
7 Burner Noise

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7.1 INTRODUCTION

The sensation of sound is a thing *sui generis*, not comparable with any of our other sensations. No one can express the relation between a sound and a color or a smell. Directly or indirectly, all questions connected with this subject must come for decision to the ear, as the organ of hearing; and from it there can be no appeal. But we are not therefore to infer that all acoustical investigations are conducted with the unassisted ear. When once we have discovered the physical phenomena which constitute the foundation of sound, our explorations are in great measure transferred to another field lying within the dominion of the principles of Mechanics. Important laws are in this way arrived at, to which the sensations of the ear cannot but conform.

— The Theory of Sound
by Lord Rayleigh III

According to recent statistics, more than 20 million Americans are exposed to hazardous sound levels on a regular basis. There are approximately 28 million Americans who have some degree of hearing loss: about one third of these — more than 9 million — have been affected, at least in part, by exposure to excessive noise.¹

Combustion equipment and processes are some of the loudest known sources of noise. Ranging from the earliest jet engines, the pulse combustors used on the V-1 bombs (Buzz Bombs) during World War II, and the large rocket engines on the Saturn rockets, down to the humble propane torch, it is obvious that combustion processes create significant to dangerous noise levels. Industrial burners are in the middle of the size range of combustion equipment as a whole, but are a ubiquitous part of all types of industry and, as such, industrial personnel as well as unrelated neighbors experience significant exposure to the noise from industrial burners

Understanding how noise is produced and propagated in industrial burners and associated equipment has grown over the years and has provided the knowledge necessary to apply effective noise control to burners. This chapter attempts to inform the reader of the various factors and mechanisms involved in the production and propagation of burner noise, as well as the techniques and strategies to mitigate its impact on the environment.

7.2 BASIC DEFINITIONS OF SOUND

Webster's Dictionary defines sound as "that which is heard." However, in keeping with Lord Rayleigh's argument that sound follows principles of mechanics in addition to human hearing, the Noise Control Reference Handbook² gives a fuller definition as follows:

Sound is a vibrational disturbance, exciting hearing mechanisms transmitted in a predictable manner determined by the medium through which it propagates. To be audible the disturbance must fall within the frequency range of approximately 20 Hz to 20,000 Hz.

Nobel laureate Lord Rayleigh (John William Strutt, Third Baron of Rayleigh), considered by some to be the father of modern acoustics, outlined most of the fundamentals of acoustics in his historic 1877 text entitled *The Theory of Sound*. In this work, Lord Rayleigh clearly explains the wave propagation of sound.¹

Sound is propagated as waves in solids, liquids, and gases. Waves transmit energy without the translation of matter in the direction of energy flow. An example is the expanding ripples caused when a pebble is dropped in a still pond. The energy is conducted out and away from the source but the particles of water only move up and down in the same place, as can be seen from the up-and-down movement of a piece of wood or leaf floating on the surface.

Lord Rayleigh pointed out that if sound propagation involved gross motion of the air, rather than wave propagation, then a cricket that is heard at considerable distance would need to move a tremendous mass of air, all at once.

Wave motion can be uniquely identified by two of its properties: namely, frequency and amplitude. For sound, frequency defines the pitch or tone of the sound; and sound pressure, which is the amplitude, defines the loudness of the sound. Sound frequency can be measured in cycles per second (cps) and, as with electricity, the commonly used unit is Hertz (1 Hertz [Hz] = 1 cycle per second [cps]). Sound pressure can be quantified in pressure units such as kPa or psig, but is most commonly expressed in units of decibels.

It is important to distinguish between sound power and sound pressure. *Sound power* usually means the capacity or power level of a sound source. *Sound pressure*, on the other hand, is the incident energy reaching any point of interest, some distance from the sound source.⁴ The difference between sound power and sound pressure is easily understood using the analogy of a 100 W light bulb. The power of the source, in the case of the light bulb, is a constant 100 W; however, the incident light energy from the 100 W light bulb at any point of interest depends on the distance from the light source as well as atmospheric conditions. Most of the time, sound pressure is of interest because it is what affects us. However, estimates of the sound power of sources is required to model and calculate the impact of sound on the environment. Sound power can be expressed in terms of power units such as watts (W), but is also most commonly expressed in decibels (dB).

“Decibel,” a term coined in the 1960s by Bell Laboratories for a $\log_{10}$ scale, is used to compress the large range of power or pressure units required to span the various sounds that we deal with every day. The abbreviation for decibel is dB. Figure 7.1 shows how decibel (dB) relates to power

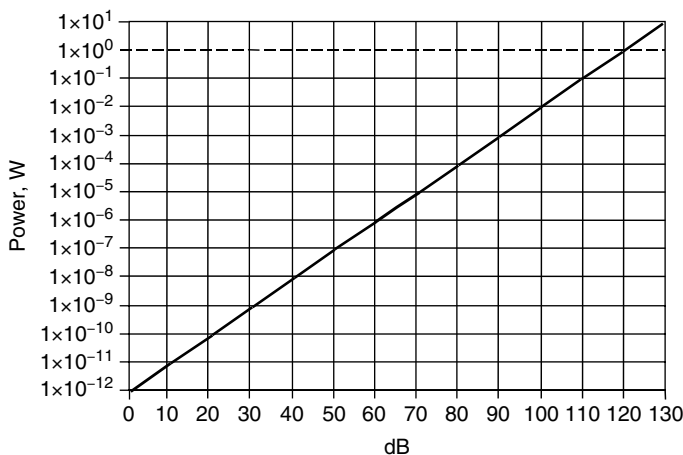


FIGURE 7.1 Relationship between decibels and watts.

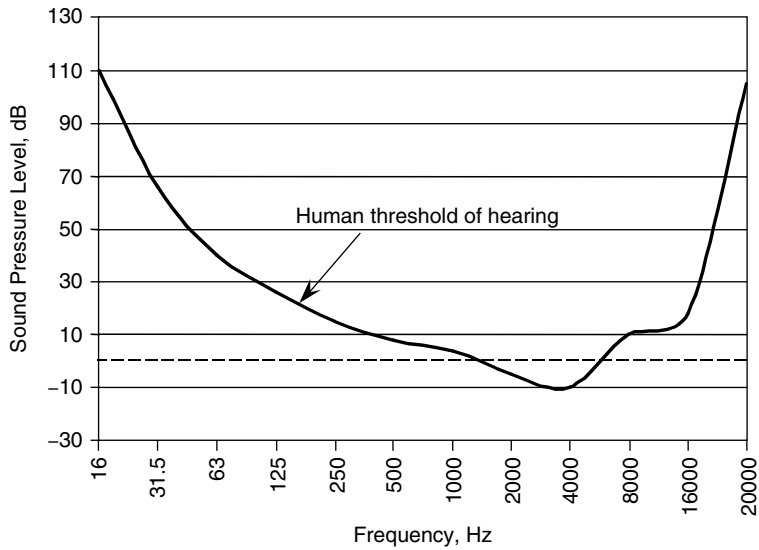


FIGURE 7.2 Hearing threshold in humans.

in watts (W). In this convention, 0 dB relates to a reference power of 1×10^{-12} W, and 1 W is equal to 120 dB.

Sound power level is the acoustic power radiated from a given sound source as related to the reference power level of 1×10^{-12} W, and can be expressed as:

$$L_w = 10 \log(W/10^{-12}) \text{ or } L_w = 10 \log W + 120 \quad (7.1)$$

where W = acoustic power in watts.

Sound pressure level in decibels (dB) is a ratio of the mean square of the actual pressure to the mean square of a reference pressure. The reference pressure, by convention, has been selected to be the threshold of hearing. That is, 0 dB has been chosen to be equal to the quietest audible sound for persons with excellent hearing (i.e., 20 μ Pa).

Human hearing sensitivity extends over a wide range of frequencies; namely, 20 to 20,000 Hz. The human ear is not equally sensitive over this entire range. Figure 7.2 shows the “Threshold of Hearing,” which represents the human ear’s sensitivity curve. The area above the curve is what can be heard. The large range of frequencies of the human hearing range can be divided into octave bands. An octave represents a doubling of frequency and, for the sake of more accurate measurement and analysis, the octave bands can be divided into one-third octave bands. The list of octave bands and the one-third octave bands is shown in Table 7.1. Any of these bands can be conveniently identified by its center frequency. The center frequency is the geometric mean of the range whose upper and lower limits are defined by the ISO and ANSI standards.

The advantages of making broad band analysis of sound using octave or one-third octave band filter sets are that less time is needed to obtain data and the instrumentation required to measure the data is less expensive. The main disadvantage is the loss of detailed information about the sound which is available from narrow band (FFT) analysis.

The upper and lower nominal octave band limits are arrived at by multiplying the center frequency by 1.414 and 0.707, respectively. One-third octave band center frequencies are given by

TABLE 7.1
Octave and One-Third Octave Bands

Band	Lower Band		Upper Band		Lower Band		Upper Band	
	Limit	Center	Limit	Limit	Center	Limit	Limit	Limit
12	11	16	22	14.1	16	17.8		
13				17.8	20	22.4		
14				22.4	25	28.2		
15	22	31.5	44	28.2	31.5	35.5		
16				35.5	40	44.7		
17				44.7	50	56.2		
18	44	63	88	56.2	63	70.8		
19				70.8	80	89.1		
20				89.1	100	112		
21	88	125	177	112	125	141		
22				141	160	178		
23				178	200	224		
24	177	250	355	224	250	282		
25				282	315	355		
26				355	400	447		
27	355	500	710	447	500	562		
28				562	630	708		
29				708	800	891		
30	710	1000	1420	891	1000	1122		
31				1122	1250	1413		
32				1413	1600	1778		
33	1420	2000	2840	1778	2000	2239		
34				2239	2500	2818		
35				2818	3150	3548		
36	2840	4000	5680	3548	4000	4467		
37				4467	5000	5623		
38				5623	6300	7079		
39	5680	8000	11,360	7079	8000	8913		
40				8913	10,000	11,220		
41				11,220	12,500	14,130		
42	11,360	16,000	22,720	14,130	16,000	17,780		
43				17,780	20,000	22,390		

$10^{n/10}$, where n is the index number (left column of Table 7.1) of the one-third octave band. As an example: for 125 Hz, the band index number is 21 and $10^{21/10} = 125.89$. Lower and upper frequencies are a function of $2^{-1/6}$ (0.89) or $2^{1/6}$ (1.12), respectively. As one can see, there is considerable approximation and truncation or round-up being used in handling these numbers. The values shown in the table are the nominal values that are most commonly used.

Sound meters have the capability to measure with equal sensitivity over the entire audible range. However, because humans do not hear with equal sensitivity at all frequencies, the sound meter’s measurement needs to be modified to quantify what really affects us. This can be done using correction curves. There are various correction curves: namely, A, B, C, and D (see Figure 7.3). The most common correction is the A-scale, which is approximately representative of the frequency response of the human ear, which is less sensitive to low-frequency than to high-frequency sound. A-scale decibel levels are expressed as “dBA.”

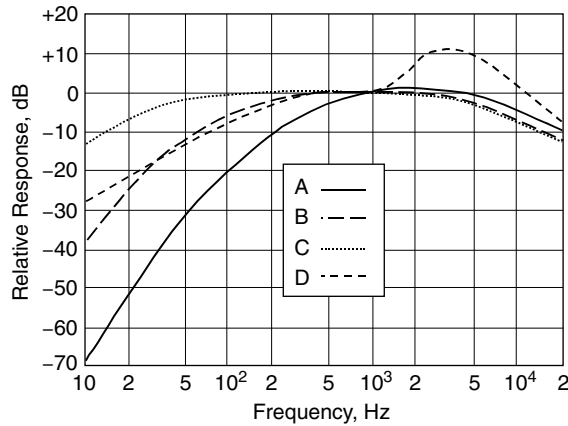


FIGURE 7.3 Weighting curves A, B, C, and D.

7.2.1 MEASUREMENT OF SOUND

Noise meters consist of a microphone, preamplifier, signal processor, output amplifier, and display. Figure 7.4 shows a typical noise meter used for measurements of burner noise. For noise analysis tasks, more sophisticated digital signal processing (DSP) equipped analyzers are used. The more sophisticated analyzers can perform Fast Fourier Transform (FFT) operations that aid accurate narrowband analysis. In general, spectrum analyzers allow the user to map the sound pressure level at different frequencies, that is, generate a curve of the sound over different frequencies. Typical measurements are made with the following resolutions:

1. Octave band measurements (one measurement per octave)
2. One-third octave band measurements (one measurement per one third of an octave)
3. Narrow band measurements (via FFT)

As the name suggests, a one-third octave band instrument makes three measurements in each octave, as opposed to the single measurement of the octave band instrument. A narrow band instrument, on the other hand, uses digital signal processing (DSP) to implement an FFT analysis and, in the current state of the art, FFT analysis allows a given frequency range to be sliced into as many as 128 intervals.

Figure 7.5 provides a comparison of the same sound spectrum as analyzed using three different band intervals: octave band, one-third octave, and narrow band. This comparison shows that the



FIGURE 7.4 Typical noise meter.

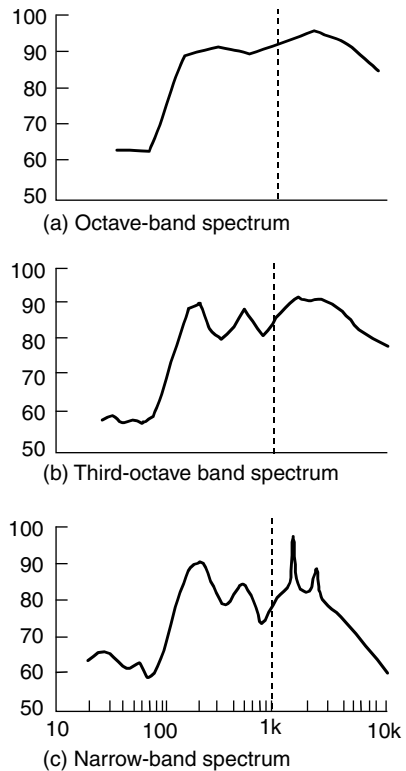


FIGURE 7.5 Same sound spectrum on three different band intervals.

additional resolution provided by narrower band methods is of vital importance. In this example, the level at 1 kHz, as recorded by the octave band instrument is 90 dB; on the one-third octave instrument, it is 85 dB; and on the narrow band instrument it is 70 dB. The lower resolution measurements produce higher measurements because the lack of resolution allows the nearby 1.8-kHz peak to influence the measurement at 1 kHz. Further, in considering implementing some noise control for this source, it is very valuable to know that it is the narrow peak at 1.8 kHz that is driving the maximum noise. This knowledge helps one to zero in on the hardware that may be causing the noise, such as an 1800-rpm motor or pump.

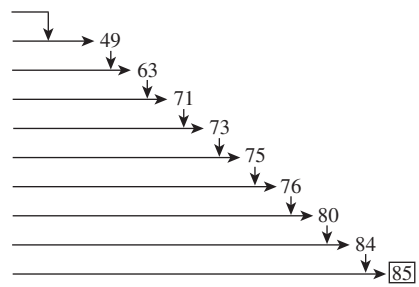
A detailed discussion of measurement issues is beyond scope of this chapter and the reader is urged to use some of the more comprehensive works in the list of references at the end of this chapter. The American Petroleum Institute has issued a recommended practice for measuring noise from fired process heaters.

7.2.2 OVERALL SOUND LEVEL AND HOW TO ADD dB VALUES

Most sounds are composites of several different levels at different frequencies. This is especially true of industrial burner noise. A typical burner noise curve is shown in Figure 7.6. As can be seen, there are significant peaks in two different frequency zones, both of which will contribute to the apparent intensity experienced by a person working in the vicinity of the burner. It is difficult to describe this sound without using either a diagram like the one shown, or a table listing various SPLs occurring in the different octave bands. The *overall sound pressure level*, a single number, has been devised to conveniently represent such composite sound curves. If a single number is to be used to represent the whole curve, then it should adequately represent the peaks in the curve, because the peaks have the most influence on the listener. Consequently, it is not practical to use the average of the various levels in the octave bands, because this number would be less than the levels at the peaks.

TABLE 7.2
Addition Rules

What is the Overall dBA Level?			
Frequency Hz	SPL dB	A-scale CF dB	SPL dBA
31.5	72	-39	33
63	75	-26	49
125	79	-16	63
250	79	-9	70
500	72	-3	69
1000	69	-0	69
2000	68	-1	69
4000	78	-1	79
8000	83	-1	82
16,000	80	-7	73



Notes: Overall sound level = 85 dBA.
Caution: Overall SPL does not = average SPL.

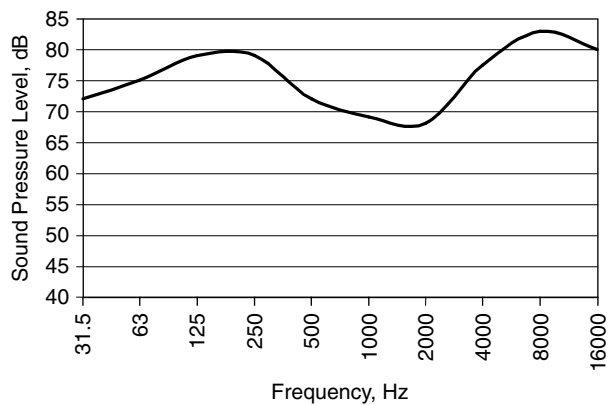


FIGURE 7.6 Typical burner noise curve.

The overall sound pressure level is calculated by adding the individual levels in the various octave bands. In columns 1 and 2 of [Table 7.2](#), the burner sound curve has been split into its component levels in each octave band. In column 3, the A-weighted correction has similarly been split and listed. Column 4 gives the A-scale corrected values for the sound curve by simply subtracting column 3 from column 2. The overall sound pressure level after A-scale correction can now be obtained by adding the values in column 4.

Decibel values cannot be added in a conventional manner due to the influence of the $\log_{10}$ scale. [Figure 7.7](#) shows a chart that can be used to add two dB values. When adding two numbers, first determine the difference between the two numbers. Locate the difference on the x-axis of the chart and determine the corresponding y-axis value and add it to the larger of the two numbers. For numbers that differ by 10 or more, the larger of the two numbers can just be used as the sum because of the decreasing significance in real power level (watts) caused by the $\log_{10}$ scale. It can also be noted that for adding two numbers of equal value, just 3 dB are added. What this translates to is that an increase of 3 dB requires a doubling in the power level.

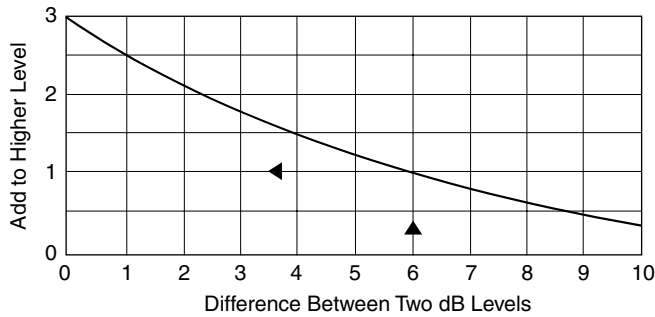


FIGURE 7.7 Graph for adding two dB values.

Alternately, the following formula can be used for dB addition:

$$L_{\text{total}} = 10 \log(\sum_{i=1 \text{ to } n} 10^{0.1L_i}) \quad (7.2)$$

where

L_{total} = overall sound pressure level

L_i = each individual level

N = number of levels to be added

And subtraction can be performed using:

$$L_{\text{diff}} = 10 \log(10^{0.1L_2} - 10^{0.1L_1}) \quad (7.3)$$

7.3 CHARACTERISTICS OF SOUND PROPAGATION

There is a great similarity between sound and light. Like light waves, sound waves can be reflected, refracted, diffracted, and diffused.⁵⁻⁷ The only difference between light and sound waves is that the latter cannot be polarized. The material nature of light allows it to be polarized. The following paragraphs present some basic characteristics of sound waves.⁷

7.3.1 ABSORPTION OF SOUND

Hard surfaces such as concrete and metal reflect almost all the sound energy that is incident upon them. However, soft fibrous surfaces, such as slabs of glass fiber or mineral fiber, are poor reflectors and absorb most of the sound energy, converting it into heat as a result of friction that occurs when the sound travels within the small air spaces between the fibers of the material.⁶⁻⁸ The absorption coefficient of a material is used to characterize the fraction of sound energy that is absorbed at the surface of the material. In the following paragraphs the absorption coefficient and the different methods of evaluating it will be discussed in detail. In addition, the factors that influence the absorption coefficient are presented.

7.3.1.1 Absorption Coefficient

The *absorption coefficient* is a material property and is defined as the fraction of incident sound energy absorbed at the surface of the material. The absorption coefficient strongly depends on frequency and varies with the angle of incidence at which the sound strikes the surface. In general, there are two types of absorption coefficients, depending on the direction of the incidence of the sound wave relative to the surface of the material.^{7,9} The normal incidence absorption coefficient is used to characterize the absorption of the sound energy that impinges on the surface in the

normal direction. The random incidence absorption coefficient is used to characterize the absorption of sound energy that impinges on the surface from various directions. Numerically, the random incidence absorption coefficient is larger than the normal incidence absorption coefficient for the same material.

7.3.1.2 Methods of Evaluating Sound Absorption

There are several methods that can be used to measure the absorption coefficient of different materials. Three of these methods are discussed below:⁷

7.3.1.2.1 Reverberation Chamber Method

In this method, a special chamber — a reverberation chamber — is used to perform the measurements. The reverberation chamber is a relatively large room with all surfaces — the walls, ceiling, and floor — being reflective surfaces. The reflective surfaces make the reverberation time of the room very long. The reverberation time is the time taken by the sound pressure level to decrease by 60 dB. After the reverberation time of just the empty room is measured, a piece of the material, for which the absorption coefficient is to be determined, is placed on the floor. Then the reverberation time for the room with the material is measured and the times with and without the specimen are compared. The shorter reverberation time of the room with the specimen is attributed to the absorption of sound by the material and size of the specimen. Comparing the reverberation times of the room with and without the specimen yields the number of absorption units the sample has added to the room. Then the absorption attributed to each square foot of the sample (i.e., the absorption coefficient) can be calculated. Usually, the absorption coefficient is reported for each of the following six frequencies: 125, 250, 500, 1000, 2000, and 4000 Hz.

7.3.1.2.2 Impedance Tube Method

A reverberant chamber is a relatively large room and requires a large piece of the material for the absorption coefficient measurement. An impedance tube, also known as standing-wave tube, is really handy and simple to use to determine the absorption coefficient of a material. It only takes a small piece of the material. The impedance tube is a circular tube with rigid walls and a rigid backing plate at one end. A loudspeaker is mounted at the open end. A slender tube for insertion of the measuring microphone passes through the center of the loudspeaker magnet. The sample is placed close to the backing plate end (either in contact with the plate or with some air gap between material and plate, depending on the intended use of the material). The loudspeaker is fired at different frequencies and for each frequency, the maxima and minima of the corresponding standing wave are determined using the microphone. The ratio of one of the maximum sound pressure to its adjacent minimum sound pressure is used to determine the absorption coefficient of the sample. It is worth noting that the absorption coefficient measured using the impedance tube is the normal incidence absorption coefficient.

7.3.1.2.3 Tone-Burst Method

Another simple and low-cost method for measuring the absorption coefficient at any incidence angle is the tone-burst method. This method allows anechoic chamber measurements to be made in ordinary rooms. Because it takes some time for the unwanted reflection from the walls to reach the measurement location, the tone-burst method utilizes short pulses of sound. A speaker and microphone are placed with some distance (X) between them, and the loudspeaker is fired with a short pulse. The sound pressure level from this short pulse is measured using the microphone. Then, a sample of the material is placed so that the path length of the reflected sound from the sample between the speaker and the microphone is the same as X . A barrier is used to prevent the direct sound path between the speaker and the microphone. Comparing the strength of the reflected pulse to the original direct pulse at distance X , the coefficient of absorption can be determined.

TABLE 7.3
Absorption Coefficients for Some Commonly Used Materials

Material	Octave Band Center Frequency (Hz)					
	125	250	500	1000	2000	4000
Fiber-glass or rock wool blanket						
25 mm thick (typical)	0.18	0.24	0.68	0.85	1.00	1.00
50 mm thick (typical)	0.25	0.83	1.00	1.00	1.00	1.00
Floors						
Wood platform with large space beneath	0.40	0.30	0.20	0.17	0.15	0.10
Concrete or terrazzo	0.01	0.01	0.01	0.02	0.02	0.02
Walls						
Polyurethane foam, 27 kg/m ³ 15-mm thick	0.08	0.22	0.55	0.70	0.85	0.75
Acoustic plaster, 10-mm thick, sprayed on solid wall	0.08	0.15	0.30	0.50	0.60	0.70
Glass, heavy plate	0.18	0.60	0.04	0.03	0.02	0.02
Water	0.01	0.01	0.01	0.015	0.02	0.03

7.3.1.3 Factors Affecting the Absorption Coefficient

There are many factors that influence the absorption coefficient of different materials.⁷⁻⁹ First, the method of mounting the test sample, whether the sample is in contact with the rigid floor of the reverberation room or there is an air gap between them that will affect the measured absorption coefficient. The air gap increases the absorption coefficient of the sample. As mentioned, the absorption coefficient is strongly dependent on frequency and usually is higher at high frequency. Incidence direction is another factor that affects the absorption coefficient. Furthermore, the thickness of the sample also affects the absorption coefficient, but its dependence varies with frequency. The dependence of the absorption coefficient on the density of the material is fairly low except for two extreme cases; namely, very high density and very low density. In both cases, the absorption coefficient is reduced. Table 7.3 gives the absorption coefficients for some commonly used materials.

7.3.2 REFLECTION OF SOUND

When a sound source is activated in an enclosed space, sound travels radially in all directions. As the sound waves encounter obstacles or rigid surfaces, such as walls, their direction of travel is changed (i.e., they are reflected).^{7,9}

7.3.2.1 Reflection from Flat Surfaces

Spherical sound waves that strike a rigid wall will be reflected and return toward the source (see Figure 7.8). As in the case of light on a mirror, the reflected wavefronts act as if they originated from a virtual image of the real sound source, on the other side of the reflecting surface. The image source is located at the same distance behind the wall as the real source is in front of the wall. This is a simple case; that is, a single reflection. In a rectangular enclosed space, there are six surfaces and the source has a virtual image with respect to each of the six surfaces, with the apparent effect of the images sending acoustic energy back to the receiver. In addition, reflected images of the virtual image sources exist and so on, resulting in a more complex situation. It is important to note that, in computing the total sound intensity at a given point, the contribution from all these images must be taken into consideration.⁷

Sound waves are reflected from objects that are large compared to the wavelength of the impinging sound. As a general rule, sound with frequencies of 300 to 400 Hz is best thought of in

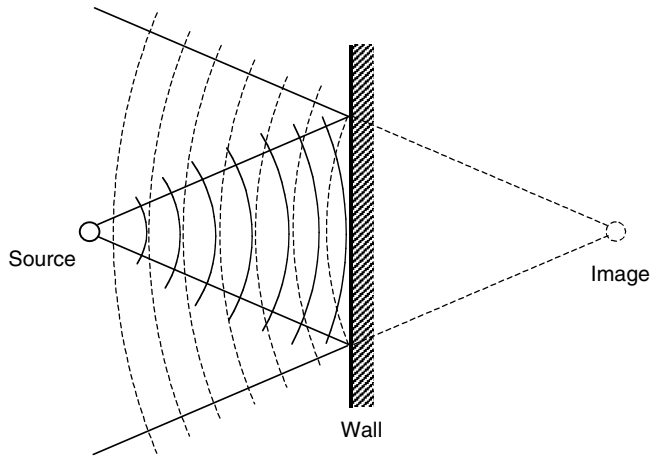


FIGURE 7.8 Reflection of sound from a point source from a flat surface. Incident waves are solid lines and reflected waves are dashed lines. The reflected waves appear to be from a virtual image source.

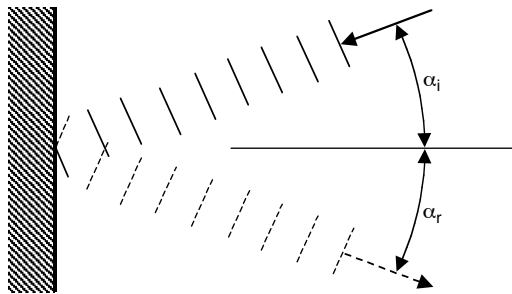


FIGURE 7.9 The angle of incidence α_i is equal to the angle of reflection, α_r .

terms of waves, and sound with frequencies above 300 to 400 Hz is best considered as traveling in rays. When a ray of sound undergoes many reflections as it bounces around an enclosed space with energy being lost at each reflection, the energy loss eventually results in the demise of that ray. Sound reflection follows the same rule as light reflection: the angle of incidence is equal to the angle of reflection^{5,7} (see Figure 7.9).

7.3.2.1.1 Doubling of Pressure at Reflection

Sound pressure on a surface normal to the incident waves is equal to the energy density of the radiation in front of that surface. If the surface is a perfect absorber, the pressure equals the energy density of the incident radiation. On the other hand, if the surface is a perfect reflector, the pressure equals the energy density of both the incident and reflected radiation. Thus, the pressure at the face of a perfectly reflecting surface is twice that of a perfectly absorbing surface. Pressure doubling has more significance in standing waves.

7.3.2.1.2 Reflecting from Convex Surfaces

Sound waves are emitted from a point source as spherical wavefronts and tend to become plane waves when further away from the source.⁷ Reflection of plane wavefronts of sound from a solid convex surface tends to scatter the sound energy in many directions as shown in Figure 7.10 and this effectively diffuses the impinging sound.

FIGURE 7.10 Plane sound waves impinging on a convex surface tend to be dispersed through a wide angle.

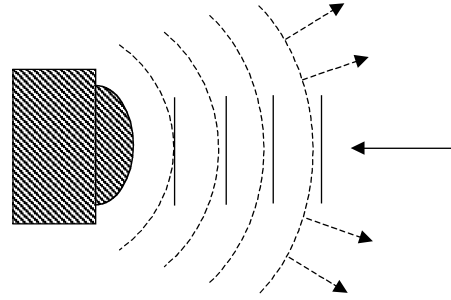


FIGURE 7.11 Plane sound waves impinging on a concave surface tend to be focused if the size of the irregularity is large compared to the wavelength of sound.

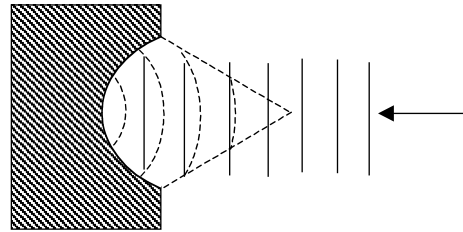
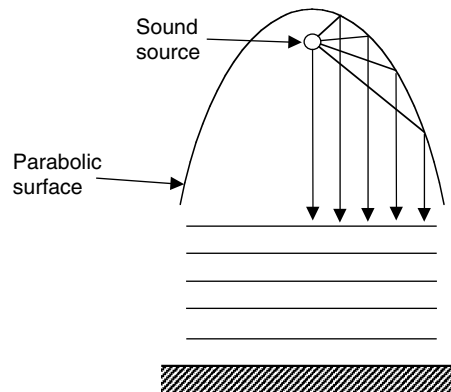


FIGURE 7.12 A parabolic reflector surface can produce plane parallel wavefronts when the sound source is placed at the focal point.



7.3.2.1.3 Reflections from Concave Surfaces

Plane waves of sound when impinging on a concave surface tend to get focused to a point as shown in Figure 7.11. The precision of focusing the sound wave depends on the shape of the concave surface. Spherical surfaces are good examples of concave reflectors. Sometimes, such a reflector is used with a microphone placed at the focal point, in order to make the microphone directional. For a concave surface to be an effective focusing reflector, the size of the reflector has to be larger than the wavelength of the sound to be focused.

7.3.2.1.4 Reflections from Parabolic Surfaces

The parabola has the characteristic of focusing the sound precisely to a point.⁷ The shape of the parabolic surface is generated based on a simple relationship, namely, $y = x^2$. A deep parabolic surface provides much better directional properties than a shallow one. Plane waves striking a parabolic surface would be focused to the focal point as shown in Figure 7.12. Conversely, sound emitted radially from a point source located at the focal point of a parabolic reflector generates plane wavefronts. Lighthouses and searchlights use parabolic mirrors to achieve a straight, low-divergence beam.

FIGURE 7.13 Sound waves reflected by a 90° corner will always return back in the same direction as the incident wave.

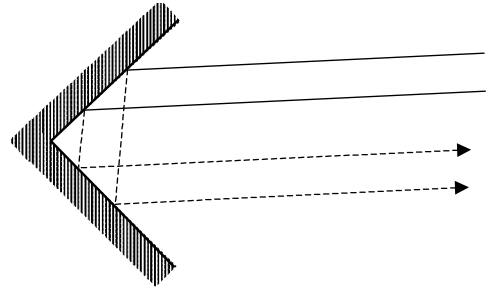


FIGURE 7.14 Large aperture has a very little effect on the plane wavefront that passes through it. Small aperture will turn the plane wavefront that penetrates the hole into a hemispherical wavefront because of the closeness of the new point sources.

7.3.2.1.5 Standing Waves

An important artifact that is a direct result of reflection is the standing wave. When a sound source is located between two parallel walls, sound waves radiated toward the two walls are reflected back. If the distance between the two walls is equal to an integer multiple of half the wavelength of the sound wave, the two waves that travel to the right and to the left will create a standing wave.⁸ This standing wave is stationary, and only sound waves that satisfy the resonance condition (integer multiple of half the wavelength) will establish a standing wave.

7.3.2.1.6 Reflections from Corners

One of the interesting consequences of sound reflection is the reflection from corners. Examine the schematic diagram in [Figure 7.13](#). It can be seen that a sound wave radiated from a source toward a corner will be reflected back to the source. For a corner with 90° , the sound wave is reflected twice and always returns in the direction of the source.⁷

7.3.3 DIFFRACTION OF SOUND

Sound waves travel in straight paths. However, when sound waves encounter an obstruction with certain geometric attributes, they change direction.^{5,7,9} The process by which sound waves change direction is called *diffraction*. The shorter the wavelength, the less is the effect of diffraction. Diffraction can be better explained using Huygens principle, which can be stated as follows: every point on the wavefront of sound that has passed an obstacle or a diffracting edge is considered to be a point source radiating sound energy back into the shadow zone (behind the obstacle).

7.3.3.1 Diffraction of Sound Waves by Large and Small Apertures

When a plane wavefront strikes a wall with an aperture, some of the energy is reflected back by the wall and some of the energy goes through the aperture. If the size of the aperture is larger than the wavelength of the sound, then the wavefront will remain a plane wave with little disturbance (see [Figure 7.14](#)). However, if the aperture size is smaller than the wavelength of the sound, the wavefront of the sound that passes through the aperture will take the form of a hemisphere. The behavior

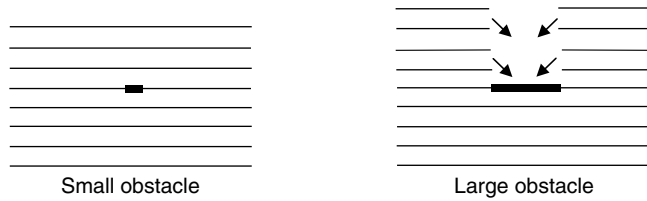


FIGURE 7.15 The small obstacle (relative to the wavelength) has very small effect on the wavefront that passes by the obstacle. The large obstacle (relative to the wavelength) has a definite effect in casting a shadow behind the obstacle.

of the two apertures can be explained with the help of the Huygens principle.⁷ In the case of the large aperture, the new virtual source points on each side of the wavefront are far apart and thus they have little effect on the wavefront; but in case of the small aperture, the new virtual source points are so close to each other that they make the wavefront hemispherical.

7.3.3.2 Diffraction of Sound Waves by Obstacles

When a plane wavefront encounters an obstacle that is small relative to the wavelength of the sound, the wave passes the obstacle with a very small shadow behind the obstacle (see Figure 7.15). In the case of an obstacle that is large with respect to the wavelength of the sound, the obstacle will lead to the generation of a significant acoustic shadow behind the obstacle see Figure 7.15). Again, consulting Huygens principle,⁷ in the case of a small obstacle, the new virtual sources are close and their combined radiation behind the obstacle maintains the plane wavefront. However, in case of the large obstacle, the new virtual sources are far from each other and their radiation behind the obstacle cannot retain the plane wavefront and hence a shadow is cast.

7.3.3.3 Diffraction of Sound Waves by a Slit

A classical experiment conducted with a narrow slit demonstrated that when sound is incident on a narrow slit, the width of the sound beam behind the slit varies inversely with the width of the slit.⁸ Thus, a narrower slit yields correspondingly more diffraction and results in a more divergent sound beam behind the slit, in keeping with the earlier discussion about the formation of a hemispherical wavefront behind a small aperture.

7.3.3.4 Diffraction of Sound by an Acoustic Lens

The acoustic lens consists of a round plate with a set of concentric annular slits.^{5,7} The radii of these slits are such that the paths from the focal point to each slot differ by multiples of $\lambda/2$, where λ is the wavelength of the sound wave (see Figure 7.16). Because these path lengths differ by $\lambda/2$, it means that all the waves will arrive at the focal point at the same time (i.e., in phase) and hence intensify the sound at the focal point.

7.3.4 REFRACTION OF SOUND

Refraction of sound is the mechanism by which the sound wave changes its direction when it propagates from one medium to another medium, when the propagation velocity of sound in the two media is different.⁷ There is a common confusion between diffraction and refraction of sound. Although the similarity of the words may lead to the confusion, the real reason stems from the perceived difficulty in understanding the difference in the two phenomena, because both phenomena

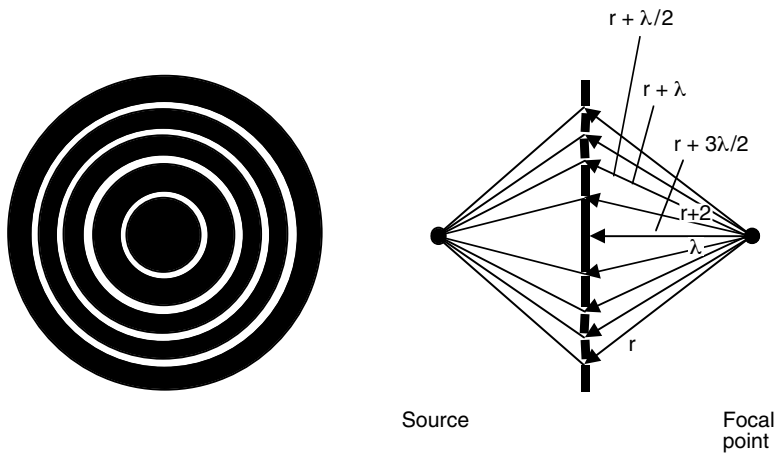


FIGURE 7.16 The slits of the acoustic lens are arranged such that the different path lengths differ by multiples of a half wavelength. Thus, the diffracted sound from all the slits arrive at the focal point in phase.

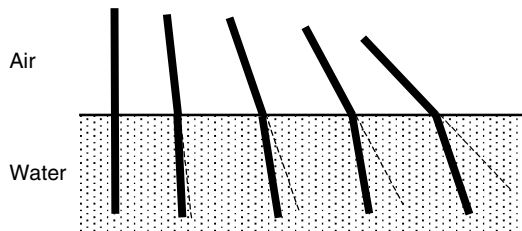


FIGURE 7.17 An immersed stick under water surface demonstrates refraction of light waves. Sound waves are also refracted by changes in the speed of sound in different materials.

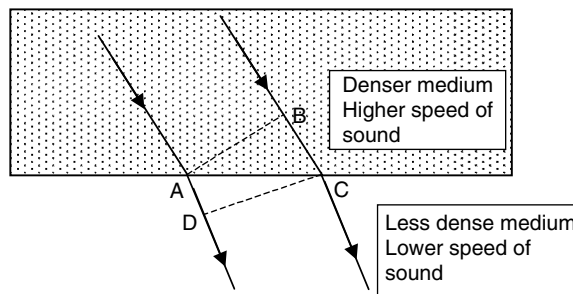


FIGURE 7.18 Sound waves traveling from denser medium.

cause change in the direction of sound waves. A good illustration of refraction is the common observation of the apparent bending of a stick as a part of it is immersed in water (see [Figure 7.17](#)). Of course, this is an illustration of the refraction of light waves.

Sound waves traveling from a denser medium having a certain speed of sound into a less dense medium having a lower speed of sound will result in change of direction of the sound wave; that is, the wavefront AB is not parallel to wavefront DC because of refraction (see [Figure 7.18](#)).

Sound waves exhibit refractive behavior when propagating from a dense solid medium to a less dense medium. The main reason for the refraction is the difference in the speed of sound in

TABLE 7.4
Speed of Sound in Different Media

Medium	Speed of Sound	
	(ft/sec)	(m/sec)
Air	1,130	344
Sea water	4,900	1,500
Wood, fir	12,500	3,800
Steel bar	16,600	5,050
Gypsum board	22,300	6,800

the two media.^{5,7} The speed of sound in the dense solid medium is greater than that in a less dense medium (see Table 7.4) for speed of sound in different media. As illustrated in Figure 7.18, one ray arrives at the boundary between the two media at A while the other ray still has some distance to go. During the time it takes the second ray to travel from B to C, the first ray travels at a slower speed from A to D, a shorter distance, in the new medium. The wavefronts AB and DC represent two different instants in time. The two wavefronts are no longer parallel because the sound has refracted at the interface between two media that have different speeds of sound.

7.3.5 DIFFUSION OF SOUND

When the sound field inside an enclosure meets certain criteria, the sound field is said to be a diffuse field. Physically, the diffuse sound field is a superposition of an infinite number of plane sound waves in which the local sound intensity does not depend on the direction. In a perfectly diffuse sound field, the pressure level is independent of the location. There are six factors that identify whether or not the sound field inside an enclosure is a diffuse sound field.⁷ The first is that the irregularities in the frequency spectrum and in the spatial distribution of the steady-state measurements should be negligible. Second, the beats in the decay characteristics must be negligible. Third, in the decay characteristics, the decay rates must be perfectly exponential; that is, they must be straight lines on a logarithmic scale. Fourth, reverberation time should be the same at all positions in the enclosure. Fifth, the character of the decay must be the same for all frequencies. Sixth and finally, the character of the decay will be independent of the directional characteristics of the microphone.

7.3.5.1 Steady-State Measurements

The frequency response of an enclosure, such as a process furnace, can be obtained by applying a variable frequency signal by means of a loudspeaker and measuring the response of the enclosure using a microphone. Also, the response should be measured at different locations in the enclosure. The typical enclosure frequency response will show some narrow and some wide peaks. The narrow peak is evidence of a single mode, while the wide peak is due to the combined effects of several adjacent modes. These deviations from the flat response of a perfectly diffuse sound field, are evidence of a sound field that is not perfectly diffuse. Also, variations in the enclosure response from one location to another are also evidence of a nondiffuse sound field.

7.3.5.1.1 Decay Beats

The nonsmooth decay response (decay response is explained in the next paragraph) of an enclosure is evidence of nonperfect diffuse sound field. Low frequencies usually exhibit more beats than high frequencies, the reason being that, at low frequencies, the number of resonance modes of enclosure is fewer than at high frequencies. Therefore, the decay at high frequencies is much smoother, resulting from averaging over many resonance modes of the enclosure.

7.3.5.1.2 Exponential Decay

Exponential decay is the characteristic of the impulse response of the enclosure. The impulse response is similar to the steady-state response except that the exciting source is an impulse sound source (a sound source that has enough power throughout the spectrum and is able to deliver a substantial amount of energy in a very short time). In the impulse response test, the impulse source is fired and the enclosure response is measured at different locations for a period of time that ends when the sound pressure level reaches the background level (i.e., all the sound energy from the impulse source has been dissipated by the enclosure). The impulse response of the enclosure is characterized by decay rate or reverberation time. The decay rate is the slope of the plot of sound pressure level vs. time, which is a straight line when the decay is exponential. The reverberation time is the time taken by the sound pressure level to reduce by 60 dB. The decay rate and the reverberation time of an enclosure are related to each other. When the impulse response, sound pressure level vs. time, deviates from being a line, this is evidence of a nonperfect diffuse field. One deviation is known as dual exponential decay, in which the impulse response is characterized by two slopes: a fast decay followed by a slow decay. The slower decay could be due to some modes that encounter low absorption. Another deviation from exponential decay results from the presence of acoustically coupled spaces and is characterized by a concave upward response curve.

7.3.6 SPATIAL UNIFORMITY OF REVERBERATION TIME

The reverberation time of an enclosure should be obtained at different locations in the enclosure. The changes in the reverberation time from one location to another in the enclosure is a sign of deviation from a perfectly diffuse sound field.

7.3.7 DECAY SHAPES

In a perfectly diffuse sound field, the decay character should be the same for all frequencies and the response should be smooth.

7.3.8 MICROPHONE DIRECTIVITY

In a totally homogeneous sound field (perfectly diffuse sound field), a highly directional microphone should register constant signal values irrespective of the direction in which it is pointed.

7.4 SOURCES OF NOISE IN BURNERS

There are four major mechanisms of noise production in burner equipment. They fall into two major categories: high-frequency and low-frequency sources:

1. Low-frequency noise sources:
 - Combustion roar
 - Fan noise
2. High-frequency noise sources:
 - Gas jet noise
 - Piping and valve noise

7.4.1 LOW FREQUENCY: COMBUSTION ROAR

To understand combustion roar, one must consider the mixing process taking place between the fuel and the oxidant on a very small scale. It is known that a well-blended mixture of fuel and oxidant will combust very rapidly if the mixture is within the flammability limits of that fuel.

However, a raw gas fuel stream that depends on turbulence and momentum to mix in the oxidant tends to create a slower combustion process due to the time delay required for mixing. In either case, when regions in the mixing process achieve a flammable mixture and encounter a source of ignition, combustion takes place rapidly.

The speed at which a fuel and oxidant mixture burns depends on the mixture fraction. For example, some hydrocarbons burn most rapidly when the fuel–oxidant mixture is near stoichiometry. The mixture proportion at which the fuel–oxidant mixture burns most rapidly has the maximum flame propagation speed. When combustion occurs at the maximum flame speed, more of the energy release is converted into noise. The rapidly expanding gases in the flame create pressure waves. The rapidly expanding gas pockets behave as acoustic monopoles. That is, they radiate sound very effectively omnidirectionally out from the source. The noise coming from each small region of rapidly combusting mixture adds up to create what we call *combustion roar*. So, combustion roar is largely a function of how rapidly the fuel is being burned. In addition, in the context of burner equipment, usually the larger the fuel flow-rate, the more turbulent the flow in the combustion process. Because turbulence directly influences the mixing rate, high-turbulence processes also produce more combustion roar. Thus, it is more accurate to state that the level of combustion roar generated from a combustion process is a function of the amount of fuel being burned and how rapidly it is burned.

High levels of turbulence in a flame are usually desirable because it helps to reduce the flame length and the tendency to produce smoke. Unfortunately, it increases the combustion roar. The signature of low-frequency combustion roar noise typically consists of a broadband spectrum with a single peak.

A common method for quantifying the *sound pressure level* (SPL) emitted from a burner is to relate the energy released from the combustion of the fuel stream (chemical energy) to the amount of that energy that is converted to noise. The ratio of noise energy released to chemical energy released from the combustion is called the *thermo-acoustic efficiency* (TAE). The TAE typically varies between 1×10^{-9} and 3×10^{-6} . The TAE value largely depends on the turbulent mixing of the fuel with combustion air and is usually determined experimentally.

A burner flame that is highly turbulent may have a TAE on the order of 1×10^{-6} . However, a flame with low levels of turbulence, such as a cigarette lighter, may have a TAE on the order of 1×10^{-9} . For every order of magnitude that the TAE changes, the sound pressure level will change by 10 dB.

Refer back to [Figure 7.6](#) for a plot showing a typical noise spectrum emitted from a burner operating under normal conditions in a furnace. Notice that the noise spectrum has two peak frequencies associated with it. The high-frequency noise contribution is from the fuel gas jets, while the low-frequency contribution is from the combustion roar. Burner combustion roar is associated with a smooth broadband spectrum having relatively low conversion efficiency from chemical energy to noise: in the range of 1×10^{-9} to 1×10^{-6} .

The combustion roar associated with burners can vary in the 200 to 500 Hz range. Burner combustion roar can have a noise spectrum shape and amplitude that can vary with many factors. These factors include the internal shape of the furnace; the design of the burner muffler, plenum, and tile; the acoustic properties of the furnace lining; the transmission of the noise into the fuel supply piping; and the transmissive and reflective characteristics of the furnace walls and stack.

7.4.2 LOW FREQUENCY: FAN NOISE

Fan noise is an issue that impacts burner noise performance in forced-draft systems. The noise emitted from industrial fans typically consists of two noise components: broadband and discrete tones. Vortex shedding from the moving blades and the interaction of the turbulence with the stationary parts of the fan create the broadband noise. This broadband noise is of the dipole type, meaning that the noise is directional. On the other hand, discrete tones are created by the periodic interactions of the rotating blades and nearby upstream and downstream surfaces. Discrete tonal

noise is usually the loudest at the frequency at which a blade passes a given point. The tonal frequency is easily calculated by multiplying the number of blades times the impeller rotation speed in revolutions per second. That is:

$$F = N \times \text{rpm}/60 \quad (7.4)$$

where F is the tonal frequency and N is the number of blades in the impeller.

The broadband and discrete tonal noise emitted from fans can radiate from both the suction and discharge side of a fan and through the fan casing. The noise can radiate downstream through the ducting and discharge into the environment at an outlet, such as a burner. Fan and duct systems should provide means to control this noise if residential areas are located nearby. Installation of mufflers and silencers on the suction and discharge sides of the fan, as well as wrapping the casing and the ducts are common methods of reducing fan noise.

7.4.3 HIGH FREQUENCY: GAS JET NOISE

Gas jet noise is very common in combustion equipment and, in many cases, it can be the dominant source of noise from a burner. The noise that is created when a high-speed gas jet exits into a relatively static ambient gas usually consists of two principal components: gas jet mixing noise and shock-associated noise.

7.4.3.1 Gas Jet Mixing Noise

Studies have shown that a high-speed gas jet, exiting a nozzle, will develop a large-scale orderly pattern as shown in [Figure 7.19](#). This orderly structure is known as the “global instability” or “preferred mode” of the jet. The presence of both the small-scale turbulent eddies within the jet and the large-scale structure is responsible for the gas jet mixing noise.

The source of gas jet mixing noise begins near the nozzle exit and extends several nozzle diameters downstream. Near the nozzle exit, the scale of the turbulent eddies is small and predominantly responsible for the high-frequency component of the jet mixing noise. The lower frequencies are generated further downstream of the nozzle exit where the large-scale orderly pattern of the gas jet exists.

Gas jet mixing creates broadband noise over a range of frequencies. The frequency at which the spectrum peaks depends on several factors, such as the diameter of the nozzle, the Mach number

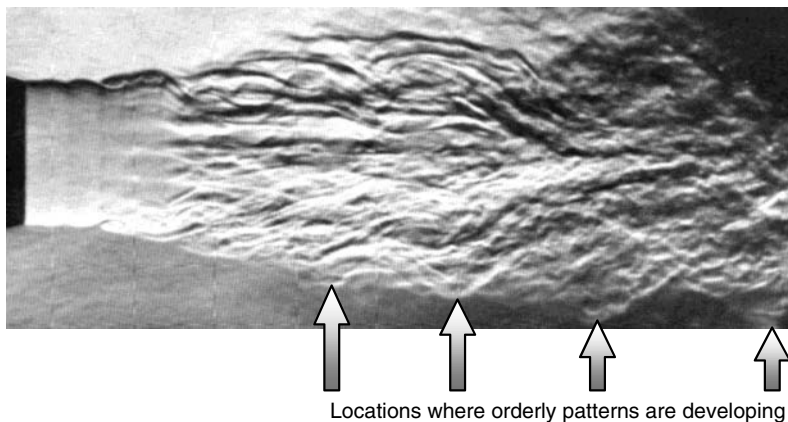


FIGURE 7.19 Development of orderly wave patterns.

of the gas jet, and the angle of observer's position relative to the exit plane of the jet and temperature ratio of the fully expanded jet to the ambient gas.

The maximum overall SPL of gas jet mixing noise occurs at an angle between approximately 15 to 30° relative to the centerline of the gas jet velocity vector. As one moves in either direction from this angle, the noise level can drop off significantly. For example, the overall SPL created by gas jet mixing can be reduced as much as 25 dB when one moves from an angle of maximum noise level to an angle directly behind the nozzle (180° away from the gas jet velocity vector's direction).

7.4.3.2 Shock-Associated Noise

When a burner operates above a certain fuel pressure, a marked change occurs in the structure of the gas jet. Above a certain pressure, called the *critical pressure*, the gas jet develops a structure of shock waves downstream of the nozzle. The critical pressure of a gas jet typically occurs at a pressure of 12 to 15 psig (0.8 to 1 barg), depending on the gas composition and temperature. The structure of shock waves, known as shock cells, consists of compression and expansion waves that repeatedly compress and expand the gas as it moves downstream. Using Schlieren photography, several investigators have seen as many as seven shock cells downstream of a nozzle. These shock cells are responsible for creating two additional components of gas jet noise: screech tones and broadband shock-associated noise.

Screech tones are distinct narrow-band frequency sounds that can be described as a “whistle” or “screech.” The literature reports that these tones are emitted from the fourth and fifth shock cell downstream of the nozzle exit, as shown in Figure 7.20.¹¹ The sound waves from these shock cells propagate upstream where they interact with the shear layer at the nozzle exit. This interaction then creates oscillating instability waves within the gas jet. When these instability waves propagate downstream, they interfere with the fourth and fifth shock cell, causing them to emit the screech tones. Unlike gas jet mixing noise, screech tone noise is not highly directional.

Broadband shock-associated noise occurs when the turbulent eddies within the gas jet pass through shock waves. The shock waves appear to suddenly distort the turbulent eddies, which creates a noise that can range over several octave bands. The broadband shock-associated noise typically peaks at a higher frequency than the screech tone peak frequency.

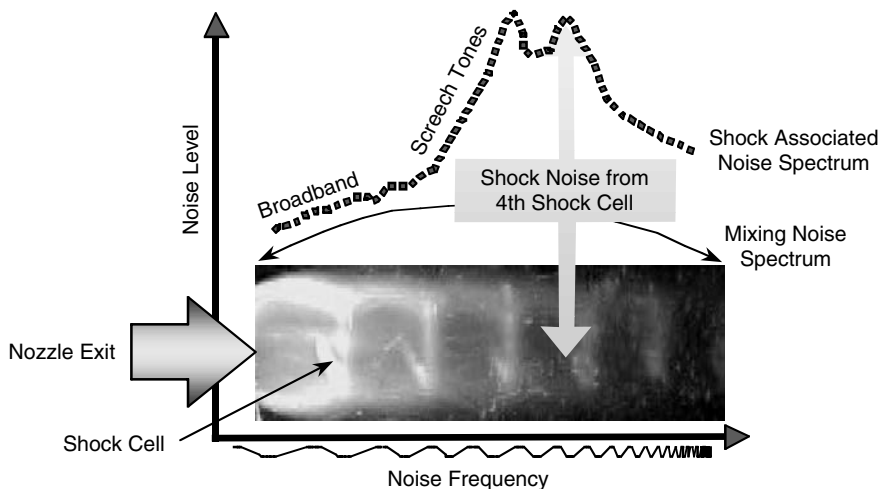


FIGURE 7.20 Location of screech tone emissions.

7.4.4 HIGH FREQUENCY: PIPING AND VALVE NOISE

When a gas flowing steadily in a pipe encounters a valve, a change in the flow pattern and pressure will occur that can create turbulence and shock waves downstream of the valve. Typically, when valves are partially closed, creating a reduction in flow area, the small flow passage behaves much like an orifice and produces jet noise. As discussed, turbulence and shock waves create mixing noise and shock-associated noise. This noise can radiate downstream through the pipe and exhaust into the environment at an outlet and/or radiate through the pipe wall into the space near the valve.

Usually, butterfly valves and ball valves are noisier than globe valves. Butterfly valves and ball valves typically have a smaller vena contracta than a globe valve operating at the same pressure drop, which results in higher levels of mixing and shock-associated noise. As a general guideline, when the pressure ratio across a valve is less than approximately 3, the mixing noise and shock-associated noise are within about the same order of magnitude.¹² With pressure ratios greater than 3, shock noise usually dominates over the mixing noise. There are several methods used for reducing the noise emitted from a valve. These include sound-absorptive wrapping of the pipes and valve casings and the installation of silencers between a valve and the connected pipes.

It should be noted that, just as with the sounding board of a piano or the body of a violin, noise radiation benefits from having a lot of surface area in contact with the source, as is the case with piping. For any sound source of a given power level, the availability of a large radiating surface area will cause the noise to sound louder than if the radiating surface were not available.

7.5 SOUND INSIDE ENCLOSURES

Sound in enclosed space is strongly affected by the reflection from the walls. A sound source in an enclosed space will generate, in general, two kinds of sound fields.⁷⁻¹⁰ The first is the direct sound field, which will dominate the immediate space around the source. The second field is the reverberant field. This field results from the multiple reflection of sound on the walls. It is important to note that the direct sound from the source, once it reflects off a wall, now belongs to the reverberation sound field. One can see that the overall sound pressure level in the cavity is a function of both the direct and reverberant sound fields. Consequently, it is difficult to determine the sound energy distribution and variation with frequency, with precision. Fortunately, there are some approximation and averaging procedures that are accepted and can be used to quantify the sound energy distribution in such sound fields. First, the sound in a reverberant field is divided into two ranges of frequency: low and high frequency.¹⁰ The ratio of the characteristic dimension of the enclosed space to the wavelength of the sound is used to mark, loosely, these two ranges. Usually, a ratio of less than 10 identifies the low-frequency range and a ratio greater than 10 marks the high-frequency range.

7.5.1 LOW-FREQUENCY RANGE

The sound field inside an enclosure in the low-frequency range is mainly governed by the standing waves that have specific characteristic frequencies.⁹ The standing waves occur because the sound travels back and forth in the enclosure due to multiple reflections on the internal surfaces and travel along the same path in opposite directions. At certain frequencies and certain locations, when the waves traveling in opposing directions reach these locations out-of-phase, pressure cancellation occurs and results in a pressure minimum called a *pressure node*. Similarly, when the opposing waves reach these locations in-phase, pressure augmentation occurs and results in a pressure maximum called a *pressure anti-node*. As one would expect, the number of resonance frequencies within a specified frequency band in the low-frequency range is relatively small. Thus, at low frequencies, the response of an enclosure as a function of the frequency and location will be irregular and, in fact, will be characterized by pressure nodes and anti-nodes.

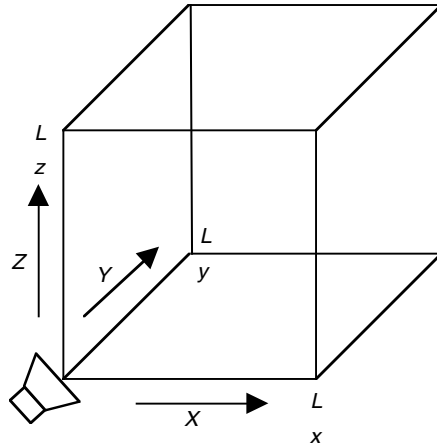


FIGURE 7.21 A schematic diagram for a rectangular enclosure.

Thus far, in the low-frequency range, the dominant response of an enclosure is determined completely by modal response of the enclosed space. It is emphasized that this modal response is by no means unique to rectangular or even other regular-shaped enclosures. In fact, modal response characterizes enclosures of all shapes. It should be understood that irregular or odd numbers of walls will not prevent resonance characterized by nodes and anti-nodes in an enclosure constructed with reasonably reflective inner surfaces.⁹ Also, this peculiar design will not result in a more uniform distribution of the resonance frequencies compared to a regular design of comparable dimensions. The only difference between the regular and irregular design of an enclosure regarding the modal response is the degree of difficulty in computing the frequencies and distributions of these modes. Next, the resonance frequencies of a rectangular enclosure (see [Figure 7.21](#)) can be obtained using the following expression:

$$F_n = \frac{c}{2} \sqrt{\left[\frac{n_x}{L_x} \right]^2 + \left[\frac{n_y}{L_y} \right]^2 + \left[\frac{n_z}{L_z} \right]^2} \quad (7.5)$$

where n_x , n_y , and n_z are the particular mode numbers and these numbers can take any positive integer number, including the zero. The subscript n on the frequency is a reference number for each mode, which depends on the mode numbers, and when c is in m/s and L in m the frequency is in hertz (Hz). Spatial and temporal distribution of these normal modes is governed by:

$$p'(x, y, z, t) = p'_o \cos \left[\frac{\pi n_x x}{L_x} \right] \cos \left[\frac{\pi n_y y}{L_y} \right] \cos \left[\frac{\pi n_z z}{L_z} \right] e^{j2\pi F_n t} \quad (7.6)$$

There are three types of normal modes of resonance for an enclosure. These types are categorized according to the corresponding combination of mode numbers, n_x , n_y , n_z , as follows:

1. Axial modes are characterized by only one mode number is nonzero.
2. Tangential modes are characterized by only one mode number is zero.
3. Oblique modes are characterized by no mode number is nonzero.

Consulting Equations (7.5) and (7.6), one can see that axial mode distribution varies in only one direction, and its frequency depends only on the dimension of the rectangular enclosure in that direction, while the tangential mode distribution varies in only two directions, and its frequency depends only on the dimensions of the enclosure in those directions. However, oblique mode varies in all directions, and its frequency depends on all dimensions of the rectangular enclosure. Also, Equation 7.5 shows that as the modes number increases (i.e., in the high-frequency range), the frequencies of the normal modes become much closer to each other than those of the small mode numbers (i.e., in the low-frequency range). That is due to the nature of the square root function in Equation 7.5.

7.5.2 HIGH-FREQUENCY RANGE

In an enclosed space in the high-frequency range, the reverberant sound field can be considered as plane waves propagate in all directions with equal probability.⁸⁻¹⁰ If the mean square amplitude of the acoustic pressure on average is the same in all directions, then the reverberant sound field inside the enclosure is said to be diffuse. This condition for a reverberant sound field to be considered a diffuse sound field can only be achieved in the high-frequency range. That is because in the high-frequency range, the resonance modes are so large that the reverberant sound field approximates uniformity throughout the enclosure. The concept of the diffuse field implies that the net power flow at any given point is negligibly small and that the flow of power is the same in all directions.

7.5.2.1 Effective Intensity in a Diffuse Sound Field

Because the power flow at a given location in a diffuse field is essentially equal in all directions, the intensity at any point in that field is zero. However, an effective intensity in a specified direction characterizing the flow of energy in that direction can be defined.

In a reverberant sound field, consider the sound energy that propagates in a narrow column with a circular cross section, as shown in Figure 7.22. Now let the column encompass a small spherical space in the reverberant field. The ratio of the volume of the spherical region to the

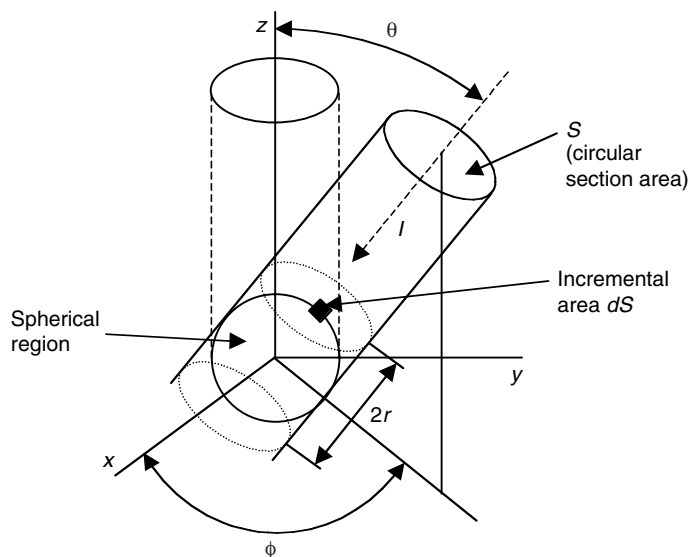


FIGURE 7.22 A schematic for the geometrical illustration for determining the effective intensity in a diffuse sound field.

volume of the cylindrical section of the column that just encloses the spherical region can be obtained as:⁹

$$\frac{4\pi r^3}{3} \cdot \frac{1}{2\pi r^3} = \frac{2}{3} \quad (7.7)$$

That is, the spherical region is occupying two thirds of the volume of the cylinder that encloses it.

In a diffuse sound field, the intensity of the incident beam in any direction would be the same and equal to I (see Figure 7.22). Now consider a sound beam that has a cross-sectional area S . The time it takes to travel through the spherical region is $2r/c$ (it is the time taken by a sound wave with a propagation speed of c to travel the length of the encompassing cylinder, $2r$). Thus, sound energy in the cylindrical region, E_c due to that beam can be obtained as:

$$E_c = I \cdot S \frac{2r}{c} \quad (7.8)$$

Also, one can compute the sound energy increment contribution from any beam in the spherical region per unit cross-sectional area of the incident beam as:

$$\Delta E = \frac{2}{3} I \frac{2r}{c} \quad (7.9)$$

The incremental area dS of the surface of a sphere with radius r is:

$$dS = r^2 \sin \theta \, d\theta \, d\phi \quad (7.10)$$

Now we can compute the total energy contribution from beams, incident from all directions, in the spherical region by integrating the incremental energy per unit area of the sphere over the whole area of the sphere. Using Equations 7.9 and 7.10, one can obtain the total energy in the spherical region as follows:

$$E = \iiint_{\text{Sphere}} \Delta E \cdot dS = \frac{4I}{3c} \int_0^{2\pi} d\phi \int_0^\pi r^3 \sin \theta \, d\theta = \frac{16I\pi r^3}{3c} \quad (7.11)$$

Assume that the energy density at the center of the spherical region is ψ . Then, the total energy in the spherical region can be obtained as:

$$E = \frac{4\pi r^3 \psi}{3} \quad (7.12)$$

Combining Equations 7.11 and 7.12, we can get an expression for the effective intensity I in any direction as a function of the local energy density, ψ :

$$I = \frac{c\psi}{4} \quad (7.13)$$

Now, for a plane wave, it is known that the intensity I_p can be written in terms of the acoustic pressure p' as follows:

$$I_p = \frac{\langle p'^2 \rangle}{(\rho c)} \quad (7.14)$$

where ρ is the mass density of the medium in which the sound is propagating. And an expression for the energy density in terms of the pressure can be obtained:

$$\psi = \frac{\langle p'^2 \rangle}{(\rho c^2)} \quad (7.15)$$

Substituting from Equation 7.15 in Equation 7.13, the effective sound intensity in a diffuse field has the following expression:

$$I = \frac{\langle p'^2 \rangle}{(4\rho c)} \quad (7.16)$$

One can see that if a diffuse sound field has the same intensity as that of a plane sound wave, the acoustic pressure associated with the diffuse field would be, on average, twice that associated with the plane wave. This is consistent with the observation of doubling the acoustic pressure at the face of a perfectly reflective wall.

7.5.2.2 Steady-State Response

In an enclosed space, the acoustic pressure at any point is the combined contribution from both the direct sound field radiated from sound sources and the reverberant field. In general, to get the combined contribution from multiple sound fields, we algebraically add the acoustic energy from each field.¹⁰ Using the notion of directivity D_θ , the acoustic pressure from the direct field at a point, which is located at a distance r from the source, can be computed as:

$$\langle p'^2 \rangle_D = \frac{W_{\rho c} D_\theta}{4\pi r^2} \quad (7.17)$$

where W is the sound power radiated by the source. It is assumed here that the point is indeed in the far field of the source (either r is sufficiently large or the source is sufficiently small). The contribution from the reverberant sound field can be obtained by carrying out sound energy balance. Basically, any sound radiated from the source has to be reflected at least once before it contributes to the reverberant sound field. The fraction of the sound power that enters the reverberant field is $(1 - \bar{\alpha})$, where $\bar{\alpha}$ is the combined absorption and transmission coefficient. By equating the sound power that is added to the reverberant field $(1 - \bar{\alpha})W$ to the sound power absorbed (actually absorbed and transmitted) through the walls while using Equation 7.16 for the sound intensity of the reverberant field, one can get the sound pressure of the reverberant field as:

$$\langle p'^2 \rangle_R = 4W_{\rho c} \frac{(1 - \bar{\alpha})}{\bar{\alpha}S} \quad (7.18)$$

where S is the total area of the enclosed walls. The sound pressure level due to combined effect of the direct and reverberant fields can be written as follows:

$$L_p = L_w + 10 \log_{10} \left\{ \left(\frac{\rho c}{400} \right) \left[\frac{D_\theta}{4\pi r^2} + \frac{4}{R} \right] \right\} \quad (7.19)$$

where $R = \frac{\bar{\alpha}S}{(1-\bar{\alpha})}$ is the constant of the enclosure (room constant). More often, the approximation $\rho c \approx 400$ ($\rho c = 414$ at 20°C) is used to simplify the expression of the sound pressure level to:

$$L_p = L_w + 10 \log_{10} \left[\frac{D_{\theta}}{4\pi r^2} + \frac{4}{R} \right] \quad (7.20)$$

One can see in Equation 7.20 that when the enclosure constant R is very small (i.e., the enclosure walls have a low absorption coefficient), the space dominated by the direct field shrinks to a small region around the sound source. On the other hand, with highly absorptive walls, the domain of the dominant direct field expands over a larger region and, at the limit, the enclosure becomes an anechoic enclosure.

7.5.2.3 Transient Response

The reverberation sound field inside an enclosure can be greatly simplified in the high-frequency range. Because the sound field in this frequency range is very close to being a diffuse sound field, it can be described in terms of a simple differential equation.¹⁰ The unsteady sound energy balance in the reverberation field can be formulated as follows: the rate of change of the energy inside the field should equal the rate of supply W_o , less the rate of losses W_a . Thus, one can write this unsteady energy balance as:

$$V \frac{d\psi}{dt} = W_o - W_a = W_o - 0.25\psi Sc\bar{\alpha} \quad (7.21)$$

where V is the volume of the enclosed space and S is the total surface area of the inside walls of the enclosure. A general solution for this first-order differential equation, Equation 7.21 can be written as follows:

$$\psi(t) = \frac{4W_o}{Sc\bar{\alpha}} + g e^{-\frac{Sc\bar{\alpha}}{4V}t} \quad (7.22)$$

where g is a constant to be determined from the initial condition of the energy density in the field. There are two conditions that this solution, Equation 7.22, can be considered. First suppose that initially the sound field was empty; that is, $\psi_o = 0$. Then at time $t = 0$, the sound source is suddenly turned on. Using the initial condition and substituting from Equation 7.15 into Equation 7.22, the response of the sound pressure inside the field can be obtained as follows:

$$\langle p'^2 \rangle = \frac{4W_o \rho c}{S \bar{\alpha}} \left(1 - e^{-\frac{Sc\bar{\alpha}}{4V}t} \right) \quad (7.23)$$

Second, we can consider that the sound field has been established and reached steady state (the sound source is steadily radiating energy to the enclosure). If the the sound source is suddenly turned off, then, at time $t = 0$, the sound power $W_o = 0$; and if the initial sound pressure is $\bar{\alpha}$, Equation 7.22 and Equation 7.29 can be used along with the initial conditions to obtain the following expression for the decaying reverberation sound field:

$$\langle p'^2 \rangle = \langle p_o'^2 \rangle e^{-\frac{Sc\bar{\alpha}}{4V}t} \quad (7.24)$$

or in terms of the sound pressure level:

$$L_{p_o} - L_p = 1.086 \left(\frac{Sc\bar{\alpha}}{V} \right) t \quad (7.25)$$

Equation 7.25 shows that the decay is proportional to the total area of the internal surface of the enclosure, the absorption coefficient, and time, but is inversely proportional to the volume of the enclosure. Now, the reverberation time (T_{60}) is readily calculated from Equation 7.25 as:

$$T_{60} = \frac{55.25 V}{Sc\bar{\alpha}} \quad (7.26)$$

This reverberation time is sometimes called Sabine's reverberation time, referring to the researcher who first introduced it.⁹ The reverberation time obtained in Equation 7.26 is somewhat approximate and that is due to the deviation of the sound field from being a perfectly diffuse field. Another derivation for the reverberation time is based on equal decay rate for all resonance modes of the enclosure. The reverberation time obtained using this approach is:

$$T_{60} = \frac{55.25V}{Sc \log_e (1 - \bar{\alpha}_{st})} \quad (7.27)$$

where $\bar{\alpha}_{st}$ is the statistical absorption coefficient of the internal surface of the enclosure.

7.6 NOISE RADIATION FROM BURNER

Industrial heaters substantially affect the level of the noise that radiates back from the air inlet of the burner (damper section). As discussed in the previous section, the sound field generated by sound sources enclosed in a confined space depends significantly on the acoustic characteristics of the internal surfaces as well as the walls of the enclosure.⁸⁻¹⁰ A burner in a process heater can be represented as a sound source inside an enclosure. Consider a simple process heater, such as the one sketched in Figure 7.23, with a single burner. There are several sound sources associated with the burner, but the major contribution comes from combustion noise and jet noise. In the high-frequency range, the sound field inside the furnace can be considered a diffuse field. For the sake of simplicity, assume that the furnace–burner combination can be represented as a single cavity (multi-cavity considerations will be given later). An acoustic energy balance to obtain the sound pressure immediately on front of the external surface of the enclosure (furnace or burner walls) p'_e can be written as follows:⁹

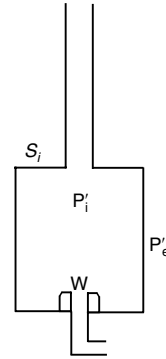


FIGURE 7.23 Schematic diagram for process heater with a single burner.

$$\frac{S_E}{\rho c} \langle p_e'^2 \rangle = W\tau_N + W(1 - \bar{\alpha}_i) \frac{S_E}{\bar{\alpha}_i S_i} \tau \quad (7.28)$$

where S_E and S_i are the external and internal surface areas of the enclosure, respectively; W is the power of all sound sources in the enclosure; τ and τ_N are the average field incidence and normal incidence transmission coefficients of the walls, respectively; and $\bar{\alpha}_i$ is the average absorption coefficient of the internal surface of the enclosure.

Numerically, $\bar{\alpha}_i$ is greater than τ .

7.6.1 MULTIPLE-CAVITY MODEL OF FURNACE-BURNER ACOUSTIC SYSTEM

An acoustic model for the furnace-burner combination will involve more than one cavity. Although all the cavities are physically connected to each other by some passages, dividing the whole internal space into multiple cavities might be required to account for the diverse acoustic properties of the internal surfaces in each cavity. For example, the internal surface of an insulated furnace has a much higher absorption coefficient than that of a burner tile with a hard surface. These cavities interact with each other through the passages that connect them. In the case of a multiple-cavity acoustic system, the whole acoustic model can be derived by applying the single-cavity model on each cavity and then considering the coupling through the interconnecting passages.

7.6.2 EVALUATION OF ACOUSTIC PROPERTIES OF INTERNAL SURFACES AND WALLS

The internal surfaces of the various cavities play an important part in the acoustic model of a furnace burner system. The effective absorption coefficient of the walls is the main property required for the acoustic model. There are many ways to evaluate the absorption coefficient of the internal surface of a cavity.^{10,11} The transient response method is the best-known method for determining the absorption coefficient of the internal surface of a cavity. The different methods of evaluating the absorption coefficient are discussed in the section on the characteristics of sound propagation. Another property required for the acoustic model is the transmission coefficient of the different walls of the cavity. The effective transmission coefficient of a wall composed of different materials can be computed from the individual coefficients of each layer of the wall.

7.6.3 EFFECT OF BACKGROUND NOISE ON BURNER NOISE LEVEL

In addition to combustion equipment, other industrial equipment such as rotating machinery, solids handling equipment, steam traps, and valve actuators emit high levels of noise, contributing to a high background noise level. Background noise creates some challenges in the process of applying noise control. First of all, a noise source that must be measured must have a noise level higher than the background noise level. If the noise source of interest is much lower than the background noise, then it is usually of no concern; but if it is close to the level of the background noise in a given location, it could be a significant contributor that is difficult to measure.

Next, if analysis needs to be performed to identify the noise source's emissions in different frequencies, it should be kept in mind that background noise can mask or provide inaccurate information in frequencies where the noise source's level is comparable to the background noise. The usual approach to tackle background noise interference is to measure close to the noise source such that its noise dominates that of surrounding sources. This may or may not be possible with industrial burners, depending on their proximity to each other and the difference in the noise levels emitted. If the noise from a burner manifests mostly inside the furnace and is poorly transmitted to the exterior of the burner, then the job of identifying, isolating, and measuring the noisy burner becomes more complicated.

It should also be noted that background noise or noise from nearby sources can elevate the noise measured at a given piece of equipment. See next section for an example of the additive effect of nearby noise sources.

7.6.4 NOISE LEVEL DUE TO MULTIPLE BURNERS

Multiple noise sources, such as burners in a furnace, create a higher level of noise than each of the individual sources separately. Of course, this additive effect depends on the number of burners that are within a certain distance of the point of interest. The importance of this effect is that if a certain noise level is to be achieved, say 85 dBA at some distance from the burners, then each individual burner will have to perform at a level lower than 85 dBA to avoid exceeding the limit due to the additive effect.

The following example demonstrates the cumulative effect of noise from neighboring burners using a simple three-burner array. Obviously, having more burners in the array will increase the measured noise, but the impact of the additional burners is reduced proportionately because they are increasingly further away from the point of interest. The formulas shown in the following example take into account the effect of distance and can be used for arrays with any number of sources.

7.6.4.1 Example

A burner manufacturer will typically guarantee a burner noise level at a location 3 ft (1 m) directly in front of the muffler. When several burners are installed in a furnace, however, the noise level 3 ft (1 m) from the burner may be higher than for a single burner due to the noise contribution from surrounding burners. The purpose of this section is to give an example that illustrates the noise level increase due to noise emitted from surrounding burners.

As an illustration, assume a furnace with a simple burner configuration as illustrated in Figure 7.24. If burner B is operating alone and the noise level is 85 dB at location 2, how is the noise level determined at location 2 when all burners are operating? First, find the sound power level L_w emitted from each burner, assuming that the source is emitted at the muffler exit at points 1A, 1B, and 1C. Assume that the noise spreads over a uniform sphere from each of these points. The sound power level can be calculated as follows:

$$L_w = L_p - 10 \log_{10} \left(\frac{1}{4\pi r^2} \right) - 10.5 \quad (7.29)$$

where L_p is the sound pressure level and r is the distance from the source (in feet). The noise level 3 ft (1 m) from burner B (location 2) is 85 dB when it is operating alone. From Equation 7.29 we find that the $L_{wB} = 95.03$ dB. Assuming that all burners are operating at the same conditions, we know that

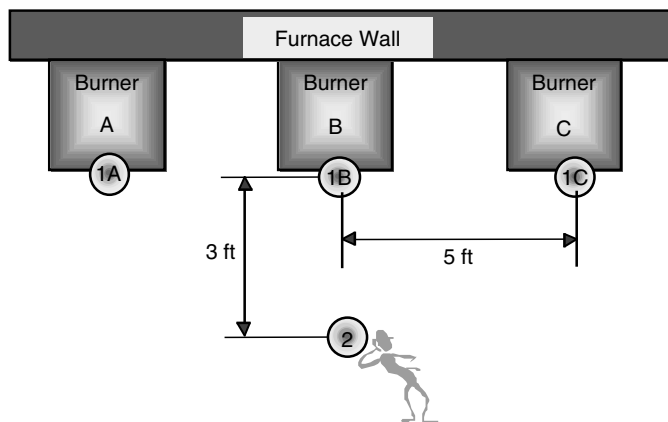


FIGURE 7.24 Burner noise example.

the sound power level must be 95.03 dB for each one. The sound pressure level contribution, L_p can now be calculated at location 2 when burner A is operating alone by solving Equation 7.29 for L_p :

$$L_p = L_w + 10 \log_{10} \left(\frac{1}{4\pi r^2} \right) + 10.5 \quad (7.30)$$

For this case, $L_{wA} = 95.03$ and $r = (5^2 + 3^2)^{0.5} = 5.83$ ft. Substituting these values into Equation 7.30 gives $L_{pA} = 79.2$ dB. This is the sound pressure level contribution emitted from burner A measured at location 2. Because the distance from burner C to location 2 is the same distance away, we know that the sound pressure level contribution from burner C at location 2, L_{pC} , is also 79.2 dB. The total sound pressure level at location 2 can be determined by adding the sound pressure level contribution from each burner (79.2 dB + 79.2 dB + 85 dB). The sound pressure levels can be added by using the following equation:

$$L_{p_{total}} = 10 \log_{10} (10^{L_{pA}/10} + 10^{L_{pB}/10} + 10^{L_{pC}/10}) = 86.8 \text{ dB} \quad (7.31)$$

For this example, the noise level will be approximately 1.6 dB higher when all the burners are operating than if burner B is operating alone.

7.7 NOISE CONTROL AND MUFFLERS

Noise control, obviously, is the desired end result of studying and characterizing noise from industrial burners. Noise control can be implemented either before the fact by including noise control features in the design of the combustion components, or after the fact by applying attenuation devices such as mufflers.

The major requirements that drive the design of industrial burners are, usually, good flame shape and low emissions. The need to achieve good combustion performance usually leaves little opportunity to aggressively design for noise reduction. However, if the opportunity arises, there are certain noise reduction strategies that can be implemented in the design phase.

Jet noise is a major contributor to industrial burner noise and can be addressed at the design phase by using multiple smaller gas orifices instead of one large one. Equation 7.32 gives an empirical relationship between the orifice diameter and the characteristic frequency of the jet noise due to that orifice:

$$F = SU/D \quad (7.32)$$

where F is the characteristic frequency, U is the velocity of the gas jet as it emerges from the orifice, D is the orifice diameter, and S is the struhel number, which is a constant that typically varies between 0.2 and 0.4.

This relationship indicates that as the orifice diameter (the denominator) reduces, the frequency increases. Because gas jet noise produces high-frequency noise, shifting the frequency to a higher realm will tend to move some of the noise out of the audible regime and past the high-frequency threshold of hearing. Figure 7.25 shows the peak noise emissions created by a jet with reference to the threshold of hearing in that range. As the orifice diameter is reduced, the peak shifts to the right and a significant part of the area under the curve is shifted out of the threshold of audibility.

It is also important to note that jet noise, being high-frequency noise, is very directional in nature. That is, the impact of the noise drops off steeply with increasing angle from the axis of the jet. Figure 7.26 shows the decrease in sound pressure level with increasing angular displacement from the axis of the jet. The impact of the directionality of the jet noise is an important parameter

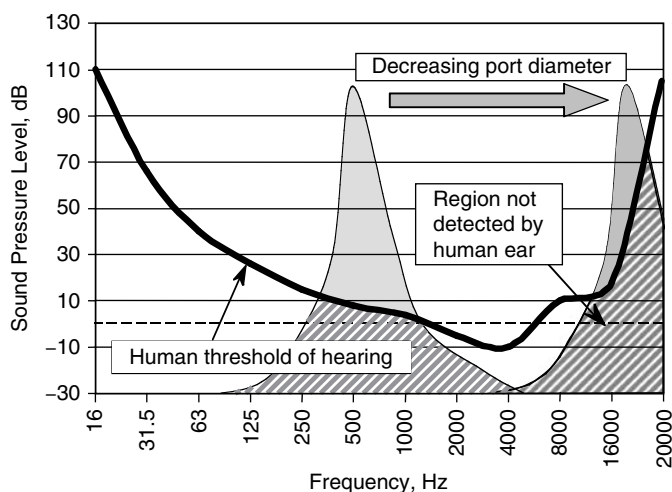


FIGURE 7.25 Threshold of hearing in humans.

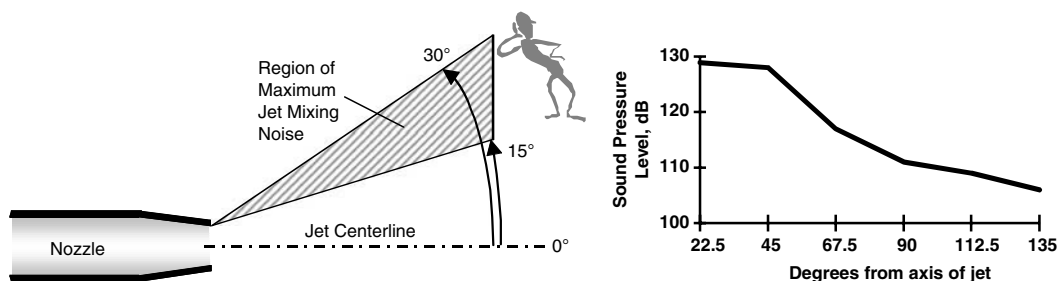


FIGURE 7.26 Region of maximum jet mixing noise and attenuation with angular displacement from axis of jet.

that can be used to reduce noise by introducing convolutions in the air passage to a burner. This use of convolutions or turns to reduce noise is usually implemented in the muffler and air plenum assembly. More information on this is presented further on in the text.

Another mechanism that reduces noise emitted from a flame, specifically combustion noise, is elongation of the flame. That is, when the combustion process is slowed down, either by delayed mixing or by dilution with inerts, the thermo-acoustic efficiency of the flame is reduced. Elongation of the flame or delayed mixing is a technique used to reduce NO_x and so it is typical to find that low-NO_x burners are less noisy than conventional burners. On the other hand, discussion about elongation of the flame is somewhat academic because it is rarely used solely as a noise reduction strategy at the design phase.

The rest of this chapter section is dedicated to discussions about mufflers. Although jet noise can sometimes be controlled through design, most often, the use of a muffler is the main noise control strategy.

A muffler is a barrier and absorber device that attenuates the noise emission from its source.^{8,9} In burner applications, mufflers are needed to meet stringent noise regulations. The muffler, as a barrier, reduces noise by directing and reflecting it away from the listener. Noise reduction in parts of the muffler, such as a bend or the inlet, is achieved through the principles of a barrier. Lined, straight ducts reduce noise by absorption, which is the conversion of acoustic energy into heat by friction in the insulating material, and they are good examples of typical absorbers. Empirical and theoretical models have been developed to predict the attenuation from various types of lined ducts. Designing a muffler to achieve certain noise reduction depends on both the noise source and the

TABLE 7.5

Predicted Octave Band Attenuation for a Rectangular Duct Lined on Two Opposite Sides. Circular or Square Lined Duct on All Four Sides Give Twice the Attenuation Shown Here. Where h is the Half Width of the Airway, and λ is the Sound Wavelength

$2h/\lambda$	0.01	0.02	0.1	0.6	0.9	1.2	2	10
Attenuation rate (dB per h duct length)	0	0	0.06	1.75	2.89	2.83	0.67	0.06

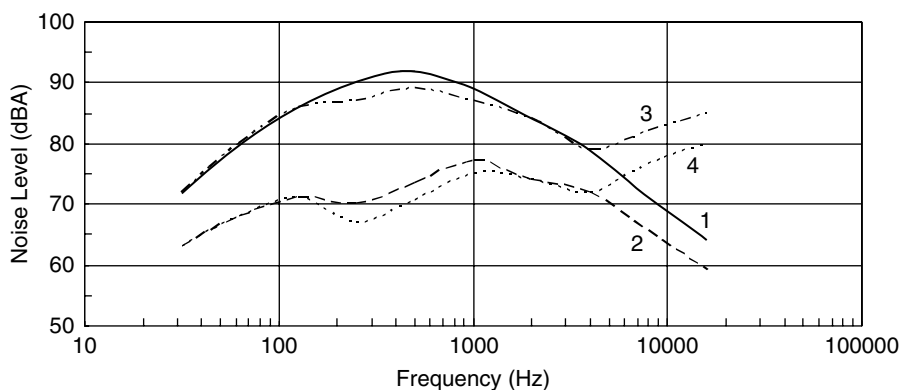


FIGURE 7.27 Noise curves for two burners with the same muffler spectral insertion loss: curve 1 is burner 1 unmuffled with total sound pressure level $L_{p1} = 96.5$ dBA; curve 2 is burner 1 muffled with $L_{p1m} = 81.5$ dBA; curve 3 is burner 2 unmuffled with $L_{p2} = 95$ dBA; and curve 4 is burner 2 muffled with $L_{p2m} = 84$ dBA. The total attenuation for burner 1 is $IL_1 = 15$ dBA, and for burner 2 is $IL_2 = 11$ dBA.

structure of the muffler. The optimal muffler for one noise source may perform poorly with another source. The spectral insertion loss of a muffler, which is the difference between noise levels before and after inserting the muffler in each spectral band, does not depend on the noise spectrum of the noise source. However, the total noise attenuation of the muffler strongly depends on the noise spectrum of the source (see the illustrative example in [Figure 7.27](#)). In the example, the overall sound pressure level for burner 1 is 96.5 dBA (curve 1), and for burner 2 is 95 dBA (curve 3); and after inserting the same muffler, the sound pressure level for burner 1 is 81.5 dBA (curve 2) and for burner 2 is 84 dBA (curve 4). Although burner 1 has higher noise, the muffler attenuation in the case of burner 1 (15 dBA) is higher than that of burner 2 (11 dBA). For example, if the muffler was designed to achieve a sound level of 82 dBA when used with burner 1, the muffler cannot achieve the same sound level with burner 2, although burner 2 has a lower sound level than burner 1.

The literature provides some theoretical models that can predict the rate of noise attenuation in straight lined ducts.^{8,9} The rate of attenuation depends on how the sound interacts with the acoustic insulation material, both locally reacting and in a bulk mode. In the local reacting regime, the sound is prevented from propagating in the material in any direction other than normal to the surface by some rigid partitions. In the bulk reacting regime, the sound can propagate in the liner parallel to the surface, and this is more practical to use. [Table 7.5](#) shows predictions for the rate of noise attenuation in a straight duct that is lined on two opposite sides when the liner thickness is equal to one-quarter of the half-width of the airway.⁹ The attenuation rate depends on the nondimensional frequency $2h/\lambda$, the nondimensional liner thickness l/h , and the nondimensional resistance of the liner $R_l l/\rho c$, where h is the half-width of the airway, l is the liner thickness, λ is the wavelength of the sound wave, c is the speed of sound in the duct, ρ is the density of fluid flowing in the duct, and R_l is the liner flow resistivity.

The noise reduction contribution from a bend has been identified in the literature from empirical information. The correction for the bend depends on the condition of the sound that is propagating in the bend as well as the bend configuration. Some of the empirical data that has been reported in the literature⁹ for different bend configurations and sound wave inputs it has been shown that the largest bend attenuation is for the case a of a rectangular lined duct and axial plane wave input. A plane wave can result in practical applications because diffuse sound that propagates for some length along the lined duct before it hits the bend, resembles a plane wave. Correction for attenuation for the case of lined duct with rectangular cross-section and diffuse input sound is given in Table 7.6. Experimental evaluations conducted by the authors also indicate a similar contribution by a bend.

Empirical data has been reported in the literature for the duct inlet correction. Because part of the sound is absorbed by the lining, sound that reflects at the wall will be more quickly attenuated than sound that passes at grazing incidence. Thus, sound at all angles of incidence at the entrance to a duct will very rapidly attenuate, until only the portion that is propagating axially remains. It has been determined that such an effect may introduce additional attenuation, the inlet correction. Table 7.7 shows that the inlet correction depends on a nondimensional frequency $S^{0.5}/\lambda$, where S is the cross-sectional area of the duct and λ is the sound wavelength.⁹ Figure 7.28 also shows that the inlet correction is more effective at high frequencies.

Exit loss has been also reported in the literature, and it has been attributed to the sudden change of cross section at the end of a duct flush mounted with a wall or ceiling. Exit loss that has been measured for circular and rectangular ducts in the literature¹⁰ is shown in Figure 7.28. One can see that the attenuation is greater in the low frequency range.

TABLE 7.6
Correction for Attenuation at a 90° Bend with Rectangular Lined Duct and Diffuse Input Sound. S is the Duct Cross-Sectional Area, and λ is the Sound Wavelength

$S^{0.5}/\lambda$	0.1	0.2	0.4	0.6	0.8	1	2	4
Bend Correction (dB)	0	0.667	0.667	11	11.3	11	10	10

TABLE 7.7
Duct Inlet Correction for Random-Incidence Sound. S is the Duct Cross-Sectional Area, and λ is the Sound Wavelength

$S^{0.5}/\lambda$	0.1	0.2	0.4	0.6	0.8	1	2	4
Bend Correction (dB)	0	3	6.834	8.834	9.5	9.8	10	10

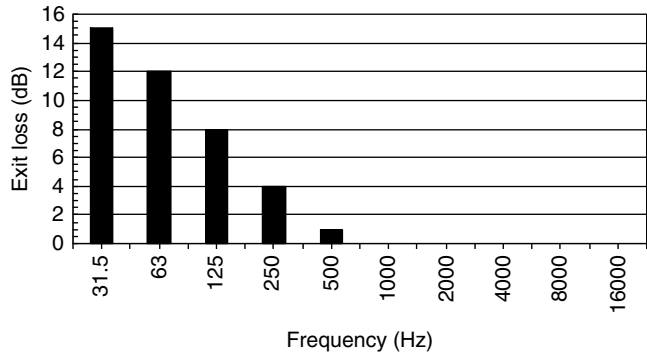


FIGURE 7.28 Exit for rectangular duct size of 0.06 m². Empirical data.

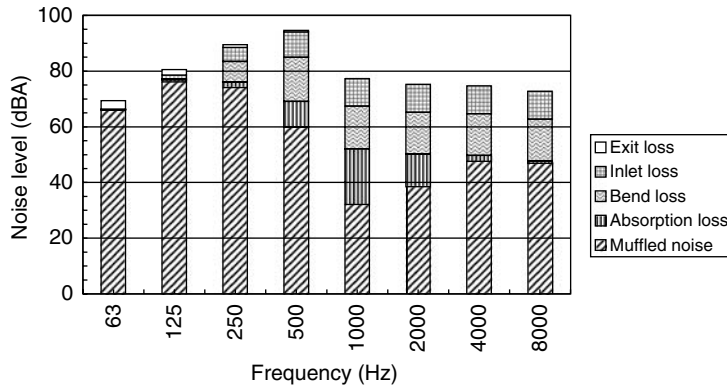


FIGURE 7.29 Insertion loss of a duct muffler with a single bend. The unmuffled noise level is 96 dBA and the muffler insertion loss is 17.5 dB.

7.7.1 EVALUATION OF MUFFLER INSERTION LOSS

Once the configuration and dimensions of a muffler are known, evaluation of the muffler insertion loss (which is the noise reduction due to the muffler insertion) is straightforward. An example of how insertion loss can be determined is shown in Figure 7.29. In this figure, the unmuffled noise in each octave band is at the top value of the column that corresponds to the octave band. To arrive at the muffled noise level—and hence the insertion loss of the muffler’s absorption—bend, inlet, and exit losses must be subtracted from the unmuffled noise level in each octave band. First, the absorption loss can be computed using the width and length of each straight part of the duct as well as the thickness of the insulation. The computation of the absorption loss can be done using Table 7.5. Second, the bend loss for each bend can be determined using the cross-sectional area of the bend and Table 7.6. Third, the inlet loss can be obtained using Table 7.7 and the cross-sectional area of the inlet. Fourth, the exit loss can be obtained using Table 7.6 and the cross-sectional area of the exit. In this example, the unmuffled noise level is 96 dBA and the muffled noise level is 78.5 dBA, making the total muffler insertion loss 17.5 dB.

7.7.2 ITERATIVE DESIGN OF MUFFLERS

Any muffler configuration involves a large number of dimensions, so muffler design is somewhat iterative in nature. Starting by satisfying the constraints that are imposed on the muffler, such as pressure drop and certain dimensional limitations, one can arrive at a rough design for the muffler configuration. Then, by computing the muffled noise level for this configuration, an iterative process can be started where the other dimensions are varied until the required noise level is achieved. It may take three to five trials to reach the final design. In general, as with any design work, the larger the number of constraints imposed on the muffler performance, the more difficult will it be to accomplish the design, and a larger number of trials will be needed.

7.8 GLOSSARY

Absorption: Conversion of sound energy into another form of energy, usually heat, when passing through an acoustical medium.

Absorption coefficient: Ratio of sound absorbing effectiveness, at a specific frequency, of a unit area of acoustical absorbent to a unit area of perfectly absorptive material.

Acoustics: Science of the production, control, transmission, reception, and effects of sound and of the phenomenon of hearing.

Ambient noise: All-pervasive noise associated with a given environment.

Anechoic room: Room whose boundaries effectively absorb all incident sound over the frequency range of interest, thereby creating essentially free field conditions.

Audibility threshold: Sound pressure level, for a specified frequency, at which persons with normal hearing begin to respond.

Background noise: Ambient noise level above which signals must be presented or noise sources measured.

Decibel scale: Linear numbering scale used to define a logarithmic amplitude scale, thereby compressing a wide range of amplitude values to a small set of numbers.

Diffraction: Scattering of radiation at an object smaller than one wavelength and the subsequent interference of the scattered wavefronts.

Diffuse field: Sound field in which the sound pressure level is the same everywhere and the flow of energy is equally probable in all directions.

Diffuse sound: Sound that is completely random in phase; sound which appears to have no single source.

Directivity factor: Ratio of the mean-square pressure (or intensity) on the axis of a transducer at a certain distance to the mean-square pressure (or intensity) that a spherical source radiating the same power would produce at that point.

Far field: The part of the sound field in which sound pressure varies inversely with distance from the source. This corresponds to a reduction of approximately 6 dB in level for each doubling of distance.

Free field: An environment in which there are no reflective surfaces within the frequency region of interest.

Hearing loss: An increase in the threshold of audibility due to disease, injury, age, or exposure to intense noise.

Hertz (Hz): Unit of frequency measurement, representing cycles per second (cps).

Infrasound: Sound at frequencies below the audible range (i.e., below about 16 Hz).

Isolation: Resistance to the transmission of sound by materials and structures.

Loudness: Subjective impression of the intensity of a sound.

Masking: Process by which the threshold of audibility of one sound is raised by the presence of another (masking) sound.

Near field: That part of a sound field, usually within about two wavelengths from a noise source, where there is no simple relationship between sound level and distance.

Noise emission level: dB(A) level measured at a specified distance and direction from a noise source, in an open environment, above a specified type of surface. Generally follows the recommendation of a national or industry standard.

Noise reduction coefficient (NRC): Arithmetic average of the sound absorption coefficients of a material at 250, 500, 1000, and 2000 Hz.

Phon: Loudness level of a sound, numerically equal to the sound pressure level of a 1-kHz free progressive wave, which is judged by reliable listeners to be as loud as the unknown sound.

Pink noise: Broadband noise whose energy content is inversely proportional to frequency (−3 dB per octave or −10 dB per decade).

Power spectrum level: Level of the power in a band 1 Hz wide referred to a given reference power.

Reverberation: Persistence of sound in an enclosure after a sound source has been stopped. Reverberation time is the time (in seconds) required for sound pressure at a specific frequency to decay 60 dB after a sound source is stopped.

Root mean square (RMS): The square root of the arithmetic average of a set of squared instantaneous values.

Sabine: Measure of sound absorption of a surface; 1 metric sabine is equivalent to 1 m² of perfectly absorptive surface.

Sound: Energy that is transmitted by pressure waves in air or other materials and is the objective cause of the sensation of hearing. Commonly called noise if it is unwanted.

Sound intensity: Rate of sound energy transmission per unit area in a specified direction.

Sound level: Level of sound measured with a sound level meter and one of its weighting networks. When A-weighting is used, the sound level is given in dB(A).

Sound level meter: An electronic instrument for measuring the RMS of sound in accordance with an accepted national or international standard.

Sound power: Total sound energy radiated by a source per unit time.

Sound power level: Fundamental measure of sound power defined as

$$L_w = 10 \log \frac{P}{P_0} \text{ dB}$$

where P is the RMS value of sound power in watts, and P_0 is 1×10^{-12} W.

Sound pressure: Dynamic variation in atmospheric pressure. The pressure at a point in space minus the static pressure at that point.

Sound pressure level: Fundamental measure of sound pressure defined as

$$L_p = 20 \log \frac{P}{P_0} \text{ dB}$$

where P is the RMS value (unless otherwise stated) of sound pressure in pascals, and P_0 is 20 μ Pa.

Sound transmission loss: Ratio of the sound energy emitted by an acoustical material or structure to the energy incident upon the opposite side.

Standing wave: A periodic wave having a fixed distribution in space that is the result of interference of progressive waves of the same frequency and kind. Characterized by the existence of maxima and minima amplitudes that are fixed in space.

Thermo-acoustic efficiency: A value used to characterize the amount of combustion noise emitted from a flame. Defined as the ratio of the acoustical power emitted from the flame to the total heat release of the flame.

Ultrasound: Sound at frequencies above the audible range (i.e., above about 20 kHz).

Wavelength: Distance measured perpendicular to the wavefront in the direction of propagation between two successive points in the wave, which are separated by one period. Equals the ratio of the speed of sound in the medium to the fundamental frequency.

Weighting network: An electronic filter in a sound level meter that approximates, under defined conditions, the frequency response of the human ear. The A-weighting network is most commonly used.

White noise: Broadband noise having constant energy per unit of frequency.

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