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 absorption, are so unpredictable that flexural vibration cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies[5], (2) adequate strength to avoid breakage, (3) fire resistance and compliance 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, and (8) freedom from odor when either dry or wet.

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 theoretically[6] 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 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.

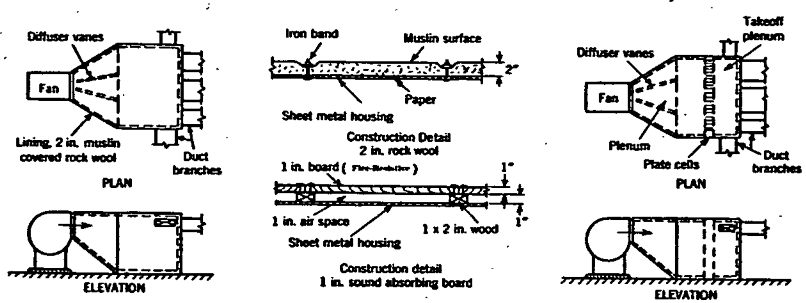

Fig. 1. Absorption Plenums With and Without Sound Cells

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, 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.

Since ventilating system noise contains many frequencies, an exception should be noted to the statement above that attenuation in decibels is directly proportional to length of duct. Most sound absorbent materials are more efficient at high frequencies than at low frequencies. In consequence, the attenuation in the first five or ten feet of lined duct will be greater because the high frequencies are being absorbed. Thereafter, since low frequencies will be predominant, the over-all noise attenuation per foot will gradually be less.

Acoustic Impedance of Absorptive Materials

In the past five years considerable literature has collected describing methods of determining the acoustic impedance of sound absorbent materials and methods of utilizing this quantity for predicting the acoustics of rooms[7] and the attenuation of sound in ventilating ducts[8]. Acoustic impedance as a concept is derived from electrical circuit theory. The effect of the sound absorbent material upon incident sound waves is described in terms of a resistive and a reactive component which may be determined by specifically devised apparatus[9].

Generally speaking, however, the use of acoustic impedance theory involves rather elaborate mathematical calculations, and the improved accuracy obtained is largely off-set by variations in the materials themselves and in their methods of mounting. It has been difficult to measure the acoustic impedance of large areas of material mounted in a manner typical of standard construction. P. E. Sabine[7] concludes that the assumptions required by acoustic impedance theory make this method of calculation of no immediate practical advantage in the measurement of sound absorption coefficients.

Beranek[8] compares results of sound attenuation observed for rectangular ducts lined with absorbent material computed by acoustic impedance theory with data reported by H. J. Sabine[10], using the methods of this chapter. Beranek concludes that the conventional P/A relation is valid for rectangular ducts not too far from square. His analysis indicates that other cases require more exact theory. However, it seems questionable whether the improvement in accuracy offered by the impedance method overbalances the additional computation time required and out-weighs other sources of error such as variations between samples of material.

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 4.

$$db \text{ (Attenuation)} = 10 \log_{10} \frac{\text{Plenum Absorption in Sabines}}{\text{Area Fan Discharge}} \tag{4}$$

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered rock wool blanket or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. With such a lining the plenum

Table 6. End Reflection of Plate Absorbers

Percentage Free Area of Absorber	Attenuation, db
50	1
40	2
30	4
25	5
20	6