sizes and shapes outside of the range specified. An empirically derived chart[7] representing the *average* experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 7 is shown in Fig. 3. Since individual materials vary, the curves in Fig. 3 are given only as representing the best available averages for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 7 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 3, as well as Equation 7, show that the attenuation in decibels is directly proportional to the length of duct lined, and that the larger the duct the greater will be the length which must be lined in order to obtain a given noise reduction.

If the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be sub-

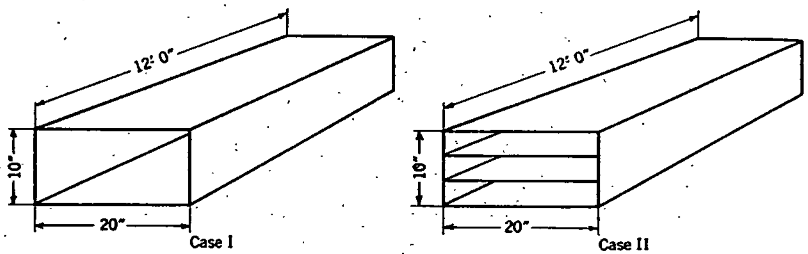

FIG. 4. DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

divided into smaller ducts, as shown in Fig. 4. The increase in noise reduction thus obtained may be calculated from Equation 8, providing the splitters are installed parallel to the long side of the duct:

$$R_s = R_o \frac{a + bn}{a + b} \tag{8}$$

where

R_s = reduction with splitters, decibels.
R_o = reduction in same length of duct, without splitters, decibels.
a = dimension of short side of duct, inches or feet.
b = dimension of long side of duct, inches or feet.
n = number of channels formed by splitters.

Example 1. An air conditioning installation is to be installed in a small theater Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theater utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in.	77 db
Acceptable room noise level (Table 1)	40 db
Required attenuation	37 db

[7]The Prediction of Noise Levels from Mechanical Equipment, by J. S. Parkinson (*Heating and Ventilating*, March, 1939, pp. 23-26). Methods of Rating the Noise from Air Conditioning Equipment, by J. S. Parkinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1940, p. 447).

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 2) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 3) 2 x 1	2.0 db
Attenuation grilles to theater air change 10 min (Table 5) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct lining or plate cells.

A similar analysis of the return duct system, shows that 15 db attenuation are to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

Example 2. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 7 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long as shown in Fig. 4, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters), From Equation 7,

$$R_o = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels), From Equation 8,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate[8]. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabines) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the preceding discussion it is presupposed that the noise level generated at the face of the grille is less than the noise level of the fan minus the attenuation of the duct system up to the face of the grille.

[8]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81).