

Table 3 Decibel Correction for Area

A sq ft	Decibel Addition	A sq ft	Decibel Addition	A sq ft	Decibel Addition
0.5	-3	8	9	70	18
1	0	16	12	100	20
2	3	20	13	200	23
4	6	40	16	400	26

Values of the decibel addition for various values of A are given in Table 3.

Example 2: Assume a deflector vane type of grille with a core area of 2 sq ft. Determine the sound power level in the speech interference bands for an air velocity of 2000 fpm.

Solution: From Fig. 8, the L_{WST} for a 1 sq ft deflection grille is 59 db. From Table 2, an addition of 3 more db is required, so that $L_{WST} = 62$ db.

Diffuser Noise

Aside from the information furnished by diffuser manufacturers, to their users, on sound levels generated by air diffusers, few other data are available. Recently, however, some test results on several types of ceiling air diffusers have yielded a satisfactory correlation of the sound generating characteristics of these devices.¹² The data, covering ceiling diffusers ranging in size from 4- to 18-in. neck diameter, show that the overall sound power level (from 75 to 10,000 cps) can be found within ± 2 db by the equation:

$$L_W = 13 \log_{10} A_{min} + 60 \log_{10} V_{max} - 2 \text{ db} \quad (20)$$

where

L_W = overall sound power level in the seven octave bands from 75 to 10,000 cps, decibels re 10^{-12} watt.

A_{min} = minimum flow area for the air passing through the diffuser, square feet. For undamped flow the minimum area will normally occur in the diffuser neck, whereas for damped flow it will occur at the damper position (see Fig. 9).

V_{max} = maximum air velocity ($g/60 A_{min}$), feet per second.

For ceiling diffusers of cone-type construction used in conjunction with deflecting vanes and a damper, Fig. 9 presents the approximate octave band spectrum relative to the overall sound power level as given by Equation 20. The curves of Fig. 9 agree within approximately 3 db of the octave band

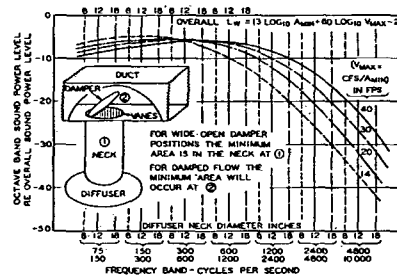


Fig. 9 Octave Band Spectrum of Ceiling Diffusers

data collected on 8-, 12-, and 18-in. neck diameter ceiling diffusers.

NOISE GENERATED BY OTHER SOURCES

Objectionable noise may be produced by fan motors, pumps, compressors, etc. The designer must give attention to these sources of noise. It is not possible to predict the noise, because units from different manufacturers differ greatly in the noise produced.

Objectionable noise is also generated in the duct system by excessive air velocity or poor duct fittings. No systematically taken data have been published from which conclusions can be drawn. The reader is referred to Chapter 21 Air Duct Design for good design practice.

SOUND ATTENUATION IN DUCTS

Unlined Straight Sheet-Metal Ducts

The attenuation of sound in straight sheet-metal ducts is a function of the length, shape, and size of the duct. If a thermal insulating material is applied to the outside of the duct this will also affect the attenuation characteristic.¹³ This was shown in an investigation¹⁴ conducted at the Massachusetts Institute of Technology where two sizes of ducts were tested, a 12 x 12 in. and a 12 x 24 in.

These ducts were investigated not only bare, but also with two types of glass fiber thermal insulation applied to the out-

Sound Control

Table 5 Natural Attenuation of Air

Frequency cps	Attenuation db/ft
40-850	neg.
1700	0.003
3400	0.008
6900	0.022
10000	0.035

side of the duct. The first type used was a 1-in. thick glass fiber insulation with a density of 0.75 lb per cu ft. The second type was a glass fiber semirigid 1-in. board with a density of 6 lb per cu ft. Attenuation values were determined by plotting L_p readings made at 3-ft intervals over a total duct length of 40 ft. The results are presented in Table 4.

It should be noted from Table 4 that the addition of thermal insulation to the outer duct walls had little or no effect on the attenuation in the four octave bands above 600 cps for the ducts tested. In the lower octave bands, however, the effect of added thermal insulation on attenuation is indeed striking. More data on larger size ductwork are needed to confirm these results.

As duct size is increased, the attenuation rate decreases toward the natural rate in free air which is shown in Table 5 for 68 F air at 60 percent relative humidity.¹⁴

Duct Lining Procedure

In the past, the most common method of obtaining sound absorption in ventilating systems has been to line the duct with absorbing material. It is usually more convenient to line all four sides of the duct, but a lining on one side over a longer length of the duct will, in general, give the same effect for the same area of applied acoustical material. Subject to certain restrictions, the attenuation, in decibels, of a fully lined duct to single-frequency sounds may be expressed by the approximate Equation 21.¹⁵

$$R = 12.6 \frac{P}{A} \alpha^{1.4} \quad (21)$$

where

R = attenuation, decibels.

l = length of lined duct, feet.

P = perimeter of duct inside the lining, inches.

A_s = cross-sectional area of duct inside the lining, square inches.

α = absorption coefficient of lining (a function of frequency).¹⁶

This formula was empirically developed for a set of duct sizes ranging from 9 x 9 in. to 18 x 18 in.; for cross-sectional dimension ratios of 1:1 to 2:1; for frequencies between 256 and 2048 cycles; and for absorption coefficients between 0.20 and 0.80. The duct lining material used was 1-in. rock wool sheet. The absorption coefficients of a material of this type in one-half and one-inch thickness are listed in Table 6.

It is also possible to calculate the absorption by a very complicated mathematical theory.^{17, 18} Such calculations are in substantial agreement with Equation 21. This equation may be in error when applied to other types of duct lining and to duct sizes and shapes greater than those specified.

With every application, the use of sound absorptive material should be considered in the dual function of insulation and

Table 6 Attenuation Data for Typical 1-in. and 1/2-in. Thick Duct Lining Board

Frequency Cycles Per Second	1-Inch Thickness			1/2-Inch Thickness		
	Absorption Coefficient α	$\alpha^{1.4}$	Attenuation db	Absorption Coefficient α	$\alpha^{1.4}$	Attenuation db
128	0.12	0.051	$0.64 l \frac{P}{A}$	0.09	0.034	$0.43 l \frac{P}{A}$
256	0.38	0.26	$3.2 l \frac{P}{A}$	0.25	0.15	$1.9 l \frac{P}{A}$
512	0.70	0.60	$7.6 l \frac{P}{A}$	0.40	0.28	$3.5 l \frac{P}{A}$
1024	0.80	0.73	$9.2 l \frac{P}{A}$	0.72	0.63	$7.9 l \frac{P}{A}$
2048	0.79	0.72	$9.1 l \frac{P}{A}$	0.78	0.71	$8.9 l \frac{P}{A}$

sound absorption. It has been shown theoretically that the reduction (in decibels per linear foot) of sound transmitted along a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the variables which will apply accurately to all cases. However, it may be stated generally that the attenuation in decibels at a given frequency is directly proportional to the length of lined duct. It decreases as the cross-sectional area increases, and increases as the aspect ratio is increased.

It has been shown¹⁹ that the attenuation rate is reduced in the presence of an air stream of appreciable velocity.

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the alternating sound waves. As the ratio of the cross-sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low-frequency range, from the flexural vibrations of the duct. In the selection and application of the absorptive material, the following points should be considered:

1. For the absorption of the low-frequencies below 500 cps the material should be 2- to 12-in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb only the high frequencies.

2. In order to provide as much low-frequency noise absorption as possible by means of flexural vibration, it is desirable to fasten the absorptive panels discontinuously. This result may be attained to some extent by spot cementing, but better results are obtained when it is possible to fasten the absorptive panels to furring strips, leaving an air space behind. However, the exact resonance characteristics of the panels, and thus their

Table 4 Octave Band Attenuation of Bare and Externally Insulated Straight Sheet-Metal Ducts

Duct Size Inches	Type of Insulation	Density	Attenuation—db per ft							
			Frequency Band—Cycles per Second							
			20-75	75-100	150-300	300-600	600-1200	1200-2400	2400-4800	4800-10,000
12 x 12 26 gage	none	—	0.26	0.26	0.20	0.08	0.08	0.12	0.12	0.16
	1-in. glass fiber blanket	0.75	0.40	0.40	0.32	0.16	0.08	0.24	0.12	0.16
	1-in. glass fiber blanket	6.0	0.66	0.66	0.50	0.26	0.08	0.24	0.12	0.16
12 x 24 24 gage	none	—	0.30	0.36	0.16	0.06	0.12	0.12	0.12	0.16
	1-in. glass fiber blanket	0.75	0.33	0.44	0.22	0.10	0.12	0.12	0.12	0.16
	1-in. glass fiber board	6.0	1.04	0.56	0.28	0.16	0.12	0.12	0.12	0.16