TABLE 2. ATTENUATION IN STRAIGHT SHEET METAL DUCT RUNS

Duct	Size, In.	Attenuation per Ft, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

2. With total pressure and tip speed constant.

$$db \text{ (change)} = 10 \operatorname{Log}_{10} \left(\frac{\text{Size}_{\text{new}}}{\text{Size}_{\text{old}}} \right)^2 \qquad (2)$$

3. With size and tip speed constant.

$$db \text{ (change)} = f \text{ (Total Pressure}_{\text{old}} - \text{Total Pressure}_{\text{new}}) \qquad (3)$$

The factor f is a function of the fan type. For a centrifugal fan with backward curved blades $f = 9.6$. For a single inlet single width type of ventilating fan with backward curved blades, the noise level at the fan discharge or intake operating at 4000 fpm tip speed and total pressure of 1 in. may be approximately 65 db. The noise level of a double inlet width fan may be 3 db higher than a single width fan at similar conditions of tip speed, total pressure, and size. For the same tip speed and size the noise level of a fan with forward curved blades is higher than for one with backward curved blades, however, the capacity of the fan with the forward curved blades will be greater.

NATURAL ATTENUATION OF DUCT SYSTEM

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[3]. Attenuation values are given in Table 2. In general, this attenuation is so negligible except for long runs that it may be disregarded for all practical purposes.

Elbows and Transformations. Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 3.

TABLE 3. ATTENUATION OF ELBOWS[a]

Elbow	Size, In.	Attenuation per Elbow, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 plus	1

[a]The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the attenuation values listed.

[3]A.S.H.V.E. RESEARCH PAPER—Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1942, p. 317).

When the area of a duct increases, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets, the area of the outlet plus the area of the duct after the outlet is greater than the duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 4 gives the *db* reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room. The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation. This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabines*. (The sound absorption of a room in *sabines* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient). The attenuation is given in Equation 4 as:

$$db \left(\begin{array}{c}\text{Attenuation between} \\ \text{grilles and room}\end{array} \right) = 10 \log_{10} \frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \qquad (4)$$

Values in Table 5 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation is that which must be supplied by the proper sound treatment of the ducts.

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, several points should be considered.

1. For the absorption of the low frequencies the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.

2. In order to take advantage of low frequency noise absorption by panel vibration, it is advisable to fasten the absorptive sheets to stripping so that the panels themselves may vibrate. However, the exact resonance characteristics of the panels and thus their absorption is so unpredictable that panel resonance cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies[4], (2) adequate strength to avoid breakage, (3) fire resistant and comply with national and local code requirements,

[4]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association*, 919 No. Michigan Ave., Chicago, Ill.