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Sound in low velocity ventilation ducts.

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THE PREPARATION OF THIS DATA ITEM

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The Internal Flow Panel, which took over the work on Internal Flow previously monitored by the Fluid Mechanics Steering Group, first met in September 1979 and had the co-operation of many engineers and scientists in industry, research establishments and universities from whom much assistance and information is being received.

SOUND IN LOW VELOCITY VENTILATION DUCTS

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1. NOTATION

		<i>SI Unit</i>
A	free cross-sectional area of duct	m^2
A_G	free area of grille	m^2
B	damper blockage factor	–
b	duct height normal to damper axis	m
c	speed of sound in gas inside duct	m/s
D_d	direct sound pressure level directivity correction	dB
D_r	direct sound pressure level distance correction	dB
d	duct diameter (for rectangular ducts equivalent $d = \sqrt{4A/\pi}$)	m
d_c	characteristic dimension of duct cross section	m
F	flow noise spectrum function (see Section 4)	dB
F_D	damper spectrum function	dB
ΔF_D	damper height correction	dB
F_G	grille spectrum function	dB
ΔF_G	grille vane thickness correction	dB
f	octave band centre frequency	Hz
f_{res}	duct resonance frequency (see Equation (3.1))	Hz
G	flow velocity function	dB
H	octave bandwidth function	dB
h	smaller cross-dimension of rectangular duct	m
K	room constant (see Equation (7.3))	m^2
K_t	damper total-pressure loss coefficient	–
k	grille constant (see Section 4.3)	–
L_p	sound pressure level	dB
L_{pd}	direct sound pressure level	dB
L_{pr}	reverberant sound pressure level	dB

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ΔL_{pr}	reverberant sound pressure level multiple grille correction	dB
L_w	octave band sound power level	dB
L_{wB}	breakout sound power level	dB
ΔL_w	octave band attenuation	dB
l	duct length	m
P	perimeter inside duct	m
Δp_t	total-pressure loss across damper	Pa
Q	volume flow rate in duct	m ³ /s
R	airborne sound reduction index	dB
R_T	reverberant sound pressure level reverberation time correction	dB
R_v	reverberant sound pressure level room volume correction	dB
r	distance from centre of duct outlet to observation point	m
Sr	Strouhal number ($Sr = fd/V$)	–
Sr_D	damper Strouhal number ($Sr_D = fd/V_e$)	–
Sr_G	grille Strouhal number ($Sr_G = ft_G/V_e$)	–
Sr_T	transition Strouhal number ($Sr_T = fd/V$)	–
s	surface area of duct	m ²
s_r	room surface area	m ²
T	reverberation time	s
t	duct lining thickness	m
t_G	grille vane thickness	m
V	mean flow velocity	m/s
V_e	effective flow velocity	m/s
v	room volume	m ³
x	length of lined duct at bends (see Table 10.3)	m
α	average reverberation sound absorption coefficient	–
ρ	density of gas in duct	kg/m ³

ρ_s	mass per unit area of duct wall	kg/m ²
θ	damper vane angle	deg
τ	static temperature of gas in duct	K

Only the above SI units should be used with this Item since many of the relationships that involve levels in dB are specific to those units.

2. INTRODUCTION

This Item provides essential data for estimating the change in acoustical energy as sound propagates through a ventilating system. From a knowledge of this sound energy the sound pressure level that results in any room served by the system may be estimated. From consideration of the estimated sound pressure levels in the most sensitive areas of the building the amount of silencing required in the system to achieve acceptable sound levels can be assessed. A guide to the types of silencers and their use in air-conditioning systems is given in ESDU 82002.

Data are provided, in either graphical or tabular form, for the estimation of the attenuation of sound in ducts and duct fittings and for the estimation of the sound generated as air flows through standard duct fittings. A means of estimating the octave band sound pressure levels, due to the air-conditioning system, in a room is also given. Other factors that affect sound in ventilation ducts (*e.g.* breakout) are discussed.

The primary noise source in ventilation systems is fan noise; it should be noted that most areas serviced by a system will be influenced by both delivery and exhaust fan noise. The energy input by a fan may be obtained from manufacturers' catalogues or estimated using the procedure given in ESDU 79037.

The calculation of noise in ventilating systems is made in terms of sound power levels which are measured in decibels (dB). The reference sound power level used in this Item is 10^{-12} W and for sound pressure levels the reference sound pressure level is 20 μ Pa. To ensure that all frequencies, over the range of interest, are considered, the data provided are for each of the octave frequency bands in the range 63 Hz to 8 kHz.

Data provided in this Item are applicable to low velocity air-conditioning systems. A guide to the maximum air velocities that may be used in a low velocity system is 10 m/s in the main risers, 7.5 m/s in the branch ducts and 4 m/s in the ducts serving the grilles. For higher air velocities, considerable attention must be given to the duct design to ensure smooth airflow paths and so minimise velocity generated noise.

For the flow generated noise data in this Item, it is implicit that the airflow upstream of each fitting is not greatly disturbed. If, for example, fittings are too close together so that highly turbulent flow emitted by the upstream fitting flows into the downstream fitting, the noise from the downstream fittings is likely to be greater than predicted using this Item. Information on good duct design to ensure smooth airflows is given in ESDU 82001.

The data provided are intended for use by engineers who are not acoustic specialists. In cases where very low noise levels must be guaranteed, for example in an auditorium where the acceptable noise criterion is likely to be less than NC 30, flow velocities may need to be severely restricted and it is recommended that the advice of an acoustics specialist should be sought. Noise design criteria are discussed in ESDU 82001.

Examples on the use of the data in this Item are given in ESDU 82003.

3. ATTENUATION IN DUCTS

3.1 Attenuation in Straight Ducts

As sound propagates along a rigid-walled straight duct there is no attenuation other than that due to gaseous absorption. This gaseous absorption is usually very small unless propagation distances are long when its effects may be evident in the higher frequency sound. In practice, duct walls are not rigid and sound is attenuated due to the energy lost through vibration of the duct wall (see Section 8). Part of this lost energy is absorbed by internal damping within the duct wall and the remainder is radiated as sound into the area through which the duct passes.

The walls of rectangular ducts are relatively less stiff than those of circular ducts so that at low frequency, where sound transmission through the duct wall is inversely proportional to its stiffness, the attenuation of sound is greatest in ducts of rectangular cross section.

3.1.1 Ducts of rectangular cross section

The attenuation against frequency curve for unlined rectangular ducts is characterised by a peak at low frequency. The attenuation drops sharply above and below this frequency. The peak frequency is a resonance phenomenon caused by the mass of the duct walls interacting with the stiffness of the enclosed air volume. At this resonance frequency, f_{res} , the energy absorbed by the walls causes high attenuation of the sound within the duct. The frequency of this resonant mode is given by

$$f_{res} = \frac{c}{2\pi} \sqrt{\frac{\rho P}{\rho_s A}}, \quad (3.1)$$

where c is the speed of sound given by

$$c = 20.05 \sqrt{\tau}, \quad (3.2)$$

where τ is the absolute static temperature in kelvin*. It is probable that many of the low frequency "rumble" noise phenomena (see Section 8) are caused by this duct wall mode being excited by flow turbulence resulting from flow separation induced by duct fittings (bends, dampers, etc).

Values of attenuation for unlined rectangular ducts are given in Table 10.1 in terms of the ratio P/A . The increase in attenuation in the octave band centred at 8000 Hz is due to atmospheric attenuation.

For internally lined rectangular ducts, the attenuation is proportional to the square of the frequency at low frequencies, and varies inversely as the square of frequency at high frequencies. For a duct lined with fibrous liner boards of thickness t and density 32 kg/m^3 , the low frequency attenuation, over a lined duct length l , is given by

$$\Delta L_w = 0.26 t^{1.38} (P/A) f l, \quad (3.3)$$

and the high frequency attenuation is given by

$$\Delta L_w = \left(\frac{P}{A} \right) \left(\frac{600}{hf} \right)^2 l. \quad (3.4)$$

* $\tau = 273.2 + \text{temperature in degrees Celsius.}$

The frequency range of applicability of the equations for ΔL_w is determined by h , the smallest cross dimension of the duct (*i. e.* minimum of width or depth of the duct free air passage), and is given in Table 3.1.

TABLE 3.1

h (m)	Octave band centre frequency (Hz)					
	63	125	250	500	1000	2000 and above
0.12 to 0.24	Calculate ΔL_w using Figure 1 or Equation (3.3)					Calculate
> 0.24 to 0.76						ΔL_w using Figure 2
> 0.76						or Equation (3.4)

For ease of estimation, the values of ΔL_w given by Equation (3.3) are plotted against $(Pl/A) f$, for a range of values of t , in Figure 1; and ΔL_w given by Equation (3.4) is plotted against (Pl/A) , for a range of values of hf , in Figure 2.

If the lined duct is not periodically broken by flexible connections, the sound attenuation along the duct is limited due to sound transmission within the duct walls (flanking transmission). If, in cases where flanking can occur, the attenuation estimated using Equations (3.3) and (3.4) is greater than 40 dB in any octave frequency band set $\Delta L_w = 40$ dB in that octave band. This limit is incorporated in Figures 1 and 2.

Figure 1 and Equation (3.3) are based on a lining density of 32 kg/m³. For lining densities of 24 kg/m³ and 48 kg/m³ multiply the estimated values of attenuation by 0.76 and 1.38 respectively. For the frequency range covered by Equation (3.4) the attenuation is independent of lining density.

3.1.2 Ducts of circular cross section

Few experimental data are available for the attenuation of sound in ducts of circular cross section. Until more accurate data are available, the approximate attenuation for lined and unlined circular ducts given in Table 10.2 should be used.

3.1.3 Effects of external insulation

Air conditioning ducts are frequently provided with external lagging to provide thermal insulation and prevent condensation. This external lagging does not affect the attenuation of sound in the duct at frequencies of 1000 Hz and above. Duct external lagging increases the attenuation at low frequencies and this may be approximately accounted for by doubling the attenuations found for the unlagged ducts in the octave bands centred at 63 Hz, 125 Hz and 250 Hz.

3.1.4 Effect of flow velocity

For unlined ducts there is no appreciable effect of flow velocity on sound attenuation below a flow velocity of 25 m/s. For the case of internally lined ducts of rectangular cross section it has been found that for flow velocities less than 10 m/s, and at all flow velocities if the sound propagates against the flow direction, the effect of flow velocity on sound attenuation is negligible. If the flow velocity exceeds 10 m/s and the sound propagates in the direction of flow, the values of attenuation rates given by Equations (3.3) and (3.4), or Figures 1 and 2, should be multiplied by 0.9.

3.2 Attenuation at Duct Bends

Sound attenuation at duct bends and elbows is due to reflection of sound back towards its source. Maximum attenuation occurs at 90° elbows in rectangular ducts where the reflecting surface is normal to the direction of the propagating sound wave. In rectangular ducts the attenuation peaks at the frequency having a wave length equal to half the width of the duct. Values of attenuation for 90° elbows are given in Table 10.3.

3.3 Attenuation at Duct Branches

At duct branches there is no change in the total sound energy propagating away from the source. The sound attenuation within the main duct is due to the division of energy between the duct branches.

The sound power level, in each octave frequency band, in a duct following a duct branch is obtained by subtracting ΔL_w from the duct sound power level immediately before the branch, where

$$\Delta L_w = 10 \log_{10} \left\{ \frac{\text{sum of cross-sectional areas of ducts after branch}}{\text{cross-sectional area of duct}} \right\}. \quad (3.5)$$

3.4 Attenuation at Duct Outlets

At an outlet into a room some of the energy is reflected back down the duct. The amount of attenuation of sound due to these end reflections is dependent on both the sound frequency and the duct outlet area. This attenuation may be considerable in the lower octave frequency bands but is small at the higher frequencies. Outlet attenuations are plotted against total outlet area, for each octave band of interest, in Figure 3.

4. FLOW GENERATED NOISE

Although fans are the major source of sound in a duct system, any obstruction or fitting within the duct that disturbs the flow is capable of generating noise. Even without fittings, surface-induced noise is generated by gas flowing over the duct walls although this noise source is usually negligible compared with noise induced by fittings. Through each duct fitting, the sound power levels in each octave band depend on the geometry of the device, the air turbulence and the air flow rate. It is not possible to provide a general procedure for the calculation of flow generated noise but, in the following sub-sections, methods of estimating flow induced noise for certain types of fitting are given.

The significance of flow generated noise through any component within the duct should be assessed by comparison with the sound power level in the duct on the serviced-area side of the duct fitting. If the flow generated noise, in each octave band, is more than say 9 dB below the ambient duct noise, it will not have made a significant contribution to the duct noise in that octave band.

The method of flow noise prediction in this section is based on limited experimental data obtained from ducts of square or rectangular cross section. Partly due to the inherent difficulty of measuring duct noise, particularly at the low frequencies, differences between measured and predicted sound power levels may, in some cases, be in excess of ± 5 dB.

The flow noise generated in each octave band for the fitting considered is estimated using the following relationship

$$L_w = F + G + H, \quad (4.1)$$

where F is a spectrum function for the particular fitting, G is a velocity function and H is an octave bandwidth function. The bandwidth function is used to convert the octave band levels to spectrum levels. Values of H for the octave bands of interest are given in Table 4.1.

TABLE 4.1

Octave band centre frequency (Hz)	63	125	250	500	1000	2000	4000	8000
H (dB)	16.4	19.5	22.5	25.5	28.5	31.5	34.5	37.5

The methods of estimating F and G are given in the appropriate following subsections.

4.1 Duct Bends

In Figure 4, the values of F are given for square cross section 90° bends with and without circular-arc turning vanes and, in Figure 5, F is given for one type of rectangular cross section 90° bend without turning vanes or with one central splitter. The values of F are given in terms of the non-dimensional Strouhal number, Sr , based on the effective duct diameter, d .

Values of G are given in Figure 6 where A is the duct area and V is the average duct flow velocity.