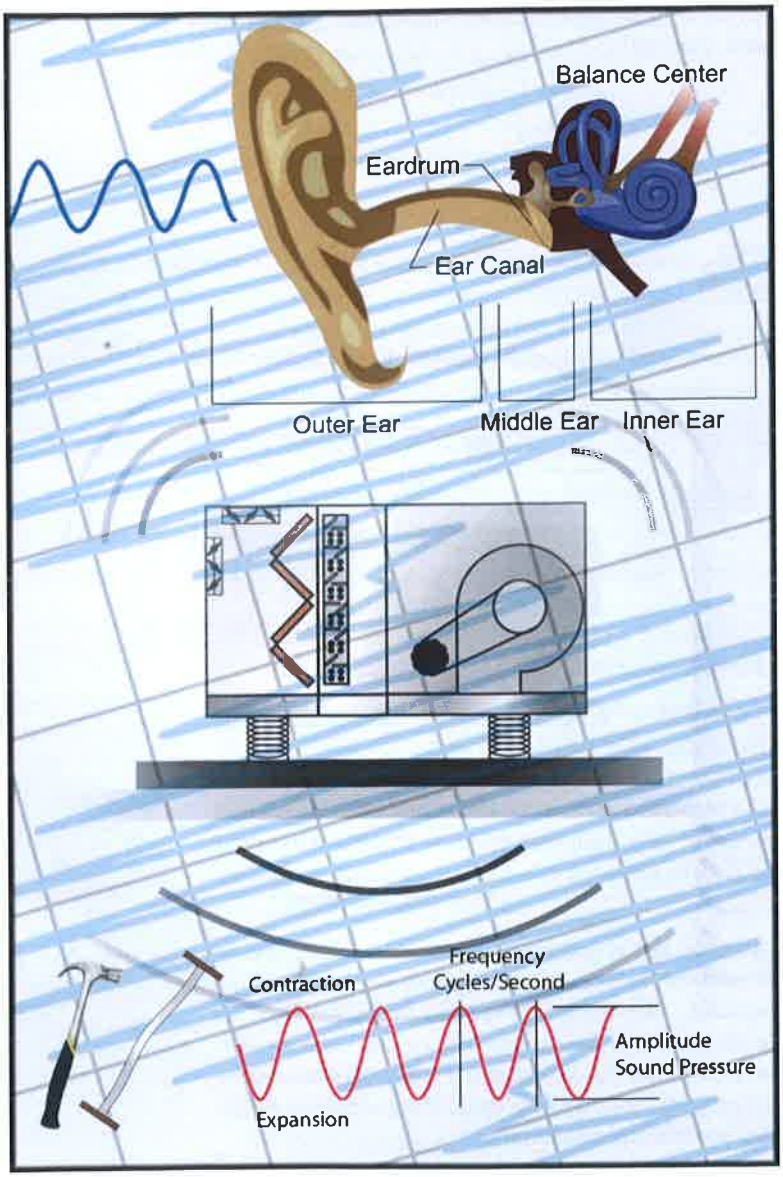
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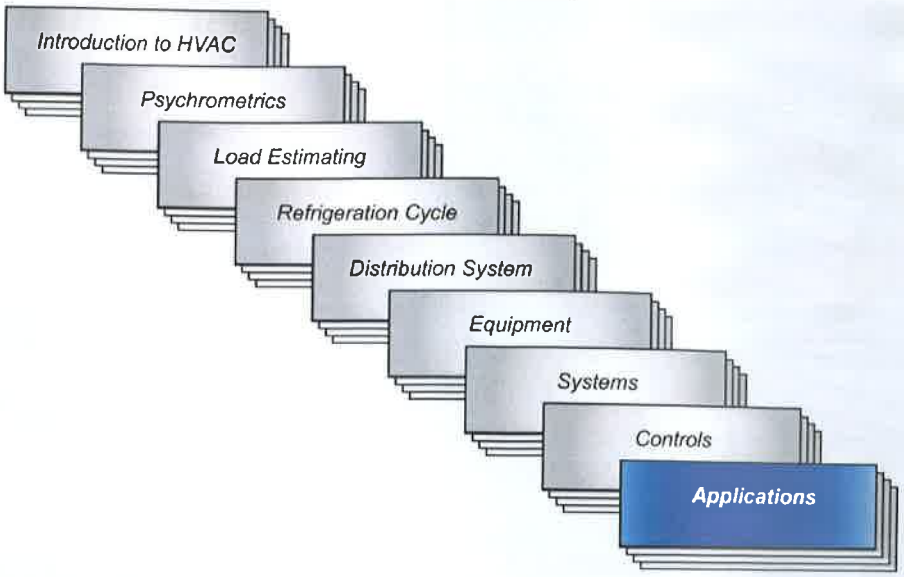
APPLICATIONS

HVAC Acoustics and Vibration

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Acoustic considerations for project designs are often overlooked. When overlooked, these issues result in noise issues that can be more expensive than if acoustic design were considered in project development. This TDP introduces system designers to the principles of acoustics and includes suggestions on how to address acoustic issues early in design. An approach is also presented on how to address an existing project with noise issues. Understanding acoustic design principles will help designers select and apply equipment and design distribution systems that more cost effectively meet the project's total environmental quality (TEQ) goals.

This module has seven sections. The first explains acoustic terms and how to add and subtract sound levels. The second section discusses the methods used to establish an acoustic rating both indoors and outdoors, including how manufacturer sound data is generated. The next two sections describe how to determine the acoustic design goal and how to estimate the sound at the receiver using the source-path-receiver concept. Specific guidelines are provided on how to estimate the sound at the equipment to control noise. The next section discusses troubleshooting existing projects, followed by controlling vibration at the design stage. Finally, guidelines are provided for preparing acoustic specifications.

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The information in this manual is offered as a general guide for the use of industry and consulting engineers in designing systems. Judgment is required for application of this information to specific installations and design applications. Carrier is not responsible for any uses made of this information and assumes no responsibility for the performance or desirability of any resulting system design.

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Introduction

This TDP module will introduce you to the fundamentals of acoustics and vibration as they apply to the HVAC field. While the system may heat and cool the building properly, a system that produces unwanted noise or vibration will not be acceptable to the tenants and the building owner. As designers and installers, we have an obligation to provide high quality installation for the customer, which means more than selecting equipment that meets only the thermal load requirements of the building. Only through proper design and application practices may we avoid problem jobs. It is more cost effective to design for acceptable sound levels than it is to correct problems once the system is installed.

This module will begin by describing sound waves and the basic terminology used in the field of HVAC acoustics. The next segment discusses measuring sound levels and rating methods used by the industry to describe and specify sound levels. The next segment provides guidance on determining the acoustical design criteria of a project. The module then illustrates how to utilize manufacturer's sound data to predict the resulting sound levels in the conditioned space or at a property line. The following sections provide recommendations of how to approach projects with acoustical problems and control vibration. Finally, the module describes how to write an acoustics and vibration specification.

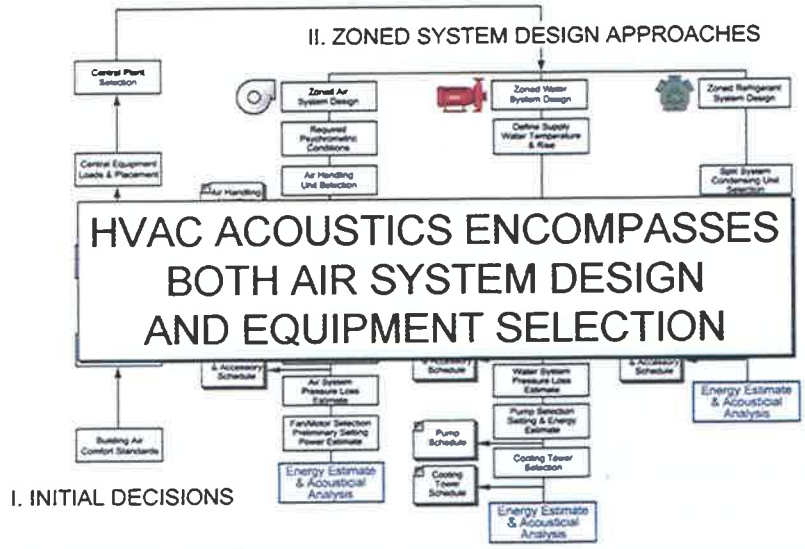


Figure 1

Acoustics is a component of the commercial design process.

Acoustics is a topic that is not well known or understood by the average HVAC technician or designer. Much of the industry's publications are intended for the acoustical specialists, making the topic confusing for most HVAC designers and contractors. This TDP module provides a working knowledge on the topic of acoustics with practical guidelines for commercial design. These guidelines will serve as a foundation of information that will help designers and technicians avoid and recognize the potential for acoustical problems in most applications. However, jobs with special or stringent acoustic requirements, such as concert halls, will often require the services of an acoustical engineer. Acoustics is an important part of the overall design process, encompassing equipment selection, application, installation, and air and piping distribution systems.

Ensuring desirable acoustical levels is more than selecting quiet equipment. If the air system is improperly sized or installed, unwanted noise will generate additional noise within the space despite the quiet equipment. Evidence of this can be found in a whistling diffuser or rumble from the ducts when the unit starts.

Sound can come from three sources: mechanical and electrical systems in the building (of which HVAC systems are usually the largest contributor), sound outside the building, and sound generated from sources in the building. The room sound level is the result of all three sources of sound. Sound control is a function of the sound sources, their intensity, frequency, and quality, and the type of activity in the space.

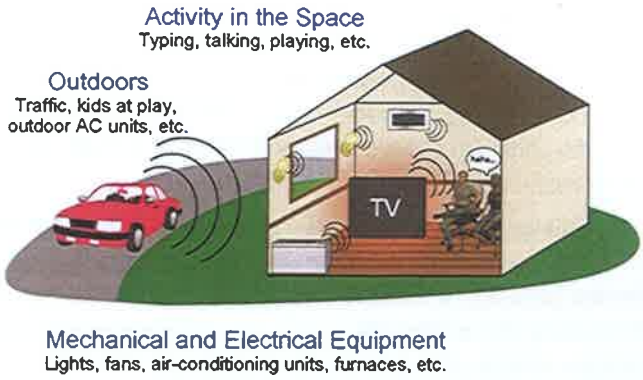


Figure 2

Sound levels in a space are a function of three sources.

The ***total environmental quality*** of a building is determined through the decisions made in the design process. Total environmental quality is more than the comfort conditions of the space; lighting levels, interior design, acoustics, and vibrations all influence the comfort and productivity of building occupants. In HVAC design, this means that the acoustic and vibration goals must be established in the design phase as part of the comfort requirements for the building. An understanding of what sound and noise are and how they are generated and controlled will allow designers to develop specifications that achieve the design goals.

Ideal Sound Quality is:

- ***Balanced***
- ***Smooth***
- ***Steady***

The purpose of acoustical design is to ensure effective generation of desirable sounds and exclusion of undesirable sounds. A sound is considered undesirable when occupants find it objectionable. Occupants may find sounds objectionable because of the volume, the vibration, or the pitch of the sound. Remember that some background noise in the space helps to mask background conversations and outside noise. The ideal background sound is one that is balanced, smooth, and steady. A ***balanced sound*** is one in which no one band is dominant over the sound spectrum. Smooth ***sound quality*** indicates no audible tones such as hums or whistles. Finally, a ***steady sound*** is one that has few fluctuations in level. For example, sound transmitted down the duct when the compressor turns on and off. These conditions are defined as sound quality.

Issues of acoustical quality within spaces have become a more important part of the design process in recent years. For some office buildings, the rental agreement may specify a maximum sound level that must not be exceeded in office spaces. In addition, the American Acoustical Society and the American National Standards Institute (ANSI) are setting much lower sound level criteria for acoustical design in school classrooms. While many projects with stringent acoustical requirements, such as concert halls, will use an engineer specializing in acoustics, every designer should have at least a rudimentary knowledge of acoustics. This knowledge will help avoid issues that may become a problem, understand the impact of manufacturer's data to actual job conditions, and know when an acoustical specialist is required.

Acoustics Fundamentals

The first step in designing for acoustics is to understand what sound is, how we hear sound, and how sound is transmitted. Secondly, we must understand how sound is measured and what constitutes good sound quality. Throughout these discussions, terms will be introduced that are essential to understanding the topic. The science of acoustics has its own vocabulary and the terms used need to be understood.

The Sound Wave and Human Hearing

Sound is energy. Sound energy is transferred through a fluid as the result of a series of minute dynamic changes in the atmospheric pressure caused by alternate compressions and expansions of the fluid. For human hearing, the fluid is usually air. These small pressure variations cause our eardrums to vibrate, which causes our hearing mechanism to work. The rate at which these pulsations occur is called the ***frequency*** and is described in cycles per second (1 cycle per second equals 1 Hz). Each cycle is one complete expansion and contraction. The speed at which these variations travel is a function of the fluid they are traveling in.

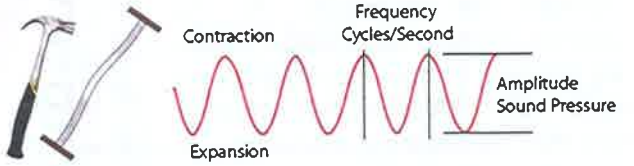


Figure 3

Sound is a wave of air pulsations.

The human ear is sensitive to a very wide range of pressure fluctuations from about 20 cycles per second to around 20,000 cycles per second. The response can vary with each person. We receive most of our sound through our ears; however, at frequencies below 31.5 Hz , we can begin to feel sound vibrating our body and this low rumble can be even more objectionable than a stronger, higher frequency sound.

The speed or frequency at which the sound waves occur is perceived as pitch or ***tone***. The sound level we hear is also a function of the pressure of the waves or the ***amplitude***. Amplitude is a measure of the strength of the sound wave. Like throwing a pebble in water, the amplitude starts out strong and slowly diminishes in strength as the wave moves out in all directions.

The human ear responds to a very wide range of sound pressure amplitudes from the sound of fluttering of butterfly wings to the roar of jet engine. The audible sound pressure varies from two ten thousand millionths of an atmosphere (2.94×10^{-9} psi), to two thousands of an atmosphere (0.0294 psi). This is one million times stronger than the lowest level.

The part of the ear that we see is the external ear, which serves to help us locate sounds in space. At the end of the ear canal is an eardrum. When ***sound pressure*** reaches the eardrum, the eardrum membrane vibrates in unison with the sound pressure. Behind the eardrum are three small bones that mechanically amplify the motion of eardrum and couple the resulting vibrations to the inner ear as pressure waves. In the inner ear, these sound waves are changed to a series of nerve impulses that are sent to the brain, where the brain decodes the impulses, resulting in our "hearing sound."

The initial part of our hearing mechanism behaves in the same manner as a microphone, by picking up the sound pressure. However, the process that takes place in the middle ear and the brain makes the hearing operation quite remarkable.

We have the capability to identify the sources of sound by "ear," something that the very best sound analysis equipment cannot do. Our hearing allows us to hear over an impressive range of sound frequencies and pressures. If our ears were any more sensitive to low sound pressures, we would hear the movement of the microscopic air molecules. At the opposite extreme, the loudest sounds we can hear without causing pain are more than 1,000,000 times the pressure than the threshold of hearing.

The figure indicates the range of frequencies and sound pressures associated with our hearing. Sound is can be both pleasant as with speech or music, or it can be annoying as in the roar of a jet engine.

Because the range of sound pressures is so large, the decibel notation system is used to describe sound levels. This is a logarithmic scale that compresses the wide hearing range to a more workable range; starting at zero decibels for the threshold of hearing and 140 decibels for the threshold of pain. The use of the decibel system will be described in more detail in the next section.

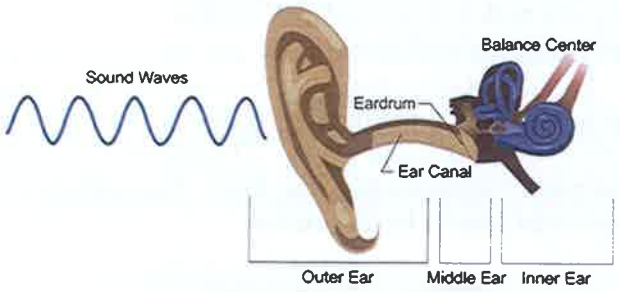


Figure 4

The human hearing mechanism senses changes in sound pressure.

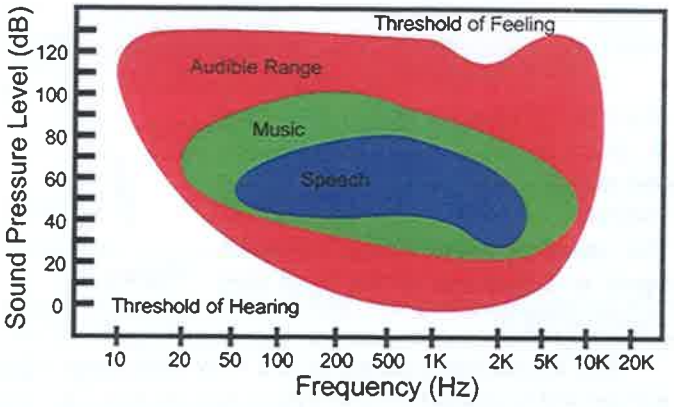


Figure 5

Areas of Hearing

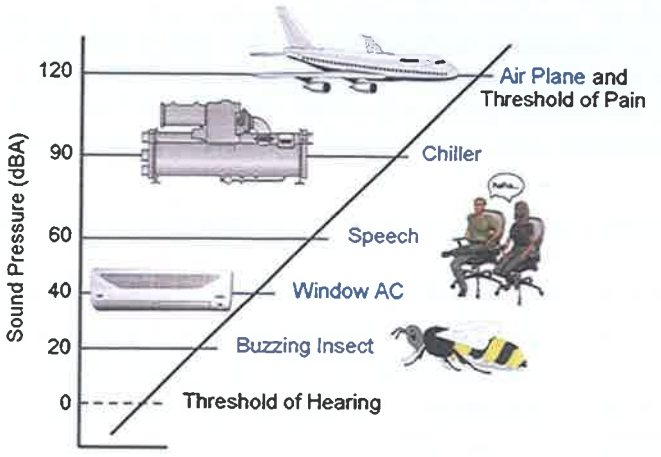


Figure 6

Range of Sound Pressure

If sound to the eardrum is too loud for too long, permanent hearing damage can result in the inner ear. OSHA has set limits on the maximum exposure time based on the sound pressure level.

Average Daily Sound Level	Time of Noise Exposure
90	8 Hours
95	4 Hours
100	2 Hours
105	1 Hour
110	30 Minutes
115	15 Minutes

Figure 7

OSHA Sound Limits

Basic Acoustic Terms

As previously explained, sound is a wave; as a result, we can use terms that describe wave action to describe sound.

In air, sound is simply pressure fluctuations (sound waves) that travel from the sound ***source*** to the sound ***receiver***. The speed (c) with which this disturbance travels depends on the density and elasticity of the medium through which it travels.

Typical values for some common materials are:

Refrigerants	500 feet/second
Air	1100 feet/second
Water	5000 feet second
Steel	16000 feet/second

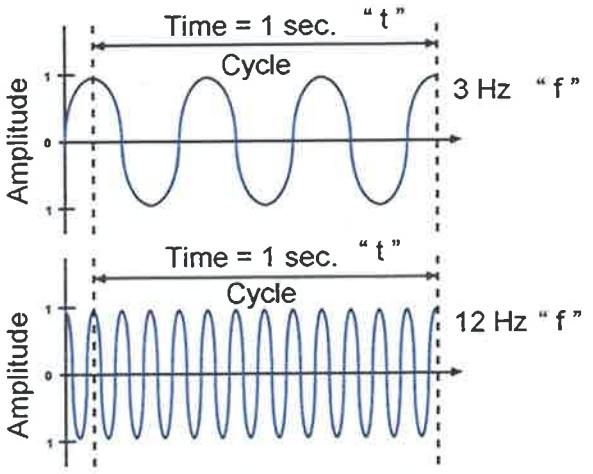


Figure 8

Sound is a wave.

The sound wave is defined by its frequency, wavelength, and amplitude:

As was defined earlier, frequency (f) is the rate that sound waves repeat with in a fixed time period (t). This is normally expressed as frequency per second, or Hz (1/t).

If the frequency and speed of the sound pressure wave is known, we can determine the distance the wave will travel. This distance is the ***wavelength*** and is measured in feet.

Important acoustic terms:

Frequency – the time at which a sound is repeated, also know as pitch or tone measured in Hz (cycles/second)

Amplitude – the intensity of the sound, measured in Pascal

Wavelength – the distance the wave travels in feet in one cycle

$$\text{Wavelength} = \lambda = c / f$$

Where: c = Speed of Sound in Air = 1127 ft./sec.

f = Frequency (Hz) (cycles per second)

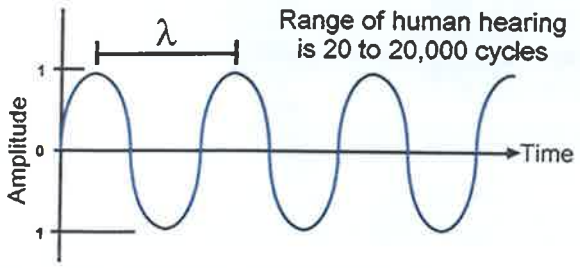


Figure 9

Wavelength is determined by the frequency

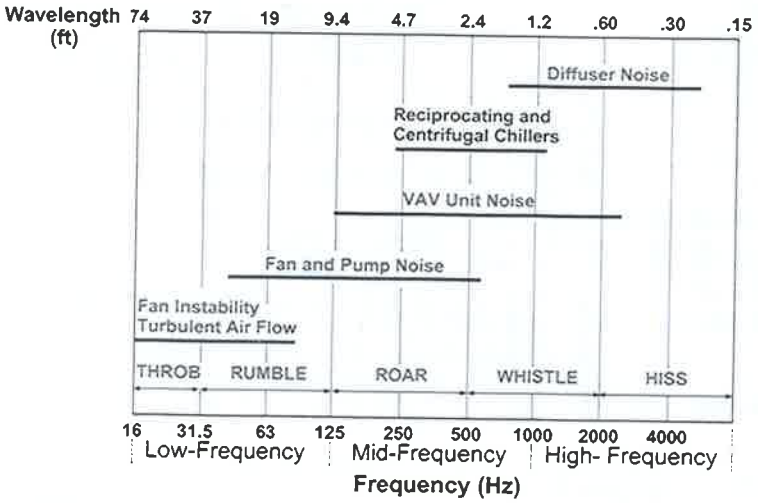


Figure 10

Wavelengths of Typical HVAC Equipment

The figure shows some typical HVAC equipment and the frequency range over which it generates sounds as well as the corresponding wavelength. Notice that the wavelengths can range from over 75 feet to less than 1/10 inch.

Wave Interference

Because sound is a repetitive wave, sound can be reflected off surfaces or come from different sources and add or subtract sound pressure. The result is wave interference. One of the classic cases of this interference was the Tacoma Narrows Bridge, where the wind built up a vibration in the bridge and the waves added together growing stronger and stronger until the bridge collapsed. Sound waves can add in the same fashion. They can also partially add or cancel.

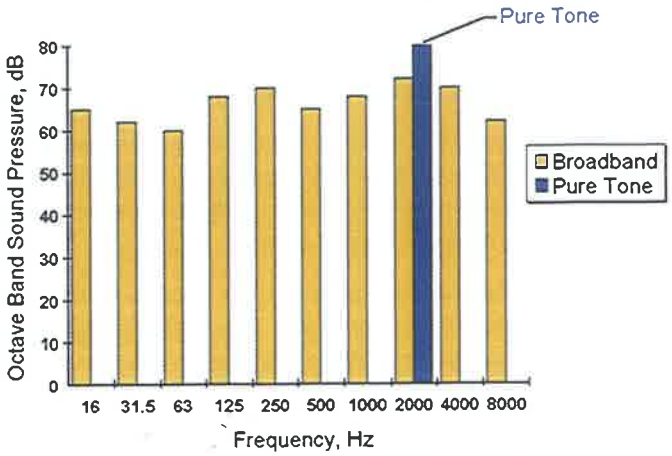


Figure 11

Pure Tone and Broad Band Sound

Sound waves with only one frequency are called ***pure tones***. When a sound source produces waves over many frequencies, as most do, it is referred to as broadband. Pure tone sound waves can combine to produce standing waves or if they differ in frequency only slightly, they produce beats.

Standing waves (space interference) occur when sound waves of the same frequency are combined. Depending on the relative phase (point in the cycle) of the waves, they will partially or totally add or subtract. When two sound waves at slightly different frequencies can be combined at a given point in time they will produce beats.

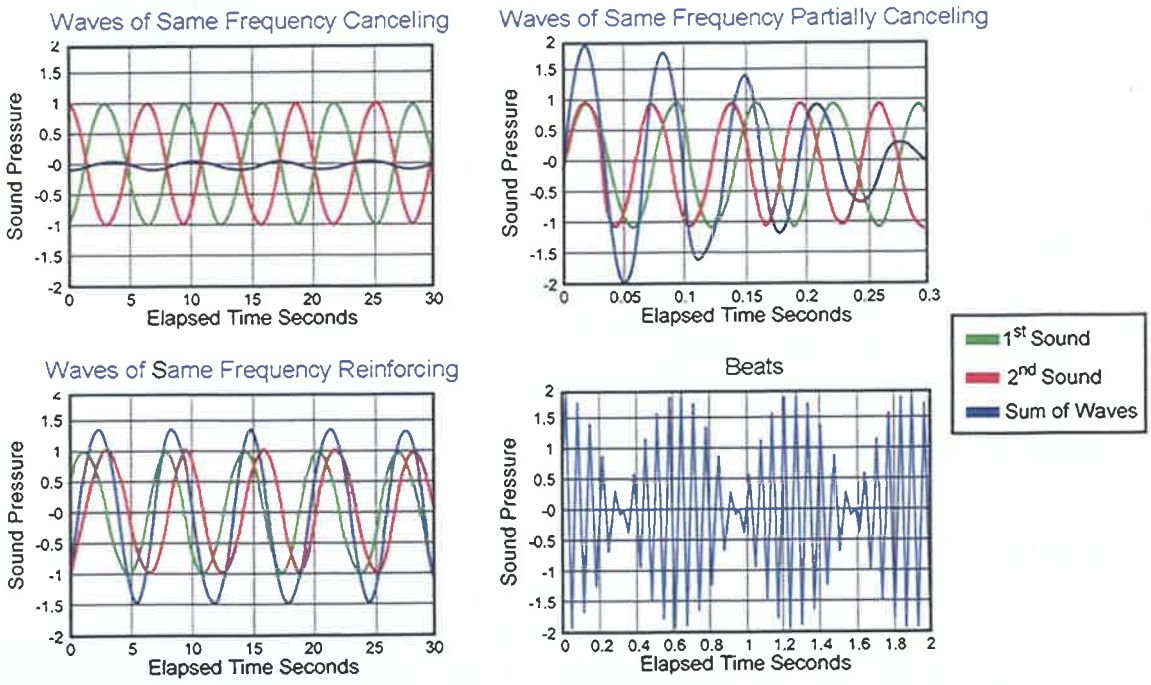


Figure 12

Standing Waves and Beats

Two pure tone sources, at relatively similar levels, and at the same frequency when radiating sound to the same point in space can be combined constructively or destructively depending on the relative phase of the resulting sound pressure levels from each source. In hard to moderately hard rooms, having surfaces that are reflective, this results in noisy (constructive) zones and quiet (destructive) zones. These zones are typically 0.4 wavelengths apart.

If two tones of slightly different frequencies radiate sound at relatively similar levels, then beats result. The resulting beat frequency is the difference in the two frequencies. For example, two tones at 60 Hz and 59.6 will produce a slow beat repeating at a frequency of 0.4 Hz, which means that the beat will repeat every 2.5 seconds.

For beat frequencies greater than approximately 10 Hz, we begin to perceive a discordant sound rather than a regular ***beat***. This happens because the hearing sense no longer follows the loudness of the envelope of the beating signals.

Sound Power and Sound Pressure

So far we have discussed sound based on its pressure wave. The intensity of the sound pressure wave decreases as the distance increases, just like it does with the rock thrown into the water. Obstacles in the wave's path and energy reflected from other sources increases or decreases the wave's amplitude. Sound pressure is important at the place the sound is received, but to evaluate sound we would like to know the power of the sound wave generated at the source.

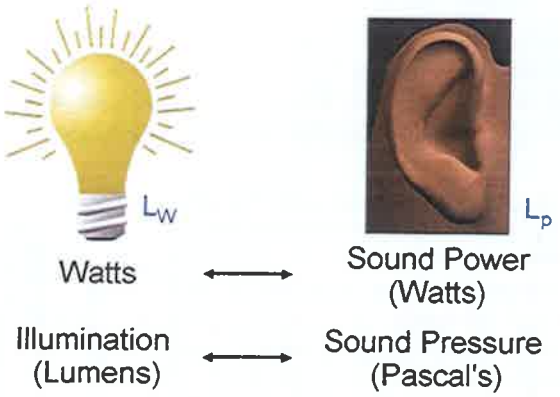


Figure 13

Sound Pressure Verses Sound Power

A good way to visualize the difference is with a light bulb. As you move away from a lamp, the lighting is less intense. The light bulb is still emitting the same light power, but its apparent illumination level (sound pressure) is reduced due to the increased distance and the fact that the room furnishings absorb some of the light. The same occurs with ***sound pressure***. As you move away from the source of the sound, it seems to be less intense. However, like the light bulb, it continues to produce the same amount of energy no matter how far we move from the lamp.

Sound Pressure (Pascal's - Lp)

Sound pressure, measured in dB, is what the ear (receiver) responds to, and it varies in relationship to the distance between the source and the receiver, the environment, and the presence of existing sounds (background noise). Sound pressure values without reference to both the distance from the source and the frequency range are essentially meaningless.

A sound pressure (Lp) level in the space at any location may be calculated from a known point source sound power (Lw) level data by knowing the acoustic characteristics of the environment surrounding the source. This will be discussed in more detail later.

This includes the effects of nearby reflecting surfaces, receiver distance, the amount of sound-absorptive material in the space, atmospheric effects, the presence of barriers, and the presence of other ambient sounds.

Typical sound pressure levels are shown in Figure 14. Note that sound pressure levels in the table are referenced to a distance from the sound source.

Typical Sound Pressure Levels	Sound Pressure Pa	Sound Pressure Level dB	Subjective Reaction
Military Jet Takeoff @ 100 ft	200.0	140	Extreme Danger
Artillery Fire @ 10 ft	63.2	130	
Passenger Ramp at Airport	20.0	120	Threshold of Pain
Loud Rock Band	6.3	110	Threshold of Discomfort
Platform of Subway Station	2.0	100	
Large Diesel Engine @ 130 ft	0.6	90	Very Loud
Computer Printout Room	0.2	80	
Freight Train @ 100 ft	0.06	70	
Conversation Speech @ 3 ft	0.02	60	
Window Air Conditioner	0.006	50	Moderate
Quit Residential Area	0.002	40	
Whisper @ 6 ft	0.0006	30	
Buzzing Insect @ 3 ft	0.0002	20	
Threshold of Good Hearing	0.00006	10	Faint
Threshold of Youthful Hearing	0.00002	0	Threshold of Hearing

Figure 14

Typical Sound Pressure Levels

Sound Power (Watts – Lw)

Sound power, often abbreviated Lw, is a measure of the total acoustic energy emitted at the source. For all practical purposes, it does not depend on the acoustical environment in which it is located. It is completely independent of the environment in which it is located. Sound power data is usually published and available from equipment manufacturers. It is derived from testing methods as outlined by the Air Conditioning & Refrigeration Institute (ARI). If the equipment sound power data and the acoustic characteristics (attenuating effects) of the environment are known, it is possible to estimate the sound pressure level in the space.

Sound power levels cannot be measured directly, they are calculated from sound pressure measurements.

Source	Watts	Watts – Exponential	Decibel re: 10^{12} W
Saturn Rocket	100,000,000	10^{8}	200
Turbojet Engine	100,000	10^{5}	170
Jet Aircraft at Takeoff	10,000	10^{4}	160
Turboprop at Takeoff	1,000	10^{3}	150
Prop Aircraft at Takeoff	100	10^{2}	140
Large Pipe Organ	10	10^{1}	130
Small Aircraft Engine	1	10^{0}	120
Blaring Radio	0.1	10^{-1}	110
Automobile at High Speed	0.001	10^{-2}	100
Voice, Shouting	0.0001	10^{-3}	90
Garbage Disposal Unit	0.00001	10^{-4}	80
Voice Conversation Level	0.000001	10^{-5}	70
Ventilation Fan	0.0000001	10^{-6}	60
Office Air Diffuser	0.00000001	10^{-7}	50
Small Electric Clock	0.000000001	10^{-8}	40
Voice, Soft Whisper	0.0000000001	10^{-9}	30
Rustling Leaves	0.00000000001	10^{-10}	20
Human Breath	0.000000000001	10^{-11}	10
Threshold of Hearing	0.0000000000001	10^{-12}	0

Figure 15

Typical Sound Power Levels

Sound power levels for some typical sound sources are shown in Figure 15. The reference level for sound power is 10^{-12} W (acoustic watts). Note this reference level is zero decibels, and is related to the threshold of human hearing.

Sound power is:

the intensity of the sound at the source and is not subject to the distance to the receiver.

The reason sound power is so important is because it is essentially independent of the environment in which it is tested, except at very low frequencies. This means that if one determines the sound power level for a given source, the level will be the same no matter where it is measured. Thus, sound power

of equipment can be easily compared. The properties of construction material and components can also be evaluated for their ability to absorb or transmit sound.

Sound pressure is dependent on the acoustical environment. The factors involved include the effects of reflecting surfaces, receiver distance, type of space, the amount, type and location of sound absorbing materials, the location in the space, the presence of barriers, and the intrusion of other ambient sound sources.

Sound pressure is:

the pressure of the sound wave at the receiver and is a function of the distance to the receiver.

Sound power and sound pressure are also different in that sound power, which is a source property, is a measure of total energy per unit time emitted by the source in all directions. Sound pressure is a measure of the sound pressure at the receiver's location.

Sometimes ***sound intensity*** (I) is used, rather than sound power. Sound intensity is sound energy per unit time. This TDP module will not use sound intensity.

Decibels

As was discussed previously, the range of sound pressure levels varies by one million times from our threshold of hearing, 0.0002 Pascal, to the threshold of pain at 1 Pascal. Because of this large range, a logarithmic scale was chosen to compress the decibel numbers we use most often into the range of 0 dB to 100 dB. The same is true of sound power and decibels are used in sound power calculations.

In acoustics, ***decibels*** (abbreviated dB) are the primary units of sound measurement and are used to measure both sound pressure and sound power. Decibels are dimensionless units used to express logarithmically the ratio of a particular sound level to a reference level. The equations used to calculate decibels is shown in Figure 16.

The sound pressure reference level is 10^{-12} Pascals. For sound power, the reference level is 10^{-12} W (acoustic watts).

- Primary unit of sound measurement
- Measures both sound pressure and sound power

Sound Power

$$dB_{Lw} = 10 \log_{10} \frac{\text{Sound level (watts)}}{\text{Reference level } (10^{-12} \text{ watts})}$$

Sound Pressure

$$dB_{Lp} = 10 \log_{10} \frac{\text{Sound level (pascal)}}{\text{Reference level } (10^{-12} \text{ pascal})}$$

Figure 16

The Decibel System for Acoustics

Combining Sound Levels

When multiple sound sources are present, a designer or installer must be able to analyze the effects of these sounds both individually and together. An example, of this might be multiple types of equipment, such as an air-handling unit and a pump, both operating in a mechanical equipment room.

The decibel scale is very useful, but mathematical operations with decibels can be confusing if you are not used to using them. This section discusses how to deal with decibel numbers and to add or subtract them.

Converting from Sound Pressure to Sound Power

Converting from sound pressure in Pascals to sound pressure levels in decibels is accomplished by multiplying by 20 times the log to the base 10 of the ratio of the sound pressure of interest (p_1) to the lowest sound pressure we can hear (p_{ref}).

Pascal to Decibel conversion:

$$L_p = 20 * Log_{10}\left(\frac{p_1}{p_{ref}}\right)$$

Where:

L_p = *sound pressure level (dB)*

p_1 = *Sound pressure level (Pascals)*

p_{ref} = *reference sound pressure level (0.0002 Pascals)*

Adding Decibels

If we have two sound sources, each with a known sound level, that are not inherently related to and interdependent such as two separate fans, the levels are called incoherent levels. Sounds from two sources which are related to each other, such as the ***pure tones*** radiated by a screw chiller from the compressor case and the condenser shell, are called coherent levels.

Coherent and incoherent sound waves

Coherent sounds occur when two sounds have identical pure tone, for example the blade passage frequency of two fans on the same shaft.

Incoherent sound occurs when the sound frequencies are not synchronized, this is the most typical situation in HVAC situations.

Adding Coherent Sound Sources

The combined effect of multiple coherent sound sources are difficult to analyze and require the use of a complex equations which consider the phase angle between the waveforms. Two coherent sound sources can combine to completely or nearly cancel each other (minus more than 10 dB; or the sounds can add completely, in which case the pressure doubles (plus 6 dB). This phenomenon is completely dependent on the relative phase angle between the two waves. Multiple waves obey the same relationship but the resultant standing wave patterns can be very complicated. Thus, regions are often experienced where the tone cannot be perceived (cancellation) and regions where the sound is quite perceptible (reinforcing regions).

The formula for combining two coherent sound sources is:

$$P_T = \left(P_1^{\,2} + P_2^{\,2} + 2 * P_1 * P_2 * \mathrm{Cos}\left(\theta_1 - \theta_2\right)\right)^{0.05}$$

Where:

P_T = total RMS (root mean squared) pressure

P_1 = RMS pressure of the first sound

P_2 = RMS pressure of the second sound

Φ_1- Φ_2 = the phase angle between the two

Fortunately, most HVAC noise is incoherent noise and may be determined by relatively simple logarithmic addition.

Adding Incoherent Sources

When two incoherent random sound sources both at the same location and producing 40dB are combined they total 43 dB, which is barely perceptible, and not 80 dB. It would require four units operating at this same level to add 6 dB at which point the increase in sound would be noticed as a significant increase.

If we have two sound sources, each with a known sound pressure level which are not synchronized (incoherent), such as an air-handling unit and a VAV terminal, we can determine the sound level of both sources operating together by combining them as indicated below.

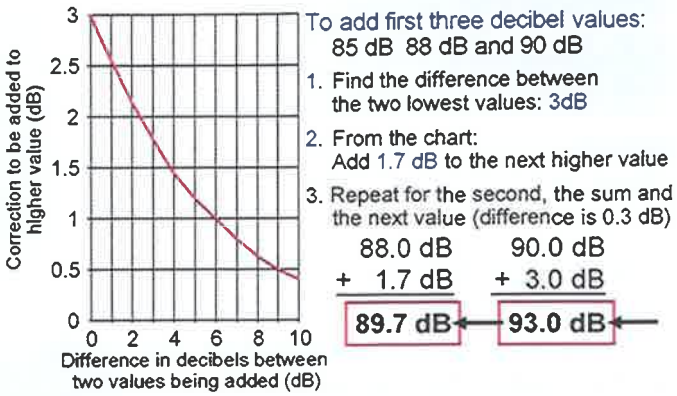


Figure 17

Decibel Addition Example

Since decibels are logarithmic, they are added together using logarithmic addition. The formula that can be used to add incoherent sound sources is:

$$Lp_T = 10 * \left(\log_{10} * 10^{(LP_1/10)} + 10^{(Lp_2/10)}\right)$$

Where:

Lp_T = total sound pressure level, dB

Lp_1 = first sound pressure level, dB

Lp_2 = second sound pressure level, dB

Lp_N = N^{th} sound pressure level, dB

Example:

Three sound sources Lp1 = 90 dB, Lp2= 88 dB and Lp3= 85 dB

$$Lp_T = 10 * \left(\log_{10} (10^{(9.0)} + 10^{(8.8)} + 10^{(8.5)}\right) = 93 \text{ dB}$$

Rather than using complicated formulas, it is easier to use a decibel addition chart such as the one shown in Figure 17 to add multiple sound sources.

Notice each value needs to be added systematically to the next value using the sum of the previous addition.

From the chart, the correct answer is 93 dB.

Subtracting Incoherent Sources

Likewise, decibel subtraction is logarithmic. It is often used to determine the sound level of a sound source by subtracting the background or ambient sound from the total sound. The formula for logarithmic subtraction is:

$$Lp_2 = 10 * (\log_{10} (10^{(Lp_T/10)} - 10^{(Lp_2/10)})$$

Where:

Lp_T = total sound pressure level, dB

Lp_1 = first sound pressure level, dB

Lp_2 = Second sound pressure level, dB

Do not subtract differences less than 3 dB because they cannot be heard.

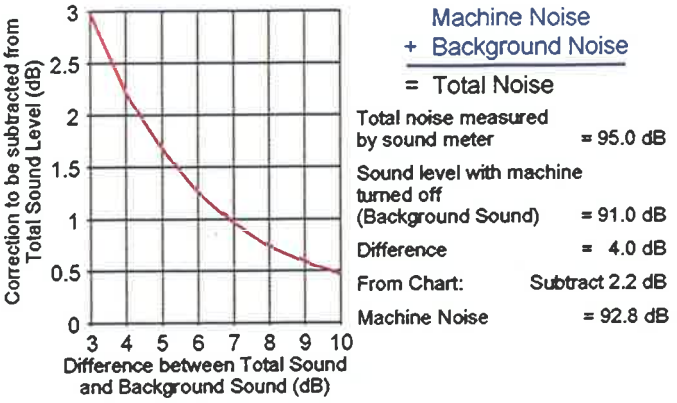


Figure 18

Decibel Subtraction Example

Example:

If we have a machine operating in an equipment room with a total noise level (machine noise + background noise) of 95 dB on a sound meter and 91 dB with the machine turned off, using decibel subtraction, the machine sound alone can be determined as 92.8 dB

$$Lp_2 = 10 * (\log_{10} (10^{(9.5)} - 10^{(9.1)}) = 92.8\,dB$$

For decibel subtraction, a similar chart has been developed that eliminates the need to perform complex arithmetic. To subtract decibels, simply use the chart in Figure 18.

Effects of Multiple Identical Sources

When multiple identical sources are located at the same or nearly the same distance from the receiver, the formula below can be used to estimate the total sound. This is a common situation in HVAC. For example, when the effect of multiple condensing fans, compressors, or identical units is to be considered.

$$Lp_N = Lp_1 + 10 * Log_{10}(N)$$

Where:

Lp_N = Sound pressure total for N machines

Lp_1 = Sound pressure total for 1 machine

N = Number of machines

To add multiple sound sources use the table

Difference in dB	dB+
• 0 to 1	3
• 2 to 4	2
• 5 to 9	1
• 10 or more	0

Example:

- 3 sound sources at 45 dB
- 45 dB + 45 dB = 45 + 3 = 48 dB
- 48 dB + 45 dB = 48 + 2 = 50 dB

Figure 19

Decibel Addition Example

Example:

Three fans all produce a sound reading of 45 dBA, the total can be calculated by:

$$Lp_N = 45 + 10 * Log_{10}(3) = 49.8 \approx 50dBA$$

The correct value is 50 dBA, which may also be determined from the chart.

Again, a table can be used as shown in Figure 19 to determine the combined sound level.

Calculating Log Average

Sometimes, it is useful to calculate the log average of decibel readings. For example, when the sound pressure level around a chiller is measured at many locations, the average level may be considered the most representative. The equation below illustrates how to calculate log average for dB readings.

$$Lp_{avg} = 10 * Log_{10}(10^{Lp1/10} + 10^{Lp2/10} + ... + 10^{Lpn/10}) - 10Log10N$$

Where:

Lp_{avg} = average sound pressure level, dB
Lp_1 = First sound pressure level, dB
Lp_2 = Second sound pressure level, dB
Lp_n = N^{th} sound pressure level, dB
N = Number of measurement point

Example:

The following readings were taken in a mechanical room:

Lp1 = 90 dB, Lp2 = 87 dB, Lp3 = 87 dB, Lp4 = 88 dB, Lp5 = 89 dB

$$Lp_{avg} = 10 * Log_{10}(10^9 + 10^{8.7} + 10^{8.7} + 10^{8.8} + 10^{8.9}) - 10 * Log_{10}5$$

$$Lp_{avg} = 88.4 \text{ dB}$$

Response to Changes in Sound Level

How do we react to changes in sound pressure levels? It depends on two things, the amount of change in dB and the frequency of the sound. The table in Figure 20 is a good guide as to how people respond to changes in sound pressure levels

CHANGES IN SOUND PRESSURE LEVEL

For High Frequencies (Above 125 Hz Octave)

1 dB	Not Noticeable
3 dB	Just Noticeable
5 dB	Noticeable
10 dB	Doubling / Halving
20 dB	Much Louder / Quieter

For Low Frequencies
(<125 Hz Octave ... Rumble Region)

3 dB	Noticeable
5 dB	Doubling / Halving
10 dB	Much Louder / Quieter

Figure 20

Response to Sound Level Changes

Source – Path – Receiver Concept

Sounds that radiate from a ***source*** travel through a ***path*** (air) to a ***receiver*** (ear) where they are transmitted to the brain via nerve endings in the inner ear. The larger the pressure fluctuations, the louder the sound is perceived. This is commonly referred to as the ***source-path-receiver*** concept. This concept will be used to evaluate design options for sound control and remedies for noise problems. The table provides a description of each element and the source of design information.

Source	Path	Receiver
Sound sources produce sound power	The sound path has elements that attenuate the sound power transmitted to the space	The sound power that arrives at the receivers space produces a resulting sound pressure level at the receivers ear which is then heard
Sound power data for HVAC products is determined in sound test chambers per ARI and ANSI Sound standards	The attenuation for sound path elements can be estimated from ASHRAE and ARI publications to determine the sound pressure	The sound pressure level at the receivers ear can be analyzed to determine if the design criteria has been met

Calculating Sound Power from Sound Pressure

To understand the sound pressure produced by a source of sound power in a space, it is helpful to understand how the sound field varies depending on the distance from the source and the acoustical environment.

Sound pressure will decrease by 6 dB in a free field (no nearby reflecting surfaces) when the distance is doubled.

Sound Fields

Consider the analogy between sound pressure and a light bulb. The 100-Watt light bulb (sound power) appears very bright when you are near it. If you move away from the light, the intensity is greatly reduced. The intensity drops off following a relationship known as the inverse square law.

According to the inverse square law, sound intensity is inversely proportional to the square of the distance from the source. Likewise, sound in a ***free-field*** (no nearby significant reflecting surfaces), will result in a 6 dB decrease when you double the distance between the source and the receiver. For example, let us assume in Figure 21 that you are in a free-field at a distance of d_1 = 10 feet from a sound source measured at sound pressure level Lp_1 = 80 dB, and move

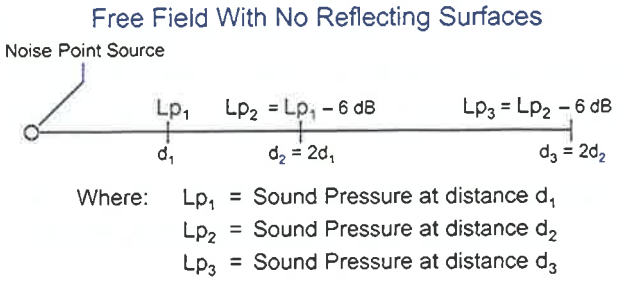


Figure 21

Inverse Square Law

10 more feet away to a distance d_2 = 20 feet. The sound intensity will be decreased by 6 dB, to a value of Lp_2 = 74 dB. To reduce the perceived sound level by another 6 dB, to Lp_3 = 68 dB, you would have to double your distance again, to d_3 = 40 feet from the source.

Six terms are important to acoustical testing and ratings. These are types of sound fields and are shown in Figure 22.

The first is ***near field;*** in this region close to the sound source, the sound pressure fluctuates dramatically. The sound waves arrive at the measuring microphone from different points out of phase, and add constructively and destructively. Fluctuations by as much as 10 dB can be experienced. The size of this region is a function of the size of the unit being tested, the location of the microphone, and the wavelength. This region is normally defined as ¼ of the distance of the longest wavelength. Near field can apply to indoor or outdoor readings.

The ***far field*** is the region located just after the near field and farther from the sound source the sound pressure wave is diminishing as a function of distance. In this region, the sound waves arrive from different points on the unit and are in phase. Sound pressure levels in this region tend to follow the inverse square law and drop off at a rate of 6dB for every doubling of distance.

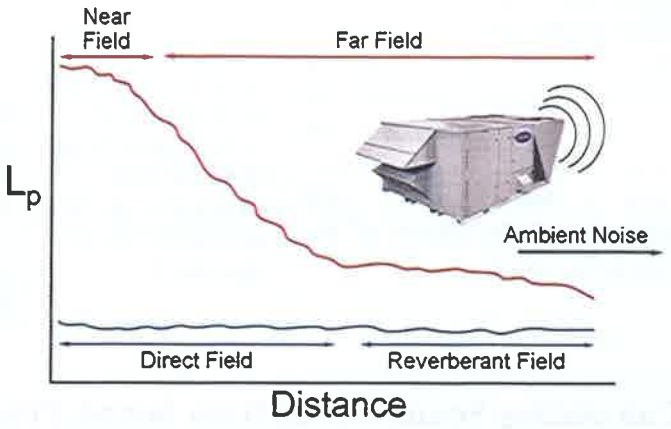


Figure 22

Sound Fields

The ***direct field*** is a region in which most of the sound waves that reach the microphone travel from the unit in a direct line with little reflected wave impacts. This region is much the same as the far field and sound pressure follows the inverse square law.

A ***free field*** is a field that is free of any significant reflecting surfaces that would reflect the sound energy and distort the sound field. Sound in this field would follow the inverse square law with a drop of 6db for each doubling of distance. This type of field is what one might expect to find outdoors, in very large rooms, or in rooms with highly absorptive surfaces. In many cases the sound source is mounted on a floor, roof, or hung from a ceiling. In these cases, the sound energy that would normally radiate in the surface direction is reflected back and combines with the radiated sound field, this is referred to as a free field above a reflecting plane.

The ***reverberant field*** is a region in which most of the sound energy reaching the microphone has been reflected one or more times giving relatively uniform contribution from all the reflecting surfaces.

Finally, the ***ambient noise*** region is where the contribution of the unit has become insignificant compared to the background sound pressure.

Directivity

Real sound sources typically radiate different amounts of sound power in different directions. A source that radiates the same sound power in all directions is said to be non-directional or omni-directional.

The directivity of the source is described by the ***directivity factor*** **(Q)** that is a function of the frequency and the direction relative to the source. It is defined as the ratio of sound pressure at a given angle to the sound source to the free field sound pressure level.

The inherent directivity of a source is also affected by the presence of nearby reflecting surfaces. These surfaces send back some of the radiated acoustic energy in the direction from which it came. This results in higher sound pressure levels in the regions in which the direct and reflected sound pressure levels combine.

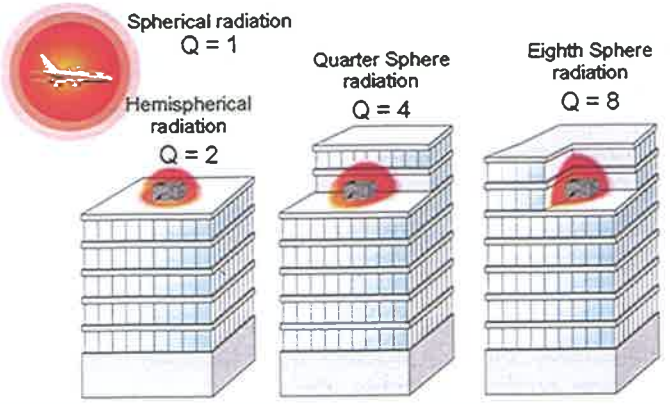


Figure 23

Directivity of Sound

The equation below can be used to calculate the sound pressure from an omnidirectional sound power source. The equation accounts for the directivity of reflecting surfaces.

$$Lp = Lw + 10 * Log_{10}\left(Q / 4\pi r^2\right) + 10.5$$

Where:

Q = Directivity factor

- Q = 0, no reflecting plans present (free field)
- Q = 2, one reflecting plane (free field above a reflecting plane)
- Q = 4, two reflecting planes (free field above a reflecting plane with a reflecting wall)
- Q = 8, three reflecting planes (free field above a reflecting plane with two reflecting walls)

R = distance from the acoustic center of the source to the location of the receiver in feet

π = 3.1415

Lw = Sound power level, dB

Lp = sound pressure level, dB

Line and Point Sources

A ***point source*** is a sound source whose largest dimension is small in relation to the wavelength it is radiating. An example would be a 3-ft diameter propeller fan that is radiating sound at 80 Hz. The wavelength for this is 13.75 feet (wavelength = 1100 speed in air / 80 Hz).

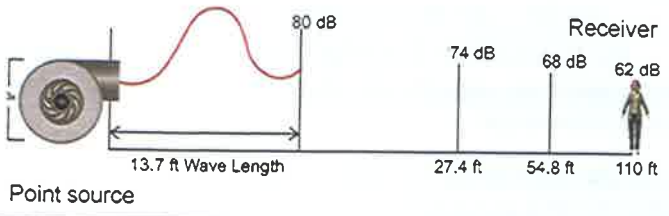


Figure 24

Point sources follow the inverse square law.

Point sources produce nearly equal sound radiation in all directions. The sound pressure levels follow the inverse square law.

A ***line source*** is a sound source composed of many point sources arranged next to each other in a line.

Line sources produce equal sound pressure at equal distances perpendicular to the axis of the line source, out to a distance approximately L/π, where L is the length of the line source. This happens because, initially, the waves are cylindrical and the energy spreads in only one direction rather than in two directions, as with a point source. At distances greater than L/π, a line source begins to behave like a point source. At this distance, the waves are more like spheres than cylinders.

In the region where cylindrical spreading takes place, sound pressure drops off 3 dB each time the distance is doubled.

Figure 25 illustrates that the falloff of sound pressure level from point, area, and line sources.

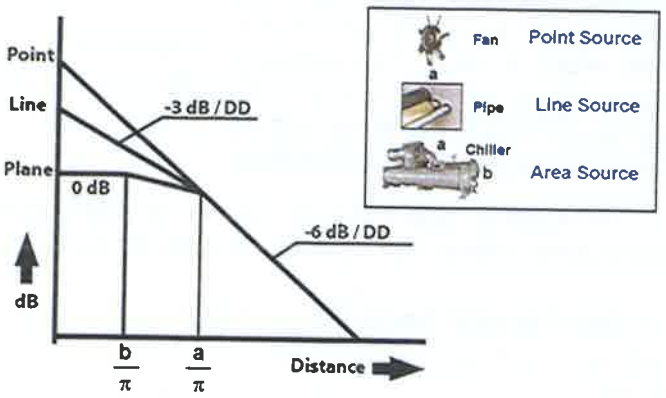


Figure 25

Line sources become point sources at sufficient distance.

Transmission of Sound Energy

Sound waves travel through a fluid (air) as a wave with the properties described thus far. The acoustic energy which arrives at the surface will either be absorbed, transmitted, reflected, diffracted, or refracted depending on the wavelength and characteristics of the surface.

Noise may reach the listener by several paths including airborne and structure-borne paths. This is illustrated in Figure 26.

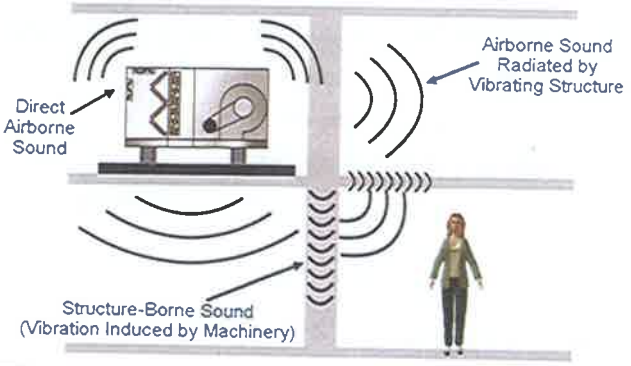


Figure 26

Transmission Paths

Airborne noise is heard in the area of the sound source and transmission generally occurs through walls to rooms directly adjacent. Structure-borne noise is generally due to vibration of mechanical equipment or components. This vibration energy travels throughout the building structure and the vibrating surfaces radiate sound, which may be heard at great distances from the sound source location.

As is indicated in the figure, the path sound energy can take between the source and the receiver could be directly radiated as airborne sound, reradiated by a surface after it has been excited by direct airborne energy or transmitted through the building as structure borne energy. Each of these paths and the energy transmitted needs to be understood to properly analyze an acoustic design or troubleshoot a problem. The actual amount of energy transmitted is a function of:

- Source surfaces
- Sound wave frequency and amplitude
- Reflecting surfaces
- Ducts in the path
- Barriers
- Room characteristics

For example, if we are trying to absorb low frequency sound, the absorber must be thick. If the issue is high frequency it may be very thin. Likewise, consider the hard surface of a filing cabinet. Depending on the frequency, the cabinet may act as either a reflector (high frequency) or as a diffractor of sound (low frequency). High frequency sounds tend to be reflected because they are short waves, while low frequency waves tend to be diffracted, or scattered.

Noise Reduction (NR), Insertion Loss (IL), and Transmission Loss (TL)

Three terms are associated with the properties of surfaces to describe their ability to reduce sound levels.

Noise reduction (NR) is in dB and is the difference in the sound pressure levels in front of and behind a sound source. The NR is used to estimate the sound pressure difference when a sound barrier is used to reduce sound transmission. For example, the sound transferred from a mechanical equipment room to the adjacent space.

Insertion loss (IL) is in dB and is the difference in sound pressure levels at a location with and without a sound control device.

Insertion loss is used to estimate the reduction in sound pressure levels produced by adding components like lined duct and sound attenuators. Since the reduction in sound pressure levels by duct elements equals the reduction in sound power by the element, insertion losses are often used to rate the reduction in sound power levels at a given point in the duct.

Transmission loss (TL) is in dB and is a measure of the reduction in sound power due to the ***absorption*** of sound waves as it passes through a wall or panel.

Noise reduction (NR)

is how many dB a sound control surface will reduce the sound pressure as it passes through the surface.

Insertion loss (IL)

is the difference in dB at a point with and without the sound control device.

Transmission loss (TL)

is a measure of the reduction in sound power due to absorption.

Reflection

Just like with a mirror, when an acoustic wave strikes a surface its reflected wave will be at an angle equal to the wave and the reflecting surface. This will only occur if the smallest dimension of the reflecting surface is significantly greater than the wavelength of the incident sound.

For example, consider a 3-ft by 5-ft surface reflecting a 2000 Hz (wavelength of 0.55 ft). Since the ***reflection*** surface is large in relation to the wavelength the wave, the sound wave will be reflected at an angle of reflection equal to the angle of incidence, as shown in Figure 27.

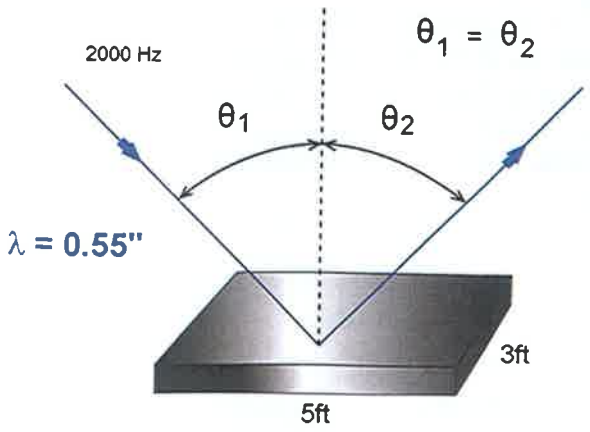


Figure 27

Sound Wave Reflection

Sound that is reflected often strikes the room surfaces many times before it reaches the receiver. Reflections are a function of the shape of the room and the sound absorption of the surfaces. Parallel surfaces can create a fluttering sound between the surfaces. Some room shapes tend to aggravate acoustical problems. For example domes and concave surfaces tend to cause the sound to be focused rather that dispersed. Absorptive materials can be used to help eliminate ***reverberation*** and reflection.

Diffraction

If the wavelength is long as compared to the surface under consideration the sound wave will be diffracted. For example, a 100 Hz wave is 11 feet long and if it were to encounter a 3-ft by 5-ft surface the sound wave would be diffracted. A diffracted wave does not follow the laws of reflection, rather the sound energy is scattered in many directions and may even go around corners. If the surface is smaller than the wavelength then diffraction dominates.

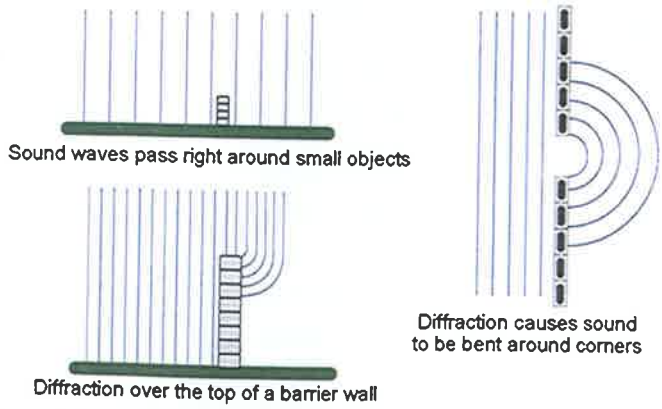


Figure 28

Sound Wave Diffraction

Sound waves at low frequency will pass around small surfaces as shown in Figure 28. If the object is smaller that the wavelength, then the sound wave will diffract around it. As shown in the figure, ***diffraction*** may cause the wave to bend over the top of the object due to wall diffraction. The sound waves do not always travel in a straight line but can diffract around corners such as will occur with holes in a mechanical equipment room wall.

Refraction

If a sound wave, either reflected or diffracted, enters a region where velocity of sound in the medium changes, then a different propagation velocity causes the wave to bend into the medium with a slower velocity. This is called ***refraction***. This is one of the reasons fiberglass ductwork works so well in reducing sound. Because the speed of sound in fiberglass is slower than in the air, sound energy propagated down the duct is turned into the absorbing liner.

Temperature changes in the atmosphere can also cause the sound wave to bend into a direction of lower speed sound. These effects often occur outdoors due to wind or air stratification, making it easier or harder to hear outdoor sound sources depending on atmospheric conditions.

Conservation of Acoustic Energy

Assume that a portion of a sound wave has one unit of acoustic energy before it strikes a wall of a large building that has an absorptive covering of fiberglass. Part of the incident sound energy is absorbed (α) by the wall and turned into heat energy. Sound waves are ordered vibrations of the air molecules, but when sound waves are absorbed, they are turned into random vibrations in the medium's molecules, which is heat. The sound energy that is not absorbed is either transmitted or reflected by the wall.

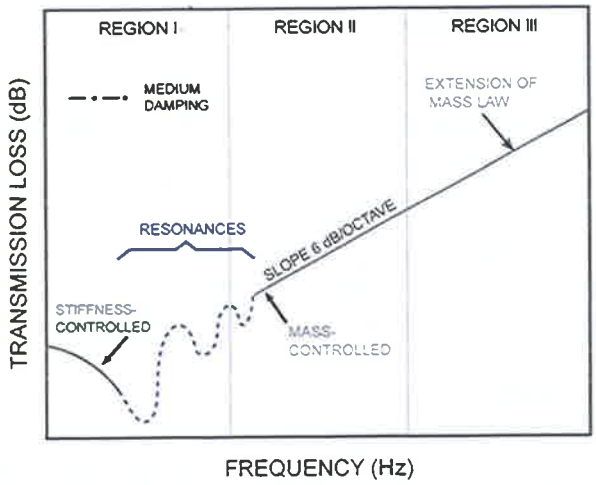


Figure 29

Typical Panel Transmission Loss

Sound energy of an incident wave can be divided into its component parts.

- Part of the incident energy is reflected (r)
- Part of the energy is absorbed (α)
- The rest is transmitted (t)

This is expressed: $1 = \alpha + r + t$

When the acoustic wave hits a building wall that does not have absorptive material, most of the energy of the incident wave is reflected or diffracted by the wall. The ability of the wall to prevent sound transmission to the adjacent space is rated in terms of ***transmission loss (TL)***. At very low frequencies, the stiffness of the wall governs the TL, while at most other frequencies the mass of the wall determines the transmission loss. Figure 29 illustrates the transmission loss versus frequency for a typical panel including the areas governed by stiffness, damping, and mass.

Stiffness Controlled Transmission Loss

At very low frequencies, where rumble and HVAC ***duct breakout*** occur, the transmission loss of the duct walls is proportional to the stiffness of the wall and drops as frequency increases

(-6 dB per octave band) first section of Region 1 in Figure 29. (Octave bands are explained in the next section).

As an example, it is often beneficial to increase the stiffness of rectangular duct by screwing drywall to the exterior. This will increase the transmission loss of a 20-gage duct by as much as 6dB in the critical 31.5 Hz octave band. Adding layers can increase the stiffness even more. Each time the stiffness doubles, the TL will increase 3 dB.

The Mass Law

The formula below approximates the TL for common construction materials at frequencies for which mass is the determining factor (Region 2 in Figure 29). Each time the thickness of the wall doubles, the mass of the wall doubles and the TL increases by 6dB.

$$TL = 20 * \mathrm{Log}_{10}(w * f) - 28$$

Where:

TL = nominal incident transmission loss, dB

W = surface mass, lb/ft^2

F = frequency, Hz

At high frequencies, (Region 3 of Figure 29) most panels do not follow the mass law because of the way in which sound waves can couple effectively with the bending waves of the panel.

Absorption

As was discussed earlier, when a sound wave strikes a reflecting surface, part of the wave is absorbed, part is reflected and the rest is transmitted. If the wall or the sides of the duct are covered with an absorbing material, most of the sound waves incident energy is absorbed and turned into heat.

The ability of the material to turn sound energy into heat energy is characterized by the material's ***sound absorption coefficient (α)***. The sound absorption coefficient is the ratio of acoustical power absorbed by the material to the acoustical power incident to the material. The sound absorption coefficient is a function of the type of material (porosity), the material thickness, the angle of incidence, the type of protective facing used, and the way in which the material is mounted. Some common absorbing materials are glass fiber, mineral fiber, foams, and acoustical tiles. Figure 30 illustrates how the absorption coefficient of a good acoustical absorber, such as

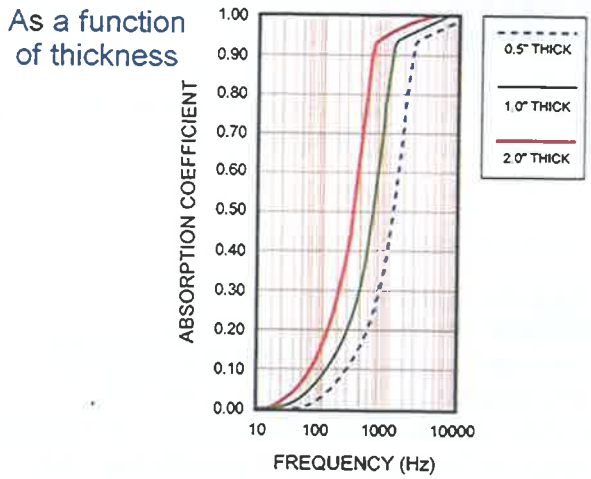


Figure 30

Absorption Coefficient for Fiberglas as a Function of Thickness

fiberglass, varies with the thickness of the fiberglass and the frequency. The curve assumes that the fiberglass is mounted over a hard reflective surface with no air gap behind it.

Figure 31 shows the relationship of absorbed and transmitted energy for an open window, acoustical absorbing material, and a hard brick wall in the 1000 Hz octave band. A large open window absorbs all the energy that impinges on it since none is reflected back into the room.



% Reflected	% Absorbed and Reflected	Sound Absorption Coefficient
0	100	1.0
20	80	0.80
98	2	0.02

Figure 31

Relationship between Absorbed, Transmitted, and Reflected Sound

Sound absorbers made of fibrous material, such as fiberglass, are most frequently used to line ductwork or absorb the incident sound from fans or air terminals. Sound absorbers are also frequently used in occupied spaces to control the reflected sound from machinery or voices. In an ordinary room, there are usually reflections from the walls, ceilings, and floors that cause some reverberation. If absorption material is added to the wall and ceiling surfaces, the reflections can be reduced to aid in speech communication and protect privacy. The resulting sound pressure level is a function of the sound power injected into the space, the amount of absorption in the space and the room volume.

Transmission loss and absorption are interrelated. The figure shows that the transmission loss of an open window is 0dB since all the incident energy is transmitted. On the other hand, we can look at this window as a perfect absorber if none of the incident energy returns to the adjacent room.

Often we do not know the amount of absorption in the room and it must be determined by test. The most common way of doing this is with the ***Sabin*** equation:

$$T60 = 0.049 * \left(V / (s * \alpha) \right)$$

Where:

T60 = ***reverberation time***, the time it takes for the sound to decay to 60d, (sec)

V = volume of the room, (ft^3)

s = surface area of the room, (ft^2)

α = average sound absorption coefficient, (Sabins)

The absorption coefficient is normally measured by placing a large sample of the material in a reverberation room and measuring the time it takes a steady state sound source to drop from its initial level by 60 dB in each frequency band of concern. If we measure T60 and the volume of the room and the surface area of the space, we can determine the average sound absorption coefficient for the space alone and for the space plus the absorbing sample. The absorption coefficient can be calculated from this information using the Sabin equation above.

End Reflection

As long as a propagating wave experiences no change in the geometry of the space in which it is propagating, the wave continues to move forward without sending any energy backwards toward the source. However, when this wave is a low frequency wave in a duct system and either the duct cross section suddenly increases as when it enters a plenum, or the duct terminates in a ceiling cavity, then a significant part of the sound energy is reflected back down the duct, leaving less energy to propagate forward from the reflecting point. This is referred to as ***end reflection***. For small ducts at frequencies below 125 Hz the end reflection is significant.

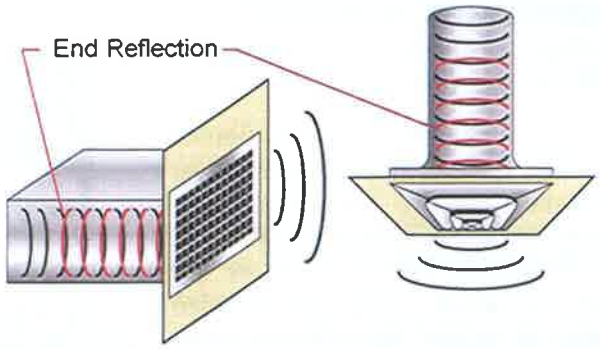


Figure 32

End Reflection of a Duct Termination

Transmission Paths

As was pointed out earlier sound travels from the sound source to the receiver by several different paths. At this point in our discussion, it may be clear that each of these paths is subject to different reflection, absorption, and transmission affects. A few additional terms related to sound paths are needed before we move on to determining sound levels in a space. Three basic types of sound paths can define the ways sound can reach the space.

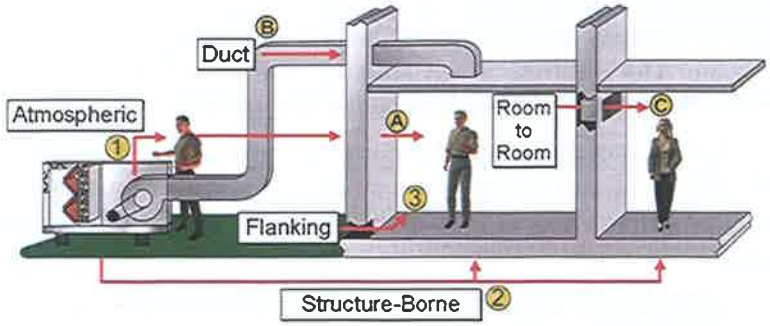


Figure 33

Transmission Paths

Airborne Transmission Paths

Airborne paths ① could be divided further into three other potential ways for sound to reach the receiver.

Atmospheric Transmission Paths

Sound travels easily through the air indoors and outdoors. This is called ***atmospheric transmission***. Indoors, the sound may be direct line of sight with the receiver and source or it may be reflected of objects and surfaces within the space Ⓐ. Detailed analysis is often required to determine the impact of the sound in indoor spaces. Outdoors, the sound tends to be free field unless large reflecting surfaces, such as a building wall, reflect part of the acoustic energy. Typically, outdoor sound follows the inverse square law.

Duct-borne transmission paths

Ductwork Ⓑ is another source of sound to indoor spaces. The ductwork can serve to transmit sound from a source like a fan or it can serve to attenuate a portion of the sound. Ductwork can also serve as a sound source when turbulent air shakes the duct, and it can serve to transmit sound from space to space, such as cross talk.

Room-to-room transmission paths

The wall surfaces between rooms Ⓒ can also serve as a sound transmission path and is a combination of airborne and structure-borne sound paths. The acoustic waves from a sound source in one room strike a surface, part of the energy is reflected, part of it is absorbed, and part of it is transmitted. The transmitted portion actually starts a sound wave in the wall material, which in turn creates an airborne sound wave in the adjacent space.

Structure-borne Transmission Paths

Sound energy may also be transmitted through the structural elements of the building ②. This is called ***structure-borne transmission***. This sound wave often results from vibration caused by rotating machinery. Normally only a small part of the sound energy is transmitted through the structure. However, it is dependent on the building mass and lightweight structures will transmit a greater amount of acoustical energy.

Flanking Transmission Paths

Flanking transmission paths ③ are the paths sound can take to travel from space to an adjoining space without being reflected or transmitted by the primary barrier. For example an open window or door, return duct openings and chases or penetrations through the wall for pipes, ducts, or electrical services. Determining flanking paths can be difficult.

Several terms are used to describe some of these sound paths:

- Discharge sound path is the sound that travels through the downstream ductwork and is delivered to the space with the conditioned air.
- Radiated sound path is sound that travels outside the air ducts or air handler casing and reaches the receiver independent of the ducts.
- Return air sound path is the path of sound that travels from the unit inlet or casing from the mechanical room or unit through the return air system to the receiver.

Sound and Noise

Noise may be defined as unwanted or undesirable sound. It may be continuous, intermittent, or transient, depending on the source. However, a more inclusive definition of noise is a sound that is a distraction and may be counterproductive in the working environment. Typical sound becomes noise when it is:

- Too loud – the ability to carry on a conversation becomes difficult or it is uncomfortable.
- Unexpected – when for example the cycling of compressors can be heard
- Uncontrolled- a sound from an adjacent building which enters your building
- Happening at the wrong time – compressors in a motel PTAC unit cycling when you are trying to fall asleep.
- Composed pure tones – a whine, whistle, or hum at one frequency
- Unwanted or distracting information – telephone conversations from adjacent offices
- Unpleasant or acoustically ugly
- A reminder of unpleasant conditions

Often it has many of these characteristics.

While loud sounds are a concern because they can cause hearing loss, sound does not have to be loud to be a problem, it just needs to be unwanted.

Source of Noise

HVAC systems have many components that produce sound that, when produced in excess or poor quality will constitute noise. The primary sources of HVAC sound are:

- Fans
- Compressors
- Pumps
- Motors
- Diffusers
- VAV fan mixing boxes
- Inverters
- Combustion
- Aerodynamically generated sound in ductwork



Figure 34

HVAC Sources of Noise

These primary sound sources can be found in a variety of product types. The potential for noise problems exists if the products are not properly selected, installed, operated, and maintained. Typical products include:

- Air terminals
- Boilers
- Central station air handlers
- Chillers
- Cooling towers
- Fan coils and unit ventilators
- Fan powered mixing boxes
- Indoor packaged units
- Water source heat pumps
- PTACs
- Rooftop units
- Split systems

Primary sources of HVAC sound from these products include sound radiated from the vibration of equipment surfaces such as equipment casings or attached ductwork; pressure fluctuations in compressors, fans, or pumps, pressure fluctuations from combustion, and vibration transmitted to the structure.

Secondary or regenerated sources include the vibration of ductwork due to air turbulence, induced by damper blades, elbows, transitions or tees or other rapid changes in ductwork geometry.

Broadband and Tonal Sound

HVAC system noise may be broadly classified as ***broadband noise***, tonal noise, or a combination of both. Broadband noise contains sound energy at many frequencies and is less annoying than similar sounds that contain dominant discrete frequencies or pure tones. Tonal noise is sound at a discrete frequency and is more annoying.

Pump whine, propeller or centrifugal compressor blade passing frequencies, screw and reciprocating compressors, fluorescent lamp ballast noise, electric motors and transformers are all examples of sources that produce pure tones that often are annoying to occupants. HVAC systems produce a combination of broadband and tonal sounds.

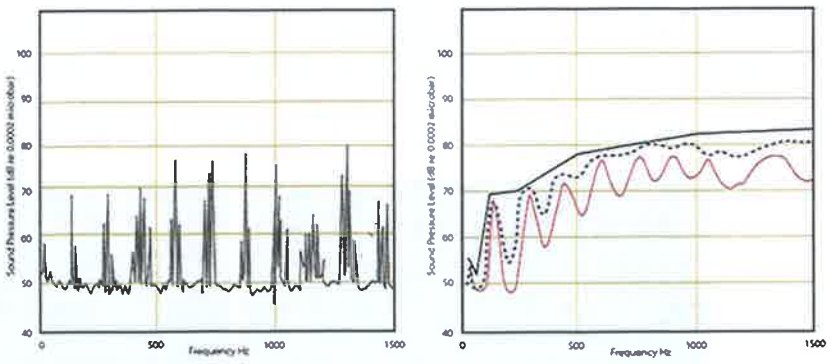


Figure 35

Broadband and Tonal Sound

Types of Noise

While noise is defined as unwanted sound, some noise actually is desirable because a lack of background noise in an office creates an environment where normal conversation may be heard at relatively long distances. You have probably experienced this phenomenon outdoors during a still, humid summer night. During those conditions, voices carry a great distance.

Random noise

is a sound that has random occurrences of both timing and frequency.

White noise

is sound with a continuous frequency spectrum with equal energy in each octave band.

Pink noise

has a continuous frequency spectrum such that the 1/3 octave band is a straight line on an NC chart.

In many offices, background noise (broadband noise) is used intentionally to mask conversations, especially in open areas where there are no walls to provide privacy. Broadband noise, has indistinguishable frequency characteristics, such as a waterfall, and is sometimes used to mask other undesirable sounds such as office machinery or conversation in adjacent areas. Depending on the desired sound levels in the space, air terminals, or diffusers may be selected to provide broadband noise, when required. ***White noise*** is defined as a noise with a continuous frequency spectrum with equal energy over a specified frequency range. White noise doubles for each successive octave band.

When sound occurs with an oscillation frequency but with random magnitude and timing, it is called ***random noise***.

Pink noise is used to define a noise with a continuous frequency spectrum but with equal energy percentage within the ***bandwidth***. Pink noise on a 1/3-octave band chart will be a straight line.

System noise levels must be evaluated and controlled to achieve a satisfactory or a quality acoustical environment. Noise transmission paths must be identified and analyzed during the design phase of a project, since remedial measures to reduce undesirable noise once the project is completed are usually expensive and in many cases only marginally effective.

Octave Bands

Actual sound is rarely a single frequency and often of various amplitudes. As indicated in Figure 35 it is a continuous spectrum of varying amplitudes. To make it easier to analyze sound, the sound is divided in to frequency bins know as ***octave bands***. The figure indicated the two used for acoustical analysis, octave bands, and ***one-third octave bands***.

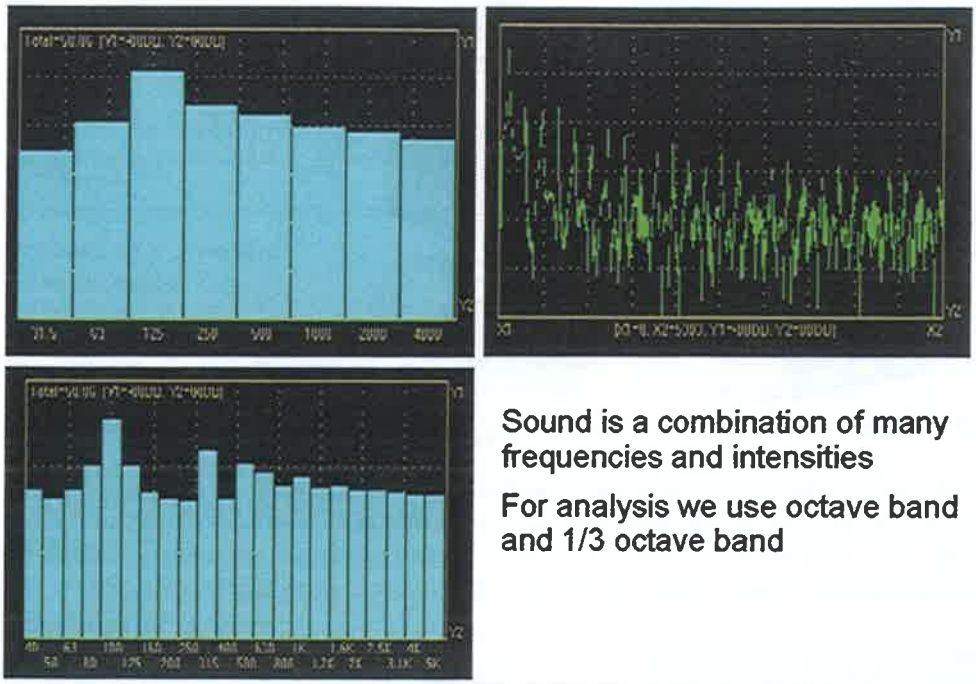


Figure 36

Sound is a combination of frequencies divided for convenience into octave bands.

The range of human hearing (audibility) is from approximately 20 Hz to 20 kHz. As a matter of convenience, the audible frequency range has been divided into 10 bands, as shown in Figure 37.

The upper band limit for each octave band is twice the lower band limit. Octave bands are identified by the center (or mid-frequency). For all practical purposes, HVAC acoustics is primarily concerned with the 31.5 Hz to 4 kHz range. Three 1/3-octave bands make up one octave band; a table of octave bands is provided in the appendix of this text.

Octave Band	Lower	Center	Upper
0	22.4	31.5	45
1	45	63	90
2	90	125	180
3	180	250	355
4	355	500	710
5	710	1,000	1,400
6	1,400	2,000	2,800
7	2,800	4,000	5,600
8	5,600	8,000	11,200
9	11,200	16,000	22,400

Figure 37

Octave bands are defined by the center frequency.

Octave bands for HVAC acoustics are the same as those in music. If you studied music, you might remember that concert pitch is the note of "A" above middle "C" on the piano, corresponding to a frequency of 440 Hz. Each octave interval either doubles or halves the frequency depending on whether or not you are increasing or decreasing the tone. In other words, the note of "A" one octave above concert pitch would be at a frequency of 880 Hz. Likewise, one octave below concert pitch would correspond to a frequency of 220 Hz.

Frequencies above 4 kHz are not normally associated with mechanical system noise. From a noise control standpoint, frequencies in the higher octave bands (above 500 Hz) are the easiest to attenuate or reduce (Figure 38). Frequencies in the lower range (16.5 Hz - 63 Hz) are the most difficult to attenuate and are generally associated with rumble and vibration of system components. In addition to being heard, noise in the lower frequency range is usually felt by the body. They can only be avoided by design.

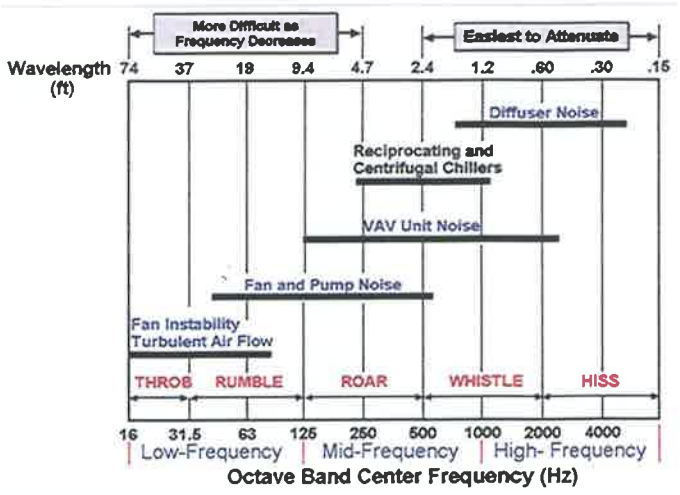


Figure 38

Typical HVAC Equipment and Octave Bands

Indoor Sound Issues

Indoor sound because of the reflecting surfaces and range of surface materials has some special terms related to evaluating indoor sound. Ductwork also presents complicated issue to be analyzed in determining room's acoustical conditions.

Reverberation

In an enclosed space, when a source of sound stops, it takes time for the sound reflections to die out and become inaudible. This is called reverberation. This is caused by the sound waves being continuously reflected off surfaces. The time it takes for sound to decay is referred to as ***reverberation time***. This affect plays a critical role in the ability to understand speech or hear music. A space with highly reflective surfaces will have a long reverberation time, as the sound level will tend to build within the space. The optimum reverberation time is different depending on the use of the space; speech and music require different reverberation times.



Figure 39

Reverberation Time

Reverberation time is the time it takes for sound to decay to 60 dB when sound is stopped. The time can vary from ½ second for a "dead" room to 5 to 10 seconds for a very "live" or reverberant room. The maximum reverberation time for clear speech is 2 seconds; over this, speech becomes exceeding difficult to understand. The ideal reverberation time for a classroom is 1 second. Ideal reverberation time for orchestral music is 1-1/2 to 2 seconds.

Space Effect

Space effect is a factor that is used to allow calculating the sound pressure level in a "real room" based on the room volume and distance to the receiver based on the sound power entering the space. Space effect equals the difference between sound power levels and sound pressure levels (Lw - Lp) and is always a positive number.

Duct Sound Issues

Breakout noise is the amount of sound that passes through the walls of the duct or through the walls of a terminal unit casing and pass into the space. Break-in noise is sound which enters a duct and is transmitted in the duct. It results from sound sources located in spaces the duct passes through. Turbulence generated by duct fittings and obstructions in the duct can result in additional sound waves.

This completes our coverage of Fundamental Terminology of Applied HVAC Acoustics. The next section will introduce you to various rating methods and design criteria. Before you continue, complete the Work Session 1. The answers can be found at the back of this book.

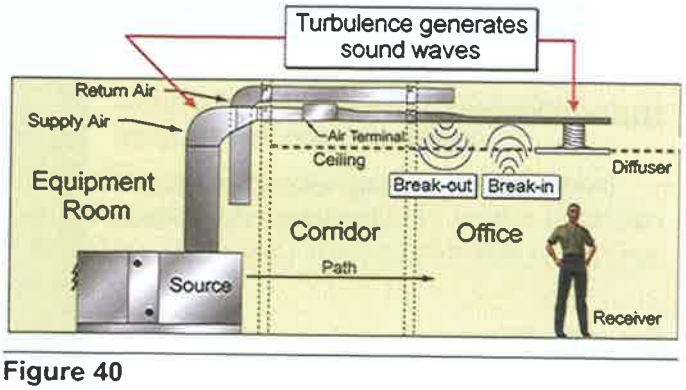


Figure 40

Duct Sound Issues

Acoustic Ratings and Methods

Sound measurements in the HVAC industry are performed for two reasons. First, to establish sound levels that are provided by manufactures for the designer to use in evaluating and estimating the acoustical contribution of equipment on a project; and second, to evaluate and troubleshoot acoustical conditions that exist with installed equipment. Different methods and equipment are used in these two situations. This section will describe how sound is measured, the ways acoustical data is presented and how manufactures data is obtained to provide sound data.

Measuring Sound

A sound level meter is used to measure sound. Sound level meters use a microphone, electronic circuits and a display to measure sound pressure levels at the measuring point. Meters may be analog display or digital. Most meters provide measurement of LIN, A, B, and C weighted networks. The more expensive meters can also provide octave band measurements with center frequencies from below 16 Hz to above 8000 Hz.

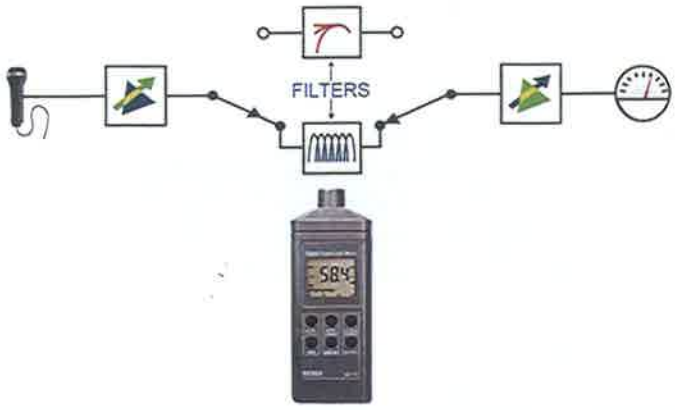


Figure 41

A sound meter measures sound using filters for different frequency ranges .

Filters

One way to analyze sound data, for both ratings and diagnosis, is to break up the sound into frequencies bands or ranges by using filters. Filters allow certain frequencies to pass un-attenuated while blocking all other frequencies. Common filters are the A-weighted filter; C-weighting filters; Octave band filters, and 1/3-octave band filters. These weighting methods will be explained shortly.

Bandwidth of Octave Band Filters

For convenience, the audible frequency spectrum is divided into octave bands, each of which is a continuous range of frequencies identified by its center frequency. The highest frequency of each octave band is twice its lowest frequency. The bandwidth of the filter is about 70 percent of its center frequency.

Bandwidth of 1/3 Octave Band Filters

To get a better definition of tones and to see details in the spectrum, each octave band can be divided into three 1/3-octave bands. The bandwidth of 1/3-octave band filters is about 23 percent of its center frequency.

Human Response to Sound

Predicting how people will respond to sound is at best a guess. Hearing perception is a very complex sense. The ability to hear certain frequencies and the perception of sound quality varies between people. The sound amplitudes and frequencies can be measured in a space or location but this is does not provide a clear indication of how people will respond to the acoustic environment. Numerous rating methods have been developed to predict how people will respond to their acoustic environment. These rating procedures all have certain advantages and disadvantages. Since noise is such a subjective issue, no one rating method is good for all applications. In the next section we will look at some of these methods. First, to understand the various methods we

need to understand that human sound perception is based on four sound related conditions, quality, loudness, tone perception and frequency spectrum.

Sound Quality

Determining if the sound in a space is acceptable is more than establishing the sound pressure level at the receiver; the quality of the sound is also important. The following factors need to be considered when determining the sound quality in the occupied space. For acoustical quality, the sound should:

- Not be too quiet, in order to preserve acoustical privacy
- Not be too loud to avoid hearing damage, speech interference, or impede learning
- Contain no pure tones that stand out above the other frequencies
- Have a balanced frequency spectrum over the range of hearing
- Not be harsh
- Be free of vibration
- Contain no significant fluctuations with time or frequency

Many of these are annoying more than damaging, like the pure tones of a hum or whine. Often, the occupant of the space will judge the sound as unacceptable if these conditions are not considered. Some of the rating methods discussed in the next section attempt to reflect the quality of the sound in the rating.

Loudness

It is very hard to determine the ***loudness level*** at which a sound will be perceived as a noise, as is evident from the level some people like to listen to music. Loudness is a subjective issue, however research has shown that the human ear is most sensitive to the mid-range frequencies.

While the ear is able to detect small changes in sound, it can take a very large change in the sound power level to make a noticeable change in the loudness. For example it takes a decrease of 10 dB for a person to perceive the sound as half the original sound. Most people will not even notice a sound pressure change of 3 dB or less. This indicates that the sound power needs to be halved just for people to notice. Figure 42 shows how people tend to perceive sound pressure levels.

EFFECTS ON HEARING PERCEPTION FOR HIGH FREQUENCIES

ABOVE 125 Hz OCT	
1 dB	Not noticeable
3 dB	Just audible
5 dB	Noticeable
10 dB	Doubling / halving
20 dB	Much louder / quieter

EFFECTS ON HEARING PERCEPTION FOR LOW FREQUENCIES

125 Hz OCT and BELOW (RUMBLE REGION)	
3 dB	Noticeable
5 dB	Doubling / halving
10 dB	Much louder / quieter

Figure 42

Subjective Response to Sound

Tone Perception

Most people are able to distinguish a single frequency tone that is above the rest of the frequency spectrum. A pure tone frequency can result in a low level sound be perceived as a noise. Some typical contributors to pure tones sounds are ***blade passage frequency*** of a fan, whine associated with pumps and turbines, and hum from electrical devices like transformers.

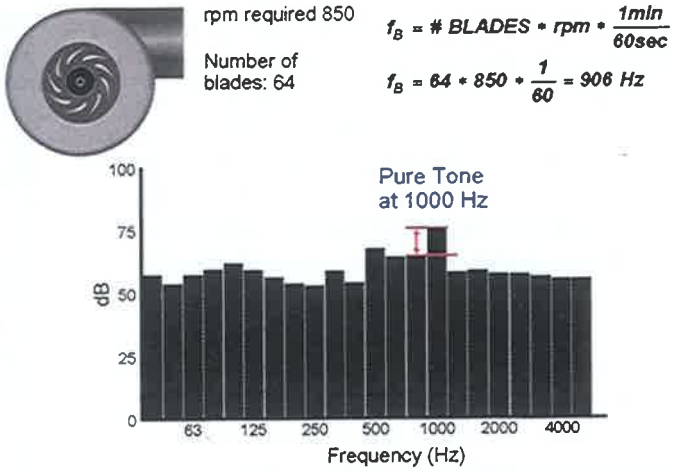


Figure 43

Pure Tone of Blade Passage Frequency

Frequency Spectrum

The other characteristic of the acoustical environment that determines how the sound is perceived is the frequency spectrum. The best frequency spectrum is a ***neutral spectrum***. A neutral spectrum does not have low or high frequency peaks. Low frequency peaks can be described as rumbly and high frequency peaks can be described as ***hissy***.

Figure 44 indicates the frequency spectrum for three different sound sources all at the same NC level and how these may be perceived. As pointed out the human ear is most sensitive to mid-frequency, approximately 2000 Hz, so some of the sound rating methods minimize the contribution of the sound pressure levels in the lower and very high frequency ranges.

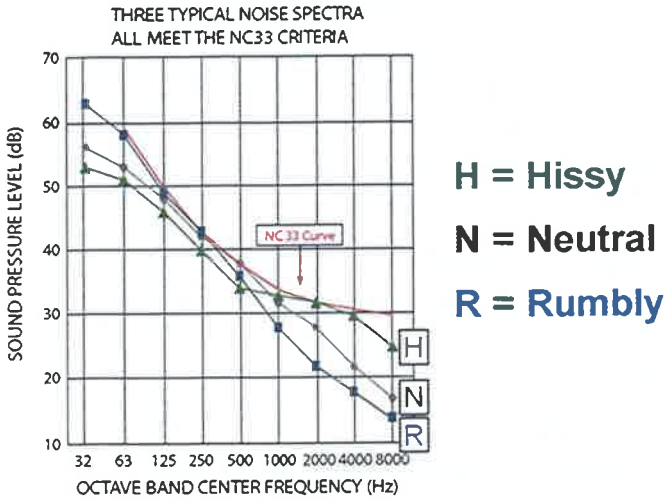


Figure 44

The same NC rating with three different frequency spectra.

Methods of Rating Sound

In HVAC situations, sound measurements and ratings may be required for one of two reasons: to estimate what the sound levels will be in a design, or to determine the magnitude and source of a noise issue on a operating system. The desired frequency spectrum could be specified for the room based on the design or on the actual room frequency spectrum from an analyzer. This information could then be used to set the design noise criteria goal or troubleshoot the problem. However, it is not always convenient to work with a frequency spectrum to specify acoustical requirements. To solve this problem many methods have been established to take these four elements of the acoustic environment and create a rating method to determine an acceptable sound level.

Methods of Rating Sound

Loudness – phons, sone and bel, used for small non-ducted ventilating fans

Weighted spectrum – A-weighted, B-weighted, C-weighted, Lin – common for outdoor sound

Noise Criteria (NC)- room sound levels

Room Criteria (RC) – room sound levels

Frequency Spectrum –manufacturer's data

Loudness Measurements (phons, sone and bel)

The human impression of the strength of a sound is its loudness. Because the ear does not hear all frequencies equally, the loudness of a sound source does not necessarily correlate with its sound level. The use of simple frequency weighting "sound level meter" alone is not an acceptable measure of determining loudness of sound sources that have a dominate sound in one small range of frequency bands. Therefore, other methods have been developed for rating loudness. Three rating methods have been used to describe a single number rating of loudness.

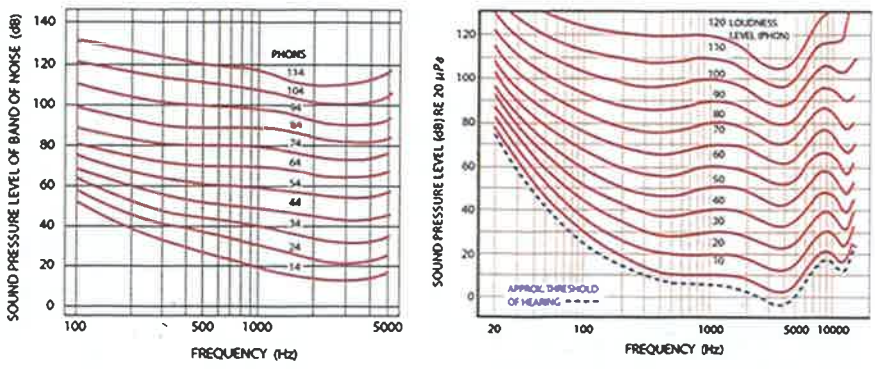


Figure 45

Loudness Rating using Phons, Sone, and Bels

The technical quantity "loudness" measures how loud or soft a listener perceives a certain sound to be. Subjective acoustic measurement rates the apparent strength of a sound. The sensitivity of the human ear varies considerably with frequency. Thus, loudness is a function of both the sound pressure and frequency of the sound. As might be expected, the loudness of any given pitch will increase as the sound intensity increases. Frequency ***weighting filters*** (A, B, and C weighting) are methods often used to provide an approximate conversion from sound intensity to loudness. However, the ear senses noise on a different basis than simple energy summation, and this can lead to discrepancy between the loudness of certain repetitive sounds and their sound level. This situation is likely to exist with equipment like fans.

The loudness of a sound may be rated using a loudness level, which has the unit "phon." The phon is based on the fact that the human ear is most sensitive to sounds at about 2000 Hz other frequency sounds are rated based on the relative response of the same sound at 1000 Hz. For example, if a sound is said to have a loudness of 40 ***phons***, this means that the listener perceives it as being just as loud as a 1000 Hz tone with a sound pressure level of 40 dB. If the actual tone were at a frequency much greater or less than 1000 Hz, the sound pressure level of the 40 phon sound tone will actually be considerably greater than 40 dB. This is because the ear is relatively insensitive to extreme bass and treble frequencies

The sone is another unit of loudness. A sone is defined as a 1000 Hz tone, 40 dB above a listener's threshold. Alternatively, 40 phon is 1 sone. The loudness of any sound that is judged by the listener to be *n* times that of the 1 sone tone is *n* ***sones***.

A 10 dB sound level increase is perceived to be about "twice as loud" in many cases. The sone is a unit of comparative loudness with:

0.5 sone = 30 phons,

1 sone = 40 phons,

2 sones = 50 phons,

4 sones = 60 phons etc.

The sone "10 dB rule" is inappropriate at very low and high sound levels where human subjective perception does not follow it. Some smaller exhaust fans are rated using sones.

The third rating is the bel. A bel is a ratio expressed logarithmically as a ratio of two acoustic power levels: 1 bel = $\log_{10} (P_1/P_2)$. $P_2 = 10^{-12}$ Watts

Bel is most commonly used with the deci prefix, that is, decibel or dB. 10 dB = 1 bel

By tradition in the acoustic community, ***bels*** are only used when expressing the sound power level of equipment in order to avoid confusion caused by reporting sound power level and sound pressure levels that are numerically similar when expressed in decibels.

Loudness level calculations take account of "***masking***." Masking is the process by which the audibility of one sound is reduced due to the presence of another at a close frequency.

A-Weighted Sound Levels

As described in the last section, single point sound ratings can be used to rate the relative loudness of a sound source. The three aforementioned methods are useful for relative rating of a single sound source. However, sound meters are useful for determining the sound pressure level in field applications. Like the first three methods, a single number rating is often helpful in measuring or estimating a sound level at some receiving point. The A-weighted (dBA) reading provides a single number measurement that can be used to determine sound pressure levels at a property line. There are four scales used on sound meters that can be used for determining this sound pressure information, A-weighted, B-weighted, C-weighted and Lin; however, the B scale is seldom used.

The ***A-weighted sound level*** is most useful when comparing the relative loudness of one acoustic environment to another similar environment. For example, in the case of diffuser noise, measuring the A-weighted sound level at various airflow deliveries could help determine if reducing the air quantity to the diffuser is an effective method of noise control. The A-weighted sound level is also used to evaluate outdoor equipment noise levels for compliance with municipal code noise limits.

The A-weighted sound level, dBA, pertains only to the relative loudness of sound; it provides no information regarding the frequency distribution of the sound, which is required to estimate sound quality and necessary noise reduction measures. A sound can be at a moderate dBA level, yet with an acoustically undesirable spectrum and therefore be judged unacceptable by an observer. Special information cannot be obtained directly from a dBA reading unless augmented by a listener's judgment, or by also measuring the Lin or C levels. Low frequencies in the "rumble" region (less than 125 Hz), are especially difficult to detect using dBA measurements. For these reasons, dBA design criteria should not be used to specify the noise control objectives of HVAC systems; sound levels specified in dBA may not be acceptable to the room occupants. In addition, noise problems should not be diagnosed with a dBA sound meter alone because a frequency spectrum shape is often required to detect and correct noise complaints.

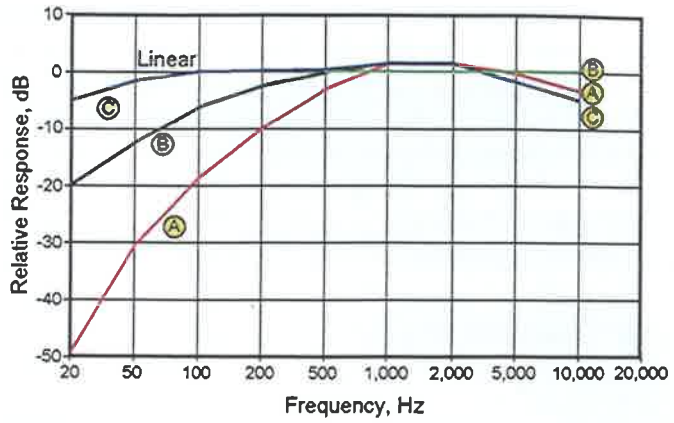


Figure 46

Filters in sound meters produce weighting curves for A, B, and C ratings.

The human ear responds to amplitude (loudness) and frequency (pitch), and is most sensitive to frequencies above 500 Hz (between 500 Hz - 4000 Hz). In other words, the ear is not as sensitive to low-frequency sounds (below 500 Hz) or very high-frequency sounds (above 4000 Hz). At 4000 Hz, a 90 dB sound seems much louder than it does at 500 Hz. See Figure 46. The A-weighted sound level was developed to approximate the human ear's response to sound at low to moderate levels, and is often used to rate HVAC noise in the areas of conversation and community noise exposure limits as well as to establish acoustical design goals. The "weighting" is achieved by subtracting (filtering) decibels from the octave band frequencies below 1000 Hz and adds some or subtracts some at higher frequencies to compensate for the human ear's lack of sensitivity in those frequency ranges. A-weighted sound level measurements are relatively easy to perform and may be measured with an inexpensive sound meter.

B-Weighted and C-Weighted Sound Levels

Most sound meters also contain ***B-weighted*** and ***C-weighted*** scales. The B and C scales are less weighted than the A scale, i.e., there is less filtering of lower frequency sounds. The B and C scales are used primarily in measuring moderate to high sound levels and when attempting to identify the presence of low-frequency sounds. This is because the loudness of the sounds measured by these scales are outside the normal hearing sensitivity range of the human ear. The C scale in particular is a relatively "flat" curve with little attenuation except in the 10 and 50 Hz frequency range.

x	Frequency (Hz)								
	31,5	63	125	250	500	1000	2000	4000	8000
dBA	-39.4	-26.2	-16.1	-8.6	-3.2	0	1.2	1	-1.1
dBB	-17	-9	-4	-1	0	0	x0	-1	-3
dBC	-3	-0.8	-0.2	0	0	0	-0.2	-0.8	-3

LIN-Weighted Sound Levels

A final type of weighting scale commonly used in HVAC system noise measurements is the *Linear weighting* scale (abbreviated LIN). ***LIN-weighting*** uses no filters to attenuate any of the octave band frequencies. Therefore, it measures the actual sound levels over the entire audible frequency range. LIN-weighted measurements are often used in conjunction with dBA measurements to analyze low frequency rumble problems. If a deviation exists between the measured dBA and the LIN sound measurements, there is the likelihood of a low frequency rumble condition.

Noise Criteria (NC)

So far, we have introduced six different single measures of sound loudness, none of which accounted for the sound quality of the frequency spectrum. When the goal is to determine the acoustical acceptability of a space by a single number rating, we need to account for the frequency spectrum; this was what led to the development of the Noise Criteria rating method.

Noise criteria (NC) ***rating curves*** were originally developed from detailed studies and engineering experience with multiple-use spaces. These "comfort" sound pressure levels take into account the intensity of sound and the frequencies in the eight octave bands. Usually the eight octave bands used are the bands from 63 Hz to 8000 Hz when analyzing potential sound problems or establishing design criteria. Figure 47 illustrates the NC curves.

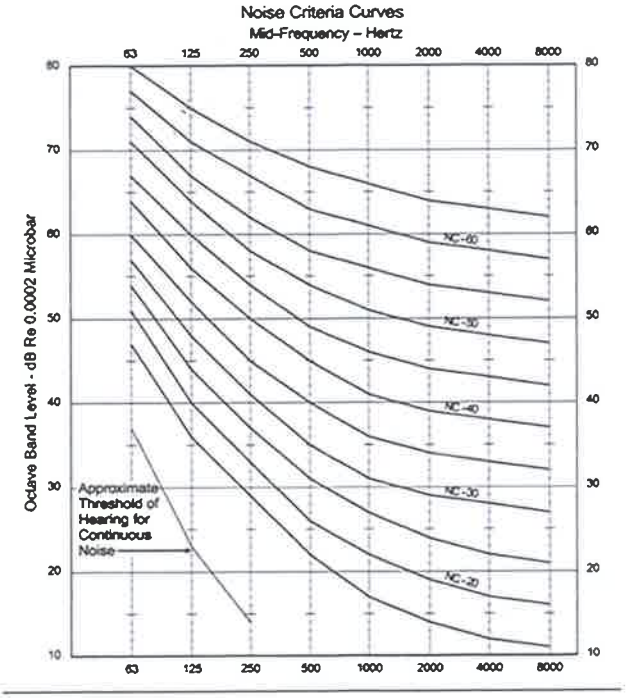


Figure 47

Noise Criteria Curves (NC)

The NC curves are a family of criteria curves that may be used to establish and evaluate acceptable noise criteria limits for occupied spaces. These curves do not adequately predict acoustic quality other than reasonable loudness levels and avoiding speech interference from HVAC and other room noise. The NC curves work well when low frequencies are not important. The curve specified as an acoustic goal for a given application depends on the type of space usage and its level of sensitivity to the level of background noise. Conceptually, the NC curve specified establishes the maximum acceptable upper limit for each octave band. In practice, however, when interpreted strictly as a limit on the level not to be exceeded in any octave band, it has been shown that noise problems are not avoided unless the shape of the actual noise spectrum approximates that of the chosen NC curve over three to four contiguous octave bands. If the actual spectrum shape approximates that of the NC curve in only one or two octave bands, the occupants may interpret the quality of the sound as rumbly or hissy.

In actual design situations, as well as in field measurement techniques, the common practice is to plot the predicted (or measured) sound spectrum data for the eight octave bands (63 Hz - 8000 Hz) on the NC chart with the NC rating corresponding to the highest NC curve tangent to

the spectrum, regardless of where in the frequency range this point occurs. Although this is a simple, common practice, it has caused confusion and frustration in many field installations where noise complaints have occurred.

Although an octave band measurement of the noise may indicate that none of the specified NC levels have been exceeded, it is possible to have an undesirable room noise situation if the actual room noise deviates from the specified NC curve over more than two or three contiguous octave bands.

To illustrate this point, consider this example.

Octave Band, Hz	63	125	250	500	1K	2K	4K	8K
Sound Level, dB	56	60	49	43	35	33	31	34

Given the octave band sound pressure data shown above, determine the room NC rating by plotting the sound pressure data on the NC chart in Figure 48.

From Figure 48, the highest point on the plotted noise spectrum curve is tangent to the NC-45 curve; therefore the rating for this sound spectrum is an NC-45. NC values are usually given to the closest 1 dB step.

Next, consider an example where a supply air grille is located 10 feet away from the room occupant (receiver). There is a central station fan in an adjacent mechanical room. The octave band frequency sound pressure data for both the grille and the fan are plotted on the NC curve as shown in Figure 49.

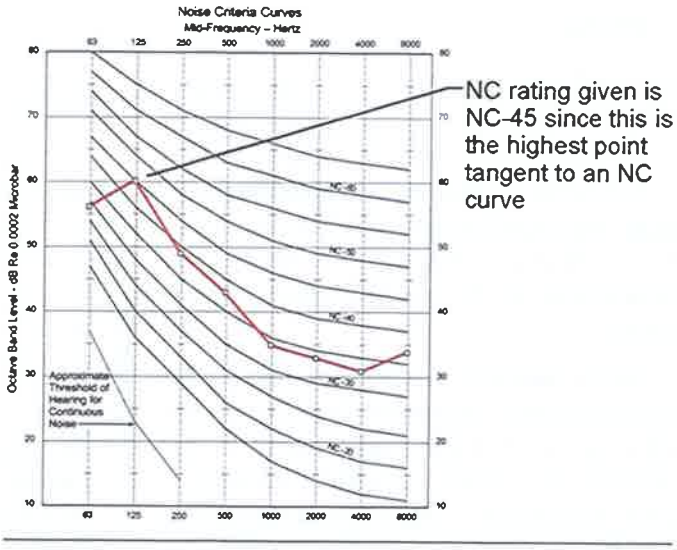


Figure 48

NC Curve Example

It may be noted from the figure that even though the shape and frequency distribution of both sound spectra are considerably different, using the NC rating method, each spectra would be rated at the same acceptable value, NC-35. Despite this, the grille spectrum would generally be judged as hissy due to the presence of excessive high frequencies. The fan spectrum would generally be judged as rumbly because of the steeply rising low-frequency characteristics of the spectra.

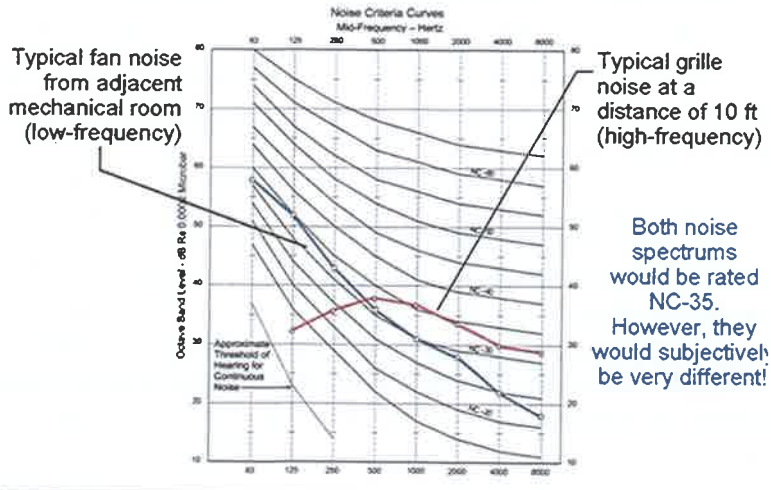


Figure 49

NC Curve Fan Example

Problems with Using Noise Criteria (NC)

The NC method was the first widely accepted octave band rating method and is still widely used today. However, the method should be used cautiously due to the following limitations:

- There is a general agreement among listeners that noise spectra that parallels the NC curve is not neutral, sounding both too rumbly and too hissy.
- The NC curve range does not cover the 16 and 31.5-Hz octave bands, which are regions where many troublesome HVAC noise occurs, especially vibration and rumble.
- The use of a single number rating based on a tangency to a rating curve at one octave band without assigning a subjective quality to the sound spectrum could possibly lead one to mistakenly rate two widely different sound spectra as equally acceptable.

A more acceptable rating procedure, which is becoming more universally utilized due to the qualitative nature of the analysis, is the ***Room Criteria*** (RC) rating method. Note that an RC rating of RC30N, in cases where significant low frequency sound energy is present, is more difficult to achieve than an NC30 rating.

The NC curve

has been one of the most popular methods of specifying the acoustics for a room. However, because it does not adequately reflect sound quality the RC rating method should be used.

Preferred Speech Interference Levels

In order to understand the RC rating we need to define an additional concept called Preferred Speech Interference Level (PSIL). The ***PSIL*** is the average of octave bands that are most important to understanding speech.

One of the prime considerations for an occupied space is the ability to understand the spoken word easily. The Preferred Speech Interference Level can be calculated from the measured octave band sound pressure levels by computing the arithmetic average, in dB, of the octave band levels centered in the 500, 1000, and 2000 Hz octave bands. For example, if the measure levels were:

500 Hz = 52 dB
1000 Hz = 45 dB
2000 Hz = 39 dB
PSIL = (52+45+39)/3=45.3 dB

DISTANCE AT WHICH SPEECH CAN BE EASILY UNDERSTOOD				
DISTANCE (ft)	LEVEL OF SPEECH			
	NORMAL	RAISED	VERY LOUD	SHOUTING
0.5	74	80	86	92
1	68	74	80	86
2	62	68	74	80
4	56	62	68	74
6	53	59	65	71
8	50	56	62	68
10	48	54	60	66
12	46	52	58	64
16	44	50	56	62

500Hz = 52dB
1000Hz = 45dB PSIL = (52 + 45 + 39) / 3 = 45.3dB
2000Hz = 39dB

Figure 50

Preferred speech interface level related to distance for speech.

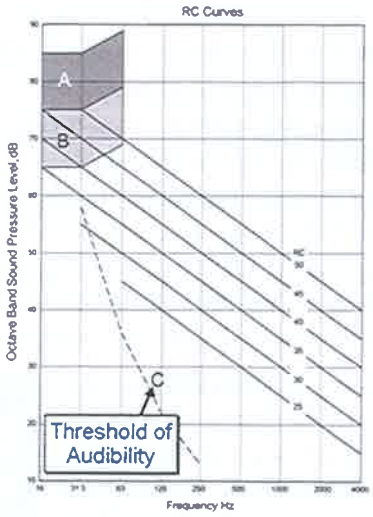
PSIL gives us a simple way to use octave band data to determine acceptability for speech interference. Using the data in Figure 50, quality speech communications in any environment can be determined when the distance between the speaker and the listener and the background sound level in dBA are known. For the example above, we can see that from the data above we could expect to communicate in a normal voice at 14 feet.

Room Criteria (RC)

The RC rating method (Figure 51) is preferred over the use of both the NC and the dBA rating methods due to the following reasons:

- Both the shape and the level of the noise spectrum is considered, making this method more of a qualitative analysis tool
- The RC rating method includes the lower frequency octave bands of 16 Hz and 31.5 Hz.

Accounts for the possibility of low-frequency acoustic energy causing structure-borne vibration and noise in light building construction.



Region A:
High probability that noise induced vibration levels in light wall and ceiling structures will be noticeable. Rattling of lightweight light fixtures, doors and windows should be anticipated.

Region B:
Moderate probability that noise-induced vibration will be noticeable in lightweight light fixtures, doors and windows.

Figure 51

Room Criteria Curves (RC)

The family of RC curves shown in Figure 51 was derived from ASHRAE research and indicates the acceptable background noises in a wide range of typical office building environments. The resulting -5 dB/octave slope of the RC curves corresponds to the average spectrum slope of the empirical data as measured in a large number of office environments that were deemed acoustically acceptable by the occupants.

The shape of these curves represent neutral sounding spectra that are approximately balanced with respect to the relative loudness between the low and high frequency regions of the spectrum. Sound that approximates the shape of the RC curves within +3 dB in high frequency and +5 dB in low frequency are considered neutral or of acceptable sound quality.

The threshold of hearing bounds the low-frequency range of the chart on one extreme and by the concern to prevent perceptible vibration of room walls and ceilings in lightly constructed buildings on the upper end of the range. The shaded regions in the upper left corner indicate the range of low-frequency sound pressure levels where the possibility of perceptible noise-induced vibration should be considered. Sound measurements in the upper shaded region (Region A) indicate a high probability that noise-induced vibration may be clearly perceptible and capable of causing acoustical ceiling systems and light fixtures to vibrate or rattle. In the lower shaded region (Region B), the effects of noise-induced vibration are marginal but still potentially capable of creating problems.

The RC rating method

Reflects the room sound quality by adding a letter qualifier to the NC rating. For most spaces, the goal should be a neutral or N qualifier.

The RC rating procedure consists of two parts, the first part is a number designation (RC number) followed by a letter descriptor (N, R, H, T, or RV) where N = Neutral, R = Rumbly, H = Hissy, T = Tonal and RV = Acoustically-Induced Perceptible Vibration.

The numerical designation of each RC curve corresponds to the ***preferred speech interference level***, or PSIL. The PSIL is derived as explained in the last section. This level (RC number) is coincident with the octave band sound pressure level at 1000 Hz. In other words, the RC number is equal to the PSIL number only at the 1000 Hz point from the intersection point at 1000 Hz and the PSIL, a line is drawn from this point with a slope of -5 dB/octave.

The second part of the RC rating designation is the letter descriptor. This letter describes the quality of the sound, e.g., RC 40(N) would represent a noise with a PSIL of 35 dB and sounding (N) Neutral to the average listener.

Neutral Spectrum (N)

The octave band levels centered at 500 Hz and below must not exceed the reference level octave band by more than 5 dB at any point within the range. This line is called the rumble roof. The octave band levels centered at 1000 Hz and above must not exceed the reference level octave band by more than 3 dB at any point within the range. This line is called the hiss roof. Levels that exceed these values will be interpreted as either ***rumbly*** or hissy.

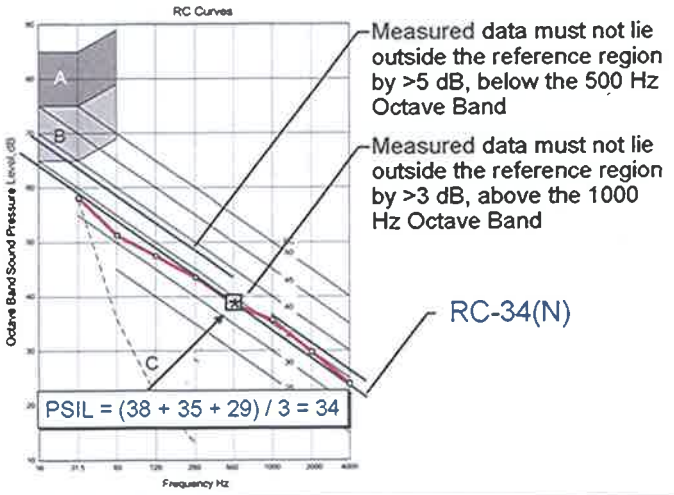


Figure 52

RC Curve Neutral

Rumbly Spectrum (R)

The octave band level centered at 500 Hz and below establishes the rumble region of the sound spectrum. Sound levels that exceed the reference octave band level by more than 5 dB at one or more points within the range will generally be judged as rumbly. This is illustrated in Figure 53.

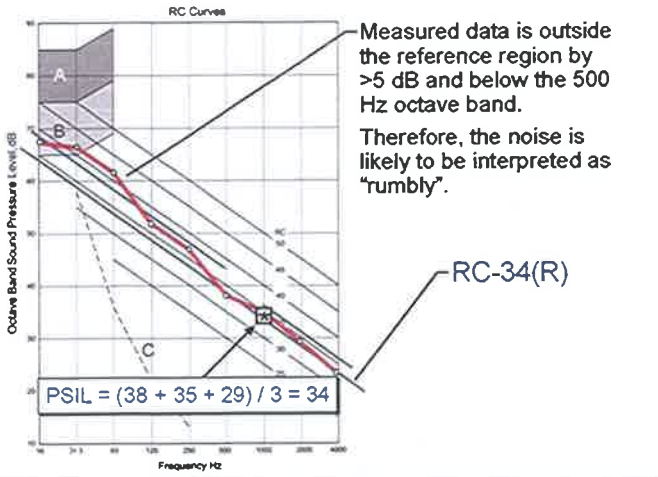


Figure 53

RC Curve Rumbly

Hissy Spectrum (H)

The octave band level centered at 1000 Hz and above establishes the hissy region of the sound spectrum. Sound levels that exceed the reference octave band level by more than 3 dB at one or more points within the range will generally be judged as hissy. This is shown in Figure 54.

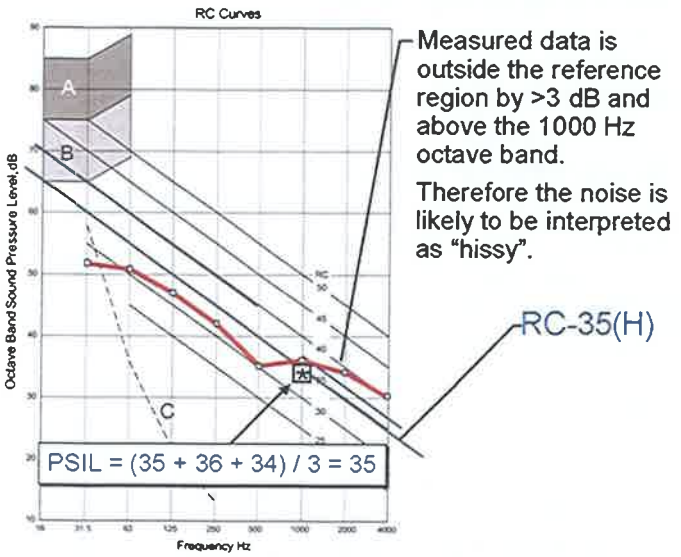


Figure 54

RC Curve Hissy

Tonal Spectrum

Noise is generally perceived as tonal (a distinct tone) when the sound spectrum shape has an abrupt change in slope causing a peak within a particular octave band relative to levels in the two adjacent octave bands. If the peak is greater than 3 dB above a line drawn between the two adjacent octave band peak points, the tone will likely be audible.

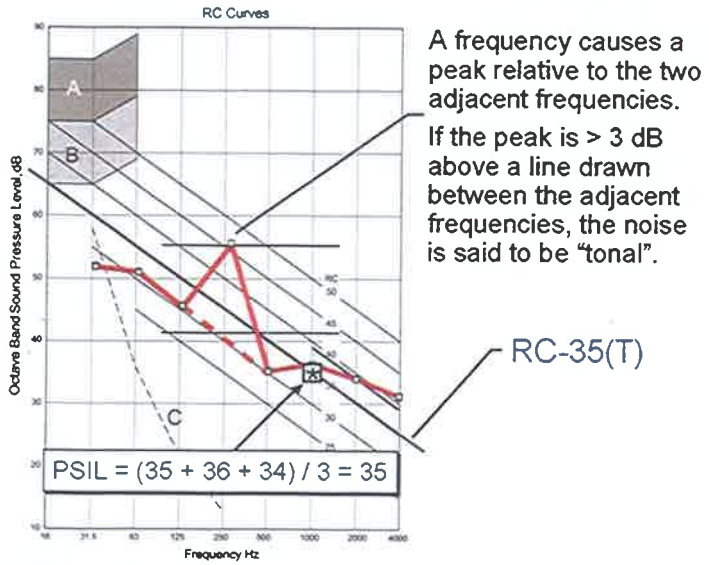


Figure 55

RC Curve Tonal

Acoustically Induced Perceptible Vibration (RV)

Sound pressure levels within the 16 Hz to 63 Hz octave band range are associated with acoustically induced vibration of building components such as walls, floors, ceilings and other furnishings in contact with any of these surfaces. These regions are indicated as regions A and B on the RC chart.

Although acoustically induced vibration typically does not interfere with normal speech, it can be particularly objectionable because we feel or see objects vibrating like the ripples in your cup of coffee sitting on the vibrating table. This is illustrated in Figure 56.

Sound in the A region has a high probability that noise-induced vibration will result. This is particularly true with light building construction. Sound in the B region has the potential to cause vibration in light construction. Carefully evaluate situations that fall in the B region to avoid low frequency rumble.

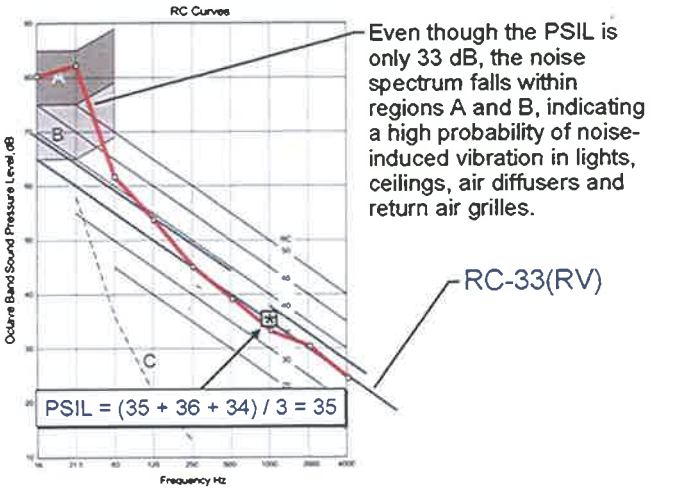


Figure 56

RC Curve for Very Rumbly

Steps for Plotting Sound Pressure Data on an RC Curve

The steps to determine the RC rating for a space can be shown using the same data we used for plotting the NC chart above. The steps are:

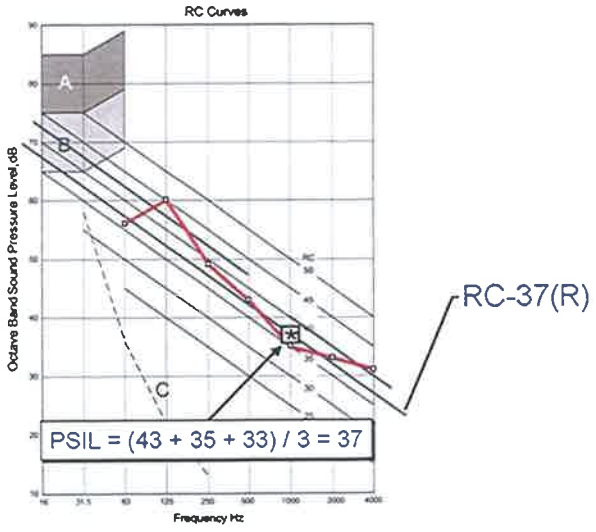


Figure 57

Example RC Curve

1. Plot the measured (or estimated) octave band noise spectrum data on the blank RC chart.
2. Calculate the PSIL value by calculating the arithmetic average, in dB, of the sound pressure data at the 500, 1000 and 2000 Hz levels (add the three values from each level together and divide by three).
3. Plot the PSIL point where it intersects the 1000 Hz octave band line. From the point where the PSIL value intersects the 1000 Hz line, draw a line with a slope of -5 dB/octave band from the 31.5 to the 4000 Hz range. This line is the reference line to be used to evaluate the sound quality of the spectrum being rated.

4. Draw another line +5 dB above and parallel to the reference line extending from the 31.5 to the 500 Hz octave band. Finally, draw a third line +3 dB above and parallel to the reference line, extending from the 1000 to the 4000 Hz octave band. The range between these two lines and the reference line represent the neutral (N) range and is the maximum permissible deviation of the noise spectrum above the reference line. Values which lie outside this neutral range will be either rumbly, hissy, or RV. When acoustically induced vibration is suspected, an alternate form of rating procedure should be used to evaluate sound levels in the 16 Hz octave band range.

NOTE: Recommendations on how to correct to a spectrum that is not neutral are given later in this text.

We have covered the six methods of single point ratings of loudness and the two methods of describing space acoustics, there are two other rating terms which HVAC designers should be acquainted with, Noise Reduction Coefficient and Sound Transmission Class. Both of these deal with a rating number for the path where the rest have dealt with the source and receiver relationship.

Noise Reduction Coefficient (NRC)

The ***Noise Reduction Coefficient*** (NRC) is a rating number for how much absorptive a material is. The rating number is based on the numerical average of the absorption coefficient or properties in the 250, 500, 1000, and 2000 Hz octave bands, which are the most important bands for speech. The NRC gives no indication of the material's ability to absorb acoustical energy in the higher or lower frequencies nor does it reflect the materials ability serve as a barrier and limit sound transmission.

Sound Transmission Class (STC)

Rather than applying to the source and receiver as most ratings do, ***Sound Transmission Class*** (STC) relates to the attenuation in the path by a material, normally walls. STC is another single-number rating, and it is a measure of the ability of a material or an assembly to resist airborne sound transfer at the frequencies between 125-4000 Hz. The STC is intended to be used in the control of speech transmission between spaces.

A partition STC rating is determined by measuring its transmission loss over the frequency range of 125 Hz to 4000 Hz. This range is consistent with the frequency range of speech. The STC rating does not account for low frequency sound transfer. In the design of mechanical equipment rooms and other space with unique sound sources in low frequency ranges should not use STC data.

Description	Estimated STC Rating	Wall Assembly
3-5/8" metal studs, 5/8" gyp (2 layers total), No insulation	38 to 40	
3-5/8" metal studs, 5/8" gyp (2 layers total), Batt insulation	43 to 44	
Staggered studs, 5/8" gyp (2 layers total), Batt insulation	46	
2 x 4 studs, 5/8" gyp (2 layers total), Batt insulation	34 to 39	
2 x 4 studs, 5/8" gyp (2 layers total), Resilient Channel, Batt insulation	45 to 52	

Figure 58

Sound Transmission Class

Loud speech will still be understood fairly well through an STC 30 wall, but speech will most likely not be audible through an STC 60 wall. An STC of 50 is a common building standard and blocks approximately 50 dB of the sound transmitted through the partition. However, people in adjoining spaces will most likely be aware of loud speech. Constructions with a higher STC (as much as 10 dB better - STC 60) should be specified in sensitive areas where sound transmission is a concern.

The International Building Code (IBC) requires a STC class for partitions between service areas, corridors, stairways, and similar spaces of STC rating of 50 (if tested in a laboratory) or 45 (if tested in the field) and a requirements for floor/ceiling assemblies: STC ratings of 50 (if tested in a laboratory) or 45 (if tested in the field).

STC is highly dependant on the construction of the partition. A partition's STC can be increased by:

- Adding mass
- Increasing air space
- Adding absorptive materials within the partition

Some examples of how these affect the STC rating are shown in Figure 58.

Manufacturer's Sound Ratings

The most common sound rating methods have been described in the last few sections of this TDP. These methods provide us a means of relating and establishing the sound levels for most applications, to use the methods described, we need to know the source sound power strength, the receiver location and distance, and the sound paths with the appropriate attenuation elements. Therefore, one of the first things to establish is the source sound power data. For HVAC systems and particularly when considering a new design this information comes from manufactures sound ratings. This section describes the various methods manufacturers use to provide sound data on their equipment. The data reported and the test procedure used changes depending on the type of equipment.

Various test procedures apply to different types of HVAC equipment. It is important for the HVAC system designer to understand how a unit is tested so that the published sound data can be properly used to estimate the acoustical performance. The basic information related to each type of HVAC product follows; a summary table is provided in the appendix of this TDP.

Sound test standards

are available for rating the acoustical performance of most HVAC equipment and components. This means the industry agreed to a procedure that can be used to test the product and present the data. A certification program, which means the testing is conducted in a qualified lab and the test results subject to verification, covers some products. Designers should always look for manufacturers data tested to the appropriate standard.

Sound Standards

A number of organizations that write acoustical testing standards have developed testing standards for HVAC equipment. Air Conditioning & Refrigeration Institute (ARI) has published sound standards that apply to most types of HVAC equipment. Most manufacturers that provide sound data for their equipment will test based on the ARI standards. Other organizations also have developed sound standards that apply to other equipment types like AMCA (Air Moving and Control Association) for fans and duct silencers.

ARI

ARI sound standards provide all the necessary requirements and procedures for rating HVAC units. ARI sound standards also specify how the resulting data should be published. Therefore, you can compare manufacturer sound data on units that have been rated according to the same ARI sound standard. If they claim compliance, the sound data is comparable. For instance, if one unit is rated at 75 dBA and a second unit us rated at 85 dBA, we can assume that the 75 dBA unit will be quieter when installed. Do not compare an ARI sound rating to data taken in the field or data that has been taken by "modifying" an ARI standard. ARI does not perform any sound tests; units that are tested in independent laboratories that test to the ARI standard. ARI does provide for some product categories a listing of products that manufacturers have "certified" as being tested to the standard. Products in a certified program are constantly audited to be sure they comply with the published data.

ARI Standards

are available for free download at ARI's web site www.ari.org.

AMCA

AMCA is primarily involved with testing related to fans, louvers, silencers, and dampers and the test room the fans are rated in. Like ARI, manufacturers will test to AMCA standards. AMCA however, does perform sound tests in their own laboratories. AMCA certifies many of the products listed with in one of its rating programs. AMCA conducts a very intensive audit program.

ANSI

ANSI is the American National Standards Institute and does not directly write product sound standards. There goal is to provide standards which can be used for rating products that are unbiased and presented in a uniform fashion. They have adapted a number of the ARI, AMCA, and ASHRAE standards for inclusion in the ANSI standards program. ANSI only administers the standards and does not test or certify.

ASHRAE

Unlike ARI and AMCA, which are organizations sponsored by manufacturers in the related industry, the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) is an organization of individuals concerned with the development of the industry. A number of standards related to sound testing and equipment sound levels have been developed. ASHRAE standards are consensus-based standards developed for the good of the industry. ASHRAE does not test or certify products.

Many of the organizations have standards that overlap in the products to which they apply. In the section that follows, we will discuss by product type the standards that are used most often to determine manufacturer sound ratings. There are a number of other standards developed by acoustical societies that are also used or referenced in these standard and are used to take field readings.

Test Chambers

Manufacturers conduct most sound tests of products in sound test chambers. A sound test chamber is a specialized test environment that allows making acoustical measurements. Test chambers provide an environment in which the relationship between sound power and sound pressure are known and reduce or eliminate interferences from other outside sound sources. In general, a test chamber is a specially constructed room with a microphone. Many different types of test chambers exist depending on the type of sound measurements to be done. Sound power is what used to determine the acoustical contribution of a unit. However, it cannot normally be measured directly. The sound pressure levels can be measured and used to determine sound power. To do this, the sound pressure waves from the unit under test need to be isolated from ambient background sound pressure waves. Sound pressure waves are picked up by a microphone and mathematically converted to sound power.



Figure 59

Sound Testing Organizations

Shown in the figure are four regions important to acoustical testing. The first region is ***near field.*** In this region close to the sound source, the sound pressure fluctuates about a mean value. The sound waves arrive at the measuring microphone from different points on the sound source. They are out of phase and add constructively and destructively. The size of this region is a function of the size of the unit being tested, the location of the microphone, and the wavelength. The ***far field*** region is located just after the near field and farther from the sound source. The sound pressure wave is diminishing as a function of distance. In this region, the sound waves arrive from different points on the unit and are in phase. In the direct field, most of the sound waves that reach the microphone travel from the unit in a direct line with little reflected impacts. In the ***reverberant field***, most of the sound energy reaching the microphone has been reflected one or more times giving relatively uniform contribution from all the reflecting surfaces. Finally, in the ***ambient noise*** region, the contribution of the unit has become insignificant to the background sound pressure.

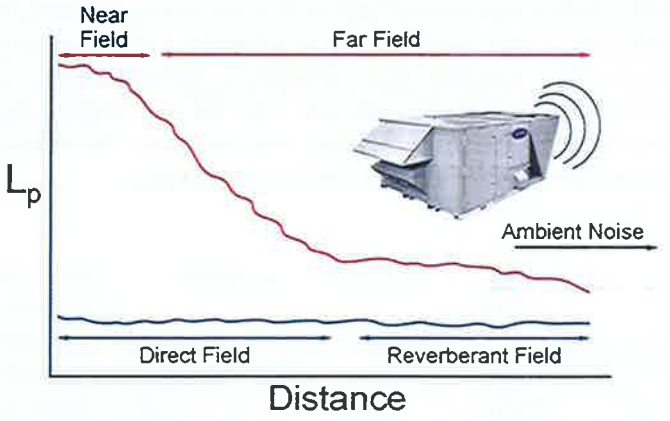


Figure 60

Sound Fields

Two basic types of test chambers are used: anechoic and reverberant. In ***anechoic test chambers***, the wall, ceiling, and even sometimes the floor are covered with acoustical material that absorbs the reflected sound waves, such as absorbent, textured foam. Sound measurements in an anechoic chamber contain little reflected sound and therefore are a direct field and can be measured by the inverse square law. In these chambers, the directivity of the sound from the source is preserved. However, a large number of microphone locations and a large test room particularly for lower frequencies is required. The microphone position is normally twice the wavelength distance from the source, making the use of this type chamber difficult to use with most HVAC products with significant low frequency (long wavelength) sound.

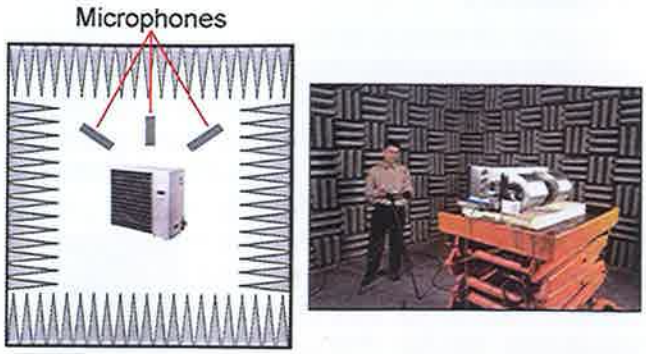


Figure 61

Anechoic Test Chamber

Reverberant chambers are the other type of sound testing chamber. In reverberant chambers, the surfaces are acoustically hard and sound is reflected off all the surfaces and readings taken in the reverberant field. This type of chamber consists of room with a rotating microphone and a ***reference sound source***. The reference sound source establishes a measured ambient sound for the space and the unit contribution is measured above this reference ambient. Reverberant rooms must also be of adequate volume to permit reasonably accurate low frequency measurements. To measure very low frequencies, 120 Hz and below, large rooms with adequate volume are required. Reverberant rooms are used for most HVAC sound testing.



Figure 62

Reverberant Test Chamber

Indoor Ratings for Non-Ducted Air-Conditioning Units

Non-ducted products include products installed in the space, fan coils, console water source heat pumps, duct free split fan coils, and packaged units that supply air directly to the room with no duct work to provide attenuation. To perform the test, the unit is mounted in the room with mounting guidelines established by ARI 350, as indicated in Figure 63. The units are furred in if so designed and the mounting wall is heavy masonry construction. The unit is operated at the conditions required for it to make its rated capacity. Units that operate in heating are tested at the heating rating as well. Sound ratings from the test are expressed in Octave Band Sound Power Levels from 125 through 8000 Hz, A-Weighted Sound Power Level and A-Weighted ***Tone corrected*** Sound Power Level. If the manufacturer chooses, ratings may also be done for other conditions for example medium and low fan speeds. These units are rated per ARI 350, *Sound Rating of Indoor Non-Ducted Air Conditioning Equipment.*

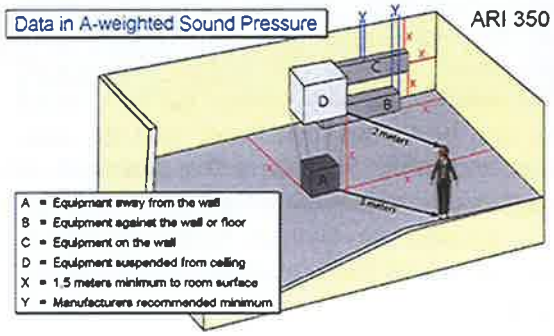


Figure 63

Testing Non-Ducted Packaged Units to ARI 350

Packaged Terminal Units

Packaged terminal equipment are units that have both an indoor and outdoor section in the same unit, mounted through the wall with free discharge. Sound ratings for these units are tested per ARI 300, *Sound Rating and Sound Transmission Loss of Packaged Terminal.* For testing, the units are mounted through the wall, as they would be in the field. These units present a special situation since they have indoor and outdoor sound sources and sound power transmission from outdoor to indoor. The unit is tested to rate all three of these conditions. Results of the testing are reported in Octave Band Sound Power Levels, A-Weighted Sound Power Level and A-Weighted Tone corrected Sound Power level; additionally this standard rates the sound transmission loss of the unit using sound transmission class numbers.

Rating Indoor Ducted Units

Units such as air handlers, ducted fan coil units, ducted water source heat pumps, rooftop units, and indoor packaged units are rated by ARI 260, *Sound Rating of Ducted Air Moving and Air Conditioning Equipment.* These units have the following unique characteristics that must be considered: they may be located in the space with the receiver or remotely; they may contain compression equipment; and the return air and the supply air may be ducted or only the supply air may be ducted. This results in different amounts of attenuation for each of the three paths.

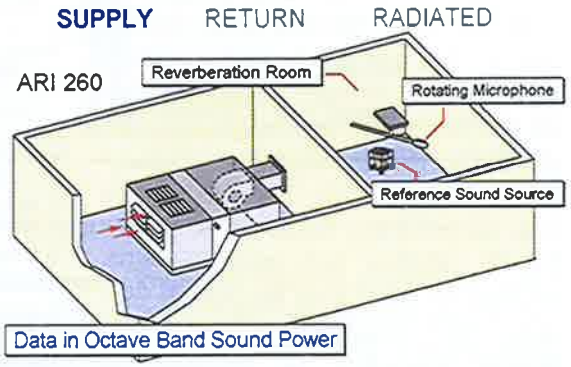


Figure 64

Testing Ducted Indoor Units – Fan Discharge Sound Power

These units are rated based on determining a sound rating for each of the three paths, one for sound through the cabinet, one for return sound and one for supply duct sound. This standard not only covers the fan's sound but also the sounds that "leak" into the ductwork from other sources in the unit such as compressors and outdoor exhaust fans, etc. This standard also includes ratings for ***casing-radiated sound*** that is important in mechanical equipment rooms.

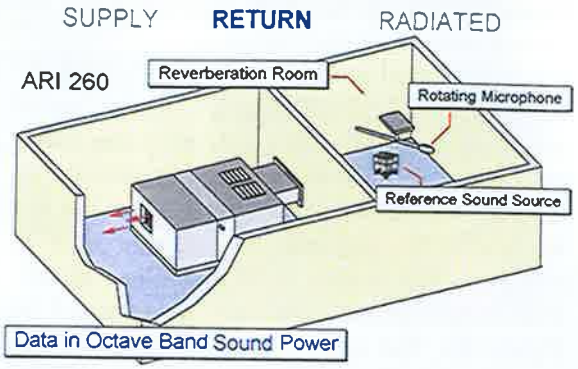


Figure 65

Testing Ducted Indoor Units – Fan Inlet Sound

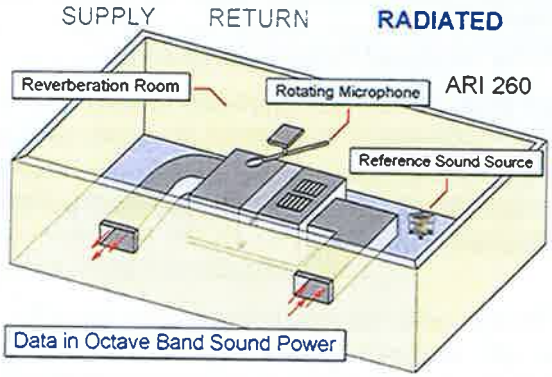


Figure 66

Casing Radiated Units – Fan Discharge Sound

Ratings for Air Terminals

Air terminals are typically devices like VAV boxes and double duct boxes that are ducted at the inlet and outlet. This does not included the grilles, registers, or diffusers, nor does it include room fan coils or induction units. These devices contain dampers that may generate sound as their position is changed, or as unit airflow is changed. They are also often located above the occupied space, which means casing-radiated sound can also be a concern. The terminal is rated for discharge and ***radiated sound power*** using ARI 880, *Air Terminals*. Ratings are in Octave Band Sound Power Levels. Airflow and static pressure settings for these tests are done at design flow conditions. The actual testing procedure is outlined in ASHRAE Standard 130, *Methods of Testing for Rating Ducted Air Terminal Units.*

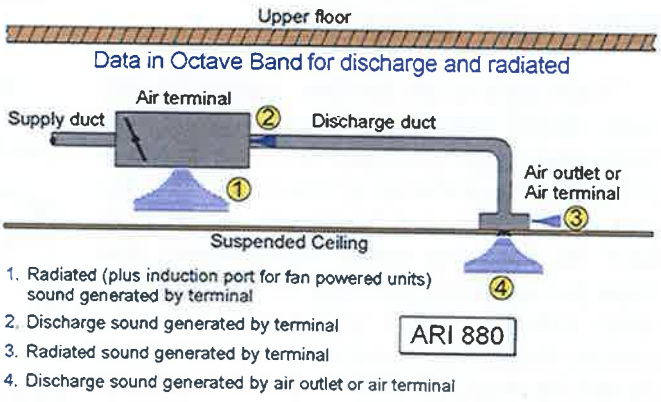


Figure 67

Testing Terminal Unit Sound Power

Because air terminals can represent such a large contribution to the occupied space sound levels a standard has been established to apply these ratings using the application standard ARI 885 *Procedures for Estimating Occupied Space Sound Pressure Levels in the Application of Air Terminals and Air Outlets.* We will use this standard for acoustical analysis later in this TDP as it provides a way to determine the actual impact of a terminal on the occupied space sound power level.

There are nine classifications of terminal units covered by this standard. Each has some unique airflow characteristics and the actual test procedures vary slightly between terminal types. The nine types are:

- Single-Duct central air terminals
- Dual- Duct central air terminals
- Induction Central air terminals
- Bypass Central air terminals
- Series flow, fan powered mixing boxes
- Parallel flow, fan powered mixing boxes
- Inlet damper, integral diffuser terminals
- Outlet damper, integral diffuser terminals
- Modulating diffuser terminals

There is a certification program for these units and certified sound power levels can be provided by manufactures that participate in the ARI certification program.

Grilles, Registers, Diffusers and Duct Silencers

The devices which deliver the air to the room and that cover the openings where the air is returned to the unit are called grilles, registers, and diffusers. Unlike terminals, these devices directly discharge air to the occupied space and do not have automatic operated airflow control dampers. These units are tested using ARI 890, *Air Diffusers and Air Diffuser Assemblies*. This standard references the rating procedure in ASHRAE Standard 70, *Methods of testing for rating the performance of air outlets and inlets*. ARI application Standard 885 may also be used to evaluate the sound power level in a space. Units are tested in an acoustical ceiling tile test setup with design airflow and pressures and adequate length of straight duct to avoid any duct generated sound. The test results are reduced by 10 dB in every frequency band to account for space effect. Test results are presented as NC rating based on the plot from the octave frequency levels. The tested sound power is used to estimate the resultant sound pressure in a 200-ft^2 average room. The industry assumes that $L_w - L_p$ for the space is 10 dB so then an L_p can be estimated. This is plotted on an NC chart to get an NC rating for the unit at this rating point.

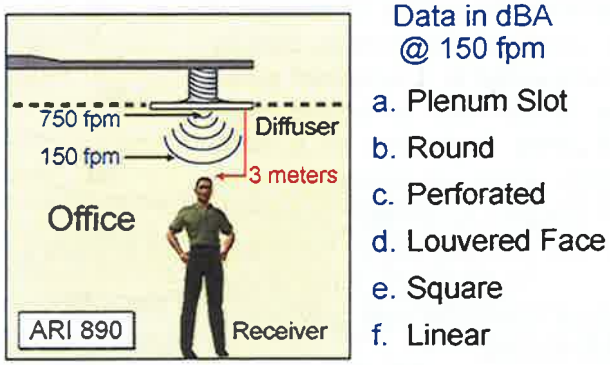


Figure 68

Testing Grilles, Registers, and Diffusers

The certification program for grilles, registers, and diffusers is not active at this time.

Packaged acoustical attenuation devices, referred to as duct silencers, may be placed in supply or return duct systems to help attenuate sound generated at the unit. These devices are certified under AMCA Standard 1011, Certified ratings program for acoustical duct silencers using ANSI standard E477, Standard test method for measuring acoustical and airflow performance of duct liner material and prefabricated silencers. The ratings list the ***insertion loss***, or how many dB are attenuated in each octave band as a function of airflow and static pressure.

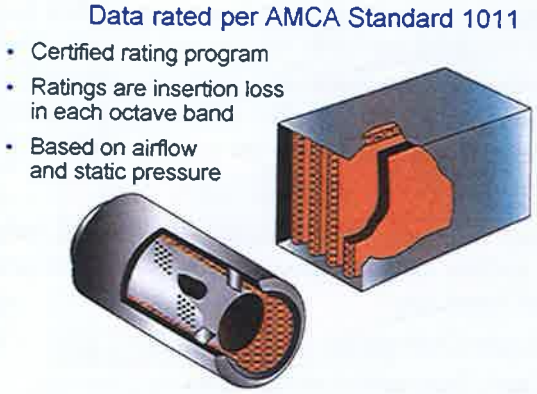


Figure 69

Prefabricated Duct Silencers

Sound Ratings for Chillers

Water chillers, pumps and other devices which are located in a machinery room are often not covered by a sound rating program, However, the sound power levels are necessary if an acoustical analysis is to be performed. To obtain this information in a consistent and usable format, ARI Standard 575, *Method of Measuring Machinery Sound Within an Equipment Space* may be used. Measuring machinery sound levels in a mechanical equipment space is not a rating standard but a method of taking the measurements within the equipment room. The standard provides guidance on the number of test points to be taken and the location. Results are documented in A-weighted sound power levels and in octave band sound power levels from 63 to 8000 Hz. Since results are dependent on the actual machine, it operating characteristics and the mechanical equipment room, results applied to other design situations need to be used understanding this caution.

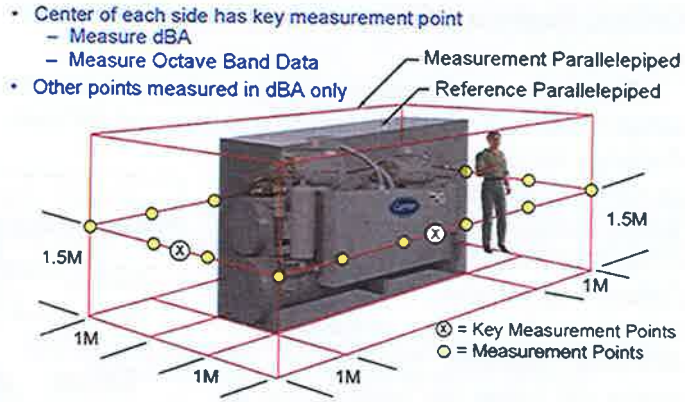


Figure 70

Sound Pressure Testing Levels for Chillers with ARI 575

Sound Ratings for Fans

Many different types of fans are used in HVAC design from central station air handlers to fan used as exhausters and return booster fans, there are several sound test standards which apply for these fans. When the fans are part of either a ducted or non-ducted HVAC product the rating is determined using ARI 260 or ARI 350 as applies, these standards also refer to some of the standards covered here.



Figure 71

Sound Testing Standards for Fans

The primary standard related to fan sound testing is AMCA 300, Reverberant room methods for sound testing fans, which describes how inlet, discharge, and casing-radiated sound should be tested. AMCA 311, a certified sound rating program for air moving devices, covers a certification program for fans. Many of the fans used in other HVAC equipment are certified under this program, such as a fan used in a rooftop unit. However, remember that once the fan is applied in the unit, the rating will be different. Many construction features of the unit are very different from the testing procedure listed in these standards, such as the amount of straight length after the fan discharge before an elbow. These changes can have significant impact on the unit sound levels. It is best to use the manufacturer's data to test the appropriate standard with the fan as applied in the HVAC product.

Ducted fans and non-ducted fans used alone in HVAC systems should use the AMCA 300 sound test data. This data is published in three categories: inlet, discharge, and radiated sound power levels by octave band. Some non-ducted fans are also rated using AMCA 302, Application of sone ratings for non-ducted air moving devices.

Compressors

When enclosed in an HVAC unit, the sound rating for a compressor is included in the sound rating of that unit. However, sometimes the compressor is a separate component. ARI Standard 530, *Method of Rating Sound and Vibration of Refrigerant Compressors*, can be used for these situations. Because compressor sound is predominately low frequency, this testing standard takes into account more on the low frequency bands down to the 63 Hz band. The sound data is reported as octave band data, A-weighted data, and vibration peak-to-peak data.

Rating Sound Levels for Outdoor Applications

The outside sound of HVAC equipment can be just as important as the indoor. When HVAC equipment is applied in residential neighborhoods and some commercial areas as well, more and more communities are specifying a maximum sound pressure level at the property line. These restrictions are called lot-line restrictions. Outdoor sound tends to follow the inverse square law, but due to reflective surfaces, this can be come a much more complicated analysis. ARI Guideline

L, *Assessing the Impact of Air Conditioning Outdoor Sound Levels in the Residential Community*, can be used to help determine the impact of outdoor ratings and complying with these requirements.

Outdoor-mounted unitary equipment sound ratings are rated according to one of two ARI standards, depending on its cooling capacity. Unitary equipment below 135,000 Btuh cooling capacity is per ARI 270, *Sound Rating of Outdoor Unitary Equipment*. This rating is an A-weighted Tone corrected value. (Future rewrite of the standard may include octave bands.) Units are tested at design performance conditions and the test results in a single number rating for the unit. Additionally, to apply the data from ARI 270, use ARI 275, *Application of Sound Rating Levels of Outdoor Unitary Equipment*.

For Outdoor Condensing Unit (less than 5 tons)

Line	Distance from equipment to evaluation point: 1. 45 ft [13.7 m] 2. 30 ft [9.1 m] 3. 38 ft [11.6m]	Evaluation Points 1	2	3
1	Unit Sound Rating Level (ARI Standard 270)	71	71	71
2	Equipment Location Factor (Table 1, Item 1)	6	6	6
3	Add Lines 1 and 2	77	77	77
4	Barrier Shielding Factor (Table 1, Item 2)	0	0	0
5	Sound Path Factor (Table 1, Item 3)	0	0	17
6	Distance Factor (Table 2)	30.5	27	29
7	Add Lines 4, 5 and 6	30.5	27	46
8	Estimated A-Weighted Sound Pressure Level (Subtract Line 7 form Line 3)	46.5	50	31
9	Estimated A-Weighted Sound Pressure Level Rounded to Nearest Whole Number	47	50	31

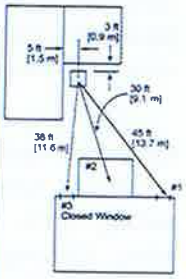


Figure 72

Outdoor Sound Power using ARI Guideline L

Data rated in application reported as dBA

ARI 370

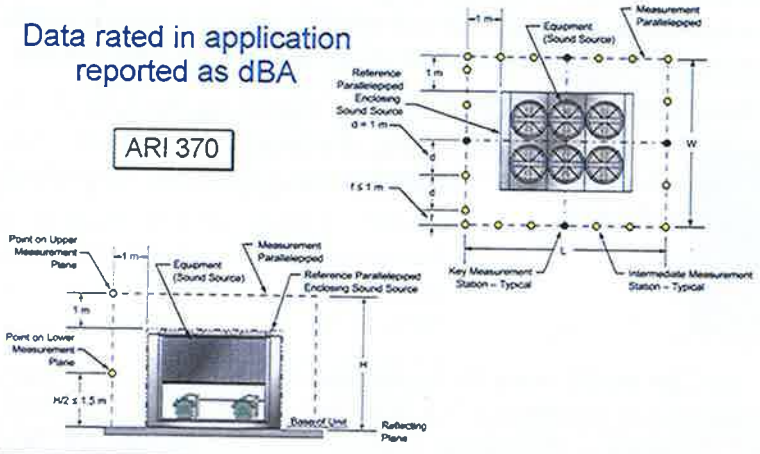


Figure 73

Outdoor Sound Power Testing of Large Air-Cooled Units

Unitary equipment above 135,00 Btuh cooling capacity is rated per ARI 370, *Sound Rating of Large Outdoor Refrigerating and Air Conditioning Equipment*. Data provided from this program includes: Octave Band Sound Power Levels, A-Weighted Sound Power Level, A-weighted Tone corrected Sound Power Level, and sometimes one-third octave band sound power levels.

ARI Standard 370 does not cover large condenser air and evaporative cooled condensers or cooling towers.

Mock-up Tests

Some projects have stringent sound requirements in the occupied space such as a lease that is based on maintaining a certain RC or NC level in a room adjacent to the equipment room. Since the actual sound is function of possibly several sound sources and sound paths each have a degree of uncertainty, a mock-up may be requested. A mock-up is a sample space constructed before the final design where application sound testing can be performed to ascertain if the design will meet the sound requirements. For a mock-up, the equipment room and the adjacent space are constructed following the actual design and equipment as closely as possible. The equipment is operated at design conditions, if possible, and sound pressure level readings are taken in the adjacent space to verify the required space sound pressure.

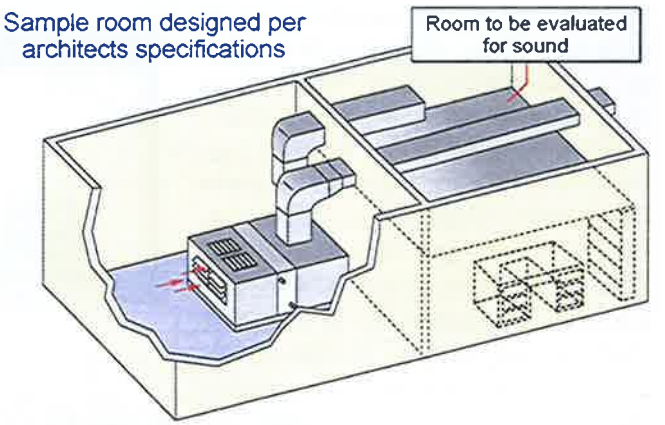


Figure 74

Mock-up Testing

While mock-ups do help the design team determine if the design will meet the requirements, they can be very expensive to construct.

OSHA Requirements

As was pointed out at the beginning of this TDP, when a person experiences long-term exposure to sounds, permanent hearing loss can result. To protect workers in industrial environments the Occupational Safety & Health Administration (OSHA) has established maximum exposure times based on both sound strength and exposure time. These are calculated based on a 90-dBA exposure with a duration of 8 hours as the ***Permissible Exposure Limit (PEL)***. An 8-hour exposure of 85 dBA is only 50 percent of the PEL. Sound control methods can be applied that will reduce the exposure to acceptable limits. A flow chart for this is shown in Figure 75.

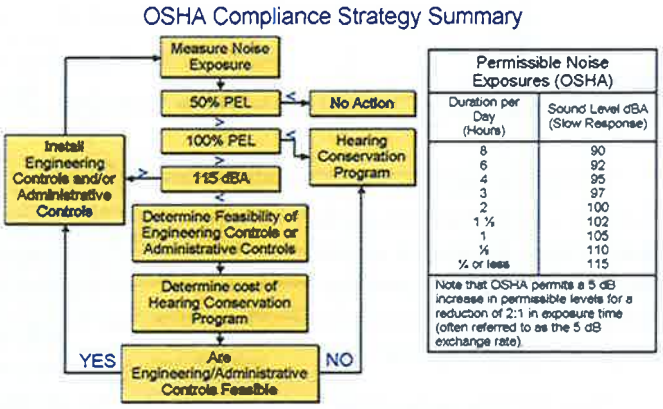


Figure 75

OSHA Sound Reauirements

School Classroom Design Standard

Some occupied spaces have special acoustical requirements based on the activity in the space and how the acoustic affects the activity. School classrooms, for example, have special acoustical requirements. Results of student testing have found a strong correlation between student's ability to learn and the characteristics of the acoustics of the classroom. Excessive noise and reverberation interfere with speech, impacting a student's ability to learn. Students with hearing loss, students learning a language, and students with English as a second language are especially susceptible. Because of these issues, schoolroom acoustics was addressed to help such students and became ANSI / ASA Standard S12.60-2002, *Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools.*

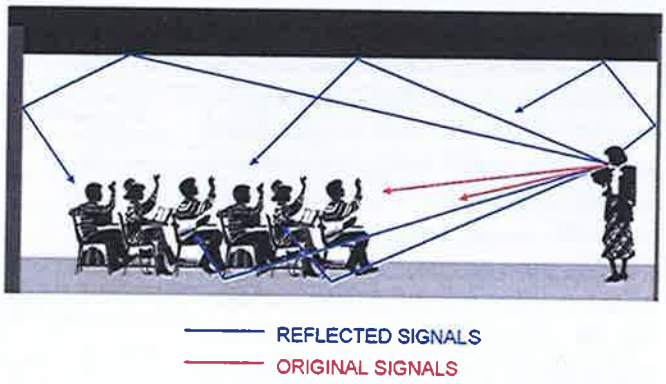


Figure 76

Reverberant Field as Applied to Classroom Acoustics

This standard sets criteria on the minimum requirements for acoustics in classroom spaces based on achieving intelligible speech. Three issues must be considered: continuous background sound level, reverberation, and isolation from other interior and exterior sound sources. Spaces within the school are divided into several different categories with set maximum noise criteria for each of the spaces. The maximum permissible background noise for a typical classroom is NC 35. Classrooms must have a maximum reverberation time depending on volume of 0.6 to 0.07 seconds.

Currently, the requirements have not been incorporated in the International Building Code. Therefore compliance with these requirements is voluntary, a few sates have adapted the requirements.

Determining Acoustical Design Requirements

In the last section, we determined how to find the sources of sound power data using manufacturer's published sound data. We now turn our attention to the receiver and how to establish a design limit for sound pressure at the receiver's location. Our first step in this process is to establish our acoustical design goal. These goals change depending on the space usage and project goals.

Select acoustic design goals

that reflect the use of the building; to stringent a goal will result in excess cost and to lenient a goal will result in dissatisfied occupants

Determining acoustical requirements can be divided into two categories: indoor and outdoor goals.

Outdoor Sound Criteria

Outdoor sound requirements are usually set by the lot-line restrictions established by local codes. These requirements are usually specified as the sound pressure in dBA at the property line. Some codes may use octave band sound pressure data rather than single number dBA, and some may incorporate requirements for pure tone. In any case, the readings are established by the data at the property line of the project. Determining the design criteria could be as simple as reading the local codebook. The designer must then convert the manufacturer's sound power data into sound pressure data at the property line. Remember that the reflection paths need to be accounted for using the directivity factors.

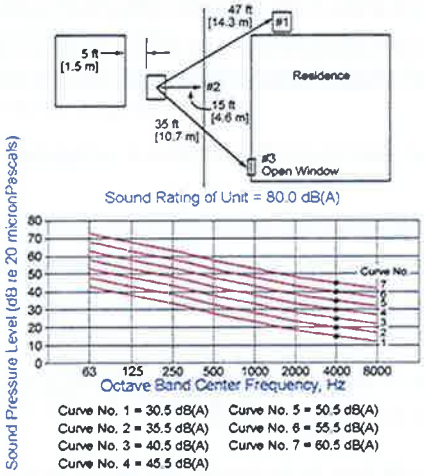


Table 1. National Estimate of Outdoor Background Noise Based on General Type of Community Area and Nearby Automotive Traffic Activity

No.	Condition	Curve No. in Figure 2
1	Nighttime, rural; no nearby traffic of concern	1
2	Daytime, rural; no nearby traffic of concern	2
5	Nighttime, urban; no nearby traffic of concern	3
7	Nighttime, business or commercial area	4
8	Daytime, business or commercial area	5
10	Daytime, industrial or manufacturing area	6
14	Within 300 ft [91.4 m] of continuous heavy-density traffic	7

Figure 77

Outdoor Design Goals

Indoor Design Criteria

ASHRAE has developed a chart outlining acceptable acoustic design criteria for various types of facilities. The chart shown in Figure 78 is published in the ASHRAE Fundamentals Handbook. The designer should understand that these ranges are somewhat conservative; the higher value in the range may be used without concern. When economic considerations dictate the project, the ranges may be increased by +5 dB, provided the sound spectrum is neutral and balanced.

Occupancy	Recommended NC or RC Range
Private Residence	25-30
Hotels/Motels	
Private, Meeting & Banquet Rooms	25-30
Offices	
Private	25-30
Executive, Conference Room	30-35
Open Areas	35-40
Computer Rooms, Public Circulation	40-45
Hospitals	
Private Rooms	25-30
Wards	30-35
Operating Rooms, Corridors, Public Areas	35-40
Churches/Schools	
Lecture/Classrooms	25-30
Open-Plan Classrooms	30-35
Libraries & Courtrooms	35-40
Movie Theatres	30-35
Restaurants	40-45
Concert Halls/Recording Studios	15-20
TV Studios	20-25

Figure 78

ASHRAE Indoor Design Goals

Steps to Meet Desired Acoustic Goals

1. Determine if outdoor noise requirements exist from local code or other property restrictions.
   - If possible, select a unit location that gives the greatest distance to the property line.
   - Locate the unit to avoid areas with multiple reflection paths
   - Select outdoor unit with sound power data that will allow achieving the sound pressure data; consider options which reduce sound if required, IE low sound condenser motors, compressor blankets.
   - If barriers are required, determine by acoustical estimates.

2. Establish appropriate and practical acoustic space sound level goals (refer to Figure 78).
   - Specifying an acoustic level that is too low will incur unnecessary costs.
   - Specifying or allowing an acoustic level that is too high will create a distracting, low-quality environment.
   - Consider desirable noise masking in spaces where adjacent noise is objectionable. In these cases, a constant white noise may be preferred.

3. Establish sound power (Lw) requirements for fans, air terminals, diffusers, grilles, and other noise-generating equipment that is likely to impact the acoustic environment.

Design Considerations to Minimize Noise in the Conditioned Space

① Use Accurate Sound Power Data

- Only select equipment that has ARI-certified sound data (if a certification program exists) or data tested to the appropriate ARI standard. For products like silencers and free standing fans, use AMCA certified data.

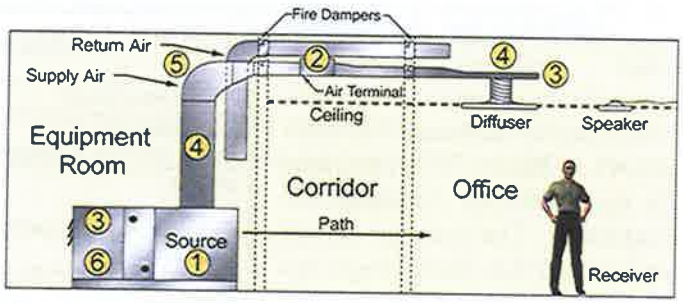


Figure 79

Minimizing Noise in the Conditioned Space

② Location of Air Terminals, Diffusers, and Grilles

- Physically locate air terminals, supply diffusers and return air grilles away from noise-sensitive areas whenever possible (be sure to consider both discharge and radiated sound).

③ Design Static Pressure

- Sound levels increase as a function of both airflow and static pressure.
- Do not overestimate fan static pressure and expect the terminal devices to throttle the excess pressure. After air balance is completed, if necessary, change fan sheaves and/or pulleys to supply only the required airflow quantities without over pressurization of the duct system.
- Fans should be selected to operate as near as possible to its rated peak efficiency point.

④ Ductwork at Inlet and Outlet of Fans and Terminals

- Sound breakout depends on aspect ratio and length of duct, sound level inside the duct, and the attenuation effects of the ductwork itself. Breakout may be better if it occurs in an unoccupied or

noisy area. Otherwise, excess breakout noise may radiate through a ceiling over a noise-sensitive area causing a room noise level problem.

- Flex duct and fiberglass ductboard allow significantly more breakout than does metal duct.
- Flex duct should only be used in short sections for connections and vibration isolation since it causes high pressure drops when used in lengths greater than 8 feet.

⑤ Duct Fitting and Elbow- Generated Flow Noise

- Avoid undersized fittings, abrupt transitions, sharp edges, and poor aerodynamics in duct systems, which are all noise-generating sources

⑥ When in Doubt, Consult Design Guides

- ASHRAE
- ARI
- SMACNA
- AMCA

Performing an Acoustical Analysis

The design goal for the receiver has now been established and the sound power of the source established. We now need to determine the sound paths between the source and receiver. Sound energy will be transmitted, refracted, and reflected along each sound path. Let us examine the source-path-receiver concept discussed previously to determine if we can achieve the design goal with the selected sound sources. Sound analysis can be done manually or with a spreadsheet or computer program.

The first step in performing an acoustical analysis is to identify and quantify all of the sound sources. Sound power data is published by the equipment manufacturers and is usually listed in dB by octave band or dBA. Octave band data is the best when performing an acoustical analysis since it allows estimating sound quality as well as sound pressure.

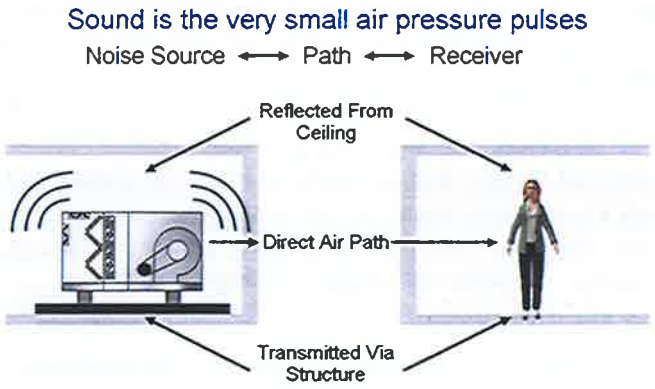


Figure 80

Source – Path – Receiver Model

Indoor Sound Analysis

The illustration in Figure 81 shows three of the most common sound sources for a fan-powered mixing box system.

The most common sound power sources for a fan-powered mixing box system are:

- Unit Casing Radiated
- Unit Discharge
- Outlet Generated

Regardless of which type of equipment you are considering, similar procedures should be followed to perform an acoustic analysis. Identifying the sound sources is the easiest part of performing an acoustic analysis.

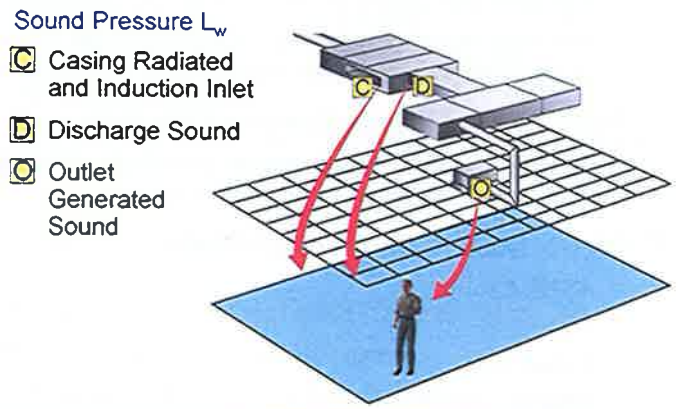


Figure 81

Sound Sources in a Fan-Powered Mixing Box System

The next step of the acoustic analysis is to determine all the expected dominant sound paths. This is all of the various routes that significant sound can travel along from the source to the receiver. Remember, sound levels measured at the receiver are *sound pressure*, not *sound power*. It is through all of these various sound paths that the sound power is attenuated before it reaches the receiver.

Referring to the original fan-powered mixing box example in Figure 81, the three sound power sources may reach the receiver space along a number of paths as shown in Figure 82.

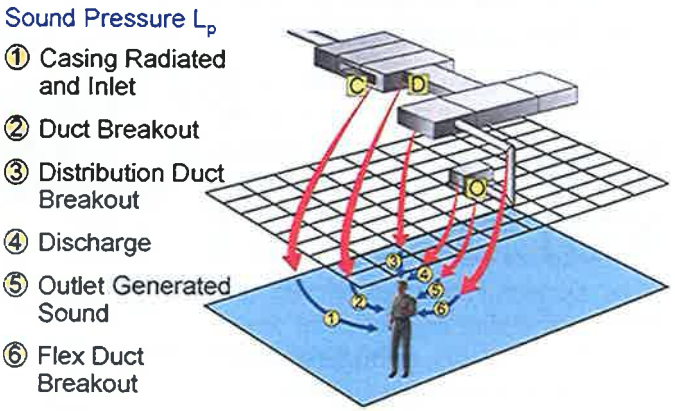


Figure 82

Possible Sound Paths in a Fan-Powered Mixing Box System

Casing-radiated sounds and induction inlet sounds are the sound energies that radiate through the equipment panels and out the induction inlet of the fan-powered mixing box, respectively. Duct breakout is the sound energy that travels down the supply or return ducts and radiates through the duct walls into the adjacent space. Duct breakout is primarily dependent on the geometry (aspect ratio) of the duct and not on the amount of lining or insulation. Rectangular ductwork with large aspect ratios (width/height) produces the most breakout noise. This is due to the large surface area that tends to vibrate rather easily, transmitting and sometimes amplifying sound energy that is present inside the duct, much like a loudspeaker. The use of round ductwork should be considered over noise-sensitive areas where duct breakout could create a noise problem.

Another phenomenon called duct break-in sometimes occurs. This happens whenever noise from one space radiates into a section of ductwork where it may travel long distances to adjacent

areas. Everyone has probably been in a location where you can hear conversation from an adjoining room coming from the diffuser. Unlined duct is a very efficient noise transfer device. When necessary, lined ductwork should be used to prevent noise carryover to adjacent spaces.

For a fan-powered mixing box, as well as ceiling mounted fan-coil units or water-source heat pumps, the casing-radiated and duct breakout noise generally radiates into a ceiling plenum. This noise then travels through the ceiling itself before reaching the space.

Discharge sound is the sound energy that travels down the duct and out the diffuser. Outlet generated sound is generated by the air diffuser itself, independent of the unit sound, and depends on the airflow (velocity) and pressure drop through the diffuser. The difficult task is estimating the contribution of each of these sound paths to the room noise level.

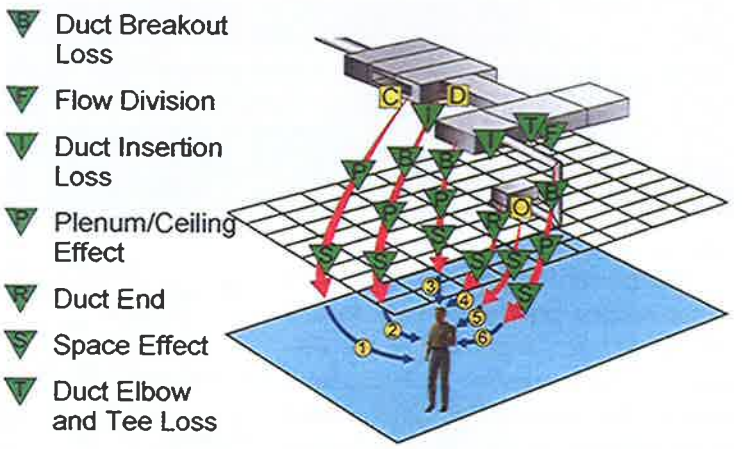


Figure 83

Sound Attenuation Elements in a Fan-Powered Mixing Box System

Once the sound paths have all been identified, the next step is to identify all of the attenuating elements that reduce the sound pressure level at the receiver location. Unlined ductwork attenuates very little sound in comparison to lined ductwork. For the fan powered mixing box example, typical attenuation elements are shown in Figure 83.

Whenever air flows through a branch or tee (flow division), a resulting attenuation in each of the flow branches is assumed to be proportional to the ratio of the flow areas of each branch. Duct lining or porous duct material inside the duct absorbs some of the sound energy. This absorption of sound energy is referred to as *insertion loss*.

Manufactured duct silencers are used primarily to attenuate discharge sound in the ductwork by absorbing higher-frequency noise. Realistically sized sound attenuators are not very effective at attenuating low frequency noise below 63 Hz.

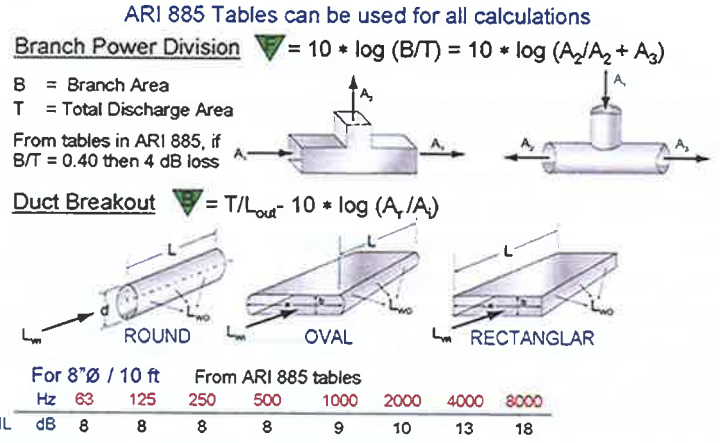


Figure 84

Sound Path Attenuation from Flow Division and Duct Breakout

Terms:

Insertion loss is how much the frequency is reduced in each octave band when a sound attenuation component is placed in a sound path.

Space effect accounts for the conversion of sound power that enters the room to sound pressure at a defined location.

Plenum effect describes the attenuation effect that occurs due to ceiling tile coupled with the adjacent plenum space, which reduces the sound that enters the occupied space.

Whenever ductwork transitions into a sudden expansion, such as at a ceiling diffuser, a portion of the discharge sound energy reflects back towards the source. This phenomenon is referred to as duct end reflection loss. This reflected sound energy results in an attenuating credit, primarily in the lower frequency bands.

Space effect is the attenuating effect of the room itself. Space effect depends on three variables:

R = Distance between noise source and receiver

V = Room volume, ft^3

F = Octave band center frequency, Hz

Elbows and tees attenuate noise by absorption and reflection. Lined elbows and tees attenuate noise much better than unlined fittings. Turning vanes, while reducing static pressure loss, reduce the attenuating effects of the fitting by directing noise energy around the bend and down the duct, therefore the use of turning vanes should be avoided in situations where discharge noise is excessive and maximum attenuation is required.

Casing-radiated and breakout sound travels through the ceiling plenum and through the ceiling into the space. ARI has several ceiling types listed in the various published standards. The more mass the ceiling material consists of, the better the attenuating effect (e.g., sheetrock attenuates better than fiberglass ceiling tile).

When the ceiling material and plenum above it absorbs sound energy, it is referred to as the plenum/ceiling effect.

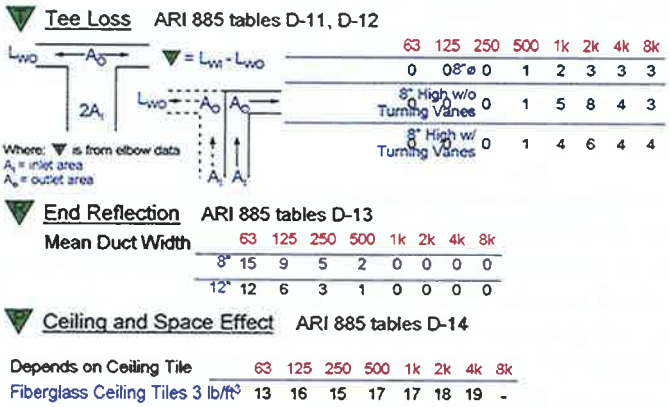


Figure 85

Ceiling Effects and End Reflection Sound Path Attenuation

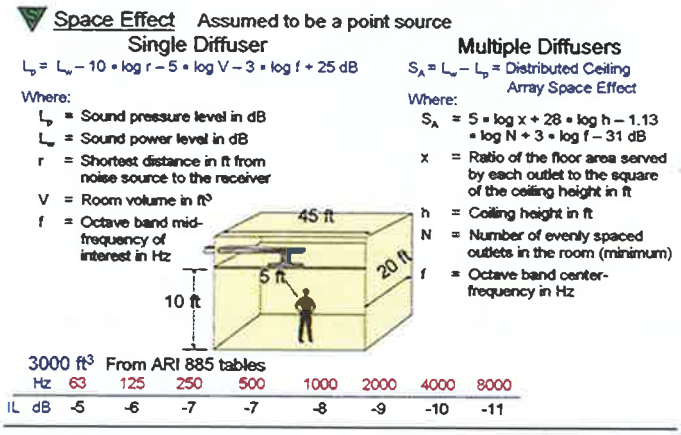


Figure 86

Space Effect

Attenuation elements reduce the noise level along the sound path as shown. Discharge sound (Path 5) through the supply duct is attenuated in direction of airflow, as follows:

- Insertion loss of main rectangular supply duct
- Flow division at tee
- Tee loss
- Insertion loss of rectangular branch duct
- Insertion loss of branch duct
- Duct end reflection at supply diffuser
- Space effect

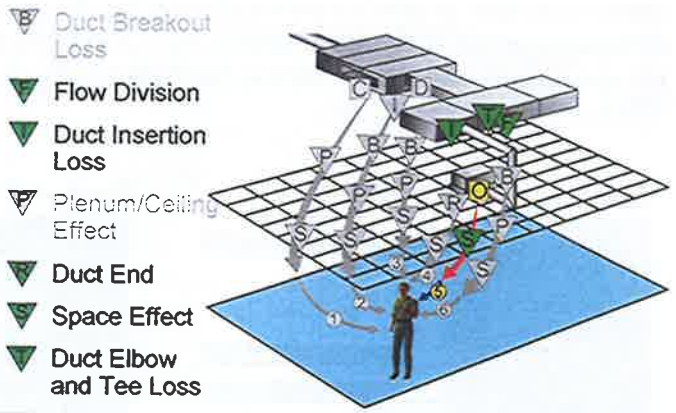


Figure 87

Attenuation Along Path 5

Casing-radiated sound energy (Path 1) is attenuated by the plenum /ceiling effect and the space effect of the room.

Outlet-generated sound (Path 6) is generated by the flow through the diffuser itself, attenuated only by the space effect.

Duct breakout (Paths 2, 3, and 4) occurs from sound energy breaking-out of the supply duct radiating through the ceiling plenum into the space. The ceiling/plenum and the space effect attenuate duct breakout.

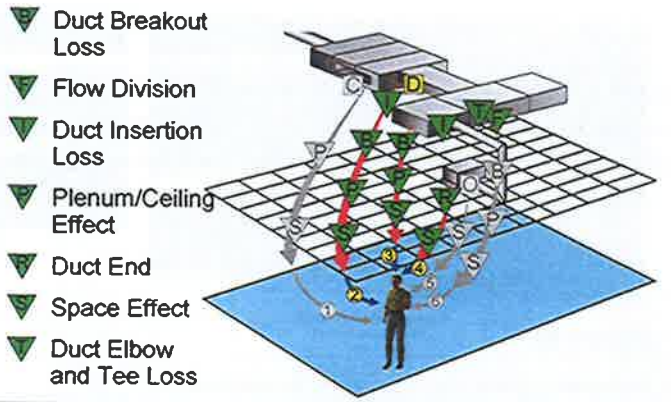


Figure 88

Duct Breakout Paths 2, 3, and 4

Whenever air flows through a restriction, a certain amount of sound energy is regenerated at the point of turbulence. This regeneration creates another sound source, e.g., an undersized air diffuser in a room that creates a rushing or whistling sound. Regeneration depends on the amount of restriction (static pressure drop), air velocity (flow rate), and the presence of turning vanes. When in doubt, add a regeneration element along the sound path to determine if regeneration occurs.

Using the ARI 885 Tables

Procedure	Octave band center frequency, Hz							
	63	125	250	500	1k	2k	4k	8k
1. Diffuser sound pressure level	59	48	40	34	30	27	25	23
2. Room and terminal effect	5	5	5	5	5	5	5	5
3. Allowance for end reflection	12	8	4	1	0	0	0	0
4. Ductwork attenuation, terminal to diffuser	17	17	9	5	5	5	5	5
5. Elbow attenuation, terminal to diffuser	0	0	7	15	25	25	19	15
6. Power-level split, branch to terminals	6	6	6	6	6	6	6	6
7. Power-level split, main duct to branch duct	10	10	10	10	10	10	10	10
Totals	109	91	78	73	78	75	67	61
8. Sound power level of fan - L_w	105	99	98	98	97	95	86	81
9. Safety factors	3	3	3	3	3	3	3	3
Total sound power level of fan - L_w	105	102	101	101	100	98	89	84
10. Silencer requirements	0	8	20	25	19	20	19	20

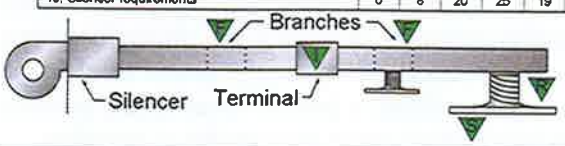


Figure 89

Calculating Sound Pressure Manually with Table

The last step is to calculate the resulting sound pressure (Lp) level in the space. If this exceeds the chosen design criteria, it will be necessary to increase the attenuation from the path that is dominant, or to choose another unit that is quieter, or as a last resort relocate the equipment farther away from the space.

Because these problems become very tedious when calculated manually,

acoustic spreadsheets or other programs are used to facilitate the computations. The good news is that the programs take the pain out of the calculations. The bad news is that the reliability of the answers are only as good as your understanding the proper use of each part of the program. Most of these programs apply the procedure from ARI 885 (described earlier) for this analysis. A simple spreadsheet may also be used to determine the sound pressure level at the receiver. A sample of one is shown in Figure 90.

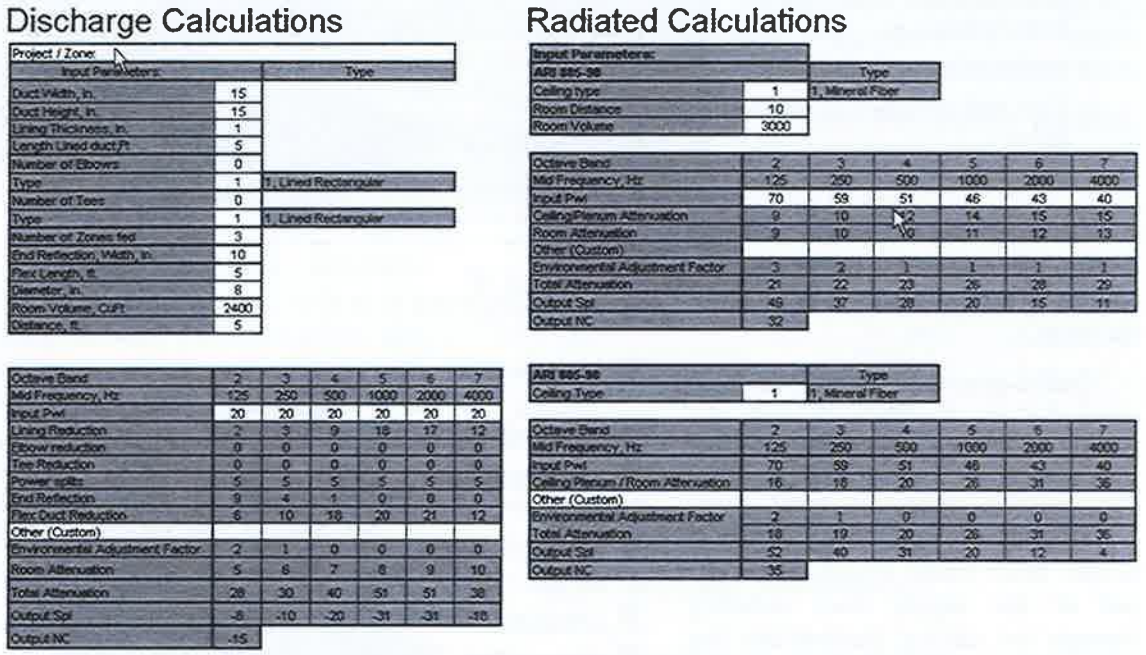


Figure 90

Calculating Sound Pressure with a Spreadsheet

To summarize, the steps to performing an acoustic analysis are:

- Identify and quantify all possible sound sources.
- Identify the important sound paths to be considered.
- Identify all attenuating elements along sound paths.
- Subtract the attenuation effects from the source sound power (Lw) to get the estimated room sound pressure (Lp) level.
- If the room noise level exceeds the desired value, do one of the following: add additional attenuation, reduce the source noise level by relocating the equipment farther away, or select a quieter piece of equipment.
- Check areas adjacent to mechanical equipment rooms for noise transmission through walls.
- Remember to repeat the analysis for the return side of the system.

It is often best to work backwards; in other words, start with desired room noise levels using room criteria (RC) data, then backward-calculate the maximum allowable equipment sound power (Lw) output based on the assumed attenuation elements. Knowing the allowable sound power output will allow the designer to establish a minimum allowable sound power level which can be written into a specification, rather than simply specifying a desired room sound pressure (RC) level and leaving the responsibility of the acoustic analysis up to the installer or supplier.

Outdoor Sound Analysis

When a design goal has been established for outdoor sound, units such as cooling towers, chillers, rooftop units, outdoor condensing units, and even large exhaust fans must be considered for their contribution to sound levels.

Outdoor sound propagation is reasonably close to free field and for many applications has an assumed directivity factor of 2 (see Figure 23). Directivity is difficult to account for in the calculations when the sound source does not approximate a point source. Several natural outdoor phenomena also influence the sound path to the receiver. These include influences from distance, topography, background sound, atmospheric absorption, temperature, and wind. These are also hard to account for and are often ignored in the calculation. These factors help to explain why field measurements may not match estimated data. ARI Standard 275 can be used to calculate the sound pressure level at the receiver.

Figure 1 Finding Community Reaction

Influencing Factor	Possible Condition	Correction
Noise Spectrum	Pure tone Components	- 5
	Wide band noise	0
Repetitiveness	Continuous to one/min	0
	10 to 60 times/hr	+ 5
	1 to 10 times/hr	+10
	4 to 20 times/day	+15
	1 to 4 times/day	+20
	1 time/day	+25
Time of Day	Daytime only	+10
	Evening	+ 5
	Nighttime	0
Season	Winter only	+ 5
	Winter and Summer	0
Type of Area	Rural	-10
	Suburban	- 5
	Residential (urban)	0
	Residential (some business)	+ 5
	Area of light industry	+10
	Area of heavy industry	+15
Peak Factor	Impulsive	- 5
	Non-impulsive	0

N1	Community Response
Less than 40	No reaction
45	Sporadic complaints
50 to 55	Widespread complaints
60 to 65	Threats of community action
70 and above	Vigorous community action

Figure 2

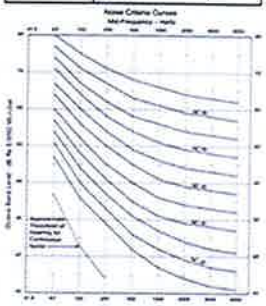


Figure 3

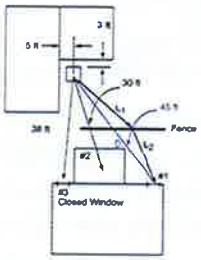
Figure 91

Community Reaction to Sound

The procedure can be summarized as:

- Obtain the equipment sound power rating
- Add the equipment location factor from table 1 of ARI 275
- Find the barrier shielding factor and subtract it from Step 2
- Subtract the sound path factor
- Subtract the distance factor
- The result is the expected sound pressure at the receiver location

An example calculation is show in Figure 92. All the required factors can be found in the ARI 275 Standard. When multiple sound sources are present, first determine the value to add to account for the multiple sources.



Barrier Calculation
D = 45 ft
L_1 = 30 ft
L_2 = 18 ft
$L = L_1 + L_2 - D$
L = 48 - 45 = 3
Barrier Shielding Factor = 12

Line	Distance from equipment to evaluation point 1. 45 ft (13.7m) 2. 30 ft (9.1 m) 3. 38 ft (11.6 m)	Evaluation Points		
		1	2	3
1	Unit Sound Rating Level (ARI Standard 270)	71	71	71
2	Equipment Location Factor (table 1, item 1)	6	6	6
3	Add Lines 1 and 2	77	77	77
4	Barrier Shielding Factor (table 1, item 2)	12	0	0
5	Sound Path Factor (table 1, item 3)	0	0	17
6	Distance Factor (table 2)	30.5	27	46
7	Add Lines 4,5, and 6	42.5	27	31
8	Estimated A-Weighted Sound Pressure Level (Subtract Line 7 from Line 3)	28.5	50	31
9	Estimated A-Weighted Sound Pressure Level Rounded to Nearest Whole Number	29	50	31

L_1 = Distance from source to barrier
L_2 = Distance from top of barrier to window (receiver)

Figure 92

Example Outdoor Sound Analysis

The ARI standard is based on making corrections to a single number dBA rating, when octave band frequency data is required use the equation below and the directivity factors explained previously.

$$Lp = Lw + 10 * Log_{10}\left(Q/_{4\pi r^2}\right) + 10.5$$

Where:

Q = Directivity factor

Q = 0, no reflecting plans present (free field)

Q = 2, one reflecting plane (free field above a reflecting plane)

Q = 4, two reflecting planes (free field above a reflecting plane with a reflecting wall)

Q = 8, three reflecting planes (free field above a reflecting plane with two reflecting walls)

	63	125	250	500	1k	2k	4k	8k
Unit Sound Power	100	97	95	93	92	88	87	88
Correction Factor	35.5	35.5	35.5	35.5	35.5	35.5	35.5	35.5
Sound Pressure*	64.5	61.5	59.5	57.5	56.5	52.5	51.5	52.5

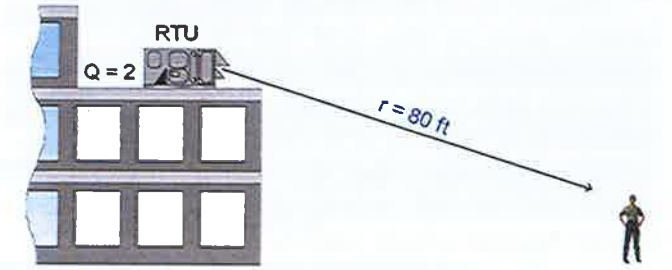


Figure 93

Rooftop Outdoor Frequency Analysis

R = distance from the acoustic center of the source to the location of the receiver in feet

π = 3.141

Lw = sound power level, dB

Lp = sound pressure level, dB

An example of the frequency band calculation is shown in Figure 93.

Obviously, an acoustic analysis does not need to be performed on most jobs. However, like many other aspects of HVAC, acoustic analysis is as much an art as it is a science. Through proper design, application, and installation practices, noise problems may be avoided. Fortunately, there are software programs available that will complete much of the manual calculations involved with an acoustic analysis.

Guidelines for Equipment Selection and Application

The following sections discuss general guidelines and illustrate various methods to assist in the design, application, and installation of HVAC products with the intent of reducing noise in the occupied space. The purpose of this section is to identify some of the more common application problems and recommended solutions and is not intended to be a product-specific design guide. Manufacturer's acoustic application data should be consulted prior to design, application, and installation of any particular piece of equipment.

Rooftop Units

Packaged rooftop units and rooftop air-handling units pose a unique and challenging design and application problem since these units are often utilized in lightweight roof deck constructions. Supply and return ductwork requires large roof penetrations within the roof-mounting curb that compounds the problem. Often ductwork is relatively short and there is insufficient room to properly apply noise-attenuation measures.

Typical Sound Data	63	125	250	500	1k	2k	4k	8k
Supply	91.1	91.2	89.4	83.8	76.8	70.4	62.6	52.7
Return	83.1	73.2	70.4	63.8	59.8	59.4	49.6	36.7
Outdoor	100	97	95	93	92	88	87	88

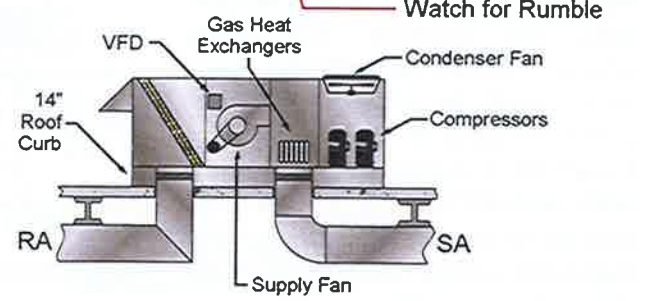


Figure 94

Rooftop Unit Sound Recommendations

The primary noise sources in rooftop unit installations are the fan(s), compressors and the turbulent airflow within the unit cabinet and supply ductwork.

The trend toward using variable air volume (VAV) systems has intensified the problem because, in these systems, the unit fans often run at lower speeds that reduce the high-frequency noise. This causes the occupied space to be rich in low frequency sound energy. This reduction in high-frequency noise creates an unbalanced sound spectrum whereby the low-frequency noise is much more detectable. Low-frequency noise (below 125 Hz) is generally associated with system rumble and is very difficult, and sometimes very expensive; to attenuate once the system is installed and running. Therefore, avoiding system rumble should be addressed during the design phase.

Selecting a relatively quiet unit during the design phase does not necessarily avoid acoustic problems. Acoustics problems are most often system problems. All components of the overall system must be considered if a quiet installations to be ensured.

Eliminating low-frequency duct rumble on an existing job usually involves major fan and ductwork modifications and/or major changes to the acoustical ceiling structures.

The first and most important consideration for roof-mounted units is the structural support of the roof itself. Placement of the unit in conjunction with a vibration isolated roof curb is recommended to control fan and compressor noise. Figure 95 illustrates recommended installation parameters to reduce noise transmittance into the conditioned space.

It is recommended that a room criteria (RC) or noise criteria (NC) rating analysis be performed for each type of building construction.

The following section outlines design and installation guidelines to reduce noise transmission into the conditioned space and does not cover duct design practices. Refer to the Ductwork section of this manual for more information.

Rooftop Design and Installation Guidelines

The roof structure should be made as stiff as possible. A maximum roof deflection of 1/4-inch during dead loading and unit operation should be permitted. A minimum of 20-foot column spacing is recommended in vicinity of unit.

Locate units at least 25 feet from any noise sensitive area, preferably over restrooms, storage areas, or other noise tolerant areas. If this is not possible, the ductwork and ceiling structure should be stiffened. Ductwork should be stiffened by adding additional mass or with lagging, see Figure 96. Ceilings should be hung with resilient mounts and be constructed of gypsum board rather than suspended ceiling tiles that offer little attenuation properties.

Supply and return air main trunk ducts should be located over acoustically non-critical spaces, such as hallways and/or public areas.

Mount unit on prefabricated vibration isolated roof curb with minimum 2-in. deflection. Factory-installed internal isolators are usually not adequate.

Utilize flexible connectors on ductwork, piping and electrical connections.

1. Max 20 ft span with ¼ in. deflection
2. Location 25 ft from noise sensitive spaces
3. Locate over hallways or service spaces
4. Use a high mass ceiling isolated from building

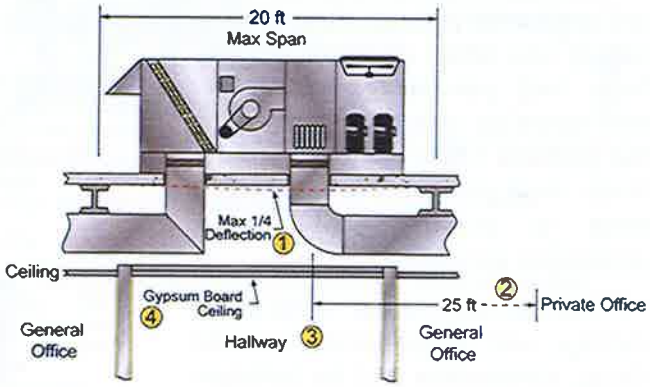


Figure 95

Rooftop Unit Installation Guidelines

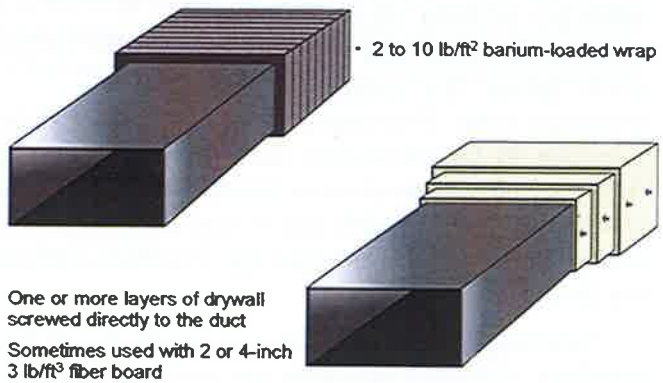


Figure 96

Stiffening Rectangular Ductwork

For units 20 tons and larger:

Use side supply units where the supply fan discharges into a supply plenum. This is common construction for gas-fired units where the air passes through a lined discharge plenum.

Supply and return air main trunk ducts should be located over non-critical hallways and/or public areas.

Return fans should be avoided unless return air ductwork is extremely long and restricted. Return fans increase the total energy required as well as the total sound power generated.

For noise-sensitive areas, mount the unit on high-deflection spring isolators connected to a steel support structure to support the unit with at least 2 in. static deflection.

If unit is located over a noisy area, consider eliminating the acoustic lining on the first sections of the supply ductwork to allow duct breakout to occur over the noisy area. This will dissipate much of the low-frequency noise close to the unit where it is unobjectionable.

Proper sizing of units should be performed by calculating a load estimate on the area served to account for diversity of the load. An oversized unit increases the probability that the fan will be operating to the left of the peak efficiency point on the fan curve, which can result in increased low-frequency rumble and fan surging. Undersized units require higher fan speeds to deliver the required airflow and static pressure, which results in increased fan noise, particularly in the higher frequencies.

To avoid rumble, never operate the unit supply fan in the region less than 10-15 percent of wide-open cfm. Change fan pulleys and sheaves to

1. Use isolation curb on noise sensitive applications
2. Insulate duct on supply and return (except if supply is to near critical space)
3. Allow at least 10 feet to the first diffuser
4. Provide Plenum Space
5. Stiffening Ductwork

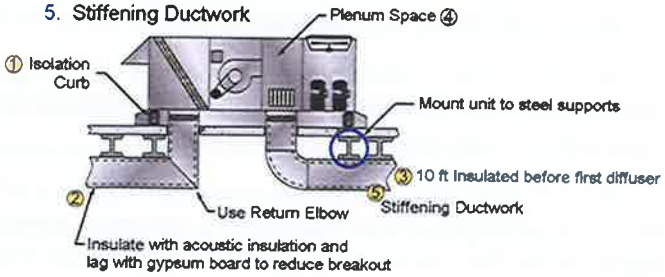


Figure 97

Rooftop Unit Sound Guidelines

1. Provide flex connections on all connections
2. Avoid return fans
3. Minimize roof penetrations and seal gaps with flexible calk

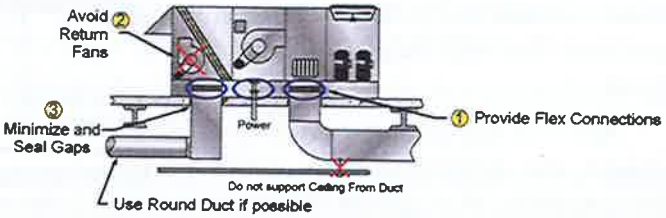
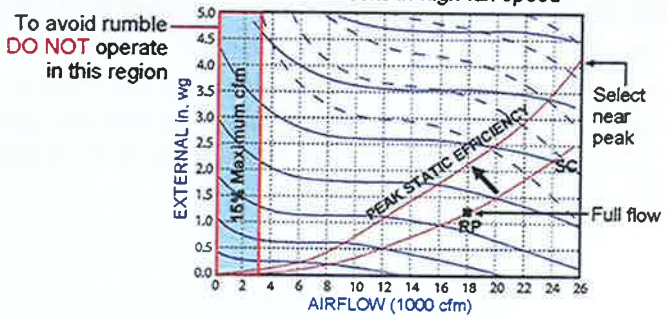


Figure 98

Rooftop Unit Sound Guidelines

- Size units for load using diversity
  - Oversized results in surge
  - Undersized results in high fan speed



- Use VFD rather than inlet guide vanes
- Be careful of sound generated by VFD

Figure 99

Rooftop Unit Sound Guidelines

provide only the airflow required meeting the load on the unit. A 10% reduction in fan speed can result in a sound level decrease of 3 dB.

Utilize round duct for the supply duct if possible. Round duct allows much less breakout noise to occur than rectangular duct.

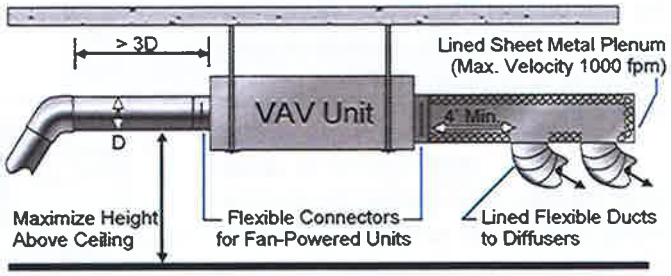
Make sure ceiling and wall subcontractors do not attach hangers or supports to ductwork.

Minimize size of roof openings to be only slightly larger than ductwork. Properly seal all roof penetrations with non-hardening caulk. For additional attenuation, curb area under unit should be lined with gypsum board and caulked airtight.

If VAV units are used, utilize variable-frequency drives (VFDs) rather than inlet guide vanes (IGVs) or other fan control methods to minimize generation of noise and maximize energy savings from the fan. However, VFDs can also generate noise.

Room Terminals

Room terminal units are the final element in the HVAC system and allow excellent zoning of the building to be achieved. They seldom pose noise problems in most general office situations. However, when located directly above or adjacent to noise-sensitive areas, they can add to the overall noise level in the space. Figure 100 indicates some common design and installation practices that will help to minimize noise.



NOTE: Parallel or series fan-powered units often require inlet attenuators to control fan noise.

Figure 100

Terminal Sound Paths

Part 3 of this training module discussed the application and sound characteristics of a typical fan powered mixing box system along with possible sound paths and attenuating elements. Reader should refer back to this section for specific sound source, path, and attenuation information.

The following section outlines design and installation guidelines to reduce noise transmission into the conditioned space and does not cover duct design practices. Refer to the Ductwork section of this manual for more information.

Design and Installation Guidelines

Establish desired room criteria (RC) level for space under design by referring to the latest ASHRAE HVAC Applications Handbook.

Obtain sound power data from various manufacturers and select the quietest unit that will supply the required airflow.

For open plenums with non-ducted return grilles in a mineral-tile suspended ceiling:

- Select each unit so that the radiated noise at design airflow rate is at least 5 dB less than the required room RC level.
- Locate unit away from ceiling yet as high as possible and at least 5 feet from any return air grille.
- If unit must be installed within 5 feet of a return grille, a fiberglass-lined sheet metal elbow may need to be installed on top of the grille, pointed away from terminal unit.

If unit is installed above a gypsum board ceiling and/or the return air system is ducted, the terminal unit may be selected for a radiated sound level 5 dB greater than the required room RC level.

Do not locate a VAV unit over a space with a required RC rating of less than 40 (conference rooms, private offices, etc.).

Do not locate a fan-powered mixing box over a space with a required RC rating of less than 40. Instead, locate unit over a non-sensitive place such as a janitors closet, restroom, or corridor.

Make sure there are at least three equivalent duct diameters of straight duct immediately upstream of the VAV unit on medium and high-pressure systems.

Use short pieces of lined flexible duct to connect supply ducts to inlet of fan-powered mixing boxes. In addition, the use of up to 6 feet of lined flexible duct downstream of a terminal unit is generally beneficial in reducing noise levels without causing duct breakout noise to be a problem.

Low-pressure ductwork leaving a VAV or fan-powered mixing box should be fiberglass lined sheet metal, not unlined sheet metal or ductboard. Ten feet of lined duct may reduce the sound level in the duct by as much as 40 dB in the mid frequencies. However, lining is ineffective at reducing low-frequency noise.

Duct taps and runouts should be at least 4 feet from the unit outlet. When a discharge duct splits into two equally sized ducts, a 3 dB reduction in noise is to be expected.

Minimize total duct pressure in the system to a level that will ensure controllability of the VAV box without producing excessive duct pressure that must be reduced by the VAV air valve. The air valve is a noise generator dependent on both airflow rate and pressure drop across it.

Fans and Air-Handling Units

Improper selection, application, and installation of fans and air-handling units (AHUs) are the main causes of noise problems in HVAC systems. Selecting the most "acoustically efficient" unit that will deliver the required cooling and/or heating capacity to condition the space is important if noise problems are to be avoided.

The illustration in Figure 101 indicates some common design and installation practices that will help to minimize noise.

The numbers in Figure 101 correspond to the following notes:

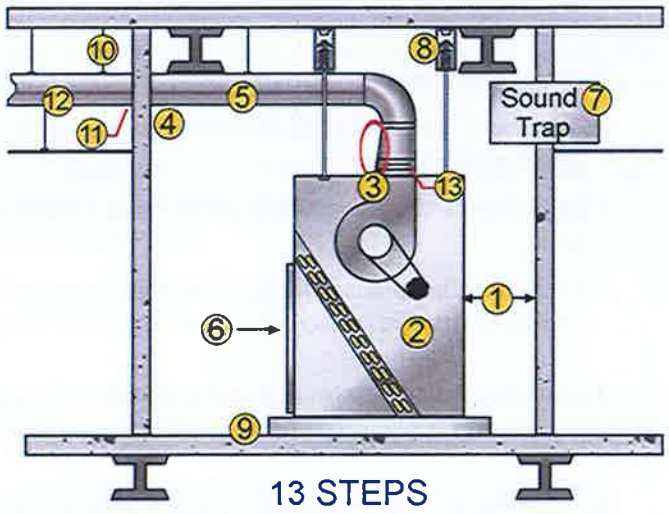


Figure 101

Air Handlers

(1) Keep a 2-foot clearance between unit and wall. A masonry wall will attenuate low frequency noise best.

(2) Use of horizontal AHU discharge eliminates requirement for turbulence-producing elbow.

(3) Use ASHRAE recommendations for gradual transition, 15° or less, at AHU outlet to minimize turbulence.

(4) Sound trap is far enough from AHU outlet to avoid excessive regenerated noise and turbulence.

(5) Round ductwork controls the transmission of low-frequency noise and rumble into the occupied space.

(6) The large clearance at the AHU inlet keeps the unit away from the walls and avoids excessive inlet turbulence.

(7) Return air sound trap to control AHU noise down return air path.

(8) Use high-deflection, steel spring vibration isolators under floor mounted units, and brackets on suspended units.

(9) The floor assembly supporting unit is resting on a housekeeping pad and at least one support beam which provides adequate mass to reduce structure-borne transmitted noise.

(10) Chilled water or DX piping suspended with vibration isolator hangers.

(11) The discharge duct does not touch the wall. A ½-in. gap surrounding the duct is filled with a non-hardening sealant.

(12) Ceiling not suspended from supply duct.

(13) Use canvas-type or elastomeric flexible connectors to attach ductwork to fan inlet and outlets (on units that do not have internally installed flexible connections between the fan discharge and the unit casing).

There are three types of noise generally associated with central station AHUs: discharge sound, radiated sound, and return air sound. Most manufacturers publish their catalog ratings with sound data for each of these three types. Discharge sound is the sound that exits the unit outlet and travels down the supply ductwork to the conditioned space. It is sometimes referred to as airborne sound. Radiated sound is the sound energy that emits from the unit casing or supply ductwork and travels either through the mechanical room walls or through the ceiling plenum to the space. Return air sound is the sound energy that is generated at the inlet of the AHU and travels down the return air duct to the space. Sound energy can propagate in the same, or opposite, direction as the airflow. The return duct system should not be overlooked when doing an acoustic design or analysis. Some types of indoor air-handling units contain the compressor section as well; guidelines in this section may be applied to these units.

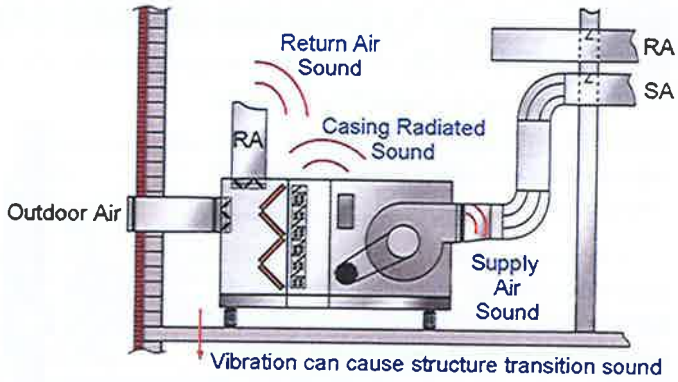


Figure 102

Sound Propagated by an Air Handler

The following section outlines design and installation guidelines to reduce noise transmission into the conditioned space and does not cover duct design practices. Refer to the Ductwork section of this manual for more information.

Design and Installation Guidelines

Select the most efficient type of AHU for the application. An efficient unit will translate into a quieter unit since noise is generated by system inefficiencies.

Select the fan to operate on the right side of the fan curve near the point of peak efficiency, safely away from the stall region. On VAV systems, be sure to consider the part-load point at which the fan will operate the majority of the time.

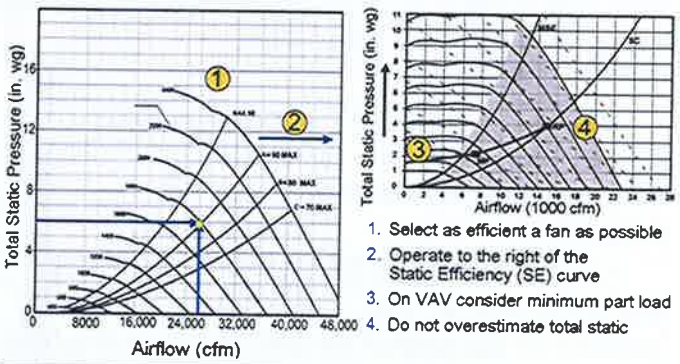


Figure 103

Air-Handling Unit Fan Guidelines

Do not overestimate external static pressure and oversize the fan. More fan is not necessarily better since it will not operate at its peak efficiency point, which will result in higher sound levels and possible rumble problems. Likewise, an undersized fan will have to operate at a higher rpm to deliver the required airflow to the space; hence creating higher sound levels at higher frequencies.

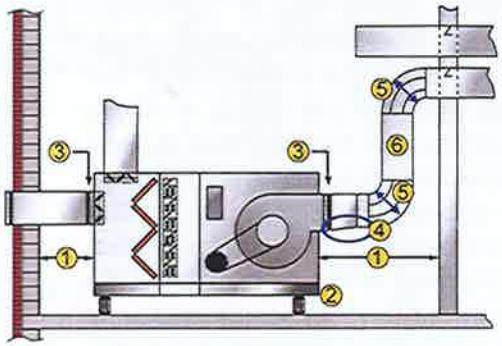
Allow clearance between the unit and the nearest wall or other obstruction of at least one fan wheel diameter for unducted inlets and one and one half fan wheel diameters for all unducted outlets.

For VAV applications, the use of a variable frequency drive (VFD) is recommended over other methods of fan control such as inlet guide vanes (IGVs), discharge dampers, and riding the fan curve. VFDs generally allow an 11 dB sound level reduction at 65 percent of full load fan speed. As compared to a system using IGV control at the same speed (65 percent), the VFD system can be as much as 17 dB less.

Use external spring-deflection type vibration isolators or hangers if internal isolators are not available from the manufacturer.

On units that do not have internally-installed flex connections between the fan discharge and the unit casing, use canvas-type or elastomeric flexible connectors to attach ductwork to fan inlet and outlets (be sure they meet National Fire Protection Association-90A Standard for smoke generation).

Duct transitions on inlet and outlet should be gradual and be at an angle of no more than 15 degrees (consult SMACNA guidelines).



1. Allow 1 to 1½ fan diameters
2. Use Spring Isolators
3. Provide flex connections
4. Fittings should have 15° or less angle
5. Use long radius elbow with turning vanes
6. Follow SMACNA recommendations

Figure 104

Central Station Air Handler Sound Control Guidelines

A minimum of three equivalent duct diameters of straight duct should be installed on the supply and return side prior to installing any duct fittings, transitions, or sound traps.

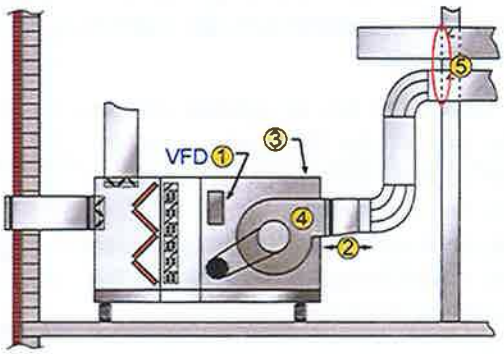
Utilize long radius elbows whenever possible.

Consider using an AHU with double-wall construction for noise-sensitive areas. A solid-faced liner will reduce casing-radiated sound but increase return air sound by 3-5 dB in the mid-frequencies (which are easily attenuated by the duct system).

Blow-thru AHUs offer significant acoustic advantages over draw-thru units, specifically by reducing the discharge sound.

Consider adding a discharge plenum lined with at least 2-in. thick insulation on the outlet of the AHU to reduce discharge noise. This allows some of the low-frequency noise to breakout at the unit rather than down the duct system.

Seal all gaps where ductwork penetrates mechanical equipment room walls with non-hardening caulk.



1. Use a VFD on VAV units
2. Allow 3 duct diameters before a duct fitting
3. Consider double wall AHU
4. Blow through units offer best attenuation
5. Seal all gaps between room walls and ductwork

Figure 105

Additional Central Station Air Handler Guidelines

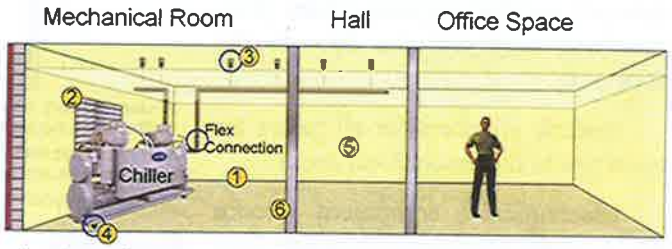
Indoor Equipment and Mechanical Rooms

Indoor equipment such as central plant chillers and air compressors produce some of the loudest sounds that the acoustical engineer or designer may ever have to deal with. The following recommendations will help attenuate noise of indoor equipment located in mechanical equipment rooms.

If possible, locate the central plant away from any noise-sensitive areas.

Attenuate ventilation air passages with acoustical louvers, duct silencers, or lined plenums to control noise escaping to the outside.

Install all central plant equipment and piping on vibration isolators. (Consult the latest version of ASHRAE Systems & Application Handbook for vibration isolator sizing and selection of type of insulator for vibration isolation.)



1. Locate central plant away from critical spaces if possible
2. Use Acoustical louvers
3. Isolate piping
4. Provide vibration isolation for equipment
5. Provide a vestibule (hall) between equipment room and occupied spaces
6. Consider adding absorption to walls if noise levels of chiller is a concern

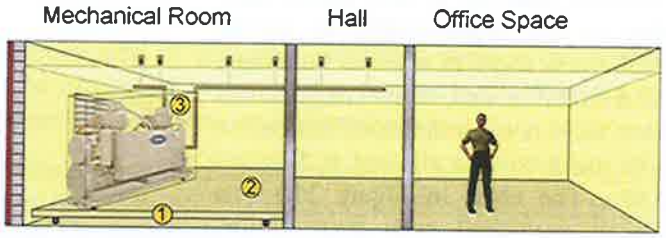
Figure 106

Mechanical Room Sound Recommendations

For vibration isolation, use slack, flexible conduits for all electrical connections to equipment. In addition, only penetrate slab with electrical conduit over areas that are not noise sensitive.

For upward noise control of equipment, consider using a floating ceiling or a noise enclosure around equipment, provided adequate ventilation air is not impeded. For downward noise control, use vibration isolators on the suspended ceiling located below the equipment room in addition to the building slab between floors or use a floating floor.

For noise control both upward and downward, use a monolithic slab weighing at least 100 pounds per square foot (psf).



1. Consider a floating floor or floating ceilings for critical areas
2. Use monolithic slab of at least 100 lb/ft^3
3. Provide noise enclosure if required

Figure 107

Recommendations to Isolate a Mechanical Equipment Room

For critical noise-sensitive areas, use floating floor or floating ceiling assemblies.

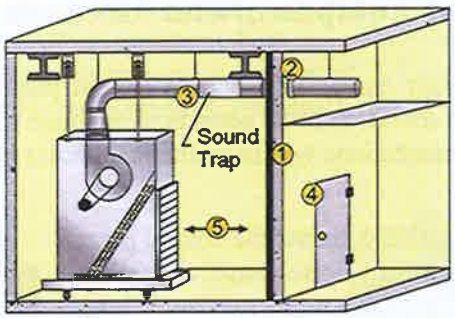
For AHU rooms, the floor area of the room should be from 10-15 square feet per 1,000 cfm of unit airflow.

When possible, consider a vestibule between the mechanical room and the occupied space.

Doors should weigh at least 5 pounds per square foot and have full perimeter seals. Doors should hinge outward so that negative room pressure will help seal the door to the doorjamb.

Specify silencers for all return air openings to the mechanical room.

Mechanical equipment room walls can be lined with fiberglass liner (1-4 inches, depending on noise the equipment contained in the room).



1. Use masonry walls with 1 to 4 in. fiberglass insulation on walls
2. Resilient sleeves on all penetration and seal all gaps
3. Provide sound traps
4. Use doors of at least 5 lb/ft^3 with door seals and hinges out
5. AHU room should have a area of 10 to 15 ft^2/1000 cfm

Figure 108

Indoor Unit Sound Control Recommendations

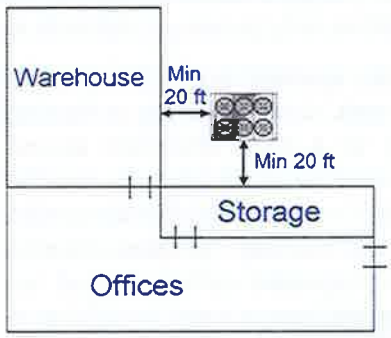
All mechanical room wall penetrations should be resiliently sleeved. A small crack or hole to the outside will allow large amounts of noise to escape.

Outdoor Equipment

Outdoor equipment should be carefully selected so that sound radiated is not objectionable to people outdoors or those in adjacent buildings. As a matter of fact, many communities have local noise ordinances that define how much noise is allowed at different times. The chart in Figure 110 lists typical municipal code noise limits. Local codes should be consulted since differences often exist.

As we discussed in the section on determining outdoor sound levels, the inverse square law can be used to calculate outdoor sound levels. That is, in a free field with no reflecting surfaces, each time you double the distance between the sound source and the receiver, you attenuate the sound level by approximately 6 dB.

Outdoor sound analysis is not always this easy. Many reflecting surfaces exist, such as roads, trees, variable terrain, and adjacent buildings that make an accurate sound analysis very difficult to perform. Therefore, it is often advantageous to



- Place units on side of building away from sensitive spaces
- When possible place minimum of 20 ft from reflecting surfaces
- Use quietest unit possible
- Use barriers
- Use low sound options and attenuators if necessary

Figure 109

Outdoor Unit Location Recommendations

	Maximum Sound Level (dBA)	
Type of District	7am to 7pm	7pm to 7am
Single Family Residential	50	45
Multi-Family Residential	55	50
Commercial	60	55
Industrial	70	70

Note: Sound levels listed are maximum values as measured at the property line.

Figure 110

Typical Municipal Code Noise Limits

attenuate the noise at the source by selecting a unit with the lowest sound power data. In addition, many manufacturers offer accessory sound attenuation on the equipment such as sound baffles on condenser fan outlets, compressor sound blankets, and condenser coil attenuators. These may be marginally effective, especially if the sound is dominated by distinct tones (tonal noise).

Often the easiest way to prevent objectionable sound from reaching adjacent occupants is by installing a sound barrier around the equipment. This must be done in a way that does not restrict airflow or violate service clearances around the unit.

Most outdoor sound energy travels in straight lines. However, diffraction of the sound occurs around corners and over the edge of barriers as shown in Figure 111.

Calculation of insertion loss associated with different barrier wall constructions can be determined from the ASHRAE Guide.

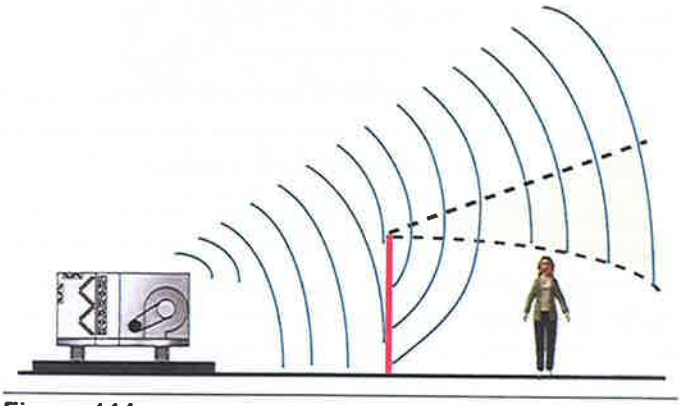


Figure 111

Outdoor Sound Barrier

Ductwork

Controlling noise in ductwork is generally one of the most important steps in the overall acoustical design process. If noise cannot be attenuated at the source, the ductwork provides another means of reducing the noise level before it reaches the conditioned space.

Proper duct design and installation practices must be followed to reduce noise. Guidelines as published by SMACNA and ASHRAE must be followed. Poorly designed or constructed fittings will result in regenerated noise. Remember, an inefficient duct system will generate unwanted noise, completely independent of the noise level at the source. This is why the entire system must be considered, not just the HVAC unit, when performing an acoustical analysis

As discussed previously in this training module, the addition of internal duct lining offers excellent attenuating properties in the higher frequency ranges; however, it does little or nothing for low frequency noise (less than 125 Hz). Adding mass or stiffness to the duct system itself may attenuate low-frequency noise. This may be accomplished either by strengthening the structural supports of the duct walls or by adding mass such as a gypsum board covering to the outside. Round ductwork is best for attenuating low-frequency rumble and preventing duct breakout. When

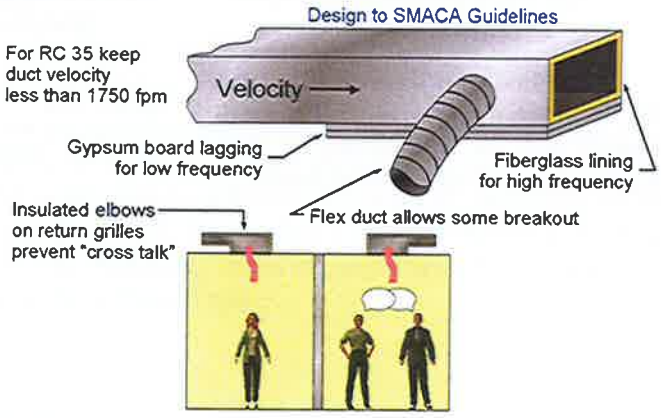


Figure 112

Ductwork Sound Recommendations

space does not permit the use of a single round duct, consider using two smaller round ducts in parallel. Duct breakout will occur rather easily in both fiberglass ductboard and flexible round duct. Sometimes breakout is allowed over noisy or unoccupied areas. However, it should be avoided or at least considered over occupied areas.

When common ductwork connects two adjacent rooms, be sure to consider the addition of a sound trap or lining to reduce *crosstalk*, i.e., the ability to hear sounds from adjacent rooms through the duct system.

Air velocities in ductwork should be kept as low as possible to prevent regenerated noise and rumble problems. This will also reduce the static pressure drop of the duct system thereby reducing the required fan speed that further reduces noise.

For an NC-35 rating with ductwork located above a suspended acoustical ceiling, air velocities should be maintained at 1,750 fpm or less for rectangular and 3,500 fpm or less for round ductwork. If the ceiling is constructed of solid drywall, velocities should be held to 2,500 fpm and 4,500 fpm, respectively. For the same NC-35 rating, when using square elbows, velocities should be held to 1,000 fpm or less. Short radius and square elbows with short turning vanes may handle velocities up to 1,300 fpm, while square elbows with long turning vanes and radius elbows with turning vanes can be designed for maximum velocities of 1,700 and 1,800 fpm, respectively.

An acoustical silencer is a prefabricated section of ductwork that contains a series of parallel or concentric baffles filled with a sound attenuating material. The acoustic performance of a sound attenuator is a function of the size, shape, and length of the internal baffles. In general, the longer the baffles are the more attenuation will occur. However, the baffles create a restriction to airflow and increase the static pressure loss of the system. The performance information for sound traps includes pressure loss and the insertion loss on each frequency band. Listed below are a few recommendations for sound traps.

- On centrifugal fans the trap should be located at least one wheel diameter downstream for each 1000 cfm. The duct transition angle should be 15° or less.
- On axial fans, the sound trap may be attached directly to the fan. The duct transition angle should be 30° or less.

The sound trap should be located in the mechanical equipment room usually near the wall before the ductwork enters the conditioned space.

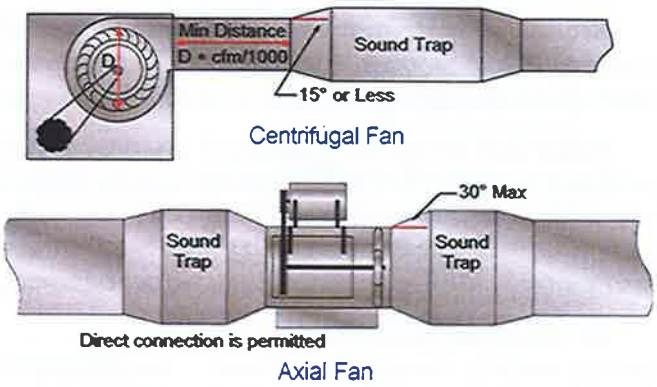


Figure 113

Sound Trap Recommendations on Fans

Acoustic plenums are another way of providing significant low frequency sound control. This method is applied in some air-handling units with plenum fans as an excellent sound reduction method. A diagram of an acoustic plenum is shown in Figure 114. The plenum is lined with acoustic material to absorb high frequency sound and the plenum dimensions act to reflect low frequency sound and help cancel it out using the wave interference phenomena discussed at the beginning of this module.

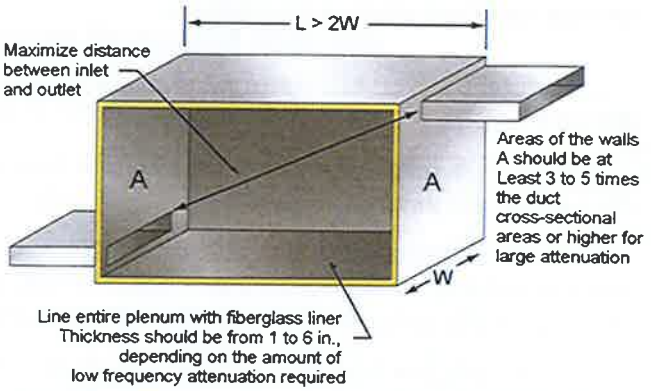


Figure 114

Acoustic plenums can be used for low-frequency sound control.

Grilles, Registers, and Diffusers

Because air terminal devices are located in the room next to the occupants, special care should be taken to control generated noise.

Specifications generally list required NC levels in the room. Manufacturer's data consists of estimated NC levels based on flow and unit size with an assumed "space effect" of 10 dB. Most diffuser noise occurs in the 1000 - 4000 Hz octave bands. This noise, when provided in controlled levels in conjunction with the supply fan noise, provides a more balanced, broadband sound spectrum in the room. This presence of high frequency noise will often help offset the presence of undesirable low-frequency noise by providing a more balanced overall noise level (remember a ***balanced sound*** spectrum is often unobjectionable even a moderate levels). For VAV systems, however, this high frequency cannot be relied on to balance the spectrum since the system will not operate at full capacity except for a few hours each year. Because of variability of air supplied, the noise generated also varies as the HVAC load varies. Constant volume (CV) systems have an advantage in this instance. The following checklist identifies some design and installation guidelines to help reduce noise in the occupied space.

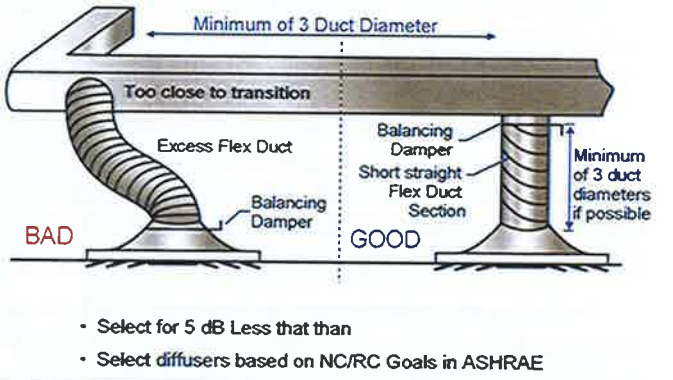


Figure 115

Grilles, Registers, and Diffuser Recommendations

Design and Installation Guidelines

Establish the NC or RC ratings for all noise sensitive areas in the building (consult ASHRAE design criteria).

Select each supply and return air device to have a manufacturer's NC or RC rating at least 5 dB less than the desired NC rating.

Supply ductwork serving an air device should be straight for at least three equivalent duct diameters upstream of the device's duct collar. If not, uneven airflow will enter the device causing additional noise and poor air distribution into the room.

Do not attach a balancing damper directly to a device's duct collar. Instead, it should be at least 3 equivalent duct diameters upstream of the device. A balancing damper connected to a air device's duct collar can add as much as 5 dB to the overall noise level.

Identifying Problems and Recommended Solutions

The sources of noise are as varied as the number of different noise-generating sources. The number one priority in noise control is to reduce the noise level at the source if possible before attempting to attenuate it along the paths.

One of the most common misconceptions about acoustics is that the addition of duct lining and sound attenuators can provide all of the required attenuation. As discussed previously, low-frequency noises are mostly unaffected by lining and silencers.

The first step in attempting to analyze a noise complaint is to determine the dominant noise source. This requires that several questions be asked: Is the problem a noise problem or a vibration problem, or both? Is the noise felt or heard? What type of noise is it – a rumble, a hiss, or a tone? Is the noise modulating, continuous, or intermittent? How often does it occur? Can the problem be linked to a particular machine or piece of equipment?

Isolating the source may be accomplished by turning on and off pieces of equipment individually. This may or may not be practical or even possible depending on the type of equipment involved. For instance, you cannot turn off a chilled-water pump or cooling tower without also turning off the associated chiller since that may cause the chiller to freeze or shut-off on a safety switch. In addition, simply isolating one source may or may not solve the problem because it is possible for two different sound sources to combine to create a problem.

Once the dominant sources are identified, determine if the equipment is installed and operating properly according to the manufacturer's guidelines. If the source is adjustable, try different settings to determine if altering the speed or set point eliminates the problem.

Once the problem is analyzed, list all possible solutions along with their probable impact on the operation of the system. For instance, adding duct mounted sound attenuators, lining or additional elbows to reduce sound may have a negative impact on the fan and consequently airflow may be adversely affected. When in doubt, consult a specialist such as an acoustician or acoustic engineer.

Recommended Noise Reduction Methods

Sound Source	Path Numbers
Circulating Fans; Grilles; Diffusers; Registers; Unitary Equipment in Room	1
Induction coil and fan-powered mixing box	1,2
Unitary equipment located outside of room served; remotely located air-handling equipment such as blowers, dampers, duct fittings, and air washers	2,3
Compressors and pumps	4,5,6
Cooling towers, air-cooled condensers	4,5,6,7
Exhaust fans: windows air conditioners	7,8
Sound transmission between rooms	9,10

Transmission Paths

Description	Recommended Noise-Reduction Method
1. Direct sound radiated from source to ear reflected sound from walls and ceiling	Direct sound can be controlled only by selecting quiet equipment reflecting sound is controlled by adding sound absorption to room and to location of equipment
2. Air and structure-borne sound radiated from casings and through walls of ducts and plenums and is transmitted through walls and ceiling into room	Design ducts and fittings for low turbulence; locate high velocity ducts in non-critical areas; isolate ducts and sound plenums from structure with neoprene or spring hangers
3. Airborne sound radiated through supply and return ducts to diffusers in room and then to listener by Path 1	Select fans for minimum sound power; use fiberglass lining in ducts; use duct silencers or sound plenums in supply and return air ducts
4. Noise is transmitted through equipment room walls and floors to adjacent rooms	Locate equipment rooms away from critical areas; use masonry blocks or concrete for equipment room walls and floor
5. Building structure transmits vibration to adjacent walls and ceilings, from which it is radiated as noise into room by Path 1	Mount all machines on properly designed vibration isolators; design mechanical equipment room for dynamic loads; balance rotating and reciprocating equipment
6. Vibration transmission along pipe and duct walls	Isolate pipe and ducts from structure with neoprene or spring hangers; install flexible connectors between pipes, ducts and vibrating machines
7. Noise radiated to outside enters room windows	Locate equipment away from critical areas; use barriers and covers to interrupt noise paths; select quiet equipment
8. Inside noise follows Path 1	Select quiet equipment
9. Noise transmitted to diffuser in a room into ducts and out through an air diffuser in another room	Design and install duct attenuation to match transmission loss of wall between rooms
10. Sound transmission through, over, and around room partitions	Extend partition to ceiling slab and tightly seal all around; seal all pipe, conduit, and duct penetrations

Figure 116

Recommended Noise Reduction Methods and Transmission Paths

Figure 117 shows possible sources of sound, their transmission paths, and recommended noise reduction methods. It demonstrates how these principles can be applied to help solve sound problems.

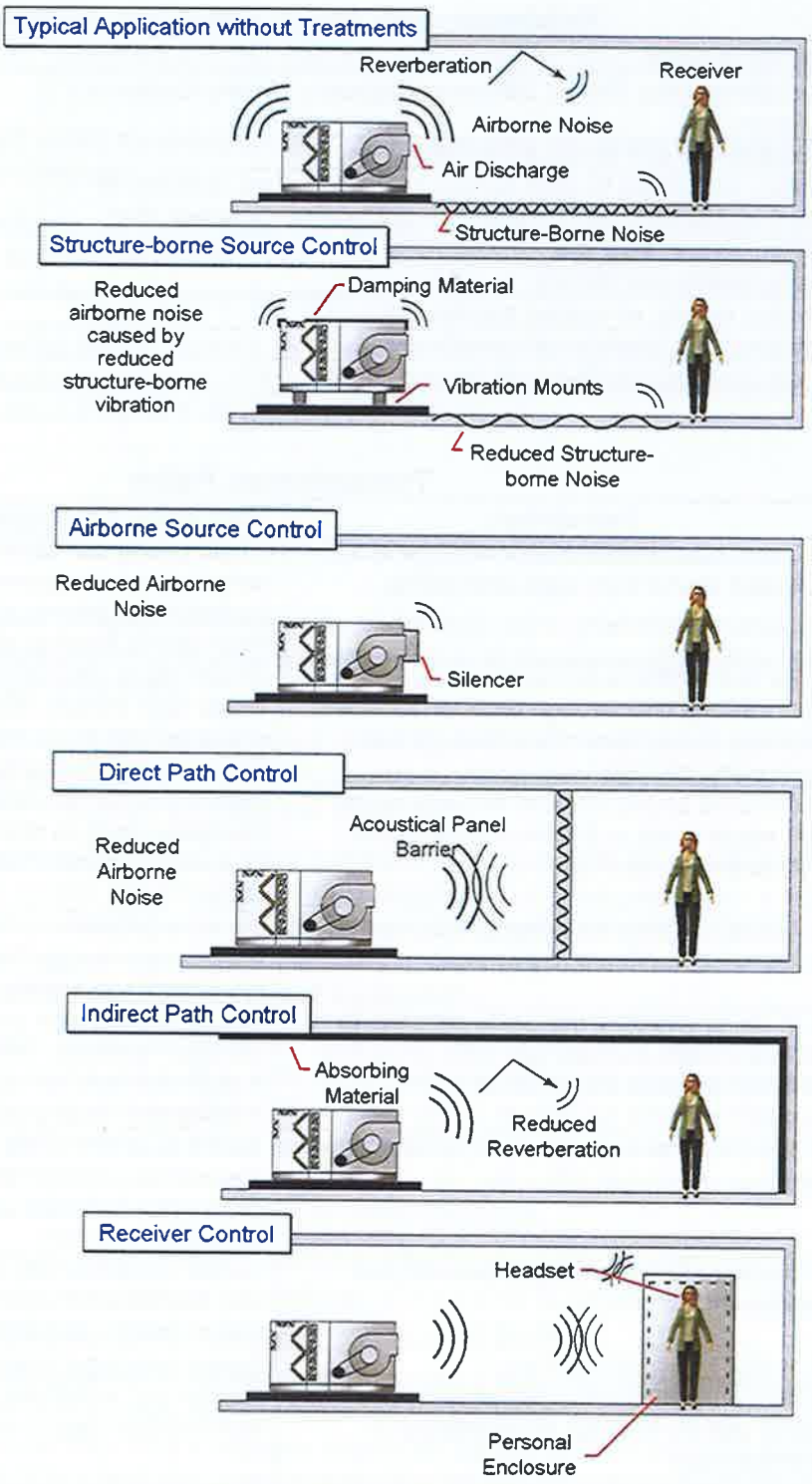


Figure 117

Sound Sources, Transmission Paths, and Noise Reduction Methods

Sound Sources, Transmission Paths and Recommended Noise Reduction Methods

(Text and pictures in this section provided by BRD Noise and Vibration Control, Inc.)

This section contains a few real life examples of solving noise control issues using the techniques described in this TDP. The solutions applied may not work for every job, but the approach may give you some ideas as to what can be done.

The first example is a liquid screw chiller that was installed in a nursing home mechanical room where several of the housing units were located in close proximity. This is a complete enclosure with flexible composite acoustical curtains on four sides and a series of rigid roof panels sitting on top of the track. The acoustical curtains have a layer of quilted fiberglass for absorption bonded to a loaded vinyl sound barrier in the middle with another absorber layer on the outside. The curtain panels with grommets are suspended from a strut type track framework (using sliding roller trolleys) and are attached to each other with Velcro. The panels slide out of the way on the overhead tracks for quick and easy service and maintenance access. Cutouts for the pipe penetrations are done in the field as shown to assure the best overall seal. The roof panels are removable to access equipment or make piping modifications. The entire system is modular and demountable. This solution used direct path control to achieve the required reduction. Notice the use of multi-layered panels with mass in the loaded barrier for low frequency absorption. Installation time was 8 hours for a three-man crew.



Figure 119

Direct Path Control for Screw Chiller in a Nursing Home
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

Another example is a liquid screw chiller installed in a mechanical room of an elementary school. Although it is buffered by utility space, it is on the same floor with many of the classrooms and the auditorium. The modular metal panel enclosure is a complete enclosure on four sides with roof panels. The construction of the panels is all galvanized steel with an 18-gauge solid outside skin and a 22-gauge perforated inside skin. The acoustic core is a high-density acoustical mineral wool. The wall panel construction for this project is 4 inches thick. There are access doors on the ends of the enclosure and three 42-inch wide removable panels in front of the control panel. The top

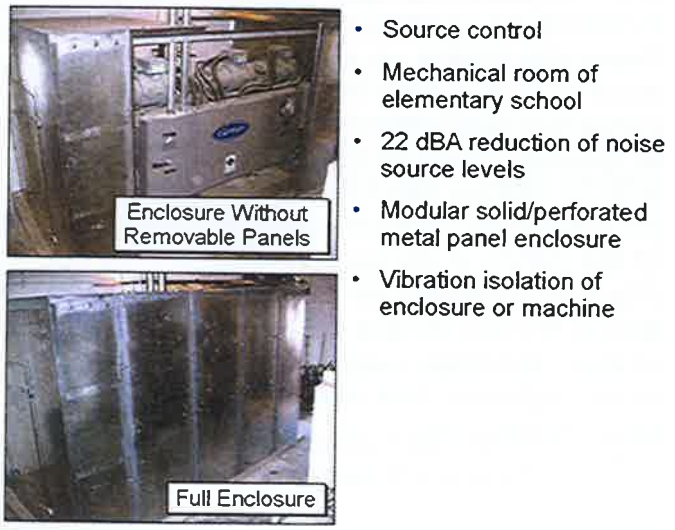


Figure 118

Direct Path Control for Screw Chiller in an Elementary School
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

picture shows the removable panels taken out and the bottom picture shows the full enclosure. The removable panels are secured with quick release tension latches and sponge rubber gasketing around the entire perimeter. They can be removed without the use of tools. Overall, reduction of noise is 22 dBA for this treatment, which again is direct path control. Installation was by a sheet metal contractor, and was installed in 16 hours by a four-man crew.

Another application involved a centrifugal machine. Removable reusable acoustical blankets have been used as source treatment. The construction of this removable blanket system is a dual density core consisting of 12 pounds per cubic foot density needlemat thermal/acoustic insulation with a layer of 1 pound per square foot density mass loaded vinyl on the outside. This dual density core is completely encased in a Teflon coated fiberglass cloth exterior jacketing. The sound test in accordance with ARI 575 showed a reduction of 6 dB. The blanket treatments provide a moderate degree of overall attenuation while also suppressing the discreet frequency compressor tones. Lacing anchors and stainless steel wire ties are used together with overlapping Velcro flaps to secure the acoustic blankets in place. The installation time was 12 hours for a two-man crew.

- Source control (airborne)
- Penthouse mechanical room of office building
- 6 dBA noise reduction
- Removable sound blankets on condenser, compressor, and discharge
- Sound test procedure: ARI575

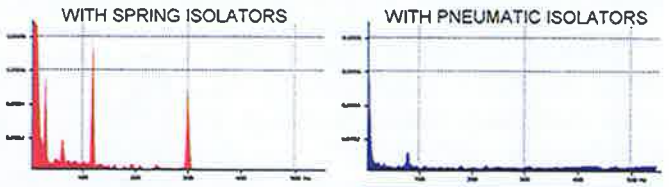


Sound Pressure Levels (dBA)										Metered Reading (dBA)
Position	31.5	63	125	250	500	1K	2K	4K	8K	
4-Before	39.6	45.8	61.3	70.4	74.8	79.6	83	82.2	85.2	88.5
4-After	34.6	49.4	63.3	68.1	73.2	75.4	79.3	80.5	74.3	83
9-Before	41.9	41.6	57.6	67.1	76.4	75.6	79.4	81.2	84.2	85.8
9-After	33.9	42.2	58.6	65.8	69.5	69.9	74.7	71	73.3	78.1
14-Before	38.6	49.4	62.5	70.4	75.5	79.7	82.9	86	87.3	90.4
14-After	39.1	53.4	66.0	68.1	72.1	72.8	76.4	72.8	74.5	82.9
19-Before	38.4	45.3	59.4	69.2	72.6	75.3	80.1	78.6	82.1	86.6
19-After	35.8	47.4	69.2	68.3	70.6	72.6	76.1	74.5	74.5	81.8

Figure 120

Centrifugal Chiller Source Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

For this mechanical room, a liquid screw chiller was installed in a space with a common wall to the school auditorium. In addition to ***airborne transmission***, there was a structure-borne issue transmitting through the spring isolation mounts. The graphs on the bottom of this cell show before and after vibration testing where pneumatic air bag isolators were installed under the chiller. This pneumatically operated and leveled isolation system virtually eliminated the predominant vibration frequencies yielding an overall efficiency of 99.99 percent. This isolation system was used in conjunction with an acoustical curtain enclosure to deal with the airborne transmission issue. An auxiliary air compressor was part of this system to provide a constant supply of compressed air. This facilitates an accurate leveling system and assures maximum isolation efficiency. Installation time was 10 hours for a two-man crew.



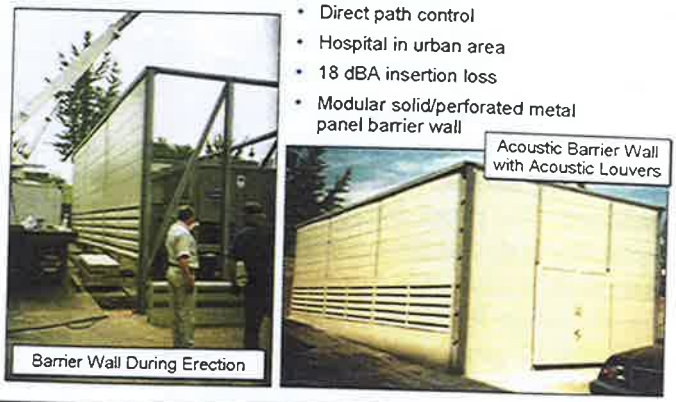
- Source control (structure-borne)
- Mechanical room with common wall to auditorium
- Pneumatic isolators to replace conventional springs
- Curtain enclosure also used



Figure 121

Structure Borne Sound Control for a Screw Chiller
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

A 3-inch thick acoustical barrier wall was installed around this air-cooled chiller at the edge of a parking lot right next to a hospital in an urban area. Patient rooms are a short distance away with windows having direct line of sight to the chiller area. There was also a concern for the community property line approximately 100 feet away. The modular barrier wall construction utilized a 3-inch thick powder coated steel construction with 16-gauge outer skin and 22-gauge perforated inside skin. The acoustic core material was a 6-pound per cubit foot density fiberglass product. There are large access doors at each end of the enclosure and a bank of air intake louvers on both sides to minimize any airflow impact on the operation of the chiller. The overall height of the open top barrier wall is 4 feet taller than the height of the condenser fans and is located with approximately 4 feet of operating clearance all around. The overall noise reduction was about 18 dBA. Installation time was 16 hours for a 5-man crew (includes dedicated crane operator).



- Direct path control
- Hospital in urban area
- 18 dBA insertion loss
- Modular solid/perforated metal panel barrier wall

Figure 122

Air-Cooled Chiller Direct Path Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

A common treatment strategy for the air-cooled chillers is to use removable sound blankets to wrap the compressors and connected components. Removable sound blankets should be custom manufactured to fit the specific equipment. Pieces are manufactured in three-dimensional designs in a number of pieces to fit the various components. These components include the compressor, the compressor discharge lines, suction lines, oil separators, and on occasion the evaporator tube. The method of assembly/installation includes overlapping Velcro flaps and lacing wires or straps with D rings. All blanket parts are tagged by the component to be treated (i.e., compressor or discharge pipe, etc.) and sequentially numbered according to the order of installation. An exploded drawing provides further assistance to locate blankets properly. To assure proper fit, the blanket manufacturer needs to know where factory installed closed cell foam insulation is to be used under the acoustic blankets. Typical noise reductions are 3 to 5 dBA if the compressors are treated and 6 to 8 dBA if the piping and oil separators are also covered.



- Source control (airborne)
- Rooftop of school
- 5 dBA noise reduction
- Removable sound blankets for compressors and piping
- Templated standard kits
- Guaranteed fit for optimum performance
- Pieces tagged by part number in sequence of installation
- Layout drawing simplifies installation

Figure 123

Applying Sound Blankets
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

Figure 124 shows a typical application where air-cooled chillers or condensing units are located on grade behind a building. This is a commercial office building located adjacent to a completely residential neighborhood with residences virtually on all sides of the building within a distance of about 40 feet. The three-sided barrier wall height was to be minimized on this project so the design included a condenser fan unitary acoustical panel systems around the top perimeter of each unit. The panel construction for the barrier wall and the unitary acoustical panel system was 4 inches thick overall, using 16-gauge solid outside skin and 22-gauge inside perforated skin and a 4-pound density acoustical mineral wool core acoustic insulation. Structural steel is selected and designed to meet local requirements for wind loads. An often-overlooked aspect of this type of application is the contribution from building reflections. You will note the installation of acoustical panels on the building façade facing the condensing units to minimize reflected noise. All components of this system have a powder coated paint finish. Installation time for this project was 24 hours for a four-man crew.

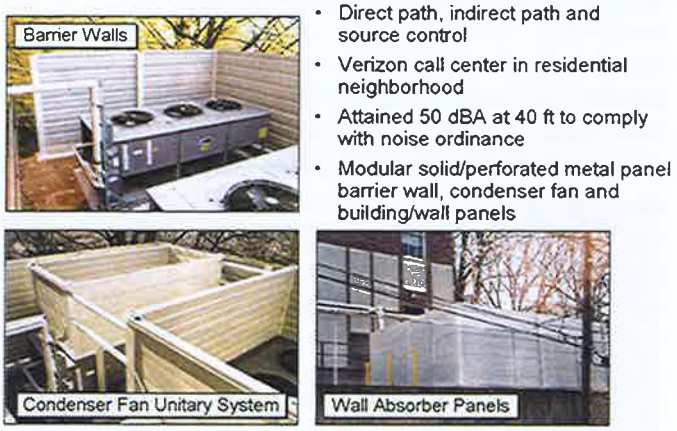


Figure 124

Air-Cooled Chiller Indirect Path and Source Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

Treating condenser fan noise on air-cooled chillers, condensing units and condenser sections of packaged air-handling units can be done in a number of ways. The preferred method of treatment is the unitary panel system shown below (top left), which has an economical cost and has a minimum impact on airflow. The typical construction of the unitary panel system is an 18-gauge solid outside skin and a 22-gauge perforated skin with 2 inches of high-density mineral wool acoustic fill. Sometimes the panels are supported by structural steel components of the equipment itself. Other times independent structures is installed to support the panels at the height of the top of the chiller. The parallel baffle and silencer bank systems are used for applications that are more critical where condenser fan noise is traveling to upper story levels of an adjacent building or where the topography for residential community property line receivers may be elevated in relation to the chiller. These treatments are often needed for projects where residential property lines are 10 to 50 feet away. The silencer stack system is utilized on an isolated basis where there may be a small quantity of one to three large diameter condenser fans. Overall impact on airflow for any condenser fan treatment is the most critical aspect of the overall



Figure 125

Condenser Fan Noise Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

design. Normally these systems add about 0.10 in. wg to 0.25 in. wg pressure loss. The pressure loss requirements are communicated in the design stage and incorporated as a key design element in the overall acoustic treatment system. The manufacturer of the condenser fan needs to verify the impact on fan performance.

Figure 126 illustrates the use of a fan silencer mounted between the roof curb and a power roof ventilator exhaust fan. Basic construction of the silencer is parallel 6-inch thick perforated acoustic baffles or splitters spaced in such a way to control overall acoustic performance and pressure loss. The splitters have approximately 12 inches of clearance to the operation of the bottom of the fan wheel blades in order to minimize turbulence airflow at the fan. Commonly, 12 to 18-inch high curbs are utilized with acoustic material on the inside perimeter of the four-sided curb for sound attenuation. However, without the parallel baffle inserts and some additional height, the overall effectiveness is minimal with no noise reduction in the frequencies below 400 Hz.



- Curb mounted silencer
- 10 to 12 dBA insertion loss
- Minimal pressure loss at 0.15 in. wg
- Easy to install
- Parallel baffle design with 3 ft low profile



Figure 126

Exhaust Fan Sound Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

Duct silencers are available in a number of configurations, most commonly rectangular and circular. Other configurations include elbows, oval and flat oval. Duct silencers are a dissipative/absorptive design whereby internal bullets, splitters, or baffles are constructed from perforated steel and packed with high-density acoustic fill. As air and noise pass through annular passageways, the noise is refracted (bent into) the liner and is dissipated as pressure fluctuations transmit through the perforated retaining skin into the acoustic fill and are then converted to heat through friction. Overall sound attenuation and pressure loss ratings are balanced together with the overall size of the attenuator based on the internal geometry and dimensional relationships among the shell lining, the splitters, the baffles, and the bullet assemblies.



Circular Duct Silencers

- Duct silencers provide excellent mid to high-frequency sound control
- Round ductwork controls low-frequency sound
- SMACNA design guidelines help minimize air turbulence and regenerated noise
- Locate first 20 feet of supply ductwork above non-sensitive spaces



Rectangular Duct Silencers



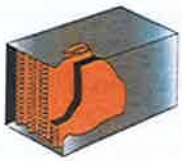


Figure 127

Duct Silencers for Duct Transmission Control
(Photos courtesy of BRD Noise and Vibration Control, Inc.)

The biggest problem with rooftop units is to understand that most installations with objectionable noise and vibration are system problems and not HVAC unit problems. The key elements to consider are the support configurations, the HVAC unit configurations, and the duct configurations as well as system integration with the building and its overall construction. For curb-mounted applications on lightweight roof constructions, location of units should be in close proximity to structural spanning steel. Many times, additional spanning steel will need to be added in the area of the unit support in order to minimize roof deflections. Fan discharge arrangements, the type of fan wheel, the use of discharge plenums, acoustic plenum liners, etc. are also important considerations. Duct configurations should be in accordance with SMACNA and ASHRAE Standards for critical applications below rooftop units. Round ductwork should be utilized to minimize duct rumble in the first 20 ft of the supply duct system. The quiet roof curb detail shows how acoustical batts and loaded vinyl sound barrier material are used to reduce radiated noise transmitting from the bottom of the unit through the deck to the spaces below. Acoustical grade sealant seals all small air gaps. The illustration also shows a vibration isolation curb that is preferred over internal isolation of the fans. Internal isolation should not be used together with vibration curbs.

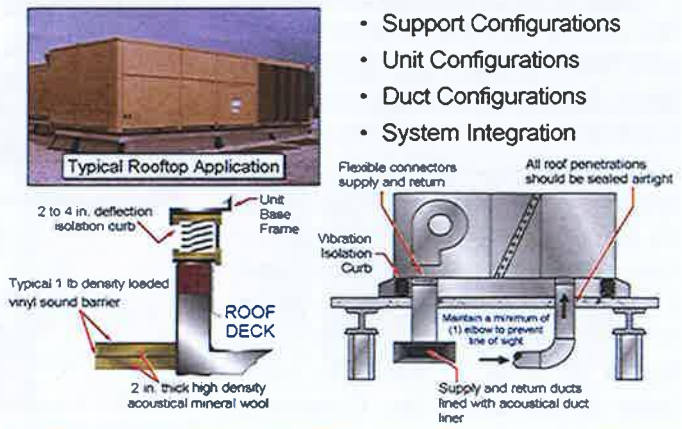


Figure 128

Rooftop Direct Path Control

Active Noise Control

Active noise control is an electronic method of reducing or removing unwanted sound by the production of a pressure wave of equal amplitude but opposite sign to the unwanted sound. When the electronically produced complement wave is added to original unwanted sound the result is a significant reduction in low frequency sound at that location. This is using the principle explained earlier in the text on interference waves.

This method has been used successfully in some duct-mounted situations to reduce fan noise. The system consist of microphone mounted down stream in the duct which picks up the acoustic wave in the

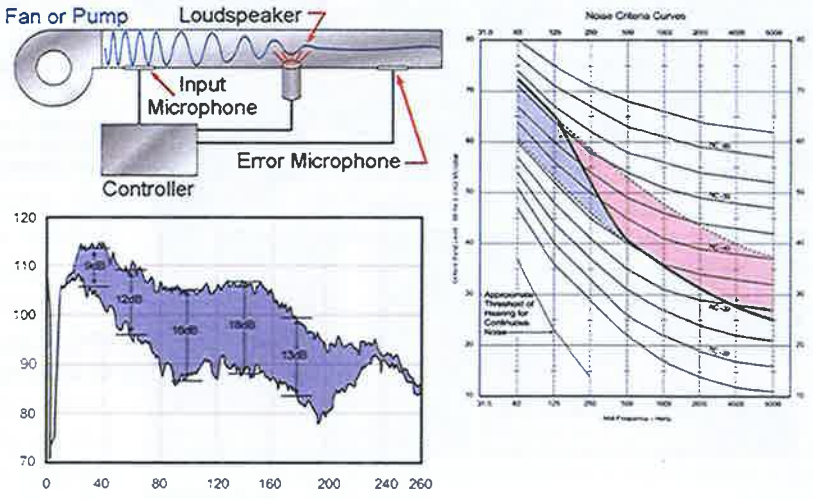


Figure 129

Active Noise Control

duct sends the signal to wave generator with an amplifier and speaker located upstream of the microphone which generates a signal of appropriate frequency and strength to cancel the selected frequencies. This requires sufficient ductwork for the microphone and speaker system to sense and respond to the sound wave.

This method of noise control is sometimes considered a "cure-all" for noise problems, but this is not the case. To be successful, a knowledgeable practitioner must be employed to design the system modification. Noise cancellation in three-dimensional spaces is difficult to achieve.

Vibration

Vibration is motion caused by a rapidly moving object. This can be transmitted to the structure on which it is mounted. This transmitted motion may be felt or cause other objects and surfaces to vibrate and radiate sound waves. Vibration motion is measured in terms of the amplitude of deflection caused by the motion. This motion may be a single, transient disturbance (shock) or may be oscillating motion (vibration). In HVAC systems, the major concern is vibration caused as the result of the imbalance of rotating parts and other repetitive forces in the equipment.

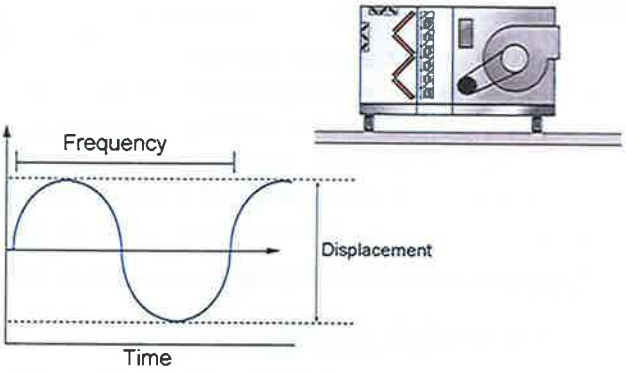


Figure 130

Vibration is Mechanical Wave Motion

All rotating equipment will have some vibrating motion associated with it because of slight imbalances or eccentricity of motion. A vibration problem exists when the amplitude of the vibration becomes large enough that it transfers into the building structure and causes detectable motion in the space or excites building surfaces causing motion or sound generation. Vibration frequencies of usual concern are between 30 Hz and 500 Hz.

The transmission of vibration is a function of three variables, the amplitude of the motion caused by the vibrating machinery, the ***transmissibility*** of the isolator between the machinery and the dynamic response of the mounting surface, and the rigidity of the mounting surface.

Vibration Fundamentals

In discussing vibration, three basic elements apply to all vibrating systems, the equipment, the vibration mount or isolator and the dynamics of the base on which the unit is mounted. As with the sound waves described earlier, the equipment is again the source, the base is the receiver, and the isolator is a device in the path. The isolator is a resilient member that acts to both absorb energy and to act as a time-delay of energy. Isolators can act as dissipaters or as absorbers of vibration energy.

HVAC vibration is often measured using an ***accelerometer*** to detect the vibration. A transducer is securely attached to the surface under investigation. The accelerometer produces an electrical charge proportional to the surface acceleration, which is then amplified by a charge amplifier and recorded or observed with a meter. The frequency and range from below 1 Hz to about 1000 Hz is the area of most interest.

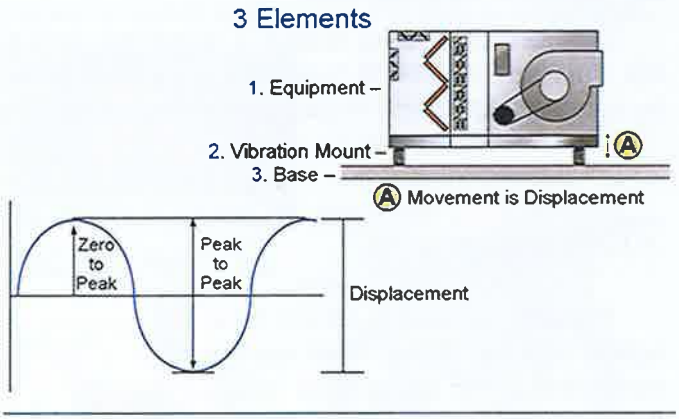


Figure 131

Vibration Terms

It is sometimes more useful to determine the vibrational velocity or ***displacement***. Often, moving coil transducers are used to directly measure vibrational velocity. A single integration of that signal provides a measure of displacement.

If only an accelerometer is available, it is necessary to integrate the acceleration signal once for velocity and twice for displacement. If the vibration displacement is sinusoidal at a known frequency (f), then an integration is calculated by dividing the original by $(2 * \pi * f)$ (noting that there is also an associated phase change).

Example: A machine is vibrating sinusoidally at 79.6 Hz with an rms acceleration of 10 m/s^2.

Its ***rms velocity*** is therefore $10/(2 * \pi * 79.6) = 20$ mm/s

Its rms displacement is $10/(4 * \pi^2 * 79.6^2) = 0.04$ mm

The final result may also be expressed in terms of ***zero-to-peak***, which is two times the square root times the rms value. The ***peak-to-peak*** value is twice the zero-to-peak value.

There are three terms, scales, and directions used to describe vibration motion. The three terms are acceleration, velocity, and displacement. Three possible scales are also used rms (root mean squared, 0-p (zero-to-peak), and p-p (peak-to-peak). The three possible directions are E-W, N-S, and up-down. Therefore, there are 27 separate possible values: vibration measurements and reporting must be explicitly clear.

Controlling Vibration

Vibration problems are analyzed by considering the system as a number of connected springs and masses with damping. The vibration source is within either the compressor motor or the mounting assembly. If a motor inside a machine produces the vibration, the ***natural frequency*** of the supporting system must be well below the frequency of motor oscillations (the forcing frequency). To achieve this, increase the mass of the motor or decrease the stiffness of the mounting assembly.

The method of vibration isolation is demonstrated with a weight held from a rubber band. If the band is moved up and down very slowly, the suspended weight will move by the same amount. At resonance the weight will move much more and possibly in the opposite direction. However, as the frequency of vertical movement is further increased, the weight will become almost stationary. Springs are more often used in compression than intension.

Attempts to reduce vibration can make the problem worse, particularly if the wrong isolators are added to a system. Vibration issues are best addressed at the time of design, manufacture, and installation.

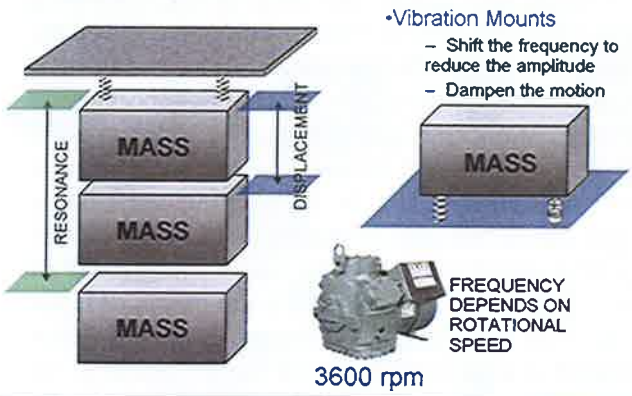


Figure 132

Vibration Control

Another approach to vibration control is to cancel the forces involved using a dynamic vibration absorber. Here an additional "tuned" mass-spring combination is added so that it exerts a force equal and opposite to the unwanted vibration. This only works when the vibration is at a single fixed frequency.

Figure 133 shows the static deflection of the mounting needed to limit the transmission of vibration to a given percentage of the total vibratory force of the equipment. The chart also recommends the maximum permissible transmissibility for assorted conditions that commonly occur in machinery and equipment installations.

To use the chart, follow these steps:

1. Determine the lowest rotational speed of the equipment and consider the disturbing frequency.
2. Move vertically to the slanted line corresponding to the percentage of transmissibility that can be tolerated.
3. Follow that line to the left of the graph to determine the natural frequency and static deflection required of the isolators.
4. Consult the catalog of the vibration mountings that will be used to select the type of isolator required with the appropriate static deflection.

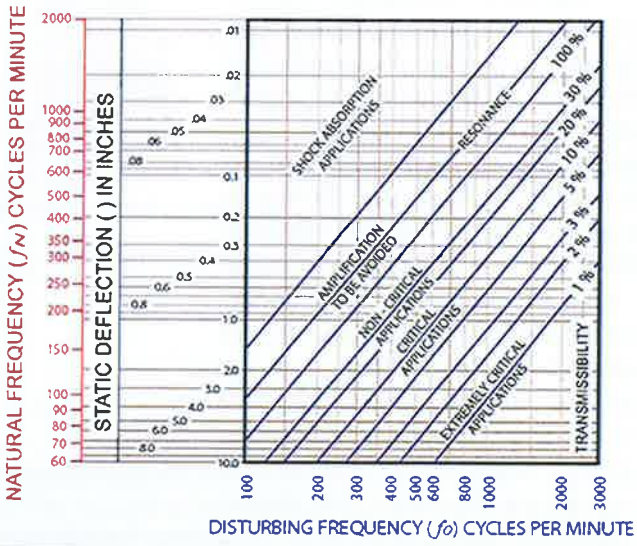


Figure 133

Vibration Selection Curve

The efficiency chart is used for models with a single degree of freedom. Other complex problems may occur. Such situations may require the assistance of a vibration specialist.

The chart in Figure 134 shows the amplification, isolation resonance, and vibration isolation region. The vertical axis shows the ratio of the disturbing source frequency (fd) to the natural frequency of the isolator system (fn). ***Resonance*** results when the sympathetic vibrations reinforce each other because the disturbing frequency is equal to the isolator natural frequency (the ratio of fd/fn= 1). At low frequencies below resonance there is no vibration isolation. Slightly above the ratio of one we are still in the amplification region until the ratio equals the square root of two. Above this point we begin the region of isolation because less energy is coming out of the isolator compared to what is going in. As a rule, a ratio of ten to one (10:1) is desirable for effective vibration isolation. A ratio below three to one is generally not recommended.

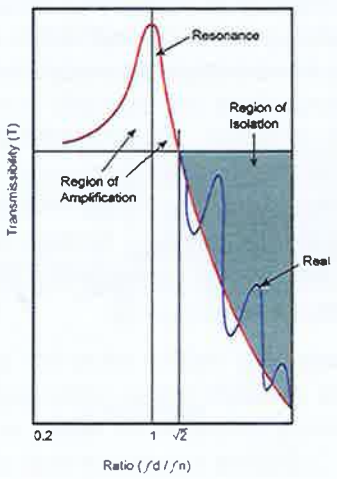


Figure 134

Vibration Amplification

Seismic (Earthquake) Protection

Earthquakes can produce vertical and sidewise vibrations up to perhaps one G or more, though it is usually much less. The immediate concerns can be divided into two categories: operating, basic earthquake" (OBE) and "safe shutdown earthquake" (SSE). OBE protection seeks to have equipment operate during and survive an earthquake. SSE protection merely assures that equipment that shuts down during an earthquake will remain viable for future use. During an earthquake, the amount of vibration experienced by equipment in a building can be worsened by building characteristics, the location of the equipment in the building, and the magnitude of the earthquake. The elasticity of the joists and columns act as springs, aiding in the protection of the equipment. However, this elasticity can also cause the building to exacerbate the vibratory motion. This motion typically occurs at anywhere from 5 Hz to 15 Hz. The best place to be is in the basement, on bedrock. Equipment bolted to the floor and walls ("hard-mounted"), survive best, provided the bolts, walls, and floor do not break. Equipment mounted on isolator springs are particularly vulnerable since those soft isolators allow the equipment to sway due to the earthquake, only to suddenly crash into the spring stops or nearby objects. Special seismic isolator springs containing soft stops that cushion the impact.



SSE (Safe Shutdown Earthquake)
- Unit survives

OBE (Operating Basic Earthquake)
- Unit continues to operate
  - Frequency is 5 to 15 Hz
  - The higher in the building the more critical
  - Special calculation by a professional engineer may be required

Figure 135

Seismic Protection Classifications

Types of Isolators

Isolators are available in many styles and for many different applications. Figure 136 shows some examples of isolator types used for a variety of applications. Vibration mountings may be as simple as soft ribbed rubber vibration pads or may be used in conjunction with springs or even steel mounting bases filled with concrete that are mounted on springs. The concrete acts as an additional mass, which vibrates less due to the increased mass of the machine and base. Selecting the right isolator depends on the equipment, which it is being used on, the location of the equipment in the building, the anticipated disturbing frequency.



Figure 136

Vibration Isolators

Preparing Acoustic and Vibration Specifications

When acoustics and vibration are of concern for a project, the construction documents and specifications should be written to address acceptable levels and target noise criteria. Because sound power will not be the same among manufacturers and even within product families, the acoustical goals should be clearly stated in the documentation. The documentation should not only state the established goals but should also reflect the acoustical performance of any barriers applied in the transmission paths to achieve the acoustic goal.

This section reviews some of the key issues to be considered in preparing these job specification documents.

General Guidelines

Where possible the specifications for noise and vibration control and for all components should be in a single specification section. The specification should establish the design goal and the rating system used to express the goal. In this section of the specifications, minimum or maximum requirements for each system component should be documented. Remember that no two manufacturers will have the same performance data, so ratings should be expressed in terms of allowable maximums or minimums that will achieve the design goal. Be sure the specifications include all the components critical to achieving the design goal. Items such as silencers, sound traps, vibration isolators, and duct/plenum liners need to be included. The construction drawings should also include adequate details of the design that are necessary for achieving the design goal.

A performance schedule specification should be included outlining the maximum allowable sound power level for each noise-sensitive piece of mechanical equipment such as: fans, air-handling units, fan-coil units, air terminal devices, chillers, and cooling towers.

Acoustic specifications should require manufacturer's data to state they have been determined using ARI rating procedures where they apply.

Establishing Space Criteria

Start with the desired room sound level (NC or RC) using ASHRAE design criteria. Then, specify an appropriate and realistic NC or RC rating. Keep in mind that background noise levels in most offices are greater than NC-35, and room noise levels as high as RC-40N is seldom objectionable in most open plan office environments.

Next, establish the best physical location for the HVAC equipment. Consider factors such as structural, discharge sound, duct breakout, and register/ diffuser generated noise. Backward-calculate the maximum allowable equipment sound power levels from desired room RC/NC rating. This maximum allowable sound power level information should be specified by octave band frequency range.

The sound path with the least amount of attenuation will determine the critical path. The calculated allowable sound power levels are then compared to the manufacturer's published sound power data to determine compliance. If allowable sound power levels are too low to be met by various manufacturers, you may have an unrealistic requirement. In this case, consider increasing the amount of attenuation, or relocating the equipment farther away from the room (receiver).

Equipment Specification Requirements

Each type of equipment has different sound rating procedures and characteristics that make the specification requirements unique. Listed below are some recommendations to be considered when writing specifications for that type equipment. Use the guidelines provided earlier in this TDP. Make sure that the specifications include adequate detail about sound and vibration construction features and manufacturer options so that the installed job will meet the designer's original design intent.

For example, specifications for VAV boxes and fan-powered mixing boxes should be specified based on the units being rated to ARI Standard 880. The ratings should give frequency band data for both radiated and discharge sound power levels. Parallel fan-powered mixing box units need to be tested with the fan on and off. If the sound calculations are based on using the box with a certain internal liner which provides a level of attenuation be sure to include that item in the specification.

Write Specifications to Avoid Ambiguous Language

When writing specifications for acoustical performance, be sure that you are clear and concise in your wording. Also, be sure that you address all aspects of acoustical performance. Here are some examples of specifications:

"The HVAC equipment shall not produce objectionable noise or vibration."

This specification is incomplete and the wording is vague. The entire system must be considered, not just the HVAC equipment. What does "objectionable" mean in this context? This subjective wording should be avoided.

"HVAC system noise shall not exceed NC-40."

This requirement places the responsibility of acoustic testing and compliance with the contractor. It also has no regard to location. Different spaces have different acoustic requirements based on activities. Newer room criteria (RC) should be used along with specific locations of where noise levels are to be maintained.

"Maximum fan noise shall not exceed 85 dB."

Decibel (dB) ratings are only one characteristic of sound.

"Sound attenuator shall attenuate fan noise by a minimum of 25 dB."

Decibel (dB) ratings are only one characteristic of sound. In addition, the attenuation effect in each octave band center frequency must be specified.

"All vibration isolators shall be 90 percent efficient."

This is impractical and expensive to verify. Large chillers and air-handling units obviously require more efficient isolators than slab-mounted pumps. A good isolator specification should contain details showing installation requirements along with a schedule listing isolators for each piece of equipment.

Summary

Acoustics is an important step in the design process. A system that is designed to provide ideal temperature and humidity control but has poor sound quality will be judged just as poorly a system that has inadequate temperature control. It should also be noted that is much easier to design to meet acoustic design objectives than it is to retrofit the system. For these reasons a designer should make an acoustic goal a part of his initial design decisions. This goal should include both indoor and outdoor requirements.

This TDP provided the HVAC designer a background of knowledge to help them deal with acoustic design issues. The first sections of the module addressed the terminology associated with acoustics. These sections described how sound is a wave of energy and how we can use principles of wave motion to describe an analyze sound. This section also described the important difference between sound power and sound pressure and how both are measured, converted to the other and how they are mathematically manipulated. This section also introduced the important concept used for acoustical analysis, which is the source – path – receiver concept.

Subsequent sections dealt with the ways we rate sound using scales for loudness, scales of acoustical bands and measures of space acoustical quality with rating methods like A-weighted, NC curves, and RC curves. The methods used by manufacturers to establish sound ratings was described and how this data could be used to determine the required source data for our acoustical analysis. Once the source sound power data was determined, design goals for indoor and outdoor locations could be established, set the requirements for our receiver.

The next two sections described how to apply equipment and the procedure to use to determine requirements for sound attenuation in the path between source and receiver. An approach and recommendations were made as to how to address projects that have not met the design requirements with a few recommendations on how these issues may be solved. In the final two sections, the topic of vibration was discussed as was some recommendations on how to address acoustical design issues in your specifications.

Acoustics can be a very complicated and involved topic, which is one of the reasons the HVAC field has professionals who deal only with acoustical design issues. This purpose of this TDP is to provide a foundation of information to allow the typical HVAC designer to design for the less critical acoustical projects, know when to call in the professionals on the critical jobs, and provide the designer with the basic concepts and terminology to discuss the issues with the professional.

Work Session 1

1. Would it be a good idea to design an HVAC system that produced negligible detectable noise in the occupied space? Why?

   _____

   _____

2. True or False? Tonal sounds are sometimes referred to as white noise. _____

3. A sound source is measured at 3 feet with a sound pressure of 85 dBA is in a free field. What is the sound pressure at the following distances?

   3 feet from the source _____ 12 feet from the source _____

   6 feet from the source _____ 24 feet from the source _____

4. An engineer desires to check the sound level of a newly installed centrifugal chiller in a mechanical room at a particular point. Other equipment in the room consists of pumps, boilers, electrical switch-gear, and a hydraulic elevator pump, all of which are running. Using a sound meter, the engineer measures a sound level of 89 dBA in the room. What is the best way to determine the sound level out-put of the chiller alone? _____

   a) Consult the manufacturer's published product data catalog, which is ARI certified.

   b) Turn off all of the equipment in the room, then turn on the chiller and measure it by itself.

   c) Use decibel subtraction, subtract the manufacturer's published sound level from the 89 dBA measured room value

   d) Turn off the chiller and take another measurement, then using decibel subtraction to determine the resulting chiller sound level.

5. Explain the source-path-receiver concept used for acoustical analysis:

   _____

   _____

   _____

   _____

   _____

6. Sound is described as a wave. The distance the wave travels during the cycles is the _____

7. True or False? Broadband sounds are considered the most annoying to most building occupants since they consist of sound energy at many frequencies. _____

8. A technician has been contracted to measure the presence of HVAC noise in a mechanical equipment room. With a sound meter, the technician has detected the presence of a sound with a center frequency level at 400 Hz. This is most likely coming from: _____

   a) An air diffuser
   b) A fan or pump
   c) A VAV box
   d) An unstable fan
   e) A centrifugal chiller or screw chiller
   f) Both (a) and (c)

9. True or False? Low frequency noise is easier to attenuate than high frequency noise. _____

10. Sound power is: (identify all that apply) _____
    a) A property of the source
    b) Dependent on the distance between the receiver and the source
    c) Measured directly by a sound meter
    d) Dependent on the path between the source and the receiver

11. Sound pressure is: (identify all that apply) _____
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    c) Measured directly by a sound meter
    d) Dependent on the path between the source and the receiver

12. Add the following decibel values 65 dB and 71 dB. The correct answer is: _____
    a) 136 dB
    b) 72 dB
    c) 66 dB
    d) 70 dB

13. Subtract the following decibel values 89 dB and 82 dB. The correct answer is: _____
    a) 88 dB
    b) 87 dB
    c) 89dB
    d) 86 dB

14. What is sound wave interference?

    _____

    _____

    _____

15. Noise is: _____

16. True or False? When two sound sources combine, like a pair of condensing fans on a condensing unit, the resulting sound power increases by 3 dB. _____

17. What is a free field?

    _____

    _____

18. Explain near field and far field:

    _____

    _____

    _____

    _____

19. Explain the Q factor:

    _____

    _____

    _____

    _____

20. Explain reverberation time:

21. True or false. Noise reduction number (NR) is the reduction in sound pressure in dB when a sound attenuating element such as duct lining is added in the sound path. _____

22. End reflection is a sound wave: (identify all that apply) _____

    a) Where not all of the energy transmitted in the duct is discharged into the room
    b) Generated in a duct that occurs at the fan discharge
    c) That tends to decrease the discharge sound level
    d) That tends to increase the discharge sound level
    e) That tends to increase the breakout sound level

23. Breakout noise is: (identify all that apply) _____

    a) Sound that passes through the walls of the duct
    b) Never contributes to the space sound levels since it is only in equipment rooms
    c) A sound wave which only occurs in the 1000 Hz frequency band
    d) That tends to increase the discharge sound level
    e) That tends to increase the radiated sound level

24. Describe an octave band:

25. What are point sources and line sources?

Work Session 2

1. What does a sound meter measure and how does it measure this value?

   _____

   _____

   _____

2. True or False? The human ear is most sensitive to low-frequency sound below 500 Hz? _____

3. The A-weighted scale was developed to: _____

   a) Approximate the human ears response to sound

   b) Measure the actual sound spectrum

   c) Compensate for the human ear's lack of response below 500 Hz

   d) Directly measure the sound power level by using a value of 1 meter

4. True or False? The A-weighted sound level scale is not a good measure of the resulting sound quality in a space? _____

5. Using an NC chart determine the resulting NC level when the following octave band reading are taken: 63 Hz – 51 db, 125 Hz – 49 dB, 250 Hz – 48 dB, 500 Hz – 42 dB, 1000 Hz – 35 dB, 2000 Hz – 49 dB, 4000 Hz – 33 dB, 8000 Hz – 30 dB.

   a) NC 51

   b) NC 30

   c) NC 40

   d) NC 35

6. NC curves should be used with caution because: (circle all that apply)

   a) The NC chart does not contain data below the 31.5 Hz range and hence cannot predict rumble conditions.

   b) ASHRAE no longer encourages the use of this rating method .

   c) A single number rating based on a tangency to a curve in only one octave band can lead one to mistakenly rate two different sound spectra as being equally acceptable.

   d) The NC chart rates the sound as a single number of sound power and thus does not account for the distance to the receiver.

7. True or False? According to ASHRAE, an RC level of 35 is recommended for private offices.

   _____

8. The room criteria (RC) method is preferred over the noise criteria method (NC) because: (identify all that apply) _____

   a) RC curves consider the room conditions where NC curves only consider the noise source levels.

   b) Modern testing instrumentation used to develop the RC curves make them more accurate than the older NC curves.

   c) RC curves contain data for both the 16 Hz and the 31.5 Hz octave bands, which makes them more useful for evaluating low-frequency rumble conditions.

   d) RC curves contain a qualitative letter descriptor, and are therefore more useful for evaluating sound quality.

9. Using an RC chart determine the resulting RC level when the following octave band reading are taken: 63 Hz – 51 db, 125 Hz – 46 dB, 250 Hz – 41 dB, 500 Hz – 37 dB, 1000 Hz – 35 dB, 2000 Hz – 32 dB, 4000 Hz – 39 dB.

   a) RC 35 N
   b) RC 32 N
   c) RC 40 H
   d) RC 35 H
   e) RC 35 R
   f) RC 45 T

10. Explain the reason different A, B, C and Lin rating scales have been developed:

    _____

    _____

    _____

11. True or False? Sound transmission Class is a rating of the attenuation provided in a sound path by a building wall or floor assembly. _____

12. Manufactures of most types of HVAC equipment rate the equipment using _____Standards and the results are presented as _____ levels.

13. When rating ducted equipment like central station air handlers or ducted rooftop units with ARI Standard 260, which of the following conditions must be tested and what results are reported?

    a) The discharge sound power level with the unit ducted into a reverberant test chamber with results reported by octave band frequency.
    b) The return sound power level with the unit ducted into a reverberant test chamber with results reported by octave band frequency.
    c) The casing-radiated sound power level with the unit in a reverberant test chamber and the supply ducted out of the chamber, with octave band results reported by frequency band.
    d) The discharge sound pressure level with the unit located in the reverberant test chamber and the discharge ducted out of the chamber with results reported by frequency band
    e) The single number sound pressure which results from the tone corrected A-weighted number of discharge sound

14. According to OSHA, how long can a worker remain in a space with a noise level of 105 dB without hearing protection? _____

    a) 30 minutes
    b) 15 minutes
    c) 4 hours
    d) 2 hours
    e) 1 hour
    f) A worker must be provided with hearing protection for any exposure at this level.

15. Which of the following are not good specifications for the acoustics in a space? _____

    a) The contractor shall show that the space can achieve a NC rating of 35.
    b) All HVAC units must be tested, rated, and certified by appropriate ARI standards.
    c) Chillers shall be as quiet as possible.
    d) Air-handling units shall not produce sound levels greater than the following in any frequency band:

63	125	250	500	1000	2000	4000	8000
101	95	98	93	90	86	81	76

    e) Air handlers shall not produce any objectionable noise within the occupied space.

16. Which of the following methods are ways to address acoustic issues that arise on an existing project? _____

    a) Provide vibration isolated acoustic shielding around a noise compressor
    b) Lag ductwork under a rooftop unit to reduce radiated breakout sound
    c) Provide sound attenuators in the ductwork to reduce fan discharge sound
    d) Provide a barrier between a lot line and a condensing unit
    e) Increase the fan speed by adding static for a more stable operation

17. True or False? Addressing acoustic specifications at the design phase of a project is important because it is much easier and less expensive to design to an acoustic goal than to attempt to correct issues that result from problems after insulation. _____

18. Which if the following need to be considered when determining the sound rating required for an outdoor unit?

    a) Local building codes that specify a lot line requirement
    b) Reflecting surfaces and trees
    c) Distance from the source to the receiver
    d) Location of barriers
    e) Sound power of the equipment
    f) Substitution of special fans, or sound blankets on the outdoor unit
    g) All the above

19. Which steps are required in performing an acoustical analysis for an indoor space? _____

    a) Determine the acoustical goal for the space.
    b) Identify all sound sources and obtain the sound power data for each.
    c) Identify dominant sound paths.
    d) Identify all attenuating elements in each sound path.
    e) Subtract the attenuating effect from the sound source to get the resulting sound pressure.
    f) Compare the results to the space design goals and add needed attenuation or change units as necessary to achieve the goal.
    g) Be sure to calculate the sound impacts of discharge, return and radiated paths as well as potential impact of adjacent mechanical equipment rooms.

20. Vibration isolation should be selected based on: (identify all that apply) _____

    a) The amplitude of the force
    b) The rotational speed of the unit
    c) Distance between structural supports
    d) Seismic requirements
    e) A minimum of 2 inch deflection of the isolators
    f) The mass of the unit relative to the mass of the building in pounds per ft^2
    g) All the above

Work Session 1 Answers

1. No, while the required room sound level depends on the activity level some sound is helpful in masking background noise like conversation in nearby rooms.
2. False
3. 3 feet from the source 85 dBA
   6 feet from the source 82 dBA
   12 feet from the source 79 dBA
   24 feet from the source 76 dBA
4. d)
5. Considering sound and sound control measures, sound analysis is best performed using the source-path-receiver model. This model considers sound across three components: the source, the path, and the receiver. The source is the element generating the sound wave and the sound power developed by the source. The path is the distance the sound travels, as well as reflection and sound-absorbing elements between the source and the receiver. The receiver is who is hearing the sound, where they are located. The acceptable sound pressure level at that location is used to analyze the sound levels.
6. wave length
7. False
8. Either b), c), or d)
9. False Low frequency noise is difficult to attenuate. Low-frequency noise must be eliminated or attenuated at the source. On the other hand, high frequency noise is easily attenuated by using duct liner, lagging, or duct-mounted silencers.
10. a)
11. b), c), d)
12. b)
13. a)
14. A sound wave can be reflected off of surfaces near the sound source, these reflected waves can combine with the wave generated by the source to add, cancel, or decrease the sound wave amplitude
15. unwanted sounds
16. True
17. When sound is generated in a region of space that has no reflecting surfaces
18. Near field is an area close to the sound source where the sound pressure fluctuates about a mean value and is normally defined as ¼ of the longest wavelength. Sound waves in this area can add constructively or destructively. The far field is the area beyond the near field and is usually considered to be any area greater than one times the longest wavelength.
19. The Q factor is a term used to account for the effect of reflecting surfaces on free field sound. A Q factor of 2 indicates one reflecting surface, a factor of 4 indicates two reflecting surfaces and a factor of 8 indicates three reflecting surfaces
20. Reverberation time is the time it takes for a sound wave to decay 60 dB after a sound source has stopped. This time is a function of the reflecting and absorbing surfaces in a space an influences the ability to hear speech and music in a space.
21. False, this is the definition of insertion loss.
22. a), c)
23. a), e)
24. Octave bands divide the range of human hearing into 10 bands or ranges of frequencies where the upper band limit is twice the lower band limit. Bands are denoted by their center frequency, which is between the upper and lower band limits.
25. A point source is a sound source that is small in comparison to the wavelength it is generat ing. A line source is a sound source which generates a sound wave that has equal strength in a direction perpendicular to it axis for a distance of L/π.

Work Session 2 Answers

1. A sound meter measures sound pressure. A series of filters are used to filter certain frequencies and measure one of several scales, these may be individual octave bands, A-weighted, B-weighted, C-weighted and liner.
2. False, the ear is most sensitive to sounds in the frequency range from 500 Hz to 3000 Hz.
3. a), c)
4. True, the A-weighting scale uses a single number to define the sound level and tells you little about the frequency spectrum distribution and hence nothing about the sound quality.
5. c)
6. a), c)
7. False. Private offices should be designed for RC 25–30.
8. c), d)
9. d)
10. The A-weighted scale adjusts the sound pressure in each octave band to more closely match the way the human ear perceives sound. The B and C scales are weighted to provide correction when measuring high to moderate sound levels with less correction to lower bands than the A-weighted. The liner scale makes no correction by octave band and is a single number rating combining each of the octave bands
11. True
12. ARI , sound power
13. a), b), c)
14. e)
15. a), c), e)
16. a), b), c), d)
17. True
18. g)
19. a), b), c), d), e), f), g)
20. a), b), c), d)

Appendix

Glossary

Term	Definition
one-third-octave band	band of frequencies extending one-third of an octave from 0.891 Fc to 1.122 Fc, where Fc is the band-center frequency; identified by the geometric mean frequency of each band; ANSI S1.6 and ISO 266 standardize one-third octave band center frequencies.
absorption	absorption of acoustical energy by acoustical materials and air changing of sound energy to heat.
absorption coefficient	the fraction of sound energy that is absorbed at any surface; has a value between 0 and 1 and varies with the frequency and angle of incidence of the sound.
accelerometer	a transducer for measuring the dynamic acceleration (vibration amplitude) of a physical device. Piezoelectric accelerometers are the most common.
airborne transmission	sound that arrives at the point of interest, such as one side of a partition, by propagation through air.
ambient noise	all sounds in a given environment both near and far, none of which is a dominant sound.
amplitude	the immediate degree of intensity of an oscillating quantity such as sound pressure; expressed in three ways: displacement (commonly in mils peak-to-peak) velocity (commonly in inches/second) and acceleration (commonly in g's rms).
anechoic test chamber	a room specially constructed with walls and a floor covered with sound-absorbing wedges to create a free-field environment; used for precision-grade measurements.
atmospheric transmission	sound that arrives at the point of interest, traveling only through the air path.
A-weighted sound level	the sound level as measured on an A-weighted sound level meter, with filtering which approximates the relative loudness.
balanced sound	a sound spectrum that does not exhibit any tonal sounds and has a uniformly sloping difference of 5 dB per octave band.
bandwidth	the total frequency range of any system as it relates to sound; specified in the following format: 20-20,000 Hz plus or minus 3 dB.
beat	periodic fluctuations that are heard when sounds of slightly different frequencies are superimposed.
bels	a common logarithm of the ratio of two powers; two powers, one of which is ten times the other, will differ by 1 bel; any power ratio may be expressed in bels; a pure number with no dimensions.
blade passage frequency	tone produced by the rotation of the blades on a fan wheel or pump impeller, which varies as the rotational speed and the number of blades; can be perceived as a tone; harmonics and sub-harmonics are possible at multiples of the blade passage frequency.
broadband noise	a sound with a continuous spectrum consisting of a large number of frequency components, none of which is individually dominate; that is, energy is present at all frequencies in a given range; often referred to as noise because it usually lacks a discernible pitch.
B-weighted sound level	weighting scale similar to the A-weighting scale with less filtering of the low frequency.

casing-radiated sound	sound that propagates from the casing of a unit or air terminal and arrives at the point of interest.
C-weighted sound level	weighting scale similar to the A-weighting scale but with less filtering of the low frequency.
dB	abbreviation for decibels, use for sound power and sound pressure readings.
dBA	abbreviation for decibels measured using the A-weighted scale, used for sound pressure readings.
decibels	primary unit of sound measurement used to measure sound power level and sound pressure level; used to identify ten times the common logarithm of the ratio of two like quantities proportional to power or energy.
diffraction	the bending or redistribution of acoustic sound waves in a room caused by some obstacle, such as a column or divider; only low frequency waveforms can be diffracted.
direct field	a region in which the majority of sound energy arriving from the sound source at a point has traveled a straight line between them, that is, without reflection.
directivity factor	the ratio of sound pressure at a given angle to the sound source to the free field sound pressure level; a measure of the extent to which a source radiates sound in a certain direction.
displacement	a measure of the amount of vibration an object experiences(dynamic displacement) caused by sound; the distance that an object moves (static displacement).
distributing frequency	the number of oscillations per unit time of an external force or displacement applied to a vibrating system.
duct breakin	sound energy that is transmitted into a duct from the surroundings.
duct breakout	sound associated with fan or airflow noise that radiates through the duct walls into the surrounding area.
ductborne transmission	sound that arrives at the point of interest, transmitted through the duct system.
end reflection	accounts for the acoustic energy that is reflected back toward the source.
far field	region farther from the sound source where the sound pressure diminishes gradually with distance; in the far field, sound waves arriving from different points on the source are in phase; begins where the near field ends; in the far field, the sound pressure decreases inversely with distance from the source; corresponds to a reduction of approximately 6 dB in level for each doubling distance.
flanking transmission	transmission of sound from the source to a receiving location by a path other than that under consideration; sound that reaches the receiving room by paths other than through the test specimen; flanking transmission paths may be airborne or structure-borne.

free field	a sound field in a homogeneous, isotropic medium where there are no obstructions imposed on sound waves; Sound waves in a free field are spread out continuously like ripples on a pond without reflections; opposite of a free field is a diffuse field, where multiple reflections result in sound incidence from all directions. an anechoic room is a good approximation to a free field.
frequency	the number of oscillations of cycles per unit of time for a vibrating object.
hissy	a description of a high frequency sound that is greater than 3 dB above the neutral sound spectrum; typical of diffuser noise.
insertion loss	the amount of sound reduction after the insertion of a noise control device in a given location; a silencer or other sound-reducing element, in a specified frequency band, will decrease then sound power level, measured at the location of the receiver, when a sound insulator or a sound attenuator is inserted in the transmission path between the source and the receiver.
line source	a sound source composed of many point sources arranged next to each other in a line; when the distance becomes longer than the frequency wavelength, the line source becomes a point source.
Lin-weighted sound level	a scale on which each equal length division represents an equal number of Hertz; a single number that represents a numeric average of all octave bands with no frequency filtering.
loudness level	a subjective measurement of how loud or soft a listener perceives a certain sound; a function of both the intensity (energy) and frequency of the sound.
masking	the amount (or the process) by which the threshold of audibility for one sound is raised by the presence of another (masking) sound.
natural frequency	the frequency of free oscillation of a system. For a multiple-degree-of-freedom system, the natural frequencies are the frequencies of the normal modes of vibration. The undamped resonance frequency of a physical system. It is expressed in cycles per unit time.
near field	locations close to the sound source between the source and the far field where the sound pressure fluctuates about a mean value; in the near field, sound waves arriving from different points on the source are out of phase and interfere constructively and destructively; the depth of the near field is a function of the geometry of the source, the measurement location, and the wavelength of sound; typically characterized by large sound pressure level variations with small changes in measurement from the source; generally within a distance from the source of less than two wavelengths.
neutral spectrum	balance across the entire frequency spectrum of the audio range.
noise	undesirable sound that is annoying or obstructs listening.
noise criteria curve (NC)	standard spectrum curves by which a given measured noise may be described by a single NC number.
noise reduction (NR)	amount of noise that is reduced using sound absorbing material; rated as the difference in sound pressure level between any two points along the path of sound propagation; used to describe the difference in sound pressure levels between the inside and outside of an enclosure.
noise reduction coefficient	the arithmetic average, to the nearest multiple of .05, of the sound absorption coefficients in the 1/3 octave bands centered at 250Hz, 500Hz, 1000Hz, and 2000 Hz.

octave band	frequency ranges in which the upper limit of each band is twice the lower limit. Octave bands are identified by their geometric mean frequency, or center frequency.
octave	an octave is a doubling or halving of frequency.
peak-to-peak	the amplitude difference between the most positive and most negative value in the time waveform.
Permissible Exposure Limit (PEL)	provides numeric standards that determine how long a worker may be exposed in the workplace to a threshold amount sound without adverse effects on hearing.
phons	a unit of perceived loudness; subjective measure of the strength (not intensity) of a sound; at a frequency of 1 kHz, 1 phon is defined to be equal to 1 dB of sound pressure level above the nominal threshold of hearing.
pink noise	noise with a continuous frequency spectrum and with equal power per constant percentage bandwidth. For example, equal power is any one-third octave band.
plenum/ceiling effect	difference between the octave band sound power level from the source located in the plenum / ceiling cavity and the sound pressure received in the space.
point source	a source of acoustic sound waves having definite position but no extension into space; an ideal single point in space; provides a good approximation for distances from the point source that are much larger compared to the actual dimensions of the source.
psil	Preferred Speech Interference Level; an indicator of the ability to understand the spoken word, numerically it is the average of the 500, 1000, and 200 Hz octave bands.
pure tones	a periodic sound consisting of a single frequency component; also referred to as a discrete tone.
radiated sound power	sound power transmitted from an air terminal or unit casing, (on fan powered air terminals it includes the induction port sound).
random noise	a noise signal that never repeats and has a flat frequency spectrum. Also known as white noise; a noise signal, commonly used in measurements, which has constantly shifting amplitude, phase, and a uniform spectral distribution of energy.
reference sound source	a portable aerodynamic sound source that produces a known and stable broad band sound output.
reflection	the bouncing back or return of sound waves from walls or other obstacles which they strike; for large surfaces compared to the wavelength of impinging sound, sound is reflected much as light is reflected, with the angle of incidence equaling the angle of reflection.
refraction	the bending of sound waves traveling through layered media with different sound velocities; a change in direction or bending of the propagation of a sound wave when it passes from one medium to another in which the velocity of sound is different.
resonance	a natural periodicity of a sound or vibration wave, or the reinforcement associated with this periodicity of the wave.
reverberant field	a region in which the majority of sound energy arriving from the sound source at a point has been reflected at least once; in the statistical limit of an infinite number of reflections, the reverberant field approaches diffuse field conditions.
reverberant test chamber	a room designed so that the reverberant sound field closely approximates a diffuse sound field, both in the steady state when the sound source is on, and during the decay after the source of sound has stopped.

reverberation	the persistence of a sound after its source has stopped.
reverberation time	at a point in an enclosure, and for a stated frequency or frequency band, the time required for the level of the mean-square sound pressure to decrease by 60 dB after the source has been stopped; symbol is T60.
rms velocity	the square root of the time average of the square of a quantity; for a periodic quantity the average is taken over one complete cycle; rms voltage is 0.707 times the peak voltage of a sine wave.
room criteria (RC)	single number noise rating system used to diagnose and rate the noise exposure in a room; best for rating noise with strong low frequency content; undesirable rumble can result if NC curves are determined mainly by low frequency noise; a hissing effect can result from NC level being controlled by higher frequency sounds; established to design spectra that meet an RC curve within + 2 dB at all frequencies to achieve a better balance between low frequency and high frequency components; a spectrum that exceeds an RC curve by more than 5 dB at frequencies below 250 Hz is likely to result in unacceptable rumble; above 2000 Hz, a spectrum more than 5 dB higher than the RC curve might have too much of a hissing quality.
room-to-room transmission	sound that arrives at the point of interest, transmitted through the walls of one room to the air in the adjoining room.
rumbly	a description of a low frequency sound that is greater than 5dB above the neutral sound spectrum.; typical of fan and reciprocating compressor noise.
Sabin	unit of sound absorption based on one square foot of material the unit of measure of sound absorption in the inch-pound system.
sones	unit of measurement for subjective loudness.
sound	energy that is transferred through a fluid as the result of a series of minute pressure pulses caused by alternate compressions and expansions of the fluid.
sound absorption	process of dissipating sound energy; accomplished by materials, objects, and structures; measured in a specified frequency band, the measure of the magnitude of the absorptive property of a material, an object, or a structure such as a room; at a specified frequency or frequency band, the process or material property whereby sound energy is converted into heat.
sound absorption coefficient (α).	amount of energy striking a material or object that is not reflected; the measure of the absorptive property of a material.
sound intensity	amount of sound power per square meter passing through a surface. In a progressive wave, it is proportional to the average magnitude of the quantity Measurement of sound intensity is W/m2.
sound power	in a specified frequency band, the rate at which acoustic energy is radiated from a source; the rate of flow of sound energy, whether from a source, through an area, or into an absorber; of airborne sound, ten times the common logarithm of the ratio of the sound power under consideration of the standard reference power of 1 pW, the quantity obtained is expressed in decibels; a function of the sound source.
sound pressure	the level, in decibels, of acoustic pressure waves; the dynamic variation of the static pressure of air; measured in force per unit area; total pressure of a point in space in the presence of a sound wave, minus the static pressure at that point; given in decibels (dB) is an expression of loudness or volume; a 10 dB increase in sound pressure level represents a doubling in volume; dependent on the distance and the path from the sound source.

sound quality	overall human experience of the pleasantness or desirability of a sound in its appropriate context.
Sound Transmission Class (STC)	measures the ability of a material or an assembly to resist airborne sound transfer at the frequencies between 125-4000 Hz; high STC rating denotes more noise being blocked from transmitting through a partition; provides an estimate of the performance of a partition in certain common sound insulation problems.
source-path-receiver concept	method of analyzing sound design issues by identifying the source and acoustic power of the sound, the location and acoustical requirements of the receiver and the paths and attenuating components between the source and receiver.
space effect	factor that is used to calculate the sound pressure level in a "real room" based on the room volume and distance to the receiver based on the sound power entering the space; equals the difference between sound power levels and sound pressure levels (Lw - Lp) and is always a positive number.
standing waves	a stationary spatial pattern of locally high and low amplitude; results from the interference of two or more progressive waves of the same frequency moving in different directions; occur in rooms because of boundaries.
structure-borne transmission	generation and propagation of time-dependent motions and forces in solid materials which result in unwanted radiated sound that is transmitted in the form of vibration and re-radiated at another point.
tone	the speed or frequency at which we perceive sound waves. A tone results in an auditory sensation of pitch.
tone-corrected	a correction made to frequency spectrum data to correct for small tones, which occur in a frequency band.
total environmental quality	an evaluation of the total environment of a space including the temperature, humidity, air motion, air quality, acoustics, lighting and other environmental factors.
transmissibility	The amount of vibration energy that is transmitted through a vibration isolation device.
transmission loss (TL).	is a measure of the reduction in sound power due to absorption; Symbol is τ.
vibration	the motion of a physical body. Measured in acceleration, velocity, or displacement.
wavelength	the distance the wave travels in feet in one cycle; the distance between one peak or crest of a sine wave and the next corresponding peak or crest; symbol is λ.
weighting filters	a family of standardized broad-band filters that attempt to simulate human response to sound by the analyzer; includes filters such as A-, B-, C-, D-, and E-weighting.
white noise	Noise with a continuous frequency spectrum and with equal power per unit bandwidth. The noise signal that does not repeat, and for which the spectral energy per Hz is independent of frequency. The spectrum looks "flat" on a spectrum analyzer display, and increases with frequency on an octave- or one-third octave band display (because there are more Hz per band at higher frequency). White noise is not necessarily random.
zero-to-peak	The amplitude difference between static position and the most positive or most negative value in the time waveform.

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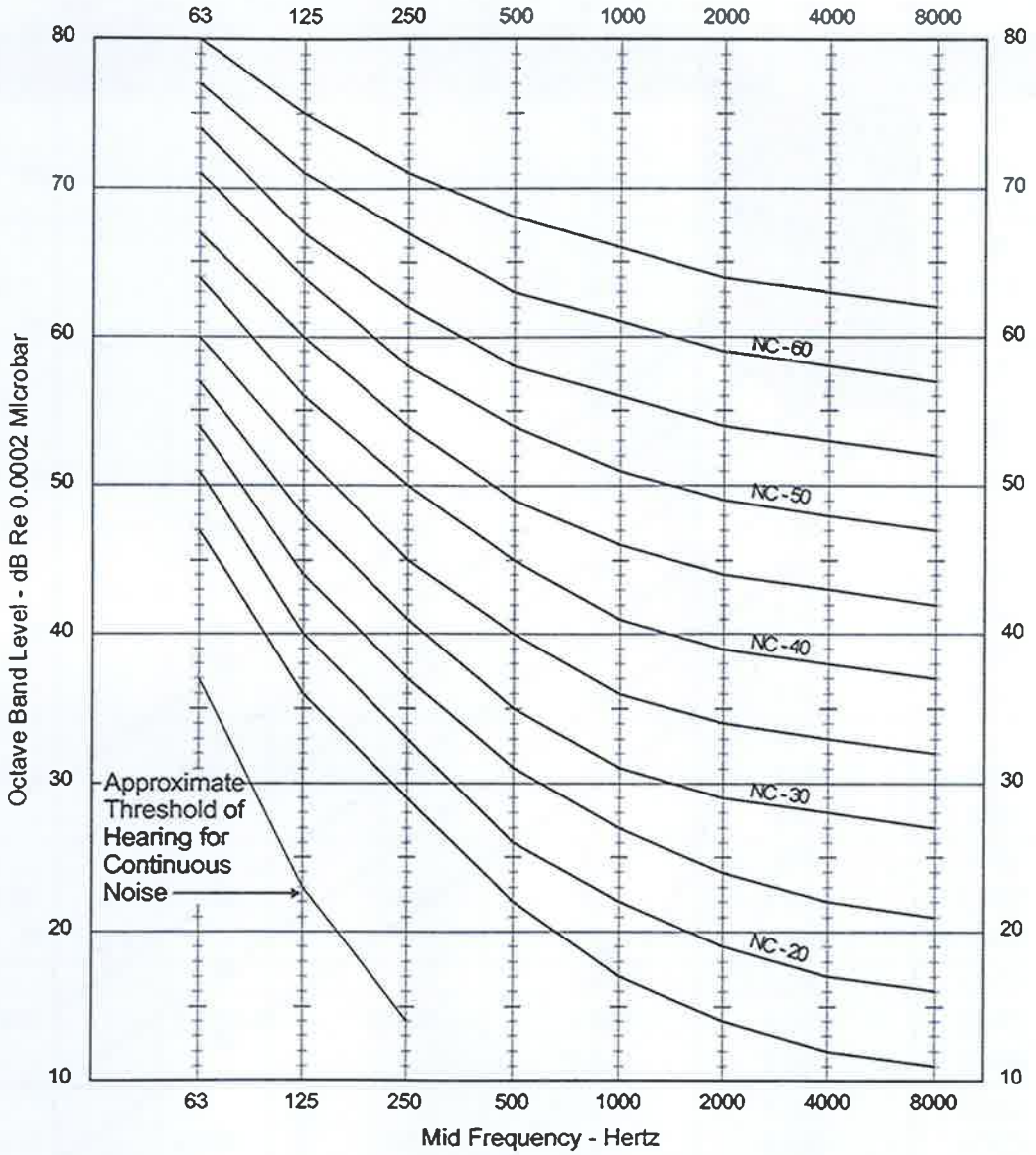
HVAC Unit Acoustical Rating Programs

Device	Type	Standard	Title
Indoor Units - Non-Ducted • Room Fan Coil • WSHP • Free Blow In-door Packaged Units • Duct-free Splits	Ratings	ARI 350-2000	Sound rating of non-ducted indoor air-conditioning equipment
	Indoor ratings for all non-ducted units mounted within the space, ratings in octave band sound power levels 125 Hz through 8000 Hz, A-weighted and tone corrected A-weighted sound pressure at 1 meter, all rated required at high fan speed. Application ratings may also be published at other fan speeds.		
Package Terminal Air Conditioners (PTAC)	Ratings	ARI 300-2000	Sound rating and sound transmission loss of packaged terminal equipment
	Ratings for package terminal air conditioners, units mounted within the space and operated at full capacity, ratings are for indoor, outdoor and transmission from indoor to outdoor, ratings in octave band sound power levels A-weighted and tone corrected A-weighted sound pressure at 1 meter, all rated required at high fan speed. Ratings reference ARI 270 and ARI 350		
Indoor Ducted Units • Rooftops • Air Handlers • Indoor Packaged	Ratings	ARI 260-2001	Sound rating of ducted air moving air condition equipment (with addendum)
	The primary ARI sound rating standard for ducted inlet and outlet sound as well as cabinet radiated sound. Utilized AMCA 300 and ANSI S12.12 reverberant room methods.		
Air Terminals	Certification	ARI 880 - 1998	Air Terminals
	Test method used rate air terminal devices like VAV boxes, covers nine different terminal types, certified ratings are in octave band levels from the 125 Hz through 4000 Hz bands.		
	Application	ARI 885 - 1998	Procedure for estimating Occupied Space Sound Levels in the application of air terminals and outlets
	Application method which allows the calculation of the discharge and radiated sound pressure levels in a space accounting for all the paths and attenuation in each path		
Grilles, Registers and Diffusers	Ratings	ASHRAE 70-1991	Method of testing for rating the performance or air outlets and inlets
	Air outlets and inlets are tested for acoustical performance in reverberant test rooms qualified to ANSI S12.51 and ISO 3741. This standard requires that an air distribution device be installed with several diameters of straight duct to avoid the introduction of duct-generated noise. Typically, the sound power is then reduced by 10 dB in every octave band for room effect, and plotted on an NC chart. Data is typically presented as NC values over a range of inlet velocities. Application guidance: In practice of course, there is seldom this ideal configuration, and rooms are not likely to absorb 10 dB in every octave band. It is accepted practice of many acoustical consultants that 5 dB be added to the catalog data to account for inlet effects and real room absorption.		
	Ratings	ARI 890-2001	Air Diffusers and air diffuser assemblies
	ARI has a GRD standard, but has not implemented a certification program. The Air Diffusion Council had a certification program, ADC 1062. Last updated in 1984, the ADC 1062 standard and certification program are no longer active, and there are presently no ADC certified products or laboratories.		
	Applications	ARI 885-1998	Procedure for estimating occupied space sound levels in the application of air terminals and air outlets
	This ARI standard can be used to determine the application rating of a diffuser in a space with the impacts from the VAV boxes and duct system		
Packaged Sound attenuators	Certification Program	AMCA 1011-1999	Certified ratings program – acoustical duct silencers
	Under the AMCA certification program unit's ratings are check tested every three years. Ratings include insertion loss in octave bands and pressure drop at a rated airflow		
	Ratings	ANSI E477-1999	Standard test method for measuring acoustical and airflow performance of duct liner materials and prefabricated silencers
	Ratings	ASTM E90.2	Standard test method for laboratory measurement of airborne sound transmission loss

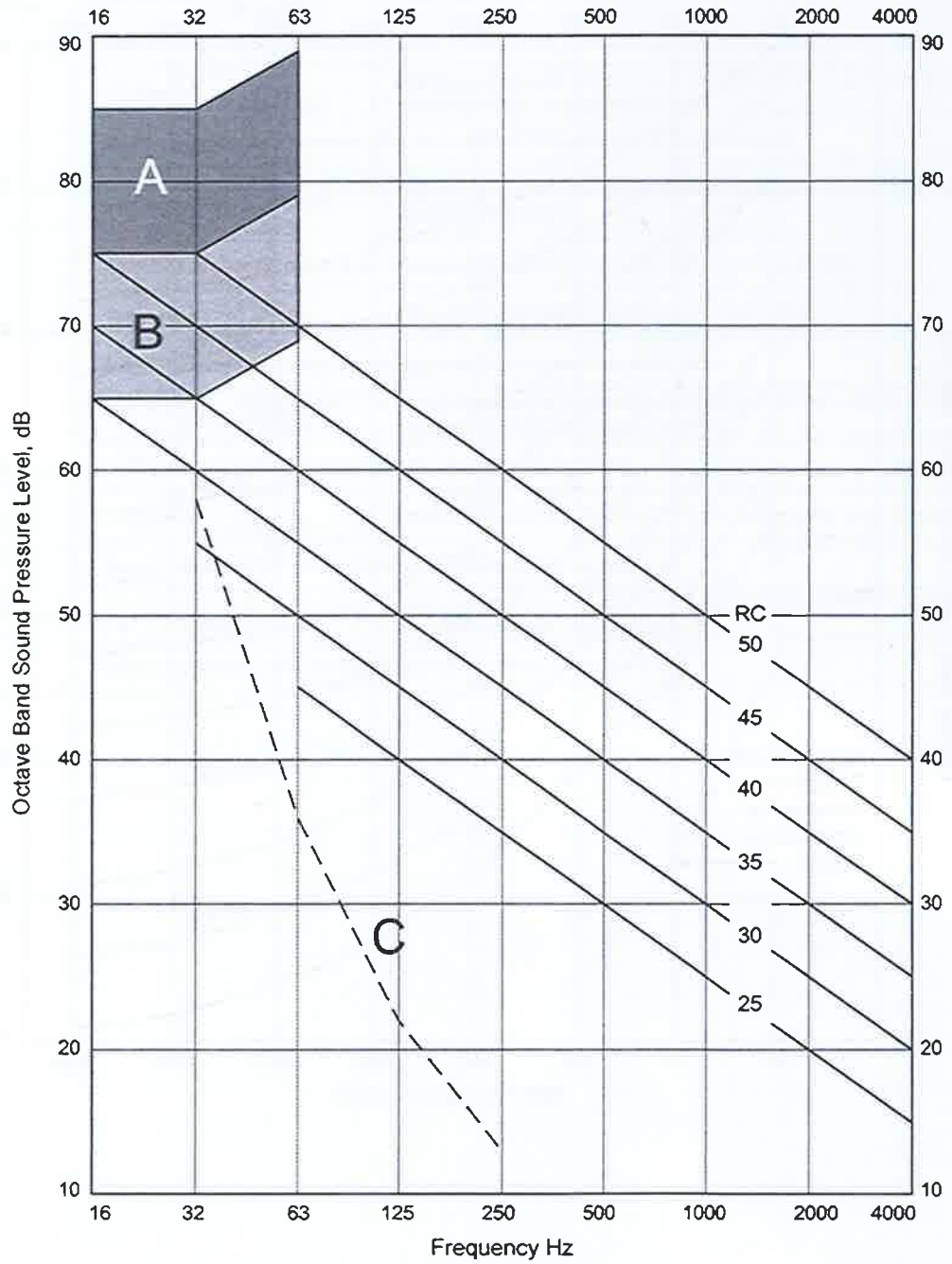
Chillers	Application	ARI 575-1994	Method of measuring machinery sound within an equipment space
	Method to determine by field measurement sound levels of equipment in equipment rooms. Sound is reported in A-weighted sound level and octave bands		
Fans	Certification Program	AMCA 311-1990	Certified sound ratings program for air moving devices
	The AMCA Certification program conducts check tests every three years on each fan line		
	Ratings	AMCA 300-1996	Reverberant room method for sound testing of fans
	The primary fan sound ratings standard. This standard addresses fans in different arrangements (ducted and non-ducted) and addresses inlet, outlet, and cabinet-radiated noise. It is the basis for ARI 260 sound testing.		
	Ratings	ANSI/AMCA/ ASHRAE 33-1997 68-1997	In duct , laboratory method of testing to sound power
	A method for rating fans by testing in a specifically designed test duct.		
	Ratings	ARI 260-2001	Sound rating of ducted air moving air condition equipment (with addendum)
	Sound rating standard for ducted inlet and outlet sound as well as cabinet radiated sound. May also be used for stand alone fans with or without a cabinet		
	Application	AMCA 302-1973	Application of sone ratings for non-ducted air moving devices
	A standard for non-ducted fans that rates fans in terms of loudness instead of sound power ratings.		
	Application	AMCA 303-1979	Application of sound power level ratings of fans.
	Notes: An applications standard that discusses how AMCA 300 sound power rating should be applied		
	Application	ANSI/AMCA 204-1996	Balance quality and vibration levels for fans
	Vibration standards for both factory and field fan balance		
	Application	AMCA 301-1990	Methods for calculating fan sound ratings room laboratory test data
	Describes how to interpolate points between test data developed in Standard 300 for different fan sizes and operating points.		
Compressors	Ratings	ARI 530-1995	Method of rating sound and vibration of refrigerant compressors
	Sound rating standard for stand-alone compressors includes a vibration rating		
Outdoor Ratings Units less than 5 tons • Rooftops • Split system condensing units	Ratings	ARI 270-1995	Sound rating of outdoor unitary equipment
	Rating test procedure for determining outdoor radiated sound. Sound values are reported in A-weighted values.		
	Applications	ARI 275-1997	Application of sound rating levels of outdoor unitary equipment
	Applications standard to using ratings from ARI 270 and applying the results to determine the sound pressure level at a given location		
	Guideline	ARI Guideline L-1997	Assessing the impact of air-conditioning outdoor sound levels in the residential community
	Information on the likelihood of sound generated complaints when applying ARI 270 sound data		
Outdoor Ratings Units greater than 5 tons • Rooftops • Condensing units • Air Cooled Chillers	Ratings	ARI 370-2001	Sound rating of large outdoor refrigerating and air-conditioning equipment
	Used to determine outdoor sound levels in a reverberant room, or in the field. Sound values are reported in octave band, A-weighted and tone corrected A-weighted with unit at full operating condition, application ratings at other conditions are permitted		

Laboratories & Test Facilities	Lab standard	ARI 280-1995	Requirements for the qualification of reverberant rooms in the 63 Hz octave band
	Reference sound source qualification		
	Test Method	ANSI S12.51-2002	Determination of sound power levels of noise sources using sound pressure – Precision methods for reverberation rooms
	This is the basis of most reverberant room test standards, Replaced ANSI S12-31 and S12.32		
	Test method	ANSI S12.12-1992 (R2002)	Engineering method for the determination of sound power levels of noise sources using sound intensity.
	Intensity is an alternate method of determining sound power, not yet in wide use for HVAC applications.		
	Test method	ARI 250-001	Performance and calibration of reference sound sources
	Used to calibrate the reference sound source used in reverberant room sound tests		
	Test method	ASTM E90-2002	Standard test method of Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions and Elements
	Used to determine the STC rating of building structural wall components		
	Test method	ISO 9614-2002	Determination of sound power levels of noise sources using sound intensity
	Sound intensity method is not in common usage for most HVAC applications		
	Test method	ISO 3741-1999	Determination of sound power levels of noise sources using sound pressure – Precision methods for reverberation rooms
	Notes: ISO 9614 and 3741 are the ISO equivalents of ANSI S12.51 and S12.12		

Noise Criteria Curves



Room Criteria Curves



Region A: High probability that noise-induced vibration levels in lightweight wall and ceiling constructions will be clearly feelable, anticipate audible rattles in light fixtures, doors, windows, etc.

Region B: Noise-induced vibration levels in lightweight wall and ceiling constructions may be moderately feelable; slight possibility of rattles in light fixtures, doors, windows, etc.

Region C: Threshold of hearing.

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

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

A-Weighting Factor Adjustments

Frequency (Hz)	Adjustments for 1/3 Octave Bands (dB)	Octave Bands
50	-30.2	
63	-26.3	-26.3
80	-22.5	
100	-19.1	
125	-16.1	-16.1
160	-13.4	
200	-10.9	
250	-8.6	-8.6
315	-6.6	
400	-4.8	
500	-3.2	-3.2
630	-1.9	
800	-0.8	
1000	0.0	0.0
1250	0.3	
1600	1.0	
2000	1.2	1.2
2500	1.3	
3150	1.2	
4000	1.0	1.0
5000	0.5	
6300	-0.1	
8000	-1.1	-1.1
10,000	-2.5	

1/3-Octave Band Adjustments for Tone Response

1-3 Octave Band Frequency	Projection of 1-3 Octave Band above the arithmetic average of the two adjacent bands.												
	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
50	†	†	†	†	†	†	†	†	†	†	†	†	†
63	†	†	†	†	†	†	†	†	†	†	†	†	†
80	†	†	†	†	†	†	†	†	†	†	†	†	†
100	†	†	†	†	†	†	†	†	†	†	†	†	†
125	†	†	†	†	†	†	†	†	†	†	†	†	†
160	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2	0.2	0.2	0.2
200	0.5	0.5	0.6	0.7	0.7	0.8	0.8	0.9	0.9	0.9	1.0	1.0	1.0
250	0.8	1.0	1.1	1.2	1.3	1.4	1.5	1.5	1.6	1.6	1.7	1.7	1.7
315	1.2	1.4	1.6	1.7	1.8	1.9	2.0	2.1	2.2	2.2	2.3	2.3	2.4
400	1.5	1.8	2.0	2.1	2.3	2.4	25	2.6	2.7	2.8	2.8	2.9	3.0
500	1.8	2.1	2.3	2.5	2.7	2.8	2.9	3.0	3.1	3.2	3.3	3.3	3.4
630	2.1	2.4	2.6	2.8	3.0	3.2	3.3	3.4	3.5	3.6	3.7	3.7	3.8
800	2.3	2.6	2.9	3.1	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.1	4.1
1000	2.5	2.8	3.1	3.4	3.6	3.7	3.9	4.0	4.1	4.2	4.3	4.4	4.4
1250	2.7	3.0	3.3	3.6	3.8	4.0	4.1	4.3	4.4	4.5	4.6	4.6	4.7
1600	2.9	3.2	3.6	3.8	4.0	4.2	4.4	4.5	4.6	4.7	4.8	4.9	5.0
2000	3.0	3.4	3.7	4.0	4.2	4.4	4.6	4.7	4.8	4.9	5.0	5.1	5.2
2500	3.1	3.5	3.9	4.1	4.4	4.5	4.7	4.9	5.0	5.1	5.2	5.3	5.3
3150	3.2	6.3	4.0	4.2	4.5	4.7	4.8	2.0	5.1	5.2	5.3	5.4	5.4
4000	3.3	37	4.0	4.3	4.5	4.7	4.9	5.0	5.2	5.3	5.4	5.4	5.5
5000	3.3	3.7	4.0	4.3	4.5	4.7	4.9	5.0	5.1	5.3	5.4	5.4	5.5
6300	3.2	3.6	4.0	4.2	4.5	4.7	48	5.0	5.1	5.2	5.3	5.4	5.5
8000	3.2	6.3	3.9	4.2	4.4	4.6	4.8	4.9	5.0	5.1	5.2	5.3	5.4
10000	†	†	†	†	†	†	†	†	†	†	†	†	†

† Tone Adjustments are not applicable to frequencies below 160 Hz or above 8000 Hz.

Room Acoustics Calculation Worksheet

Radiated Sound Calculation											
Step	Calculation	Symbol	ARI 885	63	125	250	500	1000	2000	4000	8000
1	Room Design goal NC		Table 16								
2	Environmental Adjustment	E	Table C.7	4	2	1	0	0	0	0	0
3	Ceiling / Space Effect	P	Table D.14								
4	Allowed Radiated										
5	Selected Terminal Radiated		Manufacturer								
6	Additional Attenuation Required										

Discharge Sound – Terminal Unit											
Step	Calculation	Symbol	ARI 885	63	125	250	500	1000	2000	4000	8000
1	Room design goal NC		Table 16								
2	Environmental Adjustment	E	Table C.1								
3	Space Effect	S	Table D.16								
4	Multiple Outlet	S2	Table D.17								
5	End Reflection	R	Table D.13								
6	Branch Power Division	F	Table D.2								
7	Elbow Attenuation	T	Table D.4								
8	Duct Breakout	B	Table D.2								
9	Duct Attenuation	I	Table D.3								
10	Sum of Adjustments		Sum 2 to 9								
11	Terminal Unit Correction	M									
12	Required Terminal		1-10 + 11								
13	Selected Terminal Discharge		Manufacturer								
14	Additional Attenuation Required										

Discharge Sound – AHU Unit											
Step	Calculation	Symbol	ARI 885	63	125	250	500	1000	2000	4000	8000
1	Room design goal NC		Table 16								
2	Environmental Adjustment	E	Table C.1								
3	Space Effect	S	Table D.16								
4	Multiple Outlet	S2	Table D.17								
5	End Reflection	R	Table D.13								
6	Branch Power Division	F	Table D.2								
7	Elbow Attenuation	T	Table D.4								
8	Duct Breakout	B	Table D.2								
9	Duct Attenuation	I	Table D.3								
10	Sum of Adjustments		Sum 2 to 9								
11	Terminal Unit Correction	M									
12	Required AHU Discharge		1-10 + 11								
13	Selected AHU Discharge		Manufacturer								
14	Additional Attenuation Required										

Return Sound – AHU Unit											
Step	Calculation	Symbol	ARI 885	63	125	250	500	1000	2000	4000	8000
1	Room design goal NC		Table 16								
2	Environmental Adjustment	E	Table C.1								
3	Space Effect	S	Table D.16								
4	Multiple Outlet	S2	Table D.17								
5	End Reflection	R	Table D.13								
6	Branch Power Division	F	Table D.2								
7	Elbow Attenuation	T	Table D.4								
8	Duct Breakout	B	Table D.2								
9	Duct Attenuation	I	Table D.3								
10	Sum of Adjustments		Sum 2 to 9								
11	Terminal Unit Correction	M									
12	Required AHU Return		1-10 + 11								
13	Selected AHU Return		Manufacturer								
14	Additional Attenuation Required										



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