If r, the mechanical resistance, is very small, formula 1 may be written

$$\tau = \frac{1}{\dfrac{n^2}{n_0^2} - 1} \qquad (2)$$

where n_0 is the natural frequency of the machine upon the elastic pad,

$$n_0 = \frac{1}{2\pi}\sqrt{\frac{1}{mc}}.$$

In most cases of design of resilient machine mounting the effect of frictional resistance is small, and Equation 2 may be used. In such cases it is only necessary to know the natural frequency of the elastic pad or platform used under the desired loading and the transmissibility for any vibrational frequency of the machine may be obtained. However, this formula gives the theoretical maximum insulation which may be obtained and should be used with a liberal factor of safety. (A factor of 2 is common practice.)

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ' must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 1 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies

above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approxi-

TABLE 2. 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}
Fiber Board	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^{5}
Fiber Board	Carpet lining	10	0.40×10^{-6}	....
Fiber Board	Insulating board	12	0.18×10^{-6}	....
Fiber Board	Insulating board	15	0.16×10^{-6}	....
Fiber Board	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}	....

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