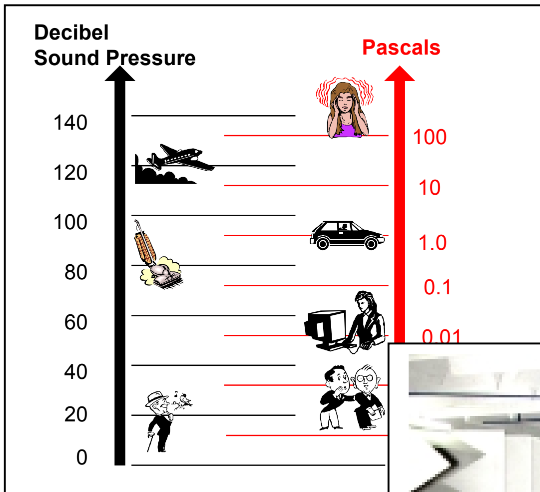
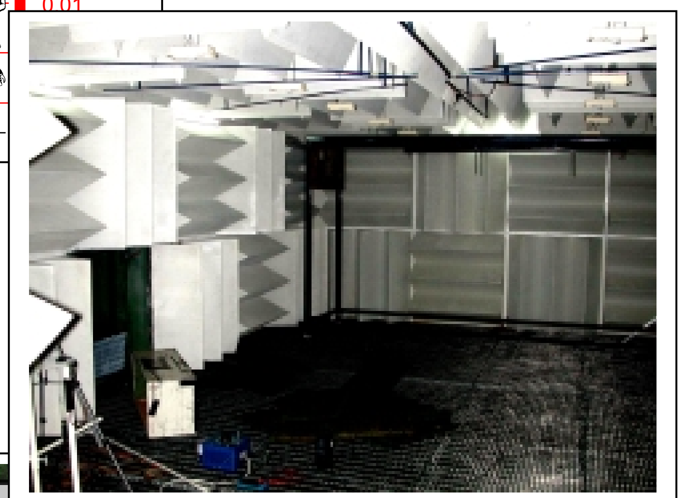
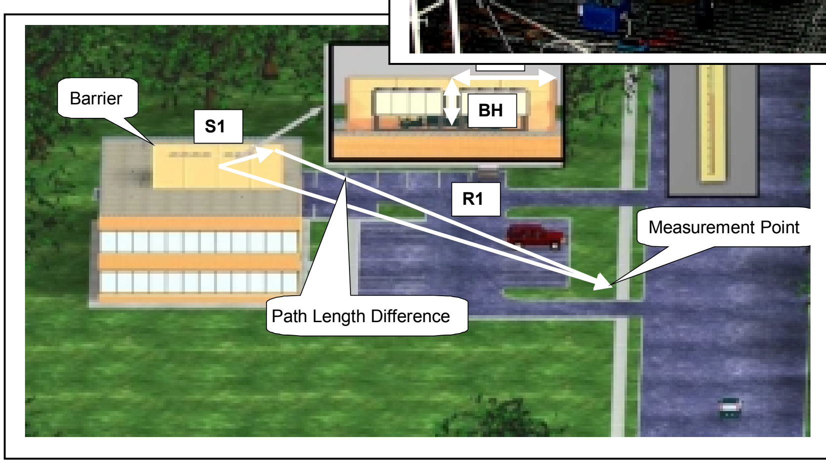
Application Guide AG 31-010

HVAC Acoustic Fundamentals







McQuay Air Conditioning

Engineered for flexibility and performance.™

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Introduction

Occupant comfort is the goal of all HVAC designers. Sound (or noise) is a key parameter in measuring comfort, in addition to temperature, humidity and Indoor Air Quality (IAQ). While acoustics consultants are usually involved in critical applications (such as performing arts centers), the task of creating a comfortable acoustic environment in most other applications falls on the HVAC engineer. This is because most background sound sources are generated by the HVAC equipment.

The purpose of this manual is to familiarize the designer will the basics of acoustics, and to apply these basics to typical HVAC designs.

Using This Manual

The manual can be used as an application guide or as a primer for using the McQuay Acoustic Analyzer™ software program. The equations and approach described herein were used in the creation of the Acoustic Analyzer program. Small differences can exist between the software and the manual because the Acoustic Analyzer program uses equations to estimate values as opposed to the tables listed in the manual. However, the difference in values will be very small.

Examples that show how to perform some analyses are included in double lined boxes. Helpful tips are also provided.

☺ *Tip; In this manual, Sound Pressure will be indicated by Lp RE 10^{-12} Pa, while Sound Power will be indicated by Lw RE 10^{-12} W.*

“Tips” box example

Decibel Example
Calculate the loudest possible sound at standard atmospheric pressure (101.3 kPa)

Lp = 20Log(101,300/0.00002)

= 194 dB RE 20μ Pa

Note: 20μ Pa is another way to write 0.00002 Pa.

Information in double lined boxes show calculation examples

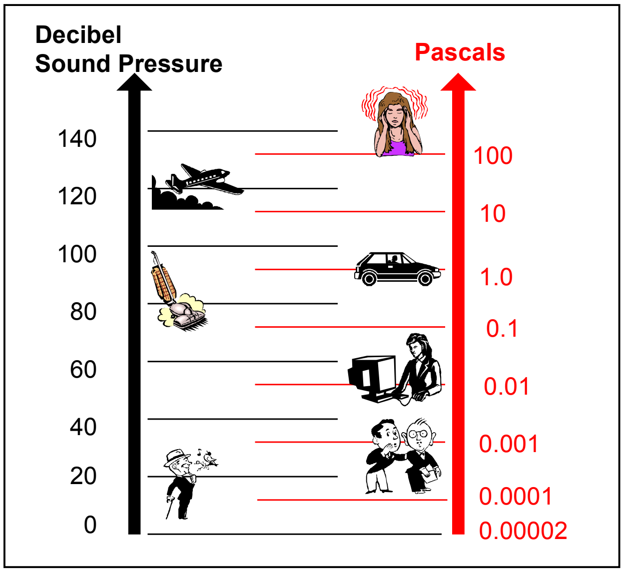
Sound Basics

General

Sound is defined as a disturbance in an elastic medium that can be detected by the human ear. The medium can be gas, liquid or solid. Noise is undesirable sound or sound without value. The pressure waves (or sound) act on the inner ear, which is what we hear. The best sound is not necessarily no sound. In an open office concept, background sound offers privacy for conversations. The quality of sound is also important. Tonal sounds are usually not desirable.

The following list will help better understand sound.

Figure 1- Typical Sound Pressure Levels



- The amplitude of the sound wave represents the loudness and is measured in decibels (Pascals). The louder the sound, the larger the amplitude. The loudest atmospheric sound has zero atmospheric pressure at the low point and two times atmospheric pressure at the high point. This is 194 dB.
- The frequency of sound represents the pitch and is measured in Hertz. The higher the frequency, the higher the sound. The human hearing range is from about 16 Hz to 16,000 Hz. Below 30 Hz, sound can be felt as well as heard.
- The wavelengths for sound can vary from 70 feet (21.3 m) at 16 Hz to 0.07 feet (0.02 m) at 16,000 Hz. This is important because sound absorbing materials tend to work well when their dimensions are close to the wavelength. Therefore, a 1-inch (25 mm) ceiling tile is effective at absorbing higher frequency sounds, but low frequency sounds are much more difficult to attenuate.
- The human ear can respond to very wide range of sound levels. At the low end, the ear is sensitive to sound pressure waves as little as 0.00002 Pa. At the high end, the human ear can hear about 20 Pa without pain, which is 1,000,000 times louder. This is a key reason why Decibel logarithmic scales are used.
- The speed of sound is dependent on the density of the medium it is travelling through. The lower the density, the slower the sound wave. At standard atmospheric conditions, the speed of sound (Mach 1) is 764 miles per hour (1120 feet per second, 341 m/s).
- Sound waves do not actually pass through walls or other solid objects. Instead, they impinge on the exterior surface of the wall or object, causing it to vibrate. This, in turn, causes the air molecules in the space to vibrate. What is actually happening is that the sound wave is making the wall or object move!

Wavelength and Frequency

The wavelength of sound in air is given by;

Eq. 1

$\lambda = c_o / f$

Where

λ is the wavelength in feet (m).

c_o is the speed of sound, which is 1120 feet per second (341 m/s) at sea level.

f is the frequency in Hz.

Decibels

The very large range in sound pressure makes a logarithmic scale more convenient. Decibels (dB) are always referenced to base signal. Knowing the base reference is critical because the term "decibels" in acoustics is used for sound pressure and sound power. In the case of sound pressure, the reference is 0.00002 Pascals, which is the threshold of hearing.

Decibel Example
Calculate the loudest possible sound at standard atmospheric pressure (101.3 kPa)

Lp = 20Log(101,300/0.00002)

= 194 dB RE 20μ Pa

Note: 20μ Pa is another way to write 0.00002 Pa.

Eq. 2

$Lp = 20 \mathrm{Log}(P/0.00002)$

Where

Lp is the sound pressure in Decibels (dB)

P is the sound pressure in Pascals.

For sound power the reference is 10^{-12} Watts.

Eq. 3

$Lw = 10 \mathrm{Log}(W/10^{-12})$

Where

Lw is the sound power in Decibels (dB)

W is the sound power in Watts

Decibel Addition and Subtraction

Since Decibels are logarithmic, they cannot simply be added. For instance, 40 dB + 40 dB is not 80 dB, it is 43 dB. Decibels can be added as follows:

Eq. 4

$Ls = 10 \mathrm{Log}(10^{L1/10} + 10^{L2/10} + 10^{L3/10} + ...)$

Decibels can be quickly added together with an accuracy of about 1 dB by using the relationship shown in ***Table 1- Decibel Addition Chart.***

Decibel Addition Example
Add the follow values together;
L_1 = 80 dB
L_2 = 82 dB
L_3 = 84 dB
L_4 = 93 dB
L_5 = 72 dB

$L_{Total} = 10 \mathrm{Log}(10^{80/10} + 10^{82/10} + 10^{84/10} + 10^{93/10} + 10^{72/10})$
= 94 dB

Using ***Table 1- Decibel Addition Chart***
Between L_1 and L_2, there is a difference of 2 dB so add 2 dB to L_2 for a total of 84 dB.
Between 84 dB and L_3, there is a difference of 0 dB so add 3 dB to 84 dB for a total of 87 dB.
Between 87 dB and L_4, there is a difference of 6 dB so add 1 dB to L_4 for a total of 94 dB.
Between 94 dB and L_5, there is a difference of 22 dB so add 0 dB to 94 dB for a total of 94 dB.

Table 1- Decibel Addition Chart

When Two Decibel Values Differ By	Add The Following Number To The Higher Number
0 or 1 dB	3 dB
2 or 3 dB	2 dB
4 to 9 dB	1 dB
10 dB or more	0 dB

Decibel subtraction is accomplished as follows:

Eq. 5

$$Ls = 10\text{Log}(10^{L1/10} - 10^{L2/10} - 10^{L3/10} - ....)$$

Sound Pressure vs. Sound Power

What you hear is sound pressure. It is the fluctuation in the atmospheric pressure that acts on your eardrum. However, sound pressure is dependant on the surroundings, making it a difficult means to measure the sound level of equipment. Sound power is the sound energy released by a sound source. It cannot be "heard", but it can be used to estimate the sound pressure levels if the space conditions are known.

Sound pressure and sound power are best explained with an example. Consider a 5 kW electric baseboard heater. The 5 kW rating is a clear, definable measure that can be used to compare one heater against another. This is the equivalent of Sound Power. However, it not possible to know whether a 5 kW heater is sufficient to keep the occupant warm and comfortable unless the temperature of the space is known. The temperature of the space is the equivalent of sound pressure. If the 5 kW heater is used in a small, single room addition to a house, it will probably provide a comfortable temperature. If the heater is used in the Toronto Skydome, it is unlikely to offer any comfort to occupants. In each application the same size heater (or sound power level) provides very different thermal comfort results (or sound pressure level).

Knowing the Sound Power levels for a piece of equipment (e.g. a fan coil) will allow a fair and direct comparison of two models. It will not, however, indicate whether the sound level will be acceptable until the space is defined. Knowing the sound pressure of a piece of equipment (e.g. a cooling tower) will allow two models to be compared (assuming the same conditions were used for testing both units). However, unless the actual space where the product is used has the same properties as the test conditions, the sound pressure level provided will not be what the occupant experiences.

In this manual, Sound Pressure will be indicated by Lp RE 20µ Pa and Sound Power will be indicated by Lw RE 10^{-12} W.

☺ *Tip; In this manual, Sound Pressure will be indicated by Lp RE 10^{-12} Pa and Sound Power will be indicated by Lw RE 10^{-12} W.*

Octave Bands

Acoustic analysis is performed over a wide frequency range. It is broken down into 10 Octave bands labeled by the center frequency. The Bandwidth covers all sound from 0.707 of the center frequency to 1.414 times the center frequency. ***Table 2 - Octave Band Properties***, page 8, shows the 10 octave bands and their properties. In many cases, the 16 and 31 Hz bands are not used because sound data is not available. Several publications reference the octave bands by the numbers 1 through 8, starting with the 63 Hz band. This can lead to misunderstanding and is not recommended. Always refer to an octave band by its center frequency (e.g. 63 Hz).

Table 2 - Octave Band Properties

Center Frequency	16	31.5	63	125	250	500	1000	2000	4000	8000
Max Freq. (Hz)	23	44	88	88	175	350	700	1400	2800	5600
Min. Freq. (Hz)	11	22	44	175	350	700	1400	2800	5600	11200
Wavelength (ft)	70	36	18	8.96	4.48	2.24	1.12	0.56	0.28	0.14

Acoustic analysis and evaluations are based on the sum of sound pressure or sound power measured over the all of the frequencies of the octave band. In some cases, a finer analysis is required and it is common to hear reference to 1/3 octave band analysis. With this approach the bands are broken down into thirds so there are 30 values to consider in the analysis.

Tonal Sounds

A pure tone is the sound pressure or power associated with a single frequency. For example, there is often a 60 Hz sound component when based on 60 Hz electric frequency. Identifying a pure tone can help in acoustic analysis. For example, if there was a 60 Hz pure tone and only octave band analysis is considered, then the 60 Hz pure tone energy would be included in the 63 Hz octave band. It would not be evident that the sound energy in this band was due the electric frequency. Equipment such as fans, compressors and pumps can produce tonal sounds.

> ☺ *Tip; Care should be taken when evaluating tonal products using the processes described in this manual and the Acoustic Analyzer.*

Caution should be taken when evaluating tonal products because they are not modeled well by octave band analysis. For instance, a tonal product such as a double helix screw compressor cannot be evaluated using the processes described in this manual or by the Acoustic Analyzer. Refer to the ASHRAE article, *Addressing Noise Problems in Screw Chillers,* (See ***Appendix 1 - References***, page 79) for additional information on evaluating tonal issues.

Human Response to Sound

The goal of acoustic analysis and design is to provide a satisfactory environment for occupants. It is necessary to consider occupant response to sound and noise as part of the analysis in the same way as occupant response to temperature and humidity are considered.

The following lists key issues about human response to sound that must be considered.

- Humans are more sensitive to sound around the 1000 Hz octave band. These are the frequencies of voice communication. For example, a 60 dB sound pressure in the 1000 Hz band will *seem* loader to an occupant than a 60 dB sound pressure in the 63 Hz band.
- It takes a very large change in sound levels to be perceived as a change to humans. Reducing the sound pressure by 50% does not *seem* 50% quieter. ***Table 3 – Perceived Sound Level Changes*** shows the perceived change relative to the actual change in sound levels.
- Humans acclimate to constant sounds and they are sensitive to changes in sound levels. For example, constantly operating the fan on a fan coil will be less of an issue than cycling the fan on and off.
- Pure tones can stand out and be uncomfortable for occupants. A broad spectrum, bland sound is just as important as the overall sound level.

Table 3 – Perceived Sound Level Changes

Sound Change Level (dB)	Acoustic Energy Loss	Relative Loudness
0	0	Reference
-3	50%	Perceptible Change
-6	75%	Noticeable Change
-10	90%	Half as loud
-20	99%	¼ as loud
-30	99.9%	1/8 as loud
-40	99.99%	1/16 as loud

- ❑ Occupants can become fixated on a sound level once it is considered irritable. This may require an even greater improvement in sound level to satisfy the occupants than if the sound had been attenuated in the first place.

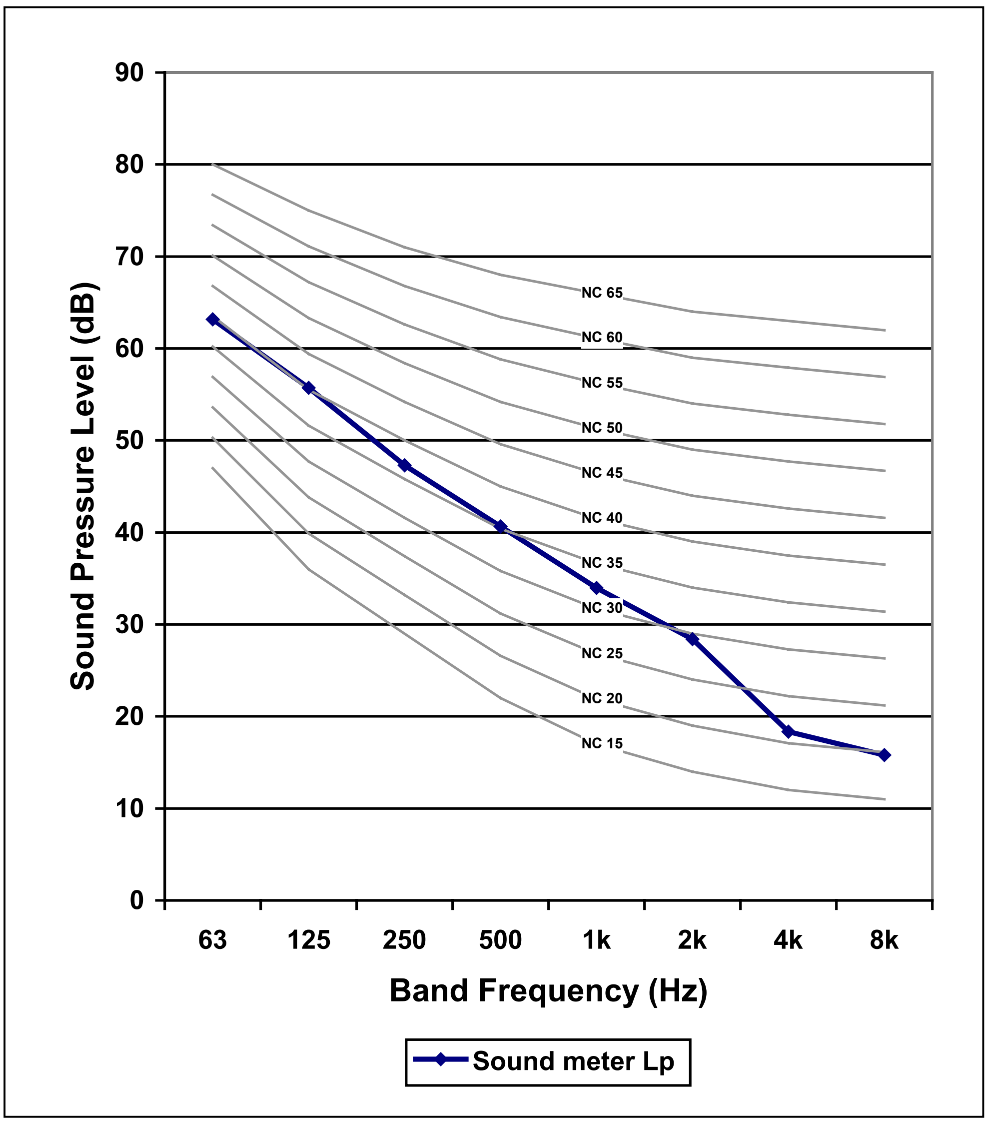
Sound Pressure Evaluation Criteria

There several recognized sound pressure criteria that are used to evaluate sound pressure levels and occupant comfort. The criteria attempt to take into account the human response to frequency, overall spectrum and level. Several of the most common in the HVAC industry are discussed here. Other criteria include Balanced Noise Criteria Method (NCB), RC mark II, Composite Noise Rating (CNR), etc. For more information on these criteria, refer to the ASHRAE Applications Handbook.

Noise Criteria (NC)

Noise Criteria or NC curves are the most common standard for indoor spaces. They were developed to take into account human response to sound pressure levels in different octave bands. Since humans are less sensitive to sound pressure in the lower bands, a higher sound pressure level was deemed acceptable. The shape of the curves can be seen in ***Figure 2 - NC Curves***, page 9.

Figure 2 - NC Curves



NC curves are based on the 63 through 8000 Hz octave band values. When the octave band values of the space sound pressure are known, they are plotted on the NC curve*. **Table 4 - Sound Pressure Levels For Each NC Level***** shows the sound pressure levels for each NC level in tabular form.

Table 4 - Sound Pressure Levels For Each NC Level

NC Level	Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
15	47	36	29	22	17	14	12	11
20	51	40	33	26	22	19	17	16
25	54	44	37	31	27	24	22	21
30	57	48	41	35	31	29	28	27
35	60	52	45	40	36	34	33	32
40	64	56	50	45	41	39	38	37
45	67	60	54	49	46	44	43	42
50	71	64	58	54	51	49	48	47
55	74	67	62	58	56	54	53	52
60	77	71	67	63	61	59	58	57
65	80	75	71	68	66	64	63	62

The NC value is not an "average" that is based on a curve drawn through the data points. Instead, it is the highest NC value in any one octave band. This is called the tangent method. An issue with the tangent method is that it is possible to have three distinct sound spectrums where the highest single NC level in any band could be the same value for all three spectrums. In this case, all three would have the same NC value, but they would sound quite different.

Another issue with the NC method is that it does not evaluate low frequency sounds below 63 Hz, which can be the most troublesome and the most difficult to attenuate.

NC Calculation Example

Using the rules described above and the following sound pressure levels, calculate the NC value

The sound pressure levels in dB RE 20μPa:

Band	63	125	250	500	1000	2000	4000	8000
Lp	63	56	47	41	34	28	18	16

Using the Tangent Rule, the highest NC level in any one band occurs in the 125 Hz band. The NC level in this band is NC 40.

Note: This curve is plotted in ***Figure 2 – NC Curves,*** page 9.

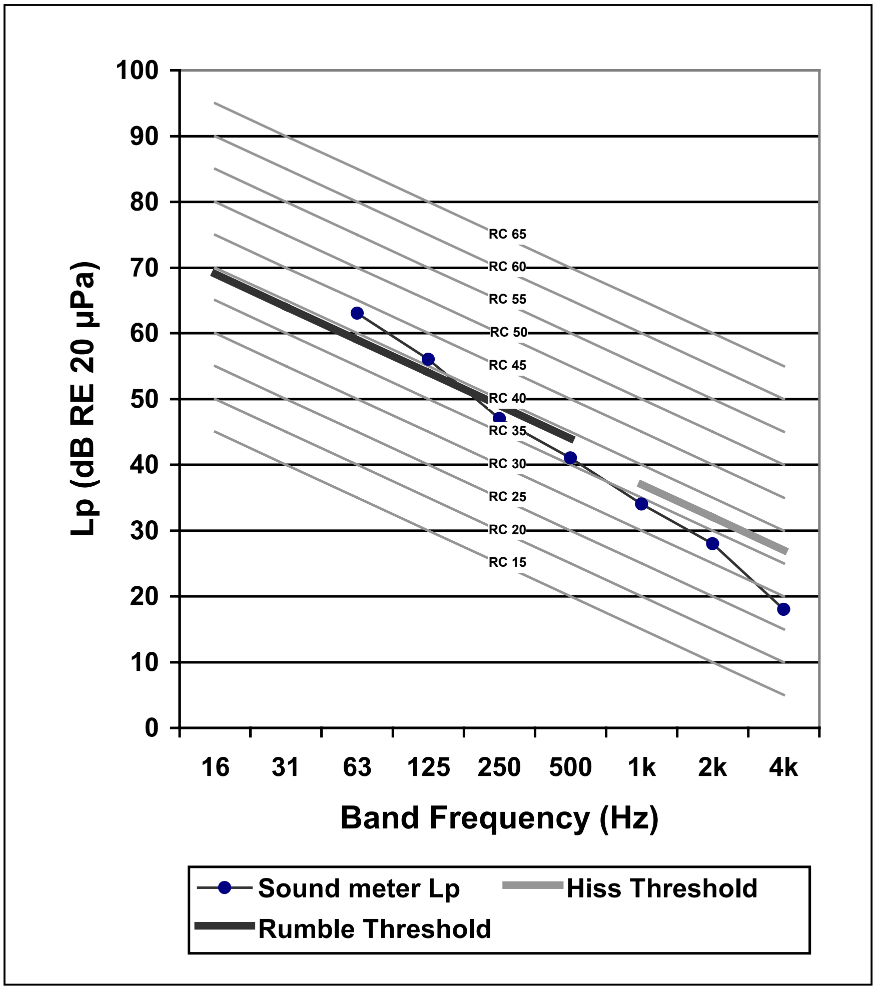
Room Criterion (RC)

Room criterion or RC curves are currently the most favored method for determining sound levels by ASHRAE. The RC system was developed to overcome the shortcomings of the NC system. They account for spectrum shape as well as sound level. The RC system also uses the 16.5 and 31 Hz bands, which allows the criteria to account for acoustically produced vibration in light building construction. However, the RC calculations should only be performed when valid data exists. If the 16.5 and 31 HZ data are not available, it should not be extrapolated from available values for RC calculations.

RC curves are based on 16 Hz through 4000 Hz octave bands. The RC value is the average of the 500, 1000 and 2000 HZ bands. The slope of each line is –5 dB per octave band as shown in ***Figure 3 - RC Curve***, page 11.

In addition to the numerical value, RC criteria also includes one or more letters to denote specifics about the spectral shape of sound. The letter R denotes Rumble. This occurs whenever any specific RC value is more than 5 dB greater than the standard curve values in the 500 Hz and below octave bands. The letter H indicates Hiss. This occurs whenever any specific RC value is more than 3 dB greater than the standard curve values above the 500 Hz octave band. If the specific dB levels are between the deadbands, then the letter N is used to denote neutral. This sound will not have any identity with frequency and will sound bland.

Figure 3 - RC Curve



RC Calculation Example

Using the rules described above and the sound power data from the NC example, calculate the RC value.

The RC value is the average of the 500, 1000 and 2000 Hz values.

RC = (41+34+28)/3 = 34

The slope of each RC line is –5dB per octave band.

Refer to the curve plotted above in ***Figure 3 – RC Curve***. The sound meter Lp line is plotted on the RC chart and shows the sound measurement points.

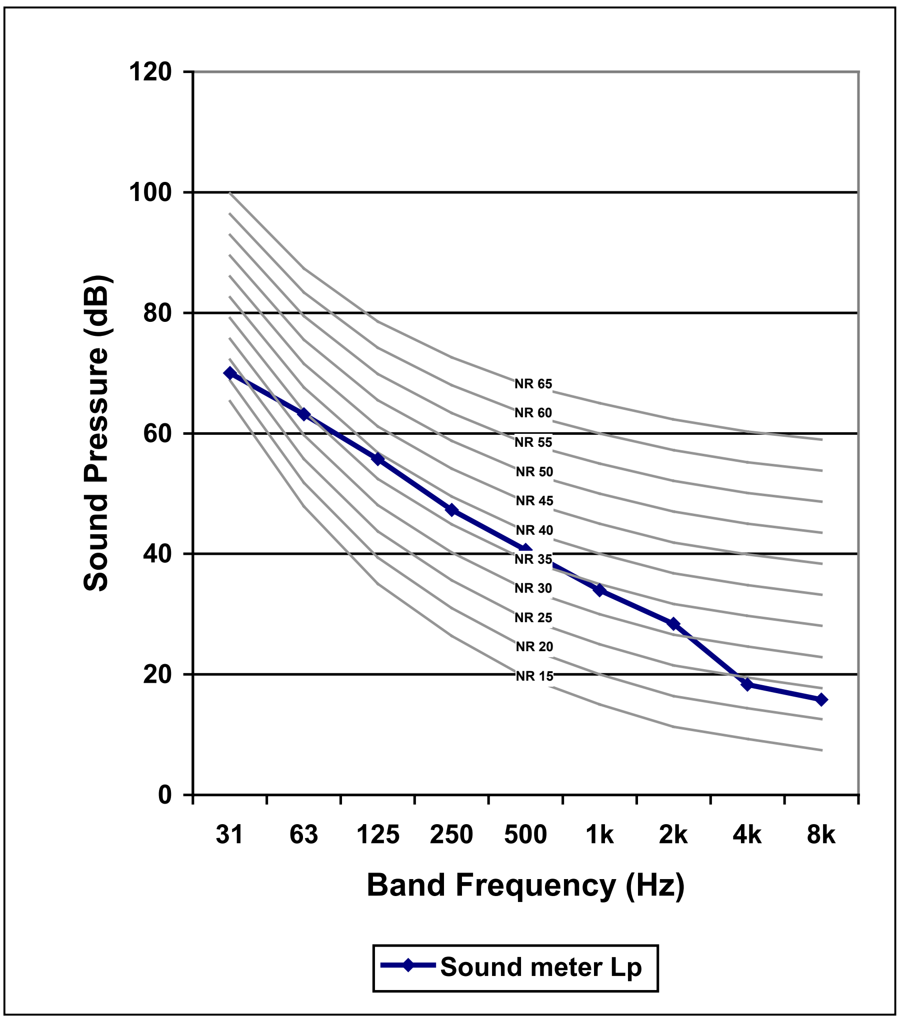
> The rumble threshold line is plotted on the chart from 500 Hz and down, 5 dB above RC 34. If any point at or below 500 Hz exceeds this line, then an R for rumble is added. In this case there is rumble.
>
> The hiss threshold line is plotted on the chart above 500 Hz, 3 dB above RC 34. If any point above 500 Hz is above this line, then an H for hiss is added. In this case there is no hiss.

Noise Rating (NR) Curves

Table 5 - Sound Pressure Level For Each NR Level

	Octave Band								
	31	63	125	250	500	1000	2000	4000	8000
NR 0	55	36	22	12	5	0	-4	-6	-8
NR 10	62	43	31	21	15	10	7	4	2
NR 20	69	51	39	31	24	20	17	14	13
NR 30	76	59	48	40	34	30	27	25	23
NR 40	83	67	57	49	44	40	37	35	33
NR 50	89	75	66	59	54	50	47	45	44
NR 60	96	83	74	68	63	60	57	55	54
NR 70	103	91	83	77	73	70	68	66	64
NR 80	110	99	92	86	83	80	78	76	74
NR 90	117	107	100	96	93	90	88	86	85
NR 100	124	115	109	105	102	100	98	96	95
NR 110	130	122	118	114	112	110	108	107	105
NR 120	137	130	126	124	122	120	118	117	116
NR 130	144	138	135	133	131	130	128	127	126

Figure 4 - NR Curves



Noise Rating curves were developed by International Organization for Standardization (ISO) and are similar to NC curves. They are based on sound pressure and the different curves represent different rooms and uses. NR levels are based on the tangent method so the highest NR value in any octave band is the NR level. NR curves are based on 31 through 8000 Hz octave bands.

A-Weighted Sound Pressure (dBA)

A-weighted sound pressure is "corrected" to more closely resemble the hearing characteristics of the human ear. The human ear approximates the A-weighted curve in the 20 to 30 dB range. At these low sound levels, the ear has relatively poor sensitivity to low frequency sound. ***Figure 5*** shows the adjustments for A-weighted sound with the very large adjustments in the lower frequency bands. There is also Band C-weighted sound data, which is meant for louder sound levels. These sound levels are not as common as A-weighted values.

A-weighted sound criteria is most commonly used in outdoor sound evaluations. It is often used in city building codes when referencing the maximum acceptable sound pressure levels at the property line. It is popular because it is a single number that most sound meters include.

Figure 5 – A-Weighted Octave Band Adjustments

Band	63	125	250	500	1000	2000	4000	8000
Adjustment	-26	-16	-9	-3	0	1	1	-1

Table 6 - Comparison of Sound Rating Methods[1]

Method	Overview	Considers Speech Interference	Evaluates Sound Quality	Components Currently Rated by Method
NC	Can rate components No quality assessment Does not evaluate low frequency rumble	Yes	No	Air Terminals Diffusers
RC	Used to evaluate systems Should not be used to evaluate components Can be used to evaluate sound quality Provides some diagnostic capability	Yes	Yes	
NR	Can rate components No quality assessment Does not evaluate low frequency rumble	Yes	No	
dBA	Can be determined using sound level meter No quality assessment Frequently used for outdoor noise ordinances	Yes	No	Cooling Towers Water Chillers Condensing Units

Acceptable Sound Levels

Acceptable sound levels depend on the use of the space. For example, the level in an office environment is established based on speech requirements. ASHRAE recommends that important space sound levels be specified by RC(N), while less stringent spaces can use NC levels.

Table 7 - Design Guidelines For Sound lists recommended RC(N) and NC levels for various spaces.

Prolonged exposure to high sound levels can cause hearing damage. OSHA (Occupational Safety and Health Administration) provides guidelines for the maximum allowable exposure. (Refer to ***Table 8 - Occupational Noise Exposure Limits***)

Table 8 - Occupational Noise Exposure Limits

Average Daily Sound Level (dBA)	Maximum Average Daily Exposure
90	88 hours
95	4 hours
100	2 hours
105	1 hour
110	30 minutes
115	15 minutes

Community noise levels are usually found in city ordinances. ***Table 9 - Typical Municipal Code Noise Limits*** shows typical acceptable sound levels. ARI Guideline L, *Assessing the Impact of Air-Conditioning Outdoor Sound Levels in the Residential Community*, discusses various methods of defining outdoor sound ordinances and a method for evaluating the sound level of residential HVAC equipment. It is recommended that the local codes should be reviewed.

Table 7 - Design Guidelines For Sound Levels in Unoccupied Spaces[2]

Space	RC(N)	NC
Private residences, apartments, condominiums	25-35	25-35
Hotels/Motels		
Individual rooms or suites	25-35	25-35
Meeting/banquet rooms	25-35	25-35
Halls, corridors, lobbies	35-45	35-45
Service, support areas	35-45	35-45
Office Buildings		
Executive and private offices	25-35	25-35
Conference rooms	25-35	25-35
Teleconference rooms	25 max	25 max
Open plan offices	30-40	30-40
Circulation and public lobbies	40-45	40-45
Hospitals and Clinics		
Private rooms	25-35	25-35
Wards	30-40	30-40
Operating Rooms	25-35	25-35
Corridors	30-40	30-40
Public areas	30-40	30-40
Performing Arts		
Drama theaters	25 max	25 max
Concert and recital halls	A	A
Music teaching studios	25 max	25 max
Music practice rooms	25 max	25 max
Laboratories		
Testing/Research, minimal speech communication	45-55	45-55
Research, extensive phone use, speech communication	40-50	40-50
Group teaching	35-45	35-45
Churches, Mosques, Synagogues	25-35	25-35
With critical music programs	A	A
Schools		
Classrooms up to 750 ft²	40 max	40 max
Classrooms over 750 ft²	35 max	35 max
Lecture rooms for than 50	35 max	35 max
Libraries	30-40	30-40
Courtrooms		
Unamplified speech	25-35	25-35
Amplified speech	30-40	30-40
Indoor Stadiums and gymnasiums		
School and College gymnasiums and natatoriums	40-50	40-50
Large seating capacity spaces	45-55	45-55

A An experienced acoustical consultant should be retained for guidance on critical space (below RC 30) and for all performing arts centers.

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Table 9 - Typical Municipal Code Noise Limits

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

Sound Testing Methods

It is very important to understand the sound levels that are measured when performing sound analysis. There many published Standards by associations such as AMCA, ASHRAE and ARI that provide a definable format for conducting sound measurements. A list of Standards is provided in ***Appendix 2 – HVAC Equipment Sound Measurement Stds***, page 80. Some methods produce Sound Power ratings, while others produce Sound Pressure ratings.

Testing Facilities

There are two primary types of acoustic testing facilities that will fit most HVAC equipment and can be used as a test chamber. In an Anechoic chamber, the surfaces are lined with sound absorbing material that does not reflect sound waves when they contact the surfaces. This is intended to emulate a free field sound test where the sound waves will travel away from the source.

Figure 6 - McQuay's Anechoic Test Facility

A reverberant chamber has surfaces designed to reflect the sound waves back into the chamber and provide a diffuse field (uniform sound level through the space).

Most large HVAC equipment cannot fit or be operated in a test chamber (e.g. chillers and cooling towers). In these cases, the test Standard applicable to the product usually describes a method for testing equipment in the manufacturer's facility. However, the lower level of test parameter control makes this data less accurate.

Measuring sound intensity is now possible due to advancements in testing equipment. Sound intensity is the amount of sound power per unit area. Sound intensity levels offer the advantage that they can be measured in the presence of background sound (within certain limitations) and yield sound directivity.

Applying Test Data

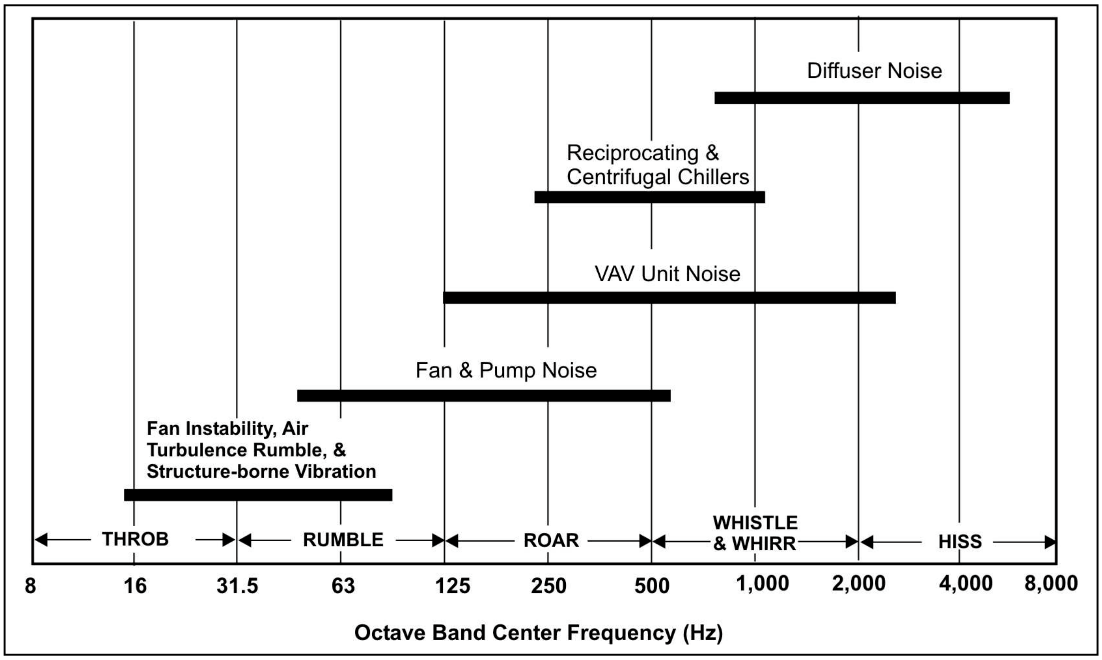
In principle, sound power ratings gathered for a given piece of equipment from either test method should be the same. However, this is not always the case. Correction factors are required to adjust sound power data gathered in accordance with a particular Standard when performing acoustic analysis. An example is ARI Standard 880, *Air Terminals*, which specifies sound power data to be measured in a reverberant method, but the equipment will act as if it is in a free field in applications. ASHRAE RP project 755, *Sound Transmission through Ceilings from Air Terminal Devices in the Plenum,* studied this effect and recommended an environmental correction factor. Acoustic Analyzer applies this correction factor where appropriate.

For more information on applying manufacturer test data, refer to ASHRAE's *Application of Manufacturers' Sound Data*, which can be purchased from ASHRAE.

HVAC Equipment Acoustics

HVAC systems and equipment generate sound at all frequencies and power levels. ***Figure 7 - Frequency Ranges*** shows the frequency ranges where various types of HVAC equipment may create noise issues. Knowing the frequency range that is not acceptable can help identify the source.

Figure 7 - Frequency Ranges Where Various HVAC Equipment Affect Sound Levels[3]



Calculating Sound Pressure from Sound Power

Sound from a Point Source

As soon as sound energy is released from a source, it interacts with the environment and creates sound pressure in a *source – path – receiver* arrangement. The most fundamental example is a point sound source emanating energy uniformly in all directions. Sound waves will travel uniformly in a spherical manner from a point sound source. The sound pressure level will decrease as a function of distance as follows;

Eq. 6

$$Lp = Lw + 10\mathrm{Log}(Q/4\pi d^2)+k$$

Where

d is the distance in feet (m) from the source to the measurement point

Q is the directivity factor, which for spherical radiation is 1

k is a constant whose value is 10.5 for I-P and 0.5 for SI

Ideally, this works out to a 6 dB drop in sound levels for every doubling of distance.

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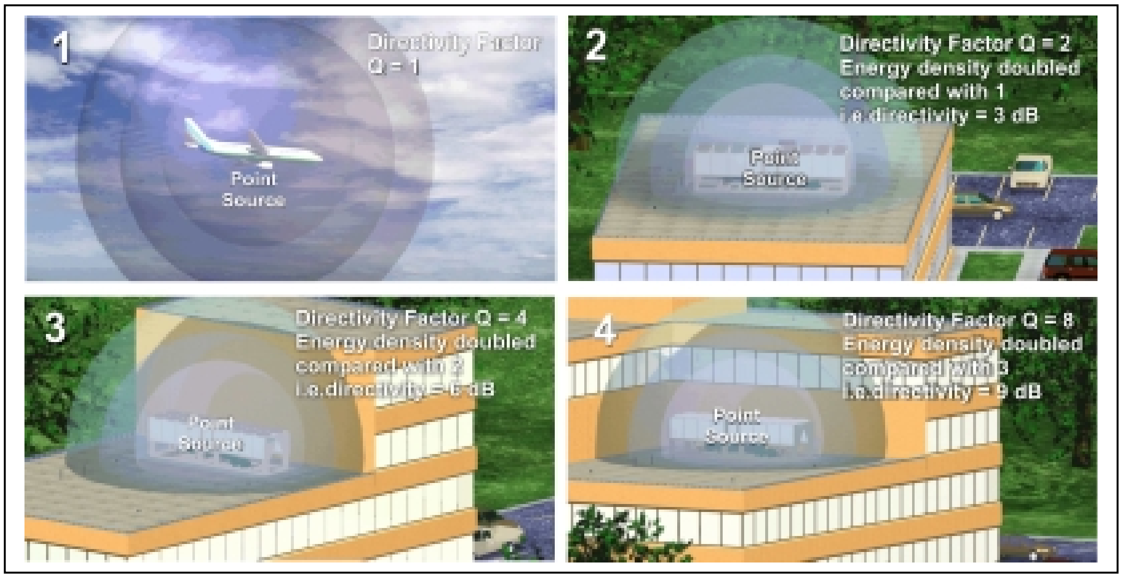
Directivity

Figure 8 - Sound Directivity Q



When a sound source is near a reflecting surface, the sound energy is concentrated and the sound waves are focused in a particular direction. This changes the *directivity* of the sound waves as shown in ***Figure 8 - Sound Directivity Q***, page 17. Sound directivity is referred to as Q and can vary from 1 to 8.

An example of directivity is a point sound source at ground level in an open field. Here the ground is a reflecting surface, which concentrates the sound energy in a hemispherical pattern. The value of Q in this case is 2.

When the sound source is near a reflecting surface such as the ground, the sound equation then becomes;

Eq. 7

$$Lp = Lw + 10Log(1/2\pi d^2)+k$$

Where

d is the distance in feet (m) from the source to the measurement point

k is a constant whose value is 10.5 for I-P and 0.5 for SI

These equations are used extensively for outdoor sound analysis and are covered in detail in ***Outdoor Sound Analysis,*** page 22.

Figure 9, Graph of Sound Pressure vs. Distance in a Free Field

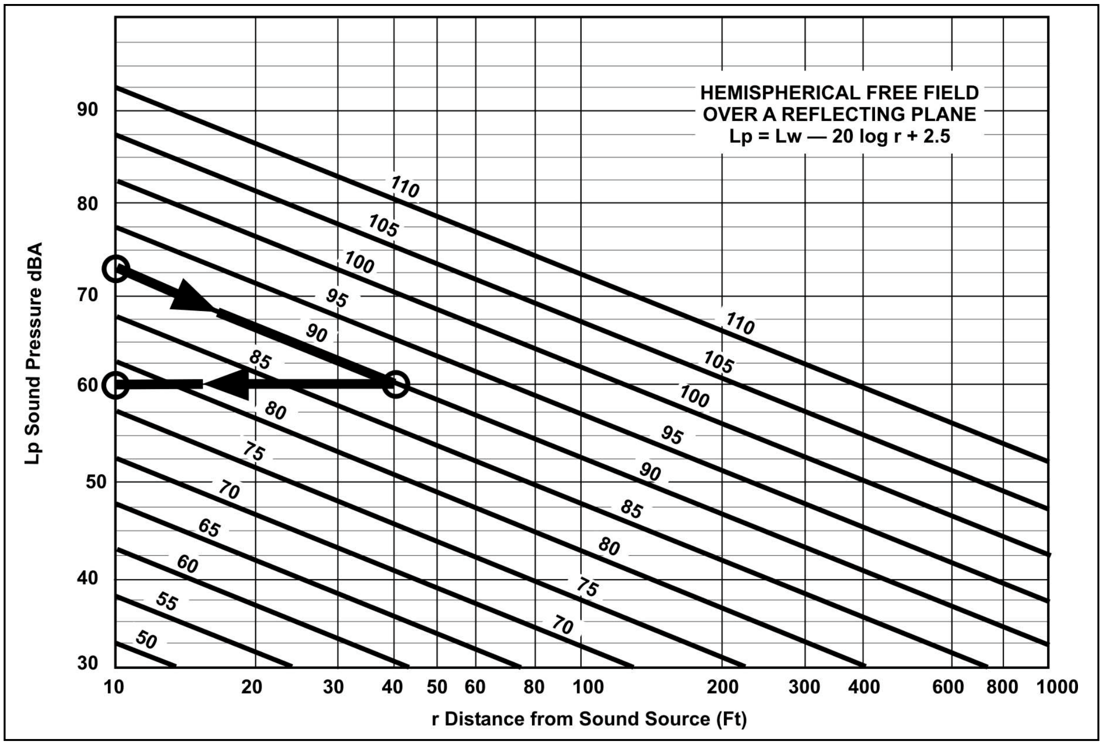


Figure 9, Graph of Sound Pressure vs. Distance in a Free Field, page 18, is a graphical means to convert a sound power source into a sound pressure level at a prescribed distance. This is only applicable in an open field. The diagonal lines represent sound power levels. Using a sound power level of 100 dB at a distance of 100 ft., the sound pressure level is about 63 dB. You can also start with a sound pressure level and estimate the sound level at a different distance. For example, start with a sound pressure level of 75 dB at 10 ft. Follow a constant sound power line (90 db) until you reach 40 feet then read horizontally back to see the new sound pressure (60 dB) at the greater distance.

Sound from a Line Source

A line source is a collection of point sources that radiate sound in a cylindrical pattern. The equation that relates line sound power to sound pressure is;

Eq. 8

$$Lp = Lw + 10\text{Log}(Q/\pi dL)+k$$

Where

d is the distance from the source to the measurement point

L is the length of the sound source in feet (m)

K is a constant whose value is 10.5 for I-P and 0.5 for SI

Line sources in HVAC systems include duct breakouts, where the breakout sound energy radiates away from the ducting and potentially into an occupied space. This is covered in more detail in ***Duct Breakout Sound Path***, page 44. Traffic noise from a highway can also be considered a line source.

Sound from a Plane Source

A plane source is a surface that radiates sound into a space. In close proximity to the wall, the sound level does not change, making plane sound sources an issue to attenuate.

The equations that relate sound power from a plane source to sound pressure are;

When $d < b/\pi$

Eq. 9

$$Lp = Lw + 10\text{Log}(\pi/(4bc)) + k$$

When $b/\pi < d < c/\pi$

Eq. 10

$$Lp = Lw - 10\text{Log}(d) - 10\text{Log}(4c) + k$$

When $d > c/\pi$

Eq. 11

$$Lp = Lw - 20\text{Log}(d) - 11 + k$$

Where

d is the distance from the source to the measurement point

c is the larger dimension of the wall in feet (m)

b is the shorter dimension of the wall in feet (m)

k is a constant whose value is 10.5 for I-P and 0.5 for SI

An example is sound transmission through a wall separating a mechanical room from occupied space. Usually the height of a room is less than either the length or the width. When this occurs and either the ceiling or the floor is the sound source, then the sound level will be uniform through the space and it will be a challenge to attenuate. Plane sources are associated with transmission loss through walls and will be dealt with in ***Transmitted Sound,*** page 64.

Comparing Point, Line and Plane Sources

Figure 10 - Attenuation With Distance From Different Source Types

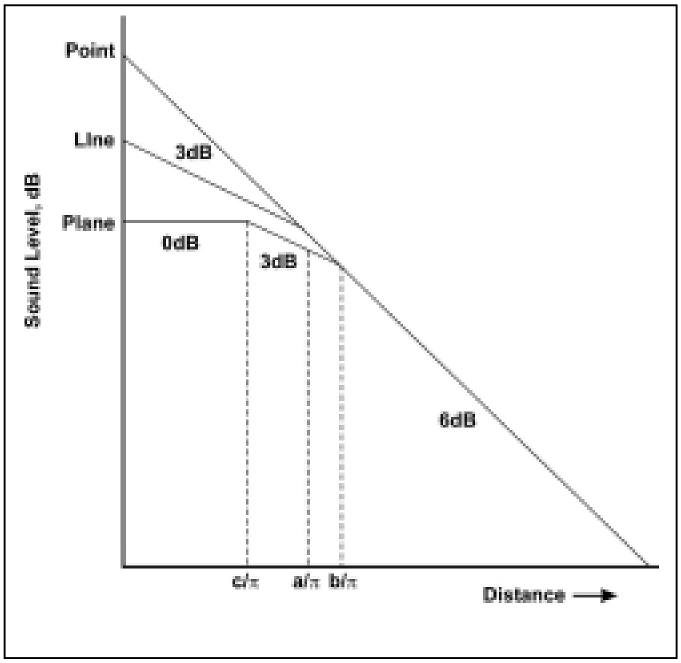


Figure 10 - Attenuation With Distance From Different Source Types shows the difference in sound attenuation as you move away from point, line and plane sources. At a great enough distance, they all behave like a point source and sound drops off at 6 dB per doubling of distance. As you move closer to a line source (around Length/π) the sound level only drops off at a rate of 3 dB per doubling of distance.

Plane sources also behave differently as you move closer. The sound level is constant within a distance equal to the shorter dimension of the plane. Beyond the shorter dimension and out to the longer dimension divided by π, the sound level drops off at a rate of 3 dB per doubling of distance.

Identifying the type of source is not always straightforward. A chiller on the roof can be considered a point source for property line evaluations if the distances are great enough. As you move closer, the condenser fan deck could be considered a line source while the compressors are considered point sources. Moving even closer, the condenser fan deck may behave as a plane sound source.

Sound Pressure in a Confined Space

When a point sound source emanates sound waves in a confined space, part of the sound energy reflects off the surfaces and back into the space. This contains the sound energy and makes the sound analysis more complex. Close to the sound source, the receiver is in the *near* sound field where no sound waves have been reflected. Further away, the receiver is in the *reverberant* sound field where there is a combination of direct and reflected sound waves. The basic formula that calculates sound pressure in a confined space is;

Eq. 12

$$Lp = Lw + 10Log(Q/(4\pi d^2)+4/R) + k$$

Where

d is the distance in feet (m) from the source to the measurement point

R is the Room Constant in Ft² (m²)

Q is the Directivity factor

k is a constant whose value is 10.5 for I-P and 0.5 for SI

Room constant is a factor that measures a room's ability to absorb sound. For example, if R equaled infinity, then the walls, floor, ceiling, etc. would absorb all sound and the sound energy would behave as if it were in a free field. ***Indoor Sound Analysis – Zoned Comfort Systems***, page 29, provides a more detailed description of Room constant. When $Q/(4\pi d^2)$ is dominant, then the receiver is in the near field as most of the sound energy is coming directly from the source. When 4/R is dominant, the receiver is in the reverberant field.

This equation is the basis for room acoustics and will be discussed in detail in later sections. Refer to ***Indoor Sound Analysis – Zoned Comfort Systems***, page 29, for a more detailed explanation.

Figure 11 - Sound Level vs. Distance in a Confined Space

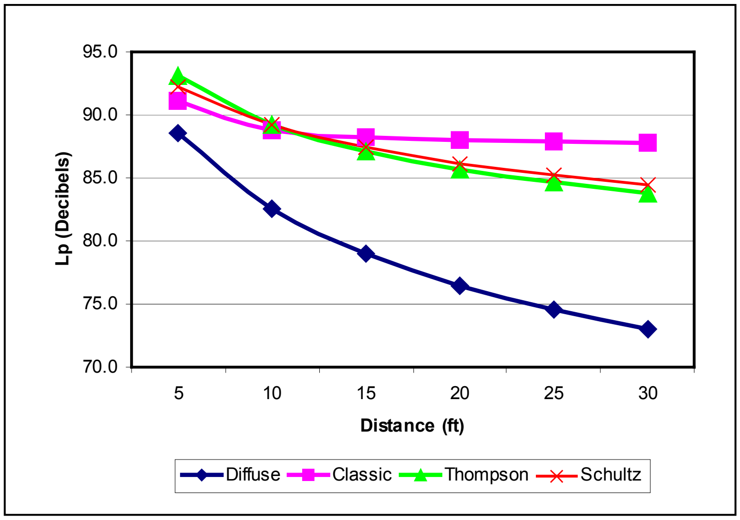


Figure 11 - Sound Level vs. Distance in a Confined Space, page 21, is based on a 30 ft by 30 ft by 9 ft (10 m by 10 m by 3 m) room with a single 100 dB sound power source against one wall. The classic equation shows the drop off in sound level as you move out of the direct field and into the reverberant field (around 10 ft (3 m)). Beyond 10 ft (3 m), there is little drop in sound level. The Thompson and Schultz equations (page 33) are considered more accurate in real world applications and show the sound level continuing to drop off as you move away from the source. The diffuse equation is included as a reference and shows how sound drops off in a free field. In this case, the sound level drops 6 dB for every doubling of distance.

Sound Transmission through a Wall

In the previous section it was shown that the sound waves will reflect off the walls, floor, etc. and back into the space. However, not all the sound energy is reflected. Some of the sound energy is absorbed and some is transmitted through the wall and into the next space. Any energy that is either absorbed or transmitted will lower the sound level in the space that has the sound source. However, the adjacent space will now have an additional sound source (the common wall) which will radiate sound energy and add to the sound level. The basic equation that considers sound transmission is;

Eq. 13

$$NR = TL - 10\text{Log}(S_w/R)$$

Where

NR is the noise reduction in dB

TL is the Transmission Loss of the wall in dB

S_w is the common wall area in ft² (m²)

R is the Room Constant of the receiving room in Ft² (m²)

This relationship is used to consider areas such as mechanical room, which can transmit sound into occupied spaces. This will be covered in more detail in future sections.

Outdoor Sound Analysis

General

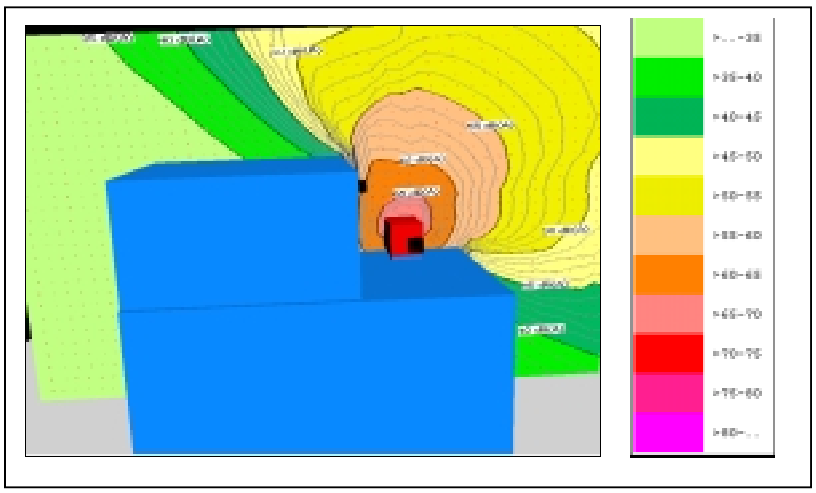
Outdoor sound analysis is commonly required to estimate the sound levels at the property line or an adjacent building. Outdoor equipment, such as cooling towers, air-cooled chillers and rooftop units, all create sound that can disturb neighbors. At a minimum, most local building codes have a maximum allowable sound requirement at the property line (refer to ***Table 9 - Typical Municipal Code Noise Limits***, page 15).

Outdoor Sound Analysis Basics

Directivity

Outdoor sound analysis is closest to a free field analysis where the directivity factor is 2. A point sound source (e.g. an air-cooled chiller) releases sound energy in a hemispherical pattern. The sound pressure level will depend on the distance from the source. In actuality, many sound sources have some level of directivity associated with them. For instance, an air-cooled chiller sends a significant amount of sound vertically from the condenser fans.

Figure 12 - Example Of Sound Directivity



In critical applications, it is necessary to account for directivity. ***Figure 12 - Example Of Sound Directivity*** shows the sound profile emanating from an air-cooled chiller on a roof with one reflecting wall. In this case, the sound energy has greater concentration from the top fan deck.

Accounting for this level of directivity is not easily done since the calculations become more involved and the sound profiles are not readily available. One method is to have a five point profile of the sound source. This profile indicates increases and decreases in the sound power level from the overall sound power levels for the four sides and vertical directions. With this information, it is possible to account for the orientation of the sound source. For example, a chiller with a control panel at one end may have a lower sound power level in that direction because the panel acts like a barrier. Orienting the unit so that the panel faces the property line may help reduce the sound pressure level at the property line.

Atmospheric and Anomalous Affects

Air absorbs sound energy, particularly in the higher frequencies. The amount of absorption is dependent on temperature and humidity. Acoustic Analyzer uses a molecular absorption coefficient (α_m) in dB per 1000 ft. (305 m) based on ambient temperature and humidity. A standard day is generally considered 59°F (15°C) and 70% relative humidity.

There are also small scale anomalous effects of refraction, sound interference, etc. that lower the sound level – particularly at large distances (over 500 feet (150 m)). The Acoustic Analyzer accounts for the anomalous effects using the molecular absorption coefficient (α_a) in dB per 1000 feet (305 m). When these two factors are added to Eq. 7;

Eq. 14

$$Lp = Lw + 10Log(1/2\pi d^2) - d(\alpha_m + \alpha_a)/1000 + k$$

Where

d is the distance from the source to the measurement point in feet (m)

α_m is the molecular absorption coefficient

α_a is the anomalous affects coefficient

k is a constant whose value is 10.5 for I-P and 0.5 for SI

Trees and Shrubs

Trees and shrubs tend to break up and scatter sound waves. This can have both a positive and a negative affect. On the positive side, they can act as a barrier and reduce sound levels on the opposite side of the sound source. However, trees and shrubs can also disperse sound waves into *shadow* areas. For example, consider a sound barrier on a highway that runs through a residential neighborhood. Trees extending above the barrier can disperse sound down behind the barrier and toward the residences.

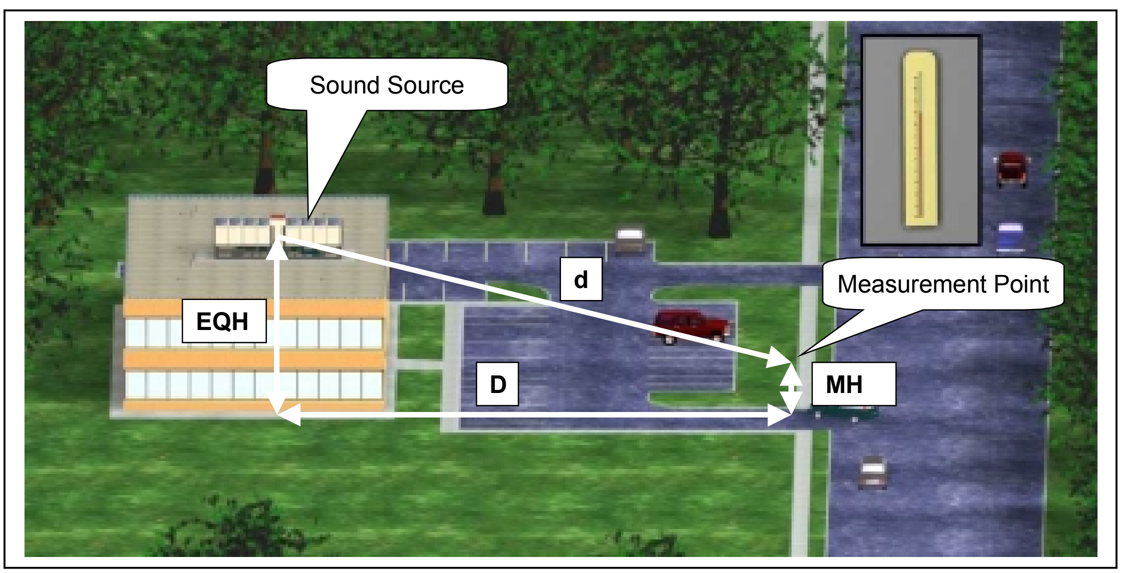
Wind, Temperature and Precipitation

Wind, temperature and precipitation can all bend sound waves and influence sound levels at large distances. Their effects are short term and generally not included in acoustic evaluations. However, they can explain differences in field testing.

Open Field Analysis

Open field or standard analysis is the most basic evaluation. An example would be rooftop equipment with no barriers or reflecting walls.

Figure 13 - Standard Open Field Analysis



Calculating the Path Length

The first requirement is to calculate the distance from the source to the receiver. Referring to ***Figure 13 - Standard Open Field Analysis***, the distance (d) from the source to the receiver is;

Eq. 15

$$d = (D^2+(EQH-MH)^2)^{1/2}$$

Where

d is the distance from the source to the receiver in feet (m)

D is the horizontal distance from the source to the receiver in feet (m)

EQH is the height to the source in feet (m)

MH is the height to the measurement point in feet (m)

Often D is the distance to the closest property line. It could also be the distance to an adjacent building on the property. The two heights are usually measured from the base of the building.

Free Field Example

Consider an air-cooled screw chiller on the roof of a 4-story building. The chiller is 100 feet horizontally from the property line. The receiver measurement point is 6 feet above grade. Given the sound power data, what is the property line dBA?

The sound power levels for the chiller are:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	103	103	102	99	99	97	90	84

First, calculate the distance from the source to the receiver using equation 15.

$d = (100^2 + ((4 \times 12) - 6)^2)^{1/2} = 108.5$ ft

Look up α_m for a Standard Day and α_a

α_m	0.1	0.2	0.4	0.7	1.5	3.0	7.6	13.7
α_a	0.4	0.6	0.8	1.1	1.5	2.2	3.0	4.0

Using equation 14, calculate the sound pressure level for each band. Here is the 63 Band for an example.

$Lp_{63} = 103 + 10\text{Log}(1/(2\pi 108.5^2) - 108.5(0.1 + 0.4)/1000 + 10 = 64$ dB

Lp	64	63	60	60	58	51	44	39

Add the correction factors for dBA and logarithmically add the bands together

dBA	-26	-16	-9	-3	0	1	1	-1
	38	47	51	57	58	52	45	38

The dBA level is 62

Sound Barriers

☺Tip; An ideal barrier has a transmission loss at least 10 dB in all frequencies greater than the insertion loss expected of it. Note the best possible insertion loss is about 24 dB.

Sound barriers can be installed to reduce the sound levels and hide equipment from view. The barrier creates an "acoustic shadow" that reduces sound levels on the opposite side of the source. The reduction in sound level, or *Insertion Loss,* is based on the path length difference. The path length difference equals the path around the barrier minus direct path from the source to the receiver in feet (or meters). Refer to ***Figure 14 - Sound Barrier***.

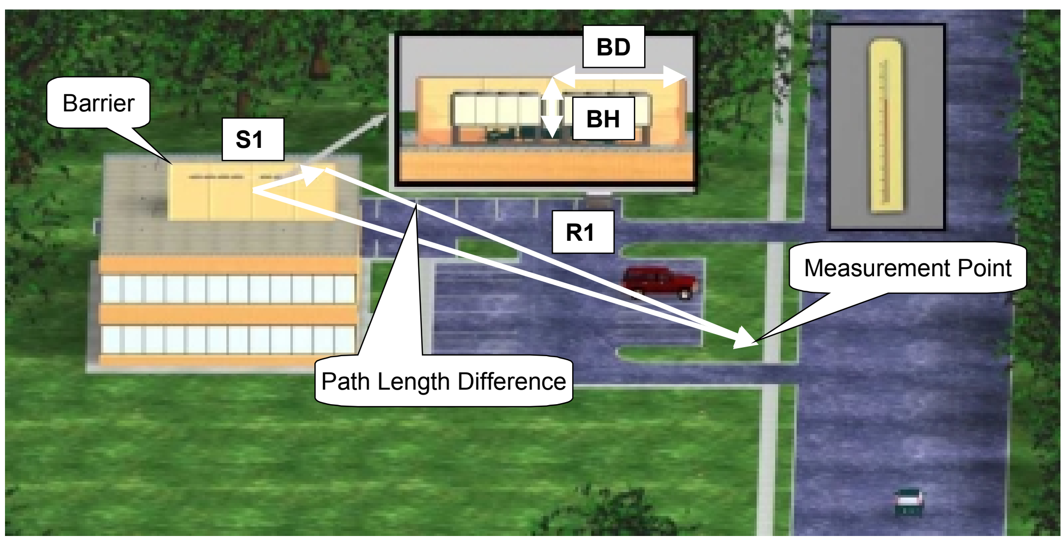
Table 10 - Insertion Loss For Ideal Barrier

Path Length Difference (ft)	Insertion Loss Per Octave Band (Hz)								
	31	63	125	250	500	1000	2000	4000	8000
0	0	0	0	0	0	0	0	0	0
0.01	5	5	5	5	5	6	7	8	9
0.02	5	5	5	5	5	6	8	9	10
0.05	5	5	5	5	6	7	9	10	12
0.1	5	5	5	6	7	9	11	13	16
0.2	5	5	6	8	9	11	13	16	19
0.5	6	7	9	10	12	15	18	20	22
1	7	8	10	12	14	17	20	22	23
2	8	10	12	14	17	20	22	23	24
5	10	12	14	17	20	22	23	24	24
10	12	15	17	20	22	23	24	24	24
20	15	18	20	22	23	24	24	24	24
50	18	20	23	24	24	24	24	24	24

Table 10 - Insertion Loss For Ideal Barrier shows the insertion loss for an ideal barrier. An ideal barrier has a transmission loss at least 10 dB in all frequencies greater than the insertion loss expected of it. Note the best possible insertion loss is about 24 dB. This is due to scattering and refraction of sound into the shadow area.

The best sound barriers surround the equipment on all four sides. Where the barrier is open, it should extend horizontally beyond the ends of the equipment to at least three times the path length difference over the top of the barrier. Reflecting walls can reduce the effectiveness of a barrier by reflecting sound into the shadow area.

Figure 14 - Sound Barrier



Calculating the Path-Length Difference

Referring to ***Figure 14 - Sound Barrier***, the path-length difference δ can be calculated as follows;

Eq. 16

$$d = (D^2+(EQH-MH)^2)^{1/2}$$

Eq. 17

$$S1 = (BD^2 + BH^2)^{1/2}$$

Eq. 18

$$R1 = ((D-BD)^2 + (EQH+BH)^2)^{1/2}$$

Eq. 19

$$\delta = S1+R1-d$$

Where

d is the distance from the source to the receiver in feet (m)

D is the horizontal distance from the source to the receiver in feet (m)

EQH is the height to the source in feet (m)

MH is the height to the measurement point in feet (m)

BD is the horizontal distance from the source to the barrier in feet (m)

BH is the vertical distance from the source to the top of the barrier in feet (m)

S1 is the distance from the source to the top of the barrier in feet (m)

R1 is the distance from the top of the barrier to the receiver in feet (m)

δ is the path length difference (m)

Barrier Example

Consider the same building used before. Now a barrier has been added around the chiller. It is 10 feet from the chiller and 2 feet taller than the chiller. What is the new dBA level? What TL is required in the barrier?

The sound pressure levels without the barrier are:

Band	63	125	250	500	1000	2000	4000	8000
Lp	64	63	60	60	58	51	44	39

First, calculate the distances d, S1, R1 and the path length difference.

$d = (100^2 + ((4 \times 12) - 6)^2)^{1/2} = 108.5$ ft

$S1 = (10^2 + 2^2)^{1/2} = 10.2$ ft

$R1 = ((100-10)^2 + (48+2)^2)^{1/2} = 103$ ft

$\delta = 10.2 + 103 - 108.5 = 4.7$ ft

Look up the insertion loss in Table 10 and subtract from the sound pressure level

δ	12	14	17	20	22	23	24	24
Lp Bar	52	49	43	40	36	28	20	15

Add the correction factors for dBA and logarithmically add the bands together

dBA	-26	-16	-9	-3	0	1	1	-1
	26	33	34	37	36	29	21	14

The dBA level is 39

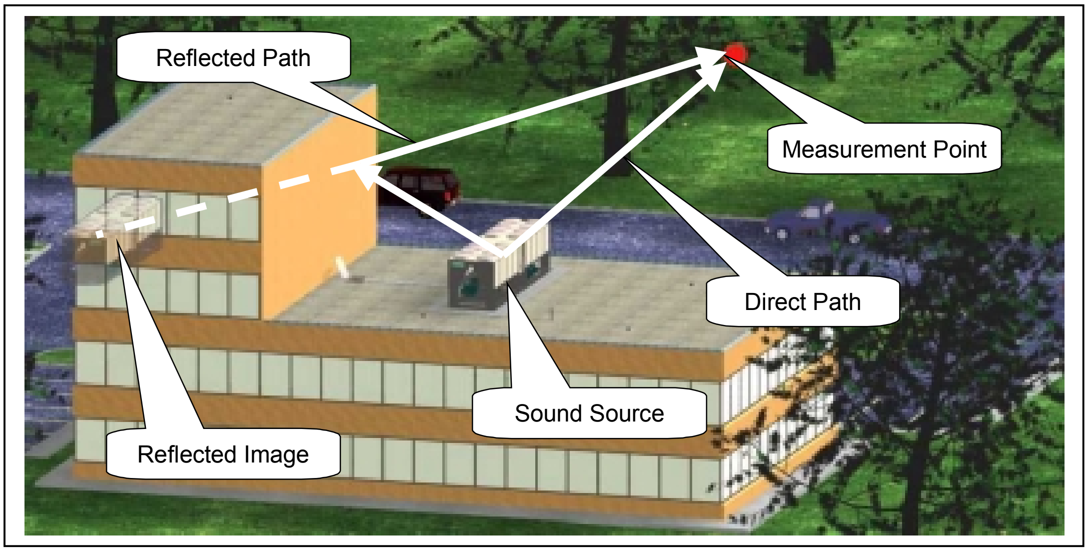
The barrier TL (Transmission Loss) must be at least 10 dB greater than the insertion loss in each band. Therefore the TL must be at least:

TL	22	24	27	30	32	33	34	34

Reflecting Walls

When a large vertical surface is located near a sound source, it can reflect and concentrate sound energy into the open field portion. In addition, reflecting walls can diminish the affect of barriers by reflecting sound into the shadow zone of a barrier.

Figure 15 - Reflected Sound Waves



Sound waves reflect off walls in a similar manner as light waves reflecting in a mirror (refer to ***Figure 15 - Reflected Sound Waves***). The reflected sound waves must travel from the sound source to the wall, where they are reflected before they travel to the receiver. The longer path is used to calculate the attenuation.

The additional sound level due to reflected sound is calculated as follows;

Eq. 20

$$\Delta Lp = 3.00 - 9.29Log(SR1/d) + 10.13(Log(SR1/d))^2 - 3.84(Log(SR1/d))^3$$

Where

d is the distance from the source to the receiver in feet

SR1 is the reflected distance in feet.

Additional reflecting walls can be included by calculating a ΔLp for each wall and adding the results to the sound pressure levels at the receiver.

One Reflected Surface Example

Consider the same building used before. In addition to the barrier, there is a reflecting wall 15 feet away from the chiller. What is the new dBA level?

The sound pressure levels with the barrier are:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lp Bar	52	49	43	40	36	28	20	15

First, calculate the distance from the source to the receiver.

$d = (100^2 + ((4 \times 12) - 6)^2)^{1/2} = 108.5$ ft

Calculate the reflected distance, SR1

SR1 = 2·15 + 108.5 = 138.5 ft

Calculate ΔLp

$\Delta Lp = 3.00 - 9.29 Log(138.5/108.5) + 10.13(Log(138.5/108.5))^2 - 3.84(Log(138.5/108.5))^3$
$= 2.1$ dB

ΔLp	2.1	.21	2.1	2.1	2.1	2.1	2.1	2.1
Lp B,W	54.1	51.1	45.1	42.1	38.1	30.1	22.1	17.1

Add the correction factors for dBA and logarithmically add the bands together

dBA	-26	-16	-9	-3	0	1	1	-1
	28.1	35.1	36.1	39.1	38.1	31.1	23.1	16.1

The dBA level is 41

Indoor Sound Analysis – Zoned Comfort Systems

General

Indoor acoustic analysis introduces surfaces that contain much of the sound energy in the space. Zoned comfort (decentralized) HVAC systems place the sound source in or near the occupied space. Three examples of decentralized systems will be considered. The first example will be a unit such as a PTAC or fan coil directly in the space. The second example will be a ducted unit away from the space, such as a WSHP in a corridor. Finally, a ducted unit above the space will be considered.

These analyses will introduce room effect, duct breakout, return air path and discharge vs. radiated sound pathways.

Sound in a Room

As discussed earlier, sound in a room will reflect off the walls, creating a near field where sound is dominated by direct sound from the source and a reverberant field where sound is dominated by the reflected sound energy. The Classic equation looks like;

Eq. 12

$$Lp = Lw + 10\mathrm{Log}(Q/(4\pi d^2)+4/R) + k$$

Where

d is the distance from the source to the measurement point

R is the Room Constant in ft² (m²)

Q is the Directivity factor

k is a constant whose value is 10.5 for I-P and 0.5 for SI

Room Constant and Sound Absorption Coefficient

Eq. 12 introduces the Room Constant measured in ft² (m²). It is a factor that describes the room's ability to absorb sound and is defined as;

Eq. 21

$$R = S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 ... + A_1 + A_2 + ...4mV$$

Where

S_1, S_2, S_3 etc are the areas of surfaces 1, 2, 3 in ft² (m²)

α_1, α_2, α_3 etc are the *sound absorption coefficients* of surfaces 1, 2, 3 etc.

+ A_1 +A_2 ... etc are lumped groups of known absorbers

m is the air absorption coefficient

V is the volume of the room in ft³ (m³)

Sound Absorption Coefficient

The sound absorption coefficient is a property of the surface material. A surface with an absorption coefficient of 1.0 would absorb all sound energy that it contacts. A coefficient of 0 would reflect all sound incident that it contacts. Sound Absorption coefficients vary with different bands, so the calculations must be performed for each band being considered. ***Table 11 - Typical Sound Absorption Coefficients***, page 30, lists several common materials and their typical sound absorption coefficients.

The term *4mV* represents the amount of sound energy the air in the space absorbs. It is most important in the 4000 and 8000 Hz bands. In some literature, the term *Sabin* is used to describe a room constant of 1 square foot (A metric Sabin is based on 1 m^2) with a sound absorption coefficient of 1.0.

Number of Occupants

Occupants in a space can have a large affect on the sound level. For example, nearly 75% of a concert hall's total absorption comes from the audience. [4] Furniture also improves sound absorption by defracting sound waves or absorbing them. Often, the goal of the designer is to obtain an acceptable background sound level in the space, which would mean there were no occupants. However, it is important that the designer knows that occupants improve sound absorption. In critical applications such as performing arts centers, accounting for the audience is required.

Table 11 - Typical Sound Absorption Coefficients

Material	Octave Bands						
	63	125	250	500	1000	2000	4000
Brick, unglazed	0.02	0.03	0.03	0.03	0.04	0.05	0.07
Brick, unglazed, painted	0	0.01	0.01	0.02	0.02	0.02	0.03
Carpet on concrete	0.01	0.02	0.06	0.14	0.37	0.6	0.65
Carpet on foam rubber	0.06	0.08	0.27	0.39	0.34	0.48	0.63
Concrete Block, light, porous	0.25	0.36	0.44	0.31	0.29	0.39	0.25
Concrete Block, dense, painted	0.07	0.1	0.05	0.06	0.07	0.09	0.08
8" Acoustic Block	0.47	0.67	0.64	0.51	0.75	0.77	0.69
12" Acoustic Block	0.67	0.95	0.89	0.55	0.74	0.81	0.72
Concrete or Terrazzo Floor	0	0.01	0.01	0.015	0.02	0.02	0.02
Linoleum, asphalt, rubber or cork tile on concrete	0.01	0.02	0.03	0.03	0.03	0.03	0.02
Wood Floor	0.1	0.15	0.11	0.1	0.07	0.06	0.07
Glass	0.25	0.35	0.25	0.18	0.12	0.07	0.04
Curtain Wall	0.126	0.18	0.06	0.04	0.03	0.02	0.02
Closed Curtains	0.05	0.07	0.31	0.49	0.75	0.7	0.6
Drywall on Stud Wall	0.2	0.29	0.1	0.05	0.04	0.07	0.09
Marble or Glazed Tile	0	0.01	0.01	0.01	0.01	0.02	0.02
Drywall on Brick wall	0.01	0.013	0.015	0.02	0.03	0.04	0.05
Wood Paneling	0.2	0.28	0.22	0.17	0.09	0.1	0.11
2" 3 pcf Fiberglass Insulation	0.15	0.22	0.82	1	1	1	1
3" 3 pcf Fiberglass Insulation	0.48	0.53	1	1	1	1	1
4" 3 pcf Fiberglass Insulation	0.76	0.84	1	1	1	1	0.97
Suspended Ceiling, 3/4 to 1" Acoustic Tile	0.4	0.58	0.59	0.69	0.86	0.84	0.75
Suspended Ceiling, 2 to 3" Acoustic Tile	0.49	0.73	0.71	0.76	0.89	0.75	0.58

Calculating Room Constants

One method to calculate the Room Constant is to use equation 17 and add all of the surface areas and their sound absorption coefficients. This level of detail is most easily done with software like the McQuay Acoustic Analyzer™. A simplified method is to use the average sound absorption coefficients in ***Table 12 - Average Sound Absorption Coefficients for Typical Receiving Rooms***, page 31, in the following equation;

Eq. 22

$$R = S \cdot \alpha_T / (1-\alpha_T)$$

And

[4] Mehta, M., Jim Johnson, Jorge Rocafort. 1999. *Architectural Acoustics, Principles and Design.* Prentice Hall, Upper Saddle River, New Jersey.

Eq. 23

$$\alpha_T = \alpha + 4mV/S$$

Where

S is the total surface area of the room in ft² (m²)

α is the average room absorption coefficient from Table 12

Table 12 - Average Sound Absorption Coefficients for Typical Receiving Rooms[5]

Type of Room	63	125	250	500	1000	2000	4000
Dead	0.26	0.30	.035	0.40	0.43	0.46	0.52
Medium Dead	0.24	0.22	0.18	0.25	0.30	0.36	0.42
Average	0.25	0.23	0.17	0.20	0.24	0.29	0.34
Medium Live	0.25	0.23	0.15	0.15	0.17	0.20	0.23
Live	0.26	0.24	0.12	0.10	0.09	0.11	0.13
Air Absorption Coefficient							
M (1/ft)	0	0	0		0	0.0009	0.0029

Most rooms where an HVAC designer would be interested in (office space, classrooms, etc.) would be rated as medium dead.

Major Factors in Room Constants

Figure 16 - Typical Sound Absorption Coefficients

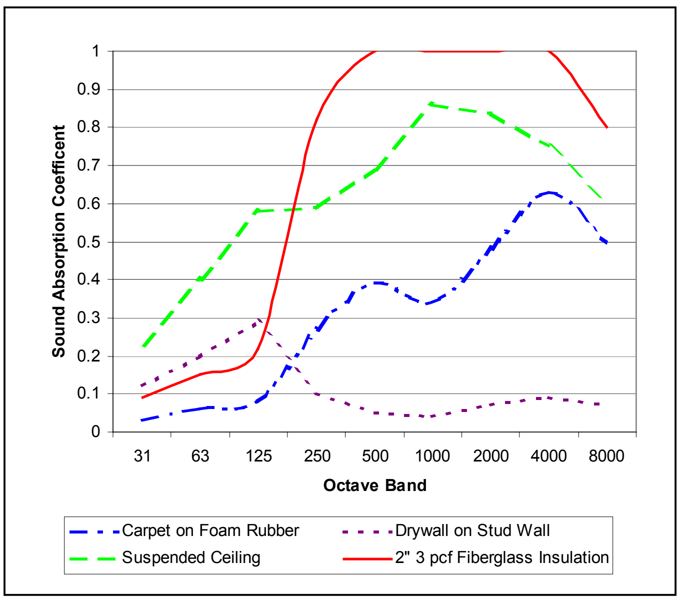


Figure 16 - Typical Sound Absorption Coefficients, page 31, shows absorption coefficients of typical wall materials. From this figure, it is clear that suspended ceiling tiles play an important role.

Curtains and carpeting can also help absorb sound. In hi-rise residential applications with poured concrete ceilings, the carpets and curtains are typically the best absorbers in the space.

Hard surfaces such as glass and concrete can reflect sound back into the space, making it very "live". An example is a gymnasium or natatorium.

[5] Reynolds, D. Jeffrey M. Bledsoe. 1991. *Algorithms for HVAC Acoustics.* American Society of Heating, Refrigerating, and Air-conditioning Engineers Inc. Atlanta, Ga.

Room Constant Example

Consider a room 20 ft long by 12 ft wide by 8 ft high. The space is typical, so it can be considered medium dead. Calculate the Room Constant.

Calculate the total room surface area

S = 2(20•12) + 2(20•8) + 2(8•12) = 992 ft²

Calculate the room Volume

V = (20•12•8) = 1920 ft³

Look up the average sound absorption coefficients for a medium dead room and air absorption coefficient.

Band	63	125	250	500	1000	2000	4000	8000
α	0.24	0.22	0.18	0.25	0.30	0.36	0.42	
m	0	0	0	0	0	0.0009	0.0029	

Calculate the average absorption coefficient for this space. Use 63 Hz band as an example.

α_T =0.24+ 4•0•1920/992 = 0.24

Calculate the Room Constant. Use 63 Hz Band as an example.

R = 992•0.24/(1-0.24) = 313 ft²

α_T	0.24	0.22	0.18	0.25	0.30	0.37	0.44	
R	313	280	218	331	425	583	779	

Note how the room constant increases in the higher bands. The space will absorb more sound energy in the higher frequencies than the lower frequencies. As the occupant moves away from the near field around the source and toward the reverberant field, the high frequency sounds will be attenuated leaving only the low frequency sounds.

Thompson and Schultz Equations

Equation 12 shows the fundamental relationship between sound power and sound pressure in an enclosed space. Based on empirical data, two equations are commonly used to provide more accurate results. These are the Thompson equation and the Schultz equation. Here is the Thompson equation;

Eq. 24

$$Lp = Lw + 10\mathrm{Log}(Q \cdot e^{-md}/(4\pi d^2) + (MFP/d) \cdot (4/R)) + 10\ \mathrm{Log}\ (N) + k$$

Where

Q is the directivity factor, which is usually 2

m is the air absorption coefficient

d is the distance from the source to the receiver in feet (m)

MFP is the mean free path in feet (m)

R is the Room constant in ft² (m²)

N is the number of point sound sources.

k is a constant whose value is 10.5 for I-P and 0.5 for SI

The mean free path is defined as;

Eq. 25

$$MFP = 4 \cdot V/S$$

Where

V is the room volume in ft^3 (m^3)

S is the total surface area of the room in ft^2 (m^2)

Here is the Schultz equation;

Eq. 26

$$Lp = Lw - 10Log(d) - 5Log(V) - 3Log(f) + 10\,Log\,(N) + k$$

Where

d is the distance from the source to the receiver in feet (m)

V is the room volume in ft^3 (m^3)

f is the center Band frequency in Hz

N is the number of point sound sources

k is a constant whose value is 25 for I-P and 12 for SI

The Schultz equation can be used for a single point sound source such as a diffuser or return air opening. The equation will work for about three point sources (N=3). For an array (four or more) of distributed ceiling diffusers, the multiple ceiling array equation can be used for a receiver point 5 feet above the floor;

Eq. 27

$$Lp = Lw_s - 27.6Log(h) - 5Log(X) - 3Log(f) + 1.3\,Log\,(N) + k$$

Where

Lw_s is the sound power level associated with a single diffuser in dB RE 10^{-12} watts

h is the ceiling height in feet (m)

X is the ratio of the floor area served by each diffuser divided by square of the ceiling height

k is a constant whose value is 30 for I-P and 15.8 for SI

When To Use Each Equation

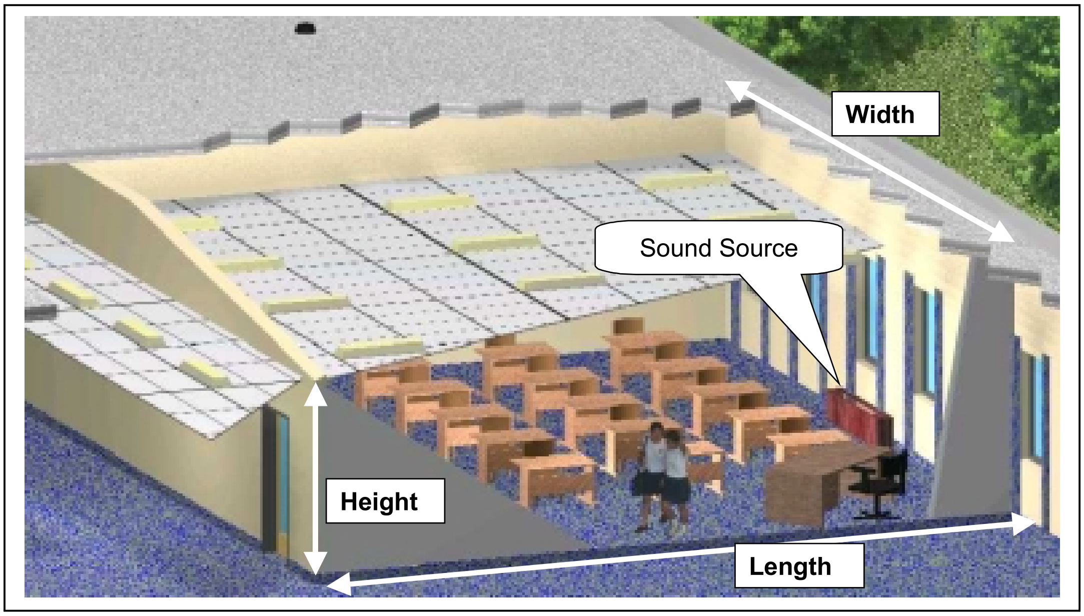
So far, this manual has listed three point source equations to estimate the sound pressure level in a confined space (Classic, Thompson and Schultz). While the classic equation is good for showing the relationship between near field and reverberant field, it is not used for actual sound calculations.

Both the Thompson and Schultz equations produce acceptable results when properly applied. The Thompson equation is based on the Classic equation (Eq. 12) with modifications based on empirical data. It also requires a Room Constant, which means the space must be understood. The Thompson equation works well with live spaces such as gymnasiums and churches.

The Schultz equation is completely empirical. It is referenced in the *ASHRAE Applications Handbook* and in *ARI Standard 885, Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets*. The Schultz equation is easier to use and works well in many typical applications (medium dead spaces).

In addition to point source equations, there are also line source equations that are used for duct breakout calculations. There are also plane source equations, which are used for wall transmission calculations. When to apply each equation is dependant on the room type (live vs. Dead) and the type of sound source (breakout duct vs. Point source). Some equations are easier to use manually. The McQuay Acoustic Analyzer™ provides recommendations on which equation to use, but it also allows the operator to choose.

Figure 17 - Decentralized Unit in the Space



Decentralized Unit In Space Example
Using the room defined in the Room Constant example, a console style fan coil with the following sound power level is being considered. Using the Thompson equation, calculate the sound pressure levels, NC and RC at 5, 10 15 and 20 feet from the fancoil unit.

The sound power levels for the Fancoil unit is:

Band	63	125	250	500	1000	2000	4000	8000
Lw	34	49	55	47	43	38	33	25

Calculate MFP
MFP = 4•1920/992 = 7.7 ft.
Q = 2 since the sound is focused by one reflecting surface
N = 1 since there is only one unit

R	313	280	218	331	425	583	779	

For d = 5 ft at 63 Hz

$Lp_{63} = 34 + 10\text{Log}(2 \cdot e^{-(0 \cdot 5)} / (4\pi 5^2) + (7.7/5) \cdot (4/313)) + 10 \text{ Log } (1) + 10.5$ (20)

Lp_{63} = 29 dB RE 20 μPa

Sound pressure levels for various distances:

Lp_5	29	44	51	41	37	31	25	
Lp_{10}	25	41	47	38	33	27	21	
Lp_{15}	23	39	46	36	31	25	19	
Lp_{20}	22	37	44	35	30	23	17	

The RC and NC levels for the various distances:

Distance	NC	RC
5	41	36(N)
10	37	33(N)
15	36	31(N)
20	34	29(N)

At all distances, the NC level was set by the sound pressure i n the 250 Hz band.

In an open field, we would expect a 6 dB drop in sound pressure levels for every doubling of distance. When the distance was 5 ft, the sound level dropped by 4 dB for the first doubling of distance because the receiver was in the near field where direct sound is dominant. Between 10 and 20 ft, the sound pressure only dropped 3 dB because the receiver was in the reverberant field.

The only options to improve the sound level would be to choose a quieter piece of equipment or locate it as far away from the receiver as possible. If the receiver is close to the sound source, changing the room properties (i.e. changing the Room Constant) will not help much since the receiver is in the near field.

Here are the sound pressure levels using the Schultz equation. They are almost the same.

Lp_5	30	44	49	40	36	30	24	
Lp_{10}	27	41	46	37	33	27	21	
Lp_{15}	25	40	45	36	31	25	19	
Lp_{20}	24	38	43	34	30	24	18	

Indoor Sound Analysis – Ducted Zoned Comfort Systems

Ducted zoned comfort systems are common in HVAC design. Ceiling WSHPs, fan coils and unit ventilators are good examples. Acoustically evaluating a ducted system introduces the concept of multiple pathways. The ducting is both an attenuator and a sound source. The sound energy from the equipment is divided into discharge and radiated sound. Ceiling plenums are commonly used with these systems, and they offer attenuation that must also be considered.

Radiated and Discharge Sound Power

Most ducted HVAC product sound data will include both radiated and discharge sound power levels. The radiated sound data is used to evaluate the sound energy that emanates from the product as if it was a single point sound source. If the unit is not above the space (a classroom HVAC unit in the corridor, for example) then radiated sound may not be important. When the unit is above the occupied space, then radiated sound will have to be considered. This is covered in ***Radiated Sound Path***, page 52.

Discharge sound power data is the sound energy that is focused into the ductwork. This sound energy will be attenuated by the ductwork with the remaining sound energy being dispersed into the occupied space at the diffusers. Discharge sound power can also break out of the ductwork, creating a path to the occupied space.

Return Air vs. Supply Air Sound Power

When considering the discharge sound energy, the designer will want to know the sound energy focused in the discharge air side (supply) and the return air side (return). Discharge and return sound power levels may be available from the manufacturer. When only the total discharge sound power levels are known, conventional wisdom is to divide the sound power in half (lower the values by 3 decibels) and assume half the energy goes one way and half the other. But this may not be true.

Multiple Path Concept

Figure 18 - Multiple Path Concept

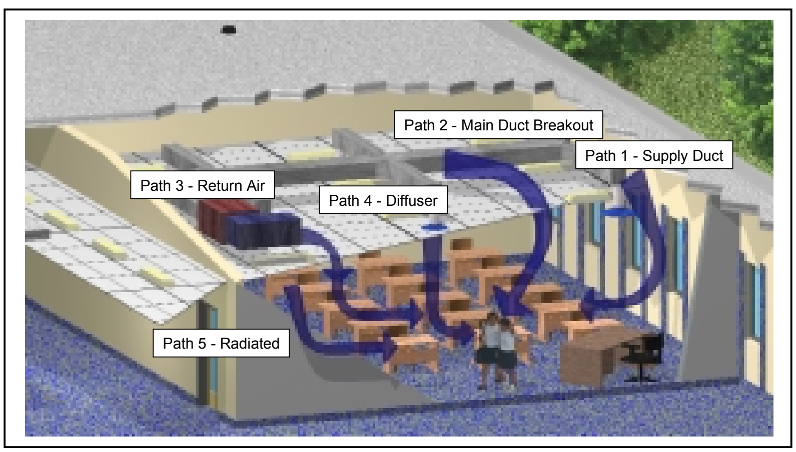


Figure 18 - Multiple Path Concept shows the many sound paths that can exist with a ducted system. To estimate the sound level in the space, each of these paths should be checked. The sound levels from all the paths are logarithmically added together to obtain the final space sound pressure level.

Ceiling Plenums

When a ceiling plenum is used, it can have a greater effect on the sound levels in the space. First, when calculating the Room Constant, acoustic tile is typically the most important surface in absorbing sound energy, particularly in the higher frequency bands. The second advantage of a ceiling plenum is its ability to “trap” sound released in the ceiling plenum. Several sound paths require the sound energy to pass through the ceiling plenum such as return air, breakout and radiated sound paths.

ASHRAE has done a considerable amount of testing on ceiling plenums and has developed the following process for estimating the plenum attenuation.

Table 13 - Ceiling Plenum Attenuation With T Bar Suspension[6]

Description	Band							
	63	125	250	500	1000	2000	4000	8000
No Suspended Ceiling	0	0	0	0	0	0	0	0
Mineral Fiber 1 lb. Density	3	6	8	10	16	21	36	21
Mineral Fiber 0.5 lb. Density	3	5	7	9	15	20	23	18
Glass Fiber 0.1 lb. Density, 5/8" Thick	3	6	5	7	7	8	9	7
Glass Fiber 0.6 lb. Density, 2" Thick	4	7	8	11	15	19	25	20
Glass Fiber with TL Backing 0.6 lb. Density, 2" Thick	4	7	8	12	27	22	29	23
Drywall Ceiling	8	11	15	15	17	17	18	14
Double Drywall Ceiling	14	17	21	21	23	23	24	19

The following qualifications have been made when using these values:

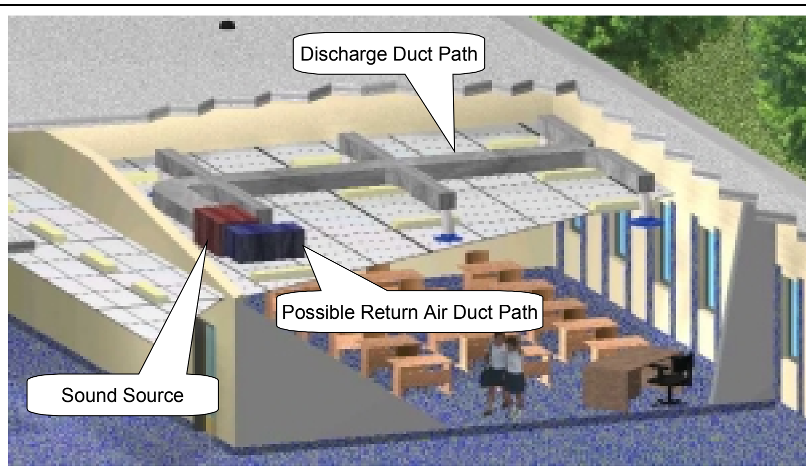
- The plenum is at least 3 ft (1 m) deep.
- The plenum space is either wide (over 30 ft (9 m)) or lined with insulation.
- The ceiling has no significant penetration directly under the unit.

For conditions other than these, the sound attenuation may be less. For instance, tests have shown that a 2-ft (0.6 m) deep plenum will be 5 to 7 dB louder below 500 Hz.[7]

Duct Sound Path

The discharge and often the return air sound energy must pass through a duct system prior to being released into the space. The duct can attenuate the sound, but it can also generate sound energy. Consider a splitter-damper that causes turbulence and makes noise as the air flows around it. This is referred to as ***regenerated*** noise (Refer to ***Regenerated Noise***, page76).

Figure 19 - Duct Path Attenuation



This room will be used for several of the upcoming examples. It is 40 ft by 50 ft by 8 ft (12 by 15 by 2.4 m). The supply air volume is 2000 cfm distributed through 6 lay in type diffusers.

Figure 19 - Duct Path Attenuation, page 37, shows a typical supply duct arrangement for a terminal unit (such as a WSHP). In this case, there is also a return air elbow. As the sound energy leaves the HVAC unit, each section of duct will generally attenuate or lower the sound level. The sound energy level at the duct

[6] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.
[7] ARI Standard 885, Procedure for Estimating Occupied Space Sound Levels in the application of Air Terminals and Air Outlets. 1998. Air-Conditioning and Refrigeration Institute, Arlington, Va.

opening close to the unit will be higher than the duct opening further from the unit because there are fewer attenuating components.

The process of estimating the attenuation requires considering each component, estimating their sound attenuation and ***arithmetically*** subtracting the attenuation from the sound power. If a component causes regenerated noise, then the noise source is ***logarithmically*** added to the sound energy level at that point in the duct. Therefore, it is important to calculate attenuation and regenerated noise in the correct order from supply to discharge.

Duct Component Attenuation

Table 14 – Typical Duct Attenuation Table[8]

	Octave Band							
Size	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
6 x 6	0.3	0.2	0.1	0.1	0.1	0.1	0.1	0.1
12 x 12	0.35	0.2	0.1	0.06	0.06	0.06	0.06	0.06
12 x 24	0.04	0.2	0.1	0.05	0.05	0.05	0.05	0.05
24 x 24	0.25	0.2	0.1	0.03	0.03	0.03	0.03	0.03
48 x 48	0.15	0.1	0.07	0.2	0.2	0.2	0.2	0.2
72 x 72	0.1	0.1	0.05	0.2	0.2	0.2	0.2	0.2

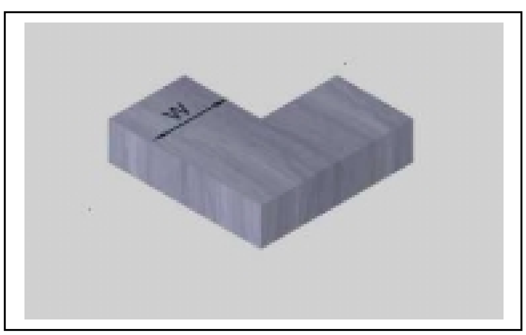
ASHRAE and other sources have developed tables and equations to estimate the attenuation of duct fittings. These can be found in the ASHRAE Applications Handbook. The McQuay Acoustic Analyzer uses algorithms based on this data. The following is an overview of several attenuating components found in ducted systems. Additional components are also described in ***Central System Duct Sound Path***, page 55.

Straight Ducting

Table 14, page 38, shows a typical table for a rectangular, unlined duct. The attenuation is based on length of duct in feet (meters). Round, oval and rectangular ducts behave quite differently, so it is important to use the correct data. Round duct has a much lower attenuation than a rectangular duct of the same size. However, round duct has much lower breakout. Duct lining on the interior surface also significantly improves duct performance, particularly in the higher frequencies. It is important to understand how much ducting is lined. Additional straight duct insertion loss tables are listed in ***Appendix 3 – Various Acoustic Properties of Materials***, page 81.

Elbows

Figure 20 – Typical Elbow Dimension

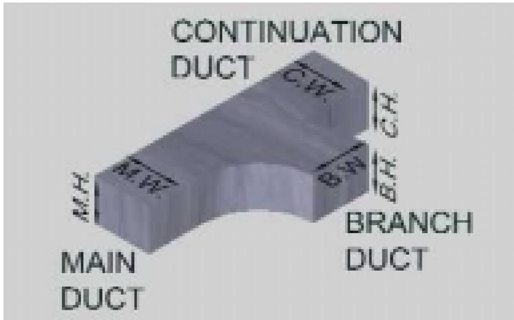


Elbows can be square (mitered) or round with a round or rectangular cross-sectional profile. They can be insulated or have turning vanes. From an acoustical perspective, a square elbow with no turning vanes is best choice if the regenerated noise is not too high. This elbow can reflect sound back up the duct. There are tables for elbows that are usually based on the elbow width (see ***Figure 20).*** Additional elbow insertion loss tables are in ***Appendix 3 – Various Acoustic Properties of Materials***, page 81.

[8] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.

Duct Branches

Figure 21 – Duct Branch



When sound energy reaches a branch in the duct, the sound energy is split. If the duct velocities all remain constant, then the ratio of the sound energy split is the same as the ratio of the duct areas.

If the duct areas on the discharge side have a different cross-sectional area than the supply duct, then the change in cross-sectional areas will cause sound energy to be reflected back up the duct. This will occur for frequencies with plane waves.

Plane waves occur at frequencies below;

Eq. 28

$$f_{co} = c_o/2a$$

Where

f_{co} is maximum frequency (Hz) that the reflection will occur

c_o is the speed of sound in air (1120 fps [341 m/s])

a is the larger cross sectional dimension (ft) (m) of a rectangular duct

The attenuation for a duct branch is given as follows;

Eq. 29

$$\Delta L_{Bi} = 10LOG*\left[\frac{\Sigma S_{Bi}/ S_M - 1}{\Sigma S_{Bi}/ S_M - 1}\right]^2 +10LOG*\left[\frac{S_{Bi}}{\Sigma S_{Bi}}\right]$$

Where

ΔL_{Bi} is the branch attenuation in dB

S_{Bi} is the cross sectional area (ft² [m²]) of branch i.

ΣS_{Bi} is the sum of the cross sectional areas (ft² [m²]) of the branches continuing on from the main duct.

S_M is the cross sectional area (ft² [m²]) of the main feeder duct

The first term in the equation is related to the plane wave reflection. The second term is related to the division of the sound power based on the ratio of air flows.

Flex Duct

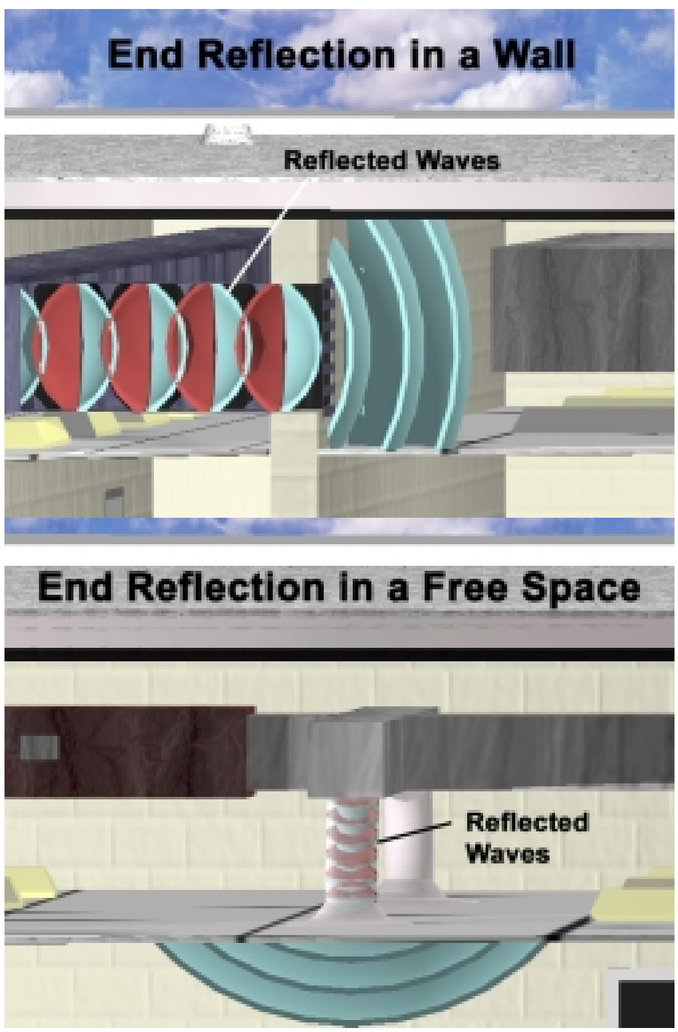
Table 15 - Flex Duct Attenuation[9]

	Octave Band							
Dia by Length	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
4 in by 3 ft	2	3	3	8	9	11	7	5
5 in by 3 ft	2	3	4	8	10	10	7	5
6 in by 3 ft	2	3	4	8	10	10	7	5
7 in by 3 ft	2	3	5	8	9	10	6	5
8 in by 3 ft	2	3	5	8	9	9	6	5
9 in by 3 ft	2	3	6	8	9	9	6	5
10 in by 3 ft	2	3	6	8	9	9	5	4
12 in by 3 ft	2	2	5	8	9	8	5	4
14 in by 3 ft	1	2	4	7	8	7	4	3
16 in by 3 ft	1	1	2	6	7	6	2	2

Flex duct is evaluated for duct attenuation in the same way as regular duct. However, its properties are different enough to require special tables and equations. Flex duct is a very good sound attenuator, but it also allows a significant amount of duct breakout (see ***Duct Breakout Sound Path***, page 44). For this reason, it is a good idea to limit flex duct to no more than 5 ft. (1.5 m).

End Reflection

Figure 22 - End Reflection



When low frequency sound waves travelling in confined spaces (ductwork) suddenly undergo a large change in cross sectional area, some of the waves are reflected back up the ductwork. This creates a significant amount of low frequency attenuation referred to as ***End Reflection***. The attenuation is greater when the duct terminates in free space versus ending flush with a wall. Since a ceiling tile array is basically transparent to low frequency sound, a duct terminating in the ceiling grid is considered to be terminating in free space. For example, a supply duct ending at a diffuser mounted in the ceiling grid is considered to end in free space. ***Table 16*** and

Table 17 show the end reflection for an equivalent size round duct. The Acoustic Analyzer uses an algorithm to estimate similar attenuation.

[9] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.

Table 16 - End Reflection Duct Terminated in Free Space[10]

Size	Octave Band							
Dia	63	125	250	500	1000	2000	4000	8000
6	20	14	9	5	2	1	0	0
8	18	12	7	3	1	0	0	0
10	16	11	6	2	1	0	0	0
12	14	9	5	2	1	0	0	0
16	12	7	3	1	0	0	0	0
20	10	6	2	1	0	0	0	0
24	9	5	2	1	0	0	0	0
28	8	4	1	0	0	0	0	0
32	8	3	1	0	0	0	0	0
36	6	3	1	0	0	0	0	0
48	5	2	1	0	0	0	0	0
72	3	1	0	0	0	0	0	0

Table 17 - End Reflection Duct Terminated in Wall[11]

Size	Octave Band							
Dia	63	125	250	500	1000	2000	4000	8000
6	18	13	8	4	1	0	0	0
8	16	11	6	2	1	0	0	0
10	14	9	5	2	1	0	0	0
12	13	8	4	1	0	0	0	0
16	10	6	2	1	0	0	0	0
20	9	5	2	1	0	0	0	0
24	8	4	1	0	0	0	0	0
28	7	3	1	0	0	0	0	0
32	6	2	1	0	0	0	0	0
36	5	2	1	0	0	0	0	0
48	4	1	0	0	0	0	0	0
72	2	1	0	0	0	0	0	0

Converting Ducted Sound Power to Sound Pressure

The processes described above will allow the designer to estimate the sound power at each duct opening in the space. This technique is used for both supply and return duct paths. Once the sound power has been estimated, the designer will have to convert the sound power to sound pressure by taking into account the room effect. For up to three point sound sources such as a supply diffuser or a return air opening, the Schultz equation can be used. For multiple diffusers (more than four) the multiple diffuser array equation can be used. Also, the Thompson equation can be used for either a single or a multiple point array.

[10] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.
[11] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.

Supply Duct Attenuation Example

Referring to the ducting system shown in ***Figure 19 - Duct Path Attenuation***, page 33, estimate the duct attenuation to the closest diffuser. The duct is lined with 1-in. insulation for 12 feet. The airflow is 2000 cfm with a duct velocity of 600 fpm.

The Discharge sound power levels for the WSHP unit is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	49	71	59	53	41	27	26	23

Since this data is based on ARI 260, it is appropriate to apply the Environmental Correction Factor:

Corr.	4	2	1	0	0	0	0	0
Corr. Lw	**45**	**69**	**58**	**53**	**41**	**27**	**26**	**23**

The first duct section is 22" by 22" by 6.2 ft long. It is lined. The attenuation is:

Duct	2.0	2.2	3.9	9.5	19.1	16.3	13.6	11.8
Lw	**43**	**66.8**	**54.1**	**43.5**	**21.9**	**10.7**	**12.4**	**11.2**

The 22" round elbow is also lined. The attenuation is:

Elbow	0	1	2	3	3	3	3	3
Lw	**43**	**65.8**	**52.1**	**40.5**	**18.9**	**7.7**	**9.4**	**8.2**

With six evenly sized diffusers and the duct velocity held constant, the branch split attenuation is:

Branch	7.8	7.8	7.8	7.8	7.8	7.8	7.8	7.8
Lw	**35.2**	**58**	**44.3**	**32.7**	**11.1**	**0**	**1.6**	**0.4**

The 9" by 9" by 10 ft Long branch duct attenuation is:

Duct	3.3	1.8	1.0	0.8	0.8	0.8	0.8	0.8
Lw	**31.9**	**56.2**	**43.3**	**31.9**	**10.3**	**0**	**0.8**	**0**

The 9 inch dia. by 2 ft flex duct attenuation is:

Flex	2	5	7	12	11	12	5	4
Lw	**29.9**	**51.6**	**36.3**	**19.9**	**0**	**0**	**0**	**0**

The end reflection is based on terminating in a free space. The attenuation is:

End Ref	15.7	10.4	5.7	2.4	0.8	0.2	0.1	0.0
Lw	**14.2**	**41.2**	**30.6**	**17.5**	**0**	**0**	**0**	**0**

The process estimated the sound power at the opening closest to the HVAC unit. Since the ductwork is symmetrical, the same sound level can be expected at the closest two diffusers. The calculations will have to be repeated for the other four diffusers. Once this is done, the Sound power can be converted to sound pressure using either the Thompson or multiple array Equations. To simplify the process, it is common to use the sound levels for the closest diffuser (conservative) and only consider the 125 Hz octave band. Generally, if this band is okay, then the ductwork is providing enough attenuation.

Supply Duct Sound Path Example
Referring to the previous example, estimate the sound pressure in the space using the multiple ceiling diffuser array equation and the Thompson equation. The space is 50 ft x 40 ft x 8 ft. The surface area is 5440 ft² and the Room Constant is given below.

The sound power level at the duct opening is

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	14.2	41.2	30.6	17.5	0	0	0	0
R	934	1355	1287	1478	1840	1852	1756	1678

Multiple Ceiling Array Method
Calculate x, the ratio of floor area served by each outlet to the square of the ceiling height.
X = (2000 ft²/6 outlets)/8²
X= 5.2

Calculate Sa
Sa = 5Log(x)+28Log(h)-1.13Log(DIFNUM)+3Log(f)-31

For 125 Hz
Sa = 5Log(5.2) + 28Log(8) – 1.13Log(6) + 3Log(125) – 31
Sa = 3.3

Lp = Lw – Sa

Lp = 41.2 – 3.3 = 37.9 dB RE 20 μPa

Sa	2.4	3.3	4.2	5.1	6.0	6.9	7.8	8.7
Lp	**11.8**	**37.9**	**26.4**	**12.4**	**0**	**0**	**0**	**0**

Thompson Equation Method

Calculate mean free path (MFP)

MFP = 4*ROOMVOL/Surface area

MFP = 4*(50 ft * 40 ft * 8 ft)/5440 ft²

MFP = 11.8

d = 16.6 ft

m can be found in Table 12

$Lp = Lw + 10Log(Q \bullet e^{-md} /(4\pi d^2)+(MFP/d) \bullet (4/R)) + 10 Log (N) +10.5$

For 125 Hz

$Lp = 41.2 +10Log[2*e^{-0*16.6}/(4\pi 16.6^2) + 11.8/16.6)*(4/1355)] + 10Log(6) +10.5$

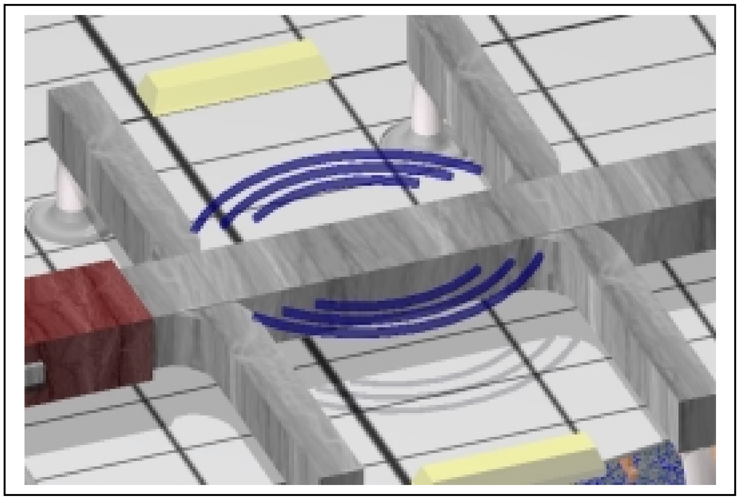
Lp = 33.7

Lp	**8.1**	**33.7**	**23.3**	**9.8**	**0**	**0**	**0**	**0**

There is a difference between the two approaches, particularly in the lower bands. Use judgement to determine which equation to use. While this is a large room, it is not quite an open plan space. When in doubt, use the more conservative values.

Duct Breakout Sound Path

Figure 23 - Duct Breakout



Estimating the Breakout Sound Power

As sound energy travels down a duct, some of the energy escapes through the duct wall. We refer to this as breakout. Breakout can be a significant sound source when evaluating sound levels in a room, and it can result from a duct passing near an occupied space that does not serve that area. This is particularly true near mechanical rooms where main supply and return ducts pass over occupied spaces as they enter the mechanical room.

Figure 23 - Duct Breakout, page 44, shows sound emanating from the duct surface. Breakout is often the source of low frequency rumble in the space.

The sound energy that breaks out from the duct is defined as;

Eq. 30

$$Lw_r = Lw_i + 10\, Log[S/A] - TL_{out}$$

Where

Lw_r is the sound power radiated from the duct in decibels

Lw_i is the sound power in the duct in decibels

S is the outer surface area of the duct in $inches^2$

A is the cross-sectional area of the duct in $inches^2$

TL_{out} is the breakout transmission loss in dB

Breakout in Rectangular Ducts

For rectangular ducts;

Eq. 31

$$S = 24*L*(w+h)$$

$$A = w*h$$

Where

L is the duct length in feet (m)

Breakout in Circular Ducts

For circular ducts;

Eq. 32

$S = 12 * L\pi d$

$A = \pi d^2/4$

Where

D is the diameter in inches

L is the duct length in feet (m)

Table 18 - Breakout TL_{out} for Rectangular Duct [12]

	Octave Bands							
Duct Size	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
12 x 12	21	24	27	30	33	36	41	45
12 x 24	19	22	25	28	31	35	41	45
12 x 48	19	22	25	28	31	37	43	45
24 x 24	20	23	26	29	32	37	43	45
24 x 48	20	23	26	29	31	39	45	45
48 x 48	21	24	27	30	35	41	45	45
48 x 96	19	22	25	29	35	41	45	45

Table 19 - Breakout TL_{out} for Long Seam Round Duct

	Octave Bands							
Duct Size	**63**	**125**	**250**	**500**	**1000**	**2000-**	**4000**	**8000**
8 in Dia	45	53	55	52	44	35	34	27
14 in Dia	50	60	54	36	34	31	25	20
22 in Dia	47	53	37	33	33	27	25	20
32 in Dia	51	46	26	26	24	22	38	30

Table 20 - Breakout TL_{out} for Spiral Round Duct

	Octave Bands							
Duct Size	**63**	**125**	**250**	**500**	**1000**	**2000-**	**4000**	**8000**
8 in Dia	48	64	75	72	56	56	56	45
14 in Dia	43	53	55	33	34	35	25	20
22 in Dia	45	50	26	26	25	22	36	29
32 in Dia	43	42	28	25	26	24	40	32

Table 18, Table 19 and ***Table 20*** are typical TL_{out} values for rectangular, long seam and spiral ducts that can be used to estimate breakout. The Tables do not account for sound energy attenuation along the duct, so they should only be used for ducts that are 20 to 30 feet (7 to10 m) in length.

Acoustic Analyzer uses algorithms to calculate breakout TL_{out}. Notice that the TL_{out} for round ducts are significantly higher than for rectangular ducts. Using round ducts can reduce breakout, but rectangular ducts are better at attenuating sound. Using rectangular duct can reduce discharge sound levels.

Insulation, such as acoustic lining for discharge sound attenuation, has little affect on breakout. To reduce breakout sound levels, the duct mass can be increased with a heavier gauge metal, or by lagging material to the outside of the duct.

Estimating the Breakout Sound Pressure in a Space

Once the breakout sound power level has been estimated, the sound pressure level in the space can be calculated. The equations for estimating breakout sound pressure are based on the basic equation 8

[12] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.

for a line source. A line source equation is used instead of a point source equation (e.g. Schultz or Thompson) because duct breakout is essentially a long line across the top of the room.

Eq. 33

$$Lp = Lw - 10Log(\pi dL)+k$$

Where

d is the distance from the source to the measurement point

L is the length of the sound source in feet (m)

K is a constant whose value is 10.5 for I-P and 0.5 for SI

Duct Breakout Example

Referring to the previous example, estimate the sound pressure in the space due to breakout from the main 22" by 22" rectangular duct. The duct length is 41.7 feet. The ceiling is made up of Mineral Fiber tiles with 1 lb. density.

The sound power level at the duct inlet is

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	43.0	67.8	56.1	46.5	24.9	13.7	15.4	14.2

To Calculate the sound power due to breakout.

Calculate the surface area:

S = 24*L*(w+h)

S = 24*40*(22 + 22)

S = 44,000 in²

Calculate the cross sectional area:

A = w*h

A = 22 * 22

A = 484 in²

Look up TL_{out} in Table 18. Use 24 x 24 for duct size:

TL_{out}	20	23	26	29	32	37	43	45

Calculate the breakout sound power level. Using 125 Hz as an example:

$Lw_r = Lw_i + 10 \text{ Log}[S/A] - TL_{out}$

$Lw_r = 67.8 + 10\text{Log}[44{,}000/484] – 23$

$Lw_r = 64.4$ dB

Lw_r	42.6	64.4	49.7	37.1	12.6	0	0	0

Subtract plenum attenuation from Table 13:

Plenum	3	6	8	10	16	21	36	21
Lw_r	39.6	58.4	41.7	27.1	0	0	0	0

Calculate the sound pressure level. Using 125 Hz as an example:

$Lp = Lw - 10\text{Log}[\pi dL]+10.5$
$Lp = 58.4 - 10\text{Log}[\pi*5*41.7] + 10.5$
$Lp = 46.8$ dB

Lp	27.9	46.8	30.0	15.5	0	0	0	0

Notice the duct wall attenuated the high frequency sound while the low frequency sound escaped. This is the source of the low frequency rumble occupants hear many ducted systems.

Return Air Sound Path

Estimating Return Air Path Sound Power

The return sound path depends on several factors. For most HVAC equipment, the return sound energy comes from a portion of the fan sound energy. This information is not always made available, so it is up to the designer to identify the sound energy levels (Refer to ***Return Air vs. Supply Air Sound Power***, page36).

Return Air Sound Power Attenuation

Return air systems may or may not include ducting. In some cases ducting is required to bring air to the unit. This is common in classroom applications where the HVAC unit is in the corridor and the corridor wall is fired rated. In an open plenum approach, ducting is not required but often considered as a sound attenuating solution. This can be a very good idea because the supply air sound energy and the return air sound energy are about equal and there is not a lot of attenuation on the return air side.

If the ducting ends in the plenum space or there is no ducting, then the plenum can offer attenuation as described in ***Ceiling Plenums****,* page 36. Return air systems that are ducted directly to the space will channel the sound energy to the space and not take advantage of the plenum.

Ducted return air systems enjoy the same attenuation benefits from ducting as supply air systems. The ducting elbows and other fittings, end reflection, etc., all come into play.

Estimating Return Air Sound Pressure in a Space

The first step in estimating return air sound pressure in a space is to identify if there is ducting and/or other sound attenuating features. If this is the case, then identify whether the ducting system path leads directly to the occupied space, or if the ducting system ends in the ceiling plenum.

If the return air path is simply the back of the HVAC unit, then the *source - path – receiver* concept becomes HVAC return air sound power levels – ceiling plenum attenuation (if any) – room effect calculation such as Schultz for a single point or Thompson.

If the HVAC unit has some ducting, such as a return air elbow, then the process is HVAC return air sound power levels – duct fitting attenuation – ceiling plenum attenuation (if any) – Room effect calculation such as Schultz for a single point or Thompson.

If the HVAC unit is ducted directly to the occupied space, then the process is HVAC return air sound power levels – duct fitting attenuation – Room effect calculation such as Schultz for a single point or Thompson. If there is a return air grille, then a new sound source may be created. This can be evaluated in a similar manner as shown in ***Diffuser Sound Path***, page 50.

Return Air Sound Path Example

Referring to the previous example, estimate the sound pressure in the space due to the return air path. The HVAC unit has a 24” by 24” return air elbow lined with 1” acoustic insulation.

The Return Air sound power levels for the WSHP unit is:

Band	63	125	250	500	1000	2000	4000	8000
Lw	49	71	59	53	41	27	26	23

Since this data is based on ARI 260, it is appropriate to apply the Environmental Correction Factor:

Corr.	4	2	1	0	0	0	0	0
Corr. Lw	**45**	**69**	**58**	**53**	**41**	**27**	**26**	**23**

The first duct section is 24” by 24” by 2 ft long. It is lined. The attenuation is:

Duct	0.6	0.6	1.2	2.9	5.8	4.9	4.2	3.7
Lw	**44.4**	**68.4**	**56.8**	**50.1**	**35.2**	**22.1**	**21.8**	**19.3**

The 24” square elbow is also lined. The attenuation is:

Elbow	0	1	6	11	10	10	10	10
Lw	**44.4**	**67.4**	**50.8**	**39.1**	**25.2**	**12.1**	**11.8**	**9.3**

The last duct section is 24” by 24” by 2 ft long. The duct attenuation is:

Duct	0.9	1.0	1.8	4.4	8.7	7.4	6.3	5.6
Lw	**43.5**	**66.4**	**49.0**	**34.7**	**16.5**	**4.7**	**5.5**	**3.7**

The end reflection is based on terminating in a free space (Table 16). The attenuation is:

End Ref	8.2	4.1	1.5	0.5	0.1	0.0	0.0	0.0
Lw	**35.3**	**62.3**	**47.5**	**34.2**	**16.4**	**4.7**	**5.5**	**3.7**

The ceiling plenum attenuation is:

Pln Eff.	3	6	8	10	16	21	36	21
Lw	**32.3**	**56.3**	**39.5**	**24.2**	**0.4**	**0**	**0**	**0**

Use the Schultz equation to convert to sound pressure. Using the 125 Hz octave band as an example:

Lp = 56.3 – 10Log(25.5) - 5Log(16,000) – 3Log(125) + 10Log(1) + 25

Lp= 40 dB

The space sound pressure due to return air sound power is:

Lw	**17**	**40**	**22**	**6**	**0**	**0**	**0**	**0**

Note, the results for no return air ducting and for return air ducted to occupied space are as follows:

No Duct	**36**	**56**	**43**	**36**	**18**	**0**	**0**	**0**
Ducted	**19**	**44**	**24**	**4**	**0**	**0**	**0**	**0**

Notice the return air elbow in this case provided the most significant attenuation due to the ceiling plenum affect. Return air elbows are an excellent way to reduce the sound levels with decentralized equipment located near the occupied space.

Diffuser Sound Path

As air passes through diffusers it generates sound (noise), so diffusers become a sound source. The sound level is proportional to the air discharge velocity, which also affects the diffuser throw. This sound energy tends to occur in the higher frequencies.

Diffuser Catalog Data

Diffuser catalogs generally rate products for air volume in cfm (m^3/h), throw in fpm (m/s), pressure drop in inches w.c. (Pa) and NC level. ***Figure 24 - Typical Diffuser Selection Catalog Showing Sound Levels*** shows performance data for a typical lay-in type ceiling diffuser. In addition to the NC level, the table also includes sound power data, which is very useful but less common.

Figure 24 - Typical Diffuser Selection Catalog Showing Sound Levels

1400 HORIZONTAL THROW **PANEL SIZES 24" x 24"** **(610 x 610) MM** **IP & (SI)**

10" neck
(254) mm neck

Ft² m²
Neck Area = .545 (.051)
Ak = .350

Velocity		Volume		Total pressure		Static pressure		Isothermal throw to terminal velocities: fpm (m/s)						Octave Band Sound Power Levels in dB RE: 10 (-12) Watts								
FPM	(m/s)	CFM	(L/s)	inch w.g.	(kPa)	inch w.g.	(kPa)	150	(.762)	100	(.508)	50	(.254)	1	2	3	4	5	6	7	8	NC
524	(3)	286	(135)	0.047	(0.012)	0.030	(0.007)	4	(1355)	7	(2032)	13	(4064)	42	26	29	29	28	*	*	*	15
581	(3)	317	(150)	0.057	(0.014)	0.036	(0.009)	5	(1476)	7	(2214)	14	(4349)	43	29	32	31	31	14	*	11	18
644	(3)	351	(166)	0.070	(0.018)	0.045	(0.011)	5	(1634)	8	(2452)	15	(4576)	45	32	35	33	34	19	*	12	21
709	(4)	387	(183)	0.086	(0.021)	0.054	(0.014)	6	(1826)	9	(2739)	16	(4837)	46	34	37	35	36	24	10	13	24
809	(4)	441	(208)	0.111	(0.028)	0.071	(0.018)	7	(2040)	10	(3059)	17	(5112)	47	37	41	38	40	30	16	15	28
889	(5)	485	(229)	0.135	(0.033)	0.085	(0.021)	7	(2198)	11	(3297)	17	(5307)	48	40	44	40	43	35	20	16	31
987	(5)	538	(254)	0.166	(0.041)	0.105	(0.026)	8	(2440)	12	(3660)	18	(5591)	49	42	46	43	46	40	24	17	34
1068	(5)	583	(275)	0.194	(0.048)	0.123	(0.031)	9	(2735)	13	(4102)	19	(5919)	50	44	48	44	48	44	28	18	37
1173	(6)	640	(302)	0.234	(0.058)	0.148	(0.037)	9	(2867)	14	(4285)	20	(6061)	51	47	51	46	51	49	32	19	40

*NC values are based on a room absorption of 10dB, re 10^{-12} watts, ANSI/ASHRAE 70-1991, 6.5.1.5. The NC levels are based on octave band Sound Power levels in the range of actual reverberation room tests. *<audible*

To provide sound performance such as an NC value requires defining the space so the diffuser sound power data can be converted from sound power to sound pressure and plotted on a NC chart. Notice the paragraph at the bottom of ***Figure 24*** which states that a 10 dB room affect was used in all octave bands to estimate NC levels. Subtracting 10 dB from the sound power levels and plotting the data on an NC chart will give you the listed NC level. While the NC value allows the designer to compare one product to another, it probably does not represent the sound pressure level in the space created by the diffuser because it is very unlikely that the space has a 10dB room effect in all octave bands.

Also, consider that the cataloged tables are only for one diffuser. If there is more than one diffuser, then the sound power levels (and the NC level) will increase. For instance, two 10-inch diffusers at 317 cfm would have a NC level 3 dB higher than the cataloged NC 18.

Estimating Diffuser Sound Pressure in a Space

Each diffuser will be a sound power source. Where possible, it is best to use the sound power data as provided in the figure above. The sound energy is then converted into sound pressure in the same manner as the sound power for ducted discharge. The Thompson or Schultz equation can be used. If there are four or more diffusers, then the multiple ceiling array equation can be used.

The distance used from the diffuser to the measurement point should be carefully considered. Two locations should be checked – directly under a diffuser and the center of the room.

Diffuser Sound Path Example

Referring to the previous example, estimate the sound pressure in the space due to 6 diffusers using the multiple ceiling diffuser array equation and the Thompson equation. The diffuser sound power levels given here.

The sound power level for each diffuser is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	44	30	33	32	32	16	0	11
R	934	1355	1287	1478	1840	1852	1756	1678

Multiple Ceiling Array Method

Calculate x, the ratio of floor area served by each outlet to the square of the ceiling height:

$X = (2000\ ft^2/6\ outlets)/8^2$

$X = 5.2$

Calculate Sa using 125 Hz as an example:

$Sa = 5Log(5.2) + 28Log(8) – 1.13Log(6) + 3Log(125) – 31$

$Sa = 3.3$

$Lp = Lw – Sa$

$Lp = 30 – 3.3 = 26.7$ dB RE 20 μPa

Sa	2.4	3.3	4.2	5.1	6.0	6.9	7.8	8.7
Lp	**41.6**	**26.7**	**28.8**	**26.9**	**26.0**	**9.1**	**0**	**2.3**

Thompson Equation Method

Calculate mean free path (MFP)

MFP = 4*ROOMVOL/Surface area

MFP = 4*(50 ft * 40 ft * 8 ft)/5440 ft²

MFP = 11.8

d = 16.6 ft to the room center and 3 ft

m can be found in Table 12

$Lp = Lw + 10Log(Q \cdot e^{-md} /(4\pi d^2)+(MFP/d)\cdot(4/R)) + 10 Log (N) +10.5$

For 125 Hz

$Lp = 30 +10Log[2*e^{-0*16.6}/(4\pi 16.6^2) + 11.8/16.6)*(4/1355)] + 10Log(6) +10.5$

Lp = 23 dB

Lp @ 16’	**38**	**23**	**26**	**24**	**24**	**8**	**0**	**3**

The sound pressure 3 feet below one diffuser is:

Lp @ 5”	**40**	**25**	**28**	**27**	**27**	**11**	**0**	**6**

Notice that the Sa values varied in every octave band and did not reach 10dB anywhere. The NC level listed in the diffuser performance table would have been very misleading in this application. Also notice the sound levels for all the diffusers to the center of the room vs. directly under one diffuser. The single diffuser is louder and more directly correlates the Thompson and multiple array equation results.

Radiated Sound Path

When a terminal unit or decentralized HVAC unit is place near the occupied space (such as in the ceiling plenum) radiated sound should be considered. Most equipment manufacturers can provide radiated sound power levels for their equipment (fan coils, WSHPs, VAV boxes, etc.). For most products it is appropriate to include the environmental correction factor.

Estimating Radiated Sound Pressure in a Space

The radiated sound path is similar to the return air sound path. The ceiling plenum (if any) can provide considerable attenuation. Converting from sound power to sound pressure can be accomplished with either the Schultz or Thompson equation.

Radiated Sound Path Example

Referring to the previous example, estimate the sound pressure in the space due to the radiated path.

The Radiated sound power levels for the WSHP unit are:

Band	63	125	250	500	1000	2000	4000	8000
Lw	49	71	59	53	41	27	35	23

Since this data is based on ARI 260, it is appropriate to apply the Environmental Correction Factor:

Corr.	4	2	1	0	0	0	0	0
Corr. Lw	**45**	**69**	**58**	**53**	**41**	**27**	**35**	**23**

The ceiling plenum attenuation is:

Pln Eff.	3	6	8	10	16	21	36	21
Lw	**42**	**63**	**50**	**43**	**25**	**6**	**0**	**2**

Use the Schultz equation to convert to sound pressure. Using the 125 Hz octave band as an example:

Lp = 63 – 10Log(25.5) - 5Log(16,000) – 3Log(125) + 10Log(1) + 25

Lp= 55 dB

The space sound pressure due to return air sound power is

Lw	**35**	**55**	**41**	**33**	**14**	**0**	**0**	**0**

Using the Thompson equation the results are:

Lw	**36**	**56**	**43**	**36**	**18**	**0**	**0**	**0**

Evaluating All the Sound Paths

The final step in evaluating sound pressure levels in a room is to evaluate how all of the various sound paths act together to create the sound level within the space. The various sound paths must be added together logarithmically. The total Lp is the logarithmic sum (See ***Decibel Addition and Subtraction***, page 6) of all the sound paths. The total Lp can then be used with the sound criteria of choice (such as NC or RC levels).

Sound Path Evaluation Example

Consider the previous sound path examples and evaluate whether the space can meet NC 35.

List the results of the five sound paths and total the sound pressure in the space using decibel addition.

Using the 125 Hz band as an example:

$$\text{Total Lp} = 10\text{Log}(10^{34/10} + 10^{40/10} + 10^{25/10} + 10^{47/10} + 10^{55/10})$$
$$= 55 \text{ dB}$$

	Octave Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Discharge	49	71	59	53	41	27	26	23
Radiated	49	71	59	53	41	27	35	23
Supply Fan Lp	8	34	23	10	0	0	0	0
Return Duct Lp	17	40	22	6	0	0	0	0
Diffuser Lp	40	25	28	27	27	11	0	6
Breakout Lp	28	47	30	16	0	0	0	0
Radiated Lp	35	55	41	33	14	0	0	0
Total Lp	41	55	41	34	27	11	3	6
Requested NC level	60	52	45	40	36	34	33	32
Required Attenuation	0	3	0	0	0	0	0	0

The NC level is 38 and is set by the level in the 125 Hz band. This demonstrates the common advice in HVAC acoustics that if you can attenuate the 125 Hz band, then the system should be okay. This can be helpful if the calculations are being done by hand.

If the 125 Hz sound pressure level was lowered 3 dB, then the requested NC 35 criteria would be met. The radiated sound level is setting the 125 Hz sound pressure level. The designer can now consider equipment options (perhaps two smaller units) or relocating the unit so the sound level is reduced.

Indoor Sound Analysis – Central Systems

General

Central systems such as air handling units serving multiple spaces are similar to the decentralized systems. Many of the sound calculation techniques used in decentralized systems can be used with central systems.

The larger scale of equipment generally means larger sound power sources, but the equipment is usually more removed from the occupants. The large fan sound energy involved in central systems introduces the concept of silencers which are added the HVAC system to reduce sound energy. Often the designer is attempting to evaluate the sound energy transmitted through ducting to the space to estimate the insertion loss required of the silencer to meet the required sound criteria. Many silencer manufacturers develop software that is focused on evaluating ducting systems for sizing silencers.

The high level of sound energy in a mechanical room creates a need to evaluate sound transmission through the wall into occupied spaces. This topic will be discussed in detail.

Regenerated noise is caused when airflow creates a sound source in the ducting system. Regenerated noise becomes an issue with larger systems with higher air flows and air velocities.

Multiple Paths

Figure 25 - Central System Multiple Paths

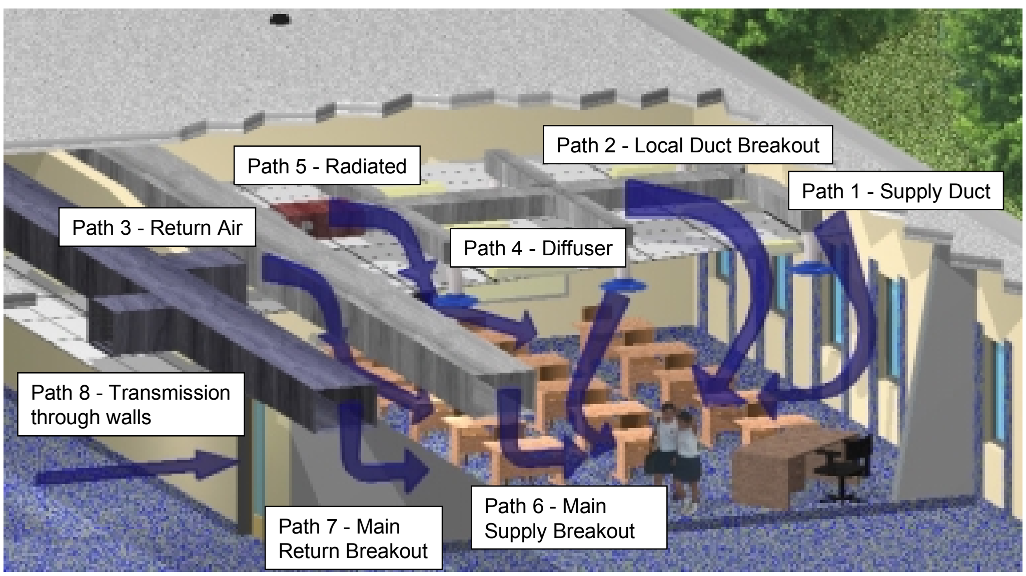


Figure 25 - Central System Multiple Paths shows a central system in a mechanical room serving several spaces. The mechanical room impacts the building with sound energy from the fan systems that is distributed throughout the building by the ductwork, and by the transmitted sound energy from the equipment (e.g. chiller).

The space being considered is evaluated in a similar manner to a decentralized system serving a room. For instance a central VAV system will have the following sound paths:

- ❑ Ducted sound energy path (Path 1)
- ❑ Return air sound path (Path 3)
- ❑ Diffuser sound path (Path 4)

- Radiated sound path with a VAV box being the radiated sound source (Path 5)
- Duct breakout path (Path 2)

In addition, a central system can introduce duct breakout from the main supply and return ducts (Path 6 and 7) because air is being distributed throughout the building and may pass over the space being considered. In addition, transmitted sound from the mechanical room (Path 8) may need to be considered.

Central systems can also serve large spaces such as an open plan office environment. These will be discussed in the following sections.

Central System Duct Sound Path

The ducted sound path for central systems is evaluated in the same manner as for decentralized systems (Refer to ***Duct Sound Path***, page 37). The main differences are that several spaces are served by a single central system and the source sound energy levels are usually much larger.

Having several occupied spaces introduces the issue of which spaces need to be evaluated. Acoustic treatment of ducted systems will affect the sound levels of all spaces downstream of the treatment. Usually the treatment (such as a duct silencer) is installed at or near the beginning of the duct system so that all spaces will have reduced sound levels. In this case, it becomes necessary to choose the "worst case" space and evaluate it.

It is not always obvious which space(s) need to be evaluated and experience is usually the best teacher. There are two key parameters to watch for when choosing. The first is the distance through the duct work from the sound source. The ductwork will naturally attenuate, so spaces further away from the source are generally better. Conversely, spaces close the source should be reviewed.

The second key issue is the percentage of supply air delivered to the space. Sound energy is almost proportional to the percentage of supply air. This means that a space close to the mechanical room that only receives a small percentage of the supply air may not be the worst case. Conversely, a large space far from the mechanical room may need to be reviewed.

It is also important to balance the supply duct sound path with other sound paths when evaluating which spaces need to be reviewed. Consider the return air paths and main duct breakouts when choosing spaces.

Air Handling Units and Return Fans

Central systems usually have larger fans, which are the main sound power source in the ducted system. Fans are, for the most part, the sound energy source for both supply and return duct systems. Fans are used when air movement is required. This can include indoor and outdoor air handling units, vertical self-contained units, applied or packaged rooftop air conditioning units. Return fans can be either inline, base mounted fans directly connected to the ductwork or cabinet fans where the fan is in a cabinet.

Figure 26 – Examples of McQuay Air Handling Products with Integral Fans

Vision™ Indoor Air Handler | **RoofPak™ Applied Rooftop Systems**

ASHRAE developed a methodology to estimate sound power levels from fans based on the physical and performance parameters of the fan. This is no longer supported by ASHRAE (since 1995), but it is still widely used.

Most air moving equipment today is tested for sound power levels and the data is readily available. It is recommended that this data be used when evaluating air movement systems. Individual fans are usually tested to *AMCA Standard 300-85*. This provides the total fan power of just the fan in eight octave bands from 63 to 8000 Hz. A common practice is to divide the sound power in half (subtract 3 dB from the cataloged sound power) with one half going to the supply and the other half going the return connection of the fan. This is not, by any means, necessarily true. The conservative answer is to use the total sound power for each direction.

Individual fan sound power is used for inline return air fans and fans used in custom air handling units. The air handling unit itself can provide a significant amount of sound attenuation. With custom air handling units, it is possible to either test the final unit to account for the unit attenuation, or to estimate the sound attenuation. What the designer should be looking for is the sound power levels at all duct connections to the air handling unit, and the radiated sound power through the casing and into the mechanical room/outdoor air area (Refer to ***Figure 27 - McQuay Vision AHU Sound Power Data***).

Figure 27 - McQuay Vision AHU Sound Power Data

UNIT SOUND	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
Radiated	73	72	67	65	57	45	38	30
Unit discharge	91	93	91	92	86	80	76	68
Unit return	88	90	85	85	74	64	53	44

Commercial or applied products are often tested to *ARI Standard 260, Sound Rating of Ducted Air Moving and Conditioning Equipment*. This standard has the advantage of testing the fan in the actual application (e.g. the fan is tested as installed in an air handling unit) as opposed to the near ideal conditions provided by the AMCA standard. If a unit fan is tightly fitted into the cabinet, it is possible for the fan to be 5 to 10 dB louder than indicated by the AMCA test standard below 250 Hz.

Duct Silencers

Duct silencers (sometimes referred to as sound attenuators) are devices that are designed to absorb or cancel sound energy in ductwork. Silencers can be static devices designed to absorb sound or dynamic devices designed to cancel out sound waves. Static silencers are either dissipative or reactive. Dissipative silencers have sound absorbing material, such as fiberglass, that is usually encased in perforated liners. Reactive silencers do not have sound absorbing material. Instead, they attenuate sound using the Helmholtz resonator principle. Dynamic or active silencers electronically generate a sound wave that is equal in amplitude, but opposite in phase to cancel out the sound source.

Most silencers used in typical HVAC applications are the dissipative type. They are rated in terms of the insertion loss they provide and their air pressure drop (refer to ***Figure 33 - Typical Silencer Selection Showing Generated Noise***, page 76). Generally, the larger the air pressure drop, the more regenerated noise they create.

Silencers are usually installed near the sound sources, such as the air handling unit or return fan. In some cases the manufacturer can integrate the silencer into the air handling or rooftop unit and provide a complete package.

Where mechanical room break-in may be a concern (e.g. sound energy entering the duct system and using it as a conduit to travel throughout the building), the silencer should be installed in the ductwork in the mechanical room as close the wall as possible. This will allow the silencer to attenuate any break-in sound from the mechanical room.

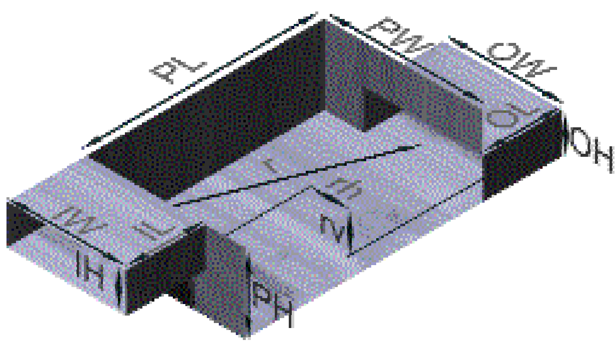
Figure 28 - Typical Duct Silencers

Straddling the wall is a very good option, but often the mechanical room wall is fire rated and requires a fire damper. Placing silencers outside the mechanical room wall is also possible, but the silencers are a sound source and the radiated sound may enter an occupied space.

HVAC designers are often focused on silencer selection as a key part of their HVAC design. Controlling supply air sound in the occupied spaces is often a key aspect to controlling the overall sound level in the space. Many silencer manufacturers provide software modeling tools to estimate the required insertion loss for silencer selection. Attenuating the supply sound source does not always provide acceptable sound levels in the space. There are many other sound paths to the space that must be considered.

Duct Plenums

Figure 29 - Typical Plenum with Required Dimensions



Plenums are common in central system designs. For example, a rooftop unit may have a return air plenum just below the roof deck to draw air from the ceiling space. Also, the outdoor inlet to an indoor air handling unit is often a plenum. Plenums can provide significant sound attenuation because the abrupt changes in cross-sectional area cause end reflection, and there can be a lot of surface area for sound absorbing material.

The transmission loss associated with a plenum can be described as follows:

Eq. 34

$$TL = -10\text{Log}[S_{out}\,[(Q\cdot\cos\theta)/(4\pi r^2) + (1 - \alpha_{avg})/(S\cdot\,\alpha_{avg})]]$$

Where

TL is the transmission loss in dB

S_{out} is the area of the out going duct in ft² (m²)

Q is the directivity which equals 2 if the inlet is near the center of the side it is located on. Q equals 4 if the inlet is in a corner of the side it is located on.

R is the distance between the centers of the inlet and outlet in ft (m).

S is the total inside surface area of the plenum minus the inlet and outlet areas in ft² (m²)

θ is the angle of the vector r and the horizontal plan and can be written as

$r = (rh^2 + rv^2)^{1/2}$

$\cos\theta = rh/r$

α_{avg} is the average absorption coefficient of the plenum lining as given by

$\alpha_{avg} = S_1\alpha_1 + S_2\alpha_2)/S$

S_1 and α_1 are the surface area and coefficient of any bare or unlined inside surfaces.

S_2 and α_2 are the surface area and coefficient of any bare or lined inside surfaces.

Table 21 - Absorption Coefficients For Plenum Materials[13]

	Octave Band							
Material	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Concrete	0.01	0.01	0.01	0.02	0.02	0.02	0.03	
Bare Sheet Metal	0.04	0.04	0.04	0.05	0.05	0.05	0.07	
3 lb density Fiberglass insulation	0.02	0.03	0.22	0.69	0.91	0.96	0.99	
4 lb density Fiberglass insulation	0.18	0.22	0.82	1.00	1.00	1.00	1.00	
5 lb density Fiberglass insulation	0.48	0.53	1.00	1.00	1.00	1.00	100	
6 lb density Fiberglass insulation	0.76	0.84	1.00	1.00	1.00	1.00	0.97	

Table 21 shows typical absorption coefficients for common plenum materials.

Eq. 34 assumes the plenum is a large enclosure and will only work if the wavelength of sound is small as compared to the characteristic dimensions of the plenum. At frequencies where plane waves exist, the equation is not valid. Plane wave propagation occurs at frequencies below the cutoff frequency (f_{co}):

Eq. 35

$f_{co} = c/2a$

Or

$f_{co} = 0.586c/d$

Where

f_{co} is the cutoff frequency in Hz

c is the speed of sound in air (1125 fps)

a is the larger cross sectional dimension of a rectangular duct in ft (m)

d is the diameter of a round duct in ft (m).

Where plane waves do occur, the plenum can be treated as an acoustically lined expansion chamber. These calculations would require computer. *Algorithms for HVAC Acoustics* contains the necessary equations.

[13] Reynolds, D. Jeffrey M. Bledsoe. 1991. *Algorithms for HVAC Acoustics.* American Society of Heating, Refrigerating, and Air-conditioning Engineers Inc. Atlanta, Ga.

Plenum Example

A plenum is to be added to the top of a McQuay SWP 055 vertical self-contained unit. The dimensions are 120" by 81" by 36". The fan discharges vertically through a 26" by 35" opening. The supply duct opening is 30" by 100". rh is 40" and rv is 18". The plenum wall has 2", 4-lb. density fiberglass insulation. What attenuation is expected?

The discharge sound power level for the SWP unit is:

Band	63	125	250	500	1000	2000	4000	8000
Lw	89	94	89	87	86	81	73	66

Calculate r:

$r = (rh^2 + rv^2)^{½} = (18^2 + 40^2)^{½} = 43.9"$

Calculate cosθ:

$\cos\theta = rh/r = 40/43.9 = 0.91$

Calculate S:

$S = [2(120 \times 81) + 2(81 \times 36) + 2(36 \times 120) – (30 \times 100 + 26 \times 35)]/144$

$S = 208\ ft^2$

Calculate α_{avg}:

The absorption coefficient for the walls and ceiling material can be looked up in Table 21, page 61. The floor of the plenum is the steel top of the HVAC unit. Bare metal can also be looked up in Table 21. Using 125 Hz as an example:

$\alpha_{avg} = S_1\alpha_1 + S_2\alpha_2)/S = [(147.2)(0.22)+(61.2)(0.04)]/208$

$\alpha_{avg} = 0.59$

α_{avg}	0.14	0.17	0.59	0.72	0.72	0.72	0.73	0.00

Calculate TL. Using 125 Hz as an example:

$TL = -10Log[S_{out}\ [(Q\cdot\cos\theta)/(4\pi r^2) + (1 – \alpha_{avg})/(S\cdot\ \alpha_{avg})]]$

$TL = -10Log(20.8[(2x0.910/4\pi 3.66^2) + (1 – 0.59)/(208 \times 0.59)]]$

TL = 1.4 dB

TL	0.7	1.4	5.3	5.8	5.8	5.8	5.8	

Check for frequency where plane wave propagation exists:

$f_{co} = c/2a = 1125/(2 \times 35/12)$

$f_{co} = 193$ Hz

Hence the calculations in the 63 and 125 Hz bands are not reliable and should be ignored. Computer software such as McQuay Acoustic Analyzer will estimate the sound level in these frequencies, assuming the plenum is an acoustically lined expansion chamber. The results in these bands are 5.5 and 8.0 db respectively.

The discharge sound power from the HVAC unit with the plenum should be:

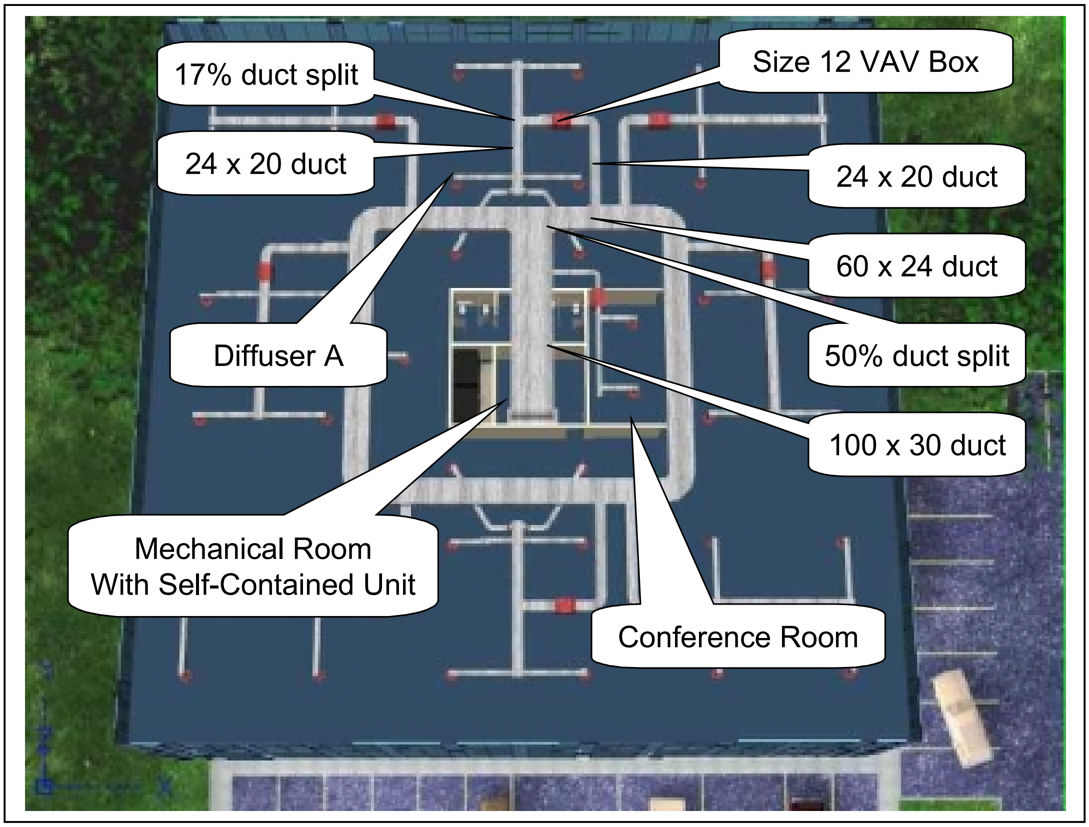
Lw	83.5	86.0	83.7	81.2	80.2	75.2	67.2	66.0

These values are conservative compared to actual equipment tests. The difference is mainly due to the actual geometry of the plenum. Whenever possible, use test data.

Evaluating Supply Duct Paths

Figure 30 - Central System Duct Plan, page 60 shows a typical central duct system for an office building that will be used to illustrate several types of acoustic analysis. The following two examples show the supply duct calculations required to estimate the required Insertion Loss for the silencer. Several silencer manufacturers provide software specifically for this type of analysis.

Figure 30 - Central System Duct Plan



Reviewing these examples shows the following:

- While the open office area is large and has many sound sources, it does not require a silencer. This is the result of good duct design and an effective discharge plenum. The conference room also does not require a silencer. Even though the conference room branch duct appears to be closely connected to the main supply air duct, the small amount of air being drawn by it (3%) introduces only a small amount of sound power to the room. Further improvement can be made by connecting the conference room branch duct to the ring supply duct instead of the main supply trunk.
- While good equipment and ducting design may resolve ducted sound transmission, it does not mean the space is acceptable. None of the other sound paths have been considered (return air, duct breakout, diffuser sound, etc.). Only sizing silencers is not enough to evaluate whether the space will be acceptable.
- Neither the large ceiling array equation or the Schultz equation provided 10 dB sound attenuation in all octave bands for the space. But many silencer sizing programs default to 10 dB in each band. In addition, the NC levels listed for diffusers are usually based on 10 dB per band.
- Even if the supply fan were perfectly attenuated by a silencer, the sound level from ducted sound in the open office area would be NC 27. This sound power is coming from the VAV box, which is downstream of the silencers. Again, there is usually more than one source for sound energy.

Supply Duct Analysis Example - Open Office Area

Figure 30 - Central System Duct Plan, page 59, shows a floor serviced by a McQuay SWP 055 unit. Calculated the insertion loss (if any) required to maintain NC 35 in the open office area.

The open office area can be evaluated using the large area array equation. Engineering judgement is required to estimate the supply air sound path that is most critical. Since the ductwork to any diffuser is relatively even, diffuser A will be considered. The ducted sound energy that makes it to diffuser A is a combination of the sound power from the vertical self contained unit and the VAV box itself.

The discharge sound power level for the SWP unit with the plenum is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	83.5	86.0	83.7	81.2	80.2	75.2	67.2	66.0

Node 1 – 100 x 30 x 38 ft Rectangular Duct:

IL	6	4	3	1	1	1	1	1
Node 1	78	82	81	80	79	74	66	65

Node 2 – Duct Split:

IL	3	3	3	3	3	3	3	3
Node 2	75	79	78	77	76	71	63	62

Node 3 – 60 x 24 x 12 ft Rectangular Duct:

IL	3	2	1	0	0	0	0	0
Node 3	72	77	77	77	76	71	63	62

Node 4 – Duct Split:

IL	6.1	6.1	6	6	6	6	6	6
Node 4	66	71	71	71	70	65	57	56

Node 5 – 24 x 20 x 6 ft Rectangular Duct:

IL	2	1	1	0	0	0	0	0
Node 5	64	69	65	63	66	62	54	53

Node 6 – 24” Round Elbow:

IL	0	1	5	6	4	3	3	3
Node 6	64	69	65	63	66	62	54	53

Node 7 – 24 x 20 x 3 ft Rectangular Duct:

IL	1	1	0	0	0	0	0	0
Node 7	63	68	65	63	66	62	54	53

Node 8 – Size 12 VAV Box Sound Power Addition:

Lw	0	65	62	59	55	54	52	42
Node 8	64	70	67	64	66	62	56	53

Supply Duct Analysis Example – Cont'd

Node 9 – 24 x 20 x 5 ft Rectangular Duct

IL	1	1	1	0	0	0	0	0
Node 9	62	69	66	64	66	62	56	53

Node 10 – Duct Split:

IL	3.1	.31	3.1	3.1	3.1	3.1	3.1	3.1
Node 10	59	66	63	61	63	59	53	50

Node 11 – 24 x 20 x 10 ft Rectangular Duct:

IL	3	2	1	0	0	0	0	0
Node 11	57	64	62	61	62	59	52	49

Node 12 – Duct Split:

IL	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1
Node 12	53	61	59	58	59	56	49	46

Node 13 – 24 x 20 x 10 ft Rectangular Duct:

IL	4	3	2	1	1	1	1	1
Node 13	49	58	57	57	59	55	49	46

Node 14 – 12" Round Elbow:

IL	0	0	1	5	8	4	3	3
Node 14	49	58	56	52	51	51	46	43

Node 15 – 8" x 3 ft Flex Duct:

IL	2	3	5	8	9	9	6	0
Node 7	47	55	51	44	42	42	40	43

Node 16 – End Reflection:

Lw	18	12	7	3	1	0	0	0
Node 8	29	43	44	41	41	42	40	43

Estimate the space sound pressure level using large area array equation. The area served by this VAV box is 40 ft x 40 ft x 8 ft high with 4 diffusers.

X = (1600 ft²/4 outlets)/8²
X= 6.25

Calculate Sa using 125 Hz as an example:
Sa = 5Log(6.25) + 28Log(8) – 1.13Log(4) + 3Log(125) – 31
Sa = 4

Sa	3	4	5	6	7	7	8	9
Lp	26	40	39	35	34	35	31	33
NC 35	60	52	45	40	36	34	33	32
Req'd IL	0	0	0	0	0	1	0	1

Supply Duct Analysis Example – Conference Room

Figure 30 - Central System Duct Plan, page 59, shows a floor serviced by a McQuay SWP 055 unit. Calculated the insertion loss (if any) required to maintain NC 35 in the conference room. The conference room receives 400 cfm.

The discharge sound power level for the SWP unit with the plenum is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	83.5	86.0	83.7	81.2	80.2	75.2	67.2	66.0

Node 1 – 100 x 30 x 25 ft Rectangular Duct:

IL	4	3	2	1	1	1	1	1
Node 1	80	83	82	80	79	74	66	65

Node 2 – Duct Split:

IL	15	15	15	15	15	15	15	15
Node 2	65	68	67	65	64	59	51	50

Node 3 – 10 x 10 x 16 ft Rectangular Duct:

IL	5	3	2	1	1	1	1	1
Node 3	60	65	65	64	63	58	50	49

Node 4 – 10” Round Elbow:

IL	0	0	1	2	3	3	3	3
Node 4	60	65	64	62	60	55	47	46

Node 5 – 10 x 10 x 8 ft Rectangular Duct:

IL	3	2	1	1	1	1	1	1
Node 5	57	63	63	61	59	54	46	45

Node 6 – Size 8 VAV Box Sound Power Addition:

Lw		62	63	57	51	47	43	34
Node 6	57	66	66	62	60	55	48	45

Node 7 – 10 x 10 x 3 ft Rectangular Duct:

IL	1	1	0	0	0	0	0	0
Node 7	56	65	66	62	60	55	48	45

Node 8 – Duct Split:

IL	3	3	3	3	3	3	3	3
Node 8	53	62	63	59	57	52	45	42

Node 9 – 10 x 10 x 6 ft Rectangular Duct:

Lw	2	1	1	0	0	0	0	0
Node 9	51	61	62	59	57	52	45	42

Supply Duct Analysis Example Cont'd – Conference Room

Node 10 – Flex Duct:

IL	2	3	4	8	10	10	7	0
Node 8	49	58	58	51	47	42	38	42

Node 11 – End Reflection:

Lw	20	14	9	5	2	1	0	0
Node 9	29	44	49	46	45	41	38	42

Estimate the space sound pressure level using the Schultz equation. The conference room is 18 x 25 x 8 ft high with two diffusers. Use 5 ft as the distance from the source.

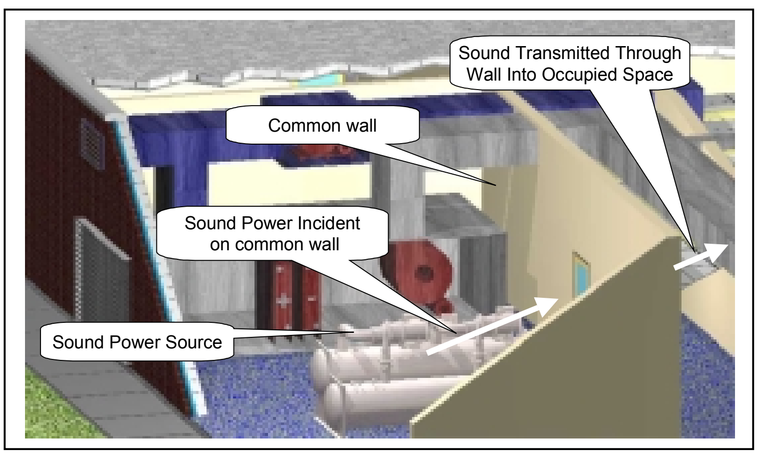
Calculate Room Effect using 125 Hz as an example:
RE = 10Log(5) + 5Log(3600) + 3Log(125) – 10Log (2) -25

Sa	5	6	7	8	9	10	11	11
Lp	24	38	42	38	35	31	27	30
NC 35	60	52	45	40	36	34	33	32
Req'd IL	0	0	0	0	0	0	0	0

Transmitted Sound

Sound energy in a space can pass through the walls, floor and ceiling and affect other spaces. An example is a mechanical room next to an occupied space. The equipment in the mechanical room can be heard in the occupied space and may be unacceptable.

Figure 31 - Transmitted Sound



Considering larger central systems introduces the concept of transmitted sound. Often the impact of mechanical rooms needs to be evaluated because the sound transmitted from them is another sound path that is logarithmically added to the other sound paths to estimate the sound level in the occupied space.

Estimating Transmitted Sound

A sound energy source such as a chiller, pump or air handling unit emits sound energy into the source room. If this space is a mechanical room, the surfaces are usually very hard and the room is acoustically live. A certain portion of the emitted sound energy acts on the common wall between the source room and the receiver room. How much energy acts on the common wall will depend on the source location with respect to the wall and the source room properties.

The common wall will reflect some sound energy, absorb some sound energy and transmit some sound energy. The transmitted sound energy will pass through the wall and act on the receiver room. From the receiver room's perspective, the common wall is a sound source. Surface sound sources create a very evenly distributed sound level in the adjacent space. As an occupant backs away from the common wall, the sound level will not change until the distance from the wall is almost equal to the larger dimension of the common wall. Beyond this point, the sound level will start to decrease at a rate that is dependent on the source room properties.

The following equations[14] can be used to estimate sound levels due to transmission. The process breaks down into the following steps:

i. Estimate the sound power incident on the common wall from the sound source.

ii. Estimate the sound power transmitted through the wall

iii. Estimate the affect of the sound power on the receiving room

Eq. 36 can be used to estimate the sound power incident on the common wall.

Eq. 36

$$Lw_{wall} = Lw_{source} + 10\mathrm{Log}[S_W[(1 - \alpha_{avg})/(S_M \cdot \alpha_{avg}) + 1/(4S_W + 4\pi l^2)]]$$

Where

Lw_{wall} is the sound power incident on the common wall in dB.

Lw_{source} is the sound power emitted by the source in dB.

S_W is the area of the common wall in ft² (m²)

α_{avg} is the average absorption coefficient of the surfaces of the source room

S_M is the surface area of the source room in ft² (m²)

l is the distance from the source to the wall.

The average absorption coefficient can be calculated using ***Eq. 37*** and the values listed in ***Table 11 - Typical Sound Absorption Coefficients***, page 30.

Eq. 37

$$\alpha_{avg} = (S_1\alpha_1 + S_2\alpha_2)/ S_M$$

Where

S_1 is the floor and ceiling area

α_1 is the absorption coefficient of floor and ceiling

S_2 is the total wall area

α_2 is the absorption coefficient of the wall

Mechanical rooms are often treated with sound absorbing material to reduce the sound energy. This can be accounted for by using ***Eq. 38***.

Eq. 38

$$\alpha_a = [PC \cdot \alpha_3 + (100\text{-}PC) \cdot \alpha_{avg}]/ 100$$

Where

PC is the percent of total room surface area covered by sound absorbing material

α_3 is the absorption coefficient of the sound absorbing material

[14] Reynolds, D. Jeffrey M. Bledsoe. 1991. *Algorithms for HVAC Acoustics.* American Society of Heating, Refrigerating, and Air-conditioning Engineers Inc. Atlanta, Ga.

The next phase is to estimate the sound energy that actually transmits through the wall. ***Table 22 - Transmission Loss Values*** shows transmission loss values that were obtained under ideal conditions. ***Eq. 39*** shows how to estimate the transmitted sound energy.

Eq. 39

$$Lw_{room} = Lw_{wall} - TL$$

Where

Lw_{wall} is the sound power incident on the common wall in dB.

Lw_{room} is the sound power that passes through the common wall in to the room.

TL is the transmission loss value for the specific wall type.

The quality of wall construction is a major factor in how much sound energy is allowed to pass through the wall. Because the transmission loss values are taken under ideal conditions, it is appropriate to take into account the wall construction as shown in ***Eq. 40***.

Eq. 40

$$TL = -10Log[(1 - \tau) \cdot 10^{-TL/10} + \tau]$$

Where

τ is correction coefficient as listed in ***Table 23 - Correction Coefficients For Wall Construction.***

Table 22 - Transmission Loss Values

Description	Octave Bands								
	31	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
4" poured concrete	29	35	36	36	41	45	50	54	58
8" poured concrete	34	36	37	41	45	49	53	57	61
12" poured concrete	36	36	38	44	48	51	55	59	63
8" Hollow Core Concrete Block	29	35	36	36	41	45	49	53	57
12" Hollow Core Concrete Block	32	36	36	37	43	47	51	55	59
4" Metal Stud Wall with 5/8" Drywall	8	11	20	30	37	47	40	44	35
4" Metal Stud Wall, 5/8" Drywall and 2" 3 lb/ft3 fiberglass insulation	9	14	23	40	45	53	47	48	38
4" Metal Stud Wall with 2 layers 5/8" Drywall	11	19	27	40	46	52	48	48	38
4" Metal Stud Wall, 2 layers 5/8" Drywall and 2" 3 lb/ft3 fiberglass insulation	13	24	32	43	50	52	49	50	40
1/8 in Single Pane Glass	2	8	13	19	23	27	27	27	31
1/4 in Single pane Glass	7	14	20	25	27	28	28	31	34
1/4 in Double Pane Glass, 1/2 in air gap	13	18	23	26	29	34	31	34	38
Wood Hollow core Door	0	2	7	12	17	18	19	22	30
Wood Solid Core Door	12	17	18	19	22	30	35	39	43
2 in Acoustic Metal Door	23	25	31	34	37	39	43	47	45
4 in Acoustic Metal Door	27	29	34	36	40	45	49	51	49
Two 4 in Acoustic Doors, 32 in Separation	42	48	54	60	67	75	84	90	95

Table 23 - Correction Coefficients For Wall Construction

Quality of Construction		τ
Excellent	No acoustic leaks or penetrations	0.00001
Good	Very few acoustic leaks and penetrations	0.0001
Average	Many acoustic leaks and penetrations	0.001
Poor	Many acoustic leaks and visible holes	0.01

The last step is to estimate the sound pressure in the space, given the sound power that passes through the wall. In this case, the sound source is a plane (the wall). Near the wall there will be no directionality. The sound level will remain relatively constant as the distance changes. At greater distances the sound will start to decrease. This usually occurs when the area of a hemisphere with a radius equal to the distance from the wall becomes greater than the area of the common wall. Beyond this point, the sound source behaves more like a point source than a plane source (see

Comparing Point, Line and Plane Sources, page 20).

For distances where $2\pi d^2 < Sw$, ***Eq. 41*** can be used.

Eq. 41

$$Lp_{room} = Lw_{room} + 10Log[1/Sw + 4(1+\alpha_{avg})/S_R\,\alpha_{avg}] + k$$

Where

S_R is the total surface area of the receiving room in ft² (m)

Sw is the surface area of the common wall in ft² (m)

α_{avg} is the average absorption coefficient for the receiving room which can be found in ***Table 12 - Average Sound Absorption Coefficients for Typical Receiving Rooms***, page 31

k is a constant whose value is 10.5 for I-P and 0.5 for SI

For distances where $2\pi d^2 > Sw$, ***Eq. 42*** should be used.

Eq. 42

$$Lp_{room} = Lw_{room} + 10Log(1/(2\pi d^2) + (MFP/d)\bullet(4/R)) + k$$

Where

d is the distance from the common wall to the receiver in feet (m)

MFP is the mean free path in feet as explained in ***Eq. 25***

R is the Room constant in ft² (m²)

k is a constant whose value is 10.5 for I-P and 0.5 for SI

Transmission Example

Figure 30 - Central System Duct Plan, page 59 shows a floor plan with a 20' x 15' x 10' high mechanical room that has a McQuay SWP 055 unit. The mechanical room walls are 8"hollow core concrete block of average construction. The floor and ceiling are 8" poured concrete. Calculate the transmitted sound in the conference room.

The discharge sound power level for the SWP unit is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	85	81	74	85	84	90	84	77

Calculate α_{avg} using ***Eq. 37*** and 125 Hz as an example:

$= (S_1\alpha_1 + S_2\alpha_2)/ S_M = [(840 \text{ ft2})(0.36) + (600\text{ft2})(0.01)]/1440 \text{ ft2} = 0.20$

α_{avg}	0.13	0.20	0.24	0.17	0.17	0.22	0.14	0.12

Calculate the sound power incident on the mechanical room wall common to the conference room. Using ***Eq. 36*** and 125 Hz as an example:

$Lw_{wall} = Lw_{source} + 10\text{Log}[S_W[(1 – \alpha_{avg})/(S_M \cdot \alpha_{avg}) + 1/(4S_W + 4\pi l^2)]]$

$= 81 + 10 \text{ Log}[150[(1\text{-}0.21)/(1300\cdot 0.21) + 1/(4\cdot 150 + 4\pi 10^2)]] = 80 \text{ dB}$

Lw	86	80	72	85	84	89	84	78

Calculate TL assuming average construction. Using ***Eq. 40*** and 125 Hz as an example:

$TL = -10\text{Log}[(1 – \tau) \cdot 10^{-TL/10} + \tau] = -10\text{Log}[(1 – 0.001) \cdot 10^{-36/10} + 0.001] = 29$

TL	29	29	29	30	30	30	30	30

Notice the TL drop from 36 to 29 due to the average construction.

Calculate Lw in the receiving room. Using ***Eq. 39*** and 125 Hz as an example:

$Lw_{room} = 80 – 29 = 51$

Lw_{room}	56	51	43	54	53	58	54	48

Calculate the sound pressure level in the conference room. First it must be ascertained whether the source behaves like a plane source or a point source. Using the room center point, $2\pi 9^2 = 509 < 1760$ (the surface area of the conference room) so the transmitted sound will be constant. Using α_{avg} for a medium dead room, ***Eq. 41*** and 125 Hz as an example:

$Lp_{room} = Lw_{room} + 10\text{Log}[1 /Sw + 4(1 + \alpha)/ S_R\, \alpha_{avg}]+k$

$= 56 + 10\text{Log}[1/150 + 4(1 + 0.22)/(1760*0.22)] + 10.5 =44$

Lp_{room}	50	44	37	48	46	50	45	40

An occupant would have to be at least 16 ft from the wall before they would start to hear a decrease in transmitted sound level. Closer than 16 ft, the common wall behaves like a giant speaker and the sound is constant throughout the space.

Main Duct Breakout

As a duct passes through an occupied space, sound energy can escape and affect the occupants. The process to account for this is covered in detail in ***Duct Breakout Sound Path***, page 44. Central systems should consider both the local ductwork that serves the space and the main supply and return ducts. The main trunks handle large air volumes and often have a significant amount of sound energy within them. If this sound energy breaks out, it can add to the background sound level in the space. This can be very problematic with low frequency fan noise.

The following example shows the calculation process using the same example as the supply duct calculations. In this case, the duct breakout from the main duct is actually louder than the sound power delivered to the diffusers. If the silencer sizing is based only on the sound level due to the ductwork, this breakout would be missed. In this case, the silencer should be sized to meet the needs of duct breakout, which is more than enough to meet the requirements of ducted discharge sound levels.

Main Duct Breakout Example

Figure 30 - Central System Duct Plan, page 59, shows a floor plan where the main supply duct passes over occupied space. Estimate the sound level due to sound breakout from the main trunk to the first duct split.

Using Node 1 from the supply duct sound analysis example, the sound power level is:

Band	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lw	74	61	58	56	61	62	51	50

To calculate the sound power due to breakout.

Calculate the surface area:

S = 24*L*(w+h)

S = 24*12*(100+ 30)

S = 37,440 in²

Calculate the cross sectional area:

A = w*h

A = 100 * 30

A = 3000 in²

Look up TL_{out} in Table 18. Use 48 x 96 for duct size

TL_{out}	19	22	25	29	35	41	45	45

Calculate the breakout sound power level using 125 Hz as an example:

$Lw_r = Lw_i + 10 \text{ Log}[S/A] - TL_{out}$

$Lw_r = 67.8 + 10\text{Log}[37{,}440/3000] – 22$

$Lw_r = 69$ dB

Lw_r	73	69	65	61	54	38	31	0

Subtract plenum attenuation from Table 13:

Plenum	3	6	8	10	16	21	36	21
Lw_r	70	63	57	51	38	17	0	0

Calculate the sound pressure level using 125 Hz as an example:

$Lp = Lw - 10\text{Log}[\pi dL]+10.5$

$Lp = 77 - 10\text{Log}[\pi*5*12] + 10.5$

$Lp = 50$ dB

Lp	57	50	44	38	25	4	0	0

This is NC 34 with a lot of sound energy in the lower bands. The occupants will hear this a duct rumble.

Return Air Path

Return air paths in central systems are often underestimated. A rooftop unit may only have an opening or short elbow to the ceiling plenum, which is a very short sound path. Air handling units used in gymnasiums often have no return fan and draw return air directly through a louver. While the supply air path has plenty of ductwork to help attenuate sound, the return path is very short. To make matters more difficult, a gymnasium is a very hard space.

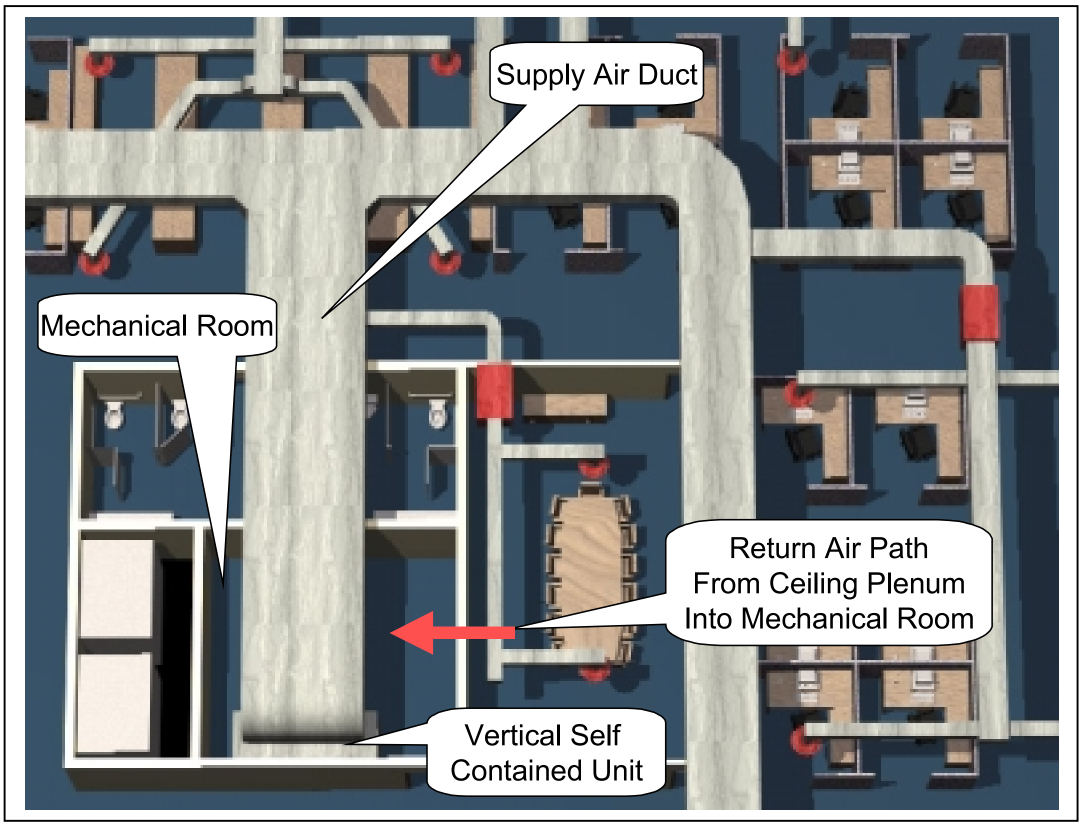
Spaces where the return air is ducted directly from the return air grille cannot take advantage of the attenuation offered by a plenum. Low static pressure on return fans often makes them very inefficient, so they generate a fair amount of noise relative to the static pressure and motor size.

***Return Air Sound Path*, page 48,** provides details on how to calculate return air sound paths. For the most part, they are similar to supply duct sound paths. Duct breakout from return ducts is also possible and should be considered.

Return Air Openings

The following example considers a special application of a vertical self-contained unit. The common application for a self-contained unit is to use the mechanical room as the return air plenum. Outdoor air is ducted directly into the mechanical room to be picked up by the supply fan and distributed throughout the occupied space. Air returns to the mechanical room from the ceiling plenum via openings above the ceiling tile. This provides a direct path from the HVAC unit to the occupied space.

Figure 32 - Return Air Path For Self-Contained Unit



Evaluating this style of mechanical room is similar to evaluating transmitted sound. A sound source (the HVAC unit) radiates sound energy into the mechanical room. The room absorbs some of the sound. The sound energy incident on the return air opening will pass through and into the ceiling plenum. The sound energy that is incident on the return air opening can be estimated as follows:

Eq. 43

$$Lw_{opening} = Lw_{source} + 10Log[S_W[(1 - \alpha_{avg})/(S_M \cdot \alpha_{avg}) + 1/(4S_W + 4\pi l^2)]]$$

Where

$Lw_{opening}$ is the sound power incident on the opening dB.

Lw_{source} is the sound power emitted by the source in dB.

S_W is the area of the common wall in ft² (m²)

α_{avg} is the average absorption coefficient of the surfaces of the source room

S_M is the surface area of the source room in ft² (m²)

l is the distance from the source to the opening.

The average absorption coefficient can be calculated using ***Eq. 37*** and the values listed in ***Table 11 - Typical Sound Absorption Coefficients***, page 30.

Eq. 44

$$\alpha_{avg} = (S_1\alpha_1 + S_2\alpha_2)/ S_M$$

Where

S_1 is the floor and ceiling area

α_1 is the absorption coefficient of floor and ceiling

S_2 is the total wall area

α_2 is the absorption coefficient of the wall

Often a mechanical room treated with sound absorbing material to reduce the sound energy. This can be accounted for by using ***Eq. 38***.

Eq. 45

$$\alpha_a = [PC \cdot \alpha_3 + (100\text{-}PC) \cdot \alpha_{avg}]/ 100$$

Where

PC is the percent of total room surface area covered by sound absorbing material

α_3 is the absorption coefficient of the sound absorbing material

Once the sound power is known in the ceiling plenum, then we can account for the attenuation for the ceiling plenum. There is minimal end reflection affect in this application.

Return Air Path Example

Figure 30 - Central System Duct Plan, page 59, shows a return air opening above the conference room from the ceiling plenum to the mechanical room. Calculate the sound pressure in the conference room due to the return air path.

The discharge sound power level for the SWP unit is:

Band	63	125	250	500	1000	2000	4000	8000
Lw	85	81	74	85	84	90	84	77

Calculate α_{avg} using ***Eq. 37*** and 125 Hz as an example:

$= (S_1\alpha_1 + S_2\alpha_2)/ S_M = [(840 \text{ ft2})(0.36) + (600\text{ft2})(0.01)]/1440 \text{ ft2} = 0.20$

α_{avg}	0.13	0.20	0.24	0.17	0.17	0.22	0.14	0.12

Calculate the sound power incident on the return air opening using ***Eq. 36*** and 125 Hz as an example:

$Lw_{opening} = Lw_{source} + 10\text{Log}[S_W[(1 – \alpha_{avg})/(S_M\cdot \alpha_{avg}) + 1/(4S_W + 4\pi l^2)]]$

$= 81 + 10 \text{ Log}[150[(1\text{-}0.21)/(1300\cdot 0.21) + 1/(4\cdot 150 +4\pi 10^2)]] = 80 \text{ dB}$

Lw	86	80	72	85	84	89	84	78

Subtract the ceiling plenum effect:

Ceiling	3	6	8	10	16	21	36	21
Lw	83	74	64	75	68	68	48	57

Estimate the space sound pressure level using the Schultz equation. The conference room is 18 x 25 x 8 ft high with one return air opening. Use 5 ft as the distance from the source.

Calculate Room Effect using 125 Hz as an example:

RE = 10Log(5) + 5Log(3600) + 3Log(125) – 10Log (1) -25

RE	5	6	7	8	9	10	11	11
Lp	78	68	57	67	59	58	37	46
NC 35	60	52	45	40	36	34	33	32
Req'd IL	18	16	12	27	29	24	4	14

A significant amount of attenuation is required to meet the desired space sound level. A length duct or a silencer with a very small air pressure drop will be required. If two silencers are used, they must have the same air pressure drop. Below is the TL for the mechanical room wall. Requiring a return air insertion loss with the same performance as the TL for wall will work, but this is more than what is usually required and it will be difficult to do.

TL	29	29	29	30	30	30	30	30

Note poor location of the return air opening. A better location would be to locate it over the top of the bathrooms where the sound will be less offensive.

Evaluating Multiple Sound Paths and Locations

To estimate the space sound level requires logarithmically adding all of the various sound paths. Central systems generally require that several locations be considered. In the example used through this section of the Manual, both the conference room and the open office area near the main trunk were considered. The conference room is a confined space, has transmitted sound via a common wall with the mechanical room and the return air opening is above. The open area has a large amount of supply air (duct sound power) and breakout from the main supply duct.

Sound Path Evaluation Example

Consider the previous sound path examples and evaluate whether the space can meet NC 35, both in the conference room and the open office area.

List the results the sound paths for the open office area and total the sound pressure in the space using decibel addition.

Using the 125 Hz band as an example:

Total Lp = $10\text{Log}(10^{40/10} + 10^{50/10}+10^{47/10} + 10^{50/10})$
= 54 dB

	Octave Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Supply Duct Disch.Lp	26	40	39	35	34	3	31	33
Radiated Lp	34	50	47	43	43	45	52	29
Return Duct Lp	nil	nil	nil	nil	nil	nil	nil	nil
Diffuser Lp	52	47	49	48	49	44	28	24
Main Duct Breakout Lp	57	50	44	38	25	4	0	0
Total Lp	58	54	52	50	50	48	52	35
Requested NC level	60	52	45	40	36	34	33	32
Required Attenuation	0	2	7	10	14	14	19	3

List the results the sound paths for the conference room and total the sound pressure in the space using decibel addition.

	Octave Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Supply Duct Disch.Lp	24	38	42	38	35	31	27	30
Radiated Lp	38	53	48	42	44	46	54	34
Return Duct Lp	78	68	57	67	59	58	37	46
Diffuser Lp	47	40	40	37	36	30	12	7
Transmitted Lp	50	44	37	48	46	50	45	40
Main Duct Breakout Lp	nil	nil	nil	nil	nil	nil	nil	nil
Total Lp	78	68	58	67	59	59	55	48
Requested NC level	60	52	45	40	36	34	33	32
Required Attenuation	18	16	13	27	23	25	22	16

Reviewing the two spaces in the above example shows each space has it own challenges. The open office area has diffuser sound issues that silencers will not help. The diffusers need to be reselected to achieve the desired sound level.

The conference room requires attention to the return air path. A silencer or a length of return air duct is required to reduce the return air sound level. Relocating the return air path over the bathrooms is also advisable. Adding sound absorbing material to the mechanical room would help reduce both the transmitted and return air sound levels. The conference room also requires a small main supply duct silencer or duct lining to lower the supply duct sound levels.

Regenerated Noise

General

Regenerated noise is a secondary sound source usually caused by air flowing through a device. An example is noise generated by a splitter-damper. Another common source of regenerated noise is silencers.

Figure 33 - Typical Silencer Selection Showing Generated Noise

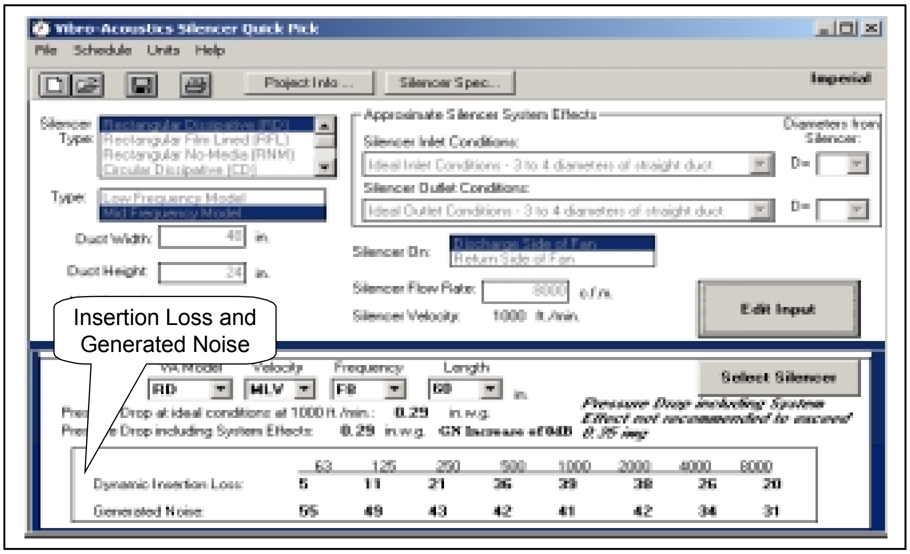


Figure 33 shows a typical silencer selection with the generated noise listed. While this silencer will attenuate 21 dB in the 250 Hz octave band, it will also add 43 dB to the same band. Whether or not this is an issue will depend on the current sound level in the duct. For example, if the duct is at 100 dB upstream on the silencer, then the silencer will attenuate the sound level to 79 dB in the 250 Hz band. It will then add 43 dB to the 79 dB, which is insignificant. The sound level will remain at 79 dB. If the sound level in the duct were 65 dB, then the silencer-generated noise would be comparable to the attenuated sound and would have to de considered.

Evaluating Regenerated Noise

There are equations and methods available to estimate regenerated noise in branches, elbows, splitters, etc. However, they are very application sensitive. The 1999 ASHRAE Applications Handbook suggests that duct related regenerated noise can be avoided by;

- Sizing ductwork or duct configurations so that air velocity is low (Refer to ***Table 24*** and ***Table 25***).
- Avoiding abrupt changes in duct cross-sectional area.
- Providing smooth transitions at duct branches, takeoffs and bends.

Table 24 - Max Recommend Duct Velocities to Achieve Specified Acoutic Design Criteria[15]

Main Duct Location	Max. Velocity (fpm)		
	Design RC	Rectangular Duct	Round Duct
	45	3500	5000
In shaft or above drywall ceiling	35	2500	3500
	25	1700	2500
	45	2500	4500
Above Suspended Acoustic Ceiling	35	1750	3000
	25	1200	2000
	45	2000	3900
Duct Located Within Occupied Space	35	1450	2600
	25	950	1700

Table 25 - Maximum Recommended "Free" Supply Outlet and Return Air Opening Velocities to Achieve Specified Acoutic Design Criteria[16]

Type of Opening	Design RC	"Free" Opening Air Velocity (fpm)
Supply Air Outlet	45	625
	40	560
	35	500
	30	425
	25	350
Return Air Opening	45	750
	40	675
	35	600
	30	500
	25	425

[15] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.
[16] ASHRAE, 1999. Applications Handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. Atlanta, Ga.

Conclusions

Acoustics can be a complex issue, but HVAC acoustics can be successfully evaluated for many typical applications. The examples shown here were generated using a computer and spreadsheet. The large number of calculations lends itself to computers. Even more helpful are software programs like McQuay's Acoustic Analyzer. This program includes defaults, databases, and typical applications to speed up the analysis. Once the user has a basic understanding of acoustics, the program can be very helpful.

More for information or assistance, please contact your local McQuay representative or contact the McQuay Applications group at www.mcquay.com

Appendix 1 - References

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2000 ASHRAE HVAC Systems and Equipment Handbook ASHRAE. Atlanta, Ga

2001 ASHRAE Fundamentals Handbook ASHRAE. Atlanta, Ga

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ARI Standard 350 – Sound Rating of Non-Ducted Indoor Air Conditioning Equipment– Air-Conditioning and Refrigeration Institute. 2000. Arlington, Va.

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Appendix 2 – HVAC Equipment Sound Measurement Stds

General

Most manufacturers can provide sound data for their equipment, so this material is not included in this manual.

The following is list most major HVAC products and the test standards they use. Pay particular attention to whether the data is sound power or sound pressure, and whether to include environmental correction factors. For an extensive analysis of manufacturer's sound data, refer to ASHRAEs ***Application of Manufacturer's Sound Data***.

Table 26 – Equipment Sound Level Rating Programs

Equipment	Test Standard	L_p or L_w	Test Location	Use Environmental Correction Factors
Air Terminal	ARI 880, ASHRAE 130P *Air Terminals*	L_w	Indoor	Yes
Air Diffuser	ARI 890 *Rating or Air Diffusers and Air Diffuser Assemblies*	L_w	Indoor	No
Fan Coil Unit	ARI 350 *Sound Rating of Non Ducted Indoor Air Conditioning Equipment*	L_w	Indoor	No
Packaged Air Handling Unit	ARI 260 *Sound Rating of Ducted Air Moving And Conditioning Equipment*	L_w	Indoor	No
Custom Air Handling Unit	AMCA 300 *Reverberant Room Method for Sound Testing of fans*:	L_w	Indoor	No
Ventilating Fan	AMCA 300 *Reverberant Room Method for Sound Testing of fans*: AMCA 301 *Methods For Calculating Fan Sound Ratings From Laboratory Test Data*	L_w	Indoor	No
Unitary Rooftop Units	ARI 370 *Sound Rating of Large Outdoor Refrigeration and Air Conditioning Equipment* ARI 260 *Sound Rating of Ducted Air Moving And Conditioning Equipment*	L_w	Indoor/ Outdoor	No
Air Cooled Chillers <10 Tons	ARI 270 *Sound Rating of Outdoor Unitary Equipment*	L_w	Outdoor	No
Air Cooled Chillers > 10 tons	ARI 370 *Sound Rating of Large Outdoor Refrigeration and Air Conditioning Equipment*	L_w	Outdoor	No
WSHPs	ARI 260 *Sound Rating of Ducted Air Moving And Conditioning Equipment*	L_w	Indoor	No
Packaged Heatpump	ARI 260 *Sound Rating of Ducted Air Moving And Conditioning Equipment*	L_w	Indoor/ Outdoor	No
Vertical Air Conditioner	ARI 260 *Sound Rating of Ducted Air Moving And Conditioning Equipment*	L_w	Indoor	No
Reciprocating Chiller	ARI 575 *Method of Measuring Machinery Sound Within Equipment Rooms*	L_p	Indoor	No
Screw Chiller	ARI 575 *Method of Measuring Machinery Sound Within Equipment Rooms*	L_p	Indoor	No
Centrifugal Chiller	ARI 575 *Method of Measuring Machinery Sound Within Equipment Rooms*	L_p	Indoor	No
Cooling Tower	CTI Code, ATC128 *Code Measuring Sound From Water-Cooling Towers*	L_p	Outdoor	No
Centrifugal Pump	ANSI/HI 9.1-9.5 *Pumps – General Guidelines*	L_p	Indoor	No
Silencers	ASTM E-477 *Standard Test Method For Measuring Acoustical and Airflow Performance of Duct Liner Materials*	Insertion Loss	N/A	No

Appendix 3 – Various Acoustic Properties of Materials

General

The following are additional tables of acoustic properties of various HVAC components. For more information refer to the *ASHRAE Applications Handbook*.

Table 27 - Insertion Loss For Rectangular Unlined Duct

Duct Size	Band							
	63	125	250	500	1000	2000	4000	8000
6 x 6	0.3	0.2	0.1	0.1	0.1	0.1	0.1	0.1
12 x 12	0.35	0.2	0.1	0.06	0.06	0.06	0.06	0.06
12 x 24	0.04	0.2	0.1	0.05	0.05	0.05	0.05	0.05
24 x 24	0.25	0.2	0.1	0.03	0.03	0.03	0.03	0.03
48 x 48	0.15	0.1	0.07	0.2	0.2	0.2	0.2	0.2
72 x 72	0.1	0.1	0.05	0.2	0.2	0.2	0.2	0.2

Table 28 - Insertion Loss For Rectangular Unlined Duct With 1 Inch Lining

Duct Size	Band							
	63	125	250	500	1000	2000	4000	8000
6 x 6	0.49	0.6	1.5	2.7	5.8	7.4	4.3	3.4
12 x 12	0.28	0.4	0.8	1.9	4	4.1	2.8	2.2
12 x 24	0.21	0.3	0.6	1.7	3.5	3.2	2.3	1.8
24 x 24	0.14	0.2	0.5	1.4	2.8	2.2	1.8	1.4
48 x 48	0.07	0.1	0.3	1	2	1.2	1.2	0.72
72 x 72	0.07	0.1	0.2	0.8	1.7	1	1	0.8

Table 29 - Insertion Loss For Rectangular Unlined Duct With 2 Inch Lining

Duct Size	Band							
	63	125	250	500	1000	2000	4000	8000
6 x 6	0.56	0.8	2.9	4.9	7.2	7.4	4.3	3.4
12 x 12	0.35	0.5	1.6	3.5	5	4.1	2.8	2.2
12 x 24	0.28	0.4	1.3	3	4.3	3.2	2.3	1.8
24 x 24	0.21	0.3	0.9	2.5	3.5	2.2	1.8	1.4
48 x 48	1.4	0.2	0.5	1.8	2.5	1.2	1.2	1
72 x 72	0.07	0.1	0.4	1.5	2.1	1	1	0.8

Table 30 - Insertion Loss For Round Unlined Duct

Duct Size	Band							
	63	125	250	500	1000	2000	4000	8000
<7	0.03	0.03	0.05	0.05	0.1	0.1	0.1	0.08
7D<15	0.03	0.03	0.03	0.05	0.07	0.07	0.07	0.05
15<D<30	0.02	0.02	0.02	0.03	0.05	0.05	0.05	0.04
30<D<60	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.016

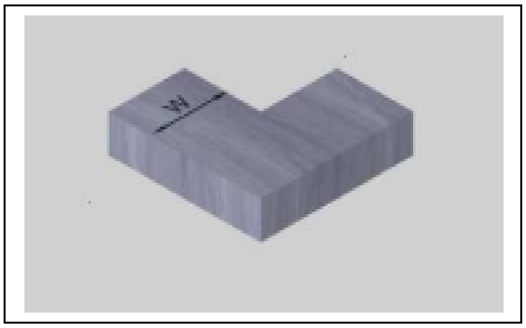
Table 31 - Insertion Loss For Round 1 Inch lined Duct

Duct Size	Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
<7	0.38	0.59	0.93	1.53	2.17	2.31	2.04	1.26
7D<15	0.19	0.42	0.77	1.43	2.14	1.79	1.34	1
15<D<30	0.03	0.19	0.49	1.2	1.46	1.04	0.74	0.74
30<D<60	0	0	0.08	0.06	0.1	0.14	0.09	0.07

Table 32 - Insertion Loss For Round 2 Inch lined Duct

Duct Size	Band							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
<7	0.56	0.8	1.37	2.25	2.17	2.31	2.04	1.26
7D<15	0.38	0.63	1.21	2.15	2.14	1.79	1.34	1
15<D<30	0.22	0.4	0.93	1.93	1.46	1.04	0.74	0.74
30<D<60	0	0	0.53	0.79	0.1	0.14	0.09	0.07

Figure 34 – Typical Elbow Fitting



The insertion loss for elbows is found by multiplying the center band frequency in kHz with the width of the elbow in inches as shown ***Figure 34 – Typical Elbow Fitting***. Using 125 Hz and a round unlined 24-inch elbow as an example:

IL = (24 x 125 Hz)/1000 = 3.0

The insertion loss is 1 dB in the 125 Hz Band.

Table 33 - Insertion Loss For Square Elbows Without Turning Vanes

	Insertion Loss	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 ≤ fw < 3.8	1	1
3.8 ≤ fw < 7.5	5	6
7.5 ≤ fw < 15	8	11
15 ≤ fw < 30	4	10
fw > 30	3	10

Table 34 - Insertion Loss For Square Elbows With Turning Vanes

	Insertion Loss	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 ≤ fw < 3.8	1	1
3.8 ≤ fw < 7.5	4	4
7.5 ≤ fw < 15	6	7
fw > 15	4	7

Table 35 - Insertion Loss For Round Elbows Without Turning Vanes

	Insertion Loss
	Unlined Elbows
fw < 1.9	0
1.9 ≤ fw < 3.8	1
3.8 ≤ fw < 7.5	2
fw > 7.5	3

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