26. SOUND AND SOUND ATTENUATION

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This chapter is a supplement to Chapter 11/Building acoustics in which the theoretical background, quantities and units used in acoustics are explained. This chapter can also be seen as a supplement to Chapter 19/ Ducting systems, which discusses the importance of designing and installing functional ducting systems, how they are integrated into buildings, the quality demands made and how airtightness is essential to function and operating costs.

This chapter takes a look at system components, primarily fans, that can create noise and how different attenuation measures can be taken, how ductwork and components create and dampen noise and how ducting can convey sound from one room to another, so-called crosstalk.

Noise is usually regarded as one of the factors threatening our living environments. In our mechanized society, we are surrounded by numerous sources of noise and, unfortunately, they seem to be ever increasing in number and strength – noise from traffic outdoors and from installations indoors. Silence is becoming an increasingly scarce commodity. When designing and building HVAC systems, to provide us with better thermal indoor climates and better air quality, it is important to realize that this must not be done at the cost of polluting the indoor environment with unwanted noise.

Many of us experience a feeling of relief when the ventilation system shuts down at the end of the working day and silence reigns once more. This type of dissatisfaction with noisy ventilation systems must be avoided and it is important that as much care is taken when planning the acoustics in a system as when tackling factors affecting our well-being and comfort.

Silence – the absence of noise – is now often a rarity and this can lead to stress and discomfort. It is also important to remember that prevention is better than cure – reengineering is a more difficult, more expensive, more time-consuming and more troublesome way of reaching acceptable solutions. In addition, it is more difficult to persuade those who have already been disturbed that they should now be happy with a new solution.

HVAC equipment, and especially fans, pumps and compressors, is the dominating source of noise in a building and it can even disturb people in the nearby surroundings. It is therefore important that all equipment is selected and located in such a manner that noise emitted disturbs neither building occupants nor neighbours.

Noise created by a component in an installation, for example, a fan, is transmitted through a building in different ways: via walls, floors and leakage points to adjacent rooms and via the supply and extract ductwork to the rooms connected to the ducting.

Vibrations from equipment can also cause structure-borne sound to be propagated through a building. These vibrations can cause walls, floors and other installations, such as piping, to vibrate and thereby create airborne noise. This, in turn, can cause disturbing noise in rooms a long way from the plant room.

INSTALLATIONS AS SOURCES OF NOISE

Building services installations for ventilation, heating, cooling and sanitary purposes have a common feature: the noises that they create come from flowing media – air, water or coolants. This applies to fans, air terminal devices, ducts, pumps, pipes, valves, compressors and flushing toilets. In every case, the amount of noise created is not only determined by the speed of the media and the pressure drop across the components but also by how well the components were originally designed from a noise point of view.

FANS AS SOURCES OF NOISE

Ventilation systems in buildings are often regarded as noisy and Figure 1 illustrates how noise can be spread.

a) Vibrations can cause structure-borne sound.
b) Airborne sound can be transmitted via the inlet to the fan and via the outlet ducts into the building.
c) Airborne sound can spread from the fan room to adjacent rooms.
d) Noise can be created in ductwork, dampers and terminal devices.

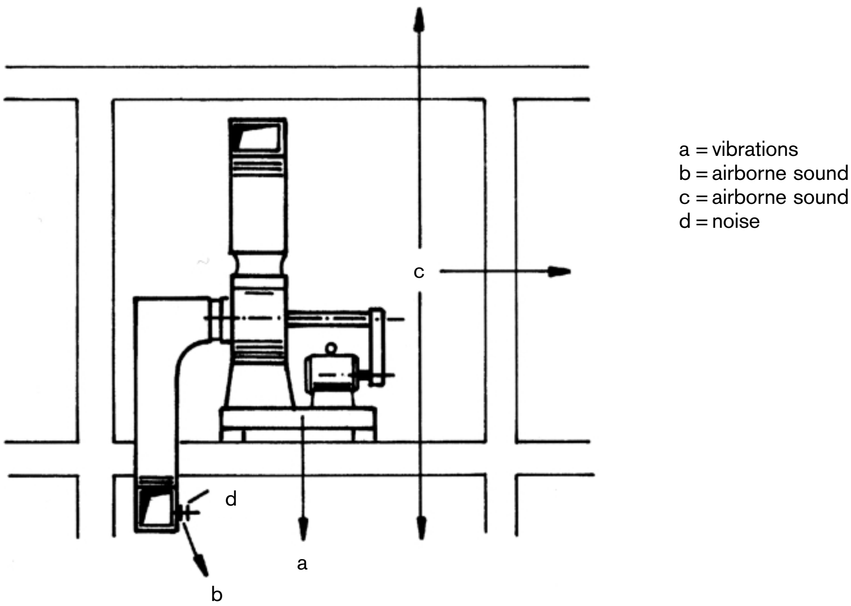

FIGURE 1. Fan noise can spread in different ways and in different directions.

Airborne sound, however, is not the only type of noise arising from a fan. The fan can also cause vibrations in the building structure, 'a' in Figure 1, if it is not statically and dynamically balanced, mounted on a correctly dimensioned and vibration-free foundation and connected to the ducting system via anti-vibration mounts. Power cables to the fan and any drainage pipe connections must also be flexible. This means that there must be a total avoidance of rigid vibration bridging between the equipment and the structure of the building. Otherwise, the other sound attenuation measures will be hardly meaningful.

If vibrations are transmitted to the structure of a building, these will create structure-borne sound that will be able to create new airborne sound in other rooms. Once structure-borne sound has occurred there is nothing that can be done, other than to remedy the source of vibration and inhibit any vibration bridging. It is not sufficient to count on the effects of other measures, as the system contains too many variables and is therefore completely indeterminate.

If the fan room has hard surfaces and, consequently, long reverberation times, see Chapter 11/ Building acoustics, the noise from the fan will result in the sound level in the room being too high and the airborne sound in the fan room, 'c' in Figure 1, will disturb surrounding rooms – above, to the side and below – as the walls and floors between them will start to vibrate and thus create new airborne sound in the adjacent rooms.

To prevent the fan noise from disturbing adjacent rooms, the walls, floors and doors must have high sound reduction indices, see Chapter 11/Building acoustics. And, as the airborne sound easily passes through small gaps and cracks, the penetration points for pipes, cables and ducts through the walls must be well sealed. The fan room doors must also be fitted with rubber sealing strips.

The most efficient and easiest way to avoid the problems described above is to locate the fan room as far as possible from sound-sensitive rooms and plan the building so that the fan room is adjacent to store rooms, corridors and other similar spaces where there are no permanent workplaces. If the fan room is located in the basement of a building, this is usually easy to arrange. It is considerably more difficult and demands much more care, if the fan room is located in a loft above the highest floor level. Top floor offices, for example, are usually the most attractive and this is where demands regarding low noise levels are the most stringent.

It is also important to have a sufficiently spacious fan room, so that the ductwork can be connected in the most suitable way possible from a flow point of view, i.e. without sudden bends or high flow speeds, and so that sound insulation material and silencers can be fitted correctly.

Back to Figure 1 – the noise from the fan will also propagate to the ducting system, 'b' in Figure 1, and this can cause high sound levels in ventilated rooms closest to the fan, as the noise will not have had a chance to attenuate in the ducting. At greater distances from the fan, the fan noise is reduced by the ducting system and this is where secondary sources of sound, from dampers and terminal devices, 'd' in Figure 1, will dominate.

Choice of fan

A low self-noise level is an important criterion when specifying and choosing equipment. Fans should be chosen so that they can operate at high levels of efficiency within their normal operating ranges. Fans that are made to run at unsuitable operating points, with subsequent poorer efficiencies, are often noisier than those that have been chosen correctly.

In CAV (constant flow) systems, fans should be chosen so that their maximum efficiencies are at the design air flows. In VAV (variable flow) systems, fans should be chosen so that they can operate with optimal efficiencies and stability in the most frequently used working ranges. A correctly chosen and installed fan reduces the need for noise attenuation in the ducting system. The following points should be kept in mind:

- Design the systems – the ducts, terminal devices and components – for low pressure drops.
- Compare sound data for different types of fans and from different manufacturers and choose the quietist.
- Choose variable speed control for air flows rather than damper control.

SOUND CREATED BY FANS

Fans generate two main types of sound:

- Rotation sound
- Turbulence and vortex sound

Rotation sound

Rotation sound in a fan occurs when the rotating field of flow passes fixed parts in the fan casing, for example, the narrowest passages in a centrifugal fan, the bars in an axial fan or the vanes in the inlet to the fan.

The speed profile at the periphery of the impeller will have a minimum at the edges of the blades and a maximum between them. The blade passing through the narrowest section in the fan casing, past the so-called tongue in a centrifugal fan, will give rise to pressure changes – and thereby sound – the sizes of which depend on how much the pressure drop is affected, i.e. the distance between the fan blade and the tongue. The frequency of the sound will depend on the speed of the fan and the number of blades on the impeller. The natural frequency of the fan, also known as the blade frequency, can be expressed as:

$$f_s = n \cdot s \qquad (1)$$

where:
f_s is the natural frequency or blade frequency in Hz
s is the number of impeller blades
n is the fan speed in rps

The blade frequency is the most characteristic and noticeable frequency but it is also often possible to discern the next two overtones. Higher overtones are generally drowned by other sounds from the fan.

The sound power radiating from the fan is therefore dependent on the air speed profile at the narrowest section of the fan casing. This means that a fan with a large number of blades will create a lower sound power level than a fan with fewer blades, on condition that the distance between the impeller blades and the casing is the same in both cases. This is because a large number of blades will even out the variations in air speeds.

For the same reason, the blade frequency will not be noticeable in a fan with a large number of blades, for example, so-called squirrel cage impellers or sirocco impellers. The natural tone and overtones, dependent on fan speed and number of blades, will drown in the sound from other sound sources.

In fans with double inlets, the rotation sound will be reduced if the impellers are evenly offset to each other. For example, two six-blade impellers would be offset by 30 degrees to each other. In this way the amplitude of the rotation sound will be displaced half a wavelength out of phase, which means that the total amount of radiated sound will greatly decrease.

Turbulence and vortex sound

Turbulence and vortex sound occurs in fans for a number of reasons, including:

1. Turbulence in the air in the fan.
2. Turbulent boundary layers next to the blade surfaces.
3. Shedding of vortices at the edges of the blades.

These types of sound dominate in fans as soon as the frequencies, at which the blade frequency and its first overtones dominate, are exceeded. Shedding of vortices is especially noticeable in centrifugal fans that are heavily throttled. The air leaving the blades causes loud vortex noises on release.

CALCULATING THE SOUND POWER OF A FAN

When sound power level data for fans is not available or when a given value has to be checked, it is often possible to calculate the sound power level to a reasonable degree of accuracy. A number of formulae are available for providing rough estimates and common to them all is that the sound power is given as a function of the flow speed to the power of five, i.e. v^5. The same speed dependency is also applicable to other components in a ventilation system with fixed damping and turbulent flow.

As pointed out above, the operating point of a fan has a great effect on the generation of sound. A fan that is chosen for its high efficiency also generates, in general, the least sound for a given flow and pressure rise.

When the air flow and, consequently, the air speed in the system are increased or decreased, the change in sound power level can be written as:

$$\Delta L = 10 \cdot \log \left(\frac{v_2}{v_1}\right)^5 = 50 \cdot \log \left(\frac{v_2}{v_1}\right) \qquad (2)$$

Example:
When doubling the flow through a fan or ventilation system the sound power level in dB will increase by:

$$\Delta L = 10 \cdot \log \left(\frac{2}{1}\right)^5 = 50 \cdot \log (2) = 50 \cdot 0.3 = 15 \text{ dB}$$

If the flow is halved, the sound power level will decrease in a similar way, by 15 dB.

Using SI units, the formula for calculating the sound power level will be as follows:

$$L_{wtot} = 40 + 10 \cdot \log q + 20 \cdot \log p_r \qquad (3)$$

where:
L_{wtot} is the total sound power level for the fan in dB(relative to 1 pW)
q is the air flow through the fan in m^3/s
p_r is the pressure rise across the fan in Pa

'40' in Equation (3) represents the so-called specific sound power level. This value assumes a normal fan efficiency (53% for axial fans and 63% for centrifugal fans) and that the constants connected to the units used are taken into account. If q and p_r are replaced by older units (m^3/h and mm water column, i.e. kp/m^2) the specific sound power level will have a value of 25 for similar fan efficiencies.

In general, the equation has an accuracy of about ±4 dB, on condition that the fan has been correctly chosen and operates within the range of maximum efficiency. The values obtained by the equation denote the sound power levels at the inlet to and outlet from the fan when the fan has been installed, i.e. when both the inlet and outlet are connected to the ducting.

The extent to which the sound power level in the fan room will be

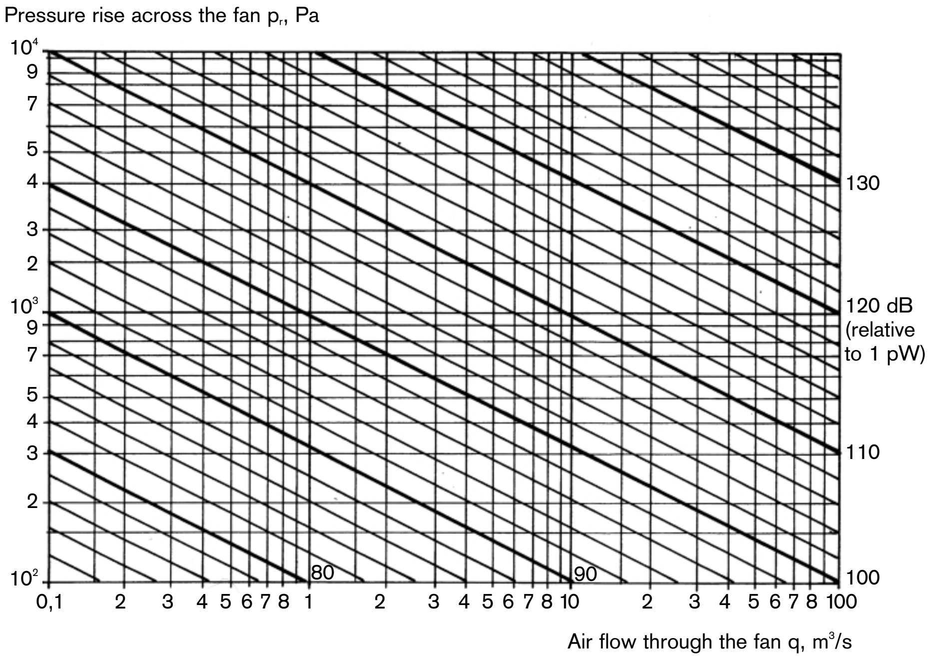

FIGURE 2. Graphical representation of Equation (3) when using SI units.

reduced depends on the type of fan, the thickness of the fan casing and the number of duct connections. The following values can normally be used as approximations:

- The total sound power level at the inlet or outlet after connecting the fan to the ducting: L_{wtot}
- The total sound power level in the fan room after connecting the fan to the ducting: L_{wtot} – approx. 10 dB
- The total sound power level in the fan room for an unconnected fan: L_{wtot} – approx. 6 dB

It is more difficult to predict the sound power level for fans installed in air conditioning units, as they will be affected by the acoustic surroundings in the unit in such a way that the level cannot be calculated but has to be measured.

OCTAVE BAND DISTRIBUTION OF THE SOUND POWER LEVEL OF A FAN

As the acoustic properties of sound sources and absorbents etc are strongly frequency-dependent, see Chapter 11/Building acoustics, sound calculations regarding the total sound power level for a fan are normally carried out for specific octave bands. If supplier's figures are not available, the following calculation method can be applied:

The total sound power level of a fan is spread over the different octave bands, with most of the sound within the octave band corresponding to the natural frequency. The natural frequency of the fan is calculated first, see above, and placed in the correct octave band.

Within the natural frequency octave band and the octave bands that have a higher frequency than the blade tone, i.e. higher up in the frequency scale, the sound power level will decrease by about 4 dB per octave band, and lower down, below the octave band of the fundamental tone, the octave levels will fall by about 3 dB per band.

Example:
Fan data:
Air flow $q = 4$ m^3/s; pressure rise $p_r = 1$ kPa; number of blades $s = 6$; speed $n = 2700$ rpm.

What is the sound power level in the 1000 Hz band?

Using the fan data in Equation (3)

$$
\begin{aligned}
L_{wtot} &= 40 + 10 \cdot \log 4 + 20 \cdot \log 1000 \\
&= 40 + 10 \cdot 0.6 + 20 \cdot 3 \\
&= 106 \text{ dB(relative to 1 pW)}
\end{aligned}
$$

Using the fan data in Equation (1)

$$f_s = \frac{2700}{60} \cdot 6 = 270 \text{ Hz}$$

This means that the blade tone belongs to the 250 Hz band, see Figure 3 in Chapter 11/Building acoustics.

Octave band, Hz	125	**250**	500	1000	2000	4000
Total sound power level L_{wtot}, dB	106					
ΔL/octave band, dB	–7	–4	–8	–12	–16	–20
Sound power level in octave band L_{woct}	99	102	98	**94**	90	86

Sound power level in the 1000 Hz band = 94 dB(relative to 1 pW).

SOUND CREATED IN VENTILATION DUCTING

Sound can be both created and dampened in the ducting system. Close to the fan, the sound from the fan will dominate but at a distance from the fan this sound will have been dampened in different ways and secondary sound sources in the ducting system will then dominate, with bends, junctions, dampers and terminal devices becoming sources of sound that will disturb room occupants more than remaining fan sound.

As the sound created by these components is strongly affected by the speed of the air in the ducting, it is important to keep speeds as low as possible and especially near occupied rooms. This can also have beneficial effects with respect to energy use. For the same reasons, ducting systems should be designed so that throttling dampers and other components that cause pressure losses, such as 90° bends and expansions and contractions to new cross-sectional areas, can be avoided.

The starting and finishing points for the air in a ducting system – the supply air and extract air terminal devices – must be chosen carefully.

The sound data supplied by the manufacturer must be checked to see whether it is applicable to the type of duct connection chosen. If there are a number of terminal devices in a room, the sound from these will be added logarithmically. As shown in Chapter 11/Building acoustics, the resultant sound level at a particular point in a room depends on the distance to the terminal devices, directivity factors and the equivalent sound absorption area of the room. Sound data for terminal devices and other components in rooms is normally given as the sound level in dB(A) in a room with an equivalent absorption area of 10 m^2 when they are placed in the reverberant field of the room.

To avoid disturbing flow-generated noise, one should:

- Design ducting and duct fittings for low air speeds.
- Avoid unnecessary turbulence by providing adequate distances between components (at least three duct diameters, but preferably more).
- Choose components that allow smooth flows through the ducting, bends, junctions and terminal devices.
- Avoid sudden cross-sectional changes or sudden changes of flow direction in the ducting system.

ATTENUATING SOUND IN THE DUCTING SYSTEM

Fan noise is reduced in a number of different ways when it is transmitted through the ducting system:

- As a result of sound power distribution.
- By attenuation in suction and pressure chambers.
- By leakage or so-called break out through duct walls.
- By attenuation in internally insulated ducts and bends.
- By using silencers in the ducting system.

Sound power distribution

The sound from a fan, like the air from a fan, is normally distributed between the branch ducts leading to the rooms served by the ventilation system. The proportion of the total sound power entering a given branch duct can be calculated from the ratio of the partial air flow to the total air flow, by using the following equation:

$$\Delta L = 10 \cdot \log \frac{q_{partial}}{q_{total}} \qquad (4)$$

where:

ΔL is the reduction of sound due to the sound power distribution in dB(relative to 1 pW)

$q_{partial}$ is the air flow in the branch duct in m^3/s

q_{total} is the total air flow from the fan in m^3/s

Example:

The air flow through the fan is 10 m^3/s and through the branch duct 100 l/s.

The attenuated sound power entering the branch duct is given by:

$$\Delta L = 10 \cdot \log \frac{q_{partial}}{q_{total}} = 10 \cdot \log \frac{0.1}{10} = 10 \cdot \log 10^{-2}$$
$$= 10 \cdot (-2) = -20 \text{ dB(relative 1 pW)}$$

If the flow conditions are expressed in percent, $[(q_{partial}/q_{total}) \cdot 100]$, the following values will be obtained:

Flow ratios, %	50	33	25	20	10	5	2	1	0.5
Attenuation, ΔL, dB	3	5	6	7	10	13	17	20	23

As in all sound calculations, sound level values have to be treated logarithmically.

Attenuation in suction and pressure chambers

A ducting system for supply air can be designed with a pressure chamber immediately downstream of the fan. Air is then distributed from this chamber via circular ducts to the different ventilated rooms. This is an excellent solution, as it prevents the transmission of disturbing fan noise through the ducting.

The sound, when it passes through the absorbent lined chamber, will be attenuated in proportion to the difficulty it has in finding its way out of the chamber. The smaller the outlet opening in relation to the total lined area, the more the sound will be forced to reverberate between the lined surfaces and the more it will be reduced. It is, of course, important that the inlet and outlet openings are not located opposite each other in the chamber, as there is a risk of the sound radiating straight across the chamber. If an outlet opening has to be located in this way, the chamber should be fitted with an internal baffle, lined on both sides with absorbent material and placed between the openings, so that the sound is forced around it.

The approximate attenuation in the chamber can be calculated as follows:

$$\Delta L = 10 \cdot \log \frac{S_0 \cdot \alpha}{S_1} \text{ dB} \qquad (5)$$

where:

S_0 is the lined surface area of the chamber including the openings in m^2

α is the sound absorption factor of the lining, see Chapter 11/Building acoustics

S_1 is the size of the outlet opening for the branch duct in question in m^2

The equation can also be solved graphically, see Figure 3. The diagram presented here is for a chamber lined with 10 cm mineral wool.

Unlined ducts

Unlined sheet steel ducts can attenuate low frequency sound, as the sound energy causes the thin walls of the duct to vibrate and thus function as a membrane absorbent, see Chapter 11/Building acoustics. The duct has a relatively high attenuating effect around the natural frequency of the sheet steel but is selective and only provides low attenuation above or below this frequency. This attenuation cannot be calculated as it is determined by too many variables, such as the thickness of the sheet steel and how it is stiffened, the ratio of the free area to the circumference of the duct, and how the ductwork is attached to the structure of the building.

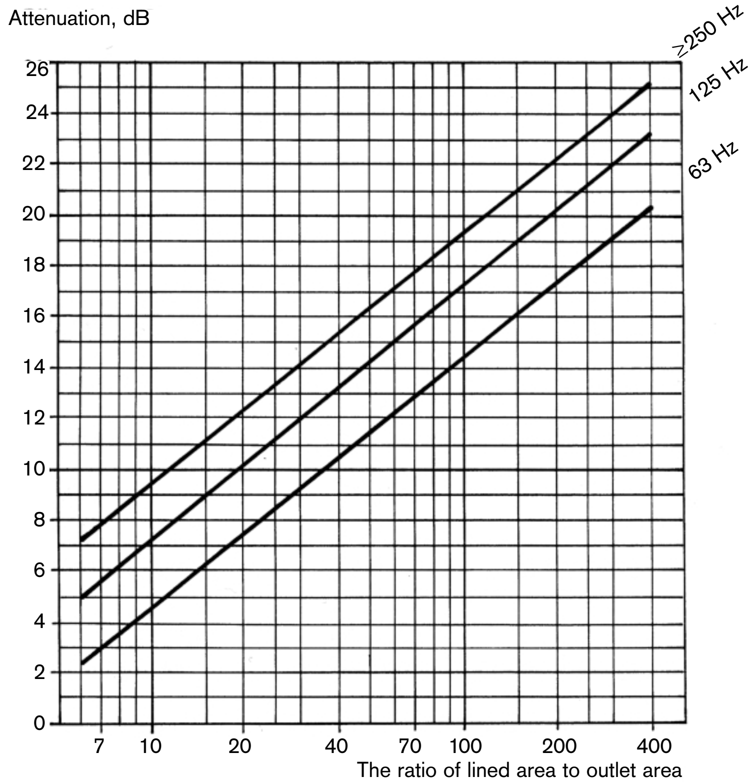

FIGURE 3. Equation (5) expressed graphically. Attenuation in a suction or pressure chamber internally lined with 10 cm of mineral wool.

When sound inside a duct causes it to vibrate, it will also vibrate externally and generate new airborne sound in a room: the inside of the duct acts as a microphone membrane and the outside of the duct as a loudspeaker membrane. Sound will leak, or break out, of the duct. This type of attenuation can, therefore, often have negative consequences, as the sound level in the surrounding space will be raised.

When rectangular ducting is installed in sound-sensitive rooms the ducts should be stiffened by cross creasing, scoring or grooving or by using external stays or stiffeners, and possibly by lining the ductwork with mineral wool or lagging it with gypsum board – see below in the section on crosstalk between ducts and rooms.

Unlined rectangular ducting

Straight unlined rectangular ducts that are not stiffened or strengthened – just like other membrane absorbents, see Chapter 11/Building acoustics – provide relatively good low frequency damping, see Table 1.

Unlined circular ducting

Circular spiral ducts, so-called spiro ducts, are considerably stiffer than rectangular ducts and therefore provide much less low-frequency attenuation and, consequently, less sound break-out, see Table 1.

TABLE 1. Attenuation in straight sections of 1 mm thick sheet steel ducts.

Duct size	**Attenuation in dB/m for different octave bands, Hz**				
	63	**125**	**250**	**500**	**≥1000**
Rectangular ducts					
75–200 mm	0.60	0.60	0.45	0.30	0.30
200–400 mm	0.60	0.60	0.45	0.30	0.20
400–800 mm	0.60	0.60	0.30	0.15	0.15
800–1000 mm	0.45	0.30	0.15	0.10	0.06
Circular ducts					
∅ 75–200 mm	0.10	0.10	0.15	0.15	0.30
∅ 200–400 mm	0.06	0.10	0.10	0.15	0.20
∅ 400–800 mm	0.03	0.06	0.06	0.10	0.15
∅ 800–1600 mm	0.03	0.03	0.03	0.06	0.06

Unlined bends

Depending on the dimensions of the ducts in relation to the particular wavelengths of the sound passing through them, a proportion of the sound will be reflected back into the duct at bends in the system. In round bends, reflection will be naturally quite small but will increase as the frequency of the sound increases, having a maximum value of about 3 dB, see Table 2.

TABLE 2. Approximate attenuation in round bends.

Duct diameter mm	**Attenuation in dB for different octave bands, Hz**					
	125	**250**	**500**	**1k**	**2k**	**≥4k**
125–250	0	0	0	1	2	3
280–500	0	0	1	2	3	3
530–1000	0	1	2	3	3	3
1050–2000	1	2	3	3	3	3

In rectangular lined bends, the damping will be considerably greater, especially at frequencies with the same approximate wavelengths as the width of the ducting. If a bend is lined with an absorbent, the damping will increase significantly.

Lined ducts and bends

Attenuation in lined ducts can be calculated using the following equation:

$$\Delta L = 1.05 \cdot \frac{P}{A} \cdot \alpha^{1.4} \qquad (6)$$

where:

ΔL is the attenuation in dB/m

P is the circumference of the lining in m

A is the cross-section area, the open area, in m^2

α is the absorption factor of the lining material in the relevant octave band

There is no point in lining a longer length of ducting than about five times its width. After this, the sound wave will have become flat and will be transmitted, primarily at its higher frequencies, relatively unattenuated in the middle of the duct (a certain amount of attenuation will, however, occur here by so-called diffraction).

When sound is attenuated using internal linings in the ducting the correct quality of material is essential – it must be able to withstand the air flow through the duct, it must not erode or emit particles, and it must be possible to clean the ducting using standard methods. This is also discussed in Chapter 19/Ducting systems. To avoid erosion of the lining material, the absorptive surface can be protected by using perforated metal sheeting, sometimes with the addition of a thin underlying layer of textile fabric. The metal sheeting will not reduce the absorption capability of the lining, compared to when the lining is completely unprotected, as long as the sheeting is chosen with an open area of at least 20%, i.e. with a maximum of 80% of the surface area of the lining covered by the sheeting.

Lining ducts with absorbents is an effective way of reducing noise, on condition that the material is placed so that the sound waves actually strike it. The absorption materials must therefore be attached to the ducts after changes in size and after bends, points where the air flow is turbulent. For example, lining the inside of the duct connecting to the fan outlet with absorption material is very effective. Here, the sound waves are very turbulent after leaving the fan impeller before they are realigned by numerous reflections against the duct walls. The placing of absorbents like this is called cross-wave attenuation.

When an absorbent lines a bend in a duct system it will attenuate directly incident sound as well as reflected sound both upstream and downstream of the bend. The extent to which the sound is attenuated in the bend will depend on the relationship between the duct size and the wavelength of the sound in the duct.

Table 3 shows the attenuation data for a bend in which the lengths of the lined duct sections before and after the internal bend are at least twice as long as the internal duct width and the lining material has a thickness t corresponding to at least 10% of the duct width, as shown in Figure 4.

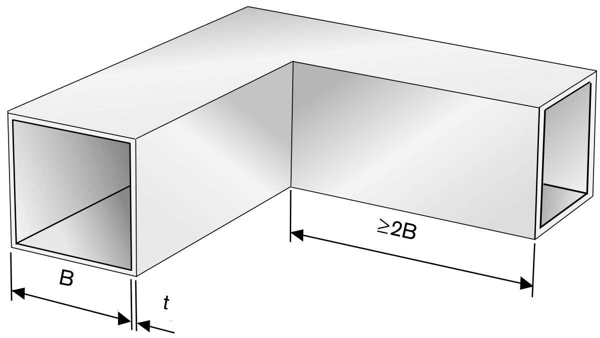

FIGURE 4. A lined bend.

TABLE 3. Attenuation data for rectangular bends with and without absorbent linings.

Internal duct width in mm	**Attenuation in dB for difference octave bands, Hz**						
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**
Bend without absorbent lining							
125				6	8	4	3
250			6	8	4	3	3
500		6	8	4	3	3	3
1000	6	8	4	3	3	3	3
Bend with absorbent lining before the bend							
125				6	8	6	8
250			6	8	6	8	11
500		6	8	6	8	11	11
1000	6	8	6	11	11	11	11
Bend with absorbent lining after the bend							
125				7	11	10	10
250			7	11	10	10	10
500		7	11	10	10	10	10
1000	7	11	10	10	10	10	10
Bend with absorbent lining before and after the bend							
125				7	12	14	16
250			7	12	14	16	18
500		7	12	14	16	18	18
1000	7	12	14	16	18	18	18

Flexible ducts

A flexible duct connection between a sheet steel duct and a supply air terminal device can be a convenient way of ensuring that the device can be positioned to fit the frame pattern of the suspended ceiling. However, care must be taken – if the flexible duct is pulled out of shape this could lead to a significant rise in the sound emitted in the terminal device, compared to that emitted when a straight connection is used.

SOUND TRANSMISSION BETWEEN DUCTS AND ROOMS

Sound leaking from ducting – break-out noise – can be regarded as a sound source in the room through which the duct passes. The sound power level created by a duct, as a source of sound in a room, can be calculated as shown below.

When a duct passes through a room, without either supplying or extracting air, the leakage of sound into or out of the room can cause problems, requiring the duct to be lined or other remedial measures to be taken.

The calculations are similar to those carried out for airborne sound insulation between two rooms, see Chapter 11/Building acoustics. The symbols and the influencing factors are, however, somewhat different:

$$L_{w(outside)} = L_{w(inside)} + 10 \cdot \log\left(\frac{S}{A}\right) - R_{duct} \qquad (7)$$

where:

$L_{w(outside)}$ is the sound power level emitted from the outside of the duct into the room in dB(relative to 1 pW)

$L_{w(inside)}$ is the sound power level of the sound in the duct in dB(relative to 1 pW)

S is the surface area of the duct emitting sound to the room in m^2

A is the cross-section area of the duct in m^2

R_{duct} is the sound reduction index of the duct wall in dB

The sound reduction index R_{duct} varies, depending on:

- Duct shape – whether rectangular or circular. A spiral circular duct is stiffer than a rectangular duct with the same free cross-sectional area and has, therefore, a higher sound reduction index.
- Duct size – the greater the width of a rectangular duct the more it will vibrate and emit noise.

- If the duct sheeting is stiffened, for example by cross creasing, it will have a higher sound reduction index than an unstiffened duct of the same dimensions.
- The frequency of the sound. The sound reduction index varies with frequency and rectangular ducts have a higher index at higher frequencies than at lower frequencies. In spiral ducts the opposite is true.

END REFLECTION

At the point where a duct is connected to a terminal device there will be a low-frequency damping effect due to reflection back into the duct. The damping will depend on the size of the opening and its location in relation to the walls and ceiling, i.e. its directivity factor Q, see Chapter 11/ Building acoustics, and Figure 5.

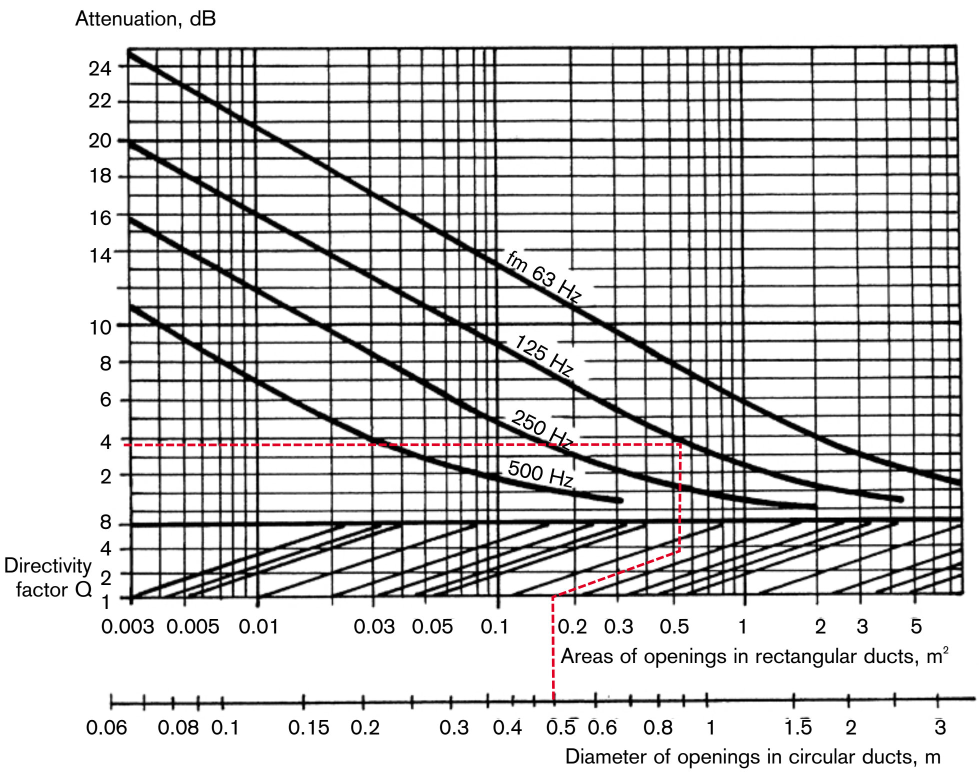

FIGURE 5. Attenuation in terminal devices and duct openings. The dashed line illustrates the following case: Directivity factor, Q = 4 and the duct diameter = 0.5 m. Attenuation at a frequency of 125 Hz will be 4 dB.

SILENCERS

Silencers, or attenuators, should normally be placed as close to the sound source as possible but at a sufficiently large distance to ensure that the air flow in the duct has a reasonably stable speed when it reaches the silencer. As in other natural phenomena, sound also flows from a higher, louder, energy level to a lower, quieter, energy level. This is why a silencer, fitted to reduce the fan sound transmitted by the ducting, should be positioned next to the fan room wall connection. This prevents the sound emitted in the fan room, which would be at a higher level than the sound in a duct section after the silencer, from being transmitted into the duct through the duct walls. If this silencer location is not possible for space reasons, this section of the duct should be lined with mineral wool or lagged using gypsum board.

Equation (6) above shows how attenuation in a lined duct depends on the sound absorption factor α and the relationship between the circumference of the lined duct area and the open area, P/A. If the P/A ratio is increased for a given open area, then the attenuation will become more effective. Advantage of this fact is taken in prefabricated silencers in which the surface area of the lining is increased by placing walls lined on both sides, baffles, parallel to the air flow. The attenuation depends on the distance between the surfaces and the shorter the distance the more effective the silencer will be. This solution will, however, cause an increase in the pressure drop across the silencer.

To improve the attenuation properties at low frequencies, where porous absorbents have low attenuation, see Table 6 in Chapter 11/ Building acoustics, porous material is sometimes combined with the excellent low frequency properties offered by membrane absorbents. This can be achieved by lining part of the porous absorbent surface with sheet steel.

The silencer not only attenuates sound but also generates sound, just like any other duct-mounted components, and, like these, its self noise increases as the air speed increases. When choosing a silencer it is important to check the manufacturers data for both of these properties, i.e. attenuation and self noise.

To facilitate operations when cleaning ducts, it should be possible to remove the silencers, especially those with baffles and those installed inside the supply air ducting. Cleaning ducts is discussed in Chapter 19/Ducting systems.

Crosstalk attenuators

It is often important to prevent speech from being transmitted between workplaces and other rooms in a building, not only for reasons of secrecy but also to reduce disturbing noise. How this affects the choice of parti-

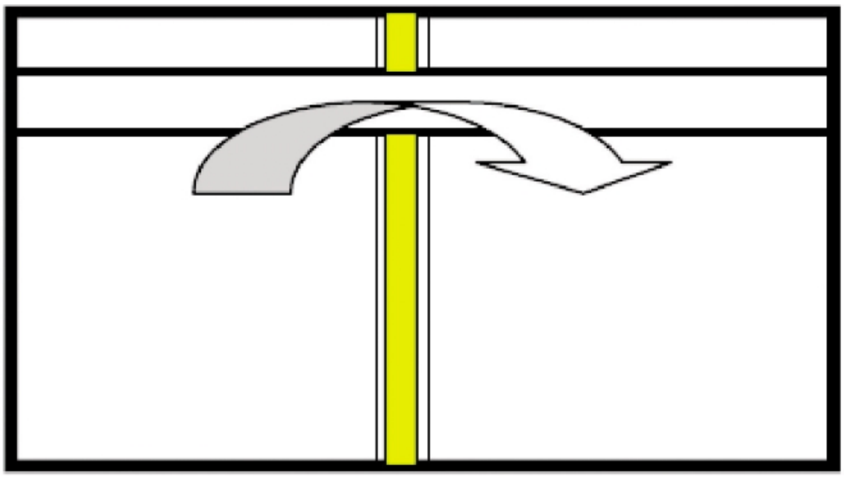

FIGURE 6. Crosstalk between two rooms via ducting.

tion walls, and their sound reduction indices and airtightness, is discussed in Chapter 11/Building acoustics.

When rooms are connected to a common supply air duct and/or extract air duct it is important that speech cannot be conveyed between the rooms, so-called crosstalk. In the daytime, when the ventilation system is in operation, this is normally not a problem – sounds from the system create a background level that is sufficiently high to drown speech. When the system is turned off at the end of the working day the conditions change and the ventilation system no longer has an attenuating effect. Supply and extract air terminal devices should therefore also provide sufficient attenuation, to prevent crosstalk, when air flows in the ducting cease. As an alternative, duct connections to the terminal devices can be fitted with attenuators to reduce crosstalk.

TERMINAL DEVICES

The location of the supply and extract air terminal devices will affect the sound distribution in a room as the walls and ceiling will reflect sound from the terminal devices. This means that terminal devices have directional characteristics, directivity factors, see Chapter 19/Ducting systems. If a device is placed in the corner of a room it will be surrounded by three reflective surfaces. A higher sound level will then result, at a given distance from the device, compared to that from a device placed at the junction of a wall and the ceiling (two reflective surfaces) or in the middle of the ceiling (one reflective surface). For each additional surface the sound at a given distance from the device will increase by 3 dB.

Sound data for a terminal device is normally expressed as the sound level in dB(A) in a room with a 10 m^2 equivalent absorption area measured in the reverberant field of the room. It is therefore important to check the following conditions, which can affect performance, in the installation in question:

- The setting of the terminal device to provide the design air flow and correct distribution pattern.
- The connection of the terminal device to the duct – is this via a straight section or a bend?
- The directivity factors.
- The effects of turbulent flow through the device (disturbances, for example, caused by balancing damper in the duct).

Adjust the sound level value if actual conditions do not agree with the product data, for example, for distances and the equivalent absorption area of the room, and add the increase due to multiple parallel devices in the room.

ACTIVE NOISE ATTENUATION

Although active silencers are unusual today, they might become more common in the future. They are especially suited to attenuating low frequency noise, for which standard attenuators are ineffective. An active attenuator produces sound waves that are out of phase with those in the ducting.

An active silencer functions as follows: A reference microphone is placed upstream on the wall of the duct and measures the sound. After it has been analysed, a mirror image of the measured noise with the same amplitude is fed into the duct via a loudspeaker placed on the duct downstream of the reference microphone. This anti-phase (180 degrees out of phase) sound will effectively interfere with the noise in the duct and attenuate it. The system is also fitted with an error microphone that measures the resulting sound levels after the attenuator and adjusts the analyser to refine the signal to the loudspeaker. As all parts are mounted remote from the air flow the silencer does not create a pressure drop or any self noise.

The air speed and the turbulence of the air flow must not be too high, if the attenuator is to work properly. The microphones used to measure the sound cannot differentiate between sound, i.e. pressure propagation in an elastic medium, and pressure changes caused by air movements in the duct. This is why TV reporters, when interviewing people outdoors, use special windshields to protect their microphones.

Active silencers should only be used in ducts where the air speed does not exceed about 8 m/s and at a sufficient distance from components creating turbulence, i.e. more than about 5 duct diameters upstream and about 3 diameters downstream from them.

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