Introduction to Building Acoustics and Noise Control

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An Introduction to Building Acoustics and Noise Control

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1. INTRODUCTION

This is an introduction to the fundamentals of acoustics and noise control in buildings. It is not an in-depth treatment, but it will introduce designers to some important principles and terminology. In simple applications on real projects the information provided here will give designers a good start in addressing acoustic control issues. For more acoustically complex projects designers will need to apply more detailed principles. There are excellent acoustical engineering treatises available commercially and from government agencies to provide practitioners with the necessary guidance in applying these more rigorous methods.

2. FUNDAMENTALS OF ACOUSTICS

2.1 This discussion presents the basic quantities used to describe acoustical properties. For the purposes of the material contained in this document perceptible acoustical sensations can be generally classified into two broad categories, these are:

- ***Sound.*** A disturbance in an elastic medium resulting in an audible sensation. Noise is by definition “unwanted sound”.

- ***Vibration.*** A disturbance in a solid elastic medium which may produce a detectable motion.

2.2 Although this differentiation is useful in presenting acoustical concepts, in reality sound and vibration are often interrelated. That is, sound is often the result of acoustical energy radiation from vibrating structures and, sound can force structures to vibrate. Acoustical energy can be completely characterized by the simultaneous determination of three qualities. These are:

- ***Level or Magnitude.*** This is a measure of the intensity of the acoustical energy.

- ***Frequency or Spectral Content.*** This is a description of an acoustical energy with respect to frequency composition.

- ***Time or Temporal Variations.*** This is a description of how the acoustical energy varies with respect to time.

2.3 The subsequent material defines the measurement parameters for each of these qualities that are used to evaluate sound and vibration.

2.4 Decibels

The basic unit of level in acoustics is the “decibel” (abbreviated dB). In acoustics, the term “level” is used to designated that the quantity is referred to some reference value, which is either stated or implied.

2.4.1 Definition and use. The decibel (dB), as used in acoustics, is a unit expressing the ratio of two quantities that are proportional to power. The decibel level is equal to 10 times the common logarithm of the power ratio; or

$$dB = 10 \log \frac{P_1}{P_2} \qquad \text{(eq 2-1)}$$

In this equation P_2 is the absolute value of the power under evaluation and P_1 is an absolute value of a power reference quantity with the same units. If the power P_1 is the accepted standard reference value, the decibels are standardized to that reference value. In acoustics, the decibel is used to quantify sound pressure levels that people hear, sound power levels radiated by sound sources, the sound transmission loss through a wall, and in other uses, such as simply “a noise reduction of 15 dB” (a reduction relative to the original sound level condition). Decibels are always related to logarithms to the base 10, so the notation 10 is usually omitted. It is important to realize that the decibel is in reality a dimensionless quantity (somewhat analogous to “percent”). Therefore when using decibel levels, reference needs to be made to the quantity under evaluation and the reference level. It is also instructive to note that the decibel level is primarily determined by the magnitude of the absolute value of the power level. That is, if the magnitude of two different power levels differs by a factor of 100 then the decibel levels differ by 20 dB.

2.4.2 Decibel addition. In many cases cumulative effects of multiple acoustical sources have to be evaluated. In this case the individual sound levels should be summed. Decibel levels are added logarithmically and not algebraically. For example, 70 dB plus 70 dB does not equal 140 dB, but only 73 dB. A very simple, but usually adequate, schedule for obtaining the sum of two decibel values is:

When two decibel values differ by	Add the following amount to the higher value
0 or 1 dB	3 dB
2 or 3 dB	2 dB
4 to 9 dB	1 dB
10 dB or more	0 dB

When several decibel values to be added equation 2-2 should be used.

$$L_{sum} = 10 \log \left[10^{\frac{L_{p1}}{10}} + 10^{\frac{L_{p2}}{10}} + \ldots + 10^{\frac{L_{pn}}{10}} \right]$$

(eq 2-2)

In the special case where decibel levels of equal magnitudes are to be added, the cumulative level can be determined with equation 2-3:

$$L_{sum} = L_p + 10 \log (n)$$

(eq 2-3)

where n is the number of sources, all with magnitude Lp.

2.4.3 Decibel subtraction. In some case it is necessary to subtract decibel levels. For example if the cumulative level of several sources is known, what would the cumulative level be if one of the sources were reduced? Decibel subtraction is given by equation 2-4.

$$L_{diff}=10\log\left[10^{\frac{L_{p1}}{10}} - 10^{\frac{L_{p2}}{10}}\right] \qquad \text{(eq 2-4)}$$

2.4.4 Decibel averaging. Strictly speaking decibels should be averaged logarithmically not arithmetically. Equation 2-5 should be used for decibel averaging.

$$L_{avg}=10\log\left[\frac{10^{\frac{L_{p1}}{10}} + 10^{\frac{L_{p2}}{10}} + \ldots + 10^{\frac{L_{pn}}{10}}}{n}\right] \qquad \text{(eq 2-5)}$$

2.5 Sound Pressure Level (L_p or SPL).

The ear responds to sound pressure. Sound waves represent tiny oscillations of pressure just above and below atmospheric pressure. These pressure oscillations impinge on the ear, and sound is heard. A sound level meter is also sensitive to sound pressure.

2.5.1 Definition, sound pressure level. The sound pressure level (in decibels) is defined by:

$$L_p=10\log\left[\left(\frac{p}{p_{ref}}\right)^2\right] \qquad \text{(eq 2-6)}$$

Where p is the absolute level of the sound pressure and p_{ref} is the reference pressure. Unless otherwise stated the pressure, p, is the effective root mean square (rms) sound pressure. This equation is also written as:

$$L_p=20\log\left[\left(\frac{p}{p_{ref}}\right)\right] \qquad \text{(eq 2-7)}$$

Although both formulas are correct, it is instructive to consider sound pressure level as the log of the pressure squared (eq 2-6). This is because when combining sound pressure levels, in almost all cases, it is the square of the pressure ratios (i.e. $\{p/P_{ref})^2\}$'s) that should be summed not the pressure ratios (i.e. not the $\{p/P_{ref}\}$'s). This is also true for sound pressure level subtraction and averaging.

2.5.2 Definition, reference pressure. Sound pressure level, expressed in decibels, is the logarithmic ratio of pressures where the reference pressure is 20 micropascal or 20 uPa (Pascal, the unit of pressure, equals 1 Newton/m^2). This reference pressure represents approximately the faintest sound that can be heard by a young, sensitive, undamaged human ear when the sound occurs in the frequency region of maximum hearing sensitivity, about 1000 Hertz (Hz). A 20 uPa pressure is 0 dB on the sound pressure level scale. In the strictest sense, a sound pressure level should be stated completely, including the reference pressure base, such as "85 decibels relative to 20 uPa." However, in normal practice and in this discussion the reference pressure is omitted, but it is nevertheless implied.

2.5.3 Abbreviations. The abbreviation SPL is often used to represent sound pressure level, and the notation L_p is normally used in equations, both in this discussion and in the general acoustics literature.

2.5.4 Limitations on the use of sound pressure levels. Sound pressure levels can be used for evaluating the effects of sound with respect to sound level criteria. Sound pressure level data taken under certain installation conditions cannot be used to predict sound pressure levels under other installation conditions unless modifications are applied. Implicit in these modifications is a sound power level calculation.

2.6 Sound Power Level (L_w or PWL)

Sound power level is an absolute measure of the quantity of acoustical energy produced by a sound source. Sound power is not audible like sound pressure. However they are related. It is the manner in which the sound power is radiated and distributed that determines the sound pressure level at a specified location. The sound power level, when correctly determined, is an indication of the sound radiated by the source and is independent of the room containing the source. The sound power level data can be used to compare sound data submittals more accurately and to estimate sound pressure levels for a variety of room conditions. Thus, there is technical need for the generally higher quality sound power level data.

2.6.1 Definition, sound power level. The sound power level (in decibels) is defined by:

$$L_w = 10 \log \frac{P}{p_{ref}} \qquad \text{(eq 2-8)}$$

Where P is the absolute level of the sound power and P_{ref} is the reference power. Unless otherwise stated the power, P, is the effective root mean square (rms) sound power.

2.6.2 Definition, reference power. Sound power level, expressed in decibels, is the logarithmic ratio of the sound power of a source in watts (W) relative to the sound power reference base of 10-12 W. Before the US joined the IS0 in acoustics terminology, the reference power in this country was 10-13W, so it is important in using old data (earlier than about 1963) to ascertain the power level base that was used. If the sound power level value is expressed in dB relative to 10-13W, it can be converted to dB relative to 10-12W, by subtracting 10 dB from the value. Special care must be used not to confuse decibels of sound pressure with decibels of sound power. It is often recommended that power level values always be followed by the notation “dB re 10-12W.” However, in this discussion this notation is omitted, although it will always be made clear when sound power levels are used.

2.6.3 Abbreviations. The abbreviation PWL is often used to represent sound power level, and the notation L_w normally used in equations involving power level. This custom is followed in the manual.

2.6.4 Limitations of sound power level data. There are two notable limitations regarding sound power level data: Sound power can not be measured directly but are calculated from sound pressure level data, and the directivity characteristics of a source are not necessarily determined when the sound power level data are obtained.

- **PWL calculated, not measured.** Under the first of these limitations, accurate measurements and calculations are possible, but nevertheless there is no simple measuring instrument that reads directly the sound power level value. The procedures involve either comparative sound pressure level measurements between a so-called standard sound source and the source under test (i.e. the "substitution method"), or very careful acoustic qualifications of the test room in which the sound pressure levels of the source are measured. Either of these procedures can be involved and requires quality equipment and knowledgeable personnel. However, when the measurements are carried out properly, the resulting sound power level data generally are more reliable than most ordinary sound pressure level data.

- **Loss of directionality characteristics.** Technically, the measurement of sound power level takes into account the fact that different amounts of sound radiate in different directions from the source, but when the measurements are made in a reverberant or semi-reverberant room, the actual directionality pattern of the radiated sound is not obtained. If directivity data are desired, measurements must be made either outdoors, in a totally anechoic test room where reflected sound cannot distort the sound radiation pattern, or in some instances by using sound intensity measurement techniques. This restriction applies equally to both sound pressure and sound power measurements.

2.7 Sound Intensity Level (L_i)

Sound intensity is sound power per unit area. Sound intensity, like sound power, is not audible. It is the sound intensity that directly relates sound power to sound pressure. Strictly speaking, sound intensity is the average flow of sound energy through a unit area in a sound field. Sound intensity is also a vector quantity, that is, it has both a magnitude and direction. Like sound power, sound intensity is not directly measurable, but sound intensity can be obtained from sound pressure measurements.

2.7.1 Definition, Sound Intensity Level. The sound intensity level (in decibels) is defined by:

$$L_i = 10 \log \frac{I}{I_{ref}} \qquad \text{(eq 2-9)}$$

Where I is the absolute level of the sound intensity and I_{ref} is the reference intensity. Unless otherwise stated the intensity, I, is the effective root mean square (rms) sound intensity.

2.7.2 Definition, reference intensity. Sound intensity level, expressed in decibels, is the logarithmic ratio of the sound intensity at a location, in watts/square meter (W/m^2) relative to the sound power reference base of 10-12 W/m^2.

2.7.3 Notation. The abbreviation L_i is often used to represent sound intensity level. The use of IL as an abbreviation is not recommended since this is often the same abbreviation for "Insertion Loss" and can lead to confusion.

2.7.4 Computation of sound power level from intensity level. The conversion between sound intensity level (in dB) and sound power level (in dB) is as follows:

$$L_w = 10 \log \left[A \left(\frac{I}{I_{ref}} \right) \right] \qquad \text{(eq 2-10)}$$

where A is the area over which the average intensity is determined in square meter (m^2). Note this can also be written as:

$$\mathbf{L_W} = L_i + 10 \log\{A\} \qquad \text{(eq 2-11)}$$

if A is in English units (sq. ft.) then equation 11 can be written as:

$$\mathbf{L_W} = L_i + 10 \log\{A\} - 10 \qquad \text{(eq 2-12)}$$

Note, that if the area A completely closes the sound source, these equations can provide the total sound power level of the source. However care must be taken to ensure that the intensity used is representative of the total area. This can be done by using an area weighted intensity or by logarithmically combining individual L_w's.

2.7.5 Determination of sound intensity. Although sound intensity cannot be measured directly, a reasonable approximation can be made if the direction of the energy flow can be determined. Under free field conditions where the energy flow direction is predictable (outdoors for example) the magnitude of the sound pressure level (L_p) is equivalent to the magnitude of the intensity level (L_i). This results because, under these conditions, the intensity (I) is directly proportional to the square of the sound pressure (p^2). This is the key to the relationship between sound pressure level and sound power level. This is also the reason that when two sounds combine the resulting sound level is proportional to the log of the sum of the squared pressures (i.e. the sum of the p^2's) not the sum of the pressures (i.e. not the sum of the p's). That is, when two sounds combine it is the intensities that add, not the pressures. Recent advances in measurement and computational techniques have resulted in equipment that determines sound intensity directly, both magnitude and direction. Using this instrumentation sound intensity measurements can be conducted in more complicated environments, where fee field conditions do not exist and the relationship between intensity and pressure is not as direct.

2.8 Vibration Levels

Vibration levels are analogous to sound pressure levels.

2.8.1 Definition, vibration level. The vibration level (in decibels) is defined by:

$$L_a = 10 \log \left[\left(\frac{a}{a_{ref}} \right)^2 \right] \qquad \text{(eq 2-13)}$$

Where a is the absolute level of the vibration and a_{ref} is the reference vibration. In the past different measures of the vibration amplitude have been utilized, these include, peak-to-peak (p-p), peak (p), average and root mean square (rms) amplitude. Unless otherwise stated the vibration amplitude, a, is the root mean square (rms). For simple harmonic motion these amplitudes can be related by:

rms value	=	0.707 x peak
average value	=	0.637 x peak
rms value	=	1.11 x average
peak-to peak	=	2 x peak

In addition vibration can be measured with three different quantities, these are, acceleration, velocity and displacement. Unless otherwise stated the vibration levels used in this manual are in terms of acceleration and are called “acceleration levels”. For simple harmonic vibration at a single frequency the velocity and displacement can be related to acceleration by:

$$\text{velocity} = \text{acceleration}/(2\pi f)$$

$$\text{displacement} = \text{acceleration}/[(2\pi f)^2]$$

Where f is the frequency of the vibration in hertz (cycles per second). For narrow bands and octave bands, the same relationship is approximately true where f is the band center frequency in hertz.

2.8.2 Definition, reference vibration. In this manual, the acceleration level, expressed in decibels, is the logarithmic ratio of acceleration magnitudes where the reference acceleration is 1 micro G(10^{-6}), where G is the acceleration of gravity (32.16 ft/sec^2 or 9.80 m/s^2). It should be noted that other reference acceleration levels are in common use, these include, 1 micro m/s^2,10 micro m/s^2 (approximately equal to 1 micro G) and 1 G. Therefore when stating an acceleration level it is customary to state the reference level, such as "60 dB relative to 1 micro G".

2.8.3 Abbreviations. The abbreviation VAL is sometimes used to represent vibration acceleration level, and the notation L_a is normally used in equations, both in this manual and in the general acoustics literature.

2.9 Frequency

Frequency is analogous to "pitch." The normal frequency range of hearing for most people extends from a low frequency of about 20 to 50 Hz (a "rumbling" sound) up to a high frequency of about 10,000 to 15,000 Hz (a "hissy" sound) or even higher for some people. Frequency characteristics are important for the following four reasons: People have different hearing sensitivity to different frequencies of sound (generally, people hear better in the upper frequency region of about 500-5000 Hz and are therefore more annoyed by loud sounds in this frequency region); high-frequency sounds of high intensity and long duration contribute to hearing loss; different pieces of electrical and mechanical equipment produce different amounts of low-, middle-, and high-frequency noise; and noise control materials and treatments vary in their effectiveness as a function of frequency (usually, low frequency noise is more difficult to control; most treatments perform better at high frequency).

2.9.1 Frequency unit, hertz, Hz. When a piano string vibrates 400 times per second, its frequency is 400 vibrations per second or 400 Hz. Before the US joined the IS0 in standardization of many technical terms (about 1963), this unit was known as “cycles per second.”

2.9.2 Discrete frequencies, tonal components. When an electrical or mechanical device operates at a constant speed and has some repetitive mechanism that produces a strong sound, that sound may be concentrated at the principal frequency of operation of the device. Examples are: the blade passage frequency of a fan or propeller, the gear-tooth contact frequency of a gear or timing belt, the whining frequencies of a motor, the firing rate of an internal combustion engine, the impeller blade frequency of a pump or compressor, and the hum of a transformer. These frequencies are designated “discrete frequencies” or “pure tones” when the sounds are clearly tonal in character, and their frequency is usually calculable. The principal frequency is known as the “fundamental,” and most such sounds also contain many “harmonics” of the fundamental. The harmonics are multiple of the fundamental frequency, i.e., 2, 3, 4, 5, etc. times the fundamental. For example, in a gear train, where gear tooth contacts occur at the rate of 200 per second, the fundamental frequency would be 200 Hz, and it is very probable that the gear would also generate sounds at 400, 600, 800, 1000, 1200 Hz and so on for possibly 10 to 15 harmonics. Considerable sound energy is often concentrated at these discrete frequencies, and these sounds are more noticeable and sometimes more annoying because of their presence. Discrete frequencies can be located and identified within a general background of broadband noise (noise that has all frequencies present, such as the roar of a jet aircraft or the water noise in a cooling tower or waterfall) with the use of narrowband filters that can be swept through the full frequency range of interest.

2.9.3 Octave frequency bands. Typically, a piece of mechanical equipment, such as a diesel engine, a fan, or a cooling tower, generates and radiates some noise over the entire audible range of hearing. The amount and frequency distribution of the total noise is determined by measuring it with an octave band analyzer, which is a set of contiguous filters covering essentially the full frequency range of human hearing. Each filter has a bandwidth of one octave, and nine such filters cover the range of interest for most noise problems. The standard octave frequencies are given in table I. An octave represents a frequency interval of a factor of two. The first column of table I gives the band width frequencies and the second column gives the geometric mean frequencies of the bands. The latter values are the frequencies that are used to label the various octave bands. For example, the 1000-Hz octave band contains all the noise falling between 707 Hz (1000/square root of 2) and 1414 Hz (1000 x square root of 2). The frequency characteristics of these filters have been standardized by agreement (ANSI S1.11 and ANSI S1.6). In some instances reference is made to “low”, “mid” and “high” frequency sound. This distinction is somewhat arbitrary, but for the purposes of this manual low frequency sound includes the 31 through 125 Hz octave bands, mid frequency sound includes the 250 through 1,000 Hz octave bands, and high frequency sound includes the 2,000 through 8,000 Hz octave band sound levels. For finer resolution of data, narrower bandwidth filters are sometimes used; for example, finer constant percentage bandwidth filters (e.g. half-octave, third-octave, and tenth-octave filters), and constant width filters (e.g. 1 Hz, 10 Hz, etc.). The spectral information presented in this manual in terms of full octave bands. This has been found to be a sufficient resolution for most engineering considerations. Most laboratory test data is obtained and presented in terms of 1/3 octave bands. A reasonably approximate conversion from 1/3 to full octave bands can be made (see below). In certain cases the octave band is referred to as a “full octave” or "1/1 octave” to differentiate it from partial octaves such as the 1/3 or 1/2 octave bands. The term “overall” is used to designate the total noise without any filtering.

2.9.4 Octave band levels (1/3). Each octave band can be further divided into three 1/3 octave bands. Laboratory data for sound pressure, sound power and sound intensity levels may be given in terms of 1/3 octave band levels. The corresponding octave band level can be determined by adding the levels of the three 1/3 octave bands using equation 2. There is no method of determining the 1/3 octave band levels from octave band data. However as an estimate one can assume that the 1/3 octave levels are approximately 4.8 dB less than the octave band level. Laboratory data for sound transmission loss is commonly given in terms of 1/3 octave band transmission losses. To convert from 1/3 octave band transmission losses to octave band transmission losses use equation 2-14.

$$TL_{ob} = 4.77 - 10 \log \left[10^{\frac{-TL_1}{10}} + 10^{\frac{-TL_2}{10}} + 10^{\frac{-TL_3}{10}} \right]$$

(eq 2-14)

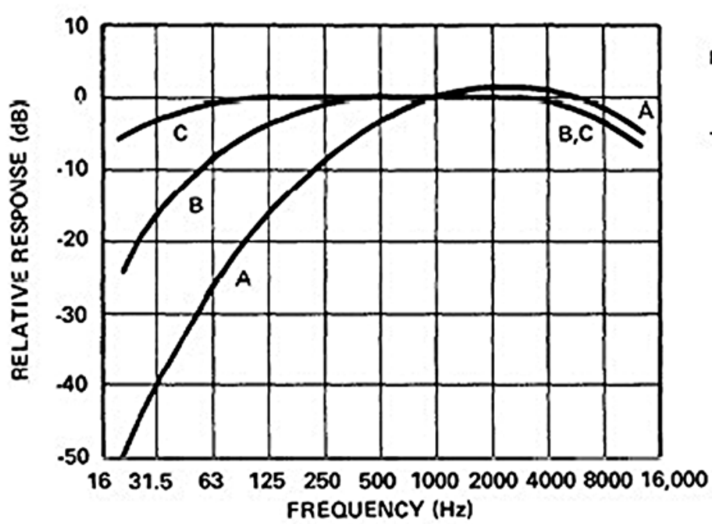
Frequency, Hz					
Octave			One-third octave		
Lower band limit	Center	Upper band limit	Lower band limit	Center	Upper band limit
11	16	22	14.1	16	17.8
			17.8	20	22.4
			22.4	25	28.2
22	31.5	44	28.2	31.5	35.5
			35.5	40	44.7
			44.7	50	56.2
44	63	88	56.2	63	70.8
			70.8	80	89.1
			89.1	100	112
88	125	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

Bandwidth and Geometric Mean Frequency of Standard Octave and 1/3 Octave Bands

Table 2-1

Where TL_{ob} is the resulting octave band transmission loss and TL1, TL2 & TL3 are the 1/3 octave band transmission losses.

2.9.5 A-, B- & C-weighted sound levels. Sound level meters are usually equipped with "weighting circuits" that tend to represent the frequency characteristics of the average human ear for various sound intensities. The frequency characteristics of the A-, B-, and C-weighting networks are shown in figure 2-1. The relative frequency response of the average ear approximates the A curve when sound pressure levels of about 20 to 30 dB are heard. For such quiet sounds, the ear has fairly poor sensitivity in the low-frequency region, The B curve represents approximately the frequency response of hearing sensitivity for sounds having 60 to 70-dB sound pressure level, and the C curve shows the almost flat frequency response of the ear for loud sounds in the range of about 90 to 100 dB. Annoyance usually occurs when an unwanted noise intrudes into an otherwise generally quiet environment. At such times, the ear is listening with a sensitivity resembling the A curve. Thus, judgment tests are often carried out on the loudness, noisiness, annoyance, or intrusiveness of a sound or noise related to the A-weighted sound level of that sound. The correlation is generally quite good, and it is a generally accepted fact that the high-frequency noise determined from the A-weighted sound level is a good indicator of the annoyance capability of a noise. Thus, noise codes and community noise ordinances are often written around A-weighted sound levels. For example: “The sound level at the property line between a manufacturing or industrial plant and a residential community must not exceed 65 dB(A) during daytime or 55 dB(A) during nighttime.” Of course, other sound levels and other details might appear in a more complete noise code. Sound levels taken on the A-, B-, and C-weighted networks have usually been designated by dB(A), dB(B), and dB(C), respectively. The parentheses are sometimes omitted, as in dBA. The weighting networks, in effect, discard some of the sound, so it is conventional not to refer to their values as sound pressure levels, but only as sound levels-as in “an A-weighted sound level of 76 dB(A).” High intensity, high-frequency sound is known to contribute to hearing loss, so the A-weighted sound level is also used as a means of monitoring factory noise for the hearing damage potential. It is very important, when reading or reporting sound levels, to identify positively the weighting network used, as the sound levels can be quite different depending on the frequency content of the noise measured. In some cases if no weighting is specified, A-weighting will be assumed. This is very poor practice and should be discouraged.



OCTAVE FREQUENCY BAND (Hz)	A-WEIGHTED FILTER RESPONSE (dB)
31	-39
63	-26
125	-16
250	-9
500	-3
1000	0
2000	+1
4000	+1
8000	-1

Approximate Electrical Frequency Response of
the A-, B-, and C-weighted Networks of Sound Level Meters.

Figure 2-1

2.9.6 Calculation of A-weighted sound level. For analytical or diagnostic purposes, octave band analyses of noise data are much more useful than sound levels from only the weighting networks. It is always possible to calculate, with a reasonable degree of accuracy, an A-weighted sound level from octave band levels. This is done by subtracting the decibel weighting from the octave band levels and then summing the levels logarithmically using equation 2-2. But it is not possible to determine accurately the detailed frequency content of a noise from only the weighted sound levels. In some instances it is considered advantageous to measure or report A-weighted octave band levels. When this is done the octave band levels should not be presented as “sound levels in dB(A)“, but must be labeled as “octave band sound levels with A-weighting”, otherwise confusion will result.

2.10 Temporal Variations

Both the acoustical level and spectral content can vary with respect to time. This can be accounted for in several ways. Sounds with short term variations can be measured using the meter averaging characteristics of the standard sound level meter as defined by ANSI S1.4. Typically two meter averaging characteristics are provided, these are termed “Slow” with a time constant of approximately 1 second and “Fast” with a time constant of approximately 1/8 second. The slow response is useful in estimating the average value of most mechanical equipment noise. The fast response if useful in evaluating the maximum level of sounds which vary widely.

2.11 Speed of Sound and Wavelength

The speed of sound in air is given by equation 2-15 where c is the speed of sound in air in ft./set, and t_F is the temperature in degrees Fahrenheit.

$$c = 49.03 \times (460 + t_F)\ 1/2 \qquad \text{(eq 2-15)}$$

2.11.1 Temperature effect. For most normal conditions, the speed of sound in air can be taken as approximately 1120 ft./sec. For an elevated temperature of about 1000 deg. F, as in the hot exhaust of a gas turbine engine, the speed of sound will be approximately 1870 ft./sec. This higher speed becomes significant for engine muffler designs, as will be noted in the following paragraph.

2.11.2 Wavelength. The wavelength of sound in air is given by equation 2-16.

$$\Lambda = c/f \qquad \text{(eq 2-16)}$$

Where Λ is the wavelength in ft., c is the speed of sound in air in ft./sec, and f is the frequency of the sound in Hz. Over the frequency range of 50 Hz to 12,000 Hz, the wavelength of sound in air at normal temperature varies from 22 feet to 1.1 inches, a relatively large spread. The significance of this spread is that many acoustical materials perform well when their dimensions are comparable to or larger than the wavelength of sound. Thus, a l-inch thickness of acoustical ceiling tile applied directly to a wall is quite effective in absorbing high-frequency sound, but is of little value in absorbing low frequency sound. At room temperature, a l0-feet long dissipative muffler is about 9 wavelengths long for sound at 1000 Hz and is therefore quite effective, but is only about 0.4 wavelength long at 50 Hz and is therefore not very effective. At an elevated exhaust temperature of 1000 deg. F, the wavelength of sound is about 2/3 greater than at room temperature, so the length of a corresponding muffler should be about 2/3 longer in order to be as effective as one at room temperature. In the design of noise control treatments and the selection of noise control materials, the acoustical performance will frequently be found to relate to the dimensions of the treatment compared to the wavelengths of sound. This is the basic reason why it is generally easier and less expensive to achieve high-frequency noise control (short wavelengths) and more difficult and expensive to achieve low frequency noise control (long wavelengths).

2.12 Loudness

The ear has a wide dynamic range. At the low end of the range, one can hear very faint sounds of about 0 to 10 dB sound pressure level. At the upper end of the range, one can hear with clarity and discrimination loud sounds of 100-dB sound pressure level, whose actual sound pressures are 100,000 times greater than those of the faintest sounds. People may hear even louder sounds, but in the interest of hearing conservation, exposure to very loud sounds for significant periods of time should be avoided. It is largely because of this very wide dynamic range that the logarithmic decibel system is useful; it permits compression of large spreads in sound power and pressure into a more practical and manageable numerical system. For example, a commercial jet airliner produced 100,000,000,000 (= 10^{11}) times the sound power of a cricket. In the decibel system, the sound power of the jet is 110 dB greater than that of the cricket ($110 = 10 \log 10^{11}$). Humans judge subjective loudness on a still more compressed scale.

2.12.1 Loudness judgments. Under controlled listening tests, humans judge that a 10 dB change in sound pressure level, on the average, represents approximately a halving or a doubling of the loudness of a sound. Yet a 10-dB reduction in a sound source means that 90 percent of the radiated sound energy has been eliminated. Table 2-2 shows the approximate relationship between sound level changes, the resulting loss in acoustic

Sound Level Change	Acoustic Energy Loss	Relative Loudness
0 dB	0	Reference
-3 dB	50%	Perceptible Change
-10 dB	90%	Half as loud
-20 dB	99%	1/4 as loud
-30 dB	99.9%	1/8 as loud
-40 dB	99.99%	1/16 as loud

Relationship Between Changes in Sound Level, Acoustic Energy Loss, and Relative Loudness of a Sound

Table 2-2

power, and the judgment of relative loudness of the changes. Toward the bottom of the table, it becomes clear that tremendous portions of the sound power must be eliminated to achieve impressive amounts of noise reduction in terms of perceived loudness.

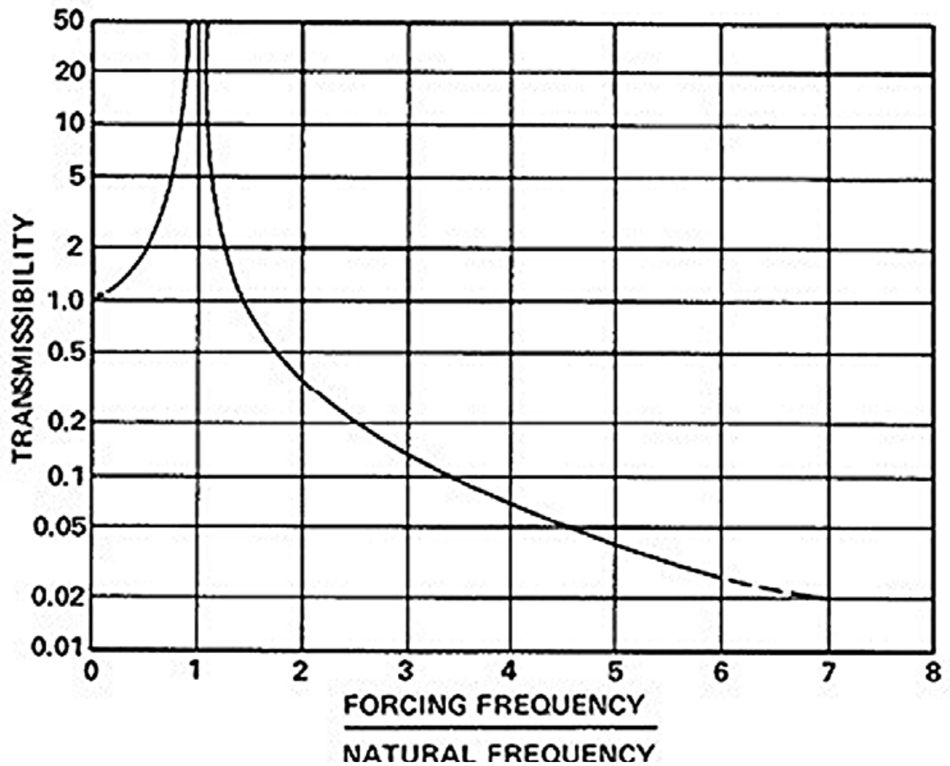
2.12.2 Sones and phons. Sones and phons are units used in calculating the relative loudness of sounds. Sones are calculated from nomograms that interrelate sound pressure levels and frequency, and phons are the summation of the sones by a special addition procedure. The results are used in judging the relative loudness of sounds, as in “a 50-phon motorcycle would be judged louder than a 40-phon motorcycle.” When the values are reduced to phon ratings, the frequency characteristics and the sound pressure level data have become detached, and the noise control analyst or engineer has no concrete data for designing noise control treatments. Sones and phons are not used in this discussion, and their use for noise control purposes is of little value. When offered data in sones and phons, the noise control engineer should request the original octave or 1/3 octave band sound pressure level data, from which the sones and phons were calculated.

2.13 Vibration Transmissibility

A transmissibility curve is often used to indicate the general behavior of a vibration-isolated system. Transmissibility is roughly defined as the ratio of the force transmitted through the isolated system to the supporting structure to the driving force exerted by the piece of vibrating equipment. Figure 2-2 is the transmissibility curve of a simple undamped single-degree-of-freedom system. The forcing frequency is usually the lowest driving frequency of the vibrating system. For an 1800-rpm pump, for example, the lowest driving frequency is 1800/60 = 30 Hz. The natural frequency, in figure 2-2, is the natural frequency of the isolator mount when loaded. An isolator mount might be an array of steel springs, neoprene-in-shear mounts, or pads of compressed glass fiber or layers of ribbed or waffle-pattern neoprene pads. When the ratio of the driving frequency to the natural frequency is less than about 1.4, the transmissibility goes above 1, which is the same as not having any vibration isolator. When the ratio of frequencies equals 1, that is, when the natural frequency of the mount coincides with the driving frequency of the equipment, the system may go into violent oscillation, to the point of damage or danger, unless the system is restrained by a damping or snubbing mechanism. Usually, the driver (the operating equipment) moves so quickly through this unique speed condition that there is no danger, but for large, heavy equipment that builds up speed slowly or runs downs slowly, this is a special problem that must be handled. At higher driving speeds, the ratio of frequencies exceeds 1.4 and the mounting system begins to provide vibration isolation, that is, to reduce the force reduce the force transmitted into the floor or other supported structure. The larger the ratio of frequencies, the more effective the isolation mount.

2.13.1 Isolation efficiency. An isolation mounting system that has a calculated transmissibility, say, of 0.05 on figure 2-2 is often described as having an “isolation efficiency” of 95 percent. A transmissibility of 0.02 corresponds to 98 percent isolation efficiency, and so on. Strict interpretation of transmissibility data and isolation efficiencies, however, must be adjusted for real-life situations.

2.13.2 Transmissibility limitations. The transmissibility curve implies that the mounted equipment (i.e. equipment plus the isolators) are supported by a structure that is infinitely massive and infinitely rigid. In most situations, this condition is not met. For example, the deflection of a concrete floor slab under static load may fall in the range



Transmissibility of a Simple, Undamped Single Degree-of-Freedom System

Figure 2-2

of 1/4 inch to 1/2 inch. This does not qualify as being infinitely rigid. The isolation efficiency is reduced as the static floor deflection increases. Therefore, the transmissibility values of figure 2-2 should not be expected for any specific ratio of driving frequency to natural frequency.

- ***Adjustment for floor deflection.*** In effect, the natural frequency of the isolation system must be made lower or the ratio of the two frequencies made higher to compensate for the resilience of the floor. This fact is especially true for upper floors of a building and is even applicable to floor slabs poured on grade (where the earth under the slab acts as a spring). Only when equipment bases are supported on large extensive portions of bedrock can the transmissibility curve be applied directly.

- ***Adjustment for floor span.*** This interpretation of the transmissibility curve is also applied to floor structures having different column spacings. Usually, floors that have large column spacing, such as 50 to 60 feet, will have larger deflections that floors of shorter column-spacing, such as 20 to 30 feet. To compensate, the natural frequency of the mounting system is usually made lower as the floor span increases. All of these factors are incorporated into the vibration isolation recommendations in this chapter.

- ***Difficulty of field measurement.*** In field situations, the transmissibility of a mounting system is not easy to measure and check against a specification. Yet the concept of transmissibility is at the heart of vibration isolation and should not be discarded because of the above weakness. The material that follows is based on the valuable features of the transmissibility concept, but added to it are some practical suggestions.

2.14 Vibration Isolation Effectiveness

With the transmissibility curve as a guide, three steps are added to arrive at a fairly practical approach toward estimating the expected effectiveness of an isolation mount.

2.14.1 Static deflection of a mounting system. The static deflection of a mount is simply the difference between the free-standing height of the uncompressed, unloaded isolator and the height of the compressed isolator under its static load. This difference is easily measured in the field or estimated from the manufacturer's catalog data. An uncompressed 6 inch high steel spring that has a compressed height of only 4 inches when installed under a fan or pump is said to have a static deflection of 2 inches. Static deflection data are usually given in the catalogs of the isolator manufacturers or distributors. The data may be given in the form of “stiffness” values. For example, a stiffness of 400 lb/in. means that a 400 lb load will produce a 1 inch static deflection, or that an 800 lb load will produce a 2 inch deflection, assuming that the mount has freedom to deflect a full 2 inches.

2.14.2 Natural frequency of a mount. The natural frequency of steel springs and most other vibration isolation materials can be calculated approximately from the formula in equation 2-17.

$$f_n = 3.13 \times \sqrt{\frac{1}{SD}} \qquad \text{(eq 2-17)}$$

where fn is the natural frequency in Hz and SD is the static deflection of the mount in inches.

- ***Example, steel spring.*** Suppose a steel spring has a static deflection of 1 inch when placed under one corner of a motor-pump base. The natural frequency of the mount is approximately:

$$f_n = 3.13 \times \sqrt{\frac{1}{1}} = 3/13 \text{ Hz} \qquad \text{(eq 2-17)}$$

- ***Example, rubber pad.*** Suppose a layer of 3/8-inch-thick ribbed neoprene is used to vibration isolate high-frequency structure borne noise or vibration. Under load, the pad is compressed enough to have a 1/16-inch static deflection. The natural frequency of the mount is approximately:

$$f_n = 3.13 \times \sqrt{\frac{1}{\left(\frac{1}{16}\right)}}$$

$$= 3.13 \times \sqrt{16}$$

$$= 3.13 \times 4 = 12 \text{ Hz} \qquad \text{(eq 2-17)}$$

This formula usually has an accuracy to within about plus or minus 20 percent for material such as neoprene-in-shear, ribbed or waffle-pattern neoprene pads, blocks of compressed glass fiber, and even pads of cork and felt when operating in their proper load range.

2.14.3 Application suggestions. Table 2-3 provides a suggested schedule for achieving various degrees of vibration isolation in normal construction. The table is based on the transmissibility curve, but suggests operating ranges of the ratio of driving frequency to natural frequency. The terms “low,” “fair,” and “high” are merely word descriptors, but they are more meaningful than such terms as 95 or 98 percent isolation efficiency which are clearly erroneous when they do not take into account the mass and stiffness of the floor slab. Vibration control recommendations given in this discussion are based on the application of this table.

- ***Example.*** Suppose an 1800-rpm motor-pump unit is mounted on steel springs having l-inch static deflection (as in the example above). The driving frequency of the system is the shaft speed, 1800 rpm or 30 Hz. The natural frequency of the mount is 3 Hz, and the ratio of driving frequency to natural frequency is about 10.

Ratio of Driving Frequency of Source to Natural Frequency of Mount	Degree of Vibration Isolation
Below 1.4	Amplification
1.4 - 3	Negligible
3 - 6	Low
6 - 10	Fair
Above 10	High

<u>Suggested Schedule for Estimating Relative Vibration</u>

<u>Isolation Effectiveness of a Mounting System</u>

Table 2-3

Table 2-3 shows that this would provide a “fair” to “high” degree of vibration isolation of the motor pump at 30 Hz. If the pump impeller has 10blades, for example, this driving frequency would be 300 Hz, and the ratio of driving to natural frequencies would be about 100; so the isolator would clearly give a “high” degree of vibration isolation for impeller blade frequency.

- ***Caution.*** The suggestions on vibration isolation offered in this discussion are based on experiences with satisfactory installations of conventional electrical and mechanical HVAC equipment in buildings. The concepts and recommendations described here may not be suitable for complex machinery, with unusual vibration modes, mounted on complex isolation systems. For such problems, assistance should be sought from a vibration specialist.

3. NOISE CRITERIA

3.1 General. This section includes data and discussions on generally acceptable indoor noise criteria for acceptable living and working environments. These criteria can be used to evaluate the suitability of existing indoor spaces and spaces under design.

3.2 Noise Criteria In Buildings. Room Criteria (RC) and Noise Criteria (NC) are two widely recognized criteria used in the evaluation of the suitability of intrusive mechanical equipment noise into indoor occupied spaces. The Speech Interference Level (SIL) is used to evaluate the adverse effects of noise on speech communication.

3.2.1 Noise Criterion (NC) Curves. Figure 3-1 presents the NC curves. NC curves have been used to set or evaluate suitable indoor sound levels resulting from the operation of building mechanical equipment. These curves give sound pressure levels (SPLs) as a function of the octave frequency bands. The lowest NC curves define noise levels that are quiet enough for resting and sleeping, while the upper NC curves define rather noisy work areas where even speech communication becomes difficult and restricted. The curves within this total range may be used to set desired noise level goals for almost all normal indoor functional areas. In a strict interpretation, the sound levels of the mechanical equipment or ventilation system under design should be equal to or be lower than the selected NC target curve in all octave bands in order to meet the design goal. In practice, however, an NC condition may be considered met if the sound levels in no more than one or two octave bands do not exceed the NC curve by more than one or two decibels.

3.2.2 Room Criterion (RC) Curves. Figure 3-2 presents the Room Criterion (RC) curves. RC curves, like NC curves, are currently being used to set or evaluate indoor sound levels resulting from the operation of mechanical equipment. The RC curves differ from the NC curves in three important respects. First, the low frequency range has been extended to include the 16 and 31.5 Hz octave bands. Secondly, the high frequency range at 2,000 and 4,000 Hz is significantly less permissive, and the 8,000 Hz octave band has been omitted since most mechanical equipment produces very little noise in this frequency region. And thirdly, the range over which the curves are defined is limited from RC 25 to RC 50 because; 1) applications below RC 25 are special purpose and expert consultation should be sought and; 2) spaces above RC 50 indicate little concern for the quality of the background sound and the NC curves become more applicable.

Table 3-1 lists representative applications of the NC curves. The evaluation of the RC curves is different than that for the NC curves. In general the sound levels in the octave bands from 250 to 2,000 Hz are lower than those of the NC curves. Should the octave band sound levels below 250 Hz be greater than the criteria a potential "rumble" problem is indicated. As a check on the relative rumble potential, the following procedure is recommended:

- Sum the sound pressure levels in the octave bands from 31.5 through 250 Hz on an energy basis.
- Sum the sound pressure levels in the octave bands from 500 through 4,000 Hz on an energy basis.
- Subtract the high frequency sum (step 2) from the low frequency sum (step 1).
- If the difference is +30 dB or greater, a positive subjective rating of rumble is expected, if the difference is between +25 and +30 dB a subjective rating of rumble is possible, if the difference is less than +20 dB a subjective rating of rumble is unlikely.

Also indicated on the RC curves (Figure 3-2) are two regions where low frequency sound, with the octave band levels indicated, can induce feelable vibration or audible rattling in light weight structures.

3.2.3 Speech interference levels. The speech interference level (SIL) of a noise is the arithmetic average of the SPLs of the noise in the 500-, 1000-, and 2000-Hz octave bands. The approximate conditions of speech communication between a speaker and listener can be estimated from Table 3-2 when the SIL of the interfering noise is known. Table 3-2 provides “barely acceptable” speech intelligibility, which implies that a few words or syllables will not be understood but that the general sense of the discussion will be conveyed or that the listener will ask for a repetition of portions missed.

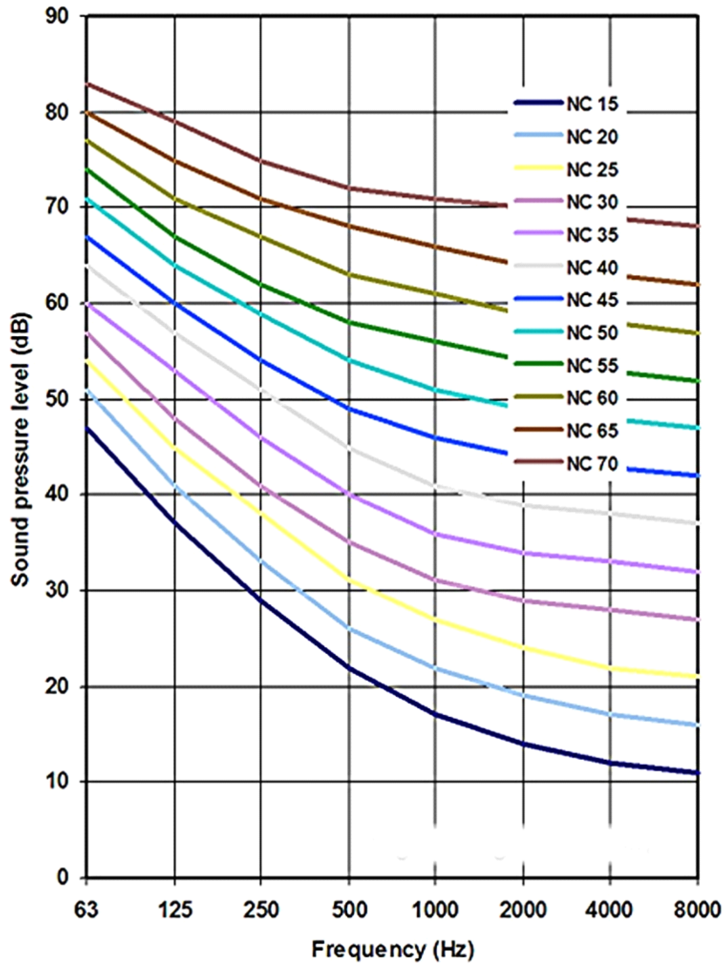


Figure 3-1
Noise Criterion (NC) Curves

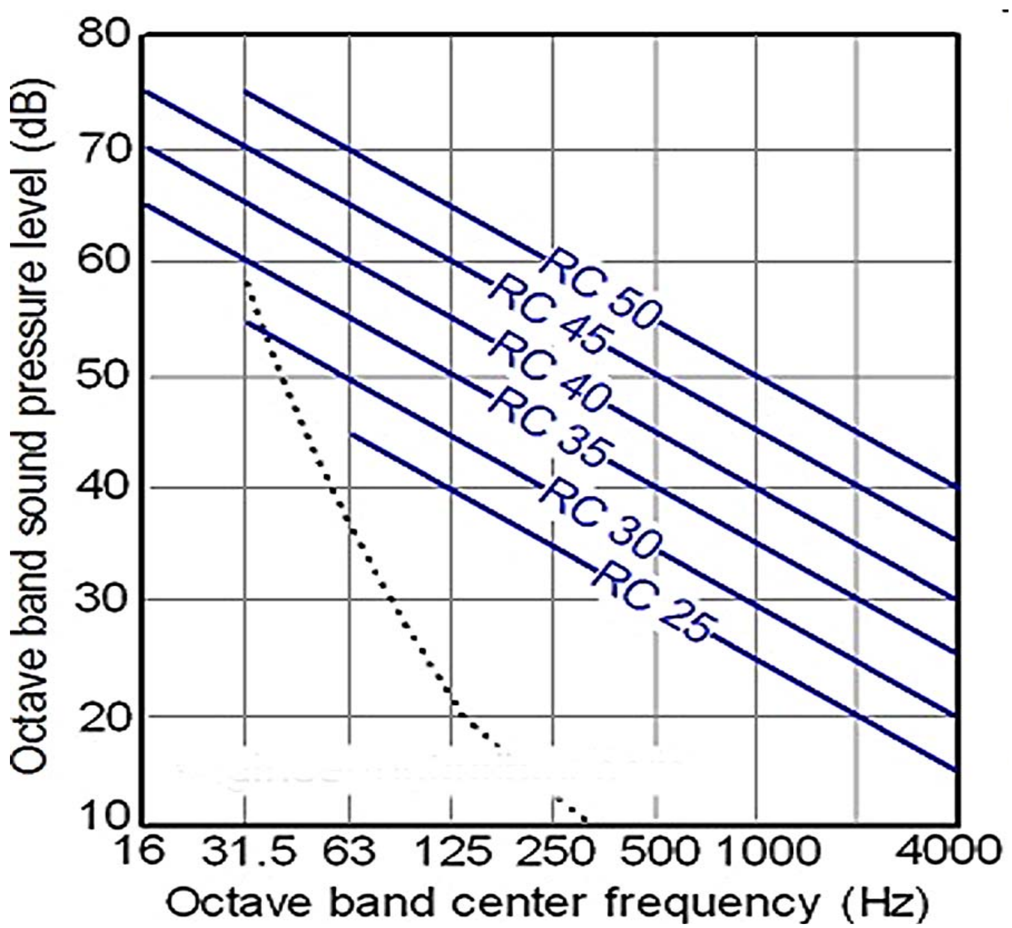


Figure 3-2
Room Criterion (RC) Curves

Category	Area (and Acoustic Requirements)	Noise Criterion[a]
1	Bedrooms, sleeping quarters, hospitals, residences, apartments, hotels, motels, etc. (for sleeping, resting, relaxing).	NC-20 to NC-30
2	Auditoriums, theaters, large meeting rooms, large conference rooms, radio studios, churches, chapels, etc. (for very good listening conditions).	NC-15 to NC-30
3	Private offices, small conference rooms, classrooms, libraries, etc. (for good listening conditions).	NC-30 to NC-35
4	Large offices, reception areas, retail shops and stores, cafeterias, restaurants, etc. (for fair listening conditions).	NC-35 to NC-40
5	Lobbies, drafting and engineering rooms, laboratory work spaces, maintenance shops such as for electrical equipment,etc. (for moderately fair listening conditions).	NC-40 to NC-50
6	Kitchens, laundries, shops, garages, machinery spaces, power plant control rooms, etc. (for minimum acceptable speech communication, no risk of hearing damage).	NC-45 to NC-65

Table 3-1
Representative Applications of the NC Curves

Distance (ft.)	Voice Level			
	Normal	Raised	Very Loud	Shouting
1/2	74	80	86	92
1	68	74	80	86
2	62	68	74	80
4	56	62	68	74
6	53	59	65	71
8	50	56	62	68
10	48	54	60	66
12	46	52	58	64
16	44	50	56	62

Table 3-2
Speech Interference Levels

4. SOUND DISTRIBUTION INDOORS

4.1 Sound Pressure Level In a Room. The sound pressure levels at a given distance or the sound power levels for individual equipment items can often be obtained from equipment suppliers. Once the characteristics of the sound source have been determined, then the sound level at any location within an enclosed space can be estimated. In an outdoor "free field" (no reflecting surfaces except the ground), the sound pressure level (SPL) decreases at a rate of 6 dB for each doubling of distance from the source. In an indoor situation, however, all the enclosing surfaces of a room confine the sound energy so that they cannot spread out indefinitely and become dissipated with distance. As sound waves bounce around within the room, there is a build-up of sound level because the sound energy is "trapped" inside the room and escapes slowly.

4.1.1 Effect of distance and absorption. The reduction of sound pressure level indoors, as one moves across the room away from the sound source, is dependent on the surface areas of the room, the amount of sound absorption material on those areas, the distances to those areas, and the distance from the source. All of this is expressed quantitatively by the curves of Figure 4-1. Figure 4-1 offers a means of estimating the amount of SPL reduction for a piece of mechanical equipment (or any other type of sound source in a room, as one moves away from some relatively close-in distance to any other distance in the room, provided the sound absorptive properties of the room (Room Constant) is known. Conversely Figure 4-1 also provides a means of estimating the sound reduction in a room, from a given source, if the distance is constant and the amount of absorptive treatment is increased. Table 4-1 represents a simplification of Figure 4-1 for a special condition of distance and room constant.

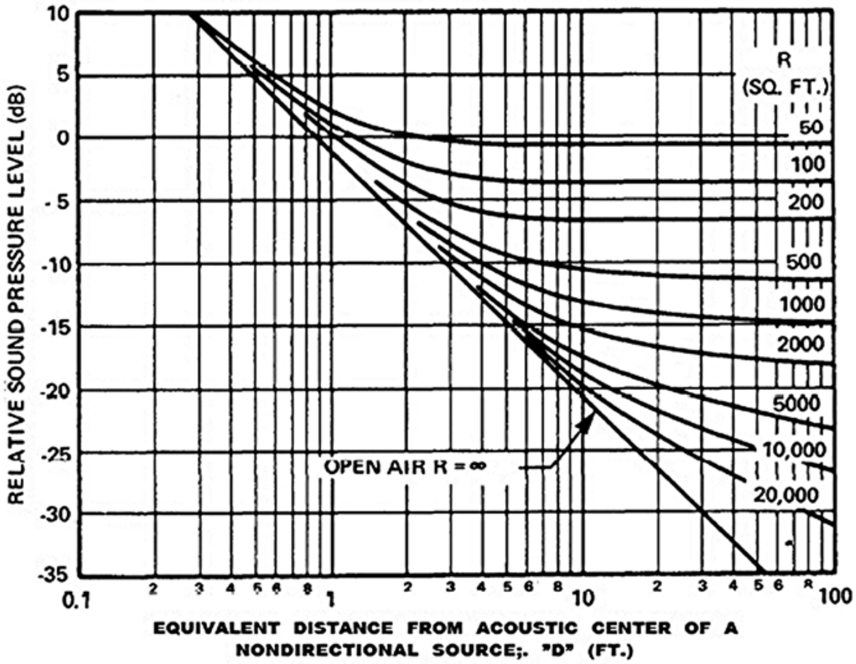


Figure 4-1

Room Constant "R" (ft.^{2})	Distance "D" (in ft.) from Equipment								
	3	5	10	15	20	30	40	60	80
100	-5	-4	-4	-4	-4	-4	-4	-4	-4
200	-3	-2	-1	-1	-1	-1	-1	-1	-1
320	-2	0	0	0	0	0	0	0	0
500	-1	1	2	3	3	3	4	4	4
700	0	2	4	4	5	5	6	6	6
1000	1	3	5	6	7	7	8	0	0
2000	1	4	7	0	9	9	10	10	10
3200	2	5	0	9	10	11	12	12	12
5000	2	6	9	11	12	13	14	14	15
7000	2	6	10	12	13	14	15	15	16
10000	2	7	11	13	14	15	16	17	10
20000	2	7	12	14	16	18	19	21	22
Infinite	2	7	13	16	19	22	25	20	31

1. Reduction of SPL in dB in going from normalized 3-feet distance and 800 sf Room Constant to any other distance and Room Constant.
2. Negative value of reduction means an increase in sound level.

Table 4-1

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4.1.2. Sound absorption coefficients. For most surfaces and materials, the sound absorption coefficients vary with frequency; hence the Room Constant must be calculated for all frequencies of interest. Even room surfaces that are not normally considered absorptive have small amounts of absorption. Usually sound absorption coefficients are not measured in the 31, 63 and 8,000 Hz frequencies. Where the data at these frequencies are not available use 40% of the value of the 125 Hz for the 31 Hz band, 70% of the 125 Hz value for the 63 Hz band and 80% of the 4,000 value for the 8,000 Hz octave band. Values of sound absorption coefficients for specialized acoustical materials must be obtained from the manufacturer.

4.1.3 Estimation of Room Constant. In the early stages of a design, some of the details of a room may not be finally determined, yet it may be necessary to proceed with certain portions of the design. An approximation of the Room Constant can be made using Table 4-2 and Table 4-3. The basic room dimensions are required but it is not necessary to have made all the decisions on side wall, floor, and ceiling materials. This simplification yields a less accurate estimate than does the more detailed procedure, but it permits rapid estimates of the Room Constant with gross, but non-specific, changes in room materials and sound absorption applications. Then, when a favored condition is found, detailed calculations can be made with an equation that is not presented here.

4.1.4 Use of Table 4-2. Table 4-2 gives a broad relationship between the volume of a typically shaped room and the Room Constant as a function of the percentage of room area that is covered by sound absorption material. Room area means the total interior surface area of floor, ceiling, and all side walls. The Room Constant values obtained from this chart strictly apply at 1000 Hz, but in this simplified procedure are considered applicable for the 2000- through 8000-Hz bands as well.

Octave Frequency Band (Hz)	Percent of Area of Thin Surfaces to Total Surface Area of Room							
	0	10	20	30	40	60	80	100
31	1	1.3	1.6	1.9	2.2	2.8	3.4	4.0
63	1	1.3	1.6	1.9	2.2	2.8	3.4	4.0
125	1	1.3	1.6	1.9	2.2	2.8	3.4	4.0
250	1	1.2	1.3	1.4	1.6	1.9	2.1	2.4
500	1	1.1	1.1	1.2	1.2	1.3	1.5	1.6

Table 4-2

Materials	Coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Brick, unglazed	.03	.03	.03	.04	.05	.07
Brick, unglazed, painted	.01	.01	.02	.02	.02	.03
Carpet, heavy, on concrete	.02	.06	.14	.37	.60	.65
Same, on 40 oz hairfelt or foam rubber	.08	.24	.57	.69	.71	.73
Same, with impermeable latex backing on 40 oz hairfelt or foam rubber	.08	.27	.39	.34	.48	.63
Concrete Block, light, porous	.36	.44	.31	.29	.39	.25
Concrete Block, dense, painted	.10	.05	.06	.07	.09	.08
Fabrics						
Light velour, 10 oz per sq yd, hung straight, in contact with wall	.03	.04	.11	.17	.24	.35
Medium velour, 14 oz per sq yd, draped to half area	.07	.31	.49	.75	.70	.60
Heavy velour, 18 oz per sq yd, draped to half area	.14	.35	.55	.72	.70	.65
Floors						
Concrete or terrazzo	.01	.01	.015	.02	.02	.02
Linoleum, asphalt, rubber or cork tile on concrete	.02	.03	.03	.03	.03	.02
Wood	.15	.11	.10	.07	.06	.07
Wood parquet in asphalt on concrete	.04	.04	.07	.06	.06	.07
Glass						
Large panes of heavy plate glass	.18	.06	.04	.03	.02	.02
Ordinary window glass	.35	.25	.18	.12	.07	.04
Gypsum Board, 1/2-in. nailed to 2×4's 16-in. o.c.	.29	.10	.05	.04	.07	.09
Marble or Glazed Tile	.01	.01	.01	.01	.02	.02
Plaster, gypsum or lime, smooth finish on tile or brick	.013	.015	.02	.03	.04	.05
Plaster, gypsum or lime, rough finish on lath	.14	.10	.06	.05	.04	.03
Same, with smooth finish	.14	.10	.06	.04	.04	.03
Plywood Paneling, 3/8-in. thick	.28	.22	.17	.09	.10	.11
Water Surface, as in a swimming pool	.008	.008	.013	.015	.020	.025
Air, Sabins per 1000 cubic feet	.09	.20	.49	1.2	2.9	7.4
Absorption of Seats and Audience						
Values given are in Sabins per square foot of seating area or per unit						
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Chairs, metal or wood seats, each, unoccupied	.15	.19	.22	.39	.38	.30
People in a room, per person (do not use for auditorium calculations)	2	3	4	5	5	4
Audience, seated in upholstered seats, per ft² of floor area, for auditorium calculations	.60	.74	.88	.96	.93	.85

Table 4-3

5. SOUND ISOLATION BETWEEN ROOMS

Discussed here are data and procedures for estimating the changes in sound levels as one follows the “energy flow” path from a sound source to a receiver, through building components, such as walls, floors, doors etc. First, the sound pressure levels in the room containing the source drop off as one moves away from the source. Then, at the walls of the room, some sound is absorbed, some is reflected back into the room, and some is transmitted by the walls into the adjoining rooms (this also occurs at the floor and ceiling surfaces). The combined effects of this absorption, reflection, and transmission are the subject of this discussion.

5.1 Sound Transmission Loss (TL), Noise Reduction (NR) And Sound Transmission Class (STC). With the knowledge of the acoustical isolation provided by walls and floors, it is possible to select materials and designs to limit noise intrusion from adjacent mechanical equipment rooms to acceptable levels. The degree of sound that is transmitted is influenced by the noise isolation properties of the demising construction, the area of the demising wall, floor or ceiling and the acoustical properties in the quiet room.

5.1.1 Transmission loss (TL) of walls. The TL of a wall is the ratio, expressed in decibels, of the sound intensity transmitted through the wall to the airborne sound intensity incident upon the wall. Thus, the TL of a wall is a performance characteristic that is entirely a function of the wall weight, material and construction, and its numerical value is not influenced by the acoustic environment on either side of the wall or the area of the wall. Procedures for determining transmission loss in the laboratory are given in ASTM E 90. This is the data usually given in most manufacturers literature and in acoustic handbooks. Laboratory ratings are rarely achieved in field installations. Transmission loss values in the laboratory are usually greater, by 4 to 5 dB, than that which can be realized in the field even when good construction practices are observed. ASTM E 336 is a corresponding standard method for determination of sound isolation in buildings (in situ). There are many references that provide transmission loss

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performance for building materials. In addition many manufacturers also provide transmission loss for their products.

5.1.2 "Noise reduction" (NR) of a wall. When sound is transmitted from one room (the "source room") to an adjoining room (the "receiving room"), it is the transmitted sound power that is of interest. The transmission loss of a wall is a performance characteristic of the wall structure, but the total sound power transmitted by the wall is also a function of its area (e.g. the larger the area, the more the transmitted sound power). The Room Constant of the receiving room also influences the Sound Power Loss (SPL) in the receiving room. A large Room Constant reduces the reverberant sound level in the room at an appropriate distance from the wall. Thus, three factors influence the SPL in a receiving room: the TL of the wall, the area of the common wall between the source and receiving rooms, and the Room Constant R2 of the receiving room. These three factors are combined in a manner that is beyond the scope of this presentation.

5.1.3 "Sound transmission class" (STC). Current architectural acoustics literature refers to the term "Sound Transmission Class" (STC). This is a one number weighting of transmission losses at many frequencies. The STC rating is used to rate partitions, doors, windows, and other acoustic dividers in terms of their relative ability to provide privacy against intrusion of speech or similar type sounds. This one-number rating system is heavily weighted in the 500- to 2000-Hz frequency region. Its use is not recommended for mechanical equipment noise, whose principal intruding frequencies are lower than the 500- to 2000 Hz region. However, manufacturers who quote STC ratings should have the 1/3 octave band TL data from which the STC values were derived, so it is possible to request the TL data when these types of partitions are being considered for isolation of mechanical equipment noise. The procedure for determining an STC rating is given in ASTM standard E 413.

5.2 Transmission Loss-Walls, Doors, Windows. Generally a partition will have better noise reduction with increasing frequency. It is therefore important to check the noise reduction at certain frequencies when dealing with low frequency, rumbletype noise. Note that partitions can consist of a combination of walls, glass and doors. Walls can generally be classified as fixed walls of drywall or masonry, or as operable walls.

5.2.1 Drywall walls. These walls consist of drywall, studs and, sometimes, fibrous blankets within the stud cavity.

5.2.1.1 Drywall. Drywall is a lightweight, low-cost material, and can provide a very high STC when used correctly. The use of Type X, or fire-rated drywall of the same nonrated drywall thickness, will have a negligible effect on acoustical ratings. Drywall is generally poor at low frequency noise reduction and is also very susceptible to poor installation. Drywall partitions must be thoroughly caulked with a non-hardening acoustical caulk at the edges. Tape and spackle is an acceptable seal at the ceiling and side walls. Electrical boxes, phone boxes, and other penetrations should not be back-to-back, but be staggered at least 2 feet, covered with a fibrous blanket, and caulked. Multiple layers of drywall should be staggered. Wood stud construction has poor noise reduction characteristics because the wood stud conducts vibration from one side to the other. This can be easily remedied by using a metal resilient channel which is inserted between the wood stud and drywall on one side. Non-load-bearing metal studs are sufficiently resilient and do not improve with a resilient channel. Load-bearing metal studs are stiff and can be improved with resilient channels installed on one side.

5.2.1.2 Fibrous blankets. Fibrous blankets in the stud cavity can substantially improve a wall's performance by as much as 10 dB in the mid and high frequency range where non-load-bearing metal studs, or studs with resilient channels, are used. A minimum 2 inch thick, 3/4 lb/ft3 fibrous blanket should be used. Blankets up to 6 inches thick provide a modest additional improvement.

5.2.1.3 Double or staggered stud walls. When a high degree of noise reduction is needed, such as between a conference room and mechanical room, use double or staggered stud wall construction with two rows of metal or wood studs without bracing them together, two layers of drywall on both sides, and a 6 inch thick fibrous blanket.

5.2.2 Masonry walls. Masonry construction is heavy, durable, and can provide particularly good low frequency noise reduction. Concrete masonry units (CMU) made of shale or cinder have good noise reduction properties when they are approximately 50 percent hollow and not less than medium weight aggregate. Parging or furring with drywall on at least one side substantially improves the noise reduction at higher frequencies. The thicker the block, the better the noise reduction. An 8 inch thick, semi-hollow medium aggregate block wall with furring and drywall on one side is excellent around machine rooms, trash chutes, and elevator shafts.

5.2.3 Doors. The sound transmission loss of both hollow and solid core doors will substantially increase when properly gasketed. Regular thermal type tape-on gaskets may not seal well because of door warpage, and can also cause difficulty in closing the door. Tube type seals fitted into an aluminum extrusion can be installed on the door stop and fitted to the door shape. Screw type adjustable tube seals are available for critical installations. Sills with a half moon seal at the bottom of the door are recommended in place of drop seals, which generally do not seal well. Two gasketed doors with a vestibule are recommended for high noise isolation. Special acoustical doors with their own jambs and door seals are available when a vestibule is not practical or very high noise isolation is required.

5.2.4 Windows. Fixed windows will be close to their laboratory TL rating. Operable sash windows can be 10 dB less than the lab rating due to sound leaks at the window frame. Gaskets are necessary for a proper seal. Some window units will have unit TL ratings which would be a rating of both the gasketing and glass type. Double-glazed units are no better than single-glazed if the air space is 1/2 inch or thinner. A 2-inch airspace between glass panes will provide better noise reduction. Laminated glass has superior noise reduction capabilities. Installing glass in a neoprene “U” channel and installing sound absorbing material on the jamb between the panes will also improve noise reduction. Special acoustical window units are available for critical installations.

5.2.5 Transmission loss values for building partitions. Table 5-1 through Table 5-10 provide octave band transmission losses for various constructions, comments or details on each structure are given in the footnotes of the tables. STC ratings are useful for cursory analysis when speech transmission is of concern. The octave band transmission losses should be used when a more thorough analysis is required, such as when the concern is for mechanical equipment.

Octave Frequency Band (Hz)	Suggested Design Values: Thickness of Concrete or Masonry (in.) 4 / Approximate Surface Weight (lb/ft.²) 48	6 / 72	8 / 96	10 / 120	12 / 144	16 / 192	"Ideal Values": Thickness (in.) 4 / Surface Wt. (lb/ft.²) 48	8 / 96	16 / 192
31	29	32	34	36	36	36	30	36	38
63	35	36	36	36	36	37	36	38	38
125	36	36	37	37	38	41	38	38	44
250	36	38	41	43	44	46	38	44	52
500	41	44	45	47	48	49	43	52	58
1000	45	48	49	50	51	53	51	58	64
2000	50	52	53	54	55	57	57	64	70
4000	54	56	57	58	59	61	63	70	76
8000	58	60	61	62	63	65	69	76	82
STC	45	48	49	51	52	53	49	55	62

Notes:

1. "Dense" concrete and masonry assumes 140-150 lb/ft.³ density.
2. For lower values of density, estimate the actual surface weight (in lb/ft.² of wall area) and use the TL from the column in the table that is closest to that surface weight.
3. If desired, install hollow-core block and fill voids completely (a course at a time) with concrete or mortar of the required density.

Table 5-1
Transmission Loss (in dB) of Dense Poured Concrete or
Solid Core Concrete Block or Masonry

Octave Frequency Band (Hz)	Suggested Design Values: Thickness of Concrete or Masonry (in.) 4	6	8	10	12	16	"Ideal Values": Thickness (in.) 4	8	16
	Approximate Surface Weight (lb/ft.²) 28	36	44	52	60	76	Surface Wt. (lb/ft.²) 28	44	76
31	26	28	29	31	32	33	26	30	34
63	31	33	35	36	36	36	32	36	38
125	35	36	36	36	36	36	37	38	38
250	36	36	36	37	37	38	38	38	39
500	37	38	41	42	43	44	38	42	48
1000	42	44	45	46	47	48	45	50	54
2000	46	48	49	50	51	52	53	56	60
4000	50	52	53	54	55	56	59	62	66
8000	54	56	57	58	59	60	65	68	72
STC	42	43	45	46	47	48	45	48	52

Notes:

1. "Dense" concrete and masonry assumes 140-150 lb/ft.³ density, if solid.
2. For lower density concrete, estimate the actual surface weight and use the TL for that value.

Table 5-2

Transmission Loss (in dB) of Hollow-Core Dense Concrete Block or Masonry

Octave Frequency Band (Hz)	Suggested Design Values				"Ideal Values"	
	Thickness of Cinder Block (in.)				Thickness (in.)	
	4	6	8	10	4	10
	Approximate Surface Weight (lb/ft.^{2})				Surface Wt. (lb/ft.^{2})	
	24	36	48	60	24	60
31	22	26	27	28	24	30
63	27	28	28	28	29	30
125	28	28	28	28	30	30
250	28	29	30	33	30	35
500	30	34	36	37	32	42
1000	36	38	40	41	41	48
2000	40	42	43	44	47	54
4000	43	45	46	47	53	60
8000	46	48	49	50	59	66
STC	36	38	39	41	39	46

Notes:

1. Lightweight block material assumes 65-75 lb/ft.^{3} density.

2. If hollow-core block or block of other density is used, select TL value for equivalent surface weight; interpolate or extrapolate if necessary.

3. Both sides of wall surfaces should be sealed with a plaster skim coat or two coats of heavy paint to achieve these values.

Table 5-3

Transmission Loss (in dB) of Cinder Block or
Other Lightweight Porous Block Material
with Impervious Skin on both Sides to Seal Pores

Octave Frequency Band (Hz)	Suggested Design Values					"Ideal Values"	
	Thickness of Plaster (in.)					Thickness (in.)	
	1/2	3/4	1	1-1/2	2	1/2	2
	Approximate Surface Weight (lb/ft.^{2})					Surface Wt. (lb/ft.^{2})	
	4-1/2	7	9	13	18	4-1/2	18
31	8	12	14	17	20	10	22
63	14	18	20	24	26	16	28
125	20	24	26	27	28	22	30
250	26	28	28	28	28	28	30
500	28	28	28	28	29	30	30
1000	28	28	29	32	34	30	38
2000	29	32	34	36	38	30	44
4000	35	37	38	40	41	38	50
8000	38	41	42	43	44	44	56
STC	29	30	31	33	34	31	37

Table 5-4
Transmission Loss (in dB) of Dense Plaster

Octave Frequency Band (Hz)	Type 1	Type 2	Type 3	Improvements A	Improvements B
31	4	9	6	2	2
63	10	16	12	2	2
125	17	24	20	3	2
250	26	34	30	3	3
500	34	42	39	4	4
1000	40	48	46	4	4
2000	46	46	52	3	5
4000	44	48	50	3	6
8000	48	52	54	3	5
STC	37	44	41	3	3

Notes :

Type 1 One layer 1/2=in. thick gypsum wallboard on each side of 2x4-in. wood studs on 16-in. centers. Fill and tape joints and edges; finish as desired. For equal width metal studs, add 2 dB in all bands and to STC.

Type 2 Two layers 5/8-h. thick gypsum wallboard on each side of 2x4-in. wood studs on 16-in. centers. Fill and tape Joints and edges; finish as desired. For equal width metal studs, add 3 dB in all bands and to STC.

Type 3 One layer 5/8-in. thick gypsum wallboard on outer edges of staggered studs, alternate studs supporting separate walls. 2x4 in. wood studs on 16-in. centers for each wall. Fill and tape joints and edges ; finish as desired. For equal width metal studs, add 1 dB in all bands and to STC.

Table 5-5

Transmission Loss (in dB) of Stud-Type Partitions

Octave Frequency Band (Hz)	Thickness of Plywood or Lumber (in.)				
	1/4	1/2	1	2	4
	Approximate Surface Weight (lb/ft.^{2})				
	1	2	4	8	16
31	0	2	7	12	17
63	2	7	12	17	18
125	7	12	17	18	19
250	12	17	18	19	22
500	17	18	19	22	30
1000	18	19	22	30	35
2000	19	22	30	35	39
4000	22	30	35	39	43
8000	30	35	39	43	47
STC	18	21	24	28	33

Notes:

1. Surface weight based on 48 lb/ft.^{3} density, or 4 lb/ft.^{2} per in. thickness.

2. Lumber construction requires tongue-and-groove Joints, overlapping joints, or sealing of joints against air leakage. For intermediate thicknesses, interpolate between thicknesses given in table.

3. For ungasketed hollow-core flush-mounted wood doors, use TL for 1/h-in. thick plywood.

4. For solid-core wood doors or approximately 2-in. thickness, well gasketed all around, use TL for 2-in. thick plywood.

5. For small-area doors or boxes, framing around Cage of panel adds effective mass and stiffness and will probably give higher TL values than shown.

Table 5-6
Transmission Loss (in dB) of Plywood, Lumber and Simple Wood Doors

Octave Frequency Band (Hz)	Thickness of Glass (in.)			
	1/8	1/4	1/2	3/4
	Approximate Surface Weight (lb/ft^2)			
	1-1/2	3	6-1/2	10
31	2	7	13	17
63	8	14	19	22
125	13	20	24	26
250	19	25	27	28
500	23	27	29	29
1000	27	28	29	30
2000	27	28	31	32
4000	27	31	36	38
8000	31	34	40	43
STC	26	28	30	31

Notes :

1. Variations in surface area and edge-clamping conditions can alter the TL values considerably. There is not much consistency among published data.

2. TL tests usually are not carried out at 31-63 Hz; values given are estimates only.

3. In typical operable windows, poor seals can reduce these values.

4. Special laminated safety glass containing one or more viscoelastic layers sandwiched between glass panels will yield 5-10 dB higher values than given here for single thicknesses of glass; available in approximately 1/4- to 3/4-in. thicknesses.

Table 5-7
Transmission Loss (in dB) of Glass Walls or Windows

Octave Frequency Band (Hz)	Width of Air Space (in.)		
	1/4	1-1/2	6
31	13	14	15
63	18	19	22
125	23	26	30
250	26	30	35
500	29	34	40
1000	34	38	43
2000	31	37	44
4000	34	41	50
8000	38	46	54
STC	31	37	43

Table 5-8

Transmission Loss (in dB) of Typical Double-Glass Windows, Using 1/4-in.-Thick Glass Panels with Different Air Space Widths

Octave Frequency Band (Hz)	Filled Metal Panel Partition[a]	Acoustic Doors, Nominal Thicknesses				
		2-in. Thick[b]	4-in. Thick[c]	6-in. ThicK[d]	10-in. Thick[e]	Two Sets 4-in. Doors in Double Walls 32-in. Air Space[f]
31	19	--	27	34	--	42
63	22	--	29	37	--	48
125	26	31	34	41	47	54
250	31	34	36	47	53	60
500	36	37	40	52	61	67
1000	43	39	45	55	66	75
2000	48	43	49	59	65	84
4000	50	47	51	62	69	90
8000	52	--	--	60	--	95
STC	41	40	45	58	64	71

Notes:

[a]Constructed of two 18 ga. steel panels filled with 3 in. of 6-8 lb/ft.[3] glass fiber or mineral wool; Joints and edges sealed airtight.

[b]Average of 4 doors, 1-3/4- to 2-5/8-in. thick, gasketed.

[c]Average of 2 doors, all 4-in. thick, gasketed around all edges, range of weight 12-23 lb/ft.

[d]Average of 4 doors, 6- to 7-in. thick, gasketed, installed by manufacturer, range of weight 23-70 lb/ft.[2]

[e]Average of 2 doors, each 10-in. thick, gasketed, installed by manufacturer, range of weight 35-38 lb/ft.[2]

[f]Estimated performance, in isolated 12-in. thick concrete walls, no leakage, no flanking paths.

Table 5-9

Transmission Loss (in dB) of a Filled Metal Panel Partition and Several Commercially Available Acoustic Doors.

	Aluminum			Steel				Lead			
Octave Frequency Band (Hz)	Thickness (in.) 1/16	1/8	1/4	Thickness (in.) 1/16	1/8	1/4	1/2	Thickness (in.) 1/32	1/16	1/8	Lead/Vinyl Curtain
	Surface Weight (lb/ft.²) 1	2	3½	Surface Weight (lb/ft.²) 2½	5	10	20	Surface Weight (lb/ft.²) 2	4	7½	Surface Weight (lb/ft.²) 1
31	0	3	9	5	11	17	23	2	8	14	--
63	3	9	15	11	17	23	29	8	14	20	--
125	9	15	21	17	23	29	35	14	20	26	13
250	15	21	27	23	29	35	40	20	26	32	17
500	21	27	29	29	35	40	40	26	32	38	20
1000	27	29	29	35	40	40	40	32	38	44	28
2000	29	29	29	40	40	40	41	38	44	50	34
4000	29	29	30	40	40	41	48	44	50	56	38
8000	29	30	40	40	41	48	54	50	56	56	--
STC	25	28	29	33	38	40	41	30	36	42	26

Notes:

1. Surface weight of aluminum based on 170 lb/ft.³ density or 14 lb/ft.² per in. thickness.
2. Surface weight of steel based on 480 lb/ft.³ density or 40 lb/ft.² per-in. thickness.
3. Surface weight of lead based on 700 lb/ft.³ density of 59 lb/ft.² per in. thickness.
4. Variations in surface area and edge clamping conditions can alter the TL values of aluminum and steel. Lead assumed "limp." Application of vibration damping material to one surface of steel or aluminum will reduce resonances and help increase TL values in resonance regions.
5. TL tests usually are not conducted at 31-63 Hz; values given are estimates only.

Table 5-10
Transmission Loss (in dB) of Aluminum, Steel and Lead