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 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. 21 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.

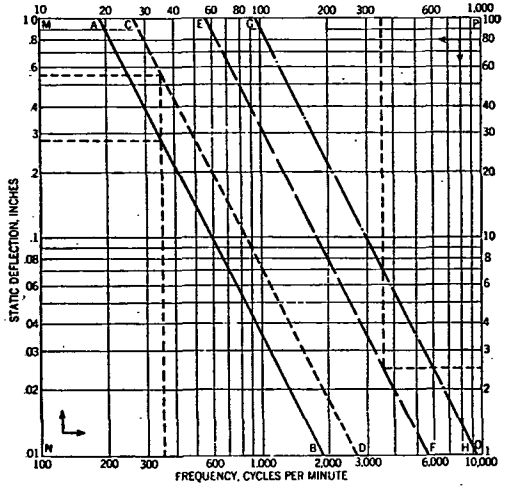


FIG. 21. STATIC DEFLECTION FOR VARIOUS FREQUENCIES

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. 21, 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 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 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 only in two ways 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 isolate properly 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.

REFERENCES

[1] *American Standard Acoustical Terminology, Z24.1—1951* (*American Standards Association*, 70 East 45th Street, New York 17, New York).

[2] *Handbook of Noise Measurement* (General Radio Company, Cambridge, Massachusetts).

[3] *The Measurement of Loudness*, by S. S. Stevens (*Journal Acoustical Society of America*, Vol. 27, September 1955, pp. 815–829).

[4] *American Standard Sound Level Meters for Measurement of Noise and other Sounds, Z24.3—1944* (*American Standards Association*).

[5] *Noise Criteria for Office Spaces*, by L. L. Beranek (*Journal Acoustical Society of America*, Vol. 28, Sept. 1956).

[6] Chapter on Fan Noise, by R. D. Madison and R. J. Wells, *Handbook of Noise Control*, Edited by C. M. Harris (McGraw-Hill Book Co., Inc., 1957).

[7] *Fan Engineering*, edited by R. D. Madison (Buffalo Forge Company, Buffalo, New York, 1949, pp. 313–314).

[8] ASHVE RESEARCH REPORT No. 1205—Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and F. R. Simons (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 267).

[9] The Absorption of Noise in Ventilating Ducts, by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

[10] For coefficients of commercial sound absorbent materials see Bulletin (*Acoustical Materials Association*, 59 East 55th Street, New York 22, N. Y.).