CHAPTER 45
NOISE AND ITS CONTROL

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45.1 INTRODUCTION

Noise is unwanted audible sound. Sound is essentially a fluctuating pressure disturbance that may act locally or propagate away from its source.

Extremely intense sound can cause structural damage or equipment malfunctions, but usually its effect on humans is the primary concern. Noise can annoy people, can lead to interference with speech communication, can interfere with the performance of mental and delicate manual tasks, and—if it is intense enough—can cause discomfort, pain, and temporary or permanent hearing damage. Fortunately, many aspects of noise and its effects are well enough understood to permit their consideration in the design process.

Sound can occur and propagate in any gas, liquid, or solid medium, but sound in air is usually of primary interest. Sound may be produced by any phenomenon that can lead to fluctuating pressure disturbances. These phenomena include (1) rapid expansion of gases or injection of fluid volumes, such as from explosions and engine exhausts; (2) repetitive interruptions or modulation of airflows, such as by siren disks or fluctuating valves; (3) turbulence, as present in fluid streams emerging from nozzles or duct grillages; and (4) vibrating solid surfaces. In many practical situations, several noise-generating phenomena may occur simultaneously; for example, an impact press may generate noise not only because of the structural vibrations it produces but also because of the air it expels from between the impacting surfaces.

Sound, being a pressure disturbance, can propagate in the medium in which it is generated. This propagation need not involve flow or net displacement of the medium; only the disturbance and the energy associated with it move away from the source.

Pressure fluctuations in air can induce fluctuations in other media in contact with the air, and vice versa. Therefore, often sound from a given source reaches an observer not only via a direct air path but also via paths that may involve several media. For example, sound radiated from vibrating gears in a housing may propagate from the air in the housing through an oil layer and through the housing wall into the ambient air. In many practical situations, several parallel paths of sound transmission from a given source to a given observer—including some relatively tortuous paths along complex structures—may be similarly important.

It usually is convenient to consider a noise problem from the "source-path-receiver" viewpoint. This approach facilitates accounting for all significant sources (noise generators), receivers (items or persons affected by noise), and paths along which the noise from the sources reaches the receivers. This approach thus encourages evaluation of all relevant facets of the problem.

The remainder of this chapter introduces noise measurement and analysis, noise effects and standards, and noise control techniques relevant to machine design. For treatment of these subjects in greater depth, texts and handbooks on acoustics should be consulted (for example, Refs. [45.1] through [45.5]), as well as the specific references given throughout this chapter.

45.2 NOISE MEASUREMENT AND ANALYSIS

45.2.1 Noise Measures

Sound or noise can be sensed by measurement of *sound pressure*, the variation in air pressure above and below its equilibrium value. The measure most commonly used is the root-mean-square (rms) sound pressure p_{rms}. The rms sound pressure is obtained by squaring the value of the sound pressure disturbance at each instant of time, averaging the squared values over the sample time, and taking the square root of the result.

Because the range of sound pressure amplitude variations that the human ear can detect extends over several factors of 10, a compressed scale based on the logarithm of the mean square pressure is used. The decibel, abbreviated dB, is a measure of this scale. The corresponding noise descriptor is called the *sound pressure level* L_p, defined as

$$L_p = 10 \log \left(\frac{p_{\text{rms}}}{p_0} \right)^2 \quad \text{dB} \tag{45.1}$$

where p_0 is a reference pressure, standardized as 20 micropascals (μPa) [2.90×10^{-9} pounds per square inch (lb/in^2)]. This very small reference pressure corresponds to 0 dB and represents approximately the weakest sound that can be heard by an average young, alert person with an undamaged hearing mechanism.

Since decibels are logarithmic measures, sound pressure levels cannot be added by ordinary arithmetic. The sound pressure level L_p (total) corresponding to the combination of n sound pressure levels $L_p(i)$ is calculated from†

$$L_p\,(\text{total}) = 10 \log \left(\sum_{i=1}^{n} 10^{L_p(i)/10} \right) \tag{45.2}$$

To describe noise adequately, one must measure not only its *amplitude,* which determines the magnitude of the pressure, but also its *frequency,* which determines its pitch. In any sound, the air pressure alternately rises and falls; for repetitive sounds, each time the pressure rises from its minimum value and returns to that value, it completes one cycle. The number of cycles occurring per second is called the *frequency* of the sound; the unit of cycles per second is hertz (Hz). Frequency is observed subjectively as the tone, or pitch, of a sound. The low frequencies (20 to 500 Hz) have a low-pitch, or bass, sound. The midfrequency range, from about 500 to 3000 Hz, is where most speech information is carried. High frequencies, from about 3000 to 20 000 Hz, tend to be prevalent in whistles, jets, and high-speed machines.

The *wavelength of a sound wave* is defined as the distance the wave travels in a stationary medium during one cycle. Wavelength and frequency are related by

$$\lambda = \frac{c}{f} \tag{45.3}$$

where c = speed of sound, ft/s (m/s)
f = frequency, Hz
λ = wavelength, ft (m).

The speed of sound in gases depends on the temperature, but not on pressure. At 70°F (21°C), for example, the speed of sound in air is 1128 ft/s (344 m/s), and the wavelength of a 1000-Hz sound wave is 1.128 ft (0.3438 m).

The basic properties of a pure-tone (that is, single-frequency) sound wave are summarized in Fig. 45.1. This figure illustrates a time-history graph of the amplitude of a sound. Note that for this sinusoidal wave, the sound pressure amplitude rises from zero to a positive maximum, then falls through zero to a negative maximum, and then returns to zero during one complete cycle. For this type of wave, the rms value is 0.707 times the absolute value of the peak (positive or negative) amplitude.

Noise from common sources, such as machinery, is usually more complex than the pure tone illustrated in Fig. 45.1. In general, noise consists of a combination of many sinusoidal components, all with different frequencies. Description of such noise requires a noise *spectrum,* which is a graph of sound pressure level versus frequency. Frequency analysis (or spectrum analysis) is essential for any comprehensive study of a noise problem for three reasons: (1) people have different hearing sensitivity and different reactions to the various frequency ranges of noise, (2) different noise sources emit differing amounts of noise at different frequencies, and (3) engineering solutions for reducing or controlling noise are different for low- and high-frequency noise.

Although a noise spectrum is useful for purposes of analysis, it is often convenient to use a single-number measure to describe a noise. The most commonly used measure of this type is the *A-weighted sound level,* expressed in units of dBA. From

† This corresponds to p^2_{rms} (total) $= \sum_{i=1}^{n} p^2_{\text{rms}}(i)$, where the individual signals are at different frequencies and/or are uncorrelated.

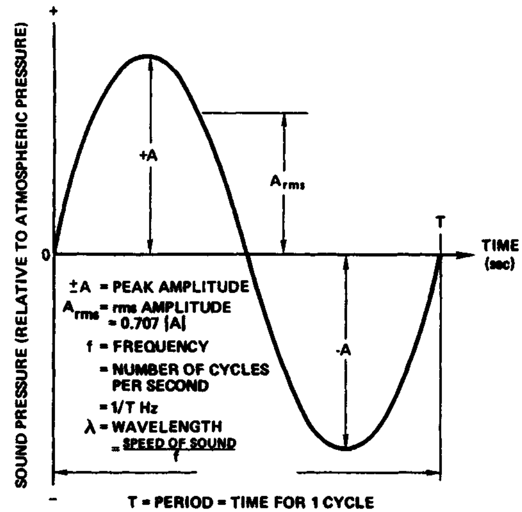

FIGURE 45.1 Basic properties of a sinusoidal (pure-tone) sound wave.

many experiments with human listeners, it was found that human hearing is more sensitive to midrange frequencies than to either low or very high frequencies. This characteristic is taken into account by adjusting, or *weighting*, the various frequency components of a sound in accordance with the sensitivity of human hearing and then combining all the weighted components. The result is a single-number measure of sound level that corresponds approximately to the human subjective perception of the severity of the noise, as well as to its annoyance and hearing damage potential.

Table 45.1 compares representative noise levels for common indoor and outdoor noise sources and environments. The extremes of noise range from 0 dBA (approximate threshold of hearing) to 120 dBA (jet aircraft at 500 ft), although most commonly encountered noise levels fall within the 40- to 100-dBA range.

An understanding of the following subjective perceptions of changes in the A-weighted sound level is useful:

- Changes of 1 dB or less cannot be perceived, except in carefully controlled laboratory experiments.
- A 3-dB increase in A-weighted level generally is just noticeable.
- A 10-dB increase in A-weighted level is perceived as approximately a doubling in loudness, independent of the initial noise level.

All the discussion thus far has been related to sound pressure, since this is the property to which human hearing and microphones respond. However, as discussed later, the magnitude of sound pressure level resulting at a given location and due to a given source depends on the "strength" of the source, on the environment in which the noise source is located, on the distance of the observation location from the source, and sometimes on the direction. Therefore, it is useful in many cases to use a noise measure that describes the intrinsic strength of a given source, that is, its *sound power*. Sound power represents the total sound energy radiated by a source per unit of time and is proportional to the square of the sound pressure at any given location.

TABLE 45.1 Comparison of Various Noise Levels

NOISE LEVEL (dBA)	EXTREMES	HOME APPLIANCES IN ROOMS	SPEECH AT 3 ft	MOTOR VEHICLES AT 50 ft	GENERAL TYPE OF OUTDOOR COMMUNITY ENVIRONMENT	GENERAL TYPE OF INDOOR WORK ENVIRONMENT
120	JET AIRCRAFT AT 500 ft					
110						
100						
90–100				DIESEL TRUCK (NOT MUFFLED)		HEAVY INDUSTRY
80–90		SHOP TOOLS	SHOUT	DIESEL TRUCK (MUFFLED)		
70–80		BLENDER	LOUD VOICE	AUTOMOBILE AT 70 mph	MAJOR METROPOLIS (DAYTIME)	LIGHT INDUSTRY
60–70		DISHWASHER	NORMAL VOICE	AUTOMOBILE AT 40 mph	URBAN (DAYTIME)	
50–60		AIR CONDITIONER	NORMAL VOICE (BACK TO LISTENER)	AUTOMOBILE AT 20 mph	SUBURBAN (DAYTIME)	OFFICE
40–50		REFRIGERATOR			RURAL (DAYTIME)	
30						
20						
10						
0	THRESHOLD OF HEARING					

As is the case for sound pressure, the range of sound power encountered in acoustics is very large. Thus, a logarithmic (decibel) scale is also used to describe sound power. The *sound power level* L_w is defined as

$$L_w = 10 \log \frac{W}{W_0} \tag{45.4}$$

where W = source sound power in watts (W) and W_0 = reference sound power, standardized as 10^{-12} W. Sound power level is typically expressed in terms of dB with respect to 10^{-12} W.

45.2.2 Sound Fields

Meaningful measurements must take into account the variation of sound pressure level with position in the vicinity of a noise source. Figure 45.2 illustrates this general relationship and indicates the various sound field regions.

For an ideal nondirectional "point source" in open space, the sound pressure level decreases at the rate of 6 dB per doubling of distance because of spherical spreading of the sound energy. This relation is usually called the *inverse-square law,* because it corresponds to the sound pressure's varying inversely as the square of distance. However, the point-source approximation breaks down at distances very close to the source. At such distances, sound variation is more complex; in this *near field,* the sound pressure level may be either more or less than predicted by the inverse-square law, as shown in Fig. 45.2. The extent of the near field depends on the

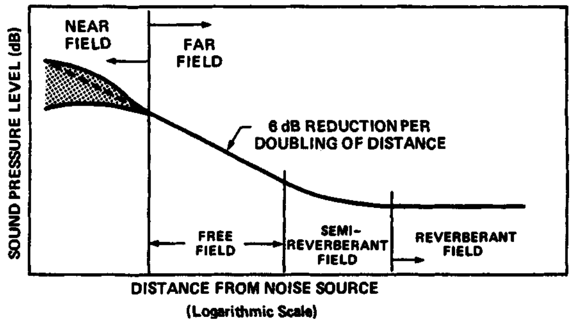

FIGURE 45.2 Sound fields in the vicinity of a noise source.

frequency of the sound, the dimensions of the source, and the phase relations of the various radiating parts of the source. As a rule of thumb, the near field may be assumed to end at a distance about twice the largest dimension of the source (or at 4 times the largest dimension, for sources resting on an acoustically reflective floor). Note that sound pressure levels measured within the near field cannot be used to predict the sound pressure levels at other distances or to evaluate the source sound power level; for these purposes, one must take care to perform measurements in the acoustic *far field* (that is, at distances beyond the near field).

In the acoustic far field, sound pressure levels decrease at a rate of 6 dB per distance doubling, as long as there exists a *free field*, which is, for all practical purposes, a field in which the effects of any air volume boundaries are negligible. Such a free field can be obtained outdoors, in a large room at locations away from the walls, or in an *anechoic chamber.* (In the latter, the walls, floors, and ceiling absorb nearly all sound incident on them.) The extent of the free-field region is characterized in Fig. 45.2 by a line with constant slope.

Sound from a source in any room—but most pronouncedly in a small room with "hard" (i.e., acoustically nonabsorptive) wall, floor, and ceiling surfaces—is reflected many times, so that the total sound at any location is composed of the sound radiated directly from the source (free-field sound) plus all the reflected components. If many reflected sound waves are arriving at an observation point from all directions, the sound field is called *reverberant.* In the reverberant field, the sound pressure level decreases less rapidly with distance than indicated by the inverse-square law, as shown in Fig. 45.2. *Reverberant rooms,* in which sound is uniform throughout, are often used to perform sound measurements that, in effect, average over all directions (for example, for the purpose of evaluating sound power levels of sources).

In practice, noise measurements often must be made in *semireverberant fields,* that is, where the sound propagation characteristics lie somewhere between free-field and reverberant conditions, as indicated by the transition zone in Fig. 45.2. The characteristics of a semireverberant environment are controlled largely by the amount of sound absorption in the room. These characteristics generally need to be evaluated and taken into account in analysis of the measured results.

45.2.3 Measurement Instrumentation

Sound Level Meters. A sound level meter consists of (1) a transducer (microphone) to convert air pressure fluctuations to an electric signal, (2) an amplifier to raise the electric signal to a usable level, (3) weighting networks to modify the frequency characteristics of the instrument's response, and (4) an indicating device (meter) to display the measured level.

Sound level meters are designated by class, depending on measurement accuracy and tolerances. International Electrotechnical Commission (IEC) standard IEC 651 defines four classes: type 0, laboratory reference; type 1, precision; type 2, general purpose; and type 3, survey. Type 0 sets the most stringent accuracy and tolerance limits, followed by types 1, 2, and 3. Type 1 meters provide sufficient accuracy for field measurements in most cases and are usually selected when cost is not a major consideration. Standards for types 0, 1, and 2 sound level meters are also provided in American National Standards Institute (ANSI) standard S1.4-1983.

The weighting network most commonly used in sound level meters is the *A-weighting* network. Its response, shown in Fig. 45.3, represents the average behavior of human hearing. Measurements made using this network are expressed in A-weighted decibels, abbreviated dBA. Other common weighting networks include the B, C, and D types, used for special purposes. Some sound level meters also include a "linear," or "flat," response, commonly employed when a sound level meter supplies an electrical signal to other instruments.

The indicating meter on a sound level meter displays the sound level in decibels, relative to a standard reference sound pressure (20 μPa, or 2.90×10^{-9} lb/in^2). The speed with which the meter electronics and indicator respond also has been standardized. Most meters include two choices for averaging time: fast, which has a time constant of about ⅛ s, and slow, which has a time constant of about 1 s. The slow response is particularly useful for estimating visually the average value of a sound that fluctuates rapidly. Some sound level meters also have peak-hold and impulse-hold features, which are useful for measuring unsteady or impulsive noises.

Microphones. A microphone is a transducer used to convert air pressure fluctuations to an electric signal. Of the different types currently available, the most commonly used are the condenser, electret, and piezoelectric types. The choice of a particular microphone depends on its intended application and required perfor-

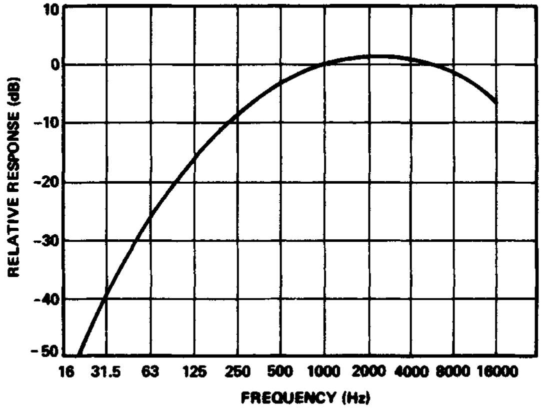

FIGURE 45.3 Frequency response specified for the A-weighting filter of sound level meters *(From ANSI S1.4-1983.)*

mance in terms of stability, precision, directivity, and frequency-response characteristics. Condenser microphones have excellent long-term stability and are insensitive to changes in temperature. However, they are sensitive to moisture. Electret microphones vary considerably in their long-term stability and sensitivity to temperature, and so are not as well suited as condenser microphones to measurement environments with large temperature variations. However, they are less sensitive to moisture. Piezoelectric microphones are generally more rugged than condenser or electret microphones.

Acoustical Calibrators. An acoustical calibrator is a device that produces a known, stable sound pressure level at the diaphragm of a microphone. The most common calibrators are the pistonphone and loudspeaker.

A pistonphone calibrator produces a known sound pressure level within a closed cavity by means of moving pistons. Calibration is usually restricted to a single frequency (typically 250 Hz), and corrections for atmospheric pressure must be applied.

Loudspeaker-type calibrators consist of a battery-operated oscillator and small loudspeaker. In contrast to the pistonphone, some loudspeaker-type calibrators operate over a wide frequency range (125 to 2000 Hz), and the sound pressure level developed is less sensitive to the atmospheric pressure.

Spectrum Analyzers. A *spectrum analyzer* essentially produces a plot of sound pressure level versus frequency. Spectrum analyzers employ electronic filters to separate the frequency components of a sound signal. The range of frequencies covered by an individual filter is called its *bandwidth.* Two basic types of filter sets are used in spectrum analyzers: those that use bands of constant bandwidth (that is, a fixed number of hertz) and those that use bands in which the upper frequency limit of the band is a fixed multiple of the lower frequency limit. Of the latter type, the bandwidth most commonly used in acoustic analysis covers a frequency range of one octave (that is, a 2-to-1 frequency range); an analyzer having filters with this bandwidth is called an *octave-band analyzer.* Other analyzers use half octaves ($\sqrt{2}$-to-1 frequency range), one-third octaves ($\sqrt[3]{2}$-to-1 range), or even narrower bands. Narrowband filters are often required to determine pure-tone components, such as those resulting from operation of cyclic (reciprocating or rotating) machinery. For narrowband analysis, digital computer-aided real-time analyzers are widely used.

The preferred center frequencies and band limits for spectrum analyzer filters are given in ANSI standard S1.6-1984 and in International Organization for Standardization (ISO) standard 266-1975. Values for octave- and one-third-octave-band filters covering the audio frequency range are given in Table 45.2. Filters that are incorporated in octave-, half-octave-, and one-third-octave-band analyzers have been standardized by ANSI (standard S1.11-1966) and by the IEC (standard 225-1966).

45.2.4 Measurement Procedures

Once the purpose and required accuracy of a measurement are defined, one must select the proper measurement, recording, and analysis equipment.

Microphone positions should be selected to yield a useful sample of the sound field in the area of interest, and the microphone orientations should be chosen on the basis of the frequency-response characteristics of the microphone and of the measurement environment (see microphone manufacturer's instructions). For outdoor measurements or for other locations where the air is not calm, the microphone

TABLE 45.2 Center and Approximate Cutoff Frequencies for Octave and One-Third-Octave Frequency Bands Covering the Audio-Frequency Range

Octave, Hz			One-third octave, Hz		
Lower band limit	Center frequency	Upper band limit	Lower band limit	Center frequency	Upper band limit
11	16	22	14.1	16	17.8
			17.8	20	22.4
			22.4	25	28.2
22	31.5	44	28.2	31.5	35.5
			35.5	40	44.7
			44.7	50	56.2
44	63	88	56.2	63	70.8
			70.8	80	89.1
			89.1	100	112
88	125	177	112	125	141
			141	160	178
			178	200	224
177	250	355	224	250	282
			282	315	355
			355	400	447
355	500	710	447	500	562
			562	630	708
			708	800	891
710	1 000	1 420	891	1 000	1 122
			1 122	1 250	1 413
			1 413	1 600	1 778
1 420	2 000	2 840	1 778	2 000	2 239
			2 239	2 500	2 818
			2 818	3 150	3 548
2 840	4 000	5 680	3 548	4 000	4 467
			4 467	5 000	5 623
			5 623	6 300	7 079
5 680	8 000	11 360	7 079	8 000	8 913
			8 913	10 000	11 220
			11 220	12 500	14 130
11 360	16 000	22 720	14 130	16 000	17 780
			17 780	20 000	22 390

SOURCE: ANSI standard S1.6-1984.

should be fitted with a windscreen to avoid extraneous noise generated by air turbulence at the microphone.

Before each set of measurements is made, all equipment should be calibrated according to the manufacturer's instructions. It is also a good idea to measure the electric noise floor (the lower measurement limit) of the instrumentation by replacing the microphone with an equivalent electric impedance (such as a capacitor) or by shielding the microphone from the acoustic background noise.

It is good practice to monitor the output of the sound level meter during the measurements by listening with the aid of a high-quality set of headphones; this permits one to detect electromagnetic pickup, signals due to wind or humidity, or other interference.

For source noise measurements, it is desirable to measure the background noise level (by turning off the noise source) to determine whether the background noise has a significant effect on the measurements. The background noise level should be at least 10 dB below the source noise level, if it is not to affect measured results significantly; otherwise, the measured noise levels must be corrected to obtain the level of the source. Table 45.3 may be used to obtain the appropriate correction.

At the conclusion of each set of measurements, the proper operation and calibration of all equipment should be rechecked, and all pertinent data should be recorded.

45.2.5 Data Evaluation

A set of measured acoustic data usually must be evaluated with regard to the problem of interest. This evaluation often requires conversion or extrapolation of the results. For example, sound pressure level measurements obtained for a machine in an anechoic chamber may need to be used to estimate the sound pressure level of the same machine at a different distance inside an industrial building. Or, one may want to use sound power level data acquired for a noise source in a reverberant room to estimate the sound pressure level at a given distance from the same source located outdoors. Such evaluations may be based on the relation between sound pressure and sound power level, as described below.

For any sound source, the sound pressure level and sound power level are related by

$$L_p = L_w + 10 \log \left(\frac{Q}{4\pi r^2} + \frac{4}{R} \right) + 10.5 \tag{45.5}$$

where
L_p = sound pressure level, dB re 20 μPa
L_w = sound power level, dB re 10^{-12} W
Q = directivity factor (dimensionless)
r = distance to observation point from acoustic center of source, ft
R = room constant, ft^2

In mks units, this equation converts to

$$L_p = L_w + 10 \log \left(\frac{Q}{4\pi r^2} + \frac{4}{R} \right) \tag{45.6}$$

where r is in meters and R is in square meters.

TABLE 45.3 Correction Factors for Background Noise

Difference between total noise level and background noise level, dB	Correction to be subtracted from total noise level to obtain source noise level, dB
8–10	0.5
6–8	1.0
4.5–6	1.5
4–4.5	2.0
3.5	2.5
3	3.0

The directivity factor Q accounts for the fact that most practical noise sources do not radiate uniformly in all directions. In the case of a nondirectional source (radiating sound uniformly in all directions), $Q = 1$. However, for a source placed on a sound-reflecting surface (for example, a machine on a concrete floor), much of, and sometimes all, the sound that would have been directed downward is reflected upward; here $Q = 2$ for a uniformly radiating source. Similarly, for noise sources located along the edge of a room and at the corner of a room, Q is 4 and 8, respectively.

The acoustic center is the location that would be occupied by a "point source" with the same sound power output as the actual source. For most practical purposes, the acoustic center can be taken as the geometric center of the controlling noise-radiating mechanism. Except for distances r very close to the noise source, errors in the estimation of the location of the acoustic center are not likely to affect the accuracy of the results significantly.

The room constant R is a measure of the sound absorption in a space. In a free field with no sound reflection at all, R is infinite, whereas in a room with no absorption, R is zero. In practice, neither of these extremes exists, and R is calculated as

$$R = \alpha_1 S_1 + \alpha_2 S_2 + \cdots + \alpha_n S_n \tag{45.7}$$

where $\alpha_1, \alpha_2, \ldots, \alpha_n$ are the sound absorption coefficients of materials on various surfaces and $S_1, S_2, \ldots, S_n$ are the areas of various surfaces, in square feet (or meters). The sound absorption coefficient α is a measure of the sound-absorptive property of a material as evaluated by ASTM Method C423, Test for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method. And α is defined as the fraction of the randomly incident sound power that is absorbed (or otherwise not reflected) by the material. Table 45.4 (Ref. [45.3]) lists sound absorption coefficients of various construction materials. Note that these coefficients usually vary with frequency, and so does room constant R.

Figure 45.4 gives a graph, obtained from Eqs. (45.5) and (45.6), showing $L_p - L_w$ as a function of $r/\sqrt{Q}$ for various values of room constant R. Thus, given r, Q, and R, the relationship between L_p and L_w can be calculated for each frequency band of interest. Application of these concepts is illustrated in the following example.

Example. Measurements of octave-band sound pressure levels 3 ft from the acoustic center of an air compressor, located inside an anechoic chamber, yielded the results shown in Table 45.5. The compressor is to be installed on the floor at the center of a workroom 60 ft long, 50 ft wide, and 30 ft high. The room has a hard concrete floor, coarse concrete block walls, and a ceiling made of 2-in-thick glass-fiber panels with plastic sheet wrapping and perforated metal facing. What will be the resulting octave-band sound pressure levels and overall A-weighted sound level at a work station located in the workroom 15 ft from the compressor?

The first step is to calculate the octave-band sound power levels for the compressor by using Eq. (45.5). For $r = 3$ ft, $Q = 1$ (spherical radiation), and $R = \infty$ (anechoic chamber), this equation indicates that $L_p - L_w = -10$, and thus $L_w - L_p = +10$ for all frequency bands. The octave-band sound power levels obtained in this way are indicated in Table 45.5.

The next step is to calculate the room constants for the workroom. First, a list is made of the octave-band sound absorption coefficients α for the floor, wall, and ceiling surfaces (see Table 45.5); these values are obtained from Table 45.4 for the materials of the various surfaces. Then these values are multiplied by the respective surface areas (3000 ft^2 for the floor or ceiling, 6600 ft^2 for the walls), and the results are summed to yield the room constant R for each octave band.

TABLE 45.4 Sound Absorption Coefficients of Construction Materials

Material	Sound absorption coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Ballast or other crushed stone						
3.18-cm (1¼-in) screened ballast 15.2 cm (6 in) deep	0.19	0.23	0.43	0.37	0.58	0.62
3.18 cm (1¼-in) 30.5 cm (12 in) deep	0.27	0.58	0.48	0.54	0.73	0.63
3.18 cm (1¼-in) 45.7 cm (18 in) deep	0.41	0.53	0.64	0.84	0.91	0.63
0.64-cm (¼-in) or less granite aggregate 15.2 cm (6 in) deep	0.22	0.64	0.70	0.79	0.88	0.72
Brick, unglazed	0.03	0.03	0.03	0.04	0.05	0.07
Brick, unglazed, painted	0.01	0.01	0.02	0.02	0.02	0.03
Carpet, heavy, on concrete	0.02	0.06	0.14	0.37	0.60	0.65
Same on 1350-g/m^2 (40-oz/yd^2) hairfelt or foam rubber	0.08	0.24	0.57	0.69	0.71	0.73
Same, with impermeable Latex backing on 1350-g/m^2 (40-oz/yd^2) hairfelt or foam rubber	0.08	0.27	0.39	0.34	0.48	0.63
Concrete block, coarse	0.36	0.44	0.31	0.29	0.39	0.25
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Drapes						
Light velour 338 g/m^2 (10 oz/yd^2) hung straight, in contact with wall	0.03	0.04	0.11	0.17	0.24	0.35
Medium velour 475 g/m^2 (14 oz/yd^2) draped to half area	0.07	0.31	0.49	0.75	0.70	0.60
Heavy velour, 610 g/m^2 (18 oz/yd^2) draped to half area	0.14	0.35	0.55	0.72	0.70	0.65

TABLE 45.4 Sound Absorption Coefficients of Construction Materials (*Continued*)

Material	Sound absorption coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Fiber-glass boards and blankets						
2.54-cm (1-in) glass wool 24 to 48 kg/m^3 (1.5 to 3.0 lb/ft^3)	0.08	0.25	0.65	0.85	0.80	0.75
5.1-cm (2-in) glass wool 24 to 48 kg/m^3 (1.5 to 3.0 lb/ft^3)	0.17	0.55	0.80	0.90	0.85	0.80
2.54-cm (1-in) glass wool, 2.54-cm (1-in) airspace	0.15	0.55	0.80	0.90	0.85	0.80
5.1-cm (2-in) glass-fiber panels with plastic sheet wrapping and perforated metal facing, as installed	0.33	0.79	0.99	0.91	0.76	0.64
Floors						
Concrete or terrazzo	0.01	0.01	0.015	0.02	0.02	0.02
Linoleum, asphalt, rubber, or cork tile on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Wood	0.15	0.11	0.10	0.07	0.06	0.07
Wood parquet in asphalt on concrete	0.04	0.04	0.07	0.06	0.06	0.07
Glass						
Large panes of heavy plate glass	0.18	0.06	0.04	0.03	0.02	0.02
Ordinary window glass	0.35	0.25	0.18	0.12	0.07	0.04
Gypsum board, 1.27 cm ($\frac{1}{2}$ in), nailed to 5.1 cm × 10.2 cm (2″ × 4″) studs 41 cm (16 in) center to center	0.29	0.10	0.05	0.04	0.07	0.09
Marble or glazed tile	0.01	0.01	0.01	0.01	0.02	0.02
Mineral spray-on materials						
1.27-cm ($\frac{1}{2}$-in) mineral fiber	0.05	0.15	0.45	0.70	0.80	0.80
1.9-cm ($\frac{3}{4}$-in) mineral fiber	0.10	0.30	0.60	0.90	0.90	0.95

TABLE 45.4 Sound Absorption Coefficients of Construction Materials (*Continued*)

Material	Sound absorption coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Mineral spray-on materials (*cont.*)						
2.5-cm (1-in) mineral fiber	0.16	0.45	0.70	0.90	0.90	0.85
1.27-cm (½-in) mineral fiber on metal lath, 2.54-cm (1-in) airspace	0.25	0.50	0.80	0.90	0.90	0.85
Plaster, gypsum or lime, smooth finish on tile or brick	0.013	0.015	0.02	0.03	0.04	0.05
Plaster, gypsum or lime, rough finish on lath	0.14	0.10	0.06	0.05	0.04	0.03
Same, with smooth finish	0.14	0.10	0.06	0.04	0.04	0.03
Plywood paneling, 1 cm (⅜ in) thick	0.28	0.22	0.17	0.09	0.10	0.11
Water surface, as in a swimming pool	0.008	0.008	0.013	0.015	0.020	0.025

SOURCE: From Harris [45.3]. Used by permission.

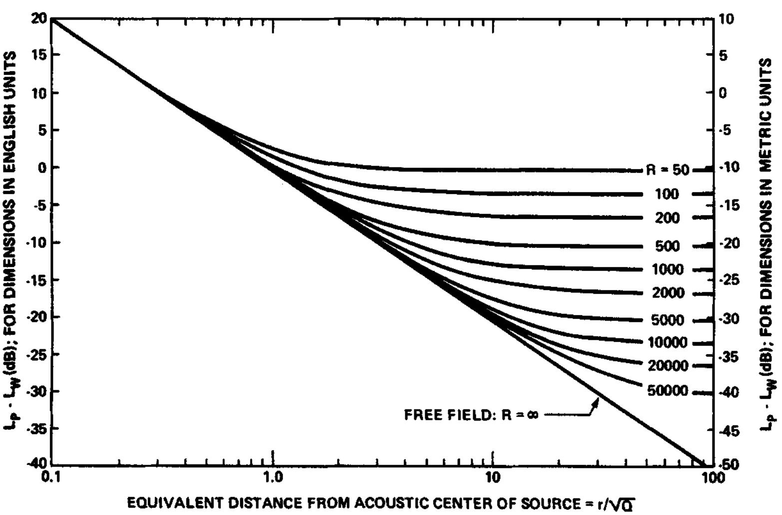

FIGURE 45.4 Relationship between sound pressure level and sound power level as a function of distance and directivity.

TABLE 45.5 Example of Noise Data Evaluation

	Octave-band center frequency, Hz					
	125	250	500	1000	2000	4000
L_p measured at 3 ft in anechoic chamber (dB)	87	86	89	92	92	90
$L_w - L_p$ (dB)	+10	+10	+10	+10	+10	+10
L_w (dB)	97	96	99	102	102	100
α (concrete floor)	0.01	0.01	0.015	0.02	0.02	0.02
α (concrete block walls)	0.36	0.44	0.31	0.29	0.39	0.25
α (glass-fiber panel ceiling)	0.33	0.79	0.99	0.91	0.76	0.64
$S\alpha$ (floor)	30	30	45	60	60	60
$S\alpha$ (walls)	2376	2904	2046	1914	2574	1650
$S\alpha$ (ceiling)	990	2370	2970	2730	2280	1920
R (ft^2)	3396	5304	5061	4344	4914	3630
L_p calculated at 15 ft in workroom (dB)	80	78	81	85	84	83
A-weighting	−16	−9	−3	0	+1	+1
L_p (A-weighted)†	64	69	78	85	85	84

†Overall, A-weighted sound level = 90 dBA.

The final step consists of calculating the octave-band sound pressure levels by using Eq. (45.5). With a directivity factor Q of 2 (for a source on a hard, reflecting surface), a distance r of 15 ft, and the calculated octave-band values of R and L_w, this equation yields the octave-band sound pressure levels L_p in the workroom 15 ft from the compressor given in Table 45.5. Next, the overall A-weighted sound level can be calculated by applying the A-weighting corrections (from Fig. 45.3) to each octave-band level (again, see Table 45.4) and then summing the results logarithmically, by using Eq. (45.2). This calculation yields an overall A-weighted sound level of 90 dBA at the work station.

45.3 NOISE EFFECTS AND STANDARDS

Figure 45.5 summarizes some of the limits and guidelines that can be used for evaluating various effects of noise on people. These effects and standards are discussed below.

45.3.1 Hearing Damage

Extremely intense sounds can produce nearly instantaneous hearing damage, but usually the development of hearing impairment is far more subtle. Persons exposed to high sound levels during part of a day will experience a temporary shift in the threshold of hearing. In other words, they will be unable to hear faint sounds for perhaps a few hours after exposure. After repeated exposure, generally over several

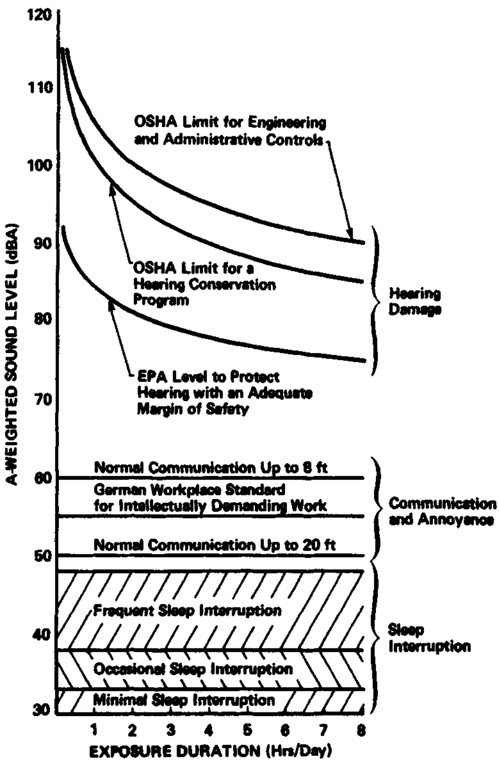

FIGURE 45.5 Noise effects and standards.

years, one's hearing threshold gradually rises and remains permanently high. After sufficiently long exposures to high sound levels, listeners are no longer able to understand normal speech and, in extreme cases, can lose hearing ability almost entirely.

Permanent threshold shifts can be induced by long-term exposures to A-weighted sound levels of about 80 dBA and become increasingly severe with higher levels of exposure [45.3]. Hearing acuity degrades to the greatest extent at frequencies around 4 kHz and less at very high frequencies (6 to 8 kHz) and at low frequencies (0.5 to 2 kHz).

For most U.S. industrial workers, the Occupational Safety and Health Administration (OSHA) established an exposure limit of 90 dBA for 8 hours (h) and uses the "5-dB increase per exposure halving rule" for sounds of lesser durations. That is, 95 dBA is allowed for 4 h, 100 dBA for 2 h, and so forth up to 115 dBA for 15 minutes (min). Exposure to continuous sounds above 115 dBA is not permitted, regardless of the duration. The OSHA exposure limit is given by the top curve in Fig. 45.5.

If workers are exposed to sounds of various durations and levels, a limit applies to the time-weighted average (TWA) sound level or to the daily dosage D. The TWA is calculated from

$$\mathrm{TWA} = 16.61 \log\left(\frac{C_1}{T_1} + \frac{C_2}{T_2} + \cdots + \frac{C_n}{T_n}\right) + 90 \tag{45.8}$$

where C_i = duration of exposure to a specified level and T_i = total time of exposure permitted at that level according to the foregoing rule. If the TWA exceeds 90 dBA, then the worker is considered to be overexposed. The daily dosage D (as a percentage of full exposure) is computed from

$$D = 100\left(\frac{C_1}{T_1} + \frac{C_2}{T_2} + \cdots + \frac{C_n}{T_n}\right) \tag{45.9}$$

If D exceeds 100 percent, then the worker is considered to be overexposed.

OSHA has also established an 85-dBA, 8-h exposure level as the threshold for which a hearing conservation program is required. The same algorithm is used for computing the time-weighted average as Eq. (45.8), except that 90 is replaced with 85. The hearing conservation program requires periodic audiometric testing of exposed workers and provision of hearing protectors for workers exhibiting significant permanent threshold shifts.

The Environmental Protection Agency (EPA) selected a level that would protect "virtually the entire population" against a hearing loss of 5 dB or less ([45.6]). Thus, the EPA recommends that exposure not exceed 70 dBA for 24 h or 75 dBA for 8 h, by the "3-dB rule" (that is, 78 dBA for 4 h, 81 dBA for 2 h, etc.). The EPA-recommended exposure limit is given in Fig. 45.5.

45.3.2 Speech Interference

A useful guide for determining when speech may be understood and the amount of effort required by the speaker is presented in Fig. 45.6 (Ref. [45.7]). Clearly, for the 15- to 20-ft distances common in many homes, schools, or workplaces, background levels should be less than about 50 dBA if communication is to be normal.

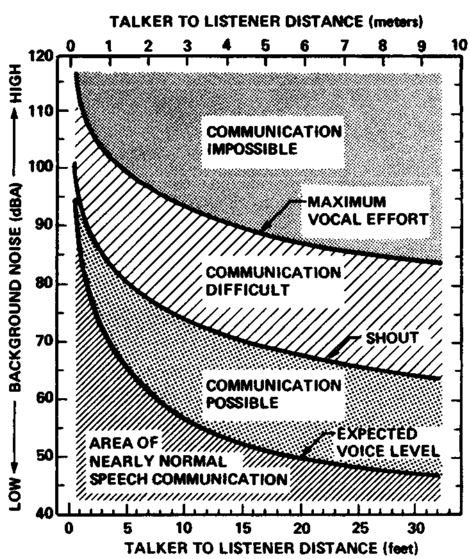

FIGURE 45.6 Quality of speech communication. *(From Miller [45.7].)*

45.3.3 Sleep Interruption

Sound can interrupt sleep in a number of complex ways that depend on the nature of the sound, the stage of sleep in which a person may be, and the individual's susceptibility to disturbance. In spite of these many factors, useful guidelines have been determined for sleep disturbance. A study of the effects of air conditioner noise [45.8] has shown the following reactions to steady noise in sleeping quarters:

Noise level, dBA	Response (complaints)
<33	None
33–38	Occasional
38–48	Frequent
>48	Unlimited

45.4 NOISE CONTROL

Noise control can be incorporated in the design of machinery either by treating the sources of machine noise or by altering the structureborne and/or airborne paths of this noise. This section contains a discussion of these different options for machinery noise control.

45.4.1 Source Control

Controlling noise at the source is often the most cost-effective procedure for machine design. Specific components, operating conditions, and/or geometries can frequently be selected that result in substantially less noise for the same function. This section discusses source noise control for some commonly encountered components and noise-generating mechanisms.

Fans and Blowers. Fans and blowers move air or other gases by lift forces on rotating fan blades or impeller vanes. This rotating pressure distribution generates some sound, but the fluctuating pressures on blades or impellers usually cause more significant noise. These pressures are generated by the turbulent boundary layers, by irregular vortex shedding at trailing edges, and by spatially and temporally varying inflow.

Where manufacturers' data or the results of special measurements are not available, it is useful to estimate fan sound power levels by the procedure recommended by the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) [45.9], which accounts for the dependence of fan noise on fan type, size, flow rate, and pressure drop as well as on the number of blades and the fan operating point.

Fan source noise control may be achieved by ensuring that the fan is operating efficiently, that is, that the fan is selected to operate at its peak efficiency at the pressure and flow conditions required by the system. For reduced noise, the inflow should be as uniform spatially and as free of turbulence as possible. Noise reductions may often be achieved by using larger fans at slower speeds and fans with swept blades instead of small high-speed fans and fans with straight blades.

Electric Motors. Electric motor noise is generated primarily by fluctuating magnetic loads, bearings, and cooling fans. Cooling fan noise is the dominant source for most motors.

Noise levels for standard and quieted totally enclosed fan-cooled motors are presented in Fig. 45.7 for a range of horsepower and operating speeds [45.10]. These data clearly show that for a given horsepower rating, sound power levels are distributed within a 10- to 20-dBA range. Standard untreated high-speed motors are invariably the noisiest, whereas quieted motors operating at low speeds are the quietest.

Noise control may be designed into motors through the combined use of high-temperature insulation and low-volume cooling fans. The insulation allows the motor to run hotter than normal, requiring less airflow to dissipate waste heat. The lower airflow and lower heat loss permit the use of smaller, quieter fans.

Noise abatement can also be achieved at the source by operational speed reduction. The data in Fig. 45.7 show that motors built to operate at 1800 instead of 3600 r/min can be as much as 17 dBA quieter and that motors built to operate at 1200 r/min are 2 to 17 dBA quieter than 3600 r/min units.

Gears. Gear noise is due to the unsteady forces associated with tooth meshing. These forces primarily result from geometric inaccuracies in the gear manufacturing process and from deflections of the teeth under load. The forces result in gear vibration, which is transmitted to the gear housing and often to contiguous structural members, all of which radiate sound. An investigation [45.11] of numerous types of gear sets has shown that radiated sound power ranges from about 2.5×10^{-6} to 10^{-8} times the mechanical power.

Source control of gear noise is best accomplished by selecting high-quality gears and gear boxes. Table 45.6 may be used to obtain a rough estimate of the corresponding reduction in gear noise and the approximate related increase in cost [45.12]. The "maximum conjugacy" tooth form indicated in Table 45.6 incorporates lengthened tooth addenda, circular arc profiles, low-pressure angles, and generous tooth-root radii.

The gear box should be designed to avoid structural resonances corresponding to tooth mesh frequencies, and bearings should be selected to minimize vibration

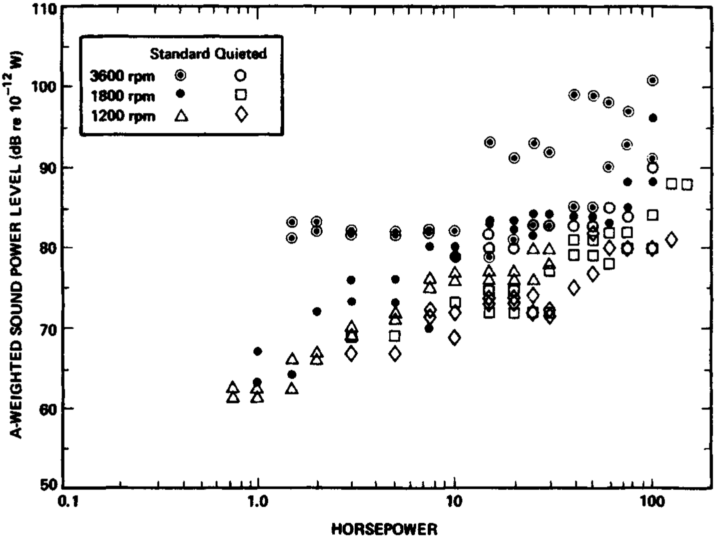

FIGURE 45.7 A-weighted sound power levels of electric motors.

transmission from gear shafts to the housing structure. Preloaded tapered-roller bearings are found to provide 4 to 5 dB of noise reduction for a given gear set when compared with unloaded ball bearings [45.11].

Hydraulic Pumps. Vane- and piston-type pumps contain oscillating components that result in pump casing vibration and sound radiation. All pumps (vane, piston, and gear) generate fluctuating pressures in working fluids that cause casing vibration and also propagate through inlet and outlet ports. This fluidborne sound radiates through tubing and can excite contiguous structures and other components.

A survey of pump noise levels taken from Ref. [45.13] and the manufacturers' literature is presented in Fig. 45.8. This figure shows how the A-weighted radiated sound power varies with hydraulic horsepower, that is, with the power delivered by the pump (equal to the pressure rise across the pump times the volume flow rate). Sound power levels can be within a large range at any value of hydraulic horsepower. Piston pumps tend to be somewhat noisier than others, and internal gear pumps tend to be somewhat quieter.

Source noise control for hydraulic pumps is best achieved by selecting pumps that have low values of radiated acoustic power, consistent with system requirements. Acoustic data can generally be obtained from manufacturers. If acoustic power levels are to be measured, care should be taken to isolate the pump from contiguous structures, because of the significant fluctuating hydraulic pressure and vibration transmitted through inlet and outlet tubing.

TABLE 45.6 Guidelines for Noise Reduction through Gear Design

Noise reduction principle	Design feature	Approximate reduction in sound pressure level, ΔdBA	Cost increase over cost of hobbed spur gears, %
Increasing total contact ratio	1. Reduce pressure angle to 14½°	0.5–1	0
	2. Lengthen addenda (usually 10 to 25%)	1–2	0–2
	3. Use substantial helix angle (20 to 35%)	3–6	0–5
	4. Use maximum-conjugacy tooth form	6–16	0–5
	5. Shot-peen roots to permit finer pitch	1–3	10–15
	6. Use stronger material to permit finer pitch	1–3	5–20
Improving kinematic accuracy	7. Relieve tooth tips, roots, or both	1–2	0
	8. Shave working surfaces	1–3	20–30
	9. Grind working surfaces	2–4	40–100
Reducing pitch line velocity	10. Maximize face-to-diameter ratio	2–6	10–30
	11. Provide multiple power paths	3–6	100–150

SOURCE: From Roverol [45.12]. Used by permission.

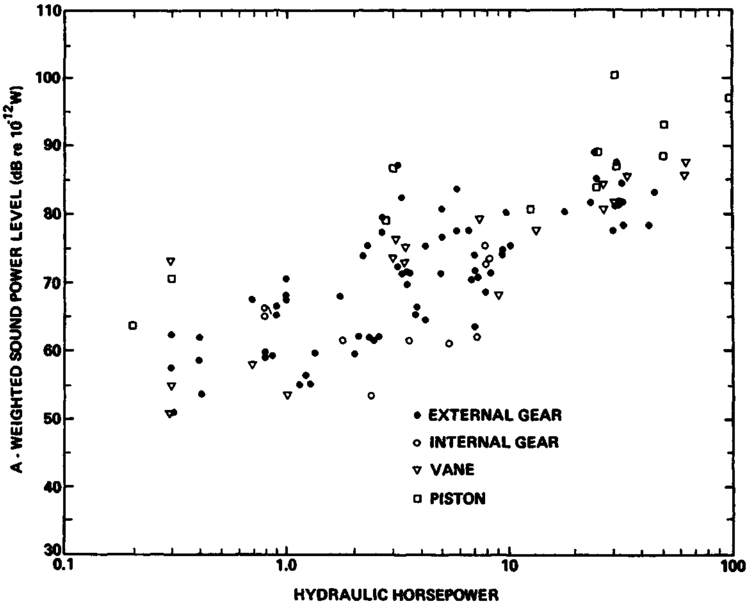

FIGURE 45.8 A-weighted sound power levels of hydraulic pumps.

Other Mechanical Components. Machines are made up of a large variety of mechanical components and mechanisms, each of which should be considered from the perspective of source noise control.

Impact mechanisms excite transient structural vibrations, which subsequently radiate sound. Impact noise can be controlled by seeking alternative mechanisms for performing a given task or by cushioning impacts. Alternative mechanisms could, for example, involve squeezing under hydraulic pressure instead of using impact devices. Cushioning is often accomplished by installing compliant materials at—or just behind—the impact point.

Rolling contact noise from bearings and other devices often results from surface irregularities. Therefore, noise can often be lessened by incorporating precision-ground components and good lubrication. Alternatively, well-lubricated journal bearings, which use an oil film to preclude metal-to-metal contact, may be substituted for ball or roller bearings.

Noise is generated from a wide variety of fluid-handling devices in addition to the pumps, blowers, and fans discussed earlier. Most common are valves and jets, which generate sound by turbulent flow. Since sound levels are strong functions of flow velocity V, source noise control is often best achieved by lowering velocities and by designing structures to reduce turbulence levels where possible.

45.4.2 Path Control

Control of Structureborne Noise. If a structural component is set into vibratory motion—by external mechanical excitation, by an adjacent vibrating structure, by impacts between it and adjacent components, or by incident sound—vibrations can propagate from the excitation region along this component to adjacent structures. Any vibrating structure can radiate sound, much as a loudspeaker membrane does; thus, any structural vibration or "structureborne sound" can give rise to audible noise or "airborne sound" if the vibrations (and sound) occur in the audible frequency range. The reduction of structureborne sound may be accomplished by

1. Reduction (or elimination) of the causes of the structureborne sound, such as vibration-producing impacts
2. Obstruction of the propagation of vibrations along structures to components that radiate sound efficiently
3. Reduction of the sound radiation effectiveness of vibrating structural surfaces

Shields (secondary walls) that stand free of the structure to be protected or that are connected to it only by soft resilient layers are often useful for reducing the vibrations induced in a structure. Such vibrations may be caused by incident sound fields or by fluctuating pressures, such as those that result from impinging fluid jets or from turbulent flows. To be effective, these shields generally need to be heavy, limp, and well damped, and the resilient layers also need to be soft and well damped—all in order to avoid significant resonances in the shield/resilient-layer/structure system and to keep the fundamental resonance frequency below the lowest frequency of interest.

Isolation from Sources. Vibration isolators placed between a vibration or impact source and a structure are generally useful for reducing the vibrations transmitted to the structure. These isolators basically consist of resilient elements, such as springs or elastomeric components or cushions, placed between the vibration sources and the structures to be protected. Isolators can be effective only if the stiffnesses of the resilient elements are (1) less than the effective dynamic stiffnesses of the structures to which they are connected, as measured at the attachment points, and (2) low enough that the fundamental resonance of the system, including the isolator, is smaller than the lowest driving frequency by a factor of at least 2.0. In all cases where isolation is used, care must be taken to avoid short-circuiting the isolation by "hard" paths, such as bolts, directly touching structures, or relatively rigid cables or ducts that can transmit vibrations relatively well.

Discontinuities. The propagation of vibrations along structures can be obstructed by the introduction of discontinuities, such as changes in stiffness or mass, as may be obtained by the addition of ribs or cross braces, by the introduction of abrupt changes in cross section (cut-outs or built-up areas) or in material properties (for example, by adding soft gaskets). Table 45.7 lists equations that may be used to estimate the attenuations provided by various discontinuities in extended beams and plates.

Added Structural Damping. The attenuation of propagating vibrations may be increased by the addition of structural damping treatments, which convert some of the vibratory energy to heat. Such treatments are useful largely only in frequency regions that encompass structural resonances or at high frequencies, where the structure is three or more wavelengths long. The reduction ΔL in the vibration level (and in the corresponding noise levels) due to adding an amount of damping η_a to an initially present amount η_i may be estimated from†

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† The symbol η, used here with various subscripts, represents the *loss factor,* that is, the ratio of the mechanical energy that is dissipated per radian (or that per cycle, divided by 2π) to the total energy of vibration. This measure of damping is related to other common ones by

$$\eta = 2\frac{c}{c_c} = \frac{\delta}{\pi} = \frac{1}{Q}$$

where c/c_c = ratio of viscous damping coefficient to critical damping coefficient, δ = logarithmic decrement, and Q = amplification at resonance.

$$\Delta L = B \log\left(1 + \frac{\eta_a}{\eta_i}\right) \tag{45.10}$$

where $B = 20$ for structures vibrating at resonance and $B = 10$ for structures subject to broadband excitation encompassing several resonances [45.14]. Note that the added damping η_a must be considerably greater than the initially present damping η_i if the added damping is to provide a significant reduction in level.

Some increases in the damping of machine structures can usually be obtained by replacing welds with bolted joints, by placing parts of the structures in contact with viscous fluids, or by replacing steel by cast iron or by special-purpose high-damping alloys. However, major damping increases usually can be achieved only by such special means as surrounding the structure with granular materials (such as sand, gravel, or lead shot) or bonding layers of high-damping viscoelastic materials to the structure. These materials typically are plastics or elastomers that are capable of dissipating considerable energy; many types are commercially available.

The increase in damping loss factor η_a that may be expected to be obtained by bonding a viscoelastic layer of thickness h to a structural plate or beam of thickness H may be estimated from

$$\eta_a = \frac{\beta}{1 + \dfrac{EH/(eh)}{3 + 3h/H + 4(h/H)^2}} \tag{45.11}$$

where β = loss factor of added viscoelastic material
e = elastic modulus of that material†
E = elastic modulus of material of basic plate or beam

Considerably increased damping often can be obtained by use of sandwich structures that incorporate high-damping viscoelastic materials. Design relations are available ([45.15]) but involve relatively complex frequency and wavelength dependences.

Reduction of Radiation. Reduction of the sound radiation from structures may be accomplished by minimizing the vibrations of radiating surfaces, as discussed earlier, and by reducing the sound-radiating capabilities of the structures. Because structures can produce sound only by pushing against air as their surfaces move perpendicularly to themselves, their sound-radiating capabilities can be reduced by keeping vibrations parallel to the major planes of structural surfaces, by reducing the areas of vibrating structures, and by providing perforations in structural surfaces.

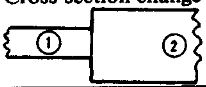
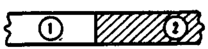
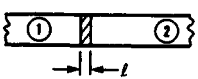
The decrease in radiated sound level resulting from reduction of the radiating area by the fraction n (or by $100n$ percent) may be estimated from

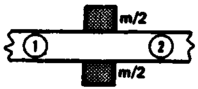
$$\Delta L = -20 \log (1 - n) \tag{45.12}$$

However, the level decrease obtained by the introduction of perforations may be greater than that resulting from the corresponding area reduction. Because perforations less than about half an acoustic wavelength apart permit the instantaneous pressure increases produced on one side of a vibrating surface essentially to be can-

† For most damping materials, β and e vary significantly with frequency and temperature; material selection must reflect these variations.

TABLE 45.7 Vibration Transmission Efficiencies of Structural Discontinuities†

	Longitudinal vibrations	Flexural vibrations
	$\lambda_L = \dfrac{\sqrt{E/\rho}}{f}$	$\lambda_F = \sqrt{2\pi R\lambda_L}$
Cross-section change 	$\dfrac{1}{\tau} = \dfrac{1}{4}\left(\sqrt{r} + \dfrac{1}{\sqrt{r}}\right)^2$	$\dfrac{1}{\tau} = \left[\dfrac{r^2/2 + \sqrt{r} + 1 + 1/\sqrt{r} + 1/(2r^2)}{r^{5/4} + r^{3/4} + r^{-3/4} + r^{-5/4}}\right]^2$
Change in material 	$\dfrac{1}{\tau} = \dfrac{1}{4}\left(\sqrt{\dfrac{E_1\rho_1}{E_2\rho_2}} + \sqrt{\dfrac{E_2\rho_2}{E_1\rho_1}}\right)^2$	$\dfrac{1}{\tau} = \left[\dfrac{a(1+b)^2 + 2b(1+a^2)}{2\sqrt{ab}(1+a)(1+b)}\right]^2$ $a = \left(\dfrac{\rho_2 E_1 R_1^2}{\rho_1 E_2 R_2^2}\right)^{1/4} \quad b = \dfrac{R_2 A_2}{R_1 A_1}\sqrt{\dfrac{E_2\rho_2}{E_1\rho_1}}$
Resilient insert 	$\dfrac{1}{\tau} = 1 = \left(\dfrac{f}{f_1}\right)^2$ $f_1 = \dfrac{k}{\pi A_1\sqrt{E_1\rho_1}}$	$\dfrac{1}{\tau} = 1 + \dfrac{f^3}{f_F^3}$ $f_F^3 = \dfrac{G^2}{2\pi^3\rho_1\sqrt{E_1\rho_1}R_1\ell^2}$

Blocking mass 	$\frac{1}{\tau} = 1 + \left(\frac{f}{f_m}\right)^2$ $f_m = \frac{A_1\sqrt{E_1\rho_1}}{\pi m}$	$\frac{1}{\tau} \approx \begin{matrix} 1 & \text{for } f < 0.5f_0 \\ 1 + f/f_B & \text{for } f > 2f_0 \end{matrix}$ $f_0 = \frac{R_1\sqrt{E_1\rho_1}}{2\pi R_m^2} \qquad f_B = \frac{2\rho_1 A_1 R_1\sqrt{E_1\rho_1}}{\pi m^2}$

†For structure on both sides of discontinuity assumed infinite—or many wavelengths long and reasonably well damped—and symmetric. (Adapted from Ref. [45.14]. Used by permission.)

τ = vibration transmission efficiency
IL = insertion loss (dB) = $10 \log (1/\tau) = 10 \log (v_{20}^2/v_2^2)$
v_{20} = vibration amplitude on receiving side in absence of discontinuity
v_2 = same, in presence of discontinuity
$r = \begin{cases} h_2/h_1 & \text{for plates} \\ A_2/A_1 & \text{for beams} \end{cases}$ λ_L = wavelength of longitudinal vibrations; λ_F = wavelength of flexural vibrations
h = plate thickness‡
A = cross-sectional area of beam or plate

E = modulus of elasticity, ρ = density — of plate or beam material‡

R = radius of gyration of beam cross section, for beam‡
$= h/\sqrt{12}$ for plate‡
G = shear modulus

ℓ = length, k = compressional stiffness — of resilient insert

m = total mass, R_m = radius of gyration — of added masses
f = frequency (Hz)

‡Subscripts 1 and 2 refer to structural components on which waves approach and leave discontinuity, respectively.

celed by the simultaneous pressure decreases produced on the other surface,† the net sound pressure may be reduced considerably.

Control of Airborne Noise. *Noise Control with Enclosures.*‡ Figure 45.9 is a schematic sketch of a typical complete enclosure. Such an enclosure is essentially a sealed box with stiffened walls. The panels of the walls have damping treatment applied to them to control their resonant vibration, and the interior of the box is covered with absorptive treatment (such as open-cell foam or glass-fiber mat) to prevent the buildup of reverberant sound in the interior. The machine to be quieted is vibration-isolated from the enclosure, so that the machine does not excite the walls of the enclosure, causing them to radiate sound and compromise the noise reduction performance. It is imperative that all openings be carefully sealed or provided with sound-absorbing ducts or mufflers. Even a small leak in a high-performance enclosure can seriously compromise its noise reduction.

A good measure of the performance of an enclosure is its insertion loss (IL), defined as 10 times the logarithm of the ratio of the sound power radiated by the untreated source to the sound power radiated by the source through the enclosure. In other words, the insertion loss is the noise reduction achievable by putting the source inside the enclosure.

The acoustic behavior of an enclosure may be understood in terms of four frequency regions, as indicated in Fig. 45.10. Region I is below the first resonant frequency of the panels of the enclosure. In this region, the insertion loss is initially

† The perforations need to be configured so as to permit airflow through them with little resistance; the cross-sectional flow area of each opening should be greater than its wetted perimeter area.

‡ See Refs. [45.16], [45.17], [45.18], and [45.19].

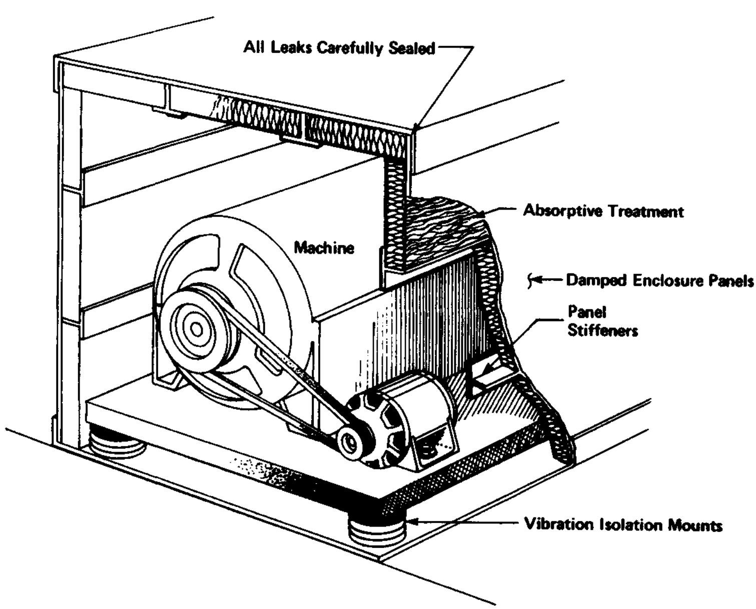


FIGURE 45.9 Cross section of a simple complete enclosure.

constant but dips rapidly near the first panel resonance. This region is often called the *stiffness-controlled* region, since the insertion loss here depends predominantly on the bending stiffness of the panels and increases with increasing stiffness.

Region II lies above the first panel resonance frequency. Here the transmission of sound through the enclosure is controlled by the panel resonances and acoustic standing waves between the enclosed machine and the walls of the enclosure. The acoustic behavior is very complicated. Insertion loss in this region can generally be increased through the use of enclosure panel damping and increased acoustic absorption in the interior of the enclosure.

In region III, the acoustic behavior is much like that in region II, except that the panel resonant modes in region III are so closely spaced in frequency that many resonant panel modes are present in any frequency bandwidth of common interest. In this region, the insertion loss decreases with increases in the transmission coefficient of the panels of the enclosure (that is, the ratio of acoustic power transmitted through the panel to the acoustic power incident on them) and increases with increased average absorption coefficient (or, the ratio of acoustic power absorbed on the surface of the inside of the enclosure to the acoustic power incident there). Since the transmission coefficient of the panels decreases with increases in their mass per unit area and the insertion loss here depends on the mass per unit area, region III is often called the *mass-controlled* region.

In region IV, a pronounced dip occurs at the *coincidence* frequency, or the frequency at which the bending wave speed in the panels equals the acoustic wave speed. At this frequency, sound passes readily through the panels, and the insertion loss can be increased by increasing the mass or damping of the panels. Alternatively, one may change the thickness or material properties of the panels so as to change the coincidence frequency and move it out of the frequency range of interest.

Table 45.8 lists the approximate formulas for estimating the insertion loss in the four frequency regions. The result of using these design formulas is shown by the dashed curves in Fig. 45.10. In region I, the design formula is most accurate well below the first panel resonance f_0. Near f_0 it overestimates the insertion loss. The table

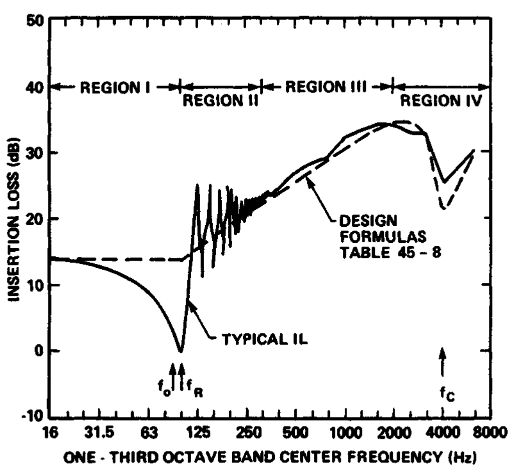


FIGURE 45.10 Insertion loss of a representative enclosure.

includes a formula for estimating the insertion loss at f_R, the frequency where the insertion loss first dips. The frequency f_R is slightly higher than the panel resonance frequency because of the panel stiffening contributed by the air in the enclosure.

As mentioned earlier, the behavior of the insertion loss in region II is quite complicated. An appropriate average value may be obtained by use of the design formula from region III.

For regions III and IV, the same design formulas may be used. The insertion loss here depends on the transmission coefficient τ (fraction of sound transmitted) and the absorption coefficient α (fraction of sound absorbed). Note that if there is no absorption in the enclosure or if α is much less than τ, then the enclosure provides no insertion loss. Absorption is crucial to the functioning of the enclosure. Adequate absorption can be ensured by installing blankets of materials such as glass-fiber mat or open-cell foam on the walls of the enclosure. Typical values of α for some of these materials are given in Table 45.4.

The transmission coefficient can be obtained from Fig. 45.11, where the ratio of τ to the normal incidence mass law transmission coefficient τ_0 is plotted against the ratio of the frequency to the critical frequency. As an aid in using Fig. 45.11, the critical frequency for various panel materials and thicknesses is given in Fig. 45.12. And τ_0 can be calculated by

$$\tau_0 = \frac{1}{1 + [2\pi f m/(2\rho c)]^2} \tag{45.13}$$

where f = sound frequency
m = panel mass per unit area
ρ = mass density of air
c = speed of sound in air

Sample Calculation. Calculate the insertion loss of an enclosure with the characteristics given in Fig. 45.13*a*.

1. Determine the stiffness S of the largest panel from the equation in Table 45.8. For

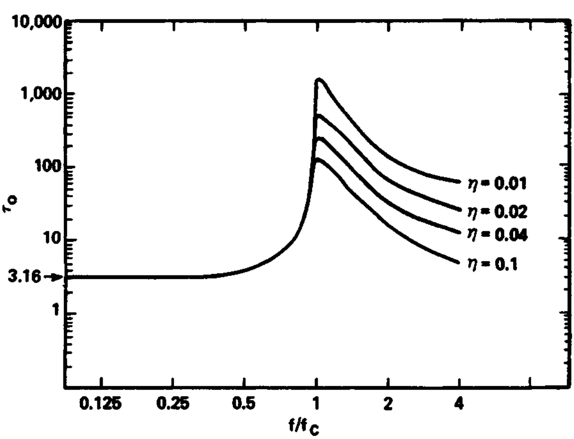


FIGURE 45.11 Transmission coefficient; m is the mass per unit area of the panel; η is the panel loss factor; ρ_0 is the density of air; c_0 is the speed of sound in air; f_c is the critical frequency (see Fig. 45.12).

TABLE 45.8 Enclosure Design Formulas

Region	Frequency range	Design formula	To improve insertion loss, increase:
I. Stiffness-controlled region	$f \ll f_0$	$\mathrm{IL_I} = 20 \log\left(1 + \frac{Sl}{\rho c^2}\right)$	• Panel stiffness • Enclosure volume
	$f_R \sim f_0\left(1 + \frac{\rho c^2}{Sl}\right)^{1/2}$	$\mathrm{IL} \sim 10 \log\left\{\left[4\pi\left(\frac{f_0 l}{c}\right)\eta + \eta^2\frac{Sl}{\rho c^2}\right]\left(1 + \frac{Sl}{\rho c^2}\right)\right\}$	• Panel stiffness • Enclosure volume • Panel damping
II. Resonance-controlled region	$f > f_0$	See region III	• Enclosure absorption • Panel damping
III. Mass-controlled	$\frac{f_c}{2} > f \gg f_0$	$\mathrm{IL_{III}} = 10 \log \frac{A_e\tau + \alpha A_\alpha}{A_e\tau}$	• Panel mass • Enclosure absorption
IV. Critical frequency	$f > \frac{f_c}{2}$	$\mathrm{IL_{IV}} = \mathrm{IL_{III}}$	• Panel mass • Enclosure absorption • Panel damping • f_c by reducing panel thickness

† S = panel stiffness/unit area
$= \frac{Et^3\pi^4}{12(1-\mu^2)}\left(\frac{1}{a^2} + \frac{1}{b^2}\right)^2$ for simply supported panel
t = panel thickness
a,b = panel dimensions
E = panel modulus
μ = Poisson's ratio for panel
l = distance from surface of noise source to enclosure walls
ρ = air density
c = acoustic wave speed
f = frequency, Hz

f_0 = first panel resonant frequency; $f_0 = \frac{1}{2\pi}\sqrt{\frac{S}{m}}$
m = mass per unit area of panels
η = panel loss factor
τ = panel transmission coefficient (see Fig. 45.11)
α = absorption coefficient (see Table 45.4)
A_e = surface area of enclosure
A_α = surface area covered by absorption
f_c = critical frequency (see Fig. 45.12)
f_R = frequency of first dip in insertion loss just above f_0

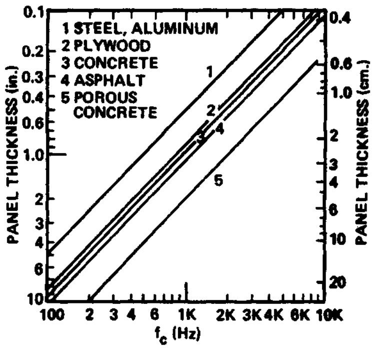


FIGURE 45.12 Critical frequency for panels of various thicknesses and materials. *(From Diehl [45.5].)* Here $f_c = c_0^2/\kappa C_l$, c_0 is the acoustic wave speed; κ is the panel radius of gyration, $\kappa = t/\sqrt{12}$ for the most simple panels; and c_l is the longitudinal wave speed in the panel $c_l = \sqrt{E/\rho_P}$, where E is the modulus and ρ_P is the density of the panel.

$$E = 1.44 \times 10^9 \text{ lb/ft}^2 \qquad t = 0.008 \text{ ft}$$

$$\mu = 0.3 \qquad a = 1.5 \text{ ft}$$

$$b = 1 \text{ ft}$$

one obtains

$$S = 1.38 \times 10^4 \text{ lb/ft}^3$$

Note that by ignoring the stiffeners in this calculation, one implicitly assumes that the panels are much less stiff than the stiffening beams. Consequently, the stiffening beams must be designed so that if a point load is applied to one of the panels of the enclosure, the deflection of the panel is much greater than that of the frame.

2. Calculate the natural frequency f_0 of that panel from the equation in Table 45.8.

Using S from the above calculation and a surface density of 1.33 lb/ft² for 0.1-in-thick aluminum, one obtains

$$m = \rho t = \frac{1.33}{32.2} = 0.0413 \text{ slug/ft}^2$$

and

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{S}{m}} = \frac{1}{2\pi}\sqrt{\frac{1.38 \times 10^4}{0.0413}} = 92 \text{ Hz}$$

Note that the equations for S and f_0 imply simply supported panels. In all likelihood, the true end conditions will tend to stiffen the panel even more, implying that the estimate here of the insertion loss is conservative; i.e., it predicts lower values than those that will occur.

3. Calculate the frequency of the first dip in the insertion loss f_R by using the equation in Table 45.8. For

$$\rho = 0.0023 \text{ slug/ft}^3 \qquad c = 1128 \text{ ft/s} \qquad l = 0.25 \text{ ft}$$

one finds

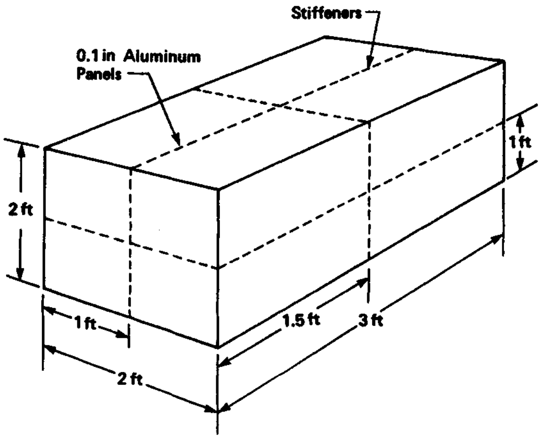
$$f_R = 125 \text{ Hz}$$

4. Determine the coincidence frequency from curve 1 in Fig. 45.12:

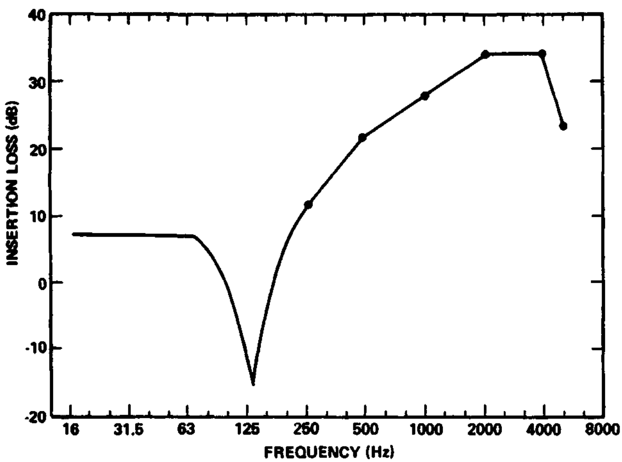
$$f_c = 5000 \text{ Hz}$$

5. Calculate IL for $f < f_R$, using the appropriate equation for region I from Table 45.8:

$$\text{IL}_\text{I} = 7 \text{ dB}$$



(a)



(b)

FIGURE 45.13 (*a*) Enclosure characteristics for illustrative calculations. Dimensions, 2 × 2 × 3 ft; material, aluminum of 160-lb/ft³ density; panel thickness = 0.1 in; panel damping = 0.05 for all frequencies; average spacing between source and panel walls is 3 in; stiffeners: stiffeners in each panel dividing each into four panels of equal size; absorption is 1 in of glass-fiber mat on all interior surfaces. (*b*) Insertion loss calculated for example.

6. Calculate IL at $f = f_R$, using the appropriate equation for region I from Table 45.8:

$$\text{IL} \approx -14.5 \text{ dB}$$

7. Calculate IL for $f > f_R$:

- Select a frequency greater than f_R and calculate τ_0 by using Eq. (45.13). For

$$f = 250 \text{ Hz} \qquad m = 0.0413 \text{ slug/ft}^2$$

$$\rho = 6.0023 \text{ slug/ft}^3 \qquad c = 1128 \text{ ft/s}$$

one obtains

$$\tau_0 = 0.00633$$

- Calculate f/f_c and determine τ/τ_0 from Fig. 45.11:

$$\frac{f}{f_c} = 0.050 \qquad \frac{\tau}{\tau_0} = 3.16$$

- Calculate τ:

$$\tau = \frac{\tau}{\tau_0}\tau_0 = 0.02$$

- Determine α at $f = 250$ Hz from Table 45.4:

$$\alpha = 0.25$$

- Calculate IL, using the equation for region III from Table 46.8. For

$$A_e = A_\alpha$$

one obtains

$$\text{IL} = 11 \text{ dB @ } 250 \text{ Hz}$$

- Repeat for other selected frequencies.

The results of similar calculations for this example for the octave bands at 500, 1000, 2000, and 4000 Hz, and the coincidence frequency 5000 Hz, are shown in Fig. 45.13*b*.

The techniques that can be used to improve the insertion loss of an enclosure depend on the frequency region in which the improvement is desired. In the example of Fig. 45.13*b*, there is a clear deficiency in region I, where the enclosure actually amplifies the noise from the source at 125 Hz. Increasing the insertion loss in this region is usually best accomplished by stiffening of the panels of the enclosure. The approaches that might be employed include

- Increasing the number of stiffeners, so that the effective panel size is reduced
- Increasing the panel thickness
- Using stiffer materials or configurations, such as graphite epoxy or honeycomb composite materials, for instance

One might also damp the panels to reduce the resonant peak in region I, although it usually is difficult to design damping treatments that will be effective at the low frequencies in this region. In regions II, III, and IV, the insertion loss can be increased by decreasing the transmission coefficient or by increasing the absorption coefficient. To increase the absorption coefficient in any region one might, for example,

- Cover more of the interior surfaces of the enclosure with absorptive material
- Select a material (from Table 45.4) that has a higher absorption coefficient in the frequency region of interest
- Use a thicker layer of absorptive material if increased absorption at low frequencies is required

Techniques for decreasing the transmission coefficient depend on the region of interest. In region II, one can

- Damp the panels by gluing on commercially available damping materials
- Increase the panel mass per unit area by using a thicker panel or a panel made from a denser material (e.g., steel instead of aluminum) or by adding layers of limp massive materials, such as leaded vinyl

The above techniques for increasing panel mass are also effective in region III. In this region, increasing panel damping will be effective only insofar as the damping material adds mass to the panel.

In region IV, increased panel mass and increased panel damping both contribute to increasing the enclosure insertion loss. In addition, one may also alter the structure to move the coincidence frequency out of the frequency range of interest by using a thinner or less stiff panel. (Of course, care must be exercised so that such treatments do not degrade the enclosure performance excessively in other frequency regions.)

Partial Enclosures. Where a complete enclosure cannot be used (for example, when openings for cooling air must be provided), mufflers can be employed to prevent noise from escaping from these openings (see Fig. 45.14). With properly designed mufflers, a partial enclosure can be as effective as a complete enclosure. The high-frequency aspect of the corresponding design is discussed in later sections of this chapter. At a certain low frequency, the *Helmholtz resonance frequency,* considerable sound transmission through the enclosure opening can occur. This frequency is given by

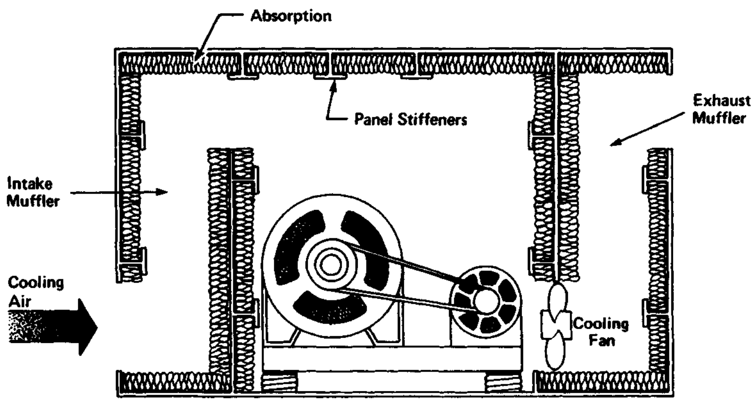


FIGURE 45.14 Schematic sectional view of a partial enclosure with mufflers.

$$f_H = \frac{c}{2\pi}\sqrt{\frac{A}{LV}} \tag{45.14}$$

where c = speed of sound
A = open area of duct
L = duct length
V = air volume contained in enclosure (enclosure volume minus volume of machine)

It is important to check that the Helmholtz resonance frequency is low enough that it is outside the frequency range of interest.

If the enclosure in the example above has two ducts 3 in square by 2 ft long and the free volume of the enclosure is 8 ft^3, then

$$V = 8\ \text{ft}^3 \qquad A = 0.125\ \text{ft}^2 \qquad \text{(two ducts)}$$

$$L = 2\ \text{ft} \qquad c = 1128\ \text{ft/s}$$

$$f_H = 15\ \text{Hz}$$

This resonance frequency is too low to be of much concern. In fact, this is the case in most practical situations. If it is not, one may increase the enclosure volume or change the duct geometry to reduce f_H.

Noise Control with Mufflers. Mufflers are special ducts or pipes or openings that allow for the free flow of air or other gases while impeding the transmission of sound.

In internal-combustion engines, these devices are almost always used at the exhaust and are sometimes used at the air inlet. Similarly, if a machine that requires cooling air is placed in an enclosure, the enclosure must be equipped with mufflers through which cooling air can be circulated.

There are two basic muffler types: *reactive* and *resistive.* Reactive mufflers rely on the reflection of acoustic waves at discontinuities (e.g., expansion chambers, side branch resonators) and on the interaction of these waves to reduce the transmission of sound. Mufflers for internal-combustion engines are primarily of this type. Reactive mufflers are essentially tuned devices providing high attenuation in some frequency bands and little attenuation in others.

Resistive, or dissipative, mufflers attenuate sound by the acoustic energy–absorbing action of absorptive material within the muffler. These devices typically provide noise attenuation over a broad frequency range. In its simplest form, a dissipative muffler is a duct with its walls lined with acoustically absorptive material.

Reactive Mufflers. Reactive mufflers for exhaust and intake silencing of internal-combustion engines are available from a number of commercial firms. These devices come in a great variety of shapes and sizes, and since their performance may be different on different engines, muffler manufacturers have usually measured the insertion loss and backpressure of their mufflers on the particular engine or class of engines for which their use is appropriate.

Techniques for design or analysis of reactive mufflers are too complex for inclusion here. However, both classical ([45.20]) and finite-element ([45.21]) techniques can be employed to estimate the insertion loss of a muffling system.

Resistive Mufflers. Resistive mufflers are often used in conjunction with enclosures, as discussed earlier. Two geometries are most commonly used: the lined duct and the lined plenum chamber.

The performance of the lined duct is indicated in Fig. 45.15, which shows the attenuation (in decibels) for sound traveling down the duct per unit length of duct of width ℓ, with absorptive lining on only two sides of the duct. If the duct is lined on four sides, one simply adds the attenuation of a duct lined on two sides to the attenuation of a duct lined top and bottom. For a parallel baffle muffler, a special type of lined duct illustrated in Fig. 45.16, the attenuation per unit length can be estimated from Fig. 45.15 by setting ℓ equal to w and t equal to $h/2$.

A lined plenum chamber is shown schematically in Fig. 45.17. Its transmission loss (TL) is given approximately by

$$\mathrm{TL} = -10 \log \left[A_0 \left(\frac{W}{2\pi q^3} + \frac{1}{A_\alpha} \right) \right] \tag{45.15}$$

where W and q are defined in Fig. 45.17, A_0 = area of the outlet, A_α = surface area covered with absorption, and α = absorption coefficient. The equation yields reasonable estimates, provided that the plenum dimensions are large compared to an acoustic wavelength.

Sample Calculation—Lined Duct. A parallel baffle muffler is to be constructed in a 1-ft^2 duct, as illustrated in Fig. 45.18. What length of duct will be required to obtain 10-dB attenuation at 1000 Hz?

1. From Fig. 45.18, ℓ is found to be equal to w or 4 in.
2. At 1000 Hz, $\lambda = c/f = 1128/1000 = 1.13$ ft and

$$\frac{\ell}{\lambda} = \frac{4}{12(1.13)} = 0.29$$

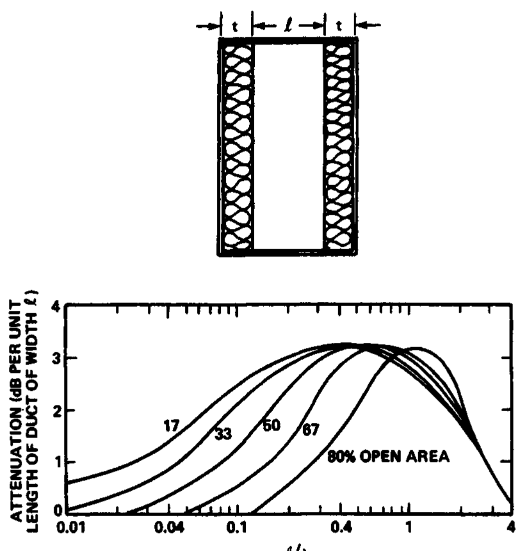


FIGURE 45.15 Attenuation of a sound by a lined duct. *(From Beranek [45.2].)*

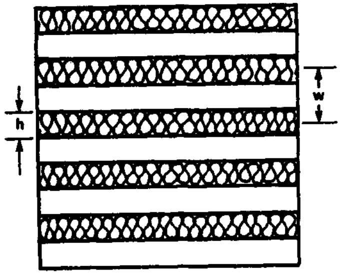


FIGURE 45.16 Parallel baffle muffler.

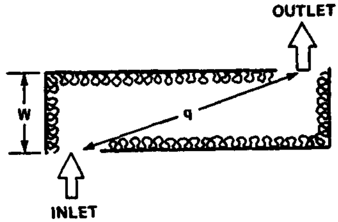


FIGURE 45.17 Plenum chamber.

3. The open area comprises 50 percent. From the corresponding curve in Fig. 45.15, the attenuation is found to be ~ 2.9 dB at $\ell/\lambda = 0.29$.
4. The desired attenuation is 10 dB. Since 2.9 dB is the attenuation every 4 in, the required length is

$$\frac{10}{2.9} \times 4 \approx 14 \text{ in}$$

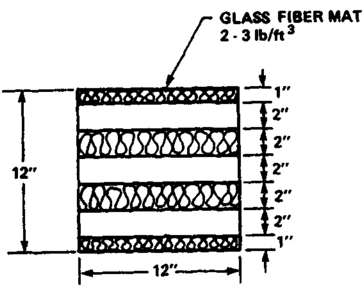


FIGURE 45.18 Parallel baffle muffler (for illustrative calculation).

Sample Calculation—Lined Plenum Chamber. Two plenum chambers are required for cooling air to enter and leave an enclosure. The walls of the enclosure are designed for a transmission loss of 10 dB at 1000 Hz. There is space available to install two plenums of the dimensions shown in Fig. 45.19. If the interior surfaces of each chamber are covered with absorptive material with an absorption coefficient of 0.90 at 1000 Hz and the required outlet area is 1 ft², will the chambers be able to provide adequate transmission loss?

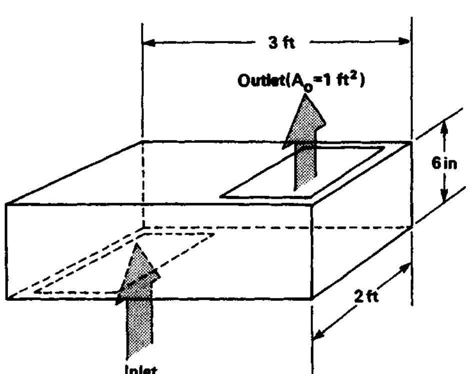


FIGURE 45.19 Plenum chamber for the sample calculation.

1. From Fig. 45.19, the parameters are

$$A_0 = 1 \text{ ft}^2 \qquad W = 0.5 \text{ ft} \qquad \alpha = 0.90$$

$$A_\alpha = 2(3 \times 0.5 + 2 \times 0.5 + 3 \times 2) = 17 \text{ ft}^2$$

$$q \sim [3^2 + (0.5)^2]^{1/2} \approx 3 \text{ ft}$$

2. Substituting these values into Eq. (45.15), one obtains

$$\text{TL} \approx 11 \text{ dB}$$

Therefore, the chamber provides more than the desired 10-dB insertion loss at 1000 Hz.

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