the first band (20–75 cps) will be about 102 − 1 or 101 db, and the power level in each succeedingly higher octave band will be 5 db less. These levels are entered in line 1 of the calculation Table 10.

The next step is to determine the natural duct attenuation between the fan discharge and the room supply grilles. Assume that all of the ducts of this system have an attenuation of 0.05 db per ft at all frequencies. The total length of the duct run from Station *A* to Station *H*, Fig. 20, is 90 ft. The straight duct attenuation is, therefore, 0.05 (90) = 5 db, which is entered in line 2 of Table 10. It should be noted here that the ducts *CD* and *DH* of 12 x 48 and 12 x 15 in. dimensions, respectively, are of about the same size as those reported in Table 4. Hence if we assume the attenuation for these two *bare* ventilation ducts can be taken from Table 4, it will be found that the straight duct attenuation between *C* and *H* (35 ft) will be approximately 10 db in the lowest two octave bands, 6 db in the third band, and 3 db for all octave bands above 300 cps. That is, using Table 4 for ducts *CD* and *DH*, and assuming 0.05 db per ft for ducts *AB* and *BC*, the first three columns in line 2, Table 10, would read 12, 12, and 9 db in that order, and the other columns would be unchanged at 5 db. It would appear, therefore, that the 5 db attenuation values for the lower three octave bands are probably overly conservative. The divided-flow fitting at *B* may be considered equivalent in attenuation to an 18-in. square bend. Values are read from Fig. 13, Part A and entered as line 3 in the calculation table. The rounded corners of the branch flow fittings at *C*, *D*, *E*, *F*, and *G* are assumed to have no attenuation. The 12 x 15 in. risers serving each of the wall grilles have a square bend (see Fig. 21), and attenuation values are read from Fig. 13, Part A and entered as line 4 of Table 10. The attenuation due to the end reflection loss at the grille face is given by Fig. 15. For the 12 x 18 in. grilles used here $L_s = \sqrt{12 \times 18} = 14.7$ in. Thus, for the first octave band (20–75 cps), which has a geometric mean frequency of 40 cps, $fL_s/1000 = (40 \times 14.7)/1000 = 0.58$ and the end loss is 13 db.

The attenuation in the other octave bands is similarly determined and entered in line 5 of Table 10. The total natural duct attenuation is now found by summing lines 2, 3, 4, and 5, and the result appears in line 6. This total attenuation may now be subtracted from the unattenuated power level in line 1 to give the estimated acoustic power level radiated into the room from the outlet grille at Station *H*. Line 7 of Table 10 gives the result.

*Determining Room Sound Pressure Levels*

The next step, then, is to calculate the sound pressure level at the position in the room occupied by the listener nearest to any one grille, and for the overall reverberant sound pressure

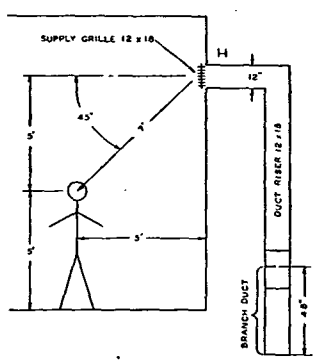


Fig. 21 . . . . Location of Nearest Listener to a Supply Grille

level in the room due to the sound power radiated from all of the grilles. Fig. 21 shows that the position of the nearest listener is 7 ft from the grille. To determine $L_p$ at this position, determine the directivity factor $Q$ at 45 deg to the grille face; this is found from Fig. 17. The parameter $fL_s/1000$ is determined by taking the characteristic length $L_s$ as the shortest side of the grille face or 12 in. Thus, for the first octave band, $fL_s/1000 = 40\ (12)/1000 = 0.48$. This value together with all the corresponding values for the other seven octave bands are entered in line 8 of Table 10. Line 9 of the table gives the corresponding directivity factors as read from Fig. 17, using curve *B* since the grille is flush with the wall but not at an edge or corner.

It was initially assumed that the library reading room had average acoustical properties, and consequently, the room constant $R$ in sq ft may be read from the line marked *average room* of Fig. 18. For a total room volume of 80,000 cu ft the result is $R = 2000$ sq ft. The relative sound pressure levels for an observer 7 ft from the grille may now be read from Fig. 19 making use of the directivity factors and the room constant as found. The dotted line in Fig. 19 shows that the relative $L_p$ for the 20–75 cps band is −22 db. This relative $L_p$ is entered in line 10 of Table 10. The sound pressure level at the 7-ft position from the grille is given by the addition of line 7 and line 10 in the calculation table. This result appears as line 11.

If there were only one grille in the room the calculation for room sound pressure level would be the only one required. However, since there are four grilles it is necessary to determine the general reverberant sound pressure level in the room due to the acoustic power radiated from these four sources to see if it is greater than the previous calculation. Neglecting the small additional duct attenuation which occurs in the somewhat longer duct runs to the other three room grilles, the total acoustic power radiated from the four grilles will be four times that from one grille. Hence the total acoustic power level from all grilles will be 10 log 4 or 6 db higher than from a single grille. In line 12 of Table 10, therefore, enter the $L_w$ of all grilles by adding 6 db to the single grille value of line 7. The relative $L_p$ for an observer in the reverberant sound field is read from the horizontal part of the $R = 2000$ sq ft curve of Fig. 19 as −26 db, and entered in line 13 of Table 10. The sum of line 12 and line 13 values gives, in line 14, the general reverberant sound pressure level in the room. Comparison of line 11 and line 14 shows that the latter is 2 db higher at the lower frequencies and, therefore, should be used in determining any noise treatment for the ventilating system or room.

*Room Criteria*

For this example the speech interference level criteria will be used. Examination of Table 1 shows that an acceptable noise level for libraries will be given by the *NC*-30 curve of Fig. 3. The values read from this curve are entered as line 15 of Table 10. The difference between these criterion sound pressure levels and the computed $L_p$ in line 14 of the table gives the required attenuation for the ventilating system. This final result appears as line 16 in Table 10.

*Treatment*

The type of treatment of the ventilating system to obtain the attenuation given in line 16 of the calculation table will depend on many factors as discussed in the section on Sound Attenuation. One satisfactory treatment might employ a 4-foot package unit whose attenuation characteristics, as given in Fig. 15, are entered in the final line 17 of Table 10. The additional 2 db of attenuation required in the second and third octave bands could be obtained by lining approximately 10 ft of the 12 x 48 inch duct with 1-inch liner. However, as was pointed out above, the straight duct attenuation in line 2 is probably too conservative in these lower octave bands, hence the package unit would most likely provide a satisfactory treatment.

A calculation is now made of the acoustic power level generated by the grille itself. The face velocity of the grille is 1500 cfm/1.5 sq ft = 1000 fpm. From Fig. 9, assuming a vertical or horizontal bar deflection type grille, the average power level in the speech interference bands (600–1200, 1200–2400, 2400–4800 cps) for a face velocity of 1000 fpm is 41 db per square foot of grille area or 41 + 10 log 1.5 = 43 db for the grille in this example.

From line 10 of the calculation table the relative $L_p$ in the



speech interference bands (600–4800 cps) is −20 db. Consequently the $L_p$ of the grille noise at 7 ft is 43 − 20 = 23 db. This is well below the 30 db criterion selected for this room and therefore, is satisfactory.

In reverberant fields all four grilles contribute so the total acoustic power level of the four grilles is 43 + 10 log 4 = 49 db. From line 13 of Table 10 the relative $L_p$ is −26 db and therefore the reverberant $L_p$ will be 49 − 26 or 23 db, which by coincidence is the same as that calculated at the 7-ft distance from one grille and, therefore, is also satisfactory.

The calculation of grille noise in this example was left to the end in order to keep a simple continuity to the principal problem of calculating the fan noise in the room. Experience shows that it is advantageous to calculate the grille noise early in the problem as indicated in the outline given in the text so that if the grille size must be changed its correct value may be used in calculating fan noise attenuation.

## CROSS TRANSMISSION BETWEEN ROOMS AND THROUGH DUCT WALLS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be subdivided to provide ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts, serves to prevent noise, originating outside the ducts, being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lagged, some of the mechanical noise from air in the equipment room may be transmitted through the wall of the duct into the air stream, and thereby carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material, such as cork, pipe covering, or other sound damping material, to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material, used in the case previously mentioned, would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays, and coils; noise due to eddying currents set up by elbows, dampers, and similar obstructions; and noise transmitted from room to room in a common duct system.

## CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise and, since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations, but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated, and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the supporting floor.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *airborne* vibrations known as sound, there is one basic formula which is important in the solution of the problem. It is the formula of transmissibility as governed by the equation:

$$T = \frac{1}{\left|\left(\frac{f}{f_n}\right)^2 - 1\right|} \qquad (26)$$

*where*

$T$ = transmissibility of the support.
$f$ = frequency of the vibratory force, cycles per second.
$f_n$ = natural frequency of the machine unit on its support (*damping* = 0), cycles per second.

Equation 26 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased, the transmissibility is also increased until at the resonant frequency, where $f = f_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency, the transmissibility becomes a smaller quantity, and at the value of $f/f_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation begins. At a ratio of 3 to 1 for $f$ to $f_n$ the isolation is effective enough for practical application, and experience and economical design have shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for $f$ to $f_n$ are easily attained and give better results for effective vibration control. For the lower speeds as experienced with compressor work the higher ratios become uneconomical. At these lower speeds the unbalanced force goes down as the square of the speed ratio, so that it is quite practical to compromise and use a lower ratio of $f$ to $f_n$, say 2 or 3 to 1.

It should be pointed out that Equation 26 is only reasonably accurate at frequencies near resonance, and even then it has been simplified by ignoring damping. It is well known that isolators seldom give an attenuation greater than 20 db or $T = 1/10$. Equation 26 should not be used to calculate $T$ because it does not take into account the facts that[24] (1) the machine itself is not a concentrated mass, (2) the normal isolator does not act as an ideal spring with a parallel dashpot, i.e., it will have standing wave resonances and damping of the