

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:

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

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 the 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.

Sound absorption material for ducts should meet the several requirements listed herewith:

1. High absorption at low frequencies⁴.
2. Adequate strength to avoid breakage.

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

3. Fire resistant—should comply with national and local code requirements.
4. Low moisture absorption.
5. Freedom from attack by bacteria and algae.
6. Low surface coefficient of friction.
7. Particles should not fray off at the higher design velocities.
8. Odor free when either dry or wet.

The discussion which follows covers the application and design of the various types of absorbers. For each absorber an attenuation formula or table is given which will give results as accurate as predictable under the present status of our knowledge. With every application the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically⁴ that the reduction, in decibels per linear foot, of sound transmitted through 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

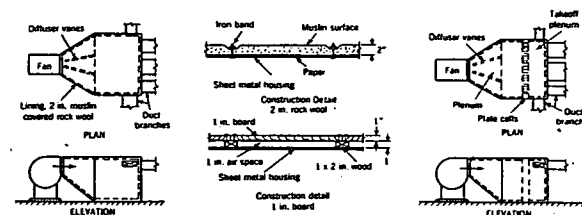


FIG. 1. ABSORPTION PLENUMS WITH AND WITHOUT SOUND CELLS

evidence likewise indicates that there is no simple formula involving the above variables which will apply accurately to all cases.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, therefore, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

Plenum Absorption

In systems, where individual ducts are directed to a number of rooms and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 1, will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 5.

$$db \text{ (attenuation)} = 10 \log_{10} \frac{\text{Plenum Absorption in Sabines}}{\text{Area Fan Discharge}} \quad (5)$$

⁴Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (Journal Acoustical Society of America, Vol. 9, p. 1937).