of accuracy. Table 4 gives the values of c and r for a number of commonly used flexible materials.

In general, there are two principal points to observe in the design of a flexible support for any piece of equipment, namely, the material should have a relatively large compliance and *it should be loaded to nearly the upper safe limit of loading*. Several flexible metallic supports have recently been developed.

Example 2. A machine weighing 1000 lb has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than ⅓ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}.)

The *transmissibility* is calculated in the following manner:

Mass of machine in grams $= 1000 \times 454 = 4.54 \times 10^{5}$.

Area of base in square centimeters $= 20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

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,

$$\tau^{1} = \sqrt{\frac{(0.28 \times 10^{9})^{2} + \frac{10^{20}}{4\pi^{2} \times 100 \times (0.134)^{2}}}{(0.28 + 10^{9})^{2} + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{10}}{2\pi \times 100 \times 0.134}\right)^{2}}} = 0.93$$

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

$$\tau^{1} = \sqrt{\frac{(0.97 \times 10^{7})^{2} + \frac{10^{18}}{4\pi^{2} \times 100 \times (0.39)^{2}}}{(0.97 \times 10^{7})^{2} + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{9}}{2\pi \times 100 \times 0.39}\right)^{2}}} = 0.037$$

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 *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.

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 x 10^{-6}	0.15 x 10^{5}
Corkboard	0.70 lb per board foot	8	0.50 x 10^{-6}	0.25 x 10^{5}
Fiber Board	1.35 lb per board foot	4 to 6	0.60 x 10^{-6}	0.50 x 10^{5}
Fiber Board	Carpet lining	10	0.40 x 10^{-6}	....
Fiber Board	Insulating board	12	0.18 x 10^{-6}	....
Fiber Board	Insulating board	15	0.16 x 10^{-6}	....
Fiber Board	Insulating board	15	0.12 x 10^{-6}	....
Anti-Vibro-Block	....	5	0.60 x 10^{-6}	1.5 x 10^{5}
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0 x 10^{-6}	....
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2 x 10^{-6}	....
Hairfelt	10 lb per cubic foot	1 to 2	1.5 x 10^{-6}	....

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

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