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Reduction of sound in ventilation and similar air distribution systems.

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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.

REDUCTION OF SOUND IN VENTILATION AND SIMILAR AIR DISTRIBUTION SYSTEMS

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REDUCTION OF SOUND IN VENTILATION AND SIMILAR AIR DISTRIBUTION SYSTEMS

1. NOTATION

		<i>SI Unit</i>
A	free cross-sectional area of duct	m^2
b	rectangular silencer splitter thickness (see Sketch 4.1)	m
d_c	cylindrical silencer reference diameter (see Sketch 4.1)	m
h	rectangular silencer airway width (see Sketch 4.1)	m
IL	silencer static insertion loss	dB
K_p	plenum chamber constant (see Equation (6.2))	m^2
L_w	silencer airflow generated noise	dB
l	silencer length	m
P	perimeter inside duct	m
Δp_t	silencer total pressure loss	Pa
r	plenum chamber direct pathlength (see Sketch 6.1)	m
s_e	plenum chamber exit area	m^2
s_p	total surface area of plenum chamber	m^2
$s_1, s_2, \dots, s_n$	individual surface areas of plenum chamber	m^2
$\bar{\alpha}$	average reverberation sound absorption coefficient	—
$\alpha_1, \alpha_2, \dots, \alpha_n$	absorption coefficients associated with plenum chamber surfaces	—
θ_p	angle between direct path and normal to exit to plenum (see Sketch 6.1)	deg
λ	sound wavelength	m

2. INTRODUCTION

This Item provides data for estimating the attenuation of sound in lined plenum chambers and ducts, and through silencers of the types used in ventilation and air-conditioning systems. These data are complementary to those presented in ESDU 81043 which provides a means of estimating the sound pressure levels in rooms that are directly attributable to the ventilation system.

Data are provided in graphical or tabular form for the estimation of typical silencer insertion loss, airflow generated noise and pressure loss. Both rectangular splitter silencers and cylindrical silencers are considered. Also, equations are provided for the estimation of plenum chamber attenuation. The attenuation of sound in lined ducts is discussed with reference to the estimation procedure given in ESDU 81043. An example is given on the selection and sizing of a silencer for a typical ventilation system, and the suitable location of silencers within the duct system is discussed.

The calculation of noise in ventilation and air-conditioning 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. 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.

During the design stage of a ventilation system there are a number of factors that should be considered as a means of avoiding possible noise problems. Firstly, the flow velocities should be kept to the practical minimum since even a small reduction in flow velocity can significantly reduce the flow generated noise due to duct fittings. An indication of the reduction of flow generated noise with air velocity may be obtained, for the case of a silencer, from Figure 3. Useful gains in low frequency attenuation can be achieved by increasing the number, and reducing the size of, the duct outlets into a room. These gains are due to an increase in end reflection loss (see Figure 3 of ESDU 81043). Further reductions can be made by locating grilles at the maximum possible distance from the room occupants so reducing the direct sound pressure levels. However, before increasing the number of, or repositioning, duct outlets it is essential to consider both the economics of increasing duct lengths and the efficiency of air distribution within the serviced space. Another way of controlling sound within a room is by decreasing reverberation time which reduces the reverberant sound pressure levels. The reverberation time in a room may be reduced, for example, by the use of carpets and acoustic ceiling tiles.

The data provided are intended for use by heating and ventilation engineers when selecting the appropriate noise reduction equipment for a ventilation and air-conditioning system. The data may be used in the initial design of a system or to investigate the appropriate remedial action for an installed air distribution system that is found to be too noisy. This Item, and the data provided, is 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, it is recommended that the advice of an acoustics specialist should be sought. Noise design criteria are discussed in ESDU 82001.

In Section 3, the reduction of sound by duct liners is discussed and reference is made to duct liner attenuation data. The types of purpose-built silencers for air distribution systems are described in Section 4, and a method of estimating their insertion loss, airflow generated noise and pressure loss is given. In Section 5 the location of silencers within the duct system is considered and in Section 6 a method is provided for the estimation of plenum chamber attenuation. A worked example that illustrates the selection and sizing of a silencer for a particular application is given in Section 8.

* An introduction to noise units and their basic relationships is given in ESDU 82001.

3. INTERNALLY LINED DUCTS

Lining a duct with porous absorbing material can significantly increase duct sound attenuation and, at the same time, increase thermal insulation. In a lined duct, sound propagates both in the free duct and in the duct lining. It is in the duct lining that sound energy is absorbed as a result of the flow resistance within the pores of the lining material converting a portion of the sound energy to heat. The energy dissipated in the lining is replaced from the sound propagating in the free duct so providing continuous sound attenuation along the lined duct. For high sound attenuation, a large ratio of lined perimeter to free area, P/A , is required and the lining thickness should not be too small compared with the wavelength of the sound to be attenuated. Also the flow resistance of the lining material should be high to dissipate a large part of the sound energy propagating in it; however, this lining flow resistance should not be so high that it inhibits the passage of sound waves into the absorber lining.

The requirement, for high sound attenuation, that the lining thickness should not be too small compared with the sound wavelength means that for significant low frequency attenuation (octave band centred at 500 Hz and lower frequency bands) the thickness of the lining material should be at least 50 mm. Insertion of thick lining material reduces the duct free cross-sectional area so increasing the flow velocity for the same volume flow rate. If this increase in flow velocity is high the flow noise generated at duct bends and fittings may be significantly increased.

At high frequencies, the sound attenuation due to duct liners is reduced since high frequency waves tend to concentrate in the centre of the duct and are therefore less affected by the lining material. This fall-off in high frequency sound attenuation is most marked when the sound wavelength is of the order of, or less than, the duct width.

In ventilation systems where mineral fibre liners are often used, it is usual to face the mineral fibre with a porous fibre mat to reduce erosion of the silencing material. In some cases where this erosion must be eliminated (*e.g.* in "clean air" areas such as pharmaceutical laboratories, operating theatres, *etc.*) the liner may be enclosed in an impervious material (*e.g.* melinex). These impervious lining covers have some effect on the sound attenuation properties of the lined duct, and may give an increased attenuation at low frequencies and a reduction at the higher frequencies.

The attenuation of sound in rectangular ducts may be estimated using the relationships given in Section 3 of ESDU 81043. In that same source, data are also provided on the attenuation of lined circular ducts and lined duct bends. Significant attenuation of the mid and high frequency sound is achieved by lining the duct leading into, and leaving, mitred bends.

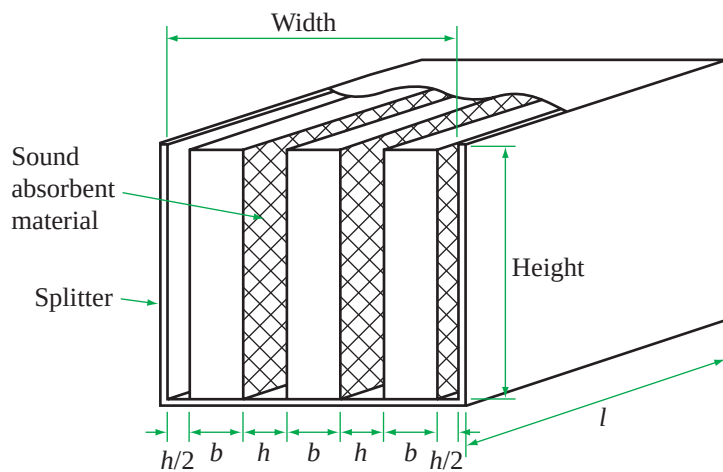
In selecting a suitable duct lining material, it is necessary to consider more than its acoustical characteristics. The material must have adequate mechanical strength to avoid breakage and crumbling, and erosion at high air velocities. Further, lining materials should be fire resistant, odourless, and should not absorb substances that will impair their sound absorption properties (*e.g.* grease from kitchens).

4. SILENCERS FOR AIR DISTRIBUTION SYSTEMS

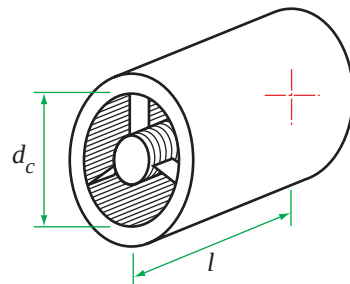
4.1 General Notes

It is not always possible, or may not be practical, to achieve the required reduction in sound in an air distribution system solely by the use of duct liners. This is particularly so when significant low frequency sound attenuation is required which can only be gained by using thick lining material. For cases where large sound attenuation is required, or where insufficient duct length is available for lining to give the required sound reduction, purpose-built silencers should be used. An advantage of using purpose-built silencers, rather than lining ducts, is that sound can be attenuated in a short distance so avoiding the problem of noise break-out from a long length of duct in which noise levels are high. Also if duct cross-sectional areas are large, which is often the case in the system connected to the primary fan, duct lining may not be really effective due to the concentration of sound waves in the centre of the duct.

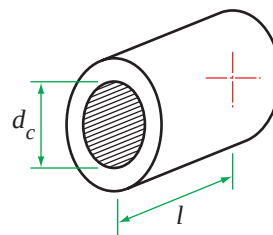
The two types of commonly used silencers that are considered in this Item are cylindrical silencers and rectangular splitter silencers. The construction of these silencers is illustrated in Sketch 4.1 where cylindrical silencers both with and without a concentric pod are shown. Particularly when low pressure losses are required, aerodynamically shaped entrance and exit pieces may be attached to the rectangular silencer splitter elements and to the cylindrical silencer central pods. Some rectangular splitter silencer manufacturers add half splitters on the silencer sides. Apart from increasing the sound absorption area, this latter addition helps to prevent flanking transmission and the resulting stiffening of the side panels reduces the possibility of their vibrating.



(a) Rectangular splitter silencer



(b) Cylinder silencer with pod



(c) Cylinder silencer without pod

Sketch 4.1 Typical silencers

The factors to be considered in assessing the acoustic performance of a silencer are its insertion loss (IL), its pressure loss (Δp_t), and its airflow generated noise (L_w). The insertion loss is the reduction in the rate of flow of sound energy, in terms of the reduction in sound power level in each octave frequency band, after a silencer due solely to the insertion of the silencer in the system. The pressure loss is the difference between the mean total pressures at the silencer inlet and outlet, and the airflow generated noise is the octave band sound power levels radiated through either the silencer inlet or outlet as a result of airflow through the silencer.

Silencer manufacturers quote either experimentally based static values of insertion loss together with airflow generated noise characteristics or dynamic values of insertion loss. Static insertion loss is measured without airflow and dynamic insertion loss is measured in the presence of airflow. Hence, for a specific value of air velocity through a silencer, the dynamic insertion loss may be used or, alternatively, the static insertion loss in conjunction with the velocity generated noise may be used. In Reference 5, methods of testing silencers for air distribution systems are described that lead to values of static insertion loss and airflow generated noise levels. It is noted in Reference 5 that significant variation of insertion loss may occur at high air velocities and consequently the values of insertion loss obtained by the recommended methods should be interpreted with caution if the air velocity within the silencer exceeds 20 m/s. In practice, it is not desirable to have airway velocities in excess of 20 m/s so static insertion loss data provided in this Item are applicable to most ventilation systems. In the case of silencers that are designed for direct coupling to a fan, the fan and attached silencer should be tested as a single unit using the test procedure described in Reference 4. These latter test procedures lead to values of dynamic insertion loss. In quoting dynamic insertion loss, it is common practice for manufacturers to give values for low, medium and high fan blade pitch settings rather than to quote specific flow velocities through the silencer. Because it is easier to attach cylindrical silencers to fan casings, dynamic insertion losses are often only quoted for cylindrical silencers.

Data provided in this Item are mean values for typical silencers designed for use in ventilation and air-conditioning systems and are applicable over the normal range of temperatures* encountered in those systems. Some manufacturers provide silencers for special application for which data provided here are not applicable. An example of such special silencers is the rectangular splitter silencer designed to give higher insertion loss in the lower frequency octave bands. Because of differences in silencer designs it is recommended that when a silencer is selected for a particular application the acoustical performance given in the manufacturers' catalogue should be used. However, when comparing the specified performance of different silencers it should be noted that, because of inaccuracies inherent in test measurements, a possible error in measured values of insertion loss and airflow generated noise of the order of ± 3 dB in the octave band centred at 125 Hz and ± 2 dB in the higher frequency octave bands may occur. Tolerance limits on the data given for the band centred at 63 Hz are likely to be greater than ± 3 dB. The tolerance on measured pressure loss is of the order of ± 5 per cent.

In using the silencer data provided, or obtained from manufacturers' catalogues, it should be realised that the values of insertion loss and pressure drop are only applicable when the flow entering the silencer is smooth. If the airflow entering the silencer is significantly disturbed, as may be the case when upstream duct fittings are close to the silencer, the achieved insertion loss will be less than that predicted by this Item and the pressure loss will be higher than predicted. Care should be taken in the design of the attachment of the duct to the silencer to ensure smooth flow conditions at both the silencer entrance and exit. If such smooth flow cannot be achieved an allowance of up to 50 per cent increase[†] in pressure loss should be made.

* Under high temperature conditions sound attenuation will be lower than predicted using this Item. The effect of a temperature rise is to increase both the speed of sound and, at a given frequency, the wavelength.

† If good ductwork design practices are not followed, pressure losses may be higher. Some illustrations of poor duct design and suggested improvements are given in Table 9.4 of ESDU 82001.

End reflection loss should also be accounted for if a silencer is directly connected to the free inlet or discharge of an axial flow fan. The additional attenuation due to the end reflection losses can be estimated from Figure 3 of ESDU 81043. Similarly end reflection losses will enhance the attenuation of a silencer that discharges into a plenum chamber. End reflection losses provide attenuation in the lower octave frequency bands and are only significant if the free exit area is small. However, it should be noted that air discharging into free space, or a plenum chamber, will result in a pressure loss penalty compared with the case of a silencer with ducted inlet and outlet.

Dynamic insertion loss data should be used with care when the direction of the airflow is opposite to the direction of sound propagation. These contra-flow conditions can exist in ducts attached to axial flow fan inlets and in return air systems. In the absence of specific data for the contra-flow condition, it is recommended that insertion loss data for a no flow condition be used.

4.2 Rectangular Splitter Silencer Selection

Silencer selection is the choice of a silencer that provides the required insertion loss with an acceptable pressure loss, and that will fit in the space available. For rectangular splitter silencers, the insertion loss is largely determined by the airway width and silencer length, while the pressure loss is largely determined by the air velocity in the airways. In general, an increase in length or a reduction in airway width will increase the insertion loss of the silencer but, if the airway width is reduced, it may be necessary to add splitter modules or increase the airway height to keep the pressure loss acceptable. Where a choice is available, the longer silencer will generally be more economical. In selecting the silencer cross section, care should be taken to achieve an aerodynamically efficient transformation to the adjacent ductwork.

4.2.1 Rectangular silencer static insertion loss

In Figures 1 and 2 static insertion loss spectra are provided for a range of values of b/h (see Sketch 4.1) for silencers having $b = 200$ mm and $b = 100$ mm, respectively. The left and right parts of these figures are used together and each figure may be entered with either a silencer length or an octave band centre frequency.

To obtain the appropriate silencer geometry for a specified insertion loss in a given octave band, enter the Figure at the centre frequency of the octave band considered. From the intersection of the frequency line on the spectrum for the selected b/h ratio, draw a line parallel to the base line. At the intersection of the drawn line and the required insertion loss, the silencer length can be read on the silencer length scale. Similarly, if the silencer length is known, enter with that length and, at the intersection with the required insertion loss, draw the line parallel to the base line to meet the frequency line considered. At this intersection the required ratio b/h can be interpolated.

The use of Figures 1 and 2 is illustrated by the example in Section 8 and the construction, applicable to that example, is shown on the Figures.

The scales for IL , in Figures 1 and 2, are terminated at a sound power level of 50 dB because this level is not usually exceeded due to sound flanking transmission paths through the silencer and duct structure.

4.2.2 Airflow generated noise

In Figure 3, airflow generated noise, L_w , is plotted against the airway velocity for a range of silencer face areas. The airway velocity is defined as the mean air velocity, at the minimum airway section, between silencing elements, and the silencer face area is the gross area of the silencer at the inlet connection (*i.e.* width $\times$ height shown on Sketch 4.1). Graphical data provided are for the octave bands centred at 63 Hz and 125 Hz; for other octave bands the appropriate correction, tabulated on Figure 3, should be added to the 125 Hz band sound power level.