Table 21 .... Typical Community and Traffic Activities Ratings Corresponding to the Background Noise Curves in Fig. 21

| Condition | Corresponding Curve in Fig. 21 |
|---|---|
| Nighttime rural; no nearby traffic of concern | 1 |
| Daytime rural; no nearby traffic of concern | 2 |
| Nighttime suburban; no nearby traffic of concern | 2 |
| Daytime suburban; no nearby traffic of concern | 3 |
| Nighttime urban; no nearby traffic of concern | 3 |
| Daytime urban; no nearby traffic of concern | 4 |
| 300 to 1000 ft from intermittent light traffic | 2 |
| 300 to 1000 ft from continuous light traffic | 3 |
| Within 300 ft of intermittent light traffic | 3 |
| Within 300 ft of continuous light traffic | 4 |
| 1000 to 2000 ft from continuous medium density traffic | 4 |
| 1000 to 2000 ft from continuous heavy traffic | 5 |
| 300 to 1000 ft from continuous medium density traffic | 5 |
| 300 to 1000 ft from continuous heavy traffic | 6 |
| Within 300 ft of continuous medium density traffic | 6 |
| Within 300 ft of continuous heavy traffic | 7 |
| Nighttime; business or commercial area | 4 |
| Daytime; business or commercial area | 5 |
| Nighttime; industrial or manufacturing area | 5 |
| Daytime; industrial or manufacturing area | 6 |

large equipment, it is recommended that large cooling towers be selected from sound pressure level ratings tabulated for various locations rather than from sound power ratings. Procedures and work sheets for this are available from Reference 69. These also allow for the shielding effect of large walls.

A comparison of the sound ratings of available equipment with the acceptable levels will indicate how much, if any, silencing treatment will be required under the prevailing conditions of distance and directivity. Considerable design information is available in References 69 and 70. Silencers for specific units are often available from the manufacturer.

Acceptable octave band sound pressure levels inside adjacent buildings can be estimated from Table 4. This introduces the additional variable of the *noise reduction* (NR) value of the building wall. Table 24 shows that the most important factor is the extent to which windows and vents are open. When the neighbor's building is not air conditioned, Row B of Table 24 should be used.

## STEP 6—ISOLATION OF MACHINE VIBRATIONS

The relatively high speeds of modern machinery and the relatively lightweight construction of modern buildings make it necessary to take the following precautions to prevent excessive vibration. Failure to observe any one of these will almost certainly cause objectionable noise and vibration, often in remote parts of the building, and may even cause fatigue failures.

1. Specify that machines shall be balanced, both statically and dynamically, within the limits of best commercial practice. Limits of balance which can be reasonably expected for various types of machines[11] are given in Table 25.
2. Evaluate the inherent quietness of various machine types. Centrifugal pumps and compressors generally run smoother than reciprocating ones, and compressors with four or more cylinders run smoother than one or two cylinder machines. Whenever dust or lint is a problem, fans should be of a design which minimizes the accumulation of foreign material that would create imbalance.
3. Specify that machines shall not have any critical speeds within ± 30 percent of any contemplated operating speed. Before changing the operating speed of a major machine, or of any machine that is running rough already, check for structural resonances and contact the manufacturer to make sure that the new speed will not be too close to any critical speed. Critical machine speeds are usually due to the natural frequency of shafts, framework, piping or fan blades. Critical speeds of multiple loaded shafts can be estimated easily from nomograms in Reference 71. Bear in mind that the critical shaft speed may be lowered by the flexibility of the bearing supports.
4. Design the supporting structure so that it does not have any natural frequencies within ± 30 percent of the operating speed of any one of the machines to be mounted on this structure. There is a distinct advantage in designing supporting structures simple enough so that their natural frequencies can be estimated from nomograms such as those in Reference 72. Foundations for heavy high-speed machines such as centrifugal compressors should be checked particularly carefully. Step-by-step design information is available in Reference 73.
5. Make sure that the machinery room is constructed massively with effective seals around all pipe and duct openings into adjacent rooms. This is necessary in order to reduce the transmission of airborne sound to other parts of the building.
6. Analyze the vibrational forces and motion for each machine installation in sufficient detail to insure that no excessive forces are transmitted to the building structure and no excessive strain imposed on connecting pipes, ducts, etc. It must be understood that no matter how rigidly a machine seems to be supported, there is a certain amount of flexibility in either the supporting structure or the framework of the machine itself. If the combination of this flexibility, the mass of the machine, and its speed approaches a certain critical value, violent vibration will result even if the machine is most carefully balanced. This condition is called resonance and is bound to cause trouble and often premature failure. It is absolutely necessary and always possible to avoid resonance, either by making the flexibility of the *entire* supporting structure *much lower* than the flexibility which produces resonance, or by using machine mounts which have a *much higher* flexibility.

For the reciprocating machines, the analysis for resonant condition has to include low-order multiples of the machine speed because there are secondary forces and couples which cannot be completely balanced out. For all electrical equipment, the analysis must include the frequency of magnetic distortion which occurs at twice the line frequency (120 cps).

It must also be borne in mind that the centrifugal force due to imbalance produces not only vertical, but also horizontal vibration at right angles to the shaft. Unbalanced couples, or the customary location of vibration mounts well below the center of gravity of equipment, cause vibration in axial, rolling, rocking and twisting modes. Thus, there are always 6 possible resonant frequencies which have to be avoided.

Table 22 .... Decibel Difference Between Power Level of Outdoor Equipment Noise and Corresponding Sound Pressure Level at Any Distance, $r$[a]

| Distance, $r$, in Feet | 10 | 15 | 20 | 30 | 40 | 60 | 80 | 120 | 200 | 500 | 1000[b] |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $(L_W - L_p)$ for $Q = 2$ | 18 | 21 | 24 | 27 | 30 | 33 | 36 | 39 | 43 | 51 | 57–62 |
| $(L_W - L_p)$ for $Q = 4$ | 15 | 18 | 21 | 24 | 27 | 30 | 33 | 36 | 40 | 48 | 54–59 |
| $(L_W - L_p)$ for $Q = 8$ | 12 | 15 | 18 | 21 | 24 | 27 | 30 | 33 | 37 | 45 | 51–56 |

[a] Table does not apply when $r$ is less than twice the maximum dimension of the equipment. Values may be up to 5 db low for distances between 2 and 5 times maximum equipment dimension.
[b] At distances above 1000 ft, air absorption and atmospheric conditions become important.

Table 23 .... Approximate Power Levels[a] for Cooling Tower Noise, in db re $10^{-12}$ Watt[67]

| Cooling Tower Fans | | Octave Band Center, cps | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Type | Horsepower | 53 63 | 106 125 | 212 250 | 425 500 | 850 1000 | 1700 2000 | 3400 4000 | 6900 8000 |
| Induced | 10 hp[b] | 97–102 | 96–101 | 94–99 | 91–96 | 87–92 | 83–89 | 80–87 | 73–83 |
| Draft | 20 hp[b] | 100–105 | 99–104 | 97–102 | 94– 99 | 90– 95 | 86–92 | 83–90 | 76–86 |
| Propeller | 50 hp[b] | 104–109 | 103–108 | 101–106 | 98–103 | 94– 99 | 90–96 | 87–94 | 80–90 |
| | 100 hp[b] | 107–112 | 106–111 | 104–109 | 101–106 | 97–102 | 93–99 | 90–97 | 83–93 |
| Forced | 20 hp | 90– 95 | 89– 94 | 87– 92 | 85– 90 | 83– 88 | 83–88 | 79–84 | 76–81 |
| Draft | 50 hp | 94– 99 | 93– 98 | 91– 96 | 89– 94 | 87– 92 | 87–92 | 83–88 | 80–85 |
| Centrifugal | 100 hp | 97–102 | 96–101 | 94– 99 | 92– 97 | 90– 95 | 90–95 | 86–91 | 83–88 |

[a] These power levels do not provide information on the radiation pattern (directivity) of the sound. A safety factor of up to 15 db should be applied, especially in the higher frequency bands when the air discharge or intake openings face a critical location. Final selections should always be based on manufacturer's ratings for the particular orientation of the cooling tower.
[b] For propeller fans add 5 db to listed level in band in which the blade frequency occurs. Blade frequency = $(RPM/60) \times$ number of blades.



Table 24 .... Approximate Noise Reduction Provided by Typical Wall Constructions, in db

| Description | Octave Band Center, cps | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 53 63 | 106 125 | 212 250 | 425 500 | 850 1000 | 1700 2000 | 3400 4000 | 6900 8000 |
| A—No wall; outside conditions | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| B—Any typical wall construction with windows normally open | 10 | 10 | 12 | 14 | 16 | 18 | 18 | 18 |
| C—Any typical wall construction, windows closed but small air vents normally open | 15 | 15 | 17 | 19 | 21 | 23 | 23 | 23 |
| D—Any typical wall construction, with no cracks or openings, windows normally closed | 20 | 20 | 23 | 26 | 29 | 32 | 32 | 32 |
| E—Approximately 20 lb/sq ft solid wall, no windows, cracks or openings | 26 | 27 | 31 | 35 | 39 | 43 | 45 | 47 |
| F—Approximately 50 lb/sq ft solid wall, no windows, cracks or openings | 32 | 33 | 37 | 41 | 45 | 49 | 51 | 53 |

## Calculation of Vertical Natural Frequency

Vibration in a vertical direction is generally most serious because of the low rigidity of common floor structures in the vertical direction. The vertical forces acting on a machine are shown schematically in Fig. 22. The symbol of a spring is used between the machine and the foundation to show that every supporting structure has a certain amount of flexibility. When the machine is rigidly mounted on a heavy concrete pad resting on soil, this flexibility may be relatively small, but not small enough to be negligible. Data on soil compressibility can be found in Reference 73.

Table 25 .... Typical Machinery Balancing Specifications.

| Source | Applicability | Balancing Limits and Parameters | | | | |
|---|---|---|---|---|---|---|
| 1. Mil-Std 167 (Ships) | Rotors for Naval Machinery | 700 rpm | 950 rpm | 1140 rpm | 1750 rpm | 3450 rpm |
| | | $1.1 \frac{\text{oz-in.}}{100\text{ lb}}$ | $0.62 \frac{\text{oz-in.}}{100\text{ lb}}$ | $0.35 \frac{\text{oz-in.}}{100\text{ lb}}$ | $0.23 \frac{\text{oz-in.}}{100\text{ lb}}$ | $0.12 \frac{\text{oz-in.}}{100\text{ lb}}$ |
| 2. Tool Engineers Handbook[14] (Chapter on Balancing Machines, by W. L. Senger, p. 84). | Rotors for any type machine with sufficiently rigid structure | 500 rpm | 1000 rpm | 5000 rpm | | |
| | | 0.0005 to 0.001 in. Displacement of freely supported bearings | 0.0005–0.001<br>0.0002 to 0.0005 in. | 0.0005–0.001<br>0.0002–0.0005<br>0.0001–0.0002 | 5%<br>10%<br>25% up | Rotor weight in percent of machine weight |
| Sources 1 and 2 | Rotors for any type machine | Dynamic (2-plane) balancing required if RPM $\geq$ 1000 or length/diameter ratio of rotor $\geq \frac{1}{2}$ | | | | |
| 3. NEMA Standard MG 1-4.23 Rev. 11-15-56 | Completely assembled electric motors elastically supported for test | *Frame Diameter Series*<br>180, 200, 210 and 220<br>250, 280 and 320<br>360, 400, 440 and 500 | | *Max. Peak-Peak Displacement*<br>0.001 in.<br>0.0015 in.<br>0.002 in. | | |

*Notes*: Limits of Mil-Std 167, although expressed in different terms, are roughly equivalent to those from Table 84-7 of Tool Engineers' Handbook, for rotors weighing 25 percent of machine weight.
NEMA MG 1-4.23 allows unbalances far in excess of those listed in the other two sources, especially for 2-pole motors. Better balance can usually be obtained at extra cost.