

of Fig. 19. 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. 20 making use of the directivity factors and the room constant as found. The dotted line in Fig. 20 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 9. 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 9, therefore, enter the PWL 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. 20 as -26 db, and entered in line 13 of Table 9. The sum of line 12 and line 13 values from Table 9 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 9. 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 9.

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 9. 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. 8, assuming a vertical or horizontal bar deflection-type grille, the average power level in the speech interference bands (600-4800 cps) is 41 dbw per square foot of grille area or 41 + 10 log 1.5 = 43 dbw 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 dbw. From line 13 of Table 9 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 eddy 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

Sound Control

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:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1} \quad (27)$$

where

T = transmissibility of the support.

f = frequency of the vibratory force.

f_n = natural frequency of the machine unit on its support (damping = 0).

Equation 27 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.

For a given installation, the speed of the compressor is fixed by the specifications; therefore the value of f is fixed. That leaves only f_n to be determined, and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame, and then the entire assembly isolated according to the rules presented in this chapter.

The value of f_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as with springs or nearly so with rubber-in-shear) the value of f_n can be determined by Equation 28:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{g}{d}} \quad (28)$$

where

g = gravitational constant = 32.2 feet per (second) (second).

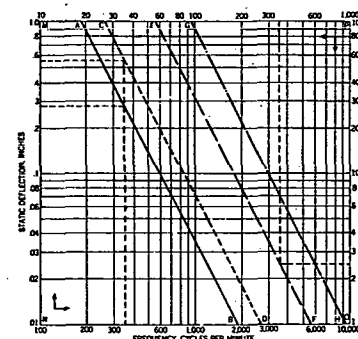


Fig. 23 . . . Static Deflection for Various Frequencies

d = static deflection of supporting material, feet.

f_n = natural frequency of the machine unit on its support (damping = 0), cycles per second.

By the use of Equation 28 a set of curves may be plotted as shown in Fig. 23. The first line AB , plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve CD , which is $\sqrt{2}$ times curve AB , shows the area $MCDN$ in which the resilient material or mounting does more harm than good. Plotting curves EF (3 times curve AB) and GH (5 times curve AB) shows area $EGHF$ which represents efficient and economical isolation. Area $GPOH$ is excellent isolation, but for all except the highest speeds, becomes rather uneconomical because of the large deflections required.

Example 4: An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm, and it is belt-connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum, and that 5 is desired. The desired natural frequency of the mounting is $360 \div 5 = 72$ cycles per minute.

From Fig. 23 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve AB for 72 cycles, or from curve GH (5 times critical) for 360 cycles. For a ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a foundation weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4500 lb
Concrete foundation	9000 lb
Total	13,500 lb

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported. Otherwise the