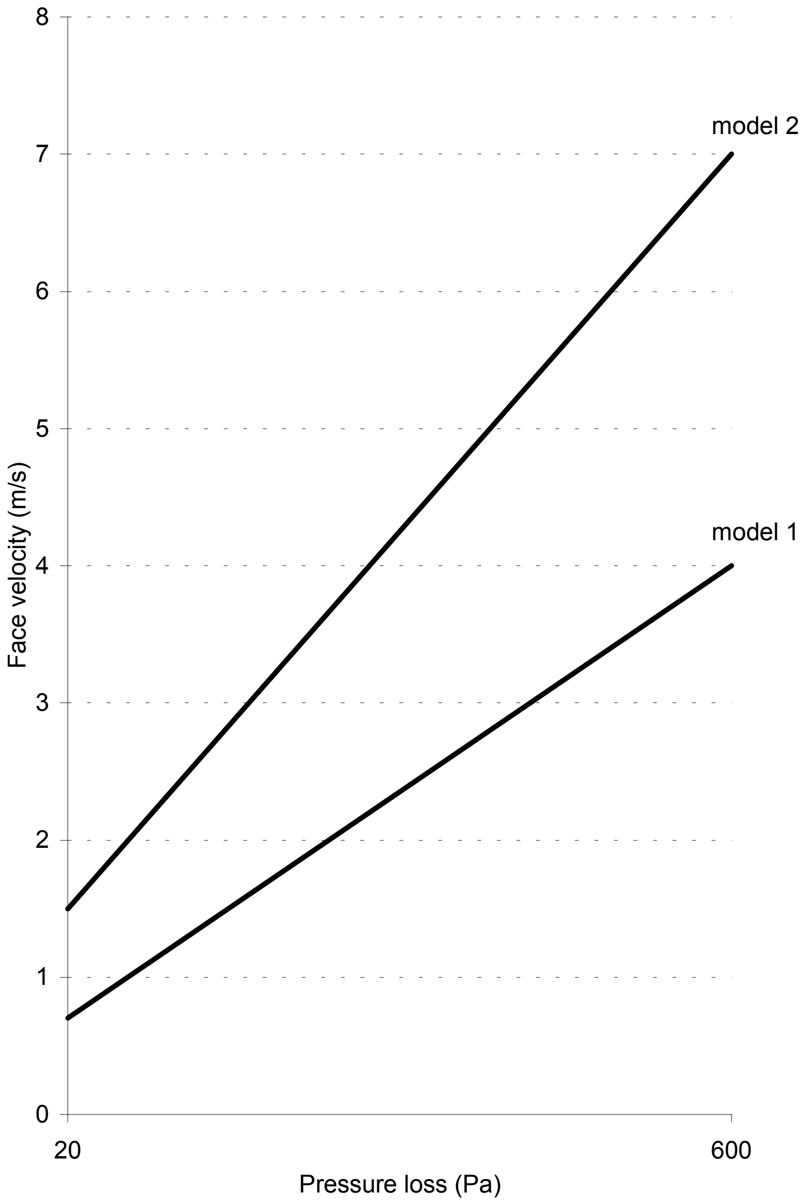


Figure 3.8 Typical Relationship Between Face Velocity and Pressure Drop for Acoustic Louvres

The work reported by White *et al* (1999) gives details of the performance of a series of domestic wall ventilators. Six through wall ventilators were tested.

Table 3.5 Description of Wall Vents Tested by White *et al* (1999)

Vent Ref	Open (Free) Area (mm^2)	Acoustic Vent?	Open	Closed
			$D_{n,e,w}$	$D_{n,e,w}$
O	4000	Yes*	49	51
P	4200	Yes*	50	52
Q	6000	No	31	35
R	6000	Yes	37	40
S	6000	Yes	43	49
T	14800	No	27	N/a
U	10000	No	31	N/a

4 Standards and Tools for Building Performance

4.1 Standards Focusing on Sound Transmission

4.1.1 Transmission Loss/Sound Reduction Index

These standards detail test method for measuring sound insulation between rooms, including facades and building elements (windows, doors etc.).

International Standards

- **EN ISO 140-1:1997** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 1: Requirements for laboratory test facilities with suppressed flanking transmission

- **EN ISO 140-2:1991** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 2: Determination, verification and application of precision data

- **EN ISO 140-3:1995** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 3: Laboratory measurements of airborne sound insulation of building elements

- **EN ISO 140-4:1998** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 4: Field measurements of airborne sound insulation between rooms

- **EN ISO 140-5:1998** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 5: Field measurements of airborne sound insulation of façade elements and façades

- **EN ISO 140-6:1998** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 6: Laboratory measurements of impact sound insulation of floors

- **EN ISO 140-7:1998** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 7: Field measurements of impact sound insulation of floors

- **EN ISO 140-8:1997** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 8: Laboratory measurements of the reduction of transmitted impact noise by floor coverings on a heavyweight standard floor

- **EN ISO 140-9:1985** Acoustics: Measurements of sound insulation in buildings and of building elements: Part 9: Laboratory measurement of room-to-room airborne sound insulation of a suspended ceiling with a plenum above it

- **EN ISO 140-10:1991** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 10: Laboratory measurement of airborne sound insulation of small building elements

- **EN ISO 140-12** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 12: Laboratory measurement of room-to-room airborne and impact sound insulation of an access floor

- **EN ISO/TR 140-13:1997** Acoustics: Measurement of sound insulation in buildings and of building elements: Part 13: Guidelines

Related Standards

- **ASTM E90** Test Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions

- **ASTM E336** Test Method for Measurement of Airborne Sound Insulation in Buildings

- **ASTM E966** Guide for Field Measurement of Airborne Sound Insulation of Building Facades and Façade Elements

  Enables measurement of outdoor to indoor transmission loss (OITL), and outdoor-indoor level reduction (OILR).

- **ASTM E1408** Test Method for Laboratory Measurement of Sound Transmission Loss of Doors and Door Systems

  This procedure is actually a supplement to E90, and details the correct installation of doors and door systems for measurement under that standard.

- **ASTM E1414** Test Method for Airborne Sound Attenuation Between Rooms Sharing a Common Ceiling Plenum

  This standard details the method to obtain the Ceiling Attenuation Class (CAC).

- **ASTM E1425-91** Standard Practice for Determining the Acoustical Performance of Exterior Windows and Doors

  This standard requires concurrent testing of air leakage and acoustic performance of the range of windows and doors.

4.1.2 Sound Transmission Rating

These standards give the calculation methodology for obtaining a single number rating (e.g. STC, R_w) of a building element. They require transmission loss data derived from one of the above standards.

International Standards

- **EN ISO 717-1:1996** Acoustics : Rating of sound insulation in buildings and of building elements : Part 1: Airborne sound insulation

- **EN ISO 717-2:1996** Acoustics : Rating of sound insulation in buildings and of building elements : Part 2: Impact sound insulation

Related Standards

- **ASTM E413-87** (Re-approved 1994) Rating of Sound Insulation (STC)

  Includes rating of STC, FSTC (Field STC), NIC (Noise Isolation Class), NNIC (Normalised NIC).

- **ASTM E597** Practice for Determining a Single-Number Rating of Airborne Sound Isolation for Use in Multi-unit Building Specifications

- **ASTM** Classification for Determination of Outdoor-Indoor Transmission Class

4.2 Material Testing

International Standards

- **EN ISO 354** Measurement of Sound Absorption in a Reverberation Room

- **EN ISO 9053:1991** Acoustics : Materials For Acoustical Applications : Determination Of Airflow Resistance

Related Standards

- **ASTM C423** Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberant Room Method

- **ASTM C522** Standard Test Method for Airflow Resistance of Acoustical Material

- **ASTM C1071-98** Standard Specification for Thermal and Acoustical Insulation (Glass Fiber, Duct Lining Material)

Covers fibrous glass insulation used as a thermal and acoustical liner for the interior surfaces of ducts, plenums, and other air handling equipment that handle air up to 121°C.

4.3 HVAC Acoustic Testing Methods

International Standards

- **EN ISO 5135:1997** Acoustics : Determination of sound power levels of noise from air-terminal devices, air-terminal units, dampers and valves by measurement in a reverberation room

- **EN ISO 5136:1990** Acoustics : Determination of sound power radiated into a duct by fans : In-duct method.
  (Also Technical Corrigendum 1:1993 to EN ISO 5136:1990.)

- **EN ISO 7235** Acoustics: Measurement Procedures for Ducted Silencers. Insertion Loss, flow noise and total pressure loss (see also ASTM E-477)

- **EN ISO 10302:1996** Acoustics : Method for the measurement of airborne noise emitted by small air-moving devices

- **EN ISO 11820:1996** Acoustics : Measurements on silencers in situ

- **EN ISO 13261-1:1998** Sound power rating of air-conditioning and air-source heat pump equipment : Part 1: Non-ducted outdoor equipment

- **EN ISO 13261-2:1998** Sound power rating of air-conditioning and air-source heat pump equipment : Part 2: Non-ducted indoor equipment

- **ISO/FDIS 14163** Acoustics : Guidelines for noise control by silencers

- **PD CR 1752:1999** Ventilation for buildings - Design criteria for the indoor environment (Draft Standard)

Related Standards

- **ASHRAE 36-72** Methods of Testing for Sound Rating Heating, Refrigerating, and Air-Conditioning Equipment

- **ASTM E477-96** Standard Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers

  This standard describes procedures for the measurement of acoustical insertion loss, airflow generated noise, and pressure drop as a function of airflow. The method also allows for a simulated semi-reflective plenum to fit around test specimens, to avoid breakout noise problems. (See also EN ISO 7235.)

- **ANSI/ASHRAE 68-1997** (AMCA Standard 330-97) Laboratory Method of Testing to Determine the Sound Power in a Duct

  This standard details the test method to determine the sound power radiated into an anechoically terminated duct on the supply and/or return side of air handling equipment. The standard applies to the following conditions:

  i) steady, broad-band, narrow band, and/or discrete frequency (50 Hz to 10,000 Hz);
  ii) air temperatures between -50°C and +70°C;
  iii) test duct diameter range from 150 mm to 2 m;
  iv) maximum flow velocity in the duct is 30 m/s;
  v) maximum swirl angle is 15°.

  This standard applies to sound sources connected to a duct. Examples of equipment covered by this standard are:

  i) fans (centrifugal, axial, mixed flow);
  ii) AHUs;
  iii) dampers;
  iv) throttling devices.

- **ANSI/ASHRAE 70-1991** Method of Testing for Rating the Performance of Air Outlets and Inlets

This standard details a standardised laboratory test method, under isothermal conditions, for rating air outlets and inlets used to terminate ducted systems for distribution and return of building ventilation air. This standard also includes specification of test instruments and facilities, test installations and test procedures, and methods of calculation for determining air flow capacity, aerodynamic performance and sound generation.

- **BS 4857:Part 2:1978** Methods for testing and rating terminal reheat units for air distribution systems. Acoustic testing and rating

- **BS 4773:Part 2:1989** Methods for testing and rating air terminal devices for air distribution systems. Acoustic testing

  This standard is for testing of air terminal devices, high/low velocity/pressure assemblies, dampers and valves used in air diffusion and distribution systems. in reverberation rooms.

- **BS 4954:Part 2:1978** Methods for testing and rating induction units for air distribution systems. Acoustic testing and rating

  Methods of acoustic testing and rating of induction units for sound power emission and terminal attenuation.

- **EN ISO 11691:1997** Acoustics. Measurement of insertion loss of ducted silencers without flow. Laboratory survey method

- **BS 848:Part 2:1985** Fans for general purposes. Methods of noise testing

  This is one of the most widespread standards adopted for the testing of fan performance. It details test methods for determining the acoustic performance of fans operating against difference of pressure. It includes four test methods:

  i) in-duct;
  ii) reverberant field;
  iii) free field;
  iv) semi-reverberant.

  Included are useful illustrations of suitable test ducting and anechoic terminations.

- **BS 4718, 1971**, Methods of test for Silencers for Air Distribution Systems

- **ARI 260P** Sound Rating of Ducted Air Moving and Conditioning Equipment

- **ARI 885-90** - Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets

  Standard published by Air-Conditioning and Refrigeration Institute (ARI).

- **AS 1277** Acoustics: Measurement Procedures for Ducted Silencers

- **ANSI S12.11-1987** (R1997) American National Standard Method for the Measurement of Noise Emitted by Small Air-Moving Devices

- **AMCA Standard 300-85**, Reverberant Room for Sound Testing of Fans

4.4 Acoustic Criteria and Requirements

One of the crucial design steps in good building acoustic design is the consideration of what is an appropriate criteria to design to in a room. This can be determined by the activity that is to take place in the room or space. The less sensitive the activity the higher the criteria that can be tolerated. There are a number of different criteria ratings that are in current use. The differences between them are slight, for example some have greater emphasis on prominence of low frequencies. The selection of the correct criterion level is also given below, as is an example of calculating a noise rating level.

4.4.1 A weighted Values ($L_{Aeq,T}$, dB)

The use of A weighting in noise measurements is widespread. Its main benefits are:

- it represents the response of the human ear to sound, and
- it is measured easily using low cost instrumentation.

Acceptable levels are published widely, in the same way that other noise criteria exist (see below). Many local authorities issue guidelines for acceptable levels in terms of dB(A), which are dependent on the planning zoning of an area. These levels should be activity dependent. Ranges for internal noise levels are normally within the range 30 - 35 dB(A) for domestic dwellings. The most recently values are from the work of Berglund (1995) carried out for the World Health Organisation. These are summarised in Table 4.1.

Table 4.1 Community Noise Guidelines
(Source: Berglund, 1995)

Location	Effects	$L_{A,eq,T}$ (dB)	Time (hours)	Time of day
Bedroom	sleep disturbance, annoyance	> 30	8	night
Living area	annoyance, speech interference	> 50	16	day
Outdoor living area	moderate annoyance serious annoyance	> 50 > 55	16	day
Outdoor living area	sleep disturbance, with open windows	> 45	8	night
School classroom	speech interference, communication disturbance	> 35	8	day
Hospitals: patient rooms	sleep disturbance, communication interference	> 30 - 35	8	day and night

The assumption of external nighttime noise levels is that there is a right to sleep with the windows open. Where low frequency noise levels are present, then lower levels than in table may be required (see comparison of criteria below).

The draft standard PD CR 1752 also gives guidance on the internal noise levels, specifically from ventilation systems. The levels are specified for three categories of indoor environment from high level of expectation to moderate level of expectation. Values are given in Table 4.2.

Table 4.2 Acceptable Noise Levels for Rooms with Ventilation Systems (PD CR 1752)

Building Type	**Space Type**	**Indoor environment expectation level category dB(A)**		
		high	***medium***	***moderate***
Child care institutions	Nursery schools	30	40	45
	Day nurseries	30	40	45
Places of assembly	Auditoriums	30	33	35
	Libraries	30	33	35
	Cinemas	30	33	35
	Court rooms	30	35	40
Commercial	Retail shops	35	40	50
	Department stores	40	45	50
	Supermarkets	40	45	50
	Computer rooms, large	40	50	60
	Computer rooms, small	40	45	50
Hospitals	Corridors	35	40	45
	Operating theatres	35	40	45
	Wards	25	30	35
Hotels	Lobbies	35	40	45
	Reception rooms	35	40	45
	Hotel rooms (daytime)	25	30	35
	Hotel rooms (nighttime)	30	35	40
Offices	Small offices	30	35	40
	Conference rooms	30	35	40
	Landscaped offices	35	40	45
	Office cubicles	35	40	45
Restaurants	Cafeterias	35	40	50
	Restaurants	35	45	50
	Kitchens	40	55	60
Schools	Classrooms	30	35	40
	Corridors	40	45	50
	Gymnasiums	35	40	45
	Teachers' rooms	30	35	40
Sport	Covered sports stadia	35	45	50
	Swimming baths	40	45	50
General	Toilets	40	45	50
	Locker rooms	40	45	50

The draft standard states that where the user has local control of the environment levels can be 5 dB(A) higher than the requirements in the table. Where tonal components occur (i.e. where a third octave band level is 5 dB(A) higher than the adjacent third octave band levels), then the levels should be 5 dB(A) lower than those given in the table.

4.4.2 Noise Rating, NR

Noise rating curves (see Figure 4.1) are octave band sound pressure level curves. They are used by plotting the spectrum on the graph. The NR value is the highest NR curve reached by the plotted spectrum. An *approximate* relationship exists between dB(A) and NR values:

$$NR = dB(A) - 6$$

This is valid for typical broadband machinery noise, but may not be applicable where prominent tonal components are present.

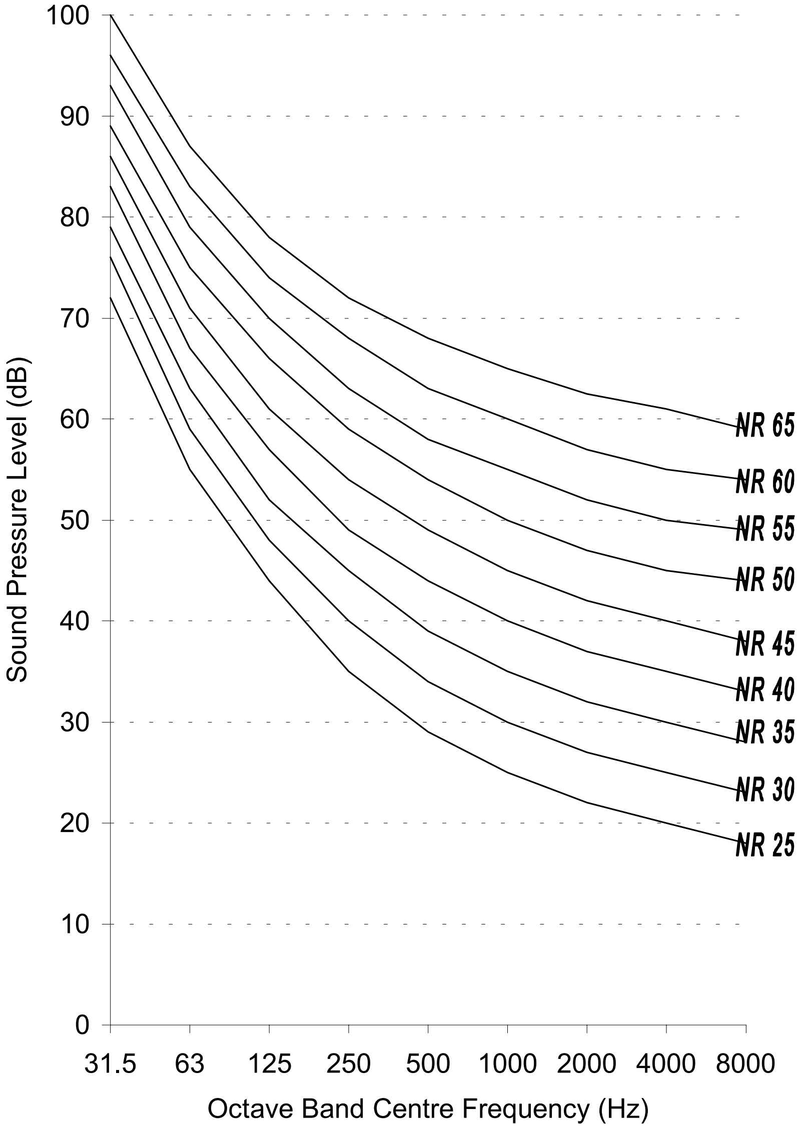


Figure 4.1 NR Curves

4.4.3 Noise Criterion, NC

Noise Criterion (NC) rating curves (see Figure 4.2) are octave band sound pressure level curves. They are used by plotting a spectrum on the graph. The NC value is the highest NC curve reached by the plotted spectrum. An *approximate* relationship exists between dB(A) and NC values:

$$NC = dB(A) + 5$$

As with NR curves this is only valid if 'well-behaved' sound sources are present!

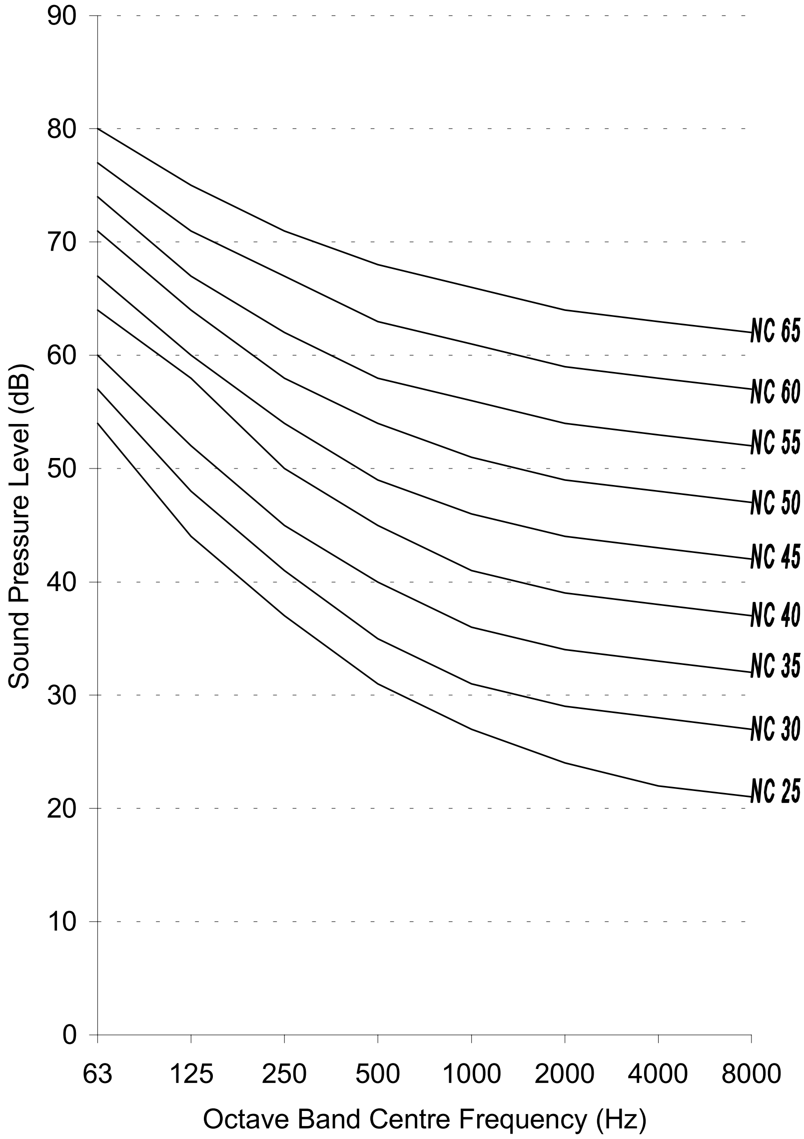


Figure 4.2 NC Curves

4.4.4 Perceived Noise Criterion, PNC

Perceived Noise Criterion (PNC) rating curves (see Figure 4.3) are octave band sound pressure level curves. They are used by plotting a spectrum on the graph. The PNC value is the highest PNC curve reached by the plotted spectrum.

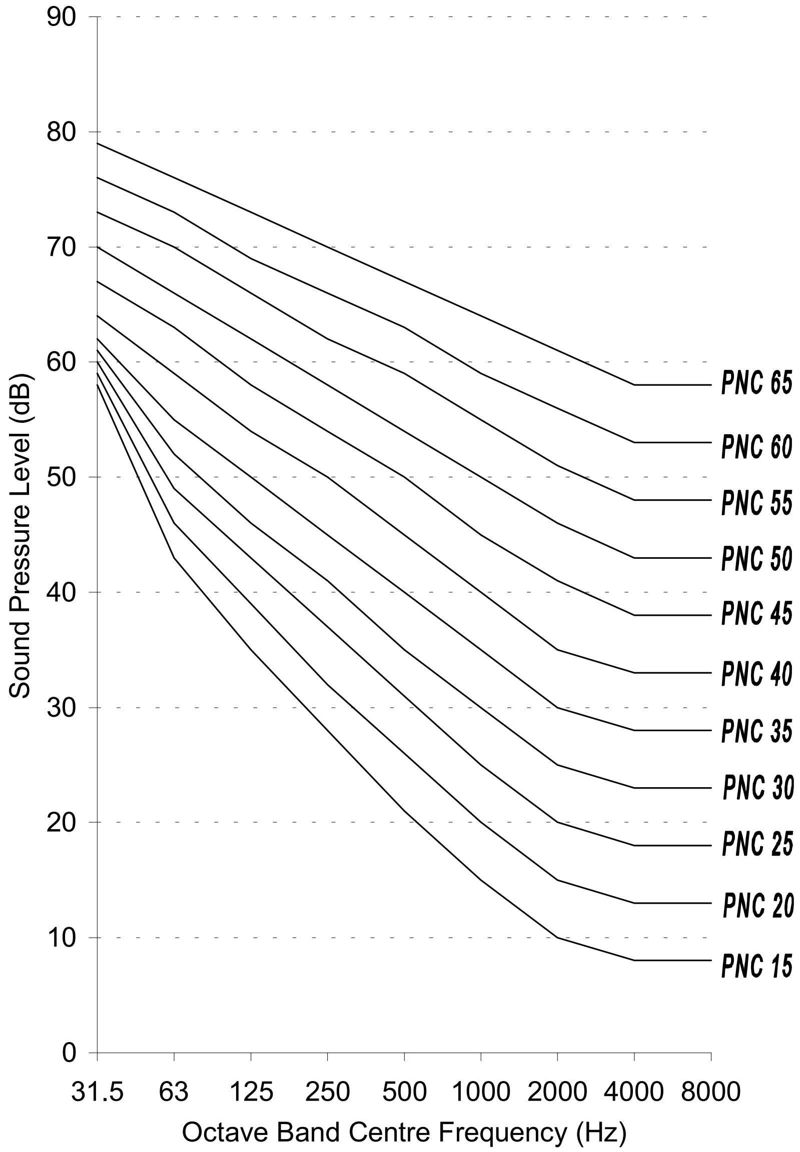


Figure 4.3 PNC Curves

4.4.5 Room Criterion (RC)

These are the ASHRAE preferred alternative to NC curves for system design. The reasons that are given for this is that it considers (i) subjective assessment of spectrum level and shape and (ii) includes the low octave bands of 16 Hz and 31.5 Hz. This second benefit thus enables potential vibration sources to be assessed. The RC curves are given in Figure 4.4.

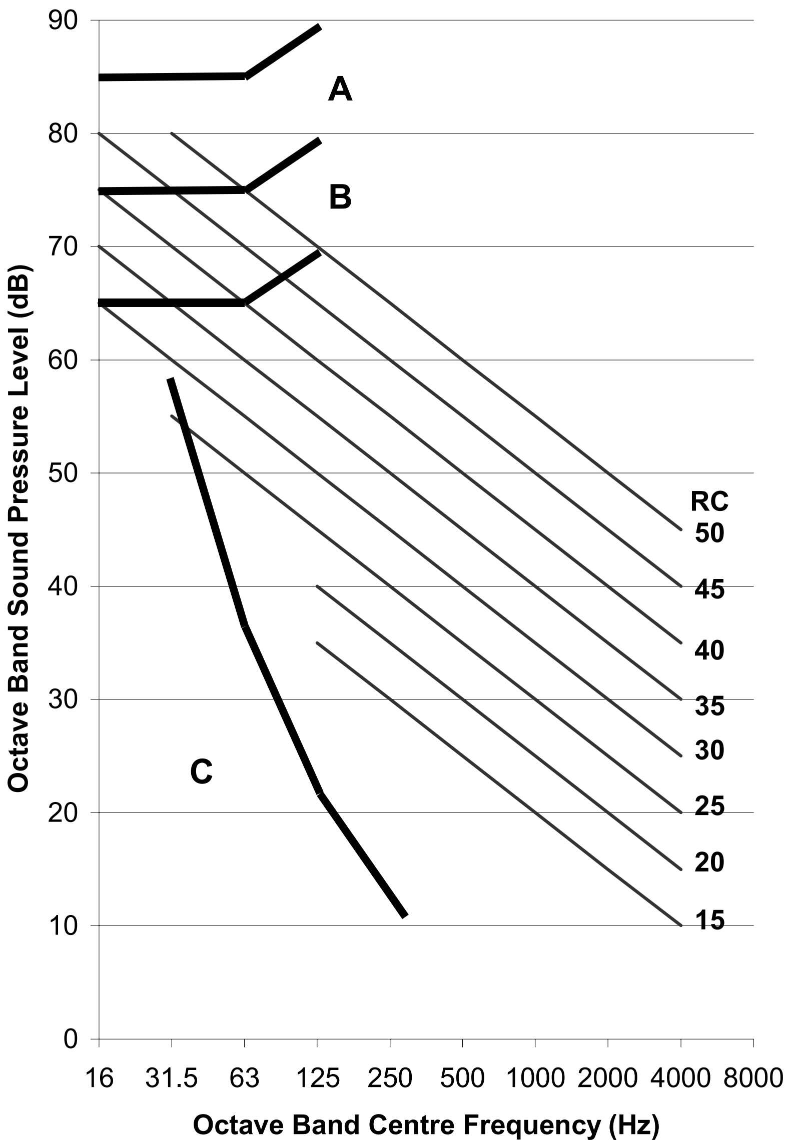


Figure 4.4 Room Criterion Curves

***Region A**:* high likelihood that noise induced vibration will occur in lightweight structures, audible rattles should be expected.
Region B: medium likelihood that noise induced vibration will occur in lightweight structures, audible rattles are possible.
Region C: below the threshold of hearing for continuous noise.

4.4.6 Comparison of the Different Criteria

Choosing the appropriate noise criteria is important when specifying to a client what level of noise is acceptable. Most organisations have elected to use a particular index based upon practical experience. However, it is still appropriate to briefly compare the indices.

In general practice NC curves are probably in most widespread use. NC curves are generally preferred over PNC curves, because the PNC criteria at lower frequencies are more stringent than that of the NC curves. However, because NC curves do not extend to the 16 and 31 Hz octave bands, problems can occur with HVAC systems, as acoustic energy is often present at these frequencies. In addition, NC curves are not balanced, thus sometimes resulting in hissy and rumbly spectra. Figure 4.5 below highlights some of the differences between 3 of the rating methods. It can be seen that for the range from 125 to 1000 Hz that the curves are mostly the same. This reinforces the opinion that it is where sources have spectra that have distinctive low or high frequency characteristics that differences in noise rating methods will be encountered.

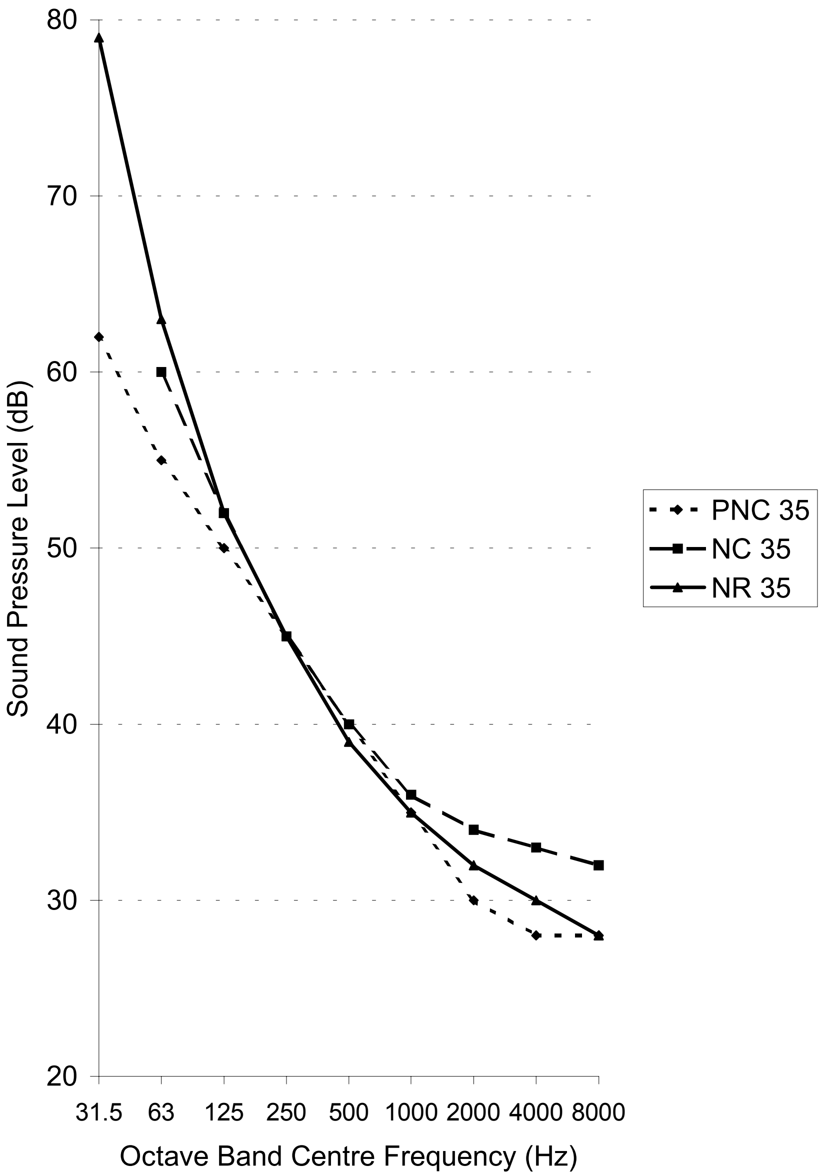


Figure 4.5 Comparison of Rating Curves

RC curves are gaining increased acceptance, partially due to their ability to provide more detail of noise levels and also because they are incorporated in ASHRAE Handbooks. Both NC and PNC have their roots in measurements made in occupied spaces, whilst RC curves were based upon non-occupied rooms, with all systems operating. NR curves were originally designed for outdoor use, although they are often applied to the inside of buildings.

Recent work by Tang and Wong (1998) compared NC, NR, PNC, RC and L_{Aeq} as rating methods in offices. They concluded that PNC correlated best with auditory sensation. Other authors (e.g. Leventhall and Wise, 1998) identify that low frequency noise is the main cause of noise complaints indoors, which may be why PNC curves perform better in the previous study. However RC curves can be seen as the most suitable when considering HVAC noise because they introduce an assessment of the presence of rumbly or hissy features.

A-weighted sound levels are also quoted for design targets. This practice should be avoided, as they do not sufficiently address low-frequency noise components. The result of this is that if internal noise levels are designed to (and indeed subsequently meet) a dB(A) value, the perceived noise comfort may not be satisfactory. However, the ease of measurement will often take precedence.

4.4.7 Setting the Criteria Level

Table 4.3 gives an *indication* of the criteria ratings that differing spaces should be designed to. Variations are seen between different publications and between different indexes, but those quoted are a reasonable guide. Remember that selection of the rating level should be done in the context of the design of required inter-room privacy.

4.4.8 Noise Criteria and Privacy in Rooms

Acoustical privacy between rooms is an important design factor in buildings. It occurs in rooms where unwanted noise (e.g. speech from an adjacent room) is masked by background noise. Rather helpfully, the background noise can be introduced by an HVAC system.

The level of acoustical privacy between rooms, or dwellings, is a function of:

- intended activity (in either room);
- background noise;
- transmission loss of partition.

This is illustrated in Figure 4.6.

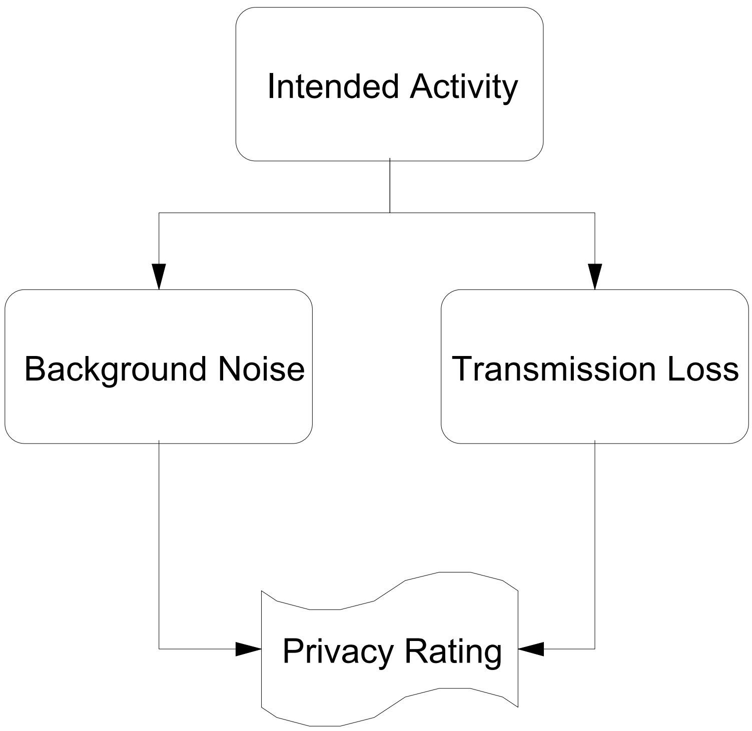


Figure 4.6 Privacy Rating Procedure

Table 4.3 Recommended Unoccupied Design Criteria for Different Areas
Note that the RC values are for neutral (N) spectrums
(Source: ASHRAE Handbook: HVAC Applications)

Occupancy	Criteria Range RC (N)
Residences, Apartments, Condominiums	25 - 35
Hotels/motels	
Individual rooms or suites	25 - 35
Meeting/banquet rooms	25 - 35
Halls, corridors, lobbies	35 - 45
Service/support areas	35 - 45
Office Buildings	
Executive and private offices	25 - 35
Conference rooms	25 - 35
Teleconference rooms	25 (max)
Open-plan areas	30 - 40
Corridors and lobbies	40 - 45
Hospitals and Clinics	
Private rooms	25 - 35
Wards	30 - 40
Operating rooms	25 - 35
Corridors and lobbies	30 - 40
Performing Arts Spaces	
Drama theatres	25 (max)
Concert and recital halls	Seek expert advice
Music teaching studios	25 (max)
Music practice rooms	35 (max)
Laboratories (with fume hoods)	
Testing/research, minimal speech communication	45 - 55
Research, extensive telephone use, speech communication	40 - 50
Group teaching	35 - 45
Church, Mosque, Synagogue	
General assembly	25 - 35
With critical music programs	Seek expert advice
Schools	
Classrooms up to 70 m^2	40 (max)
Classrooms over 70 m^2	35 (max)
Large lecture rooms, without speech amplification	35 (max) (RC 25(N) may be appropriate in some situations)
Libraries	30 - 40
Courtrooms	
Unamplified speech	25 - 35
Amplified speech	30 - 40
Indoor Stadiums, Gymnasiums	
Gymnasiums and natatoriums	40 - 50
Large seating-capacity spaces with speech amplification	45 - 55

The intended activity is easily identified. In fact it is normally the starting point of any design. The level of background noise affects the masking of activity noise, whilst the transmission loss determines the attenuation of noise between rooms. As an example, a 5 dB reduction in transmission loss between two rooms will require the background noise level to be increased by 5 dB to maintain the same level of privacy. In essence we are saying that the signal to noise ratio is what we are controlling.

This discussion leads us to see that, although the overall ratings identified in the previous table must be selected carefully, there is often scope for ventilation noise to be increased to increase the level of privacy. Thus a cost benefit can sometimes be achieved by decreasing partition performance (and thus reducing construction cost) and increasing the background noise level! This is possible because air-conditioning noise is not considered annoying.

Four speech privacy classifications are encountered in the literature (Harris 1998, Beranek 1988, Egan 1988):

- **High Satisfaction/Confidential Privacy:** speech very difficult to understand;
- **Normal Satisfaction/Normal Privacy:** speech audible but difficult to understand and therefore not a source of distraction;
- **Moderate Satisfaction/Marginal Privacy:** speech audible and largely intelligible if the listener cares to pay attention to it;
- **Poor Satisfaction /Poor Privacy:** speech is clearly audible and intelligible.

Attaching values to these categories is more difficult. There is a range of methods used to assess privacy.

One method is to obtain the speech intelligibility using the Articulation Index (AI). To measure the AI a series of words is spoken in one room. The percentage of words (or phrases) that are understood in an adjacent room is used to rate the AI. However, this method is not really practical at the design stage. Of more use is a system that relates the privacy to easily measurable or known quantities. One such privacy calculation method (Harriss 1998, Beranek, 1988) considers the effects of activity level, partition size and performance and room absorption. Pro-formas for this type of analysis are given in Egan (1988). These methods do work, but perhaps over complicate the issue for most purposes. It can be simplified to give a privacy index can by combining the single figure transmission rating (D_w, STC or R_w) and the value of the background noise rating (BNR) to give:

$$\text{Privacy Index, PI} = R_w + \text{BNR}$$

An enhancement of this is to use the D_w, which essentially already includes the effect of room absorption. Note that rating indices other than NR could also be used if preferred. Guidelines for required privacy are given in terms of the PI and the AI in Table 4.3 below.

Table 4.4 Values for Speech Privacy Calculation

Required Privacy	Background Noise Rating (BNR)			Articulation Index (AI)
	PNC	***NR***	***dB(A)***	
Confidential	70-75	>90	>85	<0.05
Normal	60-65	80-90	75-85	0.05 to 0.20
Marginal	50-55	75-80	65-75	0.20 to 0.30
Poor	<50	<70	<65	>0.30

The above discussion on privacy considered the importance of background noise upon privacy. In open plan office spaces ventilation noise can also be used to advantage. Here, the maximum acoustical separation, of even well-planned spaces, is likely to be of the order of 15 - 20 dB(A). Templeton (1998) gives a table of values relating activity and ventilation noise to % annoyed by normal speech. This is given in Table 4.5.

Table 4.5 Relationship of Background Noise and Annoyance in Offices

Background noise (including ventilation) (dB(A))	**% of people in adjacent work-space annoyed by normal speech**
35	65
40	40
45	25
47	16
55	4

National standards that are relevant to this discussion are:

- **ANSI S3.5** Methods for the Calculation of the Articulation Index (AI);
- **ANSI S3.14** Rating Noise With Respect to Speech Interference;
- **AS 2822** Acoustics: Method of Assessing and Predicting Speech Privacy and Speech Intelligibility.

4.5 Design Tool for Assessment of Building Performance

The difficulty that exists in making simple calculations of system performance is obvious from the discussion in Chapters 1 to 3 about the acoustics of ventilation systems. To try and overcome this Op 't Veld (1994) proposed a design tool to aid the non-acoustics specialist. The AIVC has indicated that it wishes to see wider use of such tools to improve internal acoustic comfort. More detail can be found in the IEA Annex 27 report.

4.5.1 Acoustic Parameters

The following acoustic parameters are used:

- incident (façade) noise level;
- façade noise reduction (20, 25, 30 and 35 dB(A));
- RT (0.5 s).

It is assumed that one façade has the dominant noise level incident upon it.

4.5.2 Room Parameters

The following room parameters are used:

- room volume (including depth);
- façade width (2 - 4 m, 4 - 6 m, 6 - 10 m);
- room height (2.5 m).

A depth factor between the room volume and surface area of the façade is assumed to be V/3S where V is the room volume (m^3) and S is the façade surface area (m^2). This is the difference between R (measured in the laboratory) and D_{nT} (measured in situ).

4.5.3 Construction Parameters

A range of materials and their performance are included:

Table 4.6 Acoustic Performance of Building Elements

Construction Type		**Overall façade sound reduction (dB(A))**
Solid façade	brickwork	46
Glazing (enclosed casement)	4 - 12 (air) - 6 mm	28
	6 - 12 (air) - 10 mm	33
	10 - 16 (air) - 10 mm (both laminated)	37
Ventilation opening	without sound proofing	-5
	with minor sound proofing	0
	with sound proofing	10
	with excellent sound proofing	15
Weather-stripping	single weather-stripping	35
	good single weather-stripping	40
	double weather-stripping	45

4.5.4 Design Tool Procedure

The procedure for using the matrices is illustrated in the flow diagram below.

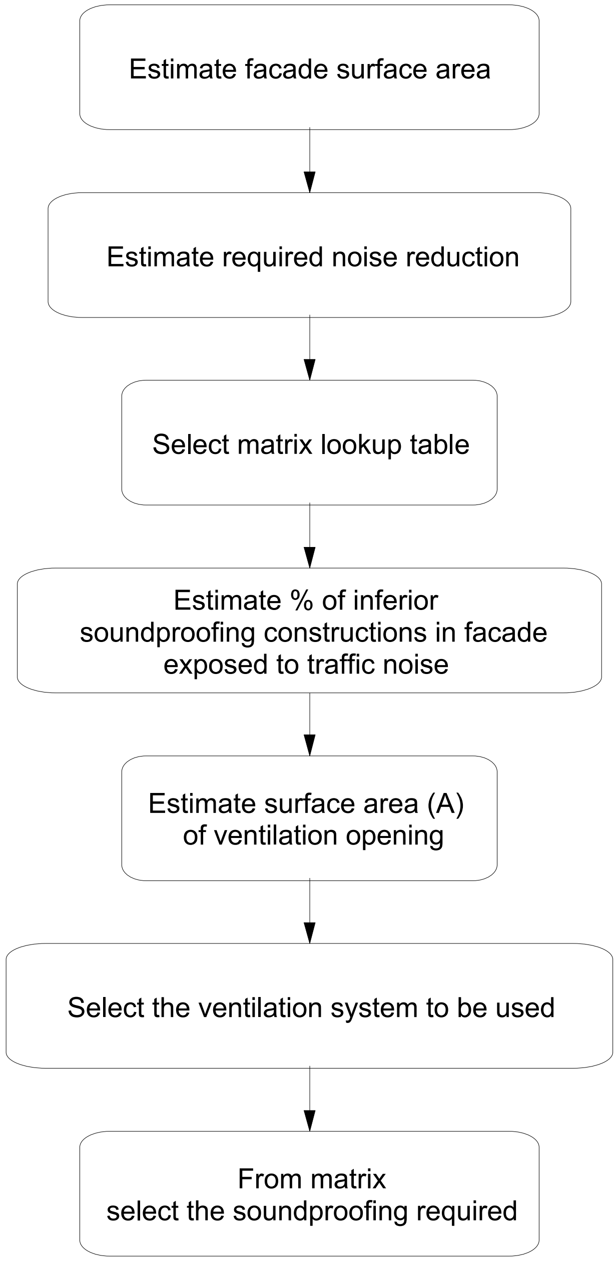


Figure 4.7 Design Procedure for Façade Insulation

4.5.5 Performance Matrix Tables

To achieve the goal of a simplified design tool, 3 sets of look-up table are provided for each range of façade areas (5 - 10 m^2, 10 - 15 m^2, 15 - 25 m^2). The ventilation systems are rated on a four point acceptability scale, which is defined as:

- - system is not applicable
- 0 system is applicable with excellent sound proofing constructions in façade
- ++ system is applicable with normal sound proofing constructions in the façade
- +++ system is applicable without extra sound proofing constructions in façade

The first four of the look-up tables (for façade area of 5 - 10 m^2) are reproduced below. (The reader is referred to the Annex 27 report for the remainder.) Note that $G_{A,c}$ is the characteristic noise reduction of the façade, as defined in Chapter 1. A is the area of the ventilation opening. The codes for the natural ventilation performance in the tables are:

a	ventilation opening without sound proofing	R_A = -5 dB(A)
b	soundproofed ventilation	R_A = 0 dB(A)
c	soundproofed ventilation	R_A = 10 dB(A)
d	soundproofed ventilation	R_A = 15 dB(A)

Table 4.7 Facade Insulation Look-Up Tables

$G_{a,c}$ = 20 dB(A)		**Percentage of inferior sound proofing construction in the façade exposed to traffic noise (%)**														
		0					< 50					≥ 50				
A (cm^2)		0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400
Natural supply	a	++	++	-	-	-	++	++	-	-	-	++	++	+	-	-
	b	++	++	++	++	-	++	++	+	+	+	++	++	++	+	0
	c	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
	d	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
Mechanical		++	++	++	++	++	++	++	++	++	++	++	++	++	++	++

$G_{a,c}$ = 25 dB(A)		**Percentage of inferior sound proofing construction in the façade exposed to traffic noise (%)**														
		0					< 50					≥ 50				
A (cm^2)		0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400
Natural supply	a	+	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	b	++	++	+	-	-	++	+	-	-	-	++	+	-	-	-
	c	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
	d	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
Mechanical		++	++	++	++	++	++	++	++	++	++	++	++	++	++	++

$G_{a,c}$ = 30 dB(A)		**Percentage of inferior sound proofing construction in the façade exposed to traffic noise (%)**														
		0					< 50					≥ 50				
A (cm^2)		0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400
Natural supply	a	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	b	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	c	++	++	++	+	-	+	+	+	0	-	+	-	-	-	-
	d	++	++	++	++	++	+	+	+	0	-	+	+	0	-	-
Mechanical		++	++	++	++	++	+	+	+	+	+	+	+	+	+	+

$G_{a,c}$ = 35 dB(A)		**Percentage of inferior sound proofing construction in the façade exposed to traffic noise (%)**														
		0					< 50					≥ 50				
A (cm^2)		0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400	0 - 50	50 - 100	100 - 200	200 - 400	≥ 400
Natural supply	a	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	b	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	c	++	-	-	-	-	0	-	-	-	-	-	-	-	-	-
	d	++	++	++	-	-	0	0	0	-	-	-	-	-	-	-
Mechanical		++	++	++	++	++	0	0	0	0	0	0	0	0	0	0

5 Conclusions

Acoustic comfort is only one aspect of the indoor air quality variables that needs to be considered. This Technical Note has attempted to overcome the difficulty of finding the right information, and then understanding how and where it should be applied. While this Note has attempted to be comprehensive, it is not exhaustive. However, it does form an essential part of the tool-box that engineers, researchers and designers need to tackle the challenge of reducing noise in building.

Good acoustics requires good design. Good design of ventilation systems requires knowledge of the impact of a building design upon its occupants and their activities (Figure 5.1).

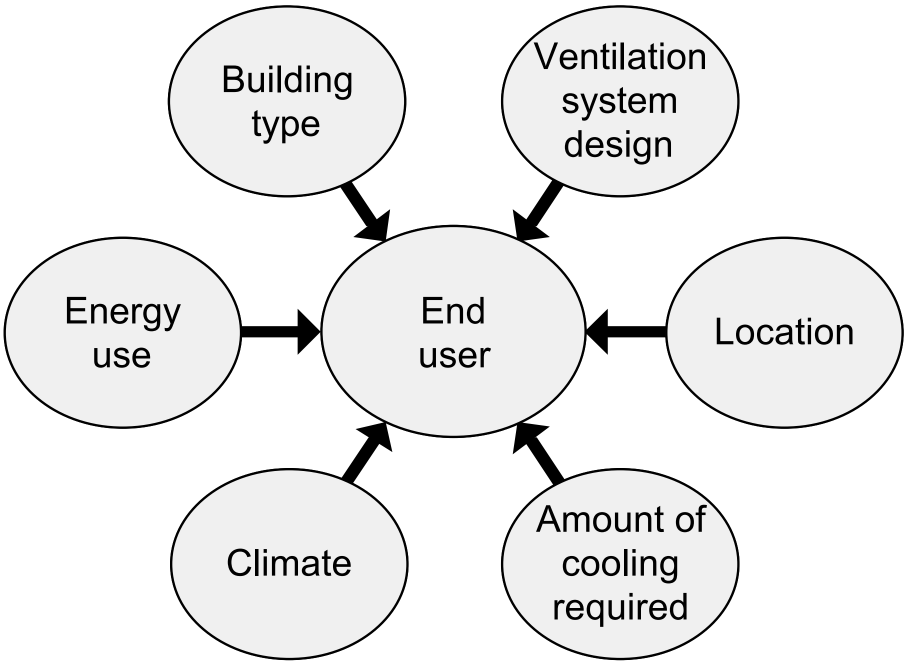


Figure 5. 1 Ventilation System Design

Chapter 1 described the foundations of acoustics. This introduced the terminology that is needed if any sense is to be made of the data in papers and manufacturer's publications. Understanding those principles is fundamental to application of the remaining information! The chapter established the model of source-receiver-path (Figure 5.2).

This simple model was introduced to focus attention on the economic principle that should underpin our equipment selection, i.e. buying quiet is the best way of containing noise break out. The model also reinforces the fact that once noise is generated it can travel to a source by a number of flanking paths, each of which need to be controlled.

Chapters 2 and 3 looked at the factors to consider when choice of ventilation systems (Figure 5.3) has been made.

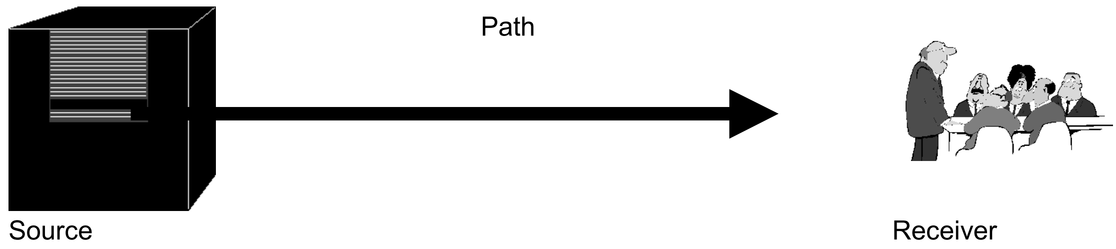


Figure 5.2 Model of Noise Transmission

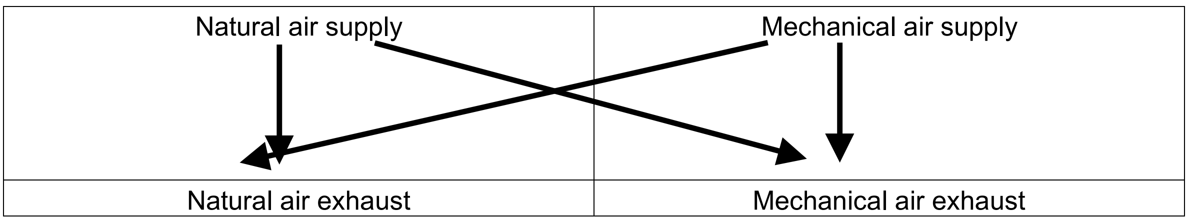


Figure 5.3 Ventilation System Types

Chapter 2 tackled the huge subject area of mechanically assisted ventilation. It discussed what it is that makes the noise, how to predict the noise levels, and some key mitigation methodologies stated. Reference was made to some of the many other sources (e.g. ASHRAE), which provide high quality application data and manuals. The importance of fan selection and connection to ductwork was explained, with examples given of good and bad installation practice.

Chapter 3 concentrated on the naturally ventilated systems. Noise from external sources can critically change system design here, due to the poor sound insulation offered by many window and door systems. Where high external noise levels are present, selection of the correctly rated window can dramatically improve internal comfort.

Chapter 4 introduced the broad range of standards that are used in ventilation acoustics. This then led onto the use of noise criteria that are appropriate for the occupants, building use and room activity. Whilst dB(A) are often used for specifying room noise levels, use of other noise criteria at design specification stage can reduce problems that might arise due to low frequency noises. Finally, the chapter discussed a design tool, which aims to simplify the selection of noise control measures in dwellings.

Chapter 6 is the final part of this Technical Note. It provides some of the supplemental information that will enable the reader to develop his understanding of acoustics and ventilation. It gives a glossary of acoustic terms and lists the references which, in conjunction with the Bibliography (Limb, 1997), provide a comprehensive source of further information. The chapter finishes with a collection of some of the useful internet web addresses that the author has encountered in writing this Note. Increasingly, manufacturers, suppliers and technical organisations are providing information on their web sites. The ability to work through tutorials (eg. The Vital Signs Project) download standards (ANSI) and review current literature (ASHRAE) ensures that this forum has numerous benefits to the practising ventilation system designer in the future.

6 Reference Section

6.1 Glossary of Terms

A-weighting

Frequency response curve which approximates to the sensitivity of the ear

Absorption

Conversion of sound energy to heat

Absorption coefficient

The proportion of sound energy absorbed, expressed as a number between 0 and 1

Airborne sound

Sound energy that is transmitted from a source by excitation of the surrounding air

Ambient noise

Totally encompassing sound in a given situation at a given time

Background noise

Usually described in terms of the $L_{A90,T}$ or $L_{A95,T}$ level: the level exceeded for 90% or 95% of the time

Blade Frequency Increment (BFI)

The correction (dB) made to the sound power level in the octave in which the BBPF occurs.

Blade Passing Frequency (BPF)

Given by N x RPM/60, where N is the number of blades on the fan. (Also used to calculate the blade frequency increment (BFI) when prediction of fan power is to be made.)

Critical frequency

The frequency at which the wavelength of sound in the air coincides with the wavelength of the associated vibration in a wall, floor or partition

Damping

Conversion of vibrational energy into heat which makes a body a less efficient sound radiator

Decibel (dB)

Unit of sound-pressure level

Diffraction

The deflection of a sound wave caused by an obstruction in a medium

Diffuse field

A space in which the sound-pressure level is constant for all positions

$\boldsymbol{D}_{\mathbf{n,e}}$

Normalised level difference of an individual element (< 1 m^2) (see ISO 140-10:1991)

$\boldsymbol{D}_{\mathbf{n,e,w}}$

The weighted single figure number of $D_{n,e}$

Equivalent continuous A-weighted sound-pressure level ($L_{Aeq,T}$)

The level of a notional steady sound which, over a defined period of time, *T.* would deliver the same A-weighted sound energy as the fluctuating sound

Filter set

A device used in sound-level meters to allow frequency analysis. For example in octave bands of a sound

Flanking transmission

Sound transmitted between rooms via elements, which are common to both rooms, other than the element that separates them

Free field

A space without sound reflecting surfaces

Frequency

The number of times in one second that a cyclic fluctuation repeats itself

Hertz (Hz)

Unit of frequency, cycles per second

Impact sound

Sound energy resulting from direct impacts on the building construction

Impact sound-pressure level *(L_1)*

The sound-pressure level in a room, resulting from impacts on the floor above generated using a standardised impact sound source

Insertion Loss *(IL)*

A measure of the noise reduction capability of an attenuator (or other element e.g. barrier). For attenuators this is measured by replacing a section of ductwork between two test rooms. One room contains a noise source and the other a sound level meter. The difference in the two levels is the insertion loss of the system.

Isolation

The introduction of a discontinuity between two elements in an energy transmission chain

L_{AN}

The A-weighted sound -pressure level that is exceeded for N% of the time

Level difference *(D)*

The arithmetic difference between the sound-pressure level in the source room and that in the receiving room (see also $D_{n,E}$)

Line source

A sound source which can be idealised as a line in space

Mass law

The empirical relationship between partition mass and sound insulation

Noise criteria (NC)

A set of octave band sound-pressure level curves used for specifying limiting values for building services noise

Noise rating (NR)

Similar to NC

Noise reduction coefficient (NRC)

The mean absorption coefficient (If a material averaged over four octave bands: 250, 500, 1000 and 2000 Hz

Noise and number index (NNI)

An index formerly used to describe aircraft noise around large airports

Octave

A frequency ratio of 2

Pascal

A unit of pressure corresponding to a force of1 Newton acting on an area of 1 m^2

Pink noise source

A source which delivers all frequencies with constant energy per octave or one-third octave band

Point source

A sound source which can be idealised as a point in space

$R_{A(traffic)}$

Value obtained using a procedure to estimate the difference between external and internal traffic noise levels in dB(A)

Rating level

The value associated with the industrial noise source under investigation. Used in British Standard BS 4142

Reference time interval

The specified interval over which an equivalent continuous A-weighted sound-pressure level is determined

Reflection

The phenomenon by which a wave is returned at a boundary between two media

Resonant Frequency

A frequency at which the response of a vibrating system to an input force reaches a maximum

Reverberation time (T)

The rate of decay of sound in a room

Root mean square (RMS)

The square root of the arithmetic average of a set of squared instantaneous values

Sabin

A measure of sound absorption. One Sabin equals 1 m^2 of perfectly-absorbing surface

Sound power

The total sound energy radiated by a source per second

Sound transmission class (STC)

A single-figure rating of sound reduction index. Similar to R_W. (USA)

Sound exposure level (L_{AE}, L_{AX} or SEL)

The level which if maintained constant for a period of one second would deliver the same A-weighted sound energy as a given noise event

Sound pressure

A dynamic variation in atmospheric pressure. The pressure at a point in space minus the static pressure at that point

Sound-pressure level

The fundamental measure of sound pressure defined as: $L_p = 10 \lg (p/p_0)^2$ dB. where p is the RMS value of sound pressure in pascals, and p_0 is 0.00002 pascals

Sound reduction index *(R)*

Ratio of the sound energy emitted by a material to the energy incident on the opposite side

Spectrum adaptation term

A value in decibels added to the single number rating to account for the characteristics of a particular sound spectrum. For example the adaptation term C_{tr} characterises the difference between the A-weighted levels for a road traffic noise spectrum

Standardised values

Sound-insulation values which have been adjusted to a receiving room reverberation time of 0.5 seconds

Standing waves

Where a wave interacts with a reflected wave, interference effects cause a pattern of pressure minima and maxima to be set up. This is called a standing wave

Structure-borne sound

Sound energy which is transmitted directly into the building construction from a source which is in contact with it

Wavelength

In a cyclic fluctuation, the distance between the ends of a single complete cycle

Weighted sound-insulation values

(for example *STC*, R_W, *NIC)*

Single figure sound insulation values obtained over the one-third octave band frequency range 100 Hz to 3150 Hz are turned into a single-figure weighted value using the procedure given in British Standard EN ISO 717

White noise source

A source which delivers all frequencies with constant energy per unit of frequency

6.2 References and Bibliography

ASHRAE, (1991), Handbook – HVAC Applications ISBN 0-9101110-79-4

Beranek LL (Ed), (1988), Noise and Vibration Control (Revised Edition), pub. Institute of Noise Control Engineering, ISBN 0-9622072-0-9

Blazier WE, (1993), Control of Low-Frequency Noise in HVAC Air-Handling Equipment and Systems, ASHRAE Transactions: Symposia, Vol. 99 Part 2, 1031-1036

Burgess MA, (1985), Resonator Effects in Window Frames, Journal of Sound and Vibration, Vol. 103, 322-332

Crocker MJ (Ed.), 1998, Handbook of Acoustics, John Wiley and Sons, ISBN 0-471-25293-X

Ebbing C., Blazier W. (Editors) (1998), Application of Manufacturers' Sound Data, ASHRAE ISBN 1-883413-62-1

Field CD, Mohajeri R., Fricke FR, (1998) Attenuation of Noise Entering Buildings using a Combined Intelligent Window and Passive Quarter-Wave Resonator System, Proc. Internoise 98, November, Christchurch, New Zealand

Gelin LJ, (1997), Active Noise Control: A Tutorial for HVAC Designers, ASHRAE Journal, August, 43-49

Harris DA, (1997), Noise Control Manual for Residential Buildings, McGraw-Hill, ISBN 0-07-026942-4

Harris CM (Ed.), (1998), Handbook of Acoustical Measurements and Noise Control, 3rd edition, Acoustical Society of America, ISBN 1-56396-774-X

Jones WP, (1997), Air Conditioning Applications and Design, Second Edition, Arnold/John Wiley and Sons, ISBN 0-340-64554-7/0-470-23595-0 (USA only)

Jorro S., (1991) The Sound Attenuation of Passive Ventilators, CIBSE Journal, 51, November

Leventhall HG, Wise SS, (1998) Making Noise Comfortable for People, ASHRAE Transactions: Symposia, SF-98-4-3, 896-900

Limb M, (1997), An Annotated Bibliography – Ventilation and Acoustics, AIVC, ISBN 0-946075-94-8

Op 't Veld PJM, (1994) Report No. 910767-1, Noise Aspects of Ventilation Systems in Dwellings, IEA Annex 27, Volume 5

Quirt JD, (1988), Sound Transmission Through Windows, National Research Council of Canada, CBD-240 (Also at http://www.cisti.nrc.ca/irc/cbd/cbd240e.html)

Quirt JD, (1991), Field Measurement of Sound from Residential Ventilation Fans, NRC Client Report for Canada Mortgage and Housing Corporation (CMHC), CR-5899.2

Rose KA, 1990, Guide to Acoustic Practice, 2nd Edition, BBC Engineering, ISBN 0-563-36079-8

Schaffer ME, (1993), A Practical Guide to Noise and Vibration Control for HVAC Systems, ASHRAE, ISBN 1-883413-04-3

Shaw CY, (1987), Mechanical Ventilation and Air Pressure in Houses, National Research Council of Canada, CBD-245, (Also at http://www.cisti.nrc.ca/irc/cbd/cbd245e.html)

Tang SK, Wong CT, (1998), Performance of Noise Indices in Office Environment Dominated by Noise from Human Speech, Applied Acoustics, Vol. 55, No. 4, 293-305

Templeton D (Ed), 1997, Acoustics in the Built Environment, Architectural Press, ISBN 07506-3644-0

Tinsdeall N J, (1994), The Sound Insulation Provided by Windows, BRE Information Paper, IP6/94

Utley WA, Sargent JW, (1989), The Insulation of Dwellings against External Noise, BRE Information Paper, IP12/89

White M., McCann G., Stephen R., Chandler M., (1999) Ventilators: Ventilation and Acoustic Effectiveness, IP 4/99, Pub. BRE/CRC, ISBN 1 86081 264 3

6.3 Internet (WWW) Addresses

Acoustic Organisations	Address
The European Acoustics Association (EAA)	http://eaa.essex.ac.uk/eaa/
The Acoustical Society of America (ASA)	http://asa.aip.org/index.html
Centre d'Information et de Documentation Sur le Bruit (CIDB), France	http://www.cidb.org/
Institute of Noise Control Engineering (INCE)	http://users.aol.com/inceusa/ince.html
International Institute of Noise Control Engineering	http://users.aol.com/iince1/
Institute of Sound and Vibration (ISVR), UK	http://www.isvr.soton.ac.uk/
Groupement de l'Ingenierie Acoustique (GIAC), France	http://www.cicf.fr/syndicats/giac.html
Societe Francaise d'Acoustique (SFA), France	http://www.loa.espci.fr/sfa/
University of Salford, School of Acoustics and Electronic Engineering, UK	http://www.salford.ac.uk/saee/
Institute of Acoustics (IOA), UK	http://www.ioa.org.uk/
The UK Quiet Pages	http://www.quiet.org.uk/

Other Organisations	Address
Air Infiltration and Ventilation Centre (AIVC)	www.aivc.org
Air Movement and Control Association (AMCA)	http://www.amca.org/
Association des Ingenieurs en Climatique Ventilation et Froid (AICVF), France	http://www.aicvf.org/
Belgian Building Research Institute (Acoustics laboratory at BBRI)	http://www.bbri.be/ http://www.bbri.be/antenne_norm/AntAco/
Building Research Establishment (BRE), UK Acoustics Centre	http://www.bre.co.uk/ http://www.bre.co.uk/acoustics/
Centre Technique des Industries Aerauliques et Thermiques (CETIAT), France	http://www.cetiat.fr/
CREST, USA	http://solstice.crest.org/

CSIRO Built Environment Sector, Australia	http://www.dbce.csiro.au/
Centre Scientifique et Technique du Batiment (CSTB), France	http://www.cstb.fr/
EREN - Infiltration, Ventilation, and Indoor Air Quality Program, USA	http://www.eren.doe.gov/buildings/iviaq/
ESDU	http://www.esdu.com/
Indoor Air Quality & Lead Poisoning Publications	http://www.iaqpubs.com/
NRC - Institute for Research in Construction, Canada	http://www.nrc.ca/
The Air Movement and Control Association International, Inc. (AMCA), USA	www.amca.org
The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), USA	www.ashrae.org
The Vital Signs Project – University of Berkeley, USA	http://www.ced.berkeley.edu/cedr/vs/index.html
World Health Organization (WHO)	http://www.who.ch/

Standards Organisation	**Address**
American National Standards Institute, USA	http://web.ansi.org/
Association Francaise de Normalisation (AFNOR), France	http://www.afnor.fr/
ASTM Technical Standards for Industry Worldwide	http://www.astm.org/
International Standards Organsiation (ISO)	www.iso.ch/welcome.html
NSSN: A National Resource for Global Standards	http://www.nssn.org/

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