Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^{5} \times 1.86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.28 \times 10^{9})^2 + \frac{10^{20}}{4\pi^2 \times 100 \times 0.134}}{(0.28 + 10^{9})^2 + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{10}}{2\pi \times 100 \times 0.134}\right)^2}} = 0.93$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^{5} \times 645 = 0.97 \times 10^{7}$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.97 \times 10^{7})^2 + \frac{10^{18}}{4\pi^2 \times 100 \times 0.39}}{(0.97 \times 10^{7})^2 + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{9}}{2\pi \times 100 \times 0.39}\right)^2}} = 0.037$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the

TABLE 4. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^{5}
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^{5}
Flax-li-num	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^{5}
Celotex	Carpet lining	10	0.40×10^{-6}	....
Celotex	Insulating board	12	0.18×10^{-6}	....
Insulite	Insulating board	15	0.16×10^{-6}	....
Masonite	Insulating board	15	0.12×10^{-6}	....
Anti-Vibro-Block	....	5	0.60×10^{-6}	1.5×10^{5}
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	....
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	....
Hairfelt	10 lb per cubic foot	1 to 2	1.5×10^{-6}	....

[a]*Architectural Acoustics*, by V. O. Knudsen, p. 278.

transmissibility is reduced to 0.037, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

CONTROL OF NOISE TRANSMISSION THROUGH DUCTS

The most troublesome sources of noise from ventilating and air conditioning equipment are fan and motor noises which are transmitted through the ducts. The reduction, in decibels, of noise transmitted through a duct, neglecting reflection from ends and bends, is proportional (1) directly to the length of the duct, (2) directly to the perimeter of the duct, (3) inversely to the area of cross section of the duct, and (4) directly (or at least approximately so) to the coefficient of sound absorption of the material which comprises the interior surface of the duct. It is apparent therefore that long narrow ducts, lined with highly absorptive material, will provide a high degree of insulation against the transmission of noise through ducts. In fact, small ducts (4 in. x 6 in.), made of material having a coefficient of sound-absorption of 0.50, will provide a noise reduction of slightly more than 1 db per linear foot.

As can be seen from an inspection of Table 2, noises of low frequency are difficult to absorb; on the other hand, these frequencies are easily reflected by elbows, branches, and duct ends whereas higher frequencies are little affected. Furthermore, the reflection effects are more pronounced in small ducts than in large ducts. Hence, by introducing into a duct a sufficient length of small, absorptive channels together with a number of elbows or other reflecting elements it is possible to reduce the transmitted noise to any required degree. This applies not only to ducts between the equipment room and other rooms in a building, but also to ducts connecting adjacent or nearly adjacent rooms. By the proper use of such *filters* it is possible to eliminate all of the difficulties which arise in connection with the transmission of sound through ventilating ducts. The problem is an engineering one which can be worked out prior to the installing of the equipment, and it can be calculated in such a way as to meet the most rigorous demands for silent operation. There is a need for quantitative data regarding the attenuation or noise-reduction provided by different types of ducts, but even with the meager data available it is possible to design filters which will suppress the ordinary noises incident to the ventilating or air conditioning of buildings[4].

In general, the motion of air resulting from the ventilating of rooms is not sufficient to introduce any appreciable difficulty in auditoriums, except where noise may originate from the issuing of high-speed air from nozzles. However, by proper stream-lining of the nozzles, it is possible to work with speeds which are adequate for all practical purposes without producing any disturbing noises. Since sound is propagated with a velocity of more than 1100 fps, the velocity of the air would have to attain speeds

[4]How Sound is Controlled, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).