ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts, serves to prevent noise, originating outside the ducts, being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined, some of the mechanical noise from air in the equipment room may be transmitted through the wall of the duct into the air stream, and thereby carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material, such as cork, pipe covering or other sound damping material, to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material, used in the case previously mentioned, would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise and, since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations, but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated, and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the supporting floor.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *airborne* vibrations known as sound, there is one basic formula which is important in the solution of the problem.

It is the formula of transmissibility as governed by the equation:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1} \tag{27}$$

where T = transmissibility of the support.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (damping = 0).

Equation 27 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased, the transmissibility is also increased until at the reasonant frequency, where $f = f_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the reasonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency, the transmissibility becomes a smaller quantity, and at the value of $f/f_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation is first accomplished. At a ratio of 3 to 1 for f to f_n the isolation is effective enough for practical application, and 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 28:

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

where g = gravitational constant.
d = static deflection of supporting material.
f_n = natural frequency of the machine unit on its support (damping = 0).

By the use of Equation 28 a set of curves may be plotted as shown in Fig. 21. 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