CHAPTER 24
NOISE MEASUREMENT AND CONTROL

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24.1 SOUND CHARACTERISTICS

Sound is a compressional wave. The particles of the medium carrying the wave vibrate longitudinally, or back and forth, in the direction of travel of the wave, producing alternating regions of compression and rarefaction. In the compressed zones the particles move forward in the direction of travel, whereas in the rarefied zones they move opposite to the direction of travel. Sound waves differ from light

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waves in that light consists of transverse waves, or waves that vibrate in a plane normal to the direction of propagation.

24.2 FREQUENCY AND WAVELENGTH

Wavelength, the distance from one compressed zone to the next, is the distance the wave travels during one cycle. Frequency is the number of complete waves transmitted per second. Wavelength and frequency are related by the equation

$$v = f\lambda$$

where v = velocity of sound, in meters per second
f = frequency, in cycles per second or hertz
λ = wavelength, in meters

24.3 VELOCITY OF SOUND

The velocity of sound in air depends on the temperature, and is equal to

$$v = 20.05\sqrt{273.2 + C^\circ}\ \text{m/sec}$$

where C° is the temperature in degrees Celsius.

The velocity in the air may also be expressed as

$$v = 49.03\sqrt{459.7 + F^\circ}\ \text{ft/sec}$$

where F° is the temperature in degrees Fahrenheit.

The velocity of sound in various materials is shown in Tables 24.1, 24.2, and 24.3.

24.4 SOUND POWER AND SOUND PRESSURE

Sound power is measured in watts. It is independent of distance from the source, and independent of the environment. Sound intensity, or watts per unit area, is dependent on distance. Total radiated sound power may be considered to pass through a spherical surface surrounding the source. Since the radius of the sphere increases with distance, the intensity, or watts per unit area, must also decrease with distance from the source.

Microphones, sound-measuring instruments, and the ear of a listener respond to changing pressures in a sound wave. Sound power, which cannot be measured directly, is proportional to the mean-square sound pressure, p^2, and can be determined from it.

24.5 DECIBELS AND LEVELS

In acoustics, sound is expressed in decibels instead of watts. By definition, a decibel is 10 times the logarithm, to the base 10, of a ratio of two powers, or powerlike quantities. The reference power is 1 pW, or 10^{-12} W. Therefore,

$$L_W = 10 \log \frac{W}{10^{-12}} \tag{24.1}$$

where L_W = sound power level in dB
W = sound power in watts
log = logarithm to base 10

Sound pressure level is 10 times the logarithm of the pressure ratio squared, or 20 times the logarithm of the pressure ratio. The reference sound pressure is 20 μPa, or 20×10^{-6} Pa. Therefore,

$$L_p = 20 \log \frac{p}{20 \times 10^{-6}} \tag{24.2}$$

where L_p = sound pressure level in dB
p = root-mean-square sound pressure in Pa
log = logarithm to base 10

24.6 COMBINING DECIBELS

It is often necessary to combine sound levels from several sources. For example, it may be desired to estimate the combined effect of adding another machine in an area where other equipment is operating. The procedure for doing this is to combine the sounds on an energy basis, as follows:

Table 24.1 Velocity of Sound in Solids

Material	Longitudinal Bar Velocity		Plate (Bulk) Velocity	
	cm/sec	fps	cm/sec	fps
Aluminum	5.24×10^5	1.72×10^4	6.4×10^5	2.1×10^4
Antimony	3.40×10^5	1.12×10^4	—	—
Bismuth	1.79×10^5	5.87×10^3	2.18×10^5	7.15×10^3
Brass	3.42×10^5	1.12×10^4	4.25×10^5	1.39×10^4
Cadmium	2.40×10^5	7.87×10^3	2.78×10^5	9.12×10^3
Constantan	4.30×10^5	1.41×10^4	5.24×10^5	1.72×10^4
Copper	3.58×10^5	1.17×10^4	4.60×10^5	1.51×10^4
German silver	3.58×10^5	1.17×10^4	4.76×10^5	1.56×10^4
Gold	2.03×10^5	6.66×10^3	3.24×10^5	1.06×10^4
Iridium	4.79×10^5	1.57×10^4	—	—
Iron	5.17×10^5	1.70×10^4	5.85×10^5	1.92×10^4
Lead	1.25×10^5	4.10×10^3	2.40×10^5	7.87×10^3
Magnesium	4.90×10^5	1.61×10^4	—	—
Manganese	3.83×10^5	1.26×10^4	4.66×10^5	1.53×10^4
Nickel	4.76×10^5	1.56×10^4	5.60×10^5	1.84×10^4
Platinum	2.80×10^5	9.19×10^3	3.96×10^5	1.30×10^4
Silver	2.64×10^5	8.66×10^3	3.60×10^5	1.18×10^4
Steel	5.05×10^5	1.66×10^4	6.10×10^5	2.00×10^4
Tantalum	3.35×10^5	1.10×10^4	—	—
Tin	2.73×10^5	8.96×10^3	3.32×10^5	1.09×10^4
Tungsten	4.31×10^5	1.41×10^4	5.46×10^5	1.79×10^4
Zinc	3.81×10^5	1.25×10^4	4.17×10^5	1.37×10^4
Cork	5.00×10^4	1.64×10^3	—	—
Crystals				
Quartz X cut	5.44×10^5	1.78×10^4	5.72×10^5	1.88×10^4
Rock salt X cut	4.51×10^5	1.48×10^4	4.78×10^5	1.57×10^4
Glass				
Heavy flint	3.49×10^5	1.15×10^4	3.76×10^5	1.23×10^4
Extra heavy flint	4.55×10^5	1.49×10^4	4.80×10^5	1.57×10^4
Heaviest crown	4.71×10^5	1.55×10^4	5.26×10^5	1.73×10^4
Crown	5.30×10^5	1.74×10^4	5.66×10^5	1.86×10^4
Quartz	5.37×10^5	1.76×10^4	5.57×10^5	1.81×10^4
Granite	3.95×10^5	1.30×10^4	—	—
Ivory	3.01×10^5	9.88×10^3	—	—
Marble	3.81×10^5	1.25×10^4	—	—
Slate	4.51×10^5	1.48×10^4	—	—
Wood				
Elm	1.01×10^5	3.31×10^3	—	—
Oak	4.10×10^5	1.35×10^4	—	—

$$L_p = 10 \log [10^{0.1L_1} + 10^{0.1L_2} + \cdots + 10^{0.1L_n}] \tag{24.3}$$

where L_p = total sound pressure level in dB
L_1 = sound pressure level of source No. 1
L_n = sound pressure level of source No. n
log = logarithm to base 10

24.7 SOUND PRODUCED BY SEVERAL MACHINES OF THE SAME TYPE

The total sound produced by a number of machines of the same type can be determined by adding 10 log n to the sound produced by one machine alone. That is,

Table 24.2 Velocity of Sound in Liquids

Material	Temperature °C	Temperature °F	Velocity cm/sec	Velocity fps
Alcohol, ethyl	12.5	54.5	1.21×10^5	3.97×10^3
	20	68	1.17×10^5	3.84×10^3
Benzene	20	68	1.32×10^5	4.33×10^3
Carbon bisulfide	20	68	1.16×10^5	3.81×10^3
Chloroform	20	68	1.00×10^5	3.28×10^3
Ether, ethyl	20	68	1.01×10^5	3.31×10^3
Glycerine	20	68	1.92×10^5	6.30×10^3
Mercury	20	68	1.45×10^5	4.76×10^3
Pentane	20	68	1.02×10^5	3.35×10^3
Petroleum	15	59	1.33×10^5	4.36×10^3
Turpentine	3.5	38.3	1.37×10^5	4.49×10^3
	27	80.6	1.28×10^5	4.20×10^3
Water, fresh	17	62.6	1.43×10^5	4.69×10^3
Water, sea	17	62.6	1.51×10^5	4.95×10^3

$$L_p(n) = L_p + 10 \log n$$

where $L_p(n)$ = sound pressure level of n machines
L_p = sound pressure level of one machine
n = number of machines of the same type

In practice, the increase in sound pressure level measured at any location seldom exceeds 6 dB, no matter how many machines are operating. This is because of the necessary spacing between machines, and the fact that sound pressure level decreases with distance.

Table 24.3 Velocity of Sound in Gases

Material	Temperature °C	Temperature °F	Velocity cm/sec	Velocity fps
Air	0	32	3.31×10^4	1.09×10^3
	20	68	3.43×10^4	1.13×10^3
Ammonia gas	0	32	4.15×10^4	1.48×10^3
Carbon dioxide	0	32	2.59×10^4	8.50×10^2
Carbon monoxide	0	32	3.33×10^4	1.09×10^3
Chlorine	0	32	2.06×10^4	6.76×10^2
Ethane	10	50	3.08×10^4	1.01×10^3
Ethylene	0	32	3.17×10^4	1.04×10^3
Hydrogen	0	32	1.28×10^5	4.20×10^3
Hydrogen chloride	0	32	2.96×10^4	9.71×10^2
Hydrogen sulfide	0	32	2.89×10^4	9.48×10^2
Methane	0	32	4.30×10^4	1.41×10^3
Nitric oxide	10	50	3.24×10^4	1.06×10^3
Nitrogen	0	32	3.34×10^4	1.10×10^3
	20	68	3.51×10^4	1.15×10^3
Nitrous oxide	0	32	2.60×10^4	8.53×10^2
Oxygen	0	32	3.16×10^4	1.04×10^3
	20	68	3.28×10^4	1.08×10^3
Sulfur dioxide	0	32	2.13×10^4	6.99×10^2
Water vapor	0	32	1.01×10^4	3.31×10^2
	100	212	1.05×10^4	3.45×10^2

24.8 AVERAGING DECIBELS

There are many occasions when the average of a number of decibel readings must be calculated. One example is when sound power level is to be determined from a number of sound pressure level readings. In such cases the average may be calculated as follows:

$$\overline{L_p} = 10 \log \left\{ \frac{1}{n} \left[10^{0.1L_1} + 10^{0.1L_2} + \cdots + 10^{0.1L_n}\right] \right\} \tag{24.4}$$

where $\overline{L_p}$ = average sound pressure level in dB
L_1 = sound pressure level at location No. 1
L_n = sound pressure level at location No. n
n = number of locations
log = logarithm to base 10

The calculation may be simplified if the difference between maximum and minimum sound pressure levels is small. In such cases arithmetic averaging may be used instead of logarithmic averaging, as follows:

If the difference between the maximum and minimum of the measured sound pressure levels is 5 dB or less, average the levels arithmetically.

If the difference between maximum and minimum sound pressure levels is between 5 and 10 dB, average the levels arithmetically and add 1 dB.

The results will usually be correct within 1 dB when compared to the average calculated by Eq. (24.4).

24.9 SOUND-LEVEL METER

The basic instrument in all sound measurements is the sound-level meter. It consists of a microphone, a calibrated attenuator, an indicating meter, and weighting networks. The meter reading is in terms of root-mean-square sound pressure level.

The A-weighting network is the one most often used. Its response characteristics approximate the response of the human ear, which is not as sensitive to low-frequency sounds as it is to high-frequency sounds. A-weighted measurements can be used for estimating annoyance caused by noise and for estimating the risk of noise-induced hearing damage. Sound levels read with the A-network are referred to as dBA.

24.10 SOUND ANALYZERS

The octave-band analyzer is the most common analyzer for industrial noise measurements. It separates complex sounds into frequency bands one octave in width, and measures the level in each of the bands.

An octave is the interval between two sounds having a frequency ratio of two. That is, the upper cutoff frequency is twice the lower cutoff frequency. The particular octaves read by the analyzer are identified by the center frequency of the octave. The center frequency of each octave is its geometric mean, or the square root of the product of the lower and upper cutoff frequencies. That is,

$$f_0 = \sqrt{f_1 f_2}$$

where f_0 = the center frequency, in Hz
f_1 = the lower cutoff frequency, in Hz
f_2 = the upper cutoff frequency, in Hz

f_1 and f_2 can be determined from the center frequency. Since $f_2 = 2f_1$ it can be shown that $f_1 = f_0/\sqrt{2}$ and $f_2 = \sqrt{2}\, f_0$.

Third-octave band analyzers divide the sound into frequency bands one-third octave in width. The upper cutoff frequency is equal to $2^{1/3}$, or 1.26, times the lower cutoff frequency.

When unknown frequency components must be identified for noise control purposes, narrow-band analyzers must be used. They are available with various bandwidths.

24.11 CORRECTION FOR BACKGROUND NOISE

The effect of ambient or background noise should be considered when measuring machine noise. Ambient noise should preferably be at least 10 dB below the machine noise. When the difference is less than 10 dB, adjustments should be made to the measured levels as shown in Table 24.4.

Table 24.4 Correction for Background Sound

Level Increase Due to the Machine (dB)	Value to Be Subtracted from Measured Level (dB)
3	3.0
4	2.2
5	1.7
6	1.3
7	1.0
8	0.8
9	0.6
10	0.5

If the difference between machine octave-band sound pressure levels and background octave-band sound pressure levels is less than 6 dB, the accuracy of the adjusted sound pressure levels will be decreased. Valid measurements cannot be made if the difference is less than 3 dB.

24.12 MEASUREMENT OF MACHINE NOISE

The noise produced by a machine may be evaluated in various ways, depending on the purpose of the measurement and the environmental conditions at the machine. Measurements are usually made in overall A-weighted sound pressure levels, plus either octave-band or third-octave-band sound pressure levels. Sound power levels are calculated from sound pressure level measurements.

24.13 SMALL MACHINES IN A FREE FIELD

A free field is one in which the effects of the boundaries are negligible, such as outdoors, or in a very large room. When small machines are sound tested in such locations, measurements at a single location are often sufficient. Many sound test codes specify measurements at a distance of 1 m from the machine.

Sound power levels, octave band, third-octave band, or A-weighted, may be determined by the following equation:

$$L_W = L_p + 20 \log r + 7.8 \tag{24.5}$$

where L_W = sound power level, in dB
L_p = sound pressure level, in dB
r = distance from source, in m
log = logarithm to base 10

24.14 MACHINES IN SEMIREVERBERANT LOCATIONS

Machines are almost always installed in semireverberant environments. Sound pressure levels measured in such locations will be greater than they would be in a free field. Before sound power levels are calculated adjustments must be made to the sound pressure level measurements.

There are several methods for determining the effect of the environment. One uses a calibrated reference sound source, with known sound power levels, in octave or third-octave bands. Sound pressure levels are measured on the machine under test, at predetermined microphone locations. The machine under test is then replaced by the reference sound source, and measurements are repeated. Sound power levels can then be calculated as follows:

$$L_{Wx} = \overline{L_{px}} + (L_{Ws} - \overline{L_{ps}}) \tag{24.6}$$

where L_{Wx} = band sound power level of the machine under test
$\overline{L_{px}}$ = average sound pressure level measured on the machine under test
L_{Ws} = band sound power level of the reference source
$\overline{L_{ps}}$ = average sound pressure level on the reference source

Another procedure for qualifying the environment uses a reverberation test. High-speed recording equipment and a special noise source are used to measure the time for the sound pressure level, originally in a steady state, to decrease 60 dB after the special noise source is stopped. This reverberation time must be measured for each frequency, or each frequency band of interest.

Unfortunately, neither of these two laboratory procedures is suitable for sound tests on large machinery, which must be tested where it is installed. This type of machinery usually cannot be shut down while tests are being made on a reference sound source, and reverberation tests cannot be made in many industrial areas because ambient noise and machine noise interfere with reverberation time measurements.

24.15 TWO-SURFACE METHOD

A procedure that can be used in most industrial areas to determine sound pressure levels and sound power levels of large operating machinery is called the two-surface method. It has definite advantages over other laboratory-type tests. The machine under test can continue to operate. Expensive, special instrumentation is not required to measure reverberation time. No calibrated reference source is needed; the machine is its own sound source. The only instrumentation required is a sound level meter and an octave-band analyzer. The procedure consists of measuring sound pressure levels on two imaginary surfaces enclosing the machine under test. The first measurement surface, S_1, is a rectangular parallelepiped 1 m away from a reference surface. The reference surface is the smallest imaginary rectangular parallelepiped that will just enclose the machine, and terminate on the reflecting plane, or floor. The area, in square meters, of the first measurement surface is given by the formula

$$S_1 = ab + 2ac + 2bc \tag{24.7}$$

where $a = L + 2$
$b = W + 2$
$c = H + 1$

and L, W, and H are the length, width, and height of the reference parallelepiped, in meters.

The second measurement surface, S_2, is a similar but larger, rectangular parallelepiped, located at some greater distance from the reference surface. The area, in square meters, of the second measurement surface is given by the formula

$$S_2 = de + 2df + 2ef \tag{24.8}$$

where $d = L + 2x$
$e = W + 2x$
$f = H + x$

and x is the distance in meters from the reference surface to S_2.

Microphone locations are usually those shown on Fig. 24.1.

First, the measured sound pressure levels should be corrected for background noise as shown in Table 24.4. Next, the average sound pressure levels, in each octave band of interest, should be calculated as shown in Eq. (24.4).

Octave-band sound pressure levels, corrected for both background noise and for the semireverberant environment, may then be calculated by the equations

$$\overline{L_p} = \overline{L_{p1}} - C \tag{24.9}$$

$$C = 10 \log \left\{ \left[\frac{K}{K-1} \right] \left[1 - \frac{S_1}{S_2} \right] \right\} \tag{24.10}$$

$$K = 10^{0.1(\overline{L_{p1}} - \overline{L_{p2}})} \tag{24.11}$$

where $\overline{L_p}$ = average octave-band sound pressure level over area S_1, corrected for both background sound and environment
$\overline{L_{p1}}$ = average octave-band sound pressure level over area S_1, corrected for background sound only
C = environmental correction
$\overline{L_{p2}}$ = average octave-band sound pressure level over area S_2, corrected for background sound

As an alternative, the environmental correction C may be obtained from Fig. 24.2.

Sound power levels, in each octave band of interest, may be calculated by the equation

$$L_W = \overline{L_p} + 10 \log \left[\frac{S_1}{S_0} \right] \tag{24.12}$$

where L_W = octave-band sound power level, in dB
$\overline{L_p}$ = average octave-band sound pressure level over area S_1, corrected for both background sound and environment
S_1 = area of measurement surface S_1, in m^2
S_0 = 1 m^2

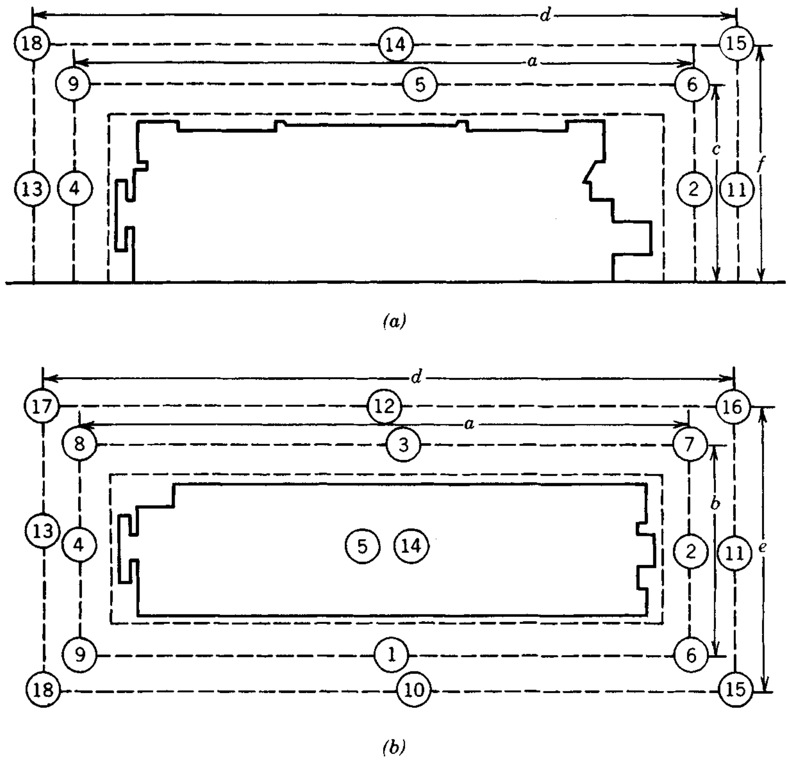

Fig. 24.1 Microphone locations: (*a*) side view; (*b*) plan view.

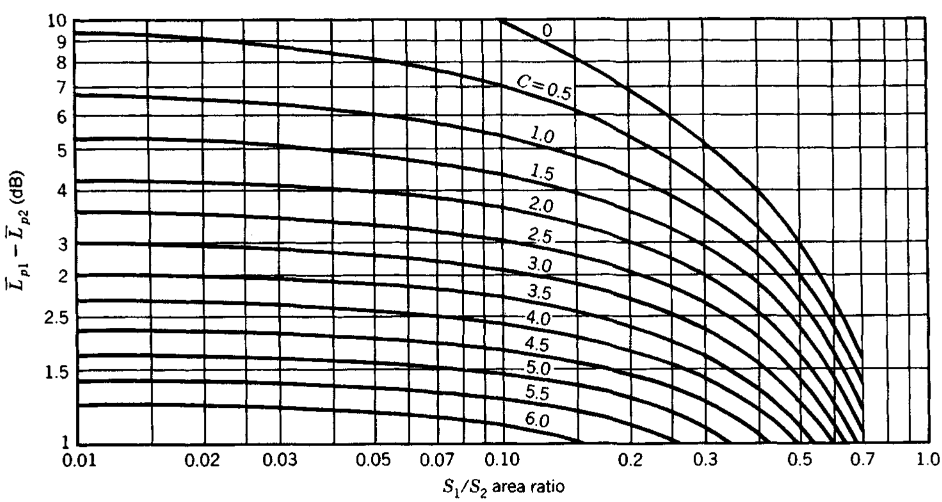

Fig. 24.2 S_1S_2 area ratio.

For simplicity, this equation can be written

$$L_W = \overline{L_p} + 10 \log S_1$$

24.16 MACHINERY NOISE CONTROL

There are five basic methods used to reduce noise: sound absorption, sound isolation, vibration isolation, vibration damping, and mufflers. In most cases several of the available methods are used in combination to achieve a satisfactory solution. Actually, most sound-absorbing materials provide some isolation, although it may be very small; and most sound-isolating materials provide some absorption, even though it may be negligible. Many mufflers rely heavily on absorption, although they are classified as a separate means of sound control.

24.17 SOUND ABSORPTION

The sound-absorbing ability of a material is given in terms of an absorption coefficient, designated by α. Absorption coefficient is defined as the ratio of the energy absorbed by the surface to the energy incident on the surface. Therefore, α can be anywhere between 0 and 1. When $\alpha = 0$, all the incident sound energy is reflected; when $\alpha = 1$, all the energy is absorbed.

The value of the absorption coefficient depends on the frequency. Therefore, when specifying the sound-absorbing qualities of a material, either a table or a curve showing α as a function of frequency is required. Sometimes, for simplicity, the acoustical performance of a material is stated at 500 Hz only, or by a noise reduction coefficient (NRC) that is obtained by averaging, to the nearest multiple of 0.05, the absorption coefficients at 250, 500, 1000, and 2000 Hz.

The absorption coefficient varies somewhat with the angle of incidence of the sound wave. Therefore, for practical use, a statistical average absorption coefficient at each frequency is usually measured and stated by the manufacturer. It is often better to select a sound-absorbing material on the basis of its characteristics for a particular noise rather than by its average sound-absorbing qualities.

Sound absorption is a function of the length of path relative to the wavelength of the sound, and not the absolute length of the path of sound in the material. This means that at low frequencies the thickness of the material becomes important, and absorption increases with thickness. Low-frequency absorption can be improved further by mounting the material at a distance of one-quarter wavelength from a wall, instead of directly on it.

Table 24.5 shows absorption coefficients of various materials used in construction.

The sound absorption of a surface, expressed in either square feet of absorption, or sabins, is equal to the area of the surface, in square feet, times the absorption coefficient of the material on the surface.

Average absorption coefficient, $\bar{\alpha}$, is calculated as follows:

$$\bar{\alpha} = \frac{\alpha_1 S_1 + \alpha_2 S_2 + \cdots + \alpha_n S_n}{S_1 + S_2 + \cdots + S_n} \tag{24.13}$$

Table 24.5 Absorption Coefficients

Material	125 cps	250 cps	500 cps	1000 cps	2000 cps	4000 cps
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
Concrete block	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
Concrete	0.01	0.01	0.015	0.02	0.02	0.02
Wood	0.15	0.11	0.10	0.07	0.06	0.07
Glass, ordinary window	0.35	0.25	0.18	0.12	0.07	0.04
Plaster	0.013	0.015	0.02	0.03	0.04	0.05
Plywood	0.28	0.22	0.17	0.09	0.10	0.11
Tile	0.02	0.03	0.03	0.03	0.03	0.02
6 lb/ft^2 fiberglass	0.48	0.82	0.97	0.99	0.90	0.86

where $\overline{\alpha}$ = the average absorption coefficient
$\alpha_1, \alpha_2, \alpha_n$ = the absorption coefficients of materials on various surfaces
S_1, S_2, S_n = the areas of various surfaces

24.18 NOISE REDUCTION DUE TO INCREASED ABSORPTION IN ROOM

A machine in a large room radiates noise that decreases at a rate inversely proportional to the square of the distance from the source. Soon after the machine is started the sound wave impinges on a wall. Some of the sound energy is absorbed by the wall, and some is reflected. The sound intensity will not be constant throughout the room. Close to the machine the sound field will be dominated by the source, almost as though it were in a free field, while farther away the sound will be dominated by the diffuse field, caused by sound reflections. The distance where the free field and the diffuse field conditions control the sound depends on the average absorption coefficient of the surfaces of the room and the wall area. This critical distance can be calculated by the following equation:

$$r_c = 0.2\sqrt{R} \tag{24.14}$$

where r_c = distance from source, in m
R = room constant of the room, in m^2

Room constant is equal to the product of the average absorption coefficient of the room and the total internal area of the room divided by the quantity one minus the average absorption coefficient. That is,

$$R = \frac{S_t\overline{\alpha}}{1 - \overline{\alpha}} \tag{24.15}$$

where R = the room constant, in m^2
$\overline{\alpha}$ = the average absorption coefficient
S_t = the total area of the room, in m^2

Essentially free-field conditions exist farther from a machine in a room with a large room constant than they do in a room with a small room constant.

The distance r_c determines where absorption will reduce noise in the room. An operator standing close to a noisy machine will not benefit by adding sound-absorbing material to the walls and ceiling. Most of the noise heard by the operator is radiated directly by the machine, and very little is reflected noise. On the other hand, listeners farther away, at distances greater than r_c, will benefit from the increased absorption.

The noise reduction in those areas can be estimated by the following equation:

$$NR = 10\log\frac{\overline{\alpha}_2 S}{\overline{\alpha}_1 S} \tag{24.16}$$

where NR = far field noise reduction, in dB
$\overline{\alpha}_1 S$ = room absorption before treatment
$\overline{\alpha}_2 S$ = room absorption after treatment

Equation (24.16) shows that doubling the absorption will reduce noise by 3 dB. It requires another doubling of the absorption to get another 3 dB reduction. This is much more difficult than getting the first doubling, and considerably more expensive.

24.19 SOUND ISOLATION

Noise may be reduced by placing a barrier or wall between a noise source and a listener. The effectiveness of such a barrier is described by its transmission coefficient.

Sound transmission coefficient of a partition is defined as the fraction of incident sound transmitted through it.

Sound transmission loss is a measure of sound-isolating ability, and is equal to the number of decibels by which sound energy is reduced in transmission through a partition. By definition, it is 10 times the logarithm to the base 10 of the reciprocal of the sound transmission coefficient. That is,

$$TL = 10\log\frac{1}{\tau} \tag{24.17}$$

Table 24.6 Transmission Loss of Building Materials

Item	*TL*
Hollow-core door (3⁄16-in. panels)	15
1¾-in. solid-core oak door	20
2½-in. heavy wood door	25–30
4-in. cinder block	20–25
4-in. cinder block, plastered	40
4-in. cinder slab	40–45
4-in. slab-suspended concrete, plastered	50
Two 4-in. cinder blocks—4-in. air space	55
4-in. brick	45
4-in. brick, plastered	47
8-in. brick, plastered	50
Two 8-in. cinder block—4-in. air space	57

where TL = the transmission loss, in dB
τ = the transmission coefficient

Transmission of sound through a rigid partition or solid wall is accomplished mainly by the forced vibration of the wall. That is, the partition is forced to vibrate by the pressure variations in the sound wave.

Under certain conditions porous materials can be used to isolate high-frequency sound, and, in general, the loss provided by a uniform porous material is directly proportional to the thickness of the material. For most applications, however, sound absorbing materials are very ineffective sound isolators because they have the wrong characteristics. They are porous, instead of airtight, and they are lightweight, instead of heavy. The transmission loss of nonporous materials is determined by weight per square foot of surface area, and how well all cracks and openings are sealed. Transmission loss is affected also by dynamic bending stiffness and internal damping. Table 24.6 shows the transmission loss of various materials used in construction.

24.20 SINGLE PANEL

The simplest type of sound-isolating barrier is a single, homogeneous, nonporous partition. In general, the transmission loss of a single wall of this type is proportional to the logarithm of the mass. Its isolating ability also increases with frequency, and the approximate relationship is given by the following equation:

$$TL = 20 \log W + 20 \log f - 33 \qquad (24.18)$$

where TL = the transmission loss, in dB
W = the surface weight, in pounds per square foot
f = the frequency, in Hz

This means that the transmission loss increases 6 dB each time the weight is doubled, and 6 dB each time the frequency is doubled. In practice, these numbers are each about 5 dB, instead of 6 dB.

In general, a single partition or barrier should not be counted on to provide noise reduction of more than about 10 dB.

24.21 COMPOSITE PANEL

Many walls or sound barriers are made of several different materials. For example, machinery enclosures are commonly constructed of sheet steel, but they may have a glass window to observe instruments inside the enclosure. The transmission coefficients of the two materials are different. Another example is when there are necessary cracks or openings in the enclosure where it fits around a rotating shaft. In this case, the transmission loss of the opening is zero, and the transmission coefficient is 1.0.

The effectiveness of such a wall is related to both the transmission coefficients of the materials in the wall and the areas of the sections. A large area transmits more noise than a small one made of the same material. Also, more noise can be transmitted by a large area with a relatively small transmission coefficient than by a small one with a comparatively high transmission coefficient. On the other hand, a small area with a high transmission coefficient can ruin the effectiveness of an otherwise excellently designed enclosure. Both transmission coefficients and areas must be controlled carefully.

The average transmission coefficient of a composite panel is

$$\bar{\tau} = \frac{\tau_1 S_1 + \tau_2 S_2 + \tau_3 S_3 + \cdots + \tau_n S_n}{S_1 + S_2 + S_3 + \cdots + S_n} \tag{24.19}$$

where $\bar{\tau}$ = the average transmission coefficient
$\tau_1, \ldots, \tau_n$ = the transmission coefficients of the various areas
$S_1, \ldots, S_n$ = the various areas

Figure 24.3 shows how the transmission loss of a composite wall or panel may be determined from the transmission loss values of its parts. It also shows the damaging effect of small leaks in the enclosure. In this case the transmission loss of the leak opening is zero.

A leak of only 0.1% in an expensive, high-quality door or barrier, constructed of material with a transmission loss of 50 dB, would reduce the transmission-loss by 20 dB, resulting in a *TL* of only 30 dB instead of 50 dB. A much less expensive 30 dB barrier would be reduced by only 3 dB, resulting in a *TL* of 27 dB. This shows that small leaks are more damaging to a high-quality enclosure than to a lower-quality one.

24.22 ACOUSTIC ENCLOSURES

When machinery noise must be reduced by 20 dB or more, it is usually necessary to use complete enclosures. It must be kept in mind that the actual decrease in noise produced by an enclosure depends on other things as well as the transmission loss of the enclosure material. Vibration resonances must be avoided, or their effects must be reduced by damping; structural and mechanical connections must not be permitted to short circuit the enclosure; and the enclosure must be sealed as well as possible to prevent acoustic leaks. In addition, the actual noise reduction depends on the acoustic properties of the room in which the enclosure is located. For this reason, published data on transmission loss of various materials should not be assumed to be the same as the noise reduction that will be obtained when using those materials in enclosures. How materials are used in machinery enclosures is just as important as which materials are used.

A better description of the performance of an acoustic enclosure is given by its noise reduction *NR*, which is defined as the difference in sound pressure level between the enclosure and the receiving room. The relation between noise reduction and transmission loss is given by the following equation:

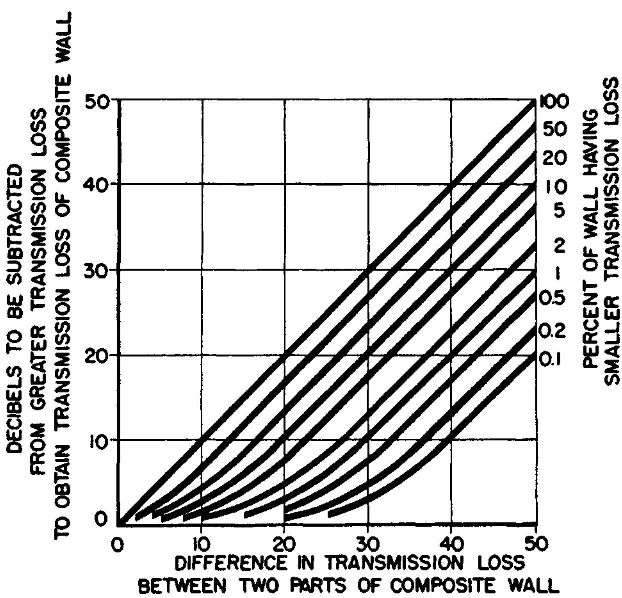

Fig. 24.3 Decibels to be subtracted from greater transmission loss to obtain transmission loss of composite wall. The difference in transmission loss between two parts of composite wall. Percentage of wall having smaller transmission loss.

$$NR = TL - 10 \log \left[\frac{1}{4} + \frac{S_{\text{wall}}}{R_{\text{room}}} \right] \tag{24.20}$$

where NR = the difference in sound pressure level, in dB, between the enclosure and the receiving room
TL = the transmission loss of the enclosure walls, in dB
S_{wall} = the area of the enclosure walls, in square feet
R_{room} = the room constant of the receiving room, in square feet

Equation (24.20) indicates that if the room constant is very large, like it would be outdoors, or in a very large area with sound-absorbing material on the walls and ceiling, the NR could exceed the TL by almost 6 dB. In most industrial areas the NR is approximately equal to, or is several dB less than, the TL.

If there is no absorption inside the enclosure, and it is highly reverberant, like smooth sheet steel, sound reflects back and forth many times. As the noise source continues to radiate noise, with none of it being absorbed, the noise continues to increase without limit. Theoretically, with zero absorption, the sound level will increase to such a value that no enclosure can contain it.

Practically, this condition cannot exist; there will always be some absorption present, even though it may be very little. However, the sound level inside the enclosure will be greater than it would be without the enclosure. For this reason, the sound level inside the enclosure should be assumed to equal the actual noise source plus 10 dB, unless a calculation shows it to be otherwise.

When absorbing material is added to the inside of the enclosure, sound energy decreases each time it is reflected.

An approximate method for estimating the noise reduction of an enclosure is

$$NR = 10 \log \left[1 + \frac{\overline{\alpha}}{\overline{\tau}} \right] \tag{24.21}$$

where NR = the noise reduction, in dB
$\overline{a}$ = the average absorption coefficient of the inside of the enclosure
$\overline{\tau}$ = the average transmission coefficient of the enclosure

Equation (24.21) shows that in the theoretical case where there is no sound absorption, there is no noise reduction.

24.23 DOUBLE WALLS

A 4-in.-thick brick wall has a transmission loss of about 45 dB. An 8-in.-thick brick wall, with twice as much weight, has a transmission loss of about 50 dB. After a certain point has been reached it is found to be impractical to try to obtain higher isolation values simply by doubling the weight, since both the weight and the cost become excessive, and only a 5 dB improvement is gained for each doubling of weight.

An increase can be obtained, however, by using double-wall construction. That is, two 4-in.-thick walls separated by an air space are better than one 8-in. wall. However, noise radiated by the first panel can excite vibration of the second one and cause it to radiate noise. If there are any mechanical connections between the two panels, vibration of one directly couples to the other, and much of the benefit of double-wall construction is lost.

There is another factor that can reduce the effectiveness of double-wall construction. Each of the walls represents a mass, and the air space between them acts as a spring. This mass–spring–mass combination has a series of resonances that greatly reduce the transmission loss at the corresponding frequencies. The effect of the resonances can be reduced by adding sound-absorbing material in the space between the panels.

24.24 VIBRATION ISOLATION

There are many instances where airborne sound can be reduced substantially by isolating a vibrating part from the rest of the structure. A vibration isolator, in its simplest form is some type of resilient support. The purpose of the isolator may be to reduce the magnitude of force transmitted from a vibrating machine or part of a machine to its supporting structure. Conversely, its purpose may be to reduce the amplitude of motion transmitted from a vibrating support to a part of the system that is radiating noise due to its vibration.

Vibration isolators can be in the form of steel springs, cork, felt, rubber, plastic, or dense fiberglass. Steel springs can be calculated quite accurately and can do an excellent job of vibration isolation. However, they also can have resonances, and high-frequency vibrations can travel through them readily, even though they are effectively isolating the lower frequencies. For this reason, springs are usually used in combination with elastomers or similar materials. Elastomers, plastics, and materials of this type have high internal damping and do not perform well below about 15 Hz. However, this is below the audible range, and, therefore, it does not limit their use in any way for effective sound control.

The noise reduction that can be obtained by installing an isolator depends on the characteristics of the isolator and the associated mechanical structure. For example, the attenuation that can be obtained by spring isolators depends not only on the spring constant, or spring stiffness (the force necessary to stretch or compress the spring one unit of length), but also on the mass load on the spring, the mass and stiffness of the foundation, and the type of excitation.

If the foundation is very massive and rigid, and if the mounted machine vibrates at constant amplitude, the reduction in force on the foundation is independent of frequency. If the machine vibrates at a constant force, the reduction in force depends on the ratio of the exciting frequency to the natural frequency of the system.

When a vibrating machine is mounted on an isolator, the ratio of the force applied to the isolator by the machine, to the force transmitted by the isolator, to the foundation is called the "transmissibility." That is,

$$\text{transmissibility} = \frac{\text{transmitted force}}{\text{impressed force}}$$

Under ideal conditions this ratio would be zero. In practice, the objective is to make it as small as possible. This can be done by designing the system so that the natural frequency of the mounted machine is very low compared to the frequency of the exciting force.

If no damping is present, the transmissibility can be expressed by the following equation:

$$T = \frac{1}{1 - (\omega/\omega_n)^2} \tag{24.22}$$

where T = the transmissibility, expressed as a fraction
ω = the circular frequency of the exciting force, in radians per second
ω_n = the circular frequency of the mounted system, in radians per second

When $\omega/\omega_n = 0$, the transmissibility equals 1.0. That is, there is no benefit obtained from the isolator.

If ω/ω_n is greater than zero but less than 1.41, the isolator actually increases the magnitude of the transmitted force. This is called the "region of amplification." In fact, when ω/ω_n equals 1.0, the theoretical amplitude of the transmitted force goes to infinity, since this is the point where the frequency of the disturbing force equals the system natural frequency.

Equation (24.22) indicates that the transmissibility becomes negative when ω/ω_n is greater than 1.0. The negative number is simply due to the phase relation between force and motion, and it can be disregarded when considering only the amount of transmitted force.

Since vibration isolation is achieved only when ω/ω_n is greater than 1.41, the equation for transmissibility can be written so that T is positive:

$$T = \frac{1}{(\omega/\omega_n)^2 - 1}$$

Also, since $\omega = 2\pi f$,

$$T = \frac{1}{(f/f_n)^2 - 1} \tag{24.23}$$

The static deflection of a spring when stretched or compressed by a weight is related to its natural frequency by the equation

$$f_n = 3.14\sqrt{\frac{1}{d}} \tag{24.24}$$

where f_n = the natural frequency, in hertz
d = the deflection, in inches

When this is substituted in the equation for transmissibility, Eq. (24.23), it can be shown that

$$d = \left(\frac{3.14}{f}\right)^2\left(\frac{1}{T} + 1\right) \tag{24.25}$$

This shows that the transmissibility can be determined from the deflection of the isolator due to its supported load.

Equations (24.23) and (24.25) can be plotted, as shown in Fig. 24.4, for convenience in selecting isolator natural frequencies or deflections.

For critical applications, the natural frequency of the isolator should be about one-tenth to one-sixth of the disturbing frequency. That is, the transmissibility should be between 1 and 3%. For less critical conditions, the natural frequency of the isolator should be about one-sixth to one-third of the driving frequency, with transmissibility between 3 and 12%.

24.25 VIBRATION DAMPING

Complex mechanical systems have many resonant frequencies, and whenever an exciting frequency is coincident with one of the resonant frequencies, the amplitude of vibration is limited only by the amount of damping in the system. If the exciting force is wide band, several resonant vibrations can occur simultaneously, thereby compounding the problem. Damping is one of the most important factors in noise and vibration control.

There are three kinds of damping. Viscous damping is the type that is produced by viscous resistance in a fluid, for example, a dashpot. The damping force is proportional to velocity. Dry friction, or Coulomb damping, produces a constant damping force, independent of displacement and velocity. The damping force is produced by dry surfaces rubbing together, and it is opposite in direction to that of the velocity. Hysteresis damping, also called material damping, produces a force that is in phase with the velocity but is proportional to displacement. This is the type of damping found in solid materials, such as elastomers, widely used in sound control.

A large amount of noise radiated from machine parts comes from vibration of large areas or panels. These parts may be integral parts of the machine or attachments to the machine. They can be flat or curved, and vibration can be caused by either mechanical or acoustic excitation. The radiated noise is a maximum when the parts are vibrating in resonance.

When the excitation is mechanical, vibration isolation may be all that is needed. In other instances, the resonant response can be reduced by bonding a layer of energy-dissipating polymeric material to the structure. When the structure bends, the damping material is placed alternately in tension and compression, thus dissipating the energy as heat.

This extensional or free-layer damping is remarkably effective in reducing resonant vibration and noise in relatively thin, lightweight structures such as panels. It becomes less effective as the structure stiffness increases, because of the excessive increase in thickness of the required damping layer.

In a vibrating structure, the amount of energy dissipated is a function of the amount of energy necessary to deflect the structure, compared to that required to deflect the damping material. If 99% of the vibration energy is required to deflect the structure, and 1% is required to deflect the damping layer, then only 1% of the vibration energy is dissipated.

Resonant vibration amplitude in heavier structures can be controlled effectively by using constrained-layer damping. In this method, a relatively thin layer of viscoelastic damping material is constrained between the structure and a stiff cover plate. Vibration energy is removed from the system by the shear motion of the damping layer.

24.26 MUFFLERS

Silencers, or mufflers, are usually divided into two categories: absorptive and reactive. The absorptive type, as the name indicates, removes sound energy by the use of sound-absorbing materials. They have relatively wide-band noise-reduction characteristics, and are usually applied to problems associated with continuous spectra, such as fans, centrifugal compressors, jet engines, and gas turbines. They are also used in cases where a narrow-band noise predominates, but the frequency varies because of a wide range of operating speed.

A variety of sound-absorbing materials are used in many different configurations, determined by the level of the unsilenced noise and its frequency content, the type of gas being used, the allowable pressure drop through the silencer, the gas velocity, gas temperature and pressure, and the noise criterion to be met.

Fiberglass or mineral wool with density approximately 0.5–6.0 lb/ft^3 is frequently used in absorptive silencers. These materials are relatively inexpensive and have good sound-absorbing characteristics. They operate on the principle that sound energy causes the material fibers to move, converting the sound energy into mechanical vibration and heat. The fibers do not become very warm since the sound energy is actually quite low, even at fairly high decibel levels.

The simplest kind of absorptive muffler is a lined duct, where the absorbing material is either added to the inside of the duct walls or the duct walls themselves are made of sound-absorbing material. The attenuation depends on the duct length, thickness of the lining, area of the air passage, type of absorbing material, and frequency of the sound passing through.

The acoustical performance of absorptive mufflers is improved by adding parallel or annular baffles to increase the amount of absorption. This also increases pressure drop through the muffler, so that spacing and area must be carefully controlled.

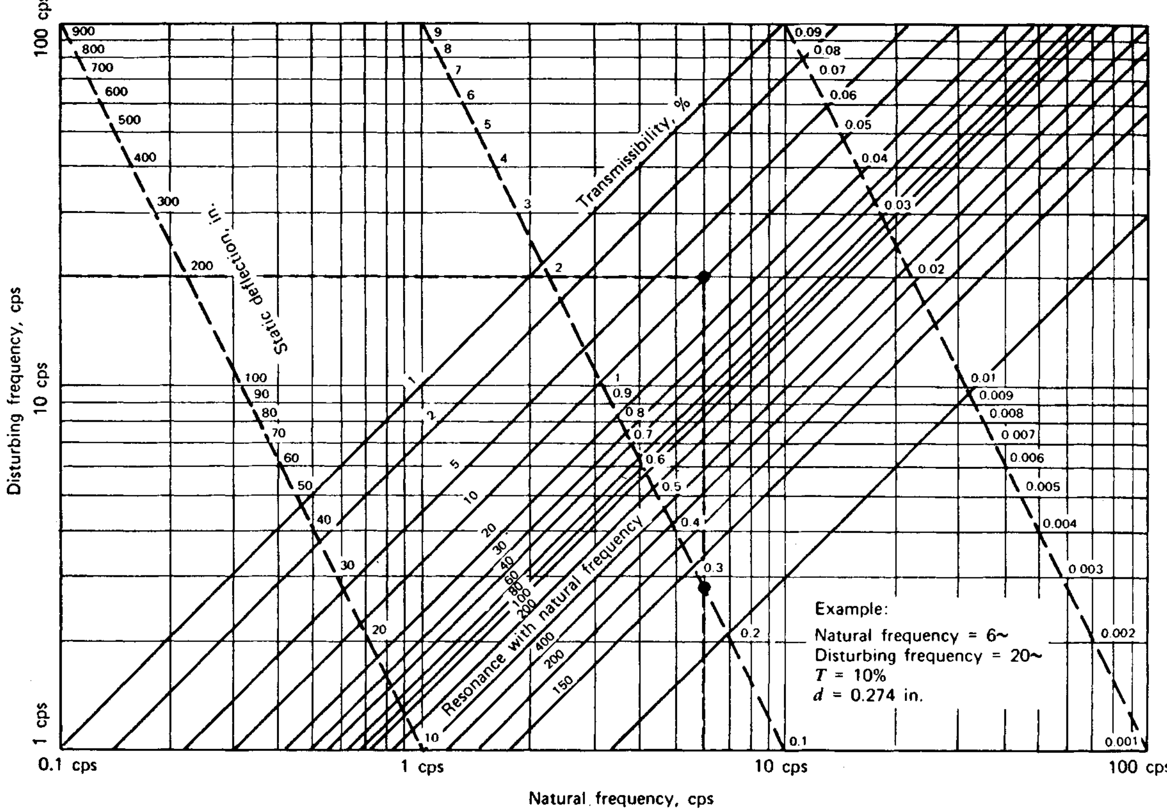

Fig. 24.4 Transmissibility of flexible mountings: $T = 1/[(\omega/\omega)^2 - 1]$.

Reactive mufflers have a characteristic performance that does not depend to any great extent on the presence of sound-absorbing material, but utilizes the reflection characteristics and attenuating properties of conical connectors, expansion chambers, side branch resonators, tail pipes, and so on, to accomplish sound reduction.

Expansion chambers operate most efficiently in applications involving discrete frequencies rather than broad-band noise. The length of the chamber is adjusted so that reflected waves cancel the incident waves, and since wavelength depends on frequency, expansion chambers should be tuned to some particular frequency. When a number of discrete frequencies must be attenuated, several expansion chambers can be placed in series, each tuned to a particular wavelength.

An effective type of reactive muffler, called a Helmholtz resonator, consists of a vessel containing a volume of air, that is connected to a noise source, such as a piping system. When a pure-tone sound wave is propagated along the pipe, the air in the vessel expands and contracts. By proper design of the area and length of the neck, and volume of the chamber, sound wave cancellation can be obtained, thereby reducing the tone. This type of resonator produces maximum noise reduction over a very narrow frequency range, but it is possible to combine several Helmholtz resonators on a piping system so that not only will each cancel out at its own frequency, but they can be made to overlap so that noise is attenuated over a wider range instead of at sharply tuned points.

Helmholtz resonators are normally located in side branches, and for this reason they do not affect flow in the main pipe.

The resonant frequency of these devices can be calculated by the equation

$$f = \frac{C}{2\pi}\sqrt{\frac{A}{LV}} \tag{24.26}$$

where f = the resonant frequency, in hertz
C = the speed of sound in the fluid, in feet per second
A = the cross-sectional area of the neck, in square feet
L = the length of the neck, in feet
V = the volume of the chamber, in cubic feet

The performance of all types of mufflers can be stated in various ways. Not everyone uses the same terminology, but in general, the following definitions apply:

INSERTION LOSS is defined as the difference between two sound pressure levels measured at the same point in space before and after a muffler is inserted in the system.

DYNAMIC INSERTION LOSS is the same as insertion loss, except that it is measured when the muffler is operating under rated flow conditions. Therefore, the dynamic insertion loss is of more interest than ratings based on no-flow conditions.

TRANSMISSION LOSS is defined as the ratio of sound power incident on the muffler to the sound power transmitted by the muffler. It cannot be measured directly, and it is difficult to calculate analytically. For these reasons the transmission loss of a muffler has little practical application.

ATTENUATION is used to describe the decrease in sound power as sound waves travel through the muffler. It does not convey information about how a muffler performs in a system.

NOISE REDUCTION is defined as the difference between sound pressure levels measured at the inlet of a muffler and those at the outlet.

24.27 SOUND CONTROL RECOMMENDATIONS

Sound control procedures should be applied during the design stages of a machine, whenever possible. A list of recommendations for noise reduction follows:

1. Reduce horsepower. Noise is proportional to horsepower. Therefore, the machine should be matched to the job. Excess horsepower means excess noise.
2. Reduce speed. Slow-speed machinery is quieter than high-speed machinery.
3. Keep impeller tip speeds low. However, it is better to keep the rpm low and the impeller diameter large than to keep the rpm high and the impeller diameter small, even though the tip speeds are the same.
4. Improve dynamic balance. This decreases rotating forces, structure-borne sound, and the excitation of structural resonances.
5. Reduce the ratio of rotating masses to fixed masses.
6. Reduce mechanical run-out of shafts. This improves the initial static and dynamic balance.
7. Avoid structural resonances. These are often responsible for many unidentified components in the radiated sound. In addition to being excited by sinusoidal forcing frequencies, they can be excited by impacting parts and sliding and rubbing contacts.

8. Eliminate or reduce impacts. Either reduce the mass of impacting parts or their striking velocities.
9. Reduce peak acceleration. Reduce the rate of change of velocity of moving parts by using the maximum time possible to produce the required velocity change, and by keeping the acceleration as nearly constant as possible over the available time period.
10. Improve lubrication. Inadequate lubrication is often the cause of bearing noise, structure-borne noise due to friction, and the excitation of structural resonances.
11. Maintain closer tolerances and clearances in bearings and moving parts.
12. Install bearings correctly. Improper installation accounts for approximately half of bearing noise problems.
13. Improve alignment. Improper alignment is a major source of noise and vibration.
14. Use center of gravity mounting whenever feasible. When supports are symmetrical with respect to the center of gravity, translational modes of vibration do not couple to rotational modes.
15. Maintain adequate separation between operating speeds and lateral and torsional resonant speeds.
16. Consider the shape of impeller vanes from an acoustic standpoint. Some configurations are noisier than others.
17. Keep the distance between impeller vanes and cutwater or diffuser vanes as large as possible. Close spacing is a major source of noise.
18. Select combinations of rotating and stationary vanes that are not likely to excite strong vibration and noise.
19. Design turning vanes properly. They are a source of self-generated noise.
20. Keep the areas of inlet passages as large as possible and their length as short as possible.
21. Remove or keep at a minimum any obstructions, bends, or abrupt changes in fluid passages.
22. Pay special attention to inlet design. This is extremely important in noise generation.
23. Item 22 applies also to the discharge, but the inlet is more important than the discharge from an acoustic standpoint.
24. Maintain gradual, not abrupt, transition from one area to the next in all fluid passages.
25. Reduce flow velocities in passages, pipes, and so on. Noise can be reduced substantially by reducing flow velocities.
26. Reduce jet velocities. Jet noise is proportional to the eighth power of the velocity.
27. Reduce large radiating areas. Surfaces radiating certain frequencies can often be divided into smaller areas with less radiating efficiency.
28. Disconnect possible sound radiating parts from other vibrating parts by installing vibration breaks to eliminate metal to metal contact.
29. Provide openings or air leaks in large radiating areas so that air can move through them. This reduces pressure build-up and decreases radiated noise.
30. Reduce clearances, piston weights, and connecting rod weights in reciprocating machinery to reduce piston impacts.
31. Apply additional sound control devices, such as inlet and discharge silencers and acoustic enclosures.
32. When acoustic enclosures are used, make sure that all openings are sealed properly.
33. Install machinery on adequate mountings and foundations to reduce structure-borne sound and vibration.
34. Take advantage of all directivity effects whenever possible by directing inlet and discharge openings away from listeners or critical areas.
35. When a machine must meet a particular sound specification, purchase driving motors, turbines, gears, and auxiliary equipment that produce 3- to 5-dB lower sound levels than the machine alone. This ensures that the combination is in compliance with the specification.