

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, but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

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 13.

$$f_n = \frac{1}{2\pi} \sqrt{\frac{g}{d}} \quad (13)$$

where

g = gravitational constant.

d = static deflection of supporting material.

f = frequency of the vibratory force.

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

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 7. 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 two more 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 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. 7 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 the minimum 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	4,500 lb
Concrete foundation	9,000 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. 7, it is

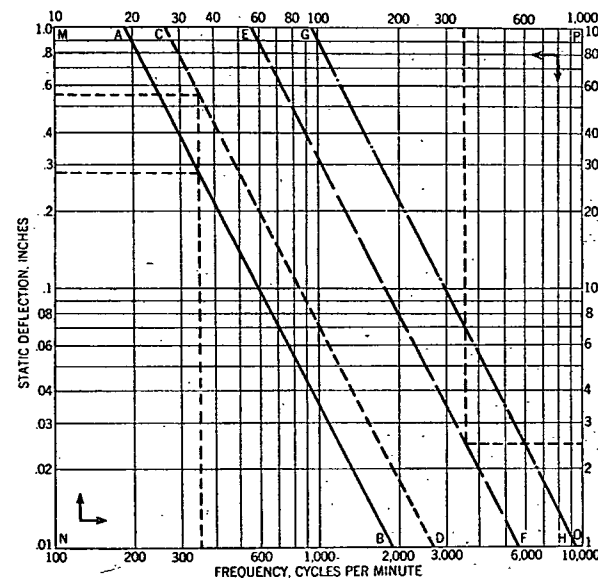


FIG. 7. STATIC DEFLECTION FOR VARIOUS FREQUENCIES

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 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork specially made for vibration damping can be applied with good results. These limitations are by no means absolute, because certain