

# SOUND AND VIBRATION

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**I**F FUNDAMENTAL PRINCIPLES of sound and vibration control are applied in the design, installation, and use of HVAC and refrigeration systems, unacceptably high noise and vibration levels and the consequent complaints can be avoided. This chapter introduces these fundamental principles, including characteristics of sound, basic definitions and terminology, human response to sound, acoustical design goals, and vibration isolation fundamentals. [Chapter 47 of the ASHRAE Handbook—Applications](#) and the references listed at the end of this chapter contain technical discussions, tables, and design examples helpful to HVAC designers.

## ACOUSTICAL DESIGN OBJECTIVE

The primary objective for the acoustical design of HVAC systems and equipment is to ensure that the acoustical environment in a given space is not degraded. Sound and vibration are created by a **source**, are transmitted along one or more **paths**, and reach a **receiver**. Treatments and modifications can be applied to any or all of these elements to achieve an acceptable acoustical environment, although it is usually most effective and least expensive to reduce noise at the source.

## CHARACTERISTICS OF SOUND

Sound is a propagating disturbance in a fluid (gas or liquid) or in a solid. In fluid media, the disturbance travels as a longitudinal compression wave. Sound in air is called **airborne sound** or simply sound. It is generated by a vibrating surface or a turbulent fluid stream. In solids, sound can travel as bending waves, compressional waves, torsional waves, shear waves and others. Sound in solids is generally called **structureborne sound**. In HVAC system design, **both** airborne and structureborne sound propagation are important.

### Speed

The speed of a longitudinal wave in a fluid is a function of the fluid's density and bulk modulus of elasticity. In air, at room temperature, the speed of sound is about 1100 fps; in water, it is about 5000 fps. In solids, there are several different types of waves, each with a different speed and some that depend on frequency. In solids, the speed of sound is usually higher than that in air.

### Sound Pressure and Sound Pressure Level

Sound waves in air are variations in pressure above and below atmospheric pressure. **Sound pressure** is measured in pascals (Pa) (SI units are used rather than I-P units because of international agreement). The human ear responds across a broad range of sound pressures; the threshold of hearing to the threshold of pain covers a

**Table 1 Typical Sound Pressures and Sound Pressure Levels**

Source	Sound Pressure, Pa	Sound Pressure Level, dB re 20 μPa	Subjective Reaction
Military jet takeoff at 100 ft	200	140	Extreme danger
Artillery fire at 10 ft	63.2	130	
Passenger jet takeoff at 100 ft	20	120	Threshold of pain
Loud rock band	6.3	110	Threshold of discomfort
Platform of subway station (steel wheels)	2	100	
Unmuffled large diesel engine at 130 ft	0.6	90	Very loud
Computer printout room	0.2	80	
Freight train at 100 ft	0.06	70	
Conversational speech at 3 ft	0.02	60	
Window air conditioner at 3 ft	0.006	50	Moderate
Quiet residential area	0.002	40	
Whispered conversation at 6 ft	0.0006	30	
Buzzing insect at 3 ft	0.0002	20	Perceptible
Threshold of good hearing	0.00006	10	Faint
Threshold of excellent youthful hearing	0.00002	0	Threshold of hearing

range of approximately 10<sup>14</sup>:1. [Table 1](#) gives approximate values of sound pressure for various sources.

The range of sound pressure in [Table 1](#) is so large that it is more convenient to use a scale that is proportional to the logarithm of this quantity. The **decibel** (dB) scale is such a scale and is the preferred method of presenting quantities in acoustics. The term **level**, when used in relation to sound power, sound intensity, or sound pressure, indicates that dB notation is being used. Numerically, the decibel is ten times the base 10 logarithm of the ratio of two like quantities proportional to acoustical power or energy. Thus, the **sound pressure level**  $L_p$  (in dB) corresponding to a sound pressure is given by

$$L_p = 10 \log(p/p_{ref})^2 \quad (1)$$

where  $p$  is the root mean square (rms) value of acoustic pressure in pascals. The ratio  $p/p_{ref}$  is squared to give quantities proportional to intensity or energy. The reference quantity  $p_{ref}$  is 20 μPa, which corresponds to the approximate threshold of hearing.

The decibel scale is used for many different descriptors relating to sound: source strength, sound level, and sound attenuation, among others; each has a different reference quantity. For this reason, it is important to be aware of the context in which the term decibel or level is used. For most acoustical quantities, there is an

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internationally accepted reference value. A reference quantity is always implied even if it does not appear.

Sound pressure level is relatively easy to measure and for this reason most noise codes and criteria use sound pressure level. (The human ear and microphones are pressure-sensitive devices.) Sound pressure levels for each source are also given in [Table 1](#).

### Frequency

Frequency is the number of oscillations (or cycles) completed per second by a vibrating object. The international unit for frequency is hertz (Hz) with dimension  $s^{-1}$ . When the motion of vibrating air particles is simple harmonic, the sound is said to be a **pure tone** and the sound pressure  $p$  as a function of time and frequency can be described by

$$p(\theta, f) = p_0 \sin 2\pi f \theta \quad (2)$$

where  $f$  is frequency in hertz and  $\theta$  is time in seconds.

The **audible frequency range** for humans extends from about 20 Hz to 20 kHz. In some cases, infrasound (< 20 Hz) or ultrasound (> 20 kHz) are important, but methods and instrumentation for these frequency regions are specialized and are not considered here.

### Wavelength

The wavelength of sound in a medium is the distance between successive maxima or minima of a simple harmonic disturbance propagating in that medium. Wavelength, speed, and frequency are related by

$$\lambda = c/f \quad (3)$$

where

- $\lambda$  = wavelength, ft
- $c$  = speed of sound, fps
- $f$  = frequency, Hz

### Sound Power and Sound Power Level

The **sound power** of a source is its rate of emission of acoustical energy and is expressed in watts. The sound power of a source depends on the operating conditions. Approximate sound power outputs for common sources are shown in [Table 2](#) together with the corresponding sound power levels. For **sound power level**  $L_w$ , the power reference is  $10^{-12}$  W. The definition of sound power level is therefore

$$L_w = 10 \log(w/10^{-12}) \quad (4)$$

where  $w$  is the acoustic power emitted by the source in watts. The acoustic power emitted by a source is not the same as the power consumed by the source. Only a small fraction of the consumed power is converted into sound. For example, a loudspeaker rated at 100 W may be only 1% to 5% efficient, generating only 1 to 5 W (acoustic).

Most mechanical equipment is rated according to sound power so equipment can be compared according to a common reference independent of distance and acoustical conditions in the room. AMCA *Publication 303* provides guidelines for the application of sound power level ratings. In addition, AMCA *Standard 301* provides standard methods for developing fan sound ratings from laboratory test data.

### Sound Intensity and Sound Intensity Level

The **sound intensity**  $I$  at a point in a specified direction is the rate of flow of sound energy through unit area at that point. The unit area is perpendicular to the specified direction, and the units of intensity are  $W/m^2$ . (SI units are used rather than I-P units because

**Table 2 Typical Sound Power Outputs and Sound Power Levels**

Source	Sound Power, W	Sound Power Level, dB re $10^{-12}$ W
Saturn rocket	$10^8$	200
Turbojet engine <sup>a</sup>	$10^5$	170
Jet aircraft at takeoff <sup>b</sup>	$10^4$	160
Turboprop at takeoff	1000	150
Prop aircraft at takeoff <sup>c</sup>	100	140
Large pipe organ	10	130
Small aircraft engine	1	120
Noisy HVAC fan	0.1	110
Automobile at highway speed	0.01	100
Voice, shouting	0.001	90
Garbage disposal unit	$10^{-4}$	80
Voice, conversation level	$10^{-5}$	70
Electronic equipment ventilation fan	$10^{-6}$	60
Office air diffuser	$10^{-7}$	50
Small electric clock	$10^{-8}$	40
Voice, soft whisper	$10^{-9}$	30
Rustling leaves	$10^{-10}$	20
Human breath	$10^{-11}$	10

<sup>a</sup> With afterburner

<sup>b</sup> Four jet engines

<sup>c</sup> Four propeller engines

of international agreement.) **Sound intensity level**  $L_I$  is expressed in dB with a reference quantity of  $10^{-12}$  W/m<sup>2</sup>, thus

$$L_I = \log(I/10^{-12})$$

### Combining Sound Levels

Because the decibel is a logarithmic unit, two sound pressure levels cannot be added arithmetically. The levels must first be converted back to energy units, summed, and then converted to a level again. Thus, the combination of two levels,  $L_1$  and  $L_2$ , produces a level  $L_{sum}$  given by

$$L_{sum} = 10 \log(10^{L_1/10} + 10^{L_2/10}) \quad (5)$$

This process may be extended to as many levels as needed. A simpler and slightly less accurate method is outlined in [Table 3](#). This method, although not exact, results in errors of 1 dB or less. The process with a series of levels may be shortened by combining the largest with the next largest. Then this sum is combined with the third largest, then the fourth largest, and so on until the next level added has little or no influence. The process may then be stopped.

The procedures outlined in [Table 3](#) and in Equation (5) are valid if the individual sound levels are not highly correlated. This is true for most (but not all) sounds encountered in HVAC work. One notable exception is the pure tone. If two or more noise signals contain pure tones at the same frequency, the combined sound level is a function of not only the level of each tone, but also the phase difference between the tones. The combined sound levels from two tones of equal amplitude can range from zero (if the two tones are 180° out of phase) up to 6 dB greater than the level of either tone (if the two tones are exactly in phase). When two tones of similar amplitude are very close in frequency, but not exactly the same, the combined sound level oscillates as the tones move in and out of phase. This effect creates an audible “beating” with a period equal to the inverse of the difference in frequency between the two tones.

**Table 3 Combining Two Sound Levels**

Difference between two levels to be combined, dB	10 and			
	0 to 1	2 to 4	5 to 9	More
Number of decibels to be added to highest level to obtain combined level	3	2	1	0

MEASURING SOUND

Instrumentation

The basic instrument for measuring sound is a **sound level meter**, which includes a microphone, electronic circuitry, and a display device. Sound pressure at a point is converted to sound pressure level and displayed by analog or digital meters. These devices are usually light, battery-operated, hand-held units with outputs that vary in complexity depending on cost and the level of current technology.

Time-Averaging

No sounds are constant; the pressure fluctuates from moment to moment and the level can vary quickly or slowly. Sound level meters can show the time fluctuations of the sound pressure level using specified time constants (slow, fast, impulse), or can hold the maximum or minimum level recorded during some specified interval. All sound level meters perform some kind of **time-averaging**. Some **integrating sound level meters** take an average of the sound pressure level over a user-definable time, then hold and display the result. As a result, an integrating meter is easier to read and more repeatable (especially if the measurement period is long). The quantity measured by the integrating sound level meter is the **equivalent continuous sound pressure level**  $L_{eq}$ .

Spectra and Analysis Bandwidths

Real sounds are much more complex than simple pure tones. **Broadband sound** contains energy at many frequencies, usually covering most of the audible frequency range. All sounds, however, can be represented as a summation of pure tones with different amplitudes. This representation of a sound is called **frequency or spectral analysis** and is similar to spectral analysis in optics.

A **constant-bandwidth analysis** expresses the energy content of a sound as a spectrum where each data point represents the same spectral width, for example, 1 Hz. This kind of analysis is useful when an objectionable sound obviously contains strong tones and the frequencies need to be accurately identified before remedial action is taken. A constant-band spectrum usually contains too much information for typical noise control work.

Measurements for most HVAC noise control work are usually made with filters that extract the energy in either **octave bands** or **one-third octave bands**. An octave band is a frequency band with its upper frequency limit twice that of its lower frequency limit. Octave and 1/3 octave bands are identified by the mid-band frequency, which is the geometric mean of the upper and lower band limits (ANSI Standards S1.6, S1.11). Three 1/3 octave bands make up an octave band. Table 4 lists the preferred series of octave and one-third octave bands and the upper and lower band limit frequencies. For HVAC sound measurements, filters for the range of 16 Hz to 8000 Hz are usually adequate.

While analysis in octave bands is usually acceptable for rating acoustical environments in rooms, 1/3 octave band analysis is often useful in product development and for remedial investigations.

Some sound level meters have octave or 1/3 octave filters for determining the frequency content of the sound. Usually, standard broadband filters that simulate the response of the average human ear to sound are provided. The most commonly used filter is the **A-weighting filter** (Figure 1). This filter simulates the response of the human ear. It de-emphasizes the low-frequency portions of a sound spectrum, automatically compensating for the lower sensitivity of the human ear to low-frequency sound.

The **C-weighting** filter is sometimes used to estimate whether a particular sound has excessive low-frequency energy present when a spectrum analyzer is not available. If the difference between the C- and A-weighted levels for the sound exceeds about 30 dB, then the sound is likely to be annoying because of excessive

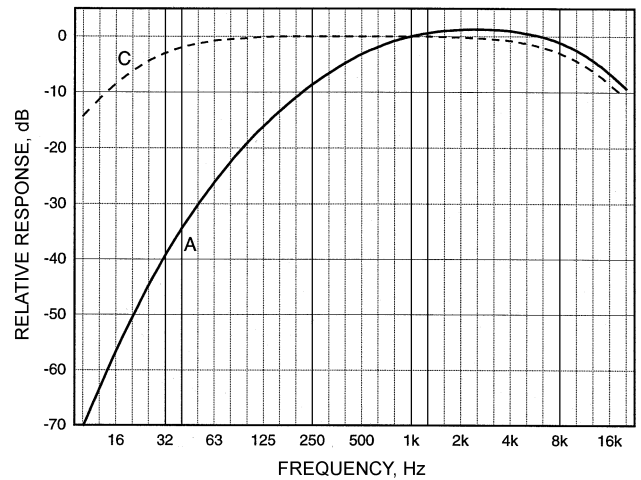


Fig. 1 Curves Showing A- and C-Weighting Responses for Sound Level Meters

Table 4 Mid-Band and Approximate Upper and Lower Cutoff Frequencies for Octave and 1/3 Octave Band Filters

Octave Bands, Hz			1/3 Octave Bands, Hz		
Lower	Mid-Band	Upper	Lower	Mid-Band	Upper
			11.2	12.5	14
11.2	16	22.4	14	16	18
			18	20	22.4
			22.4	25	28
22.4	31.5	45	28	31.5	35.5
			35.5	40	45
			45	50	56
45	63	90	56	63	71
			71	80	90
			90	100	112
90	125	180	112	125	140
			140	160	180
			180	200	224
180	250	355	224	250	280
			280	315	355
			355	400	450
355	500	710	450	500	560
			560	630	710
			710	800	900
710	1,000	1,400	900	1,000	1,120
			1,120	1,250	1,400
			1,400	1,600	1,800
1,400	2,000	2,800	1,800	2,000	2,240
			2,240	2,500	2,800
			2,800	3,150	3,550
2,800	4,000	5,600	3,550	4,000	4,500
			4,500	5,000	5,600
			5,600	6,300	7,100
5,600	8,000	11,200	7,100	8,000	9,000
			9,000	10,000	11,200
			11,200	12,500	14,000
11,200	16,000	22,400	14,000	16,000	18,000
			18,000	20,000	22,400

low-frequency noise. Note (in Figure 1) that the C-weighting curve attenuates significantly at low and high frequencies; the instrument response is not flat when this filter is used.

Sound level meters are available in several accuracy grades (ANSI Standard S1.4). A Type 1 meter has a tolerance of about ±0.7 dB. The general-purpose meter, which is less expensive, is designated Type 2 with a tolerance of about ±1 dB, and is adequate for most HVAC sound measurements.

**Table 5 Combining Decibels to Determine Overall Sound Pressure Level**

Octave Band Frequency, Hz	Octave Band Level $L_p$ , dB	$10^{L_p/10}$	
63	85	$3.2 \times 10^8$	$= 0.32 \times 10^9$
125	90	$1.0 \times 10^9$	$= 1.0 \times 10^9$
250	92	$1.6 \times 10^9$	$= 1.6 \times 10^9$
500	87	$5.0 \times 10^8$	$= 0.5 \times 10^9$
1000	82	$1.6 \times 10^8$	$= 0.16 \times 10^9$
2000	78	$6.3 \times 10^7$	$= 0.06 \times 10^9$
4000	65	$3.2 \times 10^6$	$= 0.003 \times 10^9$
8000	54	$2.5 \times 10^5$	$= 0.0002 \times 10^9$
		$3.6432 \times 10^9$	
$10 \log (3.6 \times 10^9) = 96 \text{ dB}$			

**Table 6 Guidelines for Determining Equipment Sound Levels in the Presence of Contaminating Background Sound**

Measurement A minus Measurement B	Correction to Measurement A to Obtain Equipment Sound Level
10 dB or more	0 dB
6 to 9 dB	-1 dB
4 to 5 dB	-2 dB
Less than 4 dB	Equipment sound level is more than 2 dB below Measurement A

Measurement A = Tested equipment plus background sound  
Measurement B = Background sound alone

Manually selecting filters sequentially to cover the frequency range from 16 Hz to 8000 Hz is time-consuming. An instrument that gives all filtered levels simultaneously is called a **real-time analyzer** (RTA). It speeds up measurement significantly and, on most models, the information can be saved to an internal or external digital storage device.

The process described in the previous section on Combining Sound Levels can be applied to a set of octave or 1/3 octave bands to calculate the overall broadband level (see [Table 5](#) for an example).

### Sound Measurement Basics

The sound pressure level in an occupied space can be measured directly with a sound level meter, or estimated from published sound power data after accounting for room volume, distance from the source, and other acoustical factors. Sound level meters measure the sound pressure at the microphone location. Estimation techniques calculate sound pressure at a specified point in an occupied space. Measured or estimated sound pressure levels in frequency bands can then be plotted, analyzed, and compared with established criteria for acceptance.

Measurements of HVAC sound must be done carefully to ensure repeatable and accurate results. The sound levels may not be steady, particularly at low frequencies (250 Hz and lower) and can vary significantly with time. In such cases, both the peak and average levels should be recorded.

Sophisticated sound measurements and their procedures should be carried out by experienced sound professionals. At present, only a few noise standards apply to measuring the interior noise from mechanical equipment (ASTM *Standard* E 1574, ASTM *Standard* E 1573). Most manuals for sound level meters include sections on how to measure sound.

Determining the sound spectrum in a room or investigating a noise complaint usually requires measuring the sound pressure levels in the octave bands from 16 Hz to 8000 Hz. In cases of tonal noise or rumble, narrow bands or 1/3 octave bands should be measured because their frequency resolution is higher. Whatever the measurement method, the sound pressure level is measured at a point. In a room, each measurement point often has a different

sound pressure level, so the precise location of the measurement must be detailed in the report. One might survey the room and record the location and level of the loudest position. Also, one could establish a few representative locations where occupants are normally situated. In general, the most appropriate height is 4 to 6 ft above the floor. The exact geometric center of the room should be avoided, as should any location within 3 ft of a wall.

Wherever the location, it must be defined and recorded. If the meter has an integrating-averaging function, one can use a rotating boom to sample a large area or walk slowly around the room to measure the average sound pressure level over that path. However, care must be taken that no extraneous sounds are generated by microphone movement or by walking. Also, locations where the sound levels are notably higher than average should be recorded. The section on Measurement of Room Sound Pressure Level has more details.

The contribution of other sources (plumbing noise, business machines, nearby traffic, etc.) to the sound pressure levels must also be determined. Sound from sources other than the source to be measured is designated background sound.

Sometimes the sound from a particular piece of HVAC equipment must be measured in the presence of background sound from other sources that cannot be turned off, such as automobile traffic or certain business machines. Two sets of measurements are required to determine the sound level due to selected equipment: one set with both the HVAC equipment sound and the background sound and another set with only the background sound (the HVAC equipment is turned off). This situation might also occur, for example, when determining whether the noise exposure at the property line due to a cooling tower meets a local noise ordinance. The guidelines in [Table 6](#) will help in determining the sound level of a particular machine in the presence of background sound.

The uncertainty associated with correcting for background sound depends on the uncertainty of the measuring instrument and the steadiness of the sounds being measured. In favorable circumstances, it might be possible to extend [Table 6](#). In particularly unfavorable circumstances, even values obtained from the table could be substantially in error.

Measuring sound emissions from a particular piece of equipment or group of equipment requires a measurement plan specific to the situation. The section on Standards lists several sound level measurement procedures for various laboratory and field sound measurement situations.

Outdoor measurements are somewhat easier to make than indoor because typically there are few or no boundary surfaces to affect sound build-up or absorption. Nevertheless, important issues such as the effect of large, nearby sound-reflecting surfaces and weather conditions such as wind, temperature, and precipitation must be considered. (In cases where measurements are made close to extended surfaces, sound pressure levels can be significantly increased.) These effects can be estimated through guidelines in many sources such as Harris (1991).

### Measurement of Room Sound Pressure Level

In the commissioning of HVAC systems in buildings, it often must be demonstrated that a specified room noise criterion has been met. Measurement procedures for obtaining the data to demonstrate compliance are often not specified. This can lead to confusion when different parties make measurements using different procedures, as the results often do not agree. The problem is that most rooms exhibit significant point-to-point variation in sound pressure level.

When a noise has no audible tonal components, the differences in measured sound pressure level at several locations in a room may be as high as 3 to 5 dB. However, when audible tonal components are present, especially at low frequencies, the variations due to standing waves may exceed 10 dB. These variations are generally noticeable to the average listener when moving through the room.

Although the commissioning procedures usually set precise limits for demonstrating compliance, the outcome can be controversial unless the measurement procedure has been specified in detail. At present, the industry has no general agreement regarding an acoustical measurement procedure for commissioning HVAC systems. However, ARI *Standard* 885 incorporates a “suggested procedure for field verification of NC/RC levels.”

## DETERMINING SOUND POWER

No instrument can measure the sound power of a source directly. Rather, sound power is calculated from several measurements of sound pressure or sound intensity created by a source in one of several test environments. The following four methods are commonly used.

### 1. Free-Field Method

A **free field** is a sound field where the effects of any boundaries are negligible over the frequency range of interest. In ideal conditions, there are no boundaries. Free-field conditions can be approximated in rooms having highly sound-absorbing walls, floor, and ceiling (**anechoic rooms**). In a free field the sound power of a sound source can be determined from a number of measurements of sound pressure level on an imaginary spherical surface centered on and surrounding the source. This method is based on the fact that, because absorption of sound in air can be practically neglected at small distances from the sound source, the sound power generated by a source must flow through an imagined sphere with the source at its center. The intensity of the sound is determined at each of the measuring points around the source and multiplied by the area of the imagined sphere associated with the measuring points. Total sound power is the sum of these products for each point.

ANSI *Standard* S12.35 describes various methods used to calculate the sound power level under free-field conditions. Measurement accuracy is limited at the lower frequencies because room surface treatments do not have high sound absorption coefficients at low frequencies. For example, a glass fiber wedge structure that gives significant absorption at 70 Hz must be at least 4 ft long.

Using values for the speed of sound for air at 68°F and 14.7 psi, the relationship between sound power level and sound pressure level for a nondirectional sound source is

$$L_w = L_p + 20 \log r + 0.5 \quad (6)$$

where

$$\begin{aligned} L_w &= \text{sound power level, (dB re } 10^{-12} \text{ W)} \\ L_p &= \text{sound pressure level (dB re } 20 \text{ } \mu\text{Pa)} \\ r &= \text{distance from nondirectional sound source, ft} \end{aligned}$$

**Free-Field Over Reflecting Plane.** In many cases, a completely free field is not available, and measurements can only be made in a free field over a reflecting plane. That is, the sound source is placed on a hard floor (in an otherwise sound-absorbing room) or on pavement outdoors. Since the sound is then radiated into a hemisphere rather than a full sphere, the relationship for  $L_w$  and  $L_p$  for a nondirectional sound source becomes

$$L_w = L_p + 20 \log r - 2.5 \quad (7)$$

**Source Directivity.** A sound source may radiate different amounts of sound power in different directions because various areas of its surface do not vibrate at the same level or in phase. A directivity pattern can be established by measuring sound pressure under free-field conditions, either in an anechoic room or over a reflecting plane in a hemi-anechoic space at several points around the source. The **directivity factor**  $Q$  is defined as the ratio of sound pressure at a given angle from the sound source to the sound pressure that would be produced by the same source radiating uniformly

in all directions.  $Q$  is a function of both frequency and direction. [Chapter 47 of the ASHRAE Handbook—Applications](#) provides more detailed information on sound source directivity.

### 2. Reverberation Room Method

Another method to determine sound power places the sound source in a reverberation room. Standardized methods for determining the sound power of HVAC equipment in reverberation rooms are given in ANSI *Standard* S12.31, when the sound source contains mostly broadband sound; ANSI *Standard* S12.32, when tonal sound is prominent; and AMCA *Standard* 300 for testing fans.

Sound sources that can be measured by these methods include room air conditioners, refrigeration compressors, components of central HVAC systems, and air terminal devices. AMCA *Standard* 300, ASHRAE *Standard* 130, and ARI *Standard* 880 establish special measuring procedures for some of these units. Large equipment that can operate on a large paved area, such as a parking lot, can also be measured under free-field conditions above a reflecting plane. Determining the sound power of large equipment outdoors is difficult; however, data may be available from some manufacturers.

A **diffuse field** is a sound field in which the sound intensity is the same in all directions and at every point. The **reverberant field** in a reverberation room, because of its sound reflecting walls, floor, and ceiling, provides an approximation to a diffuse field.

**Direct Method.** With the source in a reverberation room, the sound pressure level is measured at some minimum distance from the source and the surfaces of the room. The sound power level is calculated from the sound pressure level, if the rate of decay of sound (in dB/s) and the volume of the reverberation room are known.

Using the direct method, the relationship between sound power level and sound pressure level in a reverberation room is given by

$$L_w = L_p + 10 \log V + 10 \log D - 47.3 \quad (\text{direct method}) \quad (8)$$

where

$$\begin{aligned} L_p &= \text{sound pressure level averaged over room, dB re } 20 \text{ } \mu\text{Pa} \\ V &= \text{volume of room, ft}^3 \\ D &= \text{decay rate, dB/s} = 60/\theta \\ \theta &= \text{room reverberation time (time required for a 60 dB decay), s} \end{aligned}$$

**Substitution Method.** Most test standards use a calibrated **reference sound source** (RSS) to determine the sound power level of a device under test, such as a fan. The most common reference sound source is a small, direct-drive fan impeller that has no volute housing or scroll. The impeller is a forward-curved design, and a choke plate is installed on the inlet face of the impeller. The choke plate causes the fan to operate in a rotating-stall condition that is very noisy. The reference source is designed to have a stable sound power level output from 63 Hz to 8000 Hz and a relatively uniform frequency spectrum in each octave band.

To determine the sound power level of a given source, sound pressure level measurements are first made near the location of the given source with the reference sound source operating in the test room. Then the reference source is turned off and the measurements are repeated with the given source in operation. The differences in measured sound pressure levels between the reference source and unknown source represent differences in sound power level between the two. This procedure for sound power level determination is known as the **substitution method**.

Using this method, the relationship between sound power level and sound pressure level for the two sources is given by:

$$L_w = L_p + (L_w - L_p)_{\text{ref}} \quad (\text{substitution method}) \quad (9)$$

where

$$\begin{aligned} L_p &= \text{sound pressure level averaged over room, dB re } 20 \text{ } \mu\text{Pa} \\ (L_w - L_p)_{\text{ref}} &= \text{difference between sound power level and sound pressure level of reference sound source} \end{aligned}$$

### 3. Progressive Wave Method

By attaching a fan to one end of a duct, the sound energy is confined to a progressive wave field in the duct. Fan sound power can then be determined by measuring the sound pressure level inside the duct. The method is described in ASHRAE *Standard 68* (AMCA *Standard 330*). This method is not commonly used because of difficulties in constructing the required duct termination.

### 4. Sound Intensity Method

Advances in acoustical instrumentation now permit the direct determination of sound intensity, defined as the sound power per unit area flowing through a small element of a surface surrounding a source. The average sound power radiated by the source can be determined by measuring the sound intensity over the sphere or hemisphere surrounding a sound source. One of the advantages of this method is that, with certain limitations, sound intensity (and, therefore, sound power) measurements can be made in the presence of steady background sound in ordinary rooms, thereby eliminating the need for a special testing environment. Another advantage is that by measuring sound intensity over restricted areas around a sound source, sound directivity can be determined. This procedure can be particularly useful in reducing noise of products during their development.

International and United States standards that prescribe methods for making sound power measurements with sound intensity probes have been issued (ISO *Standard 9614-1*, ISO *Standard 9614-2*, ANSI *Standard S12.12*). In some situations, the sound fields may be so complex that measurements become impractical. A particular concern is that small test rooms or those having somewhat flexible boundaries (sheet metal or thin drywall) can permit a reactive sound field to exist, one in which the room's acoustical characteristics cause it to affect the sound power output of the source.

### Measurement Bandwidths for Sound Power

Sound power is normally determined in octave or 1/3 octave bands. Occasionally, a more detailed determination of the sound source spectrum is required. In these cases, narrow band analysis, using either constant fractional bandwidth (1/12 or 1/24 octave) or constant absolute bandwidth (e.g., 1 Hz) can be applied. The digital filter analyzer is most frequently used for constant percent bandwidth measurements, and the fast Fourier transform (FFT) analyzer is used for constant bandwidth measurements. Narrow band analysis results are used to determine the exact frequencies of pure tones and their harmonics in a sound spectrum.

## CONVERTING FROM SOUND POWER TO SOUND PRESSURE

The designer is often required to use the sound power level information on a source to predict the sound pressure level at a given location. The sound pressure level at a given location in a room due to a source of known sound power level depends on: (1) room volume, (2) room furnishings and surface treatments, (3) magnitude of the sound source(s), and (4) distance from the sound source(s) to the point of observation.

The classic relationship between source sound power level and room sound pressure level at some frequency is

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

where

- $L_p$  = sound pressure level, dB re 20  $\mu$ Pa
- $L_w$  = sound power level, dB re  $10^{-12}$  W
- $Q$  = directivity of the sound source (dimensionless)
- $r$  = distance from the source, ft
- $R$  = room constant,  $S\bar{\alpha}/(1 - \bar{\alpha})$
- $S$  = sum of all surface areas, ft<sup>2</sup>

$\bar{\alpha}$  = average absorption coefficient of room surfaces at given frequency

If the source is outside, far from reflecting surfaces, this relationship simplifies to

$$L_p = L_w + 10 \log (Q/4\pi r^2) + 10.5 \quad (11)$$

This relationship does not account for atmospheric absorption, weather effects and barriers. Note that the  $r^2$  term is present because the sound pressure in a free field decreases as  $1/r^2$  (the **inverse-square law**). Each time the distance from the source is doubled, the sound pressure level decreases by 6 dB.

For a simple source centered in a large, flat, reflecting surface,  $Q$  may be taken as 2. At the junction of two large flat surfaces,  $Q$  is 4, and in a corner,  $Q$  is 8.

In most typical rooms, the presence of acoustically absorbent surfaces and sound-scattering elements, such as furniture, creates a relationship between sound power and sound pressure level that is almost independent of the absorptive properties of the space. For example, hospital rooms, which have only a small amount of absorption, and executive offices, which have substantial absorption, are similar when the comparison is based on the same room volume and distance between the source and point of observation.

Equation (12) can be used to estimate the sound pressure level at a chosen observation point in a normally furnished room. The estimate is accurate to  $\pm 2$  dB (Schultz 1985).

$$L_p = L_w - 5 \log V - 3 \log f - 10 \log r + 25 \quad (12)$$

where

- $L_p$  = room sound pressure level at chosen reference point, dB re 20  $\mu$ Pa
- $L_w$  = source sound power level, dB re  $10^{-12}$  W
- $V$  = room volume, ft<sup>3</sup>
- $f$  = octave band center frequency, Hz
- $r$  = distance from source to observation point, ft

Equation (12) applies to a single sound source in the room itself, not to sources above the ceiling. With more than one source, total sound pressure level at the observation point is obtained by adding the contribution from each source in energy or power-like units, not decibels, and then converting back to sound pressure level. Warnock (1998b), reporting on ASHRAE *Research Project 755*, indicated that sound sources above ceilings may not act as point sources, and Equation (12) may not apply. [Chapter 47 of the ASHRAE Handbook—Applications](#) provides more information on this topic.

## SOUND TRANSMISSION PATHS

Sound from a source is transmitted along one or more paths to a receiver. Airborne and structureborne transmission paths are of principal concern for the HVAC system designer. Sound transmission between rooms occurs along both airborne and structureborne transmission paths. [Chapter 47 of the ASHRAE Handbook—Applications](#) has additional information on transmission paths.

### Airborne Transmission

**Atmospheric transmission.** Sound transmits readily through air, both indoors and outdoors. Indoor sound transmission paths include the direct, line-of-sight path between the source and the receiver, as well as reflected paths introduced by the presence of a room's walls, floor, ceiling and furnishings, which cause multiple sound reflection paths.

Outdoors, the effects of the reflections are small, if the source is not located near large reflecting surfaces. However, sound outdoors can refract and change propagation direction, due to the presence of wind and temperature gradient effects. Sound propagation outdoors

follows the inverse square law. Therefore, Equations (6) and (7) can generally be used to calculate the relationship between sound power level and sound pressure level for fully free-field and hemispherical free-field conditions, respectively.

**Ductborne transmission.** Ductwork can provide an effective sound transmission path because the sound is primarily contained within the boundaries of the ductwork. Sound can transmit both upstream and downstream from the source. A special case of ductborne transmission is **crostalk**, where sound is transmitted from one room to another via the duct path.

**Room-to-room transmission.** Room-to-room sound transmission generally involves both airborne and structureborne sound paths. The sound power incident on a room surface element undergoes three processes: (1) some of the sound energy is reflected from the surface element back into the room; (2) a portion of the sound energy is lost due to energy transfer into the material comprising the element, and (3) the remainder of the sound energy is transmitted through the element to the other room. Airborne sound is radiated as the element vibrates and structureborne sound can be transmitted via the studs of a partition or the floor and ceiling surfaces.

### Structureborne Transmission

Solid structures are efficient transmission paths for sound, which frequently originates as a vibration imposed on the transmitting structure. Typically, only a small amount of the input energy is radiated by the structure as airborne sound. A lightweight structure with little inherent damping radiates more sound than a massive structure with greater damping.

### Flanking Transmission

Sound from the source room can bypass the primary separating element and get into the receiving room along other paths called **flanking paths**. Common sound flanking paths include return air plenums, doors, and windows. Less obvious paths are those along floor and adjoining wall structures. Such flanking paths can seriously reduce the sound isolation between rooms. Flanking can explain poor sound isolation between spaces when the partition between spaces is known to provide very good sound insulation. Flanking can also explain sounds being heard in one room at a great distance from another room. Determining whether flanking sound transmission is important and what paths are involved can be difficult. Experience with actual situations and the theoretical aspects of flanking transmission is helpful. Sound intensity methods may be useful in determining flanking paths.

## TYPICAL SOURCES OF SOUND

Whenever mechanical power is generated or transmitted, a fraction of the power is converted into sound power and is radiated into the air. Therefore, virtually any major component of an HVAC system could be considered a sound source (e.g., fans, pumps, ductwork, piping, motors, etc.). The component's sound source characteristics depend upon its construction, its form of mechanical power and its integration with associated system components. The most important sound source characteristics include total sound power output, frequency distribution, and radiation directivity. All of these characteristics vary with frequency.

Sound sources in HVAC systems are so numerous that it is impractical to provide a complete listing here. Typical sources of sound and vibration in HVAC systems include

- Rotating and reciprocating equipment such as fans, motors, pumps, and chillers.
- Air and fluid sounds, such as those associated with flow through ductwork, piping systems, grilles, diffusers, terminal boxes, manifolds, and pressure-reducing stations.

- Excitation of surfaces—for example, friction; movement of mechanical linkages; turbulent flow impacts on ducts, plenum panels, and pipes; and impacts within equipment, such as cams and valve slap.
- Magnetostriction (transformer hum), which becomes significant in motor laminations, transformers, switchgear, lighting ballasts, and dimmers. A characteristic of magnetostrictive oscillations is that their fundamental frequency is twice the line frequency (120 Hz in a 60 Hz system.)

## CONTROLLING SOUND

### Terminology

The following noninterchangeable terms are used to describe the acoustical performance of many system components. ASTM *Standard C 634* defines additional terms.

**Sound attenuation** is a general term describing the reduction of the level of sound as it travels from a source to a receiver.

**Insertion loss (IL)** of a silencer or other sound-attenuating element is expressed in dB and is defined as the decrease in sound pressure level or sound intensity level, measured at a fixed receiver location, when the sound-attenuating element is inserted into the path between the source and the receiver. For example, if a straight, unlined piece of ductwork were replaced with a duct silencer, the sound level difference at a fixed location would be considered the silencer's insertion loss. Measurements are typically made in either octave or 1/3 octave bands.

**Sound transmission loss (TL)** of a partition or other building element is equal to 10 times the logarithm (base 10) of the ratio of the airborne sound power incident on the partition to the sound power transmitted by the partition and radiated on the other side. The quantity so obtained is expressed in decibels. Measurements are typically made in octave or 1/3 octave bands. [Chapter 47 of the ASHRAE Handbook—Applications](#) defines the special case of breakout transmission loss through duct walls.

**Noise reduction (NR)** is the difference between the average sound pressure levels produced in two enclosed spaces or rooms—a receiving room and a source room—by one or more sound sources in the source room. An alternate, non-ASTM definition of NR is the difference in sound pressure levels measured upstream and downstream of a duct silencer or sound-attenuating element. Measurements are typically made in octave or 1/3 octave bands.

**Random incidence sound absorption coefficient  $\alpha$**  is the fraction of the incident sound energy that is absorbed by a surface exposed to randomly incident sound. It is measured in a reverberation room using 1/3 octave bands of broadband sound (ASTM C 423). The sound absorption coefficient of a material in a specific 1/3 octave band depends on the material's thickness, airflow resistivity, stiffness, and method of attachment to the supporting structure.

**Spherical spreading** is the process by which sound level decreases with distance from a point source. It occurs when the sound source is located in free space or an anechoic room. Sound propagation in an anechoic space follows the *inverse square law*—6 dB level reduction per doubling of distance from a reference position.

**Scattering** is the change in direction of sound propagation due to an obstacle or inhomogeneity in the transmission medium. It results in the incident sound energy being dispersed in many directions.

### Enclosures and Barriers

Enclosing a sound source is a common means of controlling airborne sound transmission. This may be done using single- or double-leaf partitions.

The term **single-leaf partition** refers to all types of solid homogeneous panels where both faces are rigidly connected. Examples are gypsum board, plywood, concrete block, brick, and poured

concrete. The transmission loss of a single-leaf partition depends mainly on its surface mass (mass per unit area) because the heavier the partition, the less it vibrates in response to sound waves and the less sound it radiates on the side opposite the sound source. Increased surface mass can be achieved either by an increase in the partition's thickness or its density.

The **mass law** is a semi-empirical expression that may be used to predict transmission loss for randomly incident sound for thin, homogeneous single-leaf panels. It is written as

$$TL = 20 \log(w_s f) - 33 \quad (13)$$

where

$$w_s = \text{surface mass of panel, lb/ft}^2$$

$$f = \text{frequency, Hz}$$

The mass law predicts that transmission loss increases by 6 dB for each doubling of surface mass or frequency. If sound is incident only perpendicularly on the panel, the TL is about 5 dB greater than that predicted by the mass law.

Transmission loss also depends on material properties, such as stiffness and internal damping. The transmission losses of three single-leaf walls are illustrated in [Figure 2](#). For the 5/8 in. gypsum board, TL depends mainly on the surface mass of the wall at frequencies below about 1 kHz; agreement with the mass law is good. At higher frequencies, there is a dip in the TL curve, called the **coincidence dip** because it occurs at the frequency where the wavelength of flexural vibrations in the wall coincides with the wavelength of sound in the air. The frequency where the minimum value of TL occurs in the coincidence dip is called the **critical frequency**, which depends on material stiffness and thickness. The stiffer or thicker the layer of material, the lower the critical frequency. For example, the 6 in. concrete slab has a surface mass of about 75 lb/ft<sup>2</sup> and a coincidence frequency at 125 Hz. Thus, over most of the frequency range shown in [Figure 2](#), the transmission loss for the 6 in. concrete slab is well below that predicted by mass law. The coincidence dip for the 25 gage steel sheet occurs at high frequencies not shown in the figure.

The **sound transmission class (STC) rating** of a partition or assembly is a single number rating often used in architecture to classify sound isolation for speech. (ASTM E 90, ASTM E 413) Because the STC rating system was developed to deal with sound sources in the speech frequency range (125 to 4000 Hz), the rating should not be used as an indicator of an assembly's ability to control sound of any source that is rich in low frequencies. Most fan sound spectra have dominant low-frequency sound; therefore, to control fan sound, walls and slabs should be selected only on the basis of 1/3

octave or octave band sound transmission loss values, particularly at low frequencies.

Sound transmission loss values for ceiling tile are also inappropriate for estimating the reduction of sound between a sound source located in a ceiling plenum and the room below. *ARI Standard 885* has guidance on this topic.

Walls with identical STC ratings may not provide identical sound insulation at all frequencies. Because of the limited frequency range of most single number rating systems, designers should select partitions and floors on the basis of their 1/3 octave or octave band sound transmission loss values rather than single number ratings, especially when frequencies below 125 Hz are important.

For a given total mass in a wall or floor, much higher values of TL can be obtained by forming a double-leaf construction where each layer is independently or resiliently supported so vibration transmission between them is minimized. As well as mass, TL for such walls depends on cavity depth. Adding sound-absorbing material in the cavity significantly increases the TL relative to the unfilled cavity case. For further information on such walls, see [Chapter 47 of the ASHRAE Handbook—Applications](#).

If the sound fields in the rooms on each side of a panel are diffuse and the panel is the only significant path for sound between the rooms, the noise reduction NR is a function of the panel area  $S_p$  and the total sound absorption  $A_r$  in the receiving space, according to

$$NR = TL - 10 \log(S_p/A_r) \quad (14)$$

Because the total sound absorption in a room is expressed as the equivalent area of perfect sound absorption, both  $S_p$  and  $A_r$  are expressed in consistent units, usually square feet.

The sound reduction of an enclosure may be severely compromised by openings or leaks in the enclosure. Ducts that lead into or through a noisy space can carry sound to many areas of a building. Designers need to consider this factor when designing duct, piping, and electrical systems.

### Attenuation of Sound in Ducts and Plenums

Most ductwork, even a sheet metal duct without acoustical lining or silencers, attenuates sound to some degree. The natural attenuation of unlined ductwork is minimal, but can, especially for long runs of rectangular ductwork, significantly reduce ductborne sound. Acoustic lining of ductwork can greatly attenuate the propagation of sound through ducts, particularly at mid to high frequencies. [Chapter 47 of the ASHRAE Handbook—Applications](#) has a detailed discussion of lined and unlined ductwork attenuation.

If analysis shows that lined ductwork will not reduce sound propagation adequately, commercially available sound attenuators (also known as sound traps or duct silencers) can be used. There are three types: dissipative, reactive, and active. The first two are commonly known as **passive attenuators**.

- **Dissipative silencers** use absorptive media such as glass or rock fiber as the principal sound-absorption mechanism. Thick, perforated sheet metal baffles filled with low-density fiber insulation restrict the air passage width within the attenuator housing. The fiber is sometimes protected from the airstream by cloths or films. This type of attenuator is most effective in reducing mid- and high-frequency sound energy.
- **Reactive silencers** do not use any absorptive media to dissipate sound. This attenuator is typically used in HVAC systems serving hospitals, laboratories, or other areas with strict air quality standards. They are constructed only of metal, both solid and perforated. Chambers of specially designed shapes and sizes behind the perforated metal are tuned as resonators or expansion chambers to react with and reduce the sound power at selected frequencies. When designed for a broad frequency range, they are usually not as effective as dissipative attenuators and so are longer and

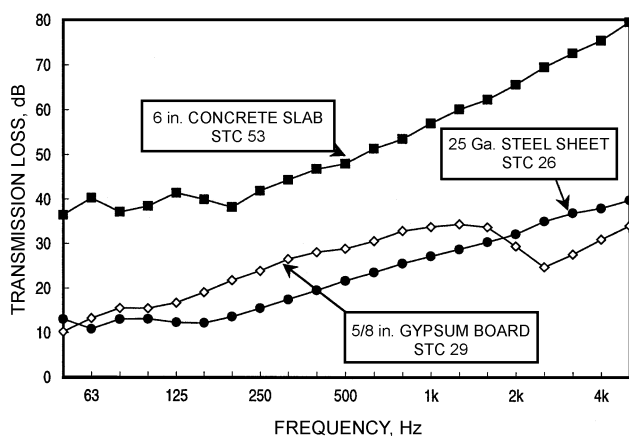


Fig. 2 Sound Transmission Loss Spectra for Single Layers of Some Common Materials

have a greater pressure drop. However, they can be highly effective and compact if designed for a limited frequency range such as for a pure tone.

- **Active silencer** systems use microphones, loudspeakers, and appropriate electronics to reduce in-duct sound by generating inverse-phase sound waves that destructively interfere with the incident sound energy. Microphones sample the sound field in the duct and loudspeakers generate signals with the opposite phase to the noise. Controlled laboratory experiments have shown that active attenuators reduce both broadband and tonal sound, but they are typically only effective in the 31.5 Hz through 250 Hz octave bands. Insertion losses of as much as 30 dB have been achieved under controlled conditions. The microphones and loudspeakers create a negligible pressure drop because they are mounted flush with the duct wall. Because active attenuators are not effective in the presence of excessively turbulent airflow, their use is limited to relatively long, straight duct sections with an air velocity less than about 1500 fpm.

Silencers are available for fans, cooling towers, air-cooled condensers, compressors, gas turbines, and many other pieces of commercial and industrial equipment. Silencers are normally installed on the intake or the discharge side (or both) of a fan or air handling unit. Also, they may be used on the receiver side of other noise generators such as terminal boxes, valves, and dampers.

Self-noise can limit an attenuator's effective insertion loss for air velocities in excess of about 2000 fpm. Use extreme caution when reviewing manufacturers' performance data for attenuators and duct liner materials to be sure that the test conditions are comparable to the specific design conditions. Short sections (3 to 5 ft) of insulated flexible duct are often very effective as attenuators. (ARI *Standard* 885 and [Chapter 47 of the ASHRAE Handbook—Applications](#) have information on typical flexible duct attenuation factors).

**End Reflections.** End reflection losses due to abrupt area changes in duct cross-section are sometimes useful in controlling low frequencies. The end reflection effect can be maximized at the end of a duct run by designing the last few feet or so of duct with the characteristic dimension of less than 15 in. Low-frequency noise reduction is inversely proportional to the characteristic dimension of the duct. However, abrupt area changes can generate high frequency noise, especially at high flow rates.

**Lined Plenums.** Where space is available, a lined plenum can provide excellent attenuation across a broad frequency range. The combination of end reflection at the plenum's entrance and exit, a large distance between the entrance and exit, and sound-absorbing lining on the plenum walls can be as effective as a sound attenuator, but with less pressure drop.

[Chapter 47 of the ASHRAE Handbook—Applications](#) has additional information on the control of sound.

### Standards for Testing Duct Silencers

Attenuators and duct liner materials are tested according to ASTM *Standard* E 477 in North America and ISO 7235 elsewhere. These define acoustical and aerodynamic performance in terms of insertion loss, self-generated noise (or self-noise), and airflow pressure drop. While many similarities exist, the ASTM and ISO standards produce differing results because of variations in loudspeaker location, orientation, duct termination conditions, and computation methods. Currently, no standard test methods are available to measure the attenuation of active silencers, although it is easy to measure in the field simply by turning the system on and off.

Insertion loss is measured in the presence of both forward and reverse flows. Forward flow occurs when the air and sound move in the same direction, as in a supply air or fan discharge system; reverse flow occurs when the air and sound travel in opposite directions, as in the case of a return air or fan intake system.

### SYSTEM EFFECTS

The way the HVAC components are assembled into a **system** affects the sound level generated by the system. Many engineers believe that satisfactory noise levels in occupied spaces can be achieved solely by using a manufacturer's sound ratings as a design tool, without consideration of the system influence.

Sound data provided by most manufacturers is obtained under standard laboratory test conditions. If the equipment is installed in a manner that differs from the test configuration, different configurations of connected ductwork, and interactions with other components of the installation, often significantly increase the operating noise level. For example, aerodynamically clean fan inlet and outlet conditions are rarely found in typical field applications. Furthermore, components such as silencers are frequently installed too close to the fan to allow a uniform velocity profile to exist at the entrance to the silencer. This results in a significantly higher than anticipated pressure drop across that component. The combination of these two effects changes the operating point on the fan curve. As a result, airflow is reduced and must be compensated for by increasing the fan speed, which may increase noise.

### HUMAN RESPONSE TO SOUND

#### Noise

Noise may be defined as any unwanted sound. Sound becomes noise when

- It is too loud—the sound is uncomfortable or makes speech difficult to understand
- It is unexpected (e.g., the sound of breaking glass)
- It is uncontrolled (e.g., a neighbor's lawn mower)
- It happens at the wrong time (e.g., a door slamming in the middle of the night)
- It contains pure tones (e.g., a whine, whistle, or hum)
- It contains unwanted information or is distracting (e.g., an adjacent telephone conversation or undesirable music)
- It is unpleasant (e.g., a dripping faucet)
- It connotes unpleasant experiences (e.g., a mosquito buzz or a siren wail)
- It is any combination of the above examples

To be noise, sound does not have to be loud, just unwanted. In addition to being annoying, loud noise can cause hearing loss, and, depending on other factors, it could affect stress level, sleep patterns and heart rate.

To increase privacy, broadband sound may be radiated into a room from a well-designed air-conditioning system to mask or hide low-level intrusive sounds from adjacent spaces. This controlled sound may be referred to as noise, but not in the context of unwanted sound; rather, it is a broadband, neutral sound that is frequently unobtrusive. Three types of broadband noise are frequently encountered in acoustics:

- **Random noise** is an oscillation, the instantaneous magnitude of which is not specified for any given instant. The instantaneous magnitudes of a random noise are specified only by probability distributions, giving the fraction of the total time that the magnitude, or some sequence of magnitudes, lies within a specified range (ANSI *Standard* S1.1).
- **White noise** is noise with a continuous frequency spectrum with equal energy per hertz over a specified frequency range. White noise is not necessarily random. Since octave bands double in width for each successive band, for white noise the energy also doubles in each successive octave band. Thus white noise displayed on a 1/3 octave or octave band chart increases by 3 dB per octave.
- **Pink noise** is noise with a continuous frequency spectrum but equal energy per constant-percentage bandwidth, such as per

octave or 1/3 octave band. Thus pink noise appears on a one-third octave or octave band chart as a horizontal line.

**Predicting Human Response to Sound**

Predicting the response of people to any given sound is, at best, only a statistical concept, and, at worst, very inaccurate. This is because response to sound is not only physiological but psychological and depends on the varying attitude of the listener. Hence, the effect of sound is often unpredictable. However, people’s response is adverse if the sound is considered too loud for the situation or if it sounds “wrong.” Therefore, most criteria are based on descriptors that account for level and spectrum shape.

**Sound Quality**

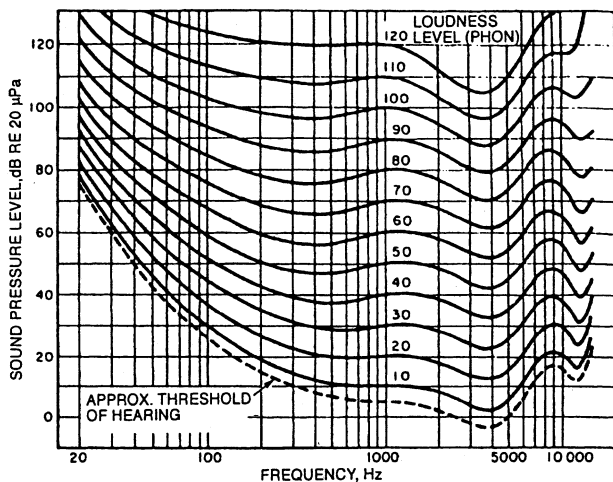
To determine the acoustic acceptability of a space to occupants, the sound pressure levels there must be known. This, however, is often not sufficient; the **sound quality** is important too. Factors influencing sound quality include (1) loudness, (2) tone perception, (3) frequency spectrum, (4) harshness, (5) time and frequency fluctuation, and (6) vibration.

People often perceive sounds with tones (like a whine or hum) as particularly annoying. A tone can cause a relatively low level sound to be perceived as noise. Studies have been done to characterize sounds with and without pure tones.

**Loudness**

The primary method used to determine a subjective estimate of loudness is to present sounds to a sample of human listeners under controlled conditions. To determine the loudness of a sound, listeners compare an unknown sound with a standard sound. (The accepted standard sound is a pure tone of 1000 Hz or a narrow band of random noise centered on 1000 Hz.) Loudness level is expressed in **phons**, and the loudness level of any sound in phons is equal to the sound pressure level in decibels of a standard sound deemed to be equally loud. Thus, a sound that is judged as loud as a 40 dB, 1000 Hz tone has a loudness level of 40 phons.

Average reactions of humans to tones are shown in [Figure 3](#) (Robinson and Dadson 1956). The reaction changes when the sound is a band of random noise (Pollack 1952), rather than a pure tone ([Figure 4](#)). The figures indicate that people are most sensitive in the mid-frequency range. The contours in [Figure 3](#) are closer together at low frequencies showing that at lower frequencies, although people are less sensitive to sound level, they are more sensitive to changes in level.



**Fig. 3 Free-Field Equal Loudness Contours for Pure Tones**

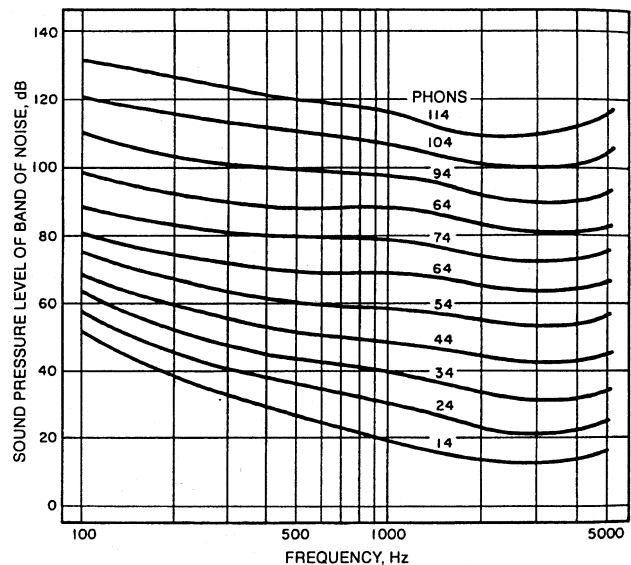
**Table 7 Subjective Effect of Changes in Sound Pressure Level, Broadband Sounds (Frequency > 250 Hz)**

Subjective Change	Objective Change in Sound Level (Approximate)
Much louder	More than +10 dB
Twice as loud	+10 dB
Louder	+5 dB
Just perceptibly louder	+3 dB
Just perceptibly quieter	-3 dB
Quieter	-5 dB
Half as loud	-10 dB
Much quieter	Less than -10 dB

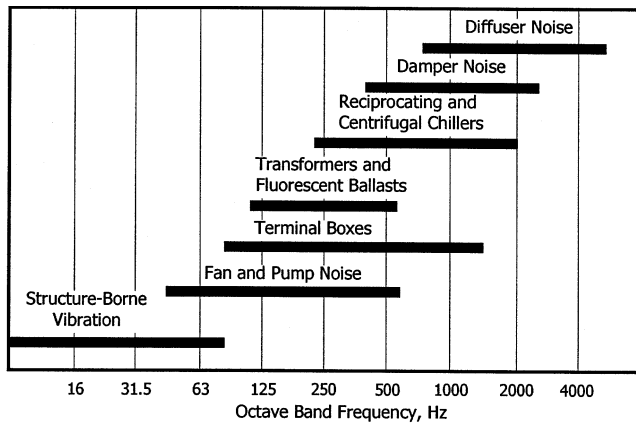
Under carefully controlled experimental conditions, humans can detect small changes in sound level. However, for humans to describe a sound as being half or twice as loud requires changes in overall sound pressure level of about 10 dB. For many people, a 3 dB change is the minimum perceptible difference. This means that halving the power output of the source causes a barely noticeable change in sound pressure level, and the power output must be reduced by a factor of 10 before humans determine that loudness has been halved. [Table 7](#) summarizes the effect of changes in sound levels for simple sounds in the frequency range of 250 Hz and higher.

The phon scale covers the large dynamic range of the ear, but it does not fit a subjective linear loudness scale. Over most of the audible range, a doubling of loudness corresponds to a change of approximately 10 phons. To obtain a quantity proportional to the loudness sensation, a loudness scale is defined in which the unit of loudness is known as a **sone**. One sone equals the loudness level of 40 phons. A rating of two sones corresponds to 50 phons and so on.

The results of such work have led to the development of standard objective methods for calculating loudness. ANSI *Standard S3.4* calculates loudness or loudness level by using octave-band sound pressure level data as a starting point. The loudness index for each octave band is obtained from a graph or by calculation. Total loudness is then calculated by combining the loudnesses for each band according to a formula given in the standard. A more complex calculation method using 1/3 octave band sound pressure levels by Zwicker (ISO *Standard 532*) or the German *Standard DIN 45631* is



**Fig. 4 Equal Loudness Contours for Relatively Narrow Bands of Random Noise**



**Fig. 5** Frequencies at Which Various Types of Mechanical and Electrical Equipment Generally Control Sound Spectra

tones. Due to its complexity, loudness has not been widely used in engineering practice in the past. However, with an increased awareness of sound quality and the availability of software for calculating loudness, this measure is now being used more frequently.

AMCA Publication 302 describes how the **sonic method** is applied to rating the relative loudness of fans and ventilators. This calculation method is usually acceptable when the measured sound spectrum has no strong tonal components.

**Acceptable Frequency Spectrum**

The most acceptable frequency spectrum for HVAC sound is a balanced or neutral spectrum. This means that it is not too “hissy” (excessive high frequency content) or too “rumbly” (excessive low-frequency content). Unfortunately, achieving a balanced sound spectrum is not always easy—there may be a multiplicity of sound sources to consider. As a guide to the designer, Figure 5 shows the more common mechanical and electrical sound sources and frequency regions that control the indoor sound spectrum. Chapter 47 of the ASHRAE Handbook—Applications provides more detailed information on treating some of these sound sources.

**SOUND RATING SYSTEMS AND ACOUSTICAL DESIGN GOALS**

Several background sound rating methods are used to rate indoor sound. They include the A-weighted sound pressure level (dBA) and noise criteria (NC), the more recent room criteria (RC) and balanced noise criteria (NCB), and the new RC Mark II. Each sound rating method was developed from data for specific applications; not all methods are equally suitable for the rating of HVAC-related sound in the variety of applications encountered.

The degree of occupant satisfaction achieved with a given level of background sound is determined by many factors. For example, large conference rooms, auditoriums, and recording studios can tolerate only a low level of background sound. On the other hand, higher levels of background sound are acceptable and even desirable in certain situations, such as in open-plan offices where a certain amount of speech and activity masking is essential. Therefore, the system sound control goal varies depending on the required use of the space.

To be unobtrusive, background sound should have the following properties:

- A balanced distribution of sound energy over a broad frequency range
- No audible tonal or other characteristics such as whine, whistle, hum, or rumble

**Table 8** Comparison of Sound Rating Methods

Method	Overview	Evaluates Sound Quality	Used For Rating of
dBA	<ul style="list-style-type: none"> <li>• Can be determined using sound level meter</li> <li>• No quality assessment</li> <li>• Frequently used for outdoor noise ordinances</li> </ul>	No	Cooling towers Water chillers Condensing units
NC	<ul style="list-style-type: none"> <li>• Can rate components</li> <li>• No quality assessment</li> <li>• Does not evaluate low frequency rumble, frequencies &lt;63 Hz</li> </ul>	No	Air terminals Diffusers
NCB	<ul style="list-style-type: none"> <li>• Can rate components</li> <li>• Some quality assessment</li> </ul>	Yes	
RC	<ul style="list-style-type: none"> <li>• Used to evaluate systems</li> <li>• Should not be used to evaluate components</li> <li>• Can be used to evaluate sound quality</li> <li>• Provides some diagnostic capability</li> </ul>	Yes	
RC Mark II	<ul style="list-style-type: none"> <li>• Evaluates sound quality</li> <li>• Provides improved diagnostics capability</li> </ul>	Yes	

- No noticeable time-varying levels from beats or other system-induced aerodynamic instability
- No fluctuations in level such as a throbbing or pulsing

At present, no acceptable process easily characterizes the effects of audible tones and level fluctuations. The preferred sound rating methods generally comprise two distinct parts: a family of criteria curves (specifying sound levels by octave bands), and a companion procedure for rating the calculated or measured sound data relative to the criterion curves. A table of recommended design goals can be found in Chapter 47 of the ASHRAE Handbook—Applications.

Table 8 summarizes the essential differences, advantages and disadvantages of the rating methods that are used to characterize HVAC-related background sound. The text following the table gives more information on each rating. Note that all the ratings in the table consider speech interference effects and all are currently used for rating background noise.

**A-Weighted Sound Level (dBA)**

The A-weighted sound level  $L_A$  is widely used to state acoustical design goals as a single number, but its usefulness is limited because it gives no information on spectrum content. The rating is expressed as a number followed by dBA, for example 40 dBA.

A-weighted sound levels correlate well with human judgments of relative loudness, but give no information on spectral balance. Thus, they do not necessarily correlate well with the annoyance caused by the noise. Many different-sounding spectra can have the same numeric rating, but have quite different subjective qualities. A-weighted comparisons are best used with sounds that sound alike but differ in level. They should not be used to compare sounds with distinctly different spectral characteristics; that is, two sounds at the same sound level but with different spectral content are likely to be judged differently by the listener in terms of acceptability as a background sound. One of the sounds might be completely acceptable, while the other could be objectionable because its spectrum shape was rumbly, hissy, or tonal in character.

A-weighted sound levels are used extensively in outdoor environmental noise standards.

**Noise Criteria (NC) Method**

The NC method is a single-number rating that is somewhat sensitive to the relative loudness and speech interference properties of

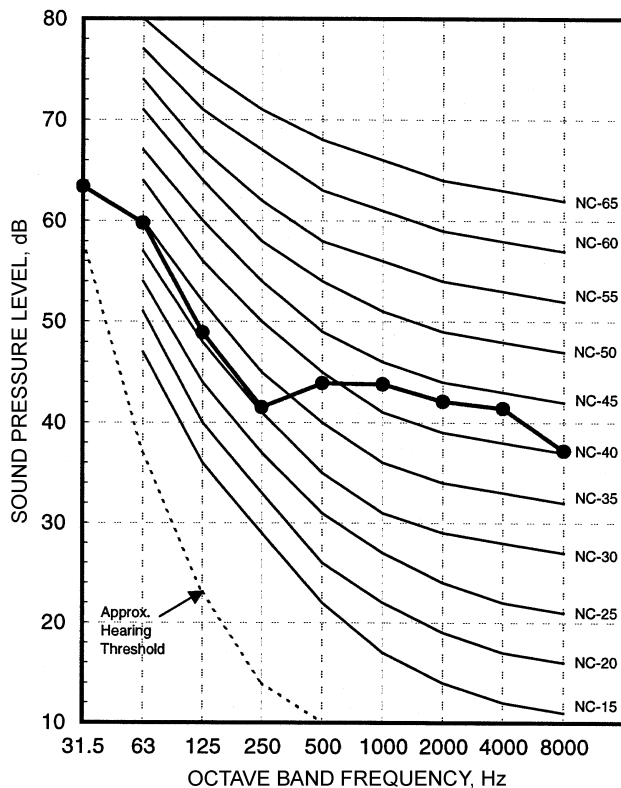
a given sound spectrum. The method consists of a family of criteria curves extending from 63 to 8000 Hz, and a **tangency rating procedure** (Beranek 1957). The criteria curves, shown in [Figure 6](#), define the limits of octave band spectra that must not be exceeded to meet occupant acceptance in certain spaces. The rating is expressed as NC followed by a number. For example, the spectrum shown in [Figure 6](#) is rated NC 45 because this is the lowest rating curve that falls entirely above the measured data. An NC 35 design goal is commonly used for private offices. The background sound level meets this goal if no portion of its spectrum lies above the designated NC 35 curve.

The NC method is sensitive to level but has the disadvantage that the tangency method used to determine the rating does not require that the sound spectrum approximate the shape of the NC curves. Thus, many different sounds can have the same numeric rating, but rank differently on the basis of subjective sound quality. In HVAC systems that do not produce excessive low frequency sound, the NC rating correlates relatively well with occupant satisfaction *if* sound quality is not a significant concern.

Two problems occur in using the NC procedure: (1) when the NC level is determined by a prominent peak in the spectrum, the actual level of resulting background sound may be quieter than that desired for masking unwanted speech and activity sounds, because the spectrum on either side of the tangent peak drops off too rapidly; and (2) when the measured spectrum closely matches the shape of the NC curve, the resulting sound is either rumbly or hissy or both.

The shape of the NC curve is not that of a well-balanced, neutral sound; thus, these curves should be used with caution in critical situations where the background sound of an air-conditioner is required to mask speech and activity sound.

NC contours are used to calculate ratings for some HVAC components such as terminal units and diffusers. NC ratings should not be used to characterize fans and air-handling units.



**Fig. 6 NC (Noise Criteria) Curves and Sample Spectrum (Curve with Symbols)**

**Balanced Noise Criteria (NCB) Method**

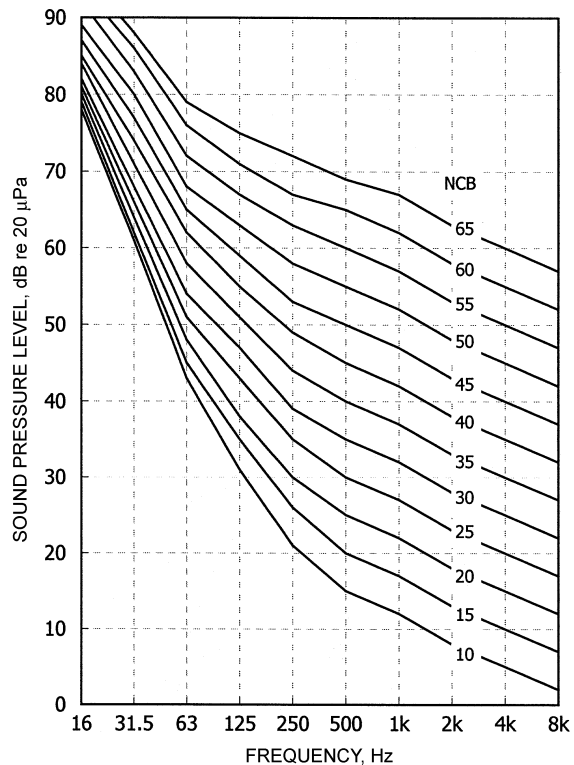
The NCB method (Beranek 1989, ANSI *Standard* S12.2) is a specification or evaluation of room sound including noise due to occupant activities. The NCB criteria curves ([Figure 7](#)) are intended as replacements for the NC curves, and include both the addition of two low-frequency octave bands (16 and 31.5 Hz) and lower permissible sound levels in the high-frequency octave bands (4000 and 8000 Hz). The NCB rating procedure is based on the **speech interference level (SIL)**, which is the arithmetic average of the sound pressure levels in the four frequency bands: 500, 1000, 2000, and 4000 Hz. Additional tests include rumble and hiss compliance. The rating is expressed as NCB followed by a number, for example, NCB 40.

The NCB method is better than the NC method in determining whether a sound spectrum has a shape sufficiently unbalanced to demand corrective action. Also, it addresses the issue of low-frequency sound. The rating procedure is somewhat more complicated than the tangency rating procedures.

**Room Criteria (RC) Method**

For some time, the RC method (Blazier 1981a,b; ANSI *Standard* S12.2) was recommended as the preferred method for rating HVAC-related sound. The RC curves were intended to establish HVAC system design goals. The revised RC Mark II method (discussed below) is now preferred.

The RC method consists of a family of criteria curves and a rating procedure. The shape of these curves differs from the NC curves to approximate a well-balanced neutral-sounding spectrum, and two additional octave bands (16 and 31.5 Hz) are added to deal with low-frequency sound. This rating procedure assesses background sound in spaces based on the effect of the sound on speech communication, and on estimates of subjective sound quality. The rating is



**Fig. 7 NCB (Noise Criteria Balanced) Curves Drawn from ANSI Standard S12.2**

expressed as RC followed by a number to show the level of the sound and a letter to indicate the quality, for example RC 35(N) where N denotes neutral.

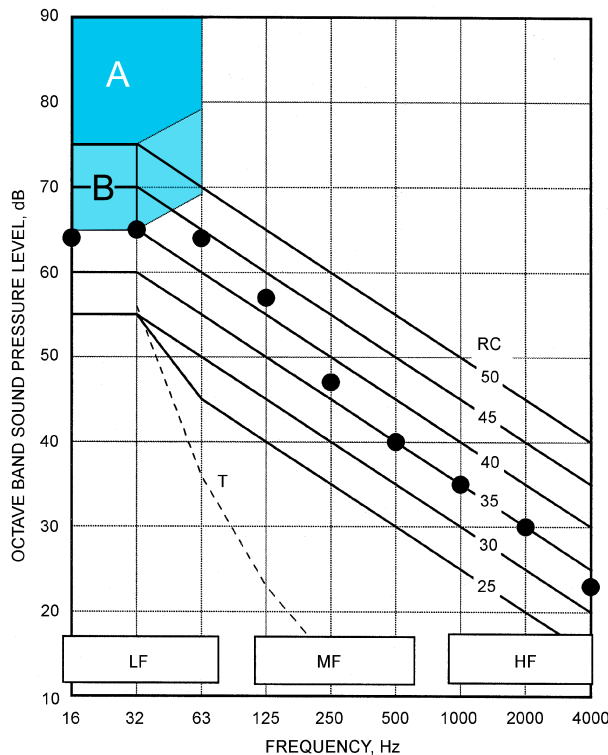
**RC Mark II Room Criteria Method**

Based on experience and the findings from ASHRAE-sponsored research, the RC method was revised to the RC Mark II method (Blazier 1997). Like its predecessor, the RC Mark II method is intended for rating the sound performance of an HVAC system as a whole. The method can also be used as a diagnostic tool for analyzing sound problems in the field. The RC Mark II method is more complicated to use than the RC method, but spreadsheet macros are available to do the calculations and graphical analysis.

The RC Mark II method of rating HVAC system sound comprises three parts:

- Family of criteria curves (Figure 8)
- Procedure for determining the RC numerical rating and the sound spectral balance (quality)
- Procedure for estimating occupant satisfaction when the spectrum does not have the shape of an RC curve (Quality Assessment Index) (Blazier 1995)

The rating is expressed as RC followed by a number and a letter, for example, RC 45(N). The number is the arithmetic average rounded to the nearest integer of the sound pressure levels in the 500, 1000, and 2000 Hz octave bands (the principal speech frequency region). The letter is a qualitative descriptor that identifies the perceived character of the sound: (N) for neutral, (LF) for low-frequency rumble, (MF) for mid-frequency roar, and (HF) for



Sound levels in Region B may generate perceptible vibration in light wall and ceiling construction. Rattles in light fixtures, doors, windows, etc., are a slight possibility. Sound levels in Region A have a high probability of generating easily perceptible sound induced vibration in light wall and ceiling construction. Audible rattling in light fixtures, doors, windows etc. may be anticipated. The text explains Regions LF, MF, and HF. The solid dots are octave band sound pressure levels for the example in the text.

**Fig. 8 Room Criteria Curves, Mark II**

high-frequency hiss. In addition, the low-frequency descriptor has two subcategories: (LF<sub>B</sub>), denoting a moderate but perceptible degree of sound induced ceiling/wall vibration, and (LF<sub>A</sub>), denoting a noticeable degree of sound induced vibration.

Each reference curve in Figure 8 identifies the shape of a neutral, bland-sounding spectrum, indexed to a curve number corresponding to the sound level in the 1000 Hz octave band. The shape of these curves is based on research by Blazier (1981a,b) and modified at 16 Hz following recommendations by Broner (1994). Regions A and B denote levels at which sound can induce vibration in light wall and ceiling constructions that can potentially cause rattles in light fixtures, furniture, etc. Curve T is the octave-band threshold of hearing as defined by ANSI Standard 12.2.

**Procedure for Determining the RC Mark II Rating for a System**

**Step 1.** Determine the appropriate RC reference curve. This is done by obtaining the arithmetic average of the sound levels in the principle speech frequency range represented by the levels in the 500, 1000, and 2000 Hz octave bands. The RC reference curve is chosen as that which has the same value at 1000 Hz as the calculated average value (rounded to the nearest integer). This curve is not to be confused with the speech-interference level (SIL), which is a four-band average obtained by including the 4000 Hz octave band.

**Step 2.** Assign a subjective quality by calculating the **Quality Assessment Index (QAI)** (Blazier 1995). This index is a measure of the degree the shape of the spectrum under evaluation deviates from the shape of the RC reference curve. The procedure requires calculation of the **energy-average spectral deviations** from the RC reference curve in each of three frequency groups: low frequency, LF (16–63 Hz), medium frequency, MF (125–500 Hz), and high frequency, HF (1000–4000 Hz). The procedure for the LF region is given by Equation (15) and is repeated in the MF and HF regions by substituting the corresponding values at each frequency. However, when evaluating typical HVAC-related sounds, a simple arithmetic average of these deviations is often adequate if the range of values does not exceed 3 dB.

$$\Delta LF = 10 \log [10^{0.1 \Delta L_{16}} + 10^{0.1 \Delta L_{31.5}} + 10^{0.1 \Delta L_{63}} / 3] \quad (15)$$

where the  $\Delta L$  terms are the differences between the spectrum being evaluated and the RC reference curve in each frequency band. In this way, three spectral deviation factors ( $\Delta LF$ ,  $\Delta MF$ ,  $\Delta HF$ ), expressed in dB with either positive or negative values, are associated with the spectrum being rated. QAI is the **range** in dB between the highest and lowest values of the spectral deviation factors.

If QAI  $\leq 5$  dB, the spectrum is assigned a neutral (N) rating. If QAI exceeds 5 dB, the sound quality descriptor of the RC rating is the letter designation of the frequency region of the deviation factor having the highest positive value. As an example, the spectrum plotted in Figure 8 is processed in Table 9.

The arithmetic average of the sound levels in the 500, 1000, and 2000 Hz octave bands in Figure 8 is 35 dB, so the RC 35 curve is selected as the reference for spectrum quality evaluation. The spectral deviation factors in the LF, MF, and HF regions are 6.6, 4.0 and -0.6, respectively, giving a QAI of 7.2. The maximum positive deviation factor occurs in the LF region, and the QAI exceeds 5, resulting in a rating of RC 35(LF). An average room occupant should perceive this spectrum as marginally rumbly in character (see Table 10).

**Estimating Occupant Satisfaction Using QAI**

The quality assessment index (QAI) is useful in estimating the probable reaction of an occupant when the system does not produce optimum sound quality. The basis for the procedure outlined here for estimating occupant satisfaction is as follows:

**Table 9 Example Calculation of RC Mark II Rating for Sound Spectrum in Figure 8**

	Frequency, Hz								
	16	31.5	63	125	250	500	1000	2000	4000
Sound pressure	64	65	64	57	47	40	35	30	23
Average sound pressure at 500-2000 Hz							35		
RC contour	60	60	55	50	45	40	35	30	25
Levels: RC contour	4	5	9	7	2	0	0	0	-2
	LF			MF			HF		
Spectral deviations	6.6			4.0			-0.6		
QAI	6.6 - (-0.6) = 7.2								
RC Mark II rating	RC 35(LF)								

- Changes in sound level of less than 5 dB do not cause subjects to change their ranking of sounds of similar spectral content. A QAI of 5 dB or less corresponds to a generally acceptable condition, provided that the perceived level of the sound is in a range consistent with the given type of space occupancy.
- A QAI that exceeds 5 dB but is less than or equal to 10 dB represents a marginal situation in which the acceptance by an occupant is questionable.
- A QAI greater than 10 dB will likely be objectionable to the average occupant.

Table 10 lists sound quality descriptors and QAI values and relates them to probable occupant reaction to the sound. An exception to this rule occurs when the sound pressure levels in the 16 Hz or 31.5 Hz octave-bands exceed 65 dB. In such cases, the potential for acoustically-induced vibration in typical lightweight office construction should be considered. If the levels in these bands exceed 75 dB, a significant problem with induced vibration is probable.

Undoubtedly situations will occur in the assessment of HVAC-related sound where the numerical part of the RC rating is less than the specified maximum for the space use, but the sound quality descriptor is other than the desirable (N). For example, a maximum of RC 40(N) is specified, but the actual sound environment turns out to be RC 35(MF). Knowledge in this area is insufficient to decide which spectrum is preferable.

Even at moderate levels, if the dominant portion of the background sound occurs in the very low-frequency region, some people experience a sense of oppressiveness or depression in the environment (Persson-Waye et al. 1997). In such situations, the basis for

complaint may result from exposure to that environment for several hours, and thus may not be noticed during a short exposure period.

### Criteria Selection Guidelines

In general, these basic guidelines are important:

- Sound levels below NCB 35 or RC 35 are not detrimental to good speech intelligibility; those at or above NCB 35 or RC 35 may interfere with or mask speech.
- Even if the occupancy sound will be significantly higher than the anticipated background sound level generated by mechanical equipment, the sound design goal should not necessarily be raised to levels approaching the occupancy sound. This avoids occupants having to raise their voices uncomfortably to be heard over the noise.

Table 11 gives recommended design criteria.

## FUNDAMENTALS OF VIBRATION

A rigidly mounted machine transmits its internal vibratory forces directly to the supporting structure. However, by inserting resilient mountings, called **vibration isolators**, between the machine and supporting structure, the magnitude of transmitted vibration can be reduced to only a fraction of the original. Vibration isolators can also be used to protect sensitive equipment from disturbing vibrations that may be present in the floor of the building structure.

### SINGLE-DEGREE-OF-FREEDOM MODEL

The simplest example of a vibration isolation system is the single-degree-of-freedom model illustrated in Figure 9. In this instance only motion along the vertical axis is considered and damping is disregarded. This is the model upon which most manufacturers of vibration isolation hardware base their catalog information.

The application of a single-degree-of-freedom model to the isolation of HVAC equipment is valid only when the stiffness of the supporting structure is large with respect to the stiffness of the vibration isolator. Under these conditions, the natural frequency  $f_n$  of the system is

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{M}} \quad (16)$$

**Table 10 Definition of Sound Quality Descriptor and Quality Assessment Index (QAI) to Aid in Interpreting RC Mark II Ratings of HVAC-Related Sound**

Sound Quality Descriptor	Description of Subjective Perception	Magnitude of QAI	Probable Occupant Evaluation, Assuming Level of Specified Criterion is Not Exceeded
(N) Neutral (Bland)	Balanced sound spectrum, no single frequency range dominant	QAI ≤ 5 dB, $L_{16}, L_{31.5} \leq 65$ QAI ≤ 5 dB, $L_{16}, L_{31.5} > 65$	Acceptable Marginal
(LF) Rumble	Low-frequency range dominant (16 – 63 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable
(LFV <sub>B</sub> ) Rumble, with moderately perceptible room surface vibration	Low-frequency range dominant (16 – 63 Hz)	QAI ≤ 5 dB, $65 < L_{16}, L_{31.5} < 75$ 5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Marginal Objectionable
(LFV <sub>A</sub> ) Rumble, with clearly perceptible room surface vibration	Low-frequency range dominant (16 – 63 Hz)	QAI ≤ 5 dB, $L_{16}, L_{31.5} > 75$ 5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Marginal Objectionable
(MF) Roar	Mid-frequency range dominant (125 – 500 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable
(HF) Hiss	High-frequency range dominant (1000 – 4000 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable

**Table 11 Design Guidelines for HVAC-Related Background Sound in Rooms**

Room Types	RC(N); QAI ≤ 5dB Criterion <sup>a,b</sup>
<b>Residences, Apartments, Condominiums</b>	25 – 35
<b>Hotels/Motels</b>	
Individual rooms or suites	25 – 35
Meeting/banquet rooms	25 – 35
Corridors, lobbies	35 – 45
Service/support areas	35 – 45
<b>Office Buildings</b>	
Executive and private offices	25 – 35
Conference rooms	25 – 35
Teleconference rooms	25 (max)
Open-plan offices	30 – 40
Corridors and lobbies	40 – 45
<b>Hospitals and Clinics</b>	
Private rooms	25 – 35
Wards	30 – 40
Operating rooms	25 – 35
Corridors and public areas	30 – 40
<b>Performing Arts Spaces</b>	
Drama theaters	25 (max)
Concert and recital halls <sup>c</sup>	
Music teaching studios	25 (max)
Music practice rooms	35 (max)
<b>Laboratories (with fume hoods)</b>	
Testing/research, minimal speech communication	45 – 55
Research, extensive telephone use, speech communication	40 – 50
Group teaching	35 – 45
<b>Churches, Mosques, Synagogues</b>	
General assembly	25 – 35
With critical music programs <sup>c</sup>	
<b>Schools <sup>d</sup></b>	
Classrooms up to 750 ft <sup>2</sup>	40 (max)
Classrooms over 750 ft <sup>2</sup>	35 (max)
Large lecture rooms, without speech amplification	35 (max)
<b>Libraries</b>	30 – 40
<b>Courtrooms</b>	
Unamplified speech	25 – 35
Amplified speech	30 – 40
<b>Indoor Stadiums, Gymnasiums</b>	
Gymnasiums, natatoriums, and large seating-capacity spaces with speech amplification <sup>e</sup>	40 – 45

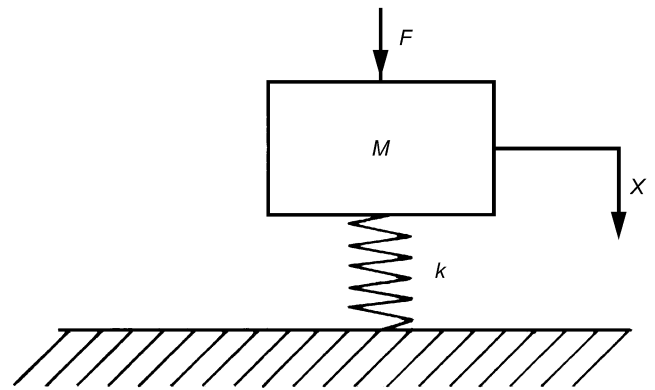
<sup>a</sup>The values and ranges are based on judgment and experience, not on quantitative evaluations of human reactions. They represent general limits of acceptability for typical building occupancies. Higher or lower values may be appropriate and should be based on a careful analysis of economics, space use, and user needs.

<sup>b</sup>When quality of sound in the space is important, specify criteria in terms of RC(N). If the quality of the sound in the space is of secondary concern, the criteria may be specified in terms of NC or NCB levels of similar magnitude.

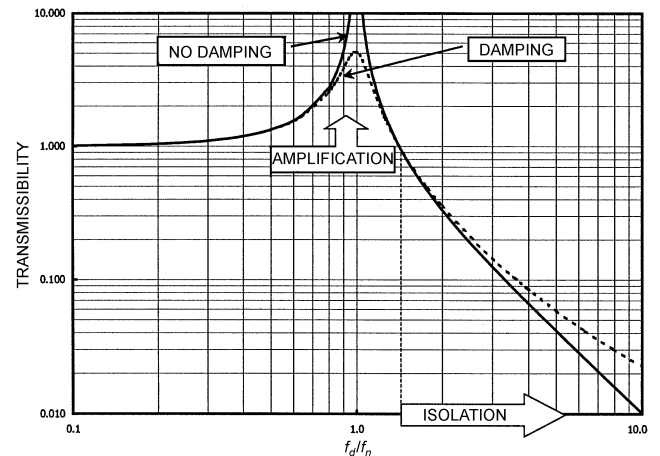
<sup>c</sup>An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces.

<sup>d</sup>HVAC-related sound criteria for schools, such as those listed in this table, may be too high and impede learning by children in primary grades whose vocabulary is limited, or whose first language is not the language of the class. Some educators and others believe that the HVAC-related background sound should not exceed RC 25(N).

<sup>e</sup>RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.



**Fig. 9 Single-Degree-of-Freedom System**



**Fig. 10 Vibration Transmissibility  $T$  as a Function of  $f_d/f_n$**

where  $k$  is the stiffness of the vibration isolator (force per unit deflection) and  $M$  is the mass of the equipment supported by the isolator. This equation simplifies to

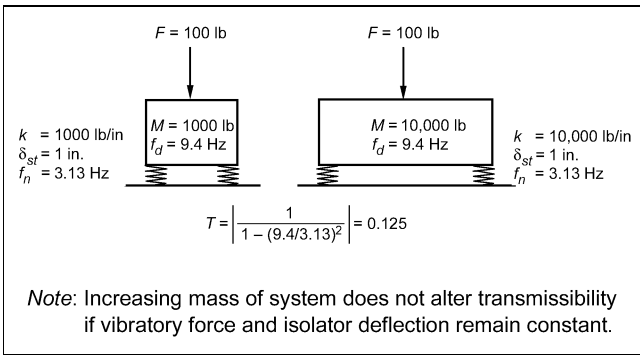
$$f_n = \frac{3.13}{\sqrt{\delta_{st}}} \tag{17}$$

where  $\delta_{st}$  is the isolator static deflection in inches (the incremental distance the isolator spring compresses under the weight of the supported equipment, or  $k/M = g/\delta_{st}$ ). Thus, to achieve the appropriate system natural frequency for a given application, the corresponding isolator static deflection and the load to be supported at each mount-point is specified.

The **transmissibility** is the ratio of the amplitudes of the force transmitted to the building structure to the exciting force produced by the vibrating equipment. Transmissibility  $T$  is inversely proportional to the square of the ratio of the disturbing frequency  $f_d$  to the system natural frequency  $f_n$ , or

$$T = \left| \frac{1}{1 - (f_d/f_n)^2} \right| \tag{18}$$

At  $f_d = f_n$ , **resonance** occurs (the denominator of Equation (18) equals zero), with theoretically infinite transmission of vibration. In practice, however, some limit on the transmission at resonance exists because inherent damping is always present to some degree. Thus, the magnitude of vibration amplification at resonance always has a finite value. Equation (18) is plotted in [Figure 10](#).



**Fig. 11 Effect of Mass on Transmissibility**

Vibration isolation does not begin to occur until  $f_d/f_n > 1.4$ . Above this ratio, the vibration transmissibility rapidly decreases. A frequency ratio of at least 3.5 is often specified, which corresponds to an isolation efficiency of about 90%, or 10% transmissibility. Higher ratios may be specified, but in practice this does not generally result in isolation efficiencies any greater than about 90%. The reason is that “wave-effects” and other nonlinear characteristics cause typical isolators to depart from the theoretical curve that limits its performance.

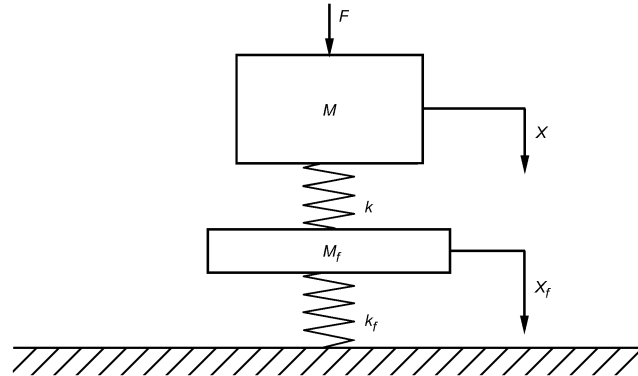
If the mass of the equipment is increased, the resonance frequency decreases, thus increasing the isolation. In practice, the load-carrying capacity of isolators usually requires that their stiffness or their number be increased. Consequently, the static deflection and the transmissibility may remain unchanged. The use of stiffer springs leads, however, to smaller vibration amplitudes—less movement of the equipment. This is one of the main reasons for placing some high-power or highly eccentric equipment on inertia pads.

For example, as shown in [Figure 11](#), a 1000 lb piece of equipment installed on isolators with stiffness  $k$  of 1000 lb<sub>f</sub>/in. results in a 1 in. deflection and a system resonance frequency  $f_n$  of 3.13 Hz. If the equipment is operated at 564 rpm (9.4 Hz) and develops a force of 100 lb<sub>f</sub>, a  $100 \times 0.125 = 12.5$  lb<sub>f</sub> force is transmitted to the structure. If the total mass is increased to 10,000 lb by placing the equipment on a concrete inertia base and the stiffness of the springs is increased to 10,000 lb<sub>f</sub>/in., the deflection is still 1 in., the resonance frequency of the system is maintained at 3.13 Hz, and the force transmitted to the structure remains at 12.5 lb<sub>f</sub>. The increased mass, however, reduces the equipment displacement.

**TWO-DEGREE-OF-FREEDOM MODEL**

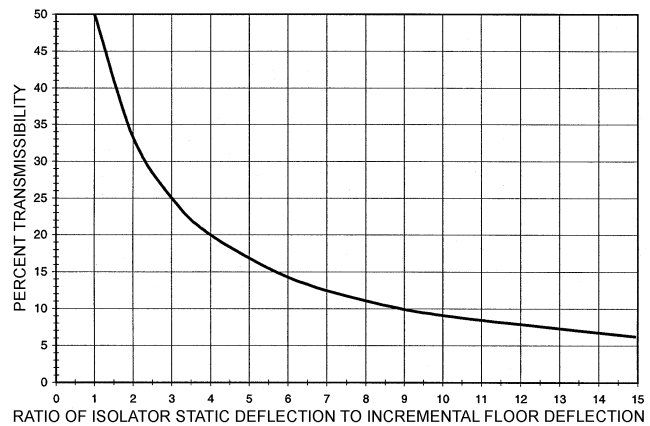
The single-degree-of-freedom model is valid only when the stiffness of the supporting structure is large with respect to the stiffness of the vibration isolator. This condition is usually satisfied for mechanical equipment in on-grade or basement locations. However, when heavy mechanical equipment is installed on a structural floor, and in particular on the roof of a building, the relative stiffness of the supporting system can no longer be ignored. Significantly “softer” vibration isolators are usually required than in the on-grade or basement case. The appropriate model for the design of vibration isolation in upper-floor locations is the two-degree-of-freedom model illustrated in [Figure 12](#).

The precise behavior of this system with respect to vibration isolation is difficult to determine. The objective is to minimize the motion of the supporting floor  $M_f$  in response to the exciting force  $F$ . This involves evaluating the interaction between two system natural frequencies and the frequency of the exciting force, which is mathematically complex. However, several engineering rules can simplify the calculations used to optimize the isolation system.



*Note: M and M<sub>f</sub> represent effective masses of vibrating equipment and supporting floor, respectively; k and k<sub>f</sub> are corresponding stiffness of isolator and floor system.*

**Fig. 12 Two-Degree-of-Freedom System**



**Fig. 13 Transmissibility of Two-Degree-of-Freedom System**  
Adapted from Plunkett (1958)

For example, the fraction of vibratory force transmitted across an isolator to the building structure (transmissibility) depends in part on the ratio between the isolator stiffness and that of the supporting floor at the point of loading. Because stiffness is inversely proportional to deflection under the applied load, this relationship can sometimes be expressed more conveniently as a ratio of deflections. To optimize isolation efficiency, the static deflection of the isolator, under the applied load, must be large with respect to the incremental static deflection of the floor that occurs due to the added equipment weight. Ideally, this ratio should be on the order of 10:1 to approach an isolation efficiency of about 90% (10% transmissibility). The relationship is illustrated in [Figure 13](#).

Note that if the static deflection of the vibration isolator is similar to the incremental deflection of the supporting floor under the added weight of the equipment, 50% or more of the vibratory force will transmit directly to the building structure. This situation is a common problem in the field where excessive vibration is attributable to upper floor or rooftop mechanical installations. Frequently, the floor stiffness has been neglected and the static deflection on the installed vibration isolators is inadequate because the selection was made on the basis of the single-degree-of-freedom model.

Problems of this nature can usually be avoided by asking the structural engineer to estimate the incremental static deflection of the floor due to the added weight of the equipment at the point of loading, before selecting a vibration isolator. Then, choose an isolator that will provide a static deflection of 8 to 10 times that of the estimated incremental floor deflection.

## VIBRATION MEASUREMENT BASICS

While the control of HVAC system sound and vibration are of equal importance, the measurement of vibration is not usually necessary for determining the sources or transmission paths of disturbing sound. Because the techniques and instrumentation used for vibration measurement and analysis are specialized, designers should consult other sources (e.g., Harris 1991) for thorough descriptions of vibration measurement and analysis methods.

The typical vibrations measured are periodic motions of a surface. This surface displacement oscillates with one or more frequencies produced by mechanical means (like gears), thermal means (like combustion), or fluid-dynamic means (like airflow through a duct or fan interactions with air). The displacement is generally inversely proportional to the frequency. In other words, if the displacements are high, the frequency is low. The frequencies of interest for most vibration measurements are between 5 Hz and 100 Hz.

A **transducer** can detect displacement, velocity, or acceleration of a surface and convert the motion to electrical signals. Displacement is the basic measure and good for low frequencies. Velocity is good for overall measurements, but requires large transducers. For most HVAC applications, the transducer of choice is an **accelerometer**, a device that detects acceleration. Readout may be as acceleration level in decibels, or acceleration with modifiers of peak, peak-to-peak, or rms.

The simplest measure is the overall signal as a function of time, be it acceleration, acceleration level, or another quantity. This is analogous to the unfiltered sound pressure level for sound. If a detailed frequency analysis is needed, there is a choice of filters similar to those available for sound measurements: octave band, 1/3 octave band or 1/12 octave band filters. In addition, narrow-band analyzers that use the fast Fourier transform (FFT) to analyze and filter a signal are available. While they are widely used, they should only be used by a specialist for accurate results.

The most important issues in vibration measurement include: (1) choosing a transducer with a frequency range appropriate to the measurement, (2) properly mounting the transducer to ensure that the frequency response claimed is achieved, and (3) not using hand-held probes for high frequencies where they are unreliable.

## STANDARDS

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