hysteresis type, and (3) the foundation cannot be infinitely stiff.

The true formula for transmissibility is

$$T = \frac{M_m + M_i}{M_m + M_i + M_b} \tag{27}$$

where

M = mobility, which is the ratio of the vibration velocity to the applied vibratory sinusoidal force. Subscripts m, b, and i refer respectively to the machine, base and isolator.

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:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{d_s}} \tag{28}$$

where

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

d_s = 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. 22. 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.

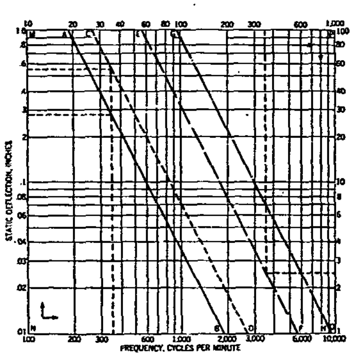

Fig. 22 . . . . Static Deflection for Various Frequencies

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 ÷ 5 = 72 cycles per minute.

From Fig. 22 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 foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor, a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 22, it is found that for 360 rpm the static deflection required for a ratio of f/f_n of 3 to 1 (line EF) is 2.5 in., and for a ratio of 5 to 1 (line GH) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should be designed with a *working height* equal to 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity, and therefore requires some additional means of preventing side drift of the unit, and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 0.80 in. For these conditions springs or rubber-in-shear can be used. Special rubber is required if the isolators are likely to be contaminated by oil. For speeds higher than 1200 rpm rubber-in-shear, as well as cork specially made for vibration isolation, can be applied with good results. These limitations are by no means absolute, because certain liberties may be taken without impairing the result if all possible degrees of freedom have been taken into account in the design of the installation.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer, it can be reduced in two ways only without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices, it is usually necessary to properly isolate the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

LETTER SYMBOLS USED IN CHAPTER 25

α = absorption coefficient of lining (a function of frequency), dimensionless.

$\bar{\alpha}$ = average sound absorption coefficient at mid-frequency of the band of noise being considered, dimensionless.

θ = angle between direction d and normal to the exit openings, degrees.

λ = wavelength of the tone, feet.

ΣN_i = sum of the loudnesses of all eight bands, sones.

ϕ = fan flow coefficient.

ψ = fan pressure coefficient (subscripts s or t refer to static or total pressure, respectively).

A = grille core area, square feet.

A_d = cross sectional area of duct inside the lining, square inches.

A_{min} = minimum flow area through diffuser, square feet.

A_o = fan outlet area, square feet.

c = speed of sound, feet per second.

D = duct dimension, inches.

D_w = fan wheel diameter, feet.

d = distance between entrance and exit of plenum, feet.

db = decibels.

E_{hp} = rated horsepower of the fan motor.

f = frequency, cycles per second.

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

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

I = sound intensity, watts per square centimeter.

I_m = sound intensity, watts per square meter.

L = length of one side of duct, inches.

L_e = equivalent length of side when duct is not square, inches.

L_I = sound intensity level, decibels.

L_N = loudness level, phons.

L_p = sound pressure level, decibels.

L_{SI} = speech interference level, decibels.

L_W = sound power level, decibels.

L_{W7} = overall sound power level in seven octave bands, decibels.

L_{WS} = overall specific sound power level, decibels.

L_{WSI} = sound power level in speech interference band, decibels.

l = length of lined duct, feet.

M = mobility, which is the ratio of the vibration velocity to the applied sinusoidal force. Subscripts m, b, and i refer respectively to the machine, base, and isolator.

M = number of corresponding decibels.

N = loudness, sones.

N_m = loudness of loudest band, sones.

P = perimeter of duct inside the lining, inches.

p = sound pressure root-mean-square incremental pressure, microbar. (A microbar is 1 dyne per square centimeter or 0.1 newton per square meter.)

p_s = static pressure, inches of water.

p_t = total pressure, inches of water.

Q = directivity factor (a dimensionless function of θ).

q = quantity of air discharged, cubic feet per minute.

R = attenuation, decibels.

R = room constant, square feet.

r = distance from duct opening, feet.

S = surface area of absorbing material, square feet.

S_e = plenum exit area, square feet.

S_w = plenum wall area, square feet.

T = transmissibility of support.

V_{max} = maximum air velocity, feet per second.

W = sound power, watts.

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