HVAC Systems Noise Control

Course No: M06-026

Credit: 6 PDH

A. Bhatia

Continuing Education and Development, Inc.
9 Greyridge Farm Court
Stony Point, NY 10980

P: (877) 322-5800
F: (877) 322-4774

info@cedengineering.com

HVAC SYSTEMS NOISE CONTROL

Unwanted noise makes a workplace uncomfortable and less productive. When people are surveyed about workplace comfort, their most prevalent complaints involve the heating, ventilating and air-conditioning (HVAC) systems. The problems they cite most frequently, aside from temperature control, have to do with excessive noise and vibration.

There are practical and economical solutions to almost all noise problems in the built environment. To approach the solution to any specific noise problem, we need to:

- Understand the basic principles of acoustics and how noise — unwanted sound is produced, how it propagates, and how it is controlled.
- Learn the basics of noise control, and how to approach the problem from three standpoints: the source of noise, the path it travels, and the point of reception.
- Become familiar with the treatments and modifications that can be applied to HVAC system to reduce unwanted noise and vibration.

That's what architects, engineers, contractors, and building owners — anyone concerned with solving noise control problems in all types of buildings — will find in this course. It includes information on how to solve specific noise control in HVAC systems. The course is divided into 4 sections:

Section-1	The Fundamentals of Acoustics
Section-2	Noise Rating Methods
Section-3	Noise Descriptors
Section - 4	Controlling HVAC Noise and Vibration
Annexure -1	Rules of Thumb
Annexure -2	Glossary of Sound Terms

SECTION -1: THE FUNDAMENTALS OF ACOUSTICS

This section provides a brief introduction to the fundamentals of building acoustics that is necessary to understand the material in the following sections. To emphasize the simplicity of the approach, equations are kept to a minimum.

What is Sound and Noise?

Sound is a form of mechanical energy transmitted by vibration of the molecules of whatever medium the sound is passing through. Noise may be defined as "unwanted or undesired sound", which interferes with speech, concentration or sleep. To control noise or sound, we need to know a little about its fundamental properties such as:

1. Frequency (pitch)
2. Wavelength
3. Amplitude (loudness)

Once these fundamental properties are understood, we can proceed to implement effective noise control measures.

What is Sound Frequency?

The frequency or pitch of a sound wave is the number of times that its basic pattern repeats itself per second. So, a musical note characterized by a pattern of pressure variations that repeats itself 1200 times per second has a frequency of 1200 Hertz.

The frequency - cycles per second - of a sound is expressed in hertz - *Hz*. The frequency can be expressed as:

$f = 1 / T$

Where

- f = frequency (s^{-1}, Hz)
- T = time for completing one cycle (s)

Example

Calculate the time of one cycle for a 500 Hz tone.

T = 1 / (500 Hz)

= <u>0.002</u> s

***Note -** Sound and noise usually are **not** pure tones*. Pure tone is described as a simple vibration of single frequency.

The human ear can perceive sounds with frequencies ranging from 20 Hz up to 20000 Hz.

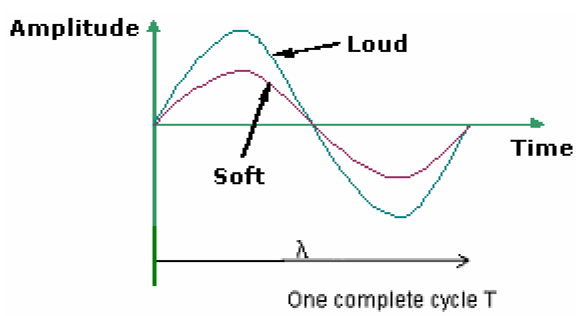
Wavelength

The wavelength of a sound wave is the distance between the start and end of a sound wave cycle or the distance between two successive sound wave pressure peaks. Numerically, it is equal to the speed of sound in the material such as air divided by the frequency of the sound wave. For example:

The wavelength of a 100 Hz tone at room temperature is 1130* ft/sec divided by 100 Hz which is equal to 11.3 ft.

*The speed of sound in air is approximately 1,130 feet per second.

***Note -** The higher the frequency the shorter shall be the wavelength.*



$\lambda = c / f$

*W*here,

- λ = wavelength (m)
- c = speed of sound (m/s) (*in air at normal atmosphere and 0^{o}C the sound of speed is 331.2 m/s)
- f = frequency (s^{-1}, Hz)

Example

Calculate the wavelength of a 500 Hz tone considering speed of sound as 331.2 m/s.

$\lambda = (331.2 \text{ m/s}) / (500 \text{ Hz})$

$= \underline{0.662}$ m

Note - The velocity is the distance moved by the sound wave per second in a fixed direction i.e. $v = f \times \lambda$

Amplitude (loudness)

The amplitude of a sound wave is the height of the wave form. It is also the maximum displacement for each air particle as it vibrates. The amplitude or loudness of a sound wave is expressed by its sound pressure level. Sounds having the same wavelength (equal frequency) may have differing loudness.

Note - *Human ears distinguish one sound from another by its loudness and pitch. Loudness is the amplitude or amount of sound energy (dB) reaching our ears. Pitch is the speed of the vibrations or frequency used to identify the source of a sound. However, both the loudness and pitch may vary depending upon where we are located relative to the sound and the surrounding environment.*

Sound Power and Sound Pressure

The difference between sound power and sound pressure is critical to the understanding of acoustics.

Sound power is the amount of acoustical power a source radiates and is measured in watts. Sound power is a fixed property of a machine irrespective of the distance and environment. HVAC equipment generates sound power.

Sound pressure is related to how loud the sound is perceived to be to a human ear in a particular environment. It depends upon the distance from the source as well as the acoustical environment of the listener (room size, construction materials, reflecting surfaces, etc.). Thus a particular noise source would be measured as producing different sound pressures in different spots. Theoretically, the sound pressure "p" is the force of sound on a surface area perpendicular to the direction of the sound. The SI-units for the Sound Pressure are N/m^2 or Pa.

The difference between the sound power that the unit generates and the sound pressure that the ear hears is caused by the reduction (attenuation) of the sound along the path to the room, and the reduction within the room (or "room effect") caused by the room layout, construction and furnishings.

How Is Sound Measured?

The term used to describe sound is “decibel” (dB) – which is represented in Logarithmic scale. Each 10 dB increase represents a tenfold increase in signal amplitude, while each 10 dB reduction represents a tenfold reduction.

What is logarithmic scale?

For instance, suppose we have two loudspeakers, the first playing a sound with power W_1, and another playing a louder version of the same sound with power W_2, but everything else (how far away, frequency) kept the same.

The difference in decibels between the two is defined to be:

- $10 \log (W_2/W_1)$ dB where the log is to base 10

If the second produces twice as much power than the first, the difference in dB is:

- $10 \log (W_2/W_1) = 10 \log 2 = 3$ dB

If the second had 10 times the power of the first, the difference in dB would be:

- $10 \log (W_2/W_1) = 10 \log 10 = 10$ dB

If the second had a million times the power of the first, the difference in dB would be:

- $10 \log (W_2/W_1) = 10 \log 1000000 = 60$ dB

This example shows one feature of decibel scales that is useful in discussing sound: they can describe very big ratios using numbers of modest size. But note that the decibel describes a *ratio*: so far we have not said what power either of the speakers radiates, only the ratio of powers.

A decibel scale can be used for many purposes, but it is meaningful for a specific purpose only when the reference level is defined. In acoustical work, two reference levels are commonly used:1) Sound Power Level (Lw) and 2) Sound Pressure Level (Lp)

Sound Power Level (Lw) is a measure of sound energy output expressed in decibels compared to a reference unit of 10^{-12} Watts. It is calculated by the formula:

$Lw = 10 \log_{10} (W_{source} / W_{ref})$

Where:

- Lw is sound power level in decibel (dB)
- W_{ref} is 10^{-12} W;
- W_{source} is sound power in W

If sound power increases by a factor of 2, this is equivalent to a 3 dB increase.

The use of sound power levels is the desirable method for equipment suppliers to furnish information because sound pressure levels can then be computed in octave bands for any desired reflective environment or space.

Sound Pressure Level (Lp)

The lowest sound pressure possible to hear (the threshold of hearing) is approximately 2 x 10^{-5} Pa (20 micro Pascal) and is given the value of 0 dB.

The threshold of pain in the ear corresponds to pressure fluctuations of about 200 Pa. This second value is ten million times the first.

These unwieldy numbers are converted to more convenient ones using a logarithmic scale (or the decibel scale) related to this lowest human hearable sound "p_{ref}" - 2 * 10^{-5} Pa, 0 dB.

Sound pressure level is the pressure of sound in relation to a fixed reference and is described by a logarithmic comparison of sound pressure output by a source to a reference sound source, P_0 (2 x 10^{-5} Pa)

$Lp = 10 \log_{10} (P^2 / P_{ref}^2)$

Where:

- Lp = Sound Pressure Level (dB)
- P_{ref} = 2 x 10^{-5} – Reference Sound Pressure (Pa);
- P = sound pressure (Pa)

Doubling the Sound Pressure raises the Sound Pressure Level with 6 dB (20 log (2)).

What does 0 dB mean?

This level occurs when the measured intensity is equal to the reference level. i.e., it is the sound level corresponding to 0.02 mPa. In this case we have

Sound level = 20 log ($P_{measured}/P_{reference}$) = 20 log 1 = 0 dB

So 0 dB does not mean no sound, it means a sound level where the sound pressure is equal to that of the reference level. This is a small pressure, but not zero. It is also possible to have negative sound levels: - 20 dB would mean a sound with pressure 10 times smaller than the reference pressure, i.e. 2 micro-Pascal.

Directionality of Sound Waves

Noise levels in rooms vary with distance from the source and with the properties of the room.

In an outdoor situation, sound levels decrease as one moves away from the source. *Sound pressures decrease inversely proportional to the distance from the source.* This is true for a source that radiates sound approximately equally in all directions and where there is only one path from the source to the receiver.

In a room, there are a very large number of possible paths from the source to the receiver, involving various reflections off the room boundaries; the combination of all these paths determines how sound behaves in the room. As a result, sound levels in a room do not continue to decrease with increasing distance from the source for all distances.

What measurements are useful?

Most of the sounds we hear are a combination of many different frequencies. Healthy young human beings normally hear frequencies as low as about 20 Hz and as high as 20,000 Hz. The human ear however is most sensitive to sound in the frequency range 1000Hz to 4000 Hz than to sound at very low or high frequencies. *This means that the noise at high or low frequencies will not be as annoying as it would be when its energy is concentrated in the middle frequencies. A higher sound pressure is therefore acceptable at lower and higher frequencies. This knowledge is important in acoustic design and sound measurement.*

The engineer must clearly distinguish and understand the difference between sound power level and sound pressure level. Even though both are expressed in dB, there is no outright conversion between sound power level and sound pressure level. A constant sound power output will result in significantly different sound pressures levels when the source is placed in different environments.

It is much more useful to know the amount of sound power produced by a source than to know the sound pressure measured under some particular condition; one should strive to obtain sound power ratings of all potential noise sources at the design stage. The acoustic engineer must take this into account when specifying noise levels.

Inverse Square Law

Most measuring instruments measure the average of the acoustic pressure over the time period of the wave.Theoretically, the power in a sound wave goes as the square of the pressure (similar to what electrical power goes as the square of the voltage). The log of the square of x is just 2 log x, so this introduces a factor of 2 when we convert to decibels for pressures. The difference in sound pressure level between two sounds with p_1 and p_2 is therefore:

$Lp = 10 \log (p_2^2/p_1^2)$ dB or

$Lp = 20 \log (p_2/p_1)$ dB

The sound levels decrease 6 dB for each doubling of distance from the source. There are however lots of things that have to be taken into account and understood before any measurement would be meaningful.

Octave bands

Typical sources in buildings emit sound at many frequencies. Because sound occurs over a range of frequencies, it is considerably more difficult to measure than temperature or pressure. The sound must be measured at each frequency in order to understand how it will be perceived in a particular environment. The human ear can perceive sounds at frequencies ranging from 20 to 16,000 Hz, whereas, HVAC system designers generally focus on sounds in the frequencies between 45 and 11,200 Hz. Despite this reduced range, measuring a sound at each frequency would result in 11,156 data points.

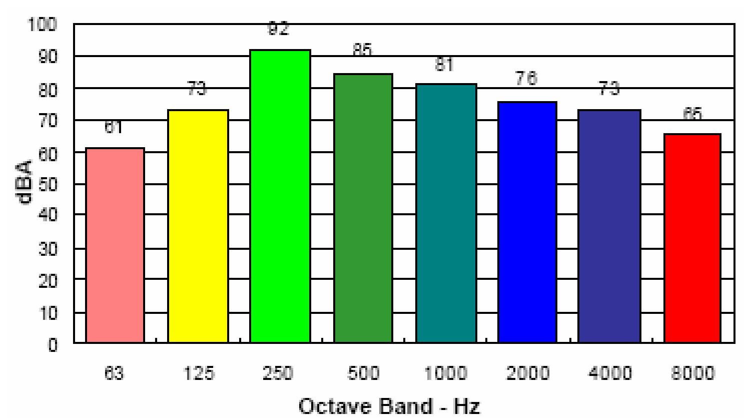
For some types of analyses, it is advantageous to measure and display the sound at each frequency over the entire range of frequencies being studied. This is called a full-spectrum analysis.

To make the amount of data more manageable, this range of frequencies is typically divided into smaller ranges called **octave bands**. Each octave band is defined such that the highest frequency in the band is two times the lowest frequency. The octave band is identified by its center frequency, which is calculated by taking the square root of the product of the lowest and highest frequencies in the band.

$$\text{Center frequency} = \sqrt{\text{Lowest frequency x highest frequency}}$$

The result is that this frequency range (45 to 11,200 Hz) is separated into eight octave bands with center frequencies of 63, 125, 250, 500, 1,000, 2,000, 4,000, and 8,000 Hz. For example, sounds that occur at the frequencies between 90 Hz and 180 Hz are grouped together in the 125 Hz octave band.

Octave bands compress the range of frequencies between the upper and lower ends of the band into a single value. A unit “Octave” defines the concept of these frequency ranges and is the interval between two points where the frequency at the second point is twice the frequency of the first.



SOUND PRESSURE LEVELS BY OCTAVE BANDS

Unfortunately, octave bands do not indicate that the human ear hears a difference between an octave that contains a tone and one that does not, even when the overall magnitude of both octaves is identical. Therefore, the process of logarithmically summing sound measurements into octave bands, though practical, sacrifices valuable information about the “character” of the sound.

Middle ground between octave-band analysis and full-spectrum analysis is provided by one-third octave-band analysis. One-third octave bands divide the full octaves into thirds. The upper cutoff frequency of each third octave is greater than the lower cutoff frequency by a factor of the cube root of two (approximately 1.2599). If tones are contained in the broadband sound, they will be more readily apparent in the third octaves.

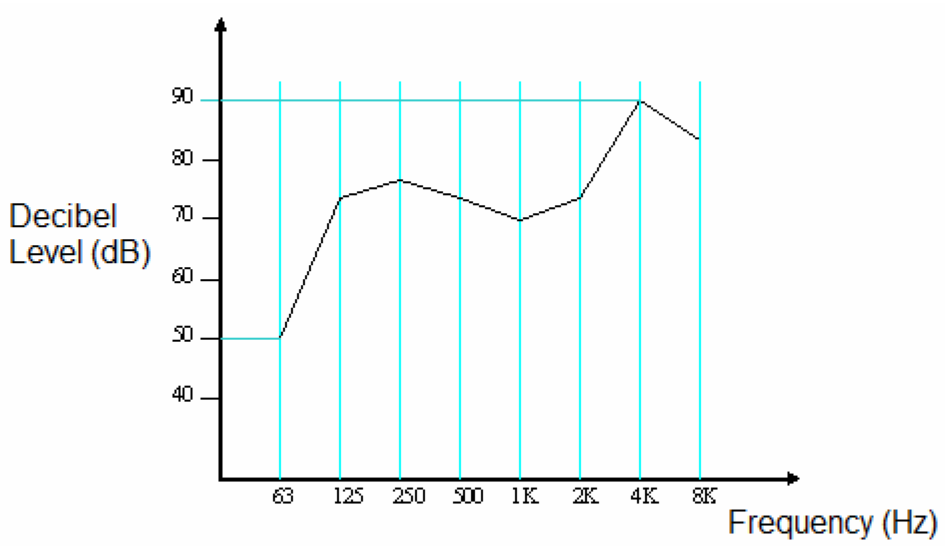
The use of octave bands is usually sufficient for rating the acoustical environment in a given space. One-third octave bands are, however, more useful for product development and troubleshooting acoustical problems.

Note - The measurement in building acoustics spectra are usually made in one-third octave bands from about 100 Hz to 4000 Hz.

Sound Measurements

Sound is measured in decibels with the use of a Sound Level Meter. The sound level meter responds to sound in approximately the same way as the human ear but it gives an objective measurement of sound level. The sound to be measured is converted into an electrical signal by a microphone. Normally condenser microphones are used for precision grade instruments. Since the signal is quite small it is necessary to amplify it and then convert for display on the meter.

When measuring the sound level from a source the meter should be held at a distance of 1 meter from the source. The sound level in decibels A- scale can be measured at each centre frequency using the filters and a graph can then be drawn for example as shown below. This gives an indication of sound levels at different frequencies. This can be useful information and can be used to design appropriate attenuation and acoustic treatment at particular frequencies.



Noise Output from a Machnine (dB)

It will be seen from the above graph that the maximum noise output of 90 dB occurs at 4K (4000) Hertz. Also the lowest noise level of 50 dB occurs at a frequency of 63 Hz.

If sound attenuation is required for the machine in this example then the engineer would concentrate at the high frequency noise spectrum (4K Hz). The insulation or proposed attenuation system could be tailored especially for high frequency attenuation rather than low frequency attenuation.

Is the Sound Pressures Levels additive from sounds emitting from various sources?

No

The decibel values of sounds can not be added or subtracted using the normal arithmetic rules. Instead, decibel values must be converted back into absolute units of power (Watts), when they can be added or subtracted directly before re-converting back into decibels.

If N sources generating the same sound pressure level are combined, the overall sound pressure level will be increased by 10 log N dB. Two sounds with average levels of 60 dB, for example, will together create a sound pressure level of 60 + 10 log 2 = 63 dB and not 120 dB. The greater the difference in noise level between two noise sources, the less effect there is on the combined level. Where the difference between two sources is more than 6 dB, the combined level will be less than 1 dB higher than the louder source alone. The table below shows the simple approximation method of obtaining the equivalent combined noise level, when the dB difference between the two levels is known.

Difference between two levels, dB	0	1	2	3	4	5	6	7	8	9	10 & more
Quantity to be added to the higher level	3	2.5	2	2	1.5	1.5	1	1	.5	.5	0

Examples of ways two sound levels contribute to each other based upon the difference between them

Procedure:

1) Select the highest level

2) Subtract the next highest level from the highest

3) Using the difference, go to the chart and find the addition to the highest level

4) Add this to the highest level

5) The result is the logarithmic sum of the two levels

Example # 1

If a fan has a sound level of 50dB and another, larger fan with a sound level of 55dB is added, the difference between the two is 5dB, so 1dB is added to the higher figure, giving a combined level of 56dB.

Example # 2

Combining Two Sound Spectrums

Band No.	1	2	3	4	5	6	7	8
Spectrum A	82	80	73	70	69	66	60	53
Spectrum B	79	77	71	68	67	64	59	52
Absolute Difference	3	3	2	2	2	2	1	1
Added to the highest	2	2	2	2	2	2	2.5	2.5

Example # 3

Combining an octave band spectrum into a single number

Band No.	1	2	3	4	5	6	7	8
Spectrum A	81	80	76	70	69	64	61	57

Procedure:

The highest number is 81, the next higher is 80; the difference is 1, the adder is 2.5, the combined number is 83.5.

The next higher number is 76, the difference is 7.5, the adder is .5, and the combined sum is 84

The next higher number is 70, the difference is 14, the adder is 0, and therefore the combined number for this spectrum is 84 dB.

If two sound levels are identical, the combined sound is three dB higher than either. If the difference is 10 dB, the highest sound level completely dominates and there is no contribution by the lower sound level.

Equivalent Noise Level with "N" Sources Generating the Same Sound Pressure Level

The table below shows the simple approximation method of obtaining the equivalent combined noise level, when the number of sources having same sound level are put in the same room (i.e. difference between two sound levels is zero).

Number of Sources	Increase in Sound Power Level (dB)	Increase in Sound Pressure Level (dB)
2	3	6
3	4.8	9.6
4	6	12
5	7	14
10	10	20
15	11.8	23.6
20	13	26

As an example, if one fan has a sound level of 50dB, and another similar fan is added, the combined sound level of the two fans will be 53dB. If two more fans are added, i.e.

the sound source is doubled again; the resultant sound level from all four fans will be 56dB.

Average Ability to Perceive Changes in Noise Levels

Studies have shown that a 3 dB A increase is barely perceptible to the human ear, whereas a change of 5 dB A is readily perceptible. The average ability of an individual to perceive changes in noise levels is well documented (see table below). These guidelines permit direct estimation of an individual's probable perception of changes in noise levels. *As a general rule, an increase or decrease of 10 dBA in noise level is perceived by an observer to be a doubling or halving of the sound, respectively.*

Average Ability to Perceive Changes in Noise Levels

Change (dB A)	**Human Perception of Sound**
2-3	Barely perceptible
5	Readily noticeable
10	A double or halving of the loudness of sound
20	A dramatic change
40	Difference between a faintly audible sound and very loud sound

Sound Intensity

Sound Intensity is the Acoustic or Sound Power of a sound (W) per unit area in relation to a fixed reference. The SI-units for Sound Intensity are W/m^2.

The Sound Intensity Level can be expressed as:

$L_I = 10 \log (I / I_{ref})$

Where:

- L_I = sound intensity level (dB)
- I = sound intensity (W/m^2)

- $I_{ref} = 10^{-12}$ ----- reference sound intensity (W/m^2)

The logarithmic sound intensity level scale matches the human sense of hearing.

Doubling the intensity increases the sound level with 3 dB (10 log (2)).

Example

The difference in intensity of 10^{-8} watts/m^2 and 10^{-4} watts/m^2 (10,000 units) can be calculated in decibels as:

$\Delta L_I = 10 \log ((10^{-4} \text{ watts/m}^2) / (10^{-12} \text{ watts/m}^2))$

$\quad - 10 \log ((10^{-8} \text{ watts/m}^2) / (10^{-12} \text{ watts/m}^2))$

$= \underline{40}$ dB

Increasing the sound intensity by a factor of:

- 10 raises its level by 10 dB
- 100 raises its level by 20 dB
- 1,000 raises its level by 30 dB
- 10,000 raises its level by 40 dB and so on

***Note!** The sound intensity level may be difficult to measure, it is common to use "sound pressure level" measured in decibels instead.*

Sound Intensity and Sound Pressure

The connection between Sound Intensity and Sound Pressure can be expressed as:

$I = p^2 / \rho c$

Where:

- p = sound pressure (Pa)
- ρ = 1.2 = density of air (kg/m^3) at 20^{o}C
- c = 340 – speed of sound (m/s)

Sound Power, Intensity and Distance to Source

The sound intensity decreases with distance to source. Intensity and distance can be expressed as:

$I = Lw / 4 \pi r^2$

Where:

- *Lw* = sound power (W)
- π = 3.14
- *r* = radius or distance from source (m)

To estimate the effect of distance from a sound source the rule of thumb calculation is very simple. Sound normally weakens by 6dB each time the distance from the sound source is doubled, i.e. a sound level measured as 60dB at 1 meter will be 54dB at 2 meters and 48 at 4 meters.

Smaller fan sound levels are sometimes measured at 1m, whereas the industry standard is becoming standardized at 3m, for comparison purposes. The difference between 1m and 3m is therefore 9dB. (41dB@3m) and 50dB@1m are in effect the same).

Note! The published noise levels are measured in dBA at a distance of 6 feet, the industry standard.

Noise Control - Why do we need it?

1. Government Regulations enforced by Environment Protection Agency (EPA), Occupation Safety and Health Administration (OSHA), Federal Housing Administration (FHA), US Department of Housing and Urban development (HUD) and other State and Local laws.
2. Insurance carriers apply pressure to prevent hearing loss claims filed under workman's compensation laws.
3. Prolonged exposure to loud environment cuts down productivity; creates stress and can lead to accidents.
4. Unwanted noise is nuisance. The most common problem in a room is too much echo or reverberation. Too much echo can garble speech clarity and intelligibility.

SECTION -2 NOISE RATING METHODS

The human ear interprets sound in terms of loudness and pitch, while electronic sound measuring equipment interprets sound in terms of pressure and frequency. This creates a problem for measuring the impact of each sound on human hearing since each frequency will be perceived differently by the ear. It is therefore important to equate and adjust sound pressure and frequency to sound levels of human hearing. The goal has been to develop a system of single-number descriptors to express both the intensity and quality of a sound.

Although a number of procedures have been developed for rating the acceptability of noise in buildings, three are most common; 1) A-weighted sound levels, 2) the noise criteria (NC) and 3) the room criteria (RC) rating procedure.

A-weighted Measurement, dB (A)

The simplest method is A-weighted Measurement, dB (A) that combines octave-band sound data into a single-number descriptor. As stated earlier, the sensitivity of the human ear to sound depends on the frequency or pitch of the sound. People hear some frequencies better than others. If a person hears two sounds of the same sound pressure but different frequencies, one sound may appear louder than the other. This occurs because people hear high frequency noise much better than low frequency noise.

Sound Level Meters are programmed to account for the human ear's sensitivity to certain frequencies. The meter, measures the sound pressures in all the different frequencies, corrects them for the human ears and then presents us with a single figure representation of the noise – dB (A).

The letter "A" indicates that the sound has been filtered to reduce the strength of very low and very high frequency sounds, much as the human ear does. The table below summarizes common noise sources with their associated typical dB A.

A – Weighted Sound Level in Decibels (dB A)

Weighted	Overall Level	Noise Environment
120	Uncomfortably loud (32 times as loud as 70 dBA)	Military Jet airplane takeoff at 50 feet

100	Very loud (8 times as loud as 70 dBA)	• Jet flyover at 1000 feet • Locomotive pass by at 100 feet
80	Very loud (2 times as loud as 70 dBA)	• Propeller plane flyover at 1000 feet • Diesel truck 40 mph at 50 feet
70	Moderately loud	• Freeway at 50 feet from pavement edge morning hours • Vacuum cleaner (indoor)
60	Relatively quiet (1/2 as loud as 70 dBA)	• Air-conditioning unit at 100 feet • Dishwasher at 10 feet (indoor)
50	Quiet (1/4 as loud as 70 dBA)	• Large transformers • Small private office (indoor)
40	Very quiet (1/8 as loud as 70 dBA)	• Birds calls • Lowest limit of urban ambient sound
10	Extremely quiet (1/64 as loud as 70 dBA)	Just audible
0		Threshold of hearing

The main advantages with the A -weighting are as follows:

1. It is adapted to the response of the human ear to sound
2. It is possible to measure easily with low cost instruments

“A-weighted” measurement is most commonly used for outdoor sound sources, such as highway traffic or outdoor equipment. For an AHU, typically only the outside air, exhaust air or casing radiated sound components would be rated in terms of dBA, not the ducted sound components. Sound power is not typically rated in dBA because the A-weighting network is intended to rate the annoyance or “effect” of outdoor noise sources, and it is a characteristic of the noise source, not the effect of the noise.

Limitations

Speech happens predominantly in the mid-frequencies. But if you take readings in dBA there may be a huge low frequency component happening that you don't know about, because dBA is a weighted average. Therefore measuring average spectra, such as dBA, typically doesn't reveal the complete spectral balance of sound and do not correlate well with the annoyance caused by noises. Different sounds can receive the same rating but retain dissimilar subjective qualities. A-weighted sounds are best for a comparison between sounds that are similar but differ in level, such as comparing the loudness between two different makes of a fan. Therefore, the A-weighted sound level is not the best tool for measuring HVAC systems as a whole, but it is better used for measuring a single component like a fan.

dB (A) and dB (C)

Similar to A-weighted measurement is C-weighted measurement denoted as dB (C). The letters "A" or "C" following the abbreviation "dB" designate a frequency-response function that filters the sounds that are picked up by the microphone in the sound level meter.

- The A weighting filters out the low frequencies and slightly emphasizes the upper middle frequencies around 2-3 kHz. A-weighting, which is most appropriately used for low-volume (or quiet) sound levels, best approximates human response to sound in the range where no hearing protection is needed.
- By comparison C weighting is almost unweighted, or no filtering at all. C-weighting is used for high volume (or loud) sound levels where the response of the ear is relatively flat.

A-weighting is used to measure hearing risk and for compliance with OSHA regulations that specify permissible noise exposures in terms of a time-weighted average sound level or daily noise dose. C-weighting is used in conjunction with A - weighting for certain computations involving computation of hearing protector attenuation.

Noise Criterion – NC values

In acoustic analysis, the air conditioning designer is concerned with what the human ear perceives. The "loudness" of the sound is based not so much on absolute numbers, as on the human ear's perception. The human ear responds differently to loudness at different frequencies.

Noise criteria (NC) curves plotted on a scale of Frequencies (Octaves) vs. Decibels (Loudness in dB); represent approximately equal loudness levels to the human ear. These curves define the limits that the octave band spectrum must not exceed. For instance, to achieve NC-35 rating, the sound spectrum must be lower than the curve in every octave band.

NC is particularly used in Heating, Ventilating and Air Conditioning (HVAC) work*, with an NC-35 being the most common requirement*.

NC Level – Methodology

The following steps describe how to calculate an NC rating:

1. Plot the octave-band sound-pressure levels on the NC chart.
2. The highest curve crossed by the plotted data determines the NC rating.

The sound is measured in octave bands from 63 to 8000 Hz and plotted against a set of curves. The point where the spectrum touches the highest point tangent to NC curve, determines the NC rating for the spectrum. The curves for a given rating allow less sound with increasing frequency. The rating numbers correspond to the curve level in the 1000-2000 Hz octaves. Two spectra can have the same NC value but quite different shapes. Although the relationship between the A-weighted level and the NC number will depend on the actual spectrum, the requirements are that the sound pressures measured at each octave band must be below the specified NC curve (within a 2dB tolerance) if they are to meet the NC rating.

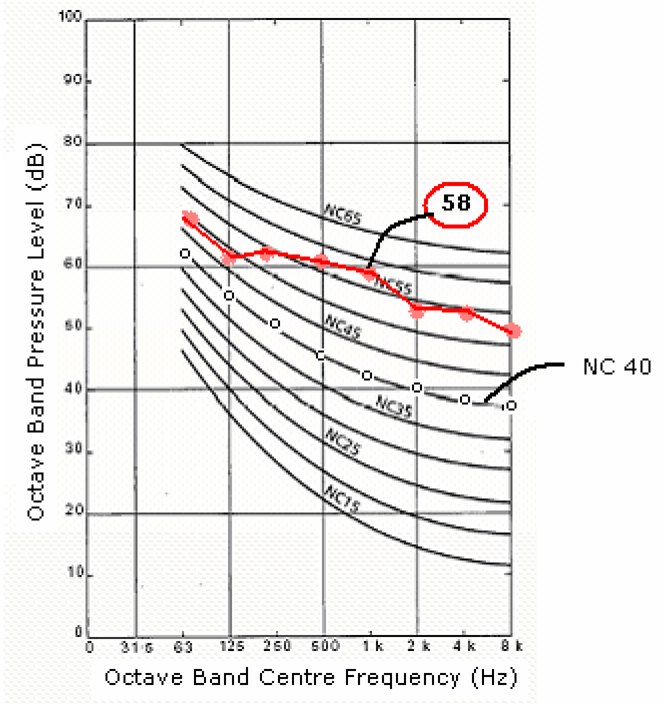
Example

Calculate the NC level and the sound insulation needed for an office with a nearby noise source as follows:

Noise Source Spectrum								
Octave Band Centre Frequency (Hz)	63	125	250	500	1K	2K	4K	8K
Measured Sound Levels (dB)	**69**	**63**	**64**	**62**	**58**	**57**	**55**	**51**

The noise source is plotted on the NC curves below.

The closest fit NC curve to the noise source is NC 58.



Noise Criteria (NC) Curves

The generally accepted sound level for office spaces is NC35 to NC 45, therefore if say NC 40 is chosen, then the amount of insulation at each frequency can be calculated.

Noise Source Spectrum								
Octave Band Centre Frequency (Hz)	63	125	250	500	1K	2K	4K	8K
Measured Sound Levels (dB)	69	63	64	62	58	57	55	51
NC 40 from Figure above (dB)	63	56	50	45	42	40	38	37
Insulation required (dB)	69 – 63 = 6	7	14	17	16	17	17	14

<u>Limitations</u>

While NC is a great improvement over dB (A) ratings, one needs to keep its limitations in mind.

It gives little indication of the sound character (quality). Two different sounding noise spectra with different acceptance from people may be rated at the same NC level. For example, equipment with a dominant single low frequency peak will sound much more offensive than equipment with a spectrum that more closely matches the NC curve. For HVAC equipment especially package and self contained units, it is important to compare the noise generated in the first (63 Hz) and second (125 Hz) octave bands. Higher noise in these octave bands can cause a rumble in the conditioned space.

NC curves do not account for sound at very low frequencies - the 16 Hz and 31.5 Hz octave bands. Although HVAC equipment manufacturers typically do not provide data in these bands (because it is very difficult to obtain reliably), these octave bands do affect the acoustical comfort of the occupied space. Higher frequency sound, which is much easier to attenuate with acoustic insulation is reduced significantly and is unable to mask unwanted sound. Low frequency noise is attenuated much less, causing the annoying rumble.

NC curves are not ideal background spectra to be sought after in rooms to guarantee occupant satisfaction but are primarily a method of rating the noise level. There is no generally accepted method of rating the subjective acceptability of the spectral and time-varying characteristics of ambient sounds. In fact, ambient noise having exactly an NC spectrum is likely to be described as both rumbly and hissy, and will probably cause some annoyance. NC ratings are often used to establish maximum noise levels associated with HVAC machinery.

Relationship between A-weighted and NC

Although the relationship between the A-weighted level and the NC number will depend on the actual spectrum, usually the A-weighted level is about 7 dB greater than the NC value.

Room Criteria (RC)

The room criterion (RC) method is currently the most favored method for determining sound levels of HVAC systems noise. The RC system overcomes the shortcomings of the NC system to some extent. The major difference is that RC curves give an additional indication of sound character. 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.

In addition, the RC rating consists of two descriptors. The first descriptor is a number representing the speech interference level (SIL) of the sound. The second descriptor is a letter denoting the character of the sound as a subjective observer might describe it.

- The letter N identifies a neutral or balanced spectrum - this sound will not have any identity with frequency and will sound bland.
- 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.
- The letter V represents “vibration”

Note the Facts below:

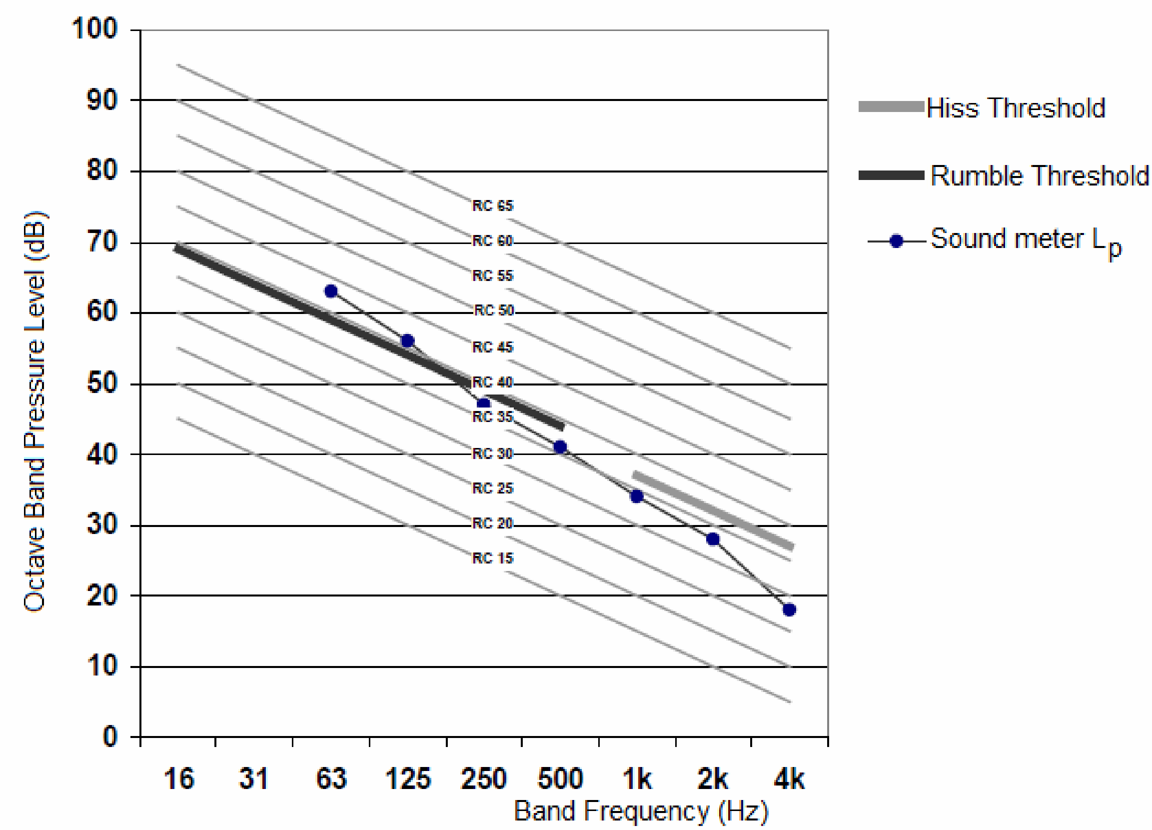
a. RC measures background noise in the building over the frequency range of 16-4000 Hz. This rating system requires determination of the mid-frequency average level and determining the perceived balance between high-frequency (HF) sound and low-frequency (LF) sound. Studies show that an RC between 35 and 45 will usually provide speech privacy in open-plan offices, while a value below 35 does not. Above RC–45, the sound is likely to interfere with speech communication.

b. The arithmetic average of the sound levels in the 500-, 1,000- and 2,000-Hz octave bands is 44.6 dB; therefore, the RC 45 curve is selected as the reference for spectrum quality evaluation.

c. The sound is rated rumbly, if any of the bands from 31.5 Hz through 250 Hz are 5 dB or more higher than the RC curve.

d. The sound is rated as a hiss, if the 2,000- of 4,000-Hz bands are greater than 3 dB above the RC curve.

RC method is a preferred rating system advocated by American Society of Heating, Refrigerating and Air-Conditioning Engineers and can be further looked at in the ASHRAE application handbook.



Room Criteria (RC) Curves

Summarizing….

When selecting a rating method it is important to consider how the rating will be used.

All three dB (A), NC and RC methods consider speech interference but A-weighted method is primarily used for outdoor noise environment.

While NC doesn't provide any quality assessment and don't evaluate low frequency rumble, RC method evaluates sound quality and provides some diagnostic capability.

NC method is used to rate components such as air terminal diffusers; RC method is not recommended to be used to rate components.

The RC method is considered to be the better measure of sound between the two methods and is slowly replacing the NC method as a means of analyzing sound.

SECTION -3 NOISE DESCRIPTORS

Acoustical design for buildings requires quantitative information about acoustical products, materials and systems so that recommended design criteria can be met. For most noise control work in buildings, the two most important acoustical properties of the materials and systems used are "sound-absorption" and "reverberation". The term sound absorption is most often used to describe single space acoustic applications where reverberation of noise or echo is the main problem. Projects such as indoor arenas, sporting facilities, cinemas, theatres and studios need to be designed to minimize this type of reverberant noise.

This section provides information on the common noise descriptors.

Reverberation

The reverberation time (RT), widely used as a design parameter in architectural acoustics, is the time required for a loud sound to fade away to inaudibility after the source has been turned off. More precisely, it indicates how long it takes until the sound pressure level in a room is decreased by 60 dB after the sound source is terminated.

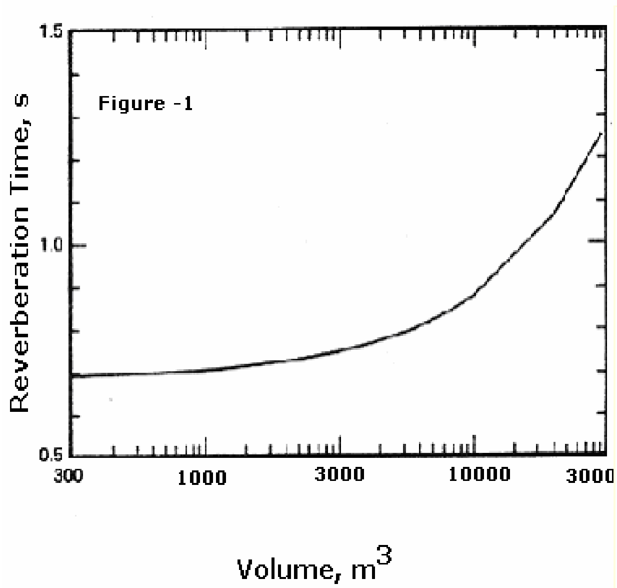
Excessively reverberant rooms are usually noisy and speech communication may be interfered with. The reverberant sound level and the RT in the room can be reduced through the installation of appropriate sound absorbing materials.

Factors Influencing Reverberation Time

Reverberation time is affected by the size of the space and the amount of reflective or absorptive surfaces within the space.

- **Size of space -** Reverberation time is directly related to the room volume. In general, larger spaces have longer reverberation times than smaller spaces. Therefore, a large space will require more absorption to achieve the same reverberation time as a smaller space.
- **Amount of absorptive surfaces -**Highly absorbent surfaces shorten the reverberation time whereas highly reflective surfaces lengthen the reverberation time.

Figure 1 (left) below gives optimum reverberation time for speech versus room volume. *As the room volume increases, the optimum reverberation time also increases.* Since this contour indicates a unique optimum reverberation time for each room volume, it is possible to calculate the related optimum total absorption necessary for each room volume.



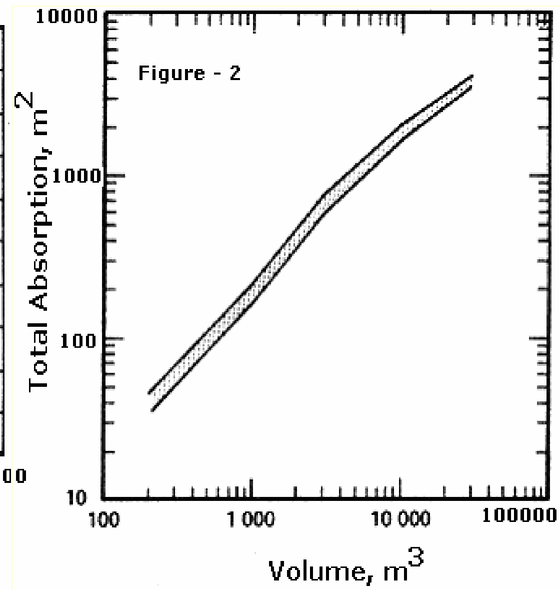


Figure 2 (right) above shows the resulting range of acceptable total room absorption (in metric sabins) versus room volume (in cubic metros) that would lead to the desired optimum reverberation time within +0.1 second. *Designing for a total room absorption as close as possible to the middle of this range should lead to nearly optimum conditions for speech.*

How to calculate Reverberation Time?

There are several formulas for calculating reverberation time; the most common formula is the Sabin Formula. The formula is based on the volume of the space and the total amount of absorption within a space.

The Reverberation Time - T_a - can be expressed as:

$T_a = 0.16\ V / A$ (1)

Where

- 0.16 is an empirical constant
- T_a = reverberation time (s)
- V = room volume (m^3)

A = Total sound absorption of the room (room surface area *x* average absorption coefficient + absorption of furniture/people, m^2).

The absorption of a room is obtained by summing the absorption of all the surfaces in the room, i.e. walls, ceiling, floor and all the furniture in the room. The total amount of absorption within a space is referred to as sabins.

What is Sabin?

The sound absorption for a sample of material or an object is measured in sabins or metric sabins. One sabin maybe thought of as the absorption of unit area (1 m^2 or 1 ft^2) of a surface that has an absorption coefficient of 1.0 (100%). When areas are measured in square meters, the term metric sabin is used. The absorption for a surface can be found by multiplying its area by its absorption coefficient. Thus for a material with an absorption coefficient of 0.5, 10 ft^2 of this material has a sound absorption of 5 sabins and 100 m^2, of 50 metric sabins.

BASIC CLASSES OF MATERIALS

Two classes of materials are often used for engineering noise control:

1. Sound absorption materials are the materials that absorb sound; reduce reflections from surfaces and decrease reverberation within spaces. Sound absorbing materials are usually fibrous, lightweight and porous.
2. Sound partitions or barriers reduce sound transmission between adjacent spaces. These materials are usually non-porous and a good reflector of sound.

A sound barrier material is a poor absorber of sound and an absorbent material is a poor barrier.

Sound Absorbing Materials

The most common types of absorbing materials are rock wool, fiberglass, cloth, polyurethane and cellulose fibers. All these materials are different in a sense that these do not absorb sound of all frequencies equally well. Sound absorption is influenced by factors such as material density thickness and, in the case of fibrous insulation products, fiber size and diameter, generally fine small diameter fibers will give superior absorption than coarser fiber blends.

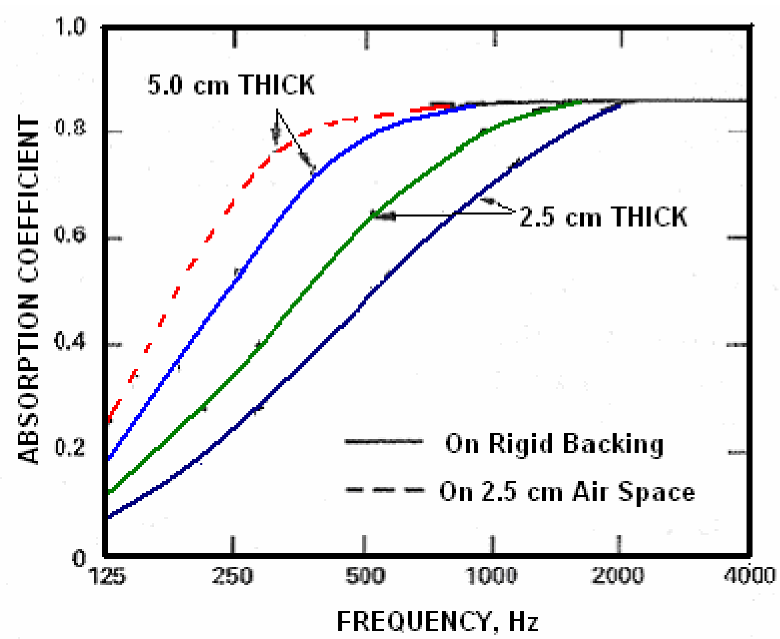
These are characterized by high absorption coefficients at high frequencies, decreasing at lower frequencies depending on the type and thickness of the material.

Sound Absorbing Coefficient (SAC)

The sound absorption coefficient indicates how much of the sound is absorbed by the material over a range of frequencies.

Example: ½" drywall on 2x4 studs has an absorption coefficient at 125 Hz of 0.29. The figure below illustrates typical absorption coefficient values versus frequency for 2.5 and 5.0 cm thick porous samples. Such absorbing materials function by resisting the air flow associated with the acoustical vibrations of the air and are most effective at higher frequencies, where they are thicker relative to the wavelength of the sound.

Since absorption decreases at lower frequencies; porous absorbing materials a few centimeters thick will never be highly absorptive at lower frequencies. Increased low frequency absorption can be obtained by adding an air space between the material and the rigid backing so that the sound absorber behaves like a thicker material.



There are three main standard methods used to test materials for absorption. Two of them are reverberation chamber methods – ASTM C423 in the U.S.A. and ISO 354 in Europe. These two methods are quite similar, but the ISO method – in general – will produce slightly lower overall numbers than the ASTM method. The other method is the impedance tube method, or ASTM C384. This method places a small sample of the material under test at the end of a tube and measures the absorption. Again, the numbers from this test are usually lower and are also not as representative of real-world applications of materials relative to the reverberation chamber methods.

Room Absorption Characteristics

The total sound absorption in a room can be expressed as:

$A = S_1\, \alpha_1 + S_2\, \alpha_2 + ... + S_n\, \alpha_n = \Sigma\, S_i\, \alpha_i$

Where:

- A = the absorption of the room (m^2 sabin)
- S_n = area of the actual surface (m^2)
- α_n = absorption coefficient of the actual surface

It is important to note that the absorption and surface area must be considered for every material within a space in order to calculate sabins.

The sound absorption coefficient can be expressed as the ratio of the absorbed sound energy to the incident energy or can also be expressed as a percentage:

$\alpha = I_a / I_i$

Where:

- I_a = absorbed sound intensity (W/m^2)
- I_i = incident sound intensity (W/m^2)

The absorption coefficient varies with frequency of sound and the angle of incidence of the sound waves on the material. These coefficients are normally determined for one-third octave bands under standard conditions. This provides an average value for each one-third-octave band and all angles of incidence. Data are usually given at six standard frequencies (125, 250, 500, 1000, 2000 and 4000 Hz), even more modern laboratories will provide data at all one-third-octave bands from about 100 Hz to 5000 Hz.

Absorption coefficients are usually obtained from manufacturers' literature and are usually obtained by testing in a reverberation test room (a room which has very long Reverberation Time) and then measure the RT so the coefficient can be derived from Sabin equation (the original version of RT calculation). There is a standard that details this procedure. The value of the coefficient for the same material varies with the type of the mounting in the test room.

Mean Absorption Coefficient

The mean absorption coefficient for the room can be expressed as:

$a_m = A / S$

Where:

- a_m = mean absorption coefficient
- A = the absorption of the room (m^2 sabin)
- S = total surface in the room (m^2)

A rooms acoustic characteristics can be calculated with the formulas above, or estimated for typical rooms.

The table below gives mean sound absorption coefficient values and reverberation time for some typical rooms.

Typical Room	Room Characteristics	Reverberation Time	Mean Sound Absorption Coefficient
Radio and TV studio	Very Soft	0.40	$0.2 < T_a < 0.25$
Restaurant Theater Lecture hall	Soft	0.25	$0.4 < T_a < 0.5$
Office, Library, Flat	Normal	0.15	$0.9 < T_a < 1.1$
Hospital, Church	Hard	0.10	$1.8 < T_a < 2.2$
Large church, Factory	Very Hard	0.05	$2.5 < T_a < 4.5$

The mean sound absorption coefficient should be calculated when more accurate values are needed.

Noise Reduction Coefficient (NRC)

In sound absorption applications the term NRC is often used as a performance indicator of how absorptive a measure is. The higher the NRC, the greater will be the sound absorption at those frequencies.

Example: ½” gypsum board (“drywall”) on 2x4 studs has an NRC of 0.05. Soft materials like acoustic foam, fiberglass, fabric, carpeting, etc. will have high NRCs; harder materials like brick, tile and drywall will have lower NRCs.

A material’s NRC is an arithmetic average of the four sound-absorption coefficients at 250, 500, 1000 and 2000 Hz rounded to the nearest 5%. *A material with NRC of 0.80 will absorb 80% of the sound that comes into contact with it and reflect 20% of the sound back into the space.* NRC is useful for a general comparison of materials. However, for materials with very similar NRCs, it is more important to compare absorption coefficients.

<u>NRC Limitations – What you should know</u>

1. The noise reduction coefficient (NRC) is only the average of the mid-frequency sound absorption coefficients (250, 500, 1000 and 2000 Hz).
2. NRC does not rate sound absorption at low frequencies less than 250hz, which is often the most critical to the application.
3. NRC does not address material’s barrier effect.
4. NRC single number rating system is convenient for ranking, the average effectiveness of different materials. However, for a more complete acoustical design, it is necessary to consider individual coefficients at each frequency.

PARTITIONS, BARRIERS & ENCLOSURES

Partitions, barriers and enclosures are the products used for restricting the passage of an airborne acoustic disturbance from one side to the other. These are nonporous and dense materials often having the structural properties. Typical barrier materials are steel, lead, plywood, gypsum wallboard, glass, brick, concrete, concrete bloc etc.

The quantity used to rate a sound attenuating material is transmission loss in dB. The sound transmission loss for different materials is given in the table below:

Material	Thickness (mm)	Surface Density (kg/m^3)	Sound Transmission Loss (dB)		
			125 Hz	500 Hz	2000 Hz
Plaster Brick Wall	125	240	36	40	54
Compressed Strawboard	56	25	22	27	35
Acoustic panel (Sandwiched type steel sheet with fiber glass)	50	27	19	31	44
Chipboard	19	11	17	25	26
Plaster board	9	7	15	24	32
Plywood	6	3.5	9	16	27

Sound Transmission Loss (STL)

When sound impinges on a partition, part of it is transmitted, some is reflected and the rest is absorbed. *The difference between the sound power incident on one side of the partition and that radiated from the other side (both expressed in decibels) is called the sound transmission loss (TL) in dB.* The larger the sound transmission loss (in decibels), the smaller will be the amount of sound energy passing through the partition.

The Mass Law is the general rule for determining the transmission loss of a partition.

$TL = 20 \log (fm) - 47.5$

Where:

- f is the frequency of sound wave, Hz
- m is the surface density of material, kg/m^2

Based on this equation, the sound insulation will increase by 6 dB for either a doubling of the frequency or surface density. In practice, however, the actual increase in transmission loss is somewhat less than that predicted by the Mass Law.

Sound Transmission Class (STC)

The Sound Transmission Class (STC) STC is a single-number rating of how effective a material or partition is at isolating sound.

Example: ½" drywall has an STC of 28. The larger the STC value, the better the partition, i.e., the less sound energy passes through it. For example, loud speech can be understood fairly well through an STC 30 wall but should not be audible through an STC 60 wall.

Hard materials like rubberized sound barriers, concrete, brick and drywall will have high STCs. Softer materials like mineral fiber, acoustic foam and carpet will have much lower STCs. Virtually every material filters out some of the sound that travels through it, but dense materials are much better at this than are porous or fibrous materials. Like NRC, STC is useful to get an overview-type comparison of one material or partition to another. However, to truly compare performance, the transmission loss numbers should be reviewed.

The transmission of sound between rooms involves not only the direct path through the separating assembly, but the flanking paths around the assembly as well. Flanking paths are the means for sound to transfer from one space to another other than through the wall. Sound can flank over, under, or around a wall. Sound can also travel through common ductwork, plumbing or corridors. Special consideration must be given to spaces where the noise transfer concern is other than speech, such as mechanical equipment or music.

Recommended Ratings

In general, loud speech can be understood fairly well through an STC 30 wall but should not be audible through an STC 60 wall. An STC of 50 is a common building standard and blocks approximately 50 dB from transmitting through the partition. Constructions with a higher STC (as much as 10dB better - STC 60) should be specified in sensitive areas where sound transmission is a concern.

The Uniform Building Code (UBC) contains requirements for sound isolation for dwelling units in Group-R occupancies (including hotels, motels, apartments, condominiums, monasteries and convents). UBC requirements for walls, floor/ceiling assemblies: STC rating of 50 (if tested in a laboratory) or 45 (if tested in the field).

STC Limitations – What you should know

1) The STC rating is based on performance with frequencies from 125 to 4000 Hertz (the speech frequencies) and does not access the low frequency sound transfer. The rating provides no evaluation of the barrier's ability to block low frequency noise, such as the bass in music or the noise of some mechanical equipment.
2) The STC rating is a lab test that does not take into consideration weak points, penetrations, or flanking paths. The field test however, does evaluate the entire assembly and includes all sound paths.
3) Improving the STC rating of a wall will probably not affect the reverberation or reflections within the space.

NRC V/s STC

NRC and STC are completely exclusive of each other.

Acoustic wall treatment with a high NRC can stop sound reflecting back into the space, possibly lowering the noise level within the space whereas acoustic wall treatments will not stop sound from passing through and into an adjacent space; therefore they do not improve the STC.

Improving the STC rating of a wall (adding mass, air space, cavity, insulation etc) will reduce the noise transfer to the adjacent space whereas improving the STC rating of a wall will probably not affect the reverberation or reflections within the space.

It is important to understand the difference between sound-absorption and sound transmission loss. Materials that prevent the passage of sound are usually solid, fairly heavy and non-porous. A good sound absorber is 15 mm of glass fiber; a good sound barrier is 150 mm of poured concrete.

SECTION -4 HVAC NOISE CONTROL

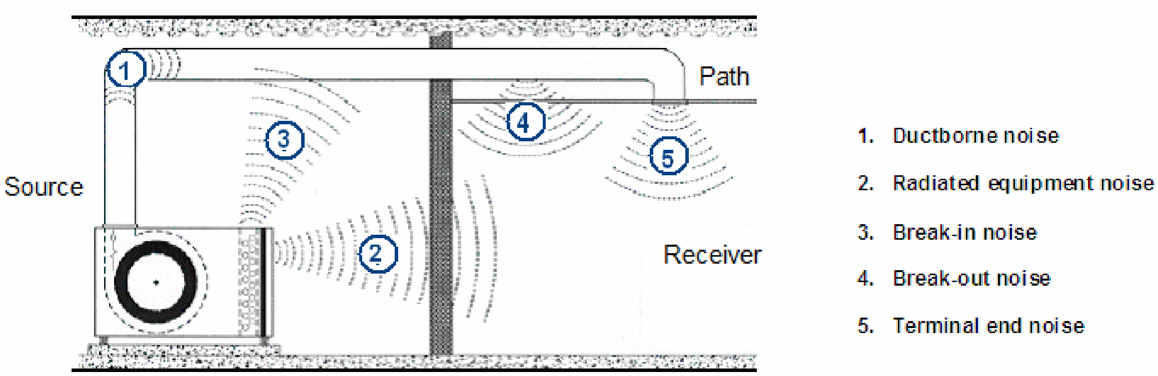
In the majority of rooms or open spaces, the most common source of disturbing noise is the HVAC noise. The primary objective for acoustical design of HVAC systems and equipment is to ensure that the acoustical environment in a given space is not unacceptably affected by HVAC system-related noise or vibration.

The ASHRAE Handbook, Chapter 43: “Sound and Vibration Control provides guidelines and procedures for noise control in HVAC systems. We won’t repeat the detailed procedures here, but present the basic principles of the design process for the building designers to grasp the general concepts. This section will discuss 25 key noise reduction strategies in HVAC Systems.

Noise Points in HVAC Systems

Sound and vibration are created by a source, are transmitted along one or more paths and reach a receiver. It is important to know which of these paths is most likely to cause acoustical problems so that it can be dealt with in the design phase of the project.

The design of ducted HVAC systems must address five distinct but related issues.



TYPICAL MECHANICAL SYSTEM NOISE COMPONENTS

Treatments and modifications can be applied to any or all of these elements to reduce unwanted noise and vibration, although it is usually most effective and least expensive to reduce noise at the source.

Duct borne Noise

Duct borne noise is generated by the flow of air and is directly dependent on the velocity of air. This component of the noise (sometimes called regenerated noise) propagates along the ductwork, follows all transitions and takeoffs, and ultimately exits at the diffuser or grille. The noise increases very rapidly as the flow velocity is increased and/or by changes in cross-section in the duct. It is affected by:

1. Objects such as dampers, grilles, registers, etc;
2. Constrictions in duct cross sectional area, silencer splitters etc;
3. Jet noise, inlet or discharge noise through orifices and air flow around bends and duct take offs (branches).

To avoid duct borne noise, note the following:

1. Select quiet fans based on sound power data and use variable frequency drives preferably. Provide good fan outlet conditions in accordance with ASHRAE guidelines;
2. Route duct in accordance with SMACNA guidelines. Do not turn the air in “the wrong direction.

Radiated Equipment Noise

Radiated equipment noise is generated by vibration of the mechanical equipment such as pump, compressor, air handler, fan and motor and is transmitted through the wall or floor into the adjacent space. Because the natural frequencies of floors, walls, beams, and columns typically range from 10 to 60 Hz, these components can be excited into greater vibration magnitude when located near equipment operating in a matching frequency range. The noise may be transmitted directly or via the duct to the building structure, or it may be radiated directly from the duct.

Excessive vibration and noise may occur due to:

1. Damaged or unbalanced fan wheel;
2. Belts too loose; worn or oily belts;
3. Speed too high;
4. Incorrect direction of rotation;
5. Improper lubrication.

<u>To reduce vibration noise paths, note the following:</u>

1. The fan and duct should be isolated from the building structure;
2. Use flexible (isolating) coupling between the fan and duct;

3. Use of vibration isolation foot pads below the rotating equipment;
4. Use of inertial base resilient mount floating floor;
5. Sound absorption treatment of the mechanical room walls and roof;
6. Routine maintenance of fan drive and motor.

Duct Break- In and Break - Out Noise

Duct breakout noise transmits through the wall of the duct, thus impacting the adjacent space. Noise breakout from ducts can occur from:

1. Fan noise passing through the duct;
2. Aerodynamic noise (also known as regenerated noise), from obstructions fittings etc in the duct;
3. Turbulent airflow causing duct walls to vibrate and rumble radiating low frequency airborne noise

<u>To reduce break-out noise, note the following:</u>

1. Design for low velocity. The air velocity in the main duct should not exceed 1500 fpm and velocity through the branch ducts should be limited to 600 fpm;
2. Design proper duct fittings for smooth flow and gradual velocity changes;
3. Make ducts stiffer. Use heavier material for duct walls and increasing damping (i.e. thicker steel sheeting). External bracing of ducts can also increase stiffness;
4. Adding damping (spray on or self adhesive compounds);
5. Add insulation on the external duct surfaces;
6. Provide acoustic silencers to reduce fan noise prior to allowing a duct to run above the acoustical tile ceiling of an occupied space.

Duct break-in noise is radiated equipment noise that enters the ductwork and propagates down the duct system. Flexible ducts are especially prone to break-in noise because of light weight.

To minimize break-in noise, note the following:

1. Avoid flexible ducts. Replace lightweight flexible ducts with heavier ducting such as sheet steel.

2. The flexible ducts can be enclosed in a solid enclosure constructed from timber, plasterboard or sheet steel, etc.
3. Add acoustic lining either glasswool or rockwool at least 1 inch thick;
4. Before enclosing flexible ducts, it should be noted that noise in the ceiling cavity will most likely penetrate the ceiling. This will happen more so if lightweight lay-in tiles are used. Fixed plasterboard ceilings give better acoustic performance than lightweight ceiling tiles.

Terminal Noise

The final links in the air distribution chain are the terminal air devices. These are the “grilles,” “diffusers”, “registers” and “vent covers” that go over the duct opening in the room. When selecting terminal devices; always select a device that has “noise criteria” rating of NC-30 or lower for the designed airflow rate.

NOISE REDUCTION STRATEGIES IN HVAC SYSTEMS

In HVAC systems, the source of noise is a combination of different processes, such as mechanical noise from fan(s), pump(s), compressor(s), motor(s), control dampers, VAV boxes and air outlets such as diffusers, grilles, dampers and registers. The HVAC noise that ultimately reach indoor space is made up of:

a. Low-frequency fan noise - Centrifugal fans generate highest noise in the low frequency range in or below the octave centered at 250-Hz.
b. Mid-frequency airflow or turbulence-generated noise – Terminal units, variable-air-volume (VAV) boxes produce their highest noise levels in the mid frequency range in the octaves centered at 250, 500 and 1000 Hz bands.
c. High-frequency damper and diffuser noises – Diffusers, and grilles, however, typically generate the highest noise levels in the octaves centered at 1000 Hz, or above.

It is important to appreciate that the sound energy can travel by several different paths. The airborne sound energy from the fan can propagate through the duct system in both directions from the fan, as well as into the fan room from the fan casing. Needless to say, each of these issues must be addressed, or else the design will fail.

The vibration energy of the fan can propagate through the fan room floor and other parts of the building structure, as well as through the walls of the duct system.

Note: *High frequency noise can be reduced using passive devices (attenuators, lining etc), but noise components at frequencies below 400-500 Hz are most difficult to attenuate.*

We will discuss various noise attenuation techniques in following paragraphs.

LOCATION OF EQUIPMENT

The installation position of equipment (chillers, air cooled condensers, cooling towers, exhaust fans etc.) is of critical importance in determining the noise level at the affected noise sensitive receivers. Where practicable, the equipment should be placed in a plant room with thick walls or at a much greater distance from the receiver or behind some large enough obstruction (e.g. a building or a barrier) such that the line of sight between receiver and the equipment is blocked. If noisy equipment has to be placed near a receiver due to spatial or other constraints, consider acoustic enclosures and barriers.

Air handlers are typically housed in mechanical rooms within the indoor space. These mechanical equipment rooms (MER) should be located away from sensitive areas and never on a roof directly over a critical space. If possible, isolate the equipment room by locating elevator cores, stairwells, rest rooms, storage rooms and corridors around its perimeter. The walls, floors and doors of MER must have high sound reduction indices and as the airborne sound easily passes through small gaps and cracks, the penetration points for pipes, cables and ducts through the walls must be well sealed. The doors must also be fitted with rubber sealing strips.

MER Size

As a rule, the larger the MER room, the quieter the HVAC system will be. It is important to have a sufficiently spacious mechanical room so that ductwork can be routed to accommodate insulation material and silencers. The air handler units and other mechanical equipment should be placed away from the walls or ceilings. There is a phenomenon called “close coupling,” in which a small air space will conduct cabinet vibratory motion to the wall or ceiling. A space of approximately 3 feet usually suffices. Provide a nominal 4 inch concrete housekeeping pad beneath equipment cabinets to minimize the effects of close coupling to the floor.

MER Wall Construction

The desired noise levels in the adjacent space and the amount of airflow determine the requirements for MER wall construction/treatment.

One method is to acoustically insulate the mechanical room walls and ceilings with fiber insulation mounted on wooden battens. The insulation shall be clad with perforated Aluminum sheet of 36 gauge. When sound strikes a surface, some of it is absorbed, some of it is reflected and some of it is transmitted through the surface. Dense surfaces, for the most part, will isolate sound well, but reflect sound back into the room. Porous surfaces, for the most part, will absorb sound well, but will not isolate. Acoustic insulation reduces the level of noise because while the sound waves are reflected back and forth in the room and interact with the sound-absorbing materials to lose some energy each time.

Wherever possible, plan the entrance to the MER from a non-critical buffer space such as a freight elevator lobby or janitor’s closet. If this is done, a solid core wood door or hollow metal door with glass fiber packing may be used. The door should have quality gaskets and drop seals to form an airtight seal all around.

MER Equipment

Right sizing of HVAC equipment and distribution system matching the turndown capabilities to the load being served is in essence, a fundamental engineering component to sustainable design. Besides high possibility of higher noise levels at source, an oversized equipment will short-cycle, frequently switching on & off which mean startup noise every time the equipment kicks in. Choosing the correct size (heating and/or cooling output) is critical not only in getting the best acoustic efficiency but also comfort, and lowest maintenance and operating costs over the life of the new system.

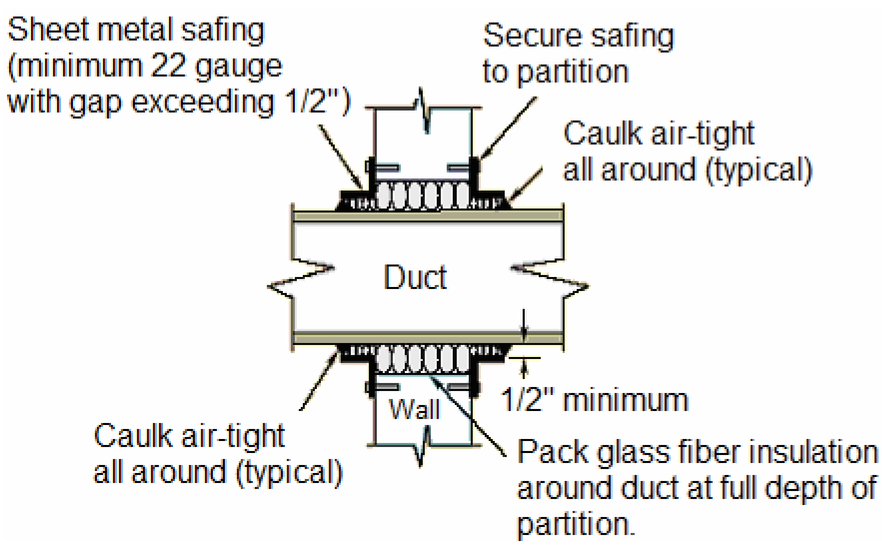
Further, the acoustical capabilities of the mechanical equipment should match the acoustical capabilities of the MER wall construction. For example, if a standard centrifugal fan system is selected for the project, and the acoustical engineer projects noise levels approximating NC 45 in the occupied space directly adjacent, the requirements for the wall and door should be in line with the known sound spectrum within the equipment room and projected noise levels outside the equipment room. A different MER wall may be selected for a mixed flow air handler and NC 35 projections.

On average, quieter equipment may generally be more expensive. However, it is almost always more economical in the long run to buy quieter equipment than to reduce noise by modification after purchase. Most equipment has a range of readily available noise control devices that are able to deal with the noise problems. It is advisable that noise levels specification is included when ordering new equipment. This allows the equipment suppliers to select appropriate equipment and optional noise control devices to suit the acoustic requirements.

MER Duct penetrations

All penetrations of MER walls must be sealed airtight. Sound, like air and water, will get through any small gap. All wall constructions must extend and sealed up to the floor construction above. If the wall is located under a beam, the space between the top of the beam and the deck must also be sealed.

The supply duct penetration must also remain resilient, avoiding rigid contact between the duct and the wall. The space between the duct and the wall should be packed with glass fiber insulation and be closed to a 1/2" wide gap. This gap should then be sealed with a permanently resilient sealant such as silicone caulk or acoustical sealant.



Duct penetration detail at wall

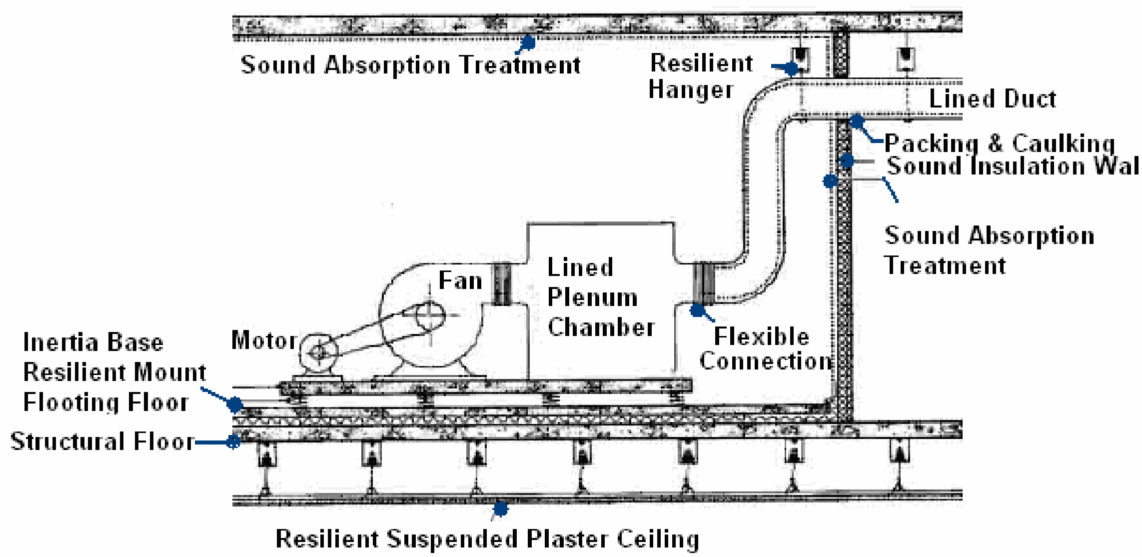
The figure above illustrates a recommended means of sealing the duct at the MER wall. The key to success is to allow no direct contact of the duct to the equipment room wall and to leave no voids between the ductwork and the wall.

VIBRATION ISOLATION

Rotating or motor-driven machinery generates lot of vibration that can travel through a building's structure and radiate from the walls, floors and ceilings in the form of airborne noise. It is therefore necessary to isolate any vibrating equipment from the surrounding structure. This can be accomplished by:

1. Balancing the machines to reduce the forces at the source; a well-balanced motor and fan impeller at as low a RPM as possible shall result in lower vibrations;
2. Using resilient, anti-vibration mountings between the machine and the supporting structure. Common materials used as vibration isolators are rubber, cork, various types of steel springs, and glass fiber pads;
3. Placing the rotating equipment on grade, in the basement, or in the sub-basement. Most pumps need isolators even in these desirable locations;
4. Using inertia blocks at the base of the equipment to reduce motion, lower the center of gravity, minimize the effect of unequal weight distribution of the support equipment, and stabilize the entire vibration isolation system. Generally an inertia block should be at least 6" thick and very stiff and rigid to avoid significant flexure in any direction;
5. Using flexible connectors on fans at each duct connection. These connectors should not be pulled taut, but should be long enough to provide folds or flexibility when the fan is off;
6. Considering floating floors particularly if the mechanical room is located on a different floor.

The figure below shows a source (air-handling unit) placed on a resilient support inside the mechanical room.



NOISE FROM FAN SYSTEMS

The most dominating source of noise in HVAC systems is because of the fans and blowers of the air handling units. The noise generated by fan is transmitted through a building in different ways: via walls, floors and leakage points to adjacent rooms and via the supply and extract ductwork to the rooms connected to the ducting.

Types of Fans

The noise generated by a fan depends on the fan type, fan speed, air flow rate, pressure and the intake/discharge arrangement. HVAC fans are mostly of the centrifugal type or the axial flow type. The primary selection criterion for a fan is based NOT on its acoustical characteristics but on its ability to move the required amount of air against the required pressure. In most cases it is not practical to substitute a fan which generates less noise, since a quieter design of the same type probably will not meet the other operating specification for the fan.

Axial-flow fans impart energy to the air by giving it a twisting motion. Axial fans generate a higher proportion of high frequency noise but less low frequency noise than centrifugal fans of similar duty. Three types are common:

- Vane-axial fans generally have the lowest amplitudes of sound at low frequency of any fan. For this reason they are often used in applications where the higher frequency noise can be managed with attenuation devices. In the useful operating range, the noise from vane-axial fans is a strong function of the inlet airflow symmetry and blade tip speed.

- The tube-axial fan generates a somewhat higher noise level than the vane-axial fan; its spectrum contains a very strong blade frequency component.
- The noise levels of the propeller fan are only slightly higher than those of the tubeaxial and vaneaxial fans, but such of the noise is at low frequencies and therefore is difficult to attenuate. Propeller fans are most commonly used on condensers and for power exhausts.

Centrifugal fans move air by centrifugal action. The centrifugal forces are created by a rotational air column which is enclosed between the blades of the fans. These produce most of their noise in the low frequencies, but in general are quieter than axial fans. Centrifugal fans may be further characterized by the type of blade used. Two common types of centrifugal fans frequently encountered in HVAC applications are forward curved and backward curved blades.

- The forward-curved fan is used primarily for HVAC work where high volume flow rates and low-pressure characteristics are required. Forward curved fans are commonly used in self-contained package units where space is at a premium. The most distinguishing acoustical concern of FC fans is the prevalent occurrence of low-frequency rumble. Forward curved blades will deliver more air than backward curved blades at a given speed, with increased noise, and if the resistance is reduced will deliver still more air and may overload the driving mechanism.
- Backward curved (BC) fans are generally much more energy efficient than FC fans at higher pressures and airflow. They require higher speed for the same air delivery and resistance as forward curved blades, but are less noisy and may be so designed as never to over load when the resistance is reduced. If the fan is close to a critical space, consider a backward curved (airfoil type) fan and silencers since the low rumble of forward-curved fans can be difficult to silence.

Many other designs fall between these two options.

Fan Characteristics

Maximum fan efficiency coincides precisely with minimum noise. Select fans that operate as near as possible to their rated peak efficiency when handling normal airflow and static pressure. This may seem obvious, but is often overlooked. Using an oversized or undersized fan can lead to higher equipment noise levels. The resulting variations in fan RPMs can also lead to airflow fluctuations and duct rumble.

Typical Sound Power Levels of Fans

Volumetric Flowrate (CFM)	Sound Power Level, dB (A) at Static Pressure	
	0.5 inch wg	3 inch wg
1000	79	95
5000	83	99
10000	85	101
20000	89	105
25000	90	107
50000	93	110

The sound power generated by fan performing a given duty is best obtained from manufacturer's test data taken under approved (ASHRAE Standard 68- 1986; also AMCA Standard 300-1986). However, if such data are not readily available, the octave-band sound power levels for various fans can be estimated by the empirical formulae described here.

$$SWL = 77 + 10 \log kW + 10 \log P$$

$$SWL = 25 + 10 \log Q + 20 \log P$$

$$SWL = 130 + 20 \log kW - 10 \log Q$$

Where:

- SWL = overall fan sound power level, dB
- kW = rated motor power, kW

- P = static pressure developed by fan, mm w.g.
- Q = volume flow delivered, m3/h

Octave band sound power levels are then found by subtracting correction factors from the overall sound power level calculated by any one of the above formulae.

Choice of Fan

A low self-noise level is an important criterion when specifying and choosing equipment. Fans should be chosen so they can operate at high levels of efficiency within their normal operating ranges. Fans that are made to run at unsuitable operating points, with subsequent poorer efficiencies, are often noisier than those that have been chosen correctly.

In CAV (constant flow) systems, fan should be chosen so that their maximum efficiencies are at the design air flows and static pressure. Operating the fan well off its duty point will create extra noise. Choosing a high-pressure air mover in a low pressure application will result in significantly greater noise. Conversely, a low-pressure air mover heavily loaded in a high-pressure application will also increase noise.

In VAV (variable flow) systems, fans should be chosen so that they can operate with optimal efficiencies and stability in the most frequently used working ranges. For example, a fan selected for peak efficiency at full output may aerodynamically stall at an operating point of 50% of full output resulting in significantly increased low frequency noise. Similarly, a fan selected to operate at the 50% output point may be very inefficient at full output, resulting in substantially increased fan noise at all frequencies. The fan for VAV applications should be selected for peak efficiency at an operating point of around 70 to 80% of the maximum required system capacity. This usually means selecting a fan that is one size smaller than that required for peak efficiency at 100% of maximum required system capacity. When the smaller fan is operated at higher capacities, it will produce up to 5 dB more noise. This occasional increase in sound level is usually more tolerable than the stall-related sound problems that can occur with a larger fan operating at less than 100% design capacity most of the time.

A correctly chosen and installed fan reduces the need for noise attenuation in the ducting system. The following points should be kept in mind:

- Design the systems: the ducts, terminal devices and components for low pressure drop;
- Compare sound data for different types of fans and from different manufactures and chose the quietest;
- Choose variable speed control for air flows rather than damper control.

Fan Speed

You need to know two things:

1. Fans are designed to push air: the faster the fan, the more air it pushes
2. Fans produce noise: the faster the fan, the more noise it produces

For our purposes, the best fan is the one that pushes the most air for the least noise.

The effect of rotational speed on noise can best be seen through one of the fan laws:

dB1 = dB2 + 50 log10 (RPM1 / RPM2)

Speed is an obvious major contributor to fan noise. For instance, if the speed of a fan is reduced by 20%, the dB level will be reduced by 5 dB."

The following table provides a guide to the trade-off that can be expected.

Fan Speed Reduction	Noise Reduction
10%	2 dB
20%	5 dB
30%	8 dB
40%	11 dB
50%	15 dB

Fan Selection

While it is true that fan noise is roughly proportional to the 5th power of fan speed, it is often mistakenly assumed that the lowest RPM fan selection is always the best selection. Note that the undersized fans operating at higher shaft speeds will produce more noise, and oversized (larger diameter) fans operating at lower shaft speeds will create more low-frequency noise (63, 125 and 250 Hz octave bands), which is much more difficult to remove.

Therefore, when choosing between two different fans of same duty and at same sound power, the one with the **lower** sound output at the **lower** frequencies is preferred from acoustic point of view. Sound power in higher-frequency bands is easier to control than low-frequency sound.

Fan discharge velocities for quiet operation

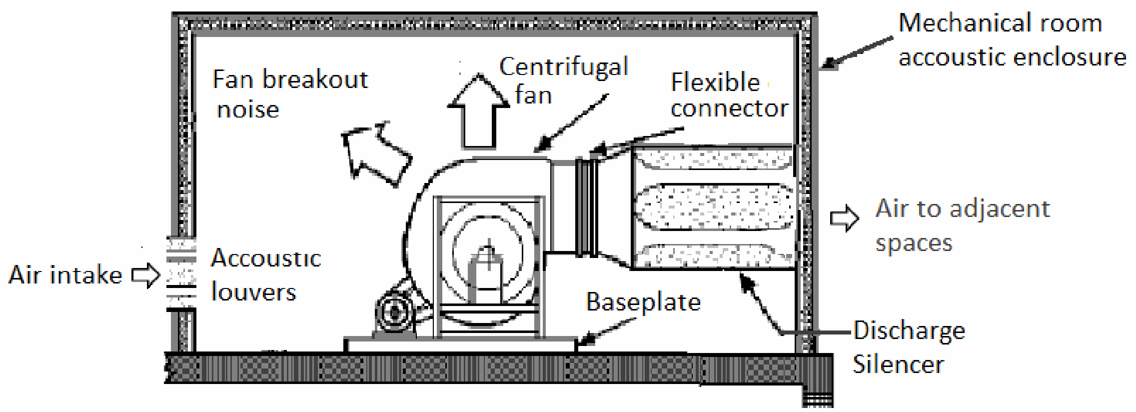
As conditioned air travels from a fan to an occupied room, it is subjected to acceleration, deceleration, changes in direction, division and a variety of surfaces and obstacles. Each of these effects disturbs the uniformity of the airflow and causes turbulence, which in turn creates noise. It is essential to limit the velocity of the airflow through all duct work systems in order to keep it from generating excessive noise. This is particularly true at the final branch ducts and the neck of the supply diffusers and return grilles where this regenerated noise is exposed directly into the occupied spaces.

Air-borne Noise from Fans

The air borne noise from fans mainly comes from the interaction of flow turbulence and solid surface of fan blades, and blade/fan vibration. The noise is transmitted upstream and downstream in the connecting ducts or to the atmosphere through the fan case.

<u>Remedies</u>

- Provide flexible connector at the fan discharge;
- Install a silencer at air discharge point of a fan so as to absorb noise generated from the fan;
- Install acoustic louvers at the fresh air intake;
- Acoustically insulate the mechanical room;
- Divert duct openings away from receivers for outdoor use;
- Provide acoustic enclosure to contain and absorb the noise energy for outdoor use, if required.



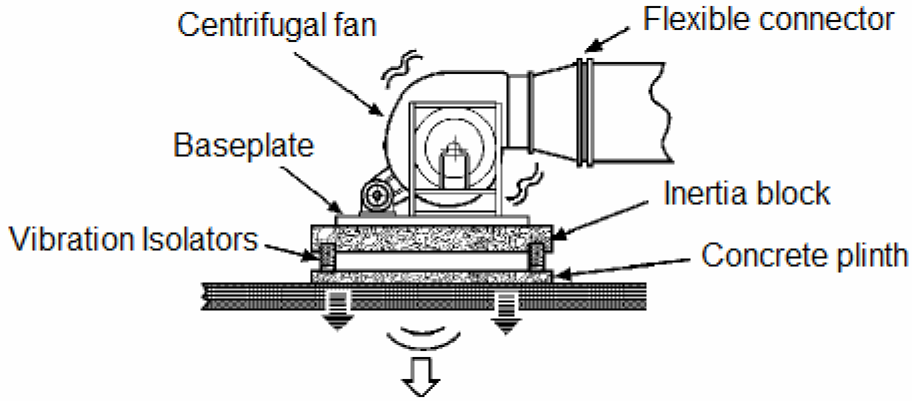
Centrifugal Fan serving the Indoor Spaces

Structure-borne Noise from Fans

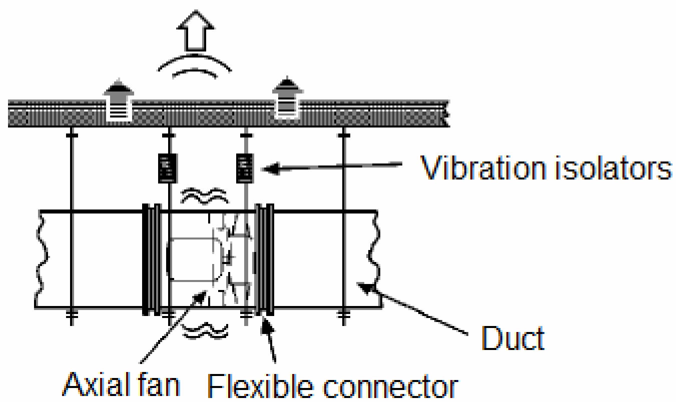
Vibration from an operating fan may be transmitted to the interior of the building through building structure when the fan is directly mounted on a supporting structure without proper isolation. The vibration noise can be very annoying and must be tackled during design.

<u>Remedies</u>

- Provide an inertia block to support the fan so as to add rigidity and stability to the ventilation system;
- Provide vibration isolators to support the inertia block, thereby isolating it from the building structure;
- Provide flexible connectors between the fan and associated ducts, thereby isolating it from the ductwork.



Vibration Isolation of Centrifugal Fan



Vibration Isolation of Axial Fan

Inertia Blocks for Fans

Inertial blocks are often used with fans to reduce the amplitude of oscillation. The table below shows a recommended weight of inertia block for various rating of ventilation equipment.

Equipment	Power	Speed (RPM)	Weight Ratio (Note 1) at		
			Min Area (Note 2)	Normal Area (Note 3)	Critical Area (Note 4)
Centrifugal Fans	<3 hp	All			2
	3 – 125 hp	<600		2	3

		600 - 1200			2
		>1200			2
	>125 hp	<600		2	3
		600 - 1200		2	2
		>1200		2	2

Note 1: Weight Ratio – Weight of inertial block over weight of equipment mounted on inertia block

Note 2: Minimum Area – Basement or on-grade slab location

Note 3: Normal Area – Upper floor location but not above or adjacent sensitive area

Note 4: Critical Area – Upper floor location above or adjacent sensitive areas.

Note: When the mass of the supported equipment is enormous, such as water cooling towers, there may be no need for additional mass in the form of inertia blocks; a rigid frame (e.g. Reinforced Concrete Beams) to support the entire assembly may be sufficient.

Vibration Isolators

Vibration isolators must be matched to the load they carry. A spring that is fully compressed doesn't offer any isolation and an uncompressed spring is also just as ineffective. There are many types of off-the-shelf anti-vibration mounts available, for instance rubber/neoprene or spring types. The type of isolator that is most appropriate will depend on, among other factors, the mass of the plant and the frequency of vibration to be isolated. Any supplier of anti-vibration mounts will be able to advise you on this.

Metal Spring:

Springs are particularly applicable where heavy equipment is to be isolated or where the required static deflections exceed 0.5” (12.5mm). Static deflection of a spring is a value specified by the suppliers. Selection of appropriate springs is important as this may result in poor isolation efficiency or even amplification of vibration, especially in the case that the vibration frequency is extremely low.

Vibration Isolators - Metal Springs

The most important feature of spring mountings is to provide good isolation due to its ability of withstanding relatively large static deflection. Metal springs however have the disadvantage that at very high frequencies vibration can travel along the spring into the adhered structure. This is normally overcome by incorporating a neoprene pad in the spring assembly so that there is no metal-to-metal contact. Most commercially available springs contain such a pad as a standard feature.

The table below provides the minimum static deflection required for achieving particular isolation efficiency at different equipment speeds.

Minimum Static Deflection for Various Speeds

Machine Speed (rpm)	Minimum Static Deflection at Various Isolation Efficiency (mm)			
	1%	5%	10%	15%
3600	14.0	1.5	1.0	0.5
2400	30.5	3.5	2.0	1.5
1800	56.0	6.0	3.0	2.0
1600	71.5	7.5	4.0	3.0
1400	91.5	10.0	5.5	4.0
1200	124.5	13.5	7.0	5.0
1100	150.0	15.5	8.5	6.0
1000	180.5	19.0	10.0	7.0
900	223.0	23.5	12.5	9.0
800	282.0	30.5	15.5	11.0
700	--	38.5	20.5	14.0
600	--	53.5	28.0	19.5
550	--	63.5	33.0	23.0
400	--	117.0	61.0	43.5
350	--	155.0	81.5	56.0
300	--	211.0	109.5	76.5
250	--	--	157.5	109.5

Isolation Pads:

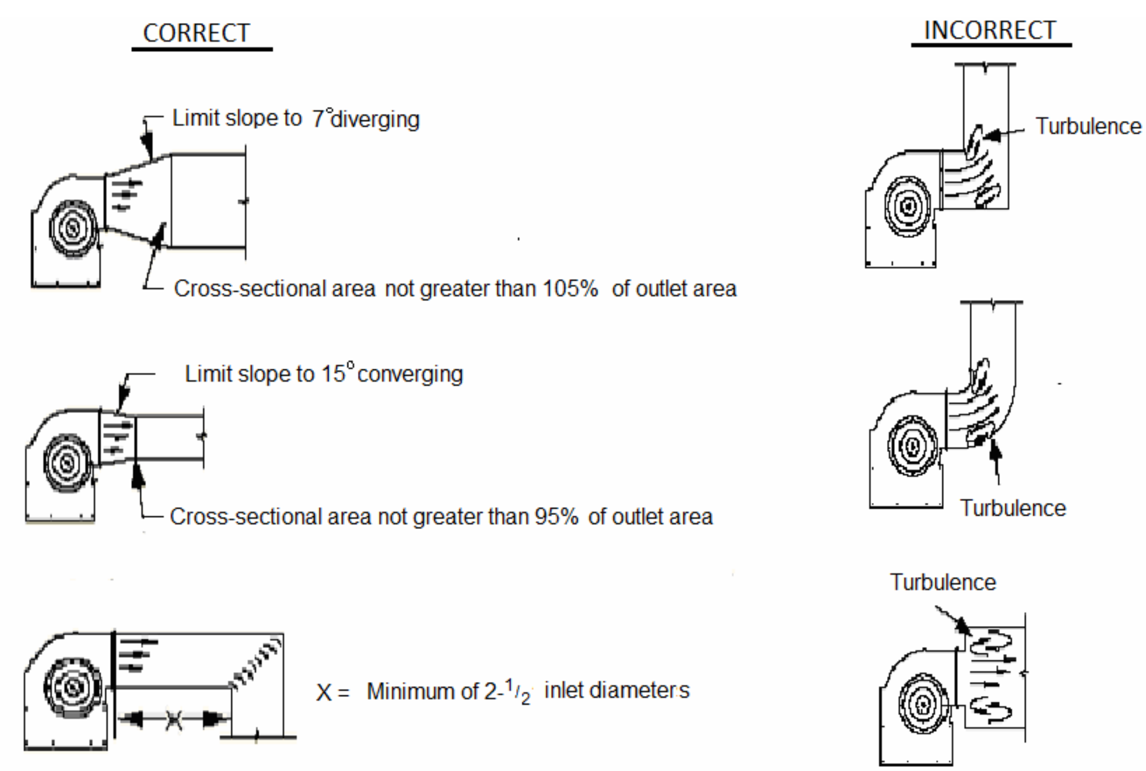
Isolation pads can be made of rubber, neoprene, glass fiber or combination of them. They are relatively cheap, easy for installation and replacement, and have the advantage of good high-frequencies isolation. However, attention should be given to the life of the isolation pads as some of them can be damaged by overload or low temperature.

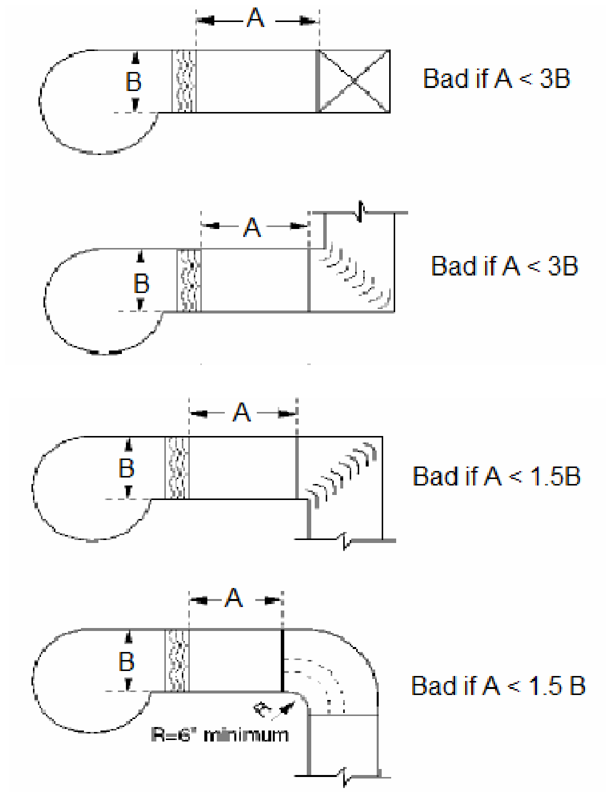
Guidelines for Centrifugal and Axial Fan Installations

Turbulence results in the generation of noise and an increased static pressure drop in the system. Therefore, the airflow at the entrance and exit of a fan should be as smooth as possible to minimize the generation of turbulence. For this reason, fitting (such as elbows and transitions) should not be placed closer than 3 to 6 duct diameters downstream from a fan and the duct transitions, particularly at the unit inlet, and discharge openings should be gradual, 15° maximum to minimize air pressure drop and turbulence.

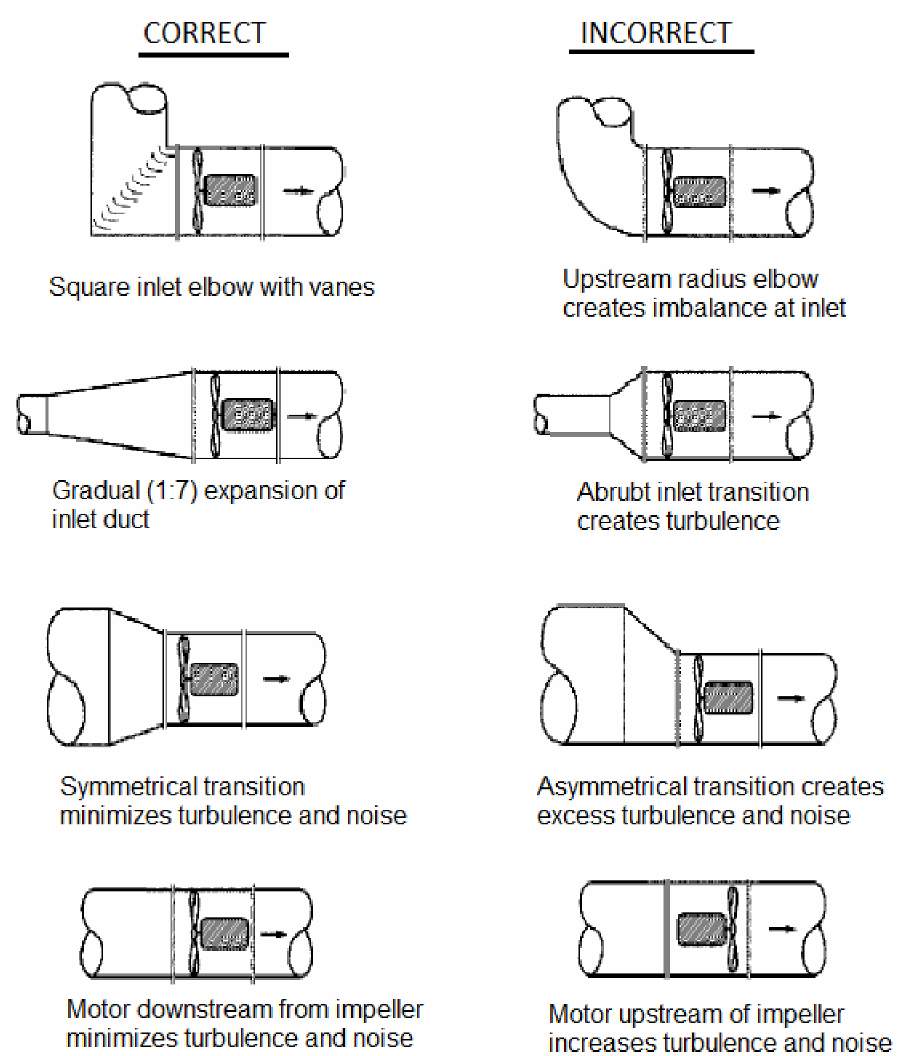
The figures below show examples of good and bad airflow conditions for fan installation:

<u>Guidelines for Centrifugal Fan Installation</u>




A
B
Bad if A < 3B
A
B
Bad if A < 3B
A
B
Bad if A < 1.5B
A
B
Bad if A < 1.5 B
R=6" minimum

Guidelines for Axial Fan Installations



Using Multiple Units

Sometimes, it may be prudent to use several smaller units instead of selecting single bigger equipment. This reduces the air volume per unit, cutting high air velocities. Smaller units operating at efficiency point can reduce overall equipment noise, if properly placed. However, this solution must be carefully planned in advance for several reasons. First, there must be space for the units and secondly it may be that more equipment needs to be located closer to the building occupants. Apply discretion while locating the equipment. Note that the use of in-room or portable air-conditioning units often makes noise problems worse, not better, for this very reason. Furthermore, multiple units may cost more initially and result in increased maintenance costs.

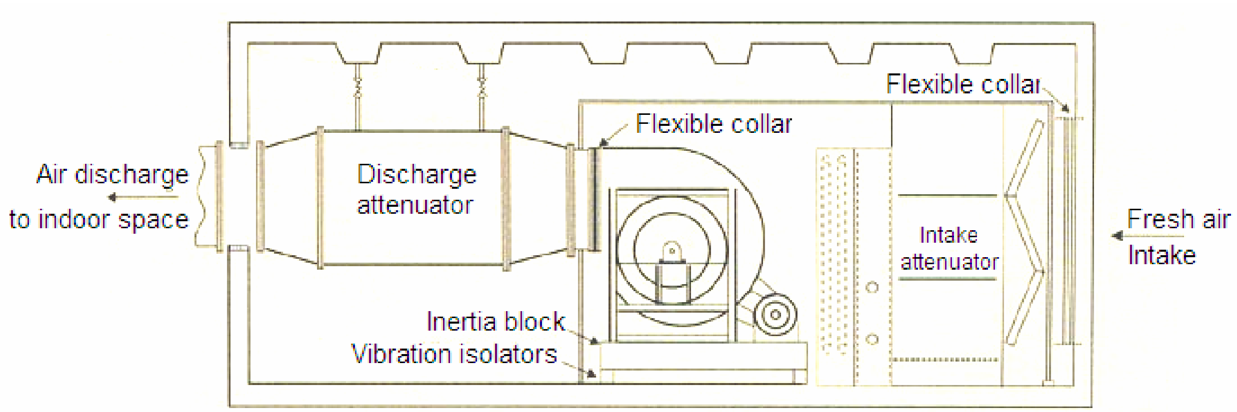
Typically a single air handling capacity should be restricted to 30,000 CFM maximum in commercial buildings. Use of multiple units of smaller capacity is preferred for noise control as well providing the flexibility of standby operation.

Sound Attenuators for Fans

If lower sound levels are required than are generated by well-designed fans, it is necessary to add attenuation to the system. This may be provided by sound attenuators or silencers, which allow the passage of air while restricting the passage of sound generated from air distribution equipment. They subdivide the airflow into several passages each lined with perforated sheet backed by mineral wool or glass fiber. A silencer usually has a cross section greater than the duct in which it is installed such that noise induced by high air flow velocity passing through the silencer can be avoided.

Silencers are available for circular or rectangular ducts and are fabricated in modular form in cross section, and in lengths of 0.6, 0.9, 1.2 and 1.5m, etc. They are generally specified by the insertion loss in decibels (dB) in each octave band, so that the degree of match with the sound power distribution of the noise source over the frequencies may be judged. The other important parameter associated with silencers is the resistance to airflow. The use of silencer will inevitably increase the load of the fan and it is important to address both the acoustic and air flow performances during the design stage.

The figure below shows a centrifugal fan, with sound attenuators on both the inlet and the outlet, used in the supply system of a central station ventilating system.



Sound Attenuation System for a Centrifugal Fan Installation

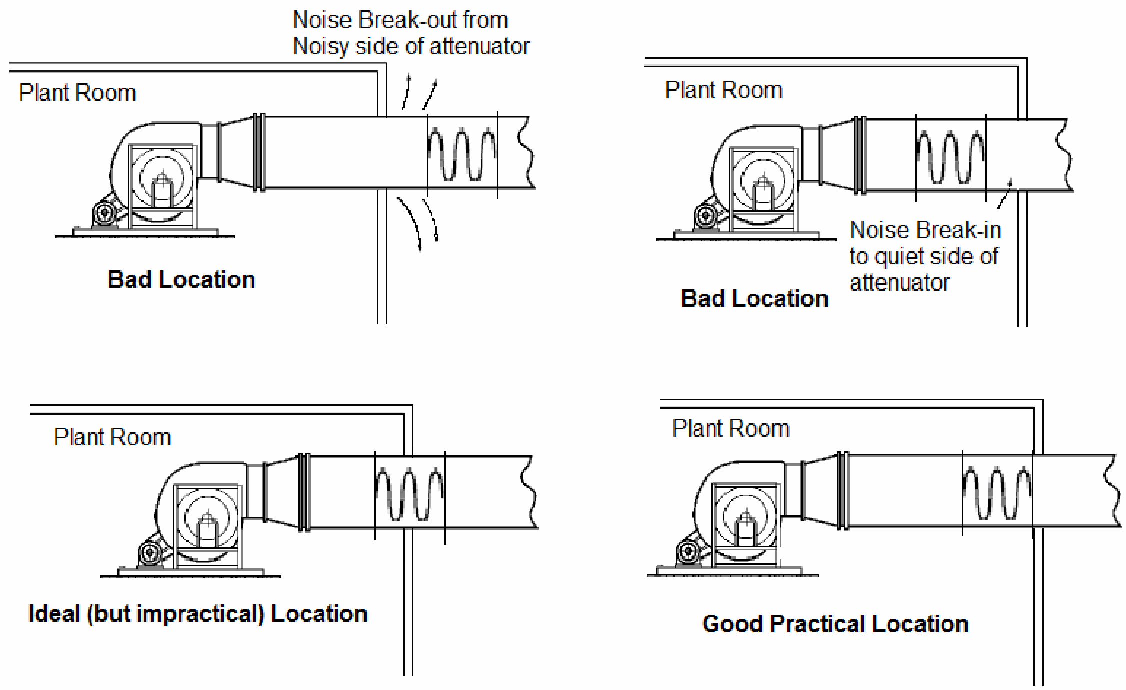
The following facts on silencers are worth noting:

1. Most sound attenuators for fan noise attenuation are of the absorption type because this type has broad band attenuation characteristics. These silencers perform well in mid-and high-frequency range and usually consist of sheet steel duct housing containing sound absorbent ‘splitters’ usually made of rockwool or glasswool.
2. Silencers restrict airflow, and may require more powerful fans to maintain the desired air volume. If air flow is restricted too much, the fan works harder, uses more energy and emits greater sound power levels. However, the less restrictive the silencer, the less effective it is in reducing low-frequency noise. One means of countering this problem is to extend the length of the silencer.
3. They are also much larger and heavier than the air ducts, often requiring 6 to 10 feet of unoccupied space. ASHRAE recommends 3-5 duct diameters on either side of a silencer and the fit-the-system silencers help overcome space restrictions that most HVAC designers face on a regular basis.

Location of Silencers

The position of the silencer in the duct system can be very important in determining its effectiveness in reducing the noise at the reception point. The optimum position can be governed by the possibility of noise breaking into the duct after (downstream as far as noise is concerned) the silencer, e.g. from noisy plant-rooms, or by possible break-out of noise from the duct before (upstream of) the silencer. Positioning the silencer close to bends can cause increases in pressure drop and self-generated noise.

Silencers should ideally be located where the duct leaves the plant room. Care must be taken to avoid plant room noise from entering the quiet side of the silencer.



If a silencer is located within a mechanical room, noise may enter the system through the sheet metal duct after the silencer. If a silencer is located away from the mechanical room walls, noise may escape the system through the sheet metal duct before it is attenuated by the silencer. Ideally, a sound attenuator should be located within or immediately adjacent to the mechanical room's walls.

Silencers should be located far enough upstream of any acoustically sensitive space to ensure that the noise they generate is adequately attenuated before it reaches the occupied room.

DUCTWORK NOISE

Air flowing through ducts induces vibration at the duct wall, which generates rumbling noise. In addition, the noise inside the duct can be transmitted to the atmosphere through the duct surface. To avoid air-borne noise, one should:

1. Size duct for low resistance and for low air velocities. Note that the airflow generated noise is proportional to the fifth power of the velocity. The general guidelines are:

Application	Supply systems ft/min	Extract systems ft/min
Sound studios, churches, libraries	800-1000	1000-1400
Cinemas, theatres, ballrooms	1000 - 1500	1200-1600
Restaurants, offices, hotels, shops	1200 -1600	1500-1800

In practice, airflow generated noise can be ignored when velocities are below 1500 ft/min in the main duct and 600 ft/min in branch ducts.

2. Smaller ducts tend to be noisier and leakier than larger ducts due to higher air speeds and pressures. Over-sizing the ducts is a good practice. Figure out the airflow required for each room in "CFM" (cubic feet per minute). To find out what size you should make the ducts, divide the CFM by the cross-sectional area of the duct in square feet (ft^2). Example: 500 CFM requiring 12" round duct yields 0.785 ft^2 cross-sectional area. Therefore, the air flow velocity will be 500/0.785 = ~637 FPM (feet per minute). Any result you get for the above under 1,000 FPM is good;
3. Avoid unnecessary turbulence by providing adequate distances between components (at least three duct diameters, but preferably more);
4. Choose components that allow smooth flows through the ducting, bends, junctions and terminal devices;
5. Avoid sudden cross-sectional changes or sudden changes of flow direction in the ducting system;
6. Stiffen the vibrating duct surface with supporting webs so as to reduce the movement of the vibrating surface;
7. Apply damping material to the vibrating duct surface so as to reduce the movement of vibrating surface;
8. Apply composite lagging of sound absorbing materials to contain the radiation of noise;

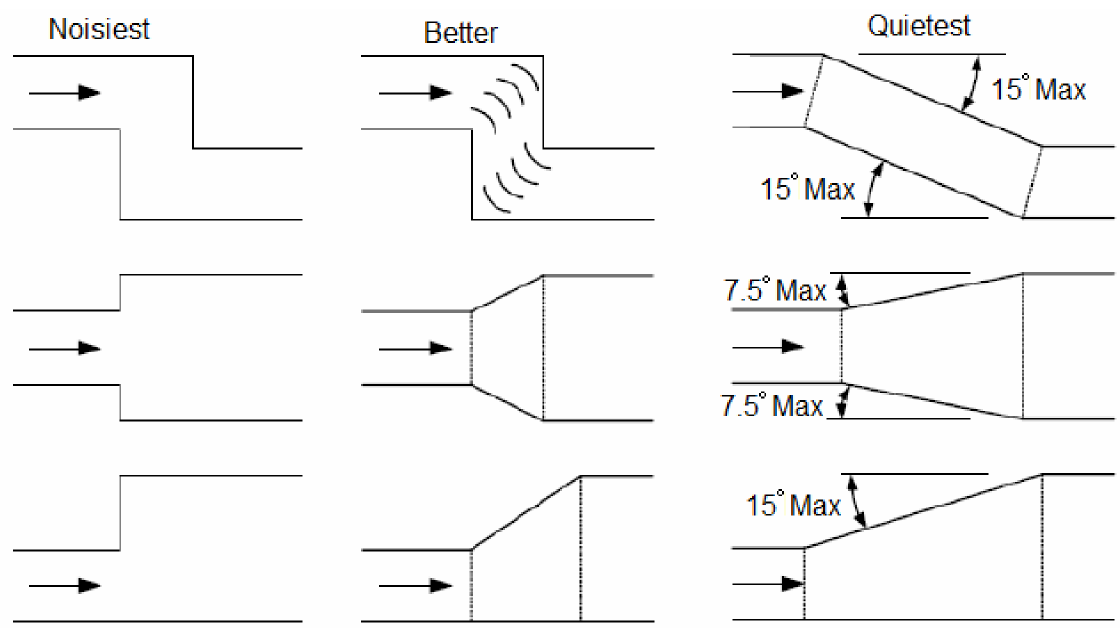
9. With centrifugal-type fans, the low frequency content of the fan noise is the most prevalent and difficult to attenuate. The best way to attenuates low frequency noise is by adding mass, i.e. by increasing the duct gauge from SMACNA standards, one can increase the mass and enhance the sound barrier;
10. The best duct configuration to reduce fan noise breakout is circular duct. The hoop strength of the circular duct is very good; therefore, minimal noise breaks out of the ductwork. While we know that most project ceiling plenums cannot accommodate circular ductwork, the idea in establishing aspect ratios of ductwork is to size the duct to approach circular;
11. Use round duct near fans when the duct must pass over critical areas. Use rectangular duct near fans over non-critical areas;
12. Locate main duct systems away from acoustically sensitive areas. The greater the length of supply duct over non-critical spaces, the more likely the duct is to reduce its acoustical impact on occupied spaces. Generally, 15 to 20 ft of ductwork is desired prior to entering the ceiling plenum of the occupied space;
13. Avoid very sharp bends in the ducts. Where bends are necessary, make sure they are gradual and, if possible, include long, radiuses turning vanes. Use low pressure drop elbows & fittings (per SMACNA guidelines). Note that the bends reduce noise, but only if they are gradual and preferably equipped with turning vanes;
14. If the ducts are sheet metal, you may need to isolate them from the building using spring isolating hangers;
15. Split larger ducts (>54" wide) into two and if possible, exit the equipment room using multiple ducts. *When the air stream is divided, the sound carried in each downstream branch is less than the sound upstream of the branch take-off.* Thus if the main duct divides into two equal branches, the sound power level in each branch just below the junction is 3 dB less than in the main branch just above the junction. The appropriate level of attenuation can then be determined from table below.

Duct Splits, dB

% of total air flow	5	10	15	20	30	40	50	60
Attenuation	31	10	8	7	5	4	3	1

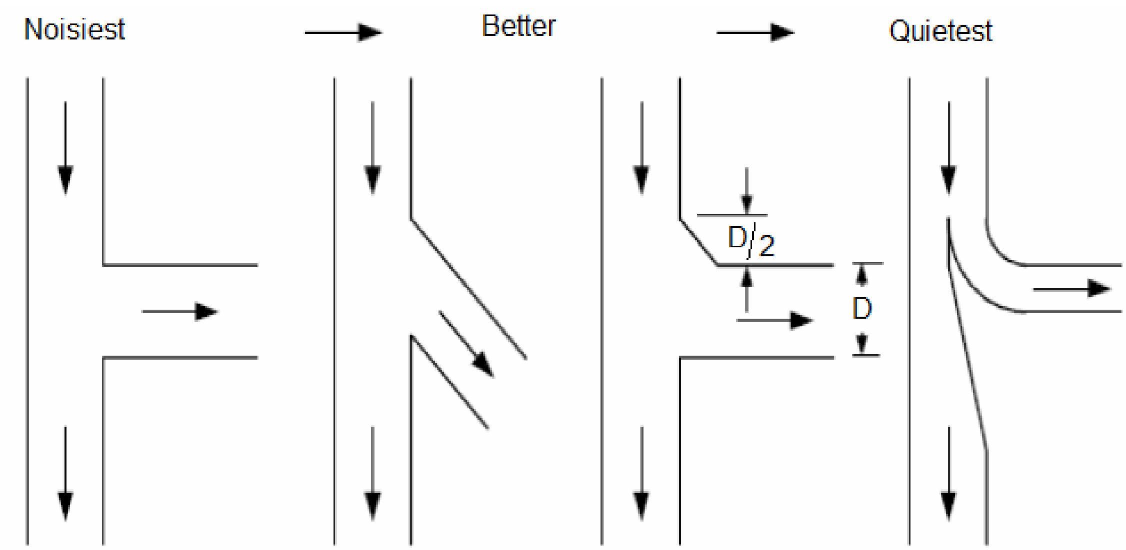
If the flow had divided into two parts down one branch and three parts down the other, then in the smaller branch the branch attenuation would be 10 $\log_{10}$ (5/2) = 4 dB, i.e. the sound power level in the smaller branch would be 4 dB less than in the main branch. In the other branch the attenuation would be 2.2 dB.

The following figures illustrate some good engineering practices:



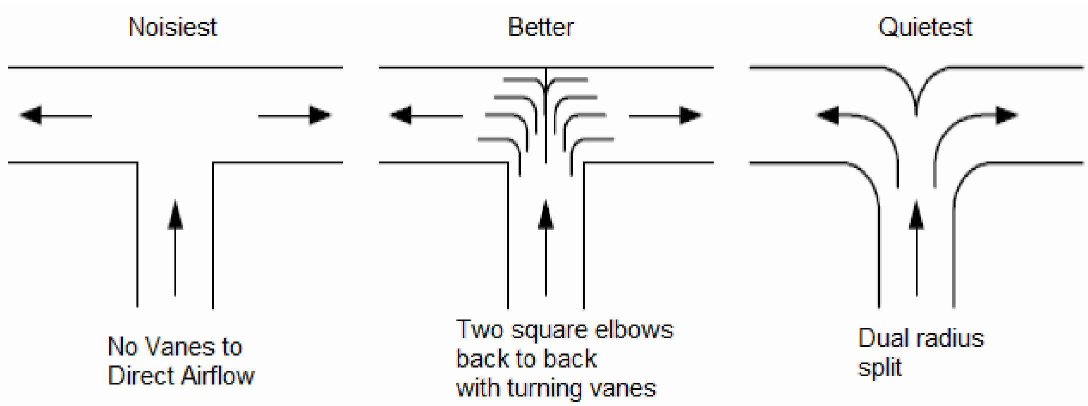
Recommendations for Minimizing Airflow generated Noise in Duct Transitions

Source: SMACNA



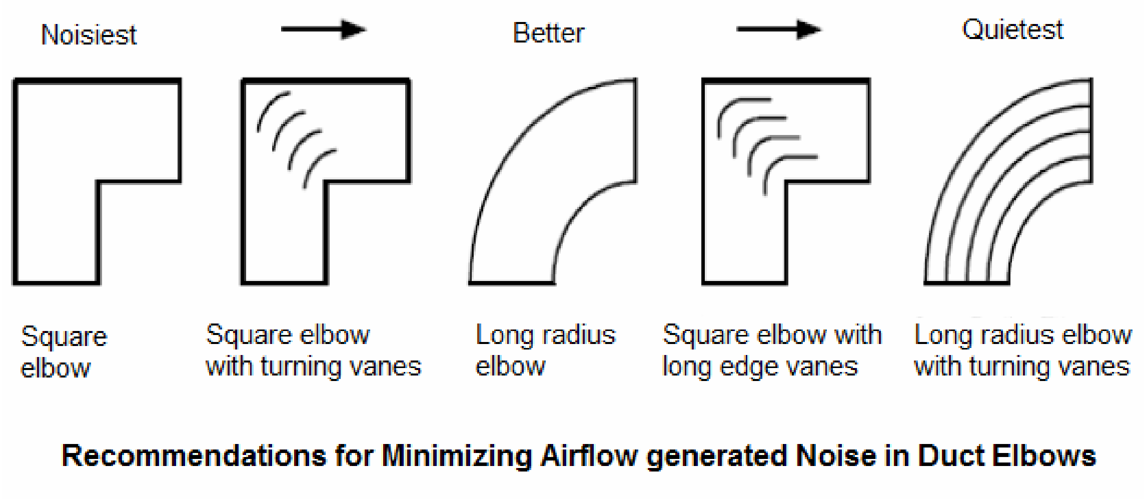
Recommendations for Minimizing Airflow generated noise in Duct Branch Takeoffs

Source: SMACNA



Recommendations for Minimizing Airflow generated Noise in Duct Tees

Source: SMACNA



Source: SMACNA

Duct Lining

Internal lining of ductwork with suitable sound absorbing materials provides attenuation to the air-borne noise. The ARI Standard (ARI 885), as well as the ASHRAE Handbook, provides guidance on insertion loss or decrease in sound power level over a length of duct.

The table below illustrates why it is important to pay particular attention to lower octave bands. The insertion loss in the 125 Hz octave band for a 24"X48" duct with 2" lining is 2 dB over a 10' length of duct. For the same duct, the insertion losses for the 250 Hz and 500 Hz octave bands are 7 dB and 22 dB respectively (see Table below).

Insertion Loss for Common Sized Ducts

Type	Lining	Dimension(s)	Insertion loss, dB/ft					
			Octave Band Center Frequency, Hz					
			125	250	500	1000	2000	4000
Rectangular	1"	18" x 36"	0.2	0.5	1.4	2.8	2.2	1.8
Rectangular	1"	18" x 54"	0.2	0.4	1.3	2.7	2.0	1.7
Rectangular	1"	24" x 48"	0.2	0.4	1.2	2.4	1.7	1.5
Rectangular	2"	18" x 36"	0.3	0.9	2.5	3.5	2.2	1.8
Rectangular	2"	18" x 54"	0.3	0.8	2.3	3.3	2.0	1.7
Rectangular	2"	24" x 48"	0.2	0.7	2.2	3.0	1.7	1.5
Round	1"	30" Diameter	0.16	0.45	1.16	1.33	0.95	0.69
Round	1"	36" Diameter	0.08	0.35	1.02	0.93	0.71	0.60
Round	1"	38" Diameter	0.06	0.31	0.96	0.80	0.64	0.58

Source – ASHRAE 1999, Application Handbook

According to the 1999 ASHRAE HVAC Applications Handbook, this pattern also holds true for transmission losses through ceiling tile, walls, floors and other building materials. Consult the Handbook for more detailed information about insertion losses. Generally speaking, if the equipment sound levels are lower in the first and second octave bands (63 Hz and 125 Hz); the project will be quiet.

The common duct liner materials are fiber glass, polymide and elastomeric foam. Sound absorption is directly proportional to exposed surface area and liner thickness. A common reference for measuring sound absorption of duct liner is the Noise Reduction Coefficient (NRC). A one-inch-thick fiber glass duct insulation has an NRC of 0.70; that means it effectively absorbs 70 percent of the sound at the most common frequencies. Two-inch-thick duct liner has an NRC of 0.90, absorbing 90 percent of the sound.

Note the following facts:

1. The attenuation produced by duct liner (in dB per meter run of duct), depends on three factors:

    a. Sound absorption coefficient (SAC) of the duct lining material: The attenuation produced by duct lining is greatest at medium frequencies and poor at low frequencies.

    b. Thickness of the absorbing lining: An increase in thickness produce improved attenuation particularly at low frequencies.

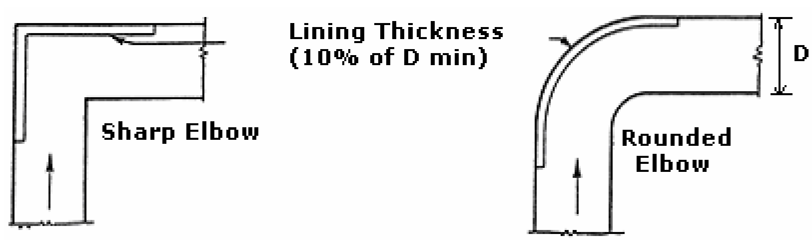
    c. Perimeter to cross-sectional area of the duct (P/S): In order to achieve maximum attenuation the shape of the duct should be designed to give the highest possible P/S ratio, which in effect means that for a given cross-sectional area of duct the sound is exposed to the greatest possible surface area of sound-absorbing material.

2. The location of duct lining can be a critical factor. It is normally placed at the start of a duct system to attenuate fan noise and near the outlets to correct air flow generated noise from dampers and fittings.

3. Noise goals depend on the use of a space. An office, apartment, or classroom usually requires a moderate amount of HVAC-noise reduction for normal activities to be accommodated. For these projects, as little as 10 to 20 ft of lined

duct, usually immediately downstream from the fan or approximately 10 ft from fan discharge, is all that is needed for noise goals to be met.

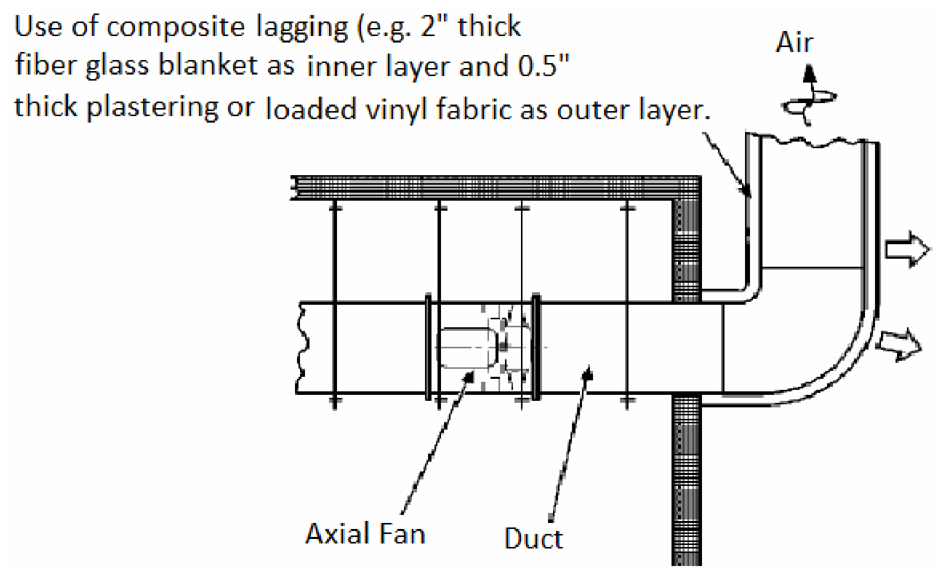
4. Turns and thin linings help reduce high-frequency noise but have little effect on low rumble. Thick linings, true plenums (large in comparison to connected ducts), or wraps on rectangular duct provide the most reduction of low-frequency sound.
5. Internal liners reduce duct cross-sectional area. For that reason, liners usually are kept to a minimum thickness, often 1 inch. To illustrate the impact of liner thickness, for an 18-in.-by-24-in. duct, going from a 1-in.-thick liner to a 2-in.-thick liner decreases flow area by about 20 percent, resulting in an increase in air velocity of 26 percent.
6. For acoustic purposes, square elbows are preferred to radius bends but the pressure drop will be higher. The lining should have a thickness at least 10% of D, the clear width between the two linings and the length of lining should extend a distance not less than 2D before and after the bend.



Unfounded concerns about health issues related to fiber glass being carcinogenic have been laid to rest by a definitive determination issued by the International Agency for Research on Cancer, an arm of the World Health Organization.

External Duct Insulation

In addition to the air-borne noise, sound may also transmit directly through the wall of the ducting and into the surrounding room. This breakout noise can be dampened by insulating the outside of the duct with a rockwool or glasswool. Further attenuation is achievable when an additional mass layer is applied as a covering onto the insulation.



The sound attenuation achieved inside the duct is also enhanced by external duct lagging particularly at low frequencies, up to about 500Hz.

Duct Vibrations

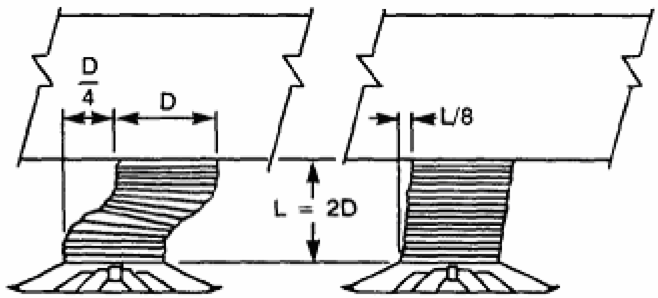
Structure-borne noise often arises due to the mounts (duct hangers or supports) attaching the ductwork to walls or ceilings and travels through the building in the form of vibration. When airflow fluctuates, due to a change in fan speed or other variables, it can make the duct surface vibrate. This rumble by itself can produce sound levels from 65 to 95 decibels at frequencies that range from 10 to 100 Hz. At the mid range, this noise is as noticeable as normal human speech at a distance of three feet – not easily ignored.

To solve this problem, special flexible fittings should be used which acoustically isolate the duct system from the structural elements of the building.

Flexible Ducts

Flexible ducts are usually provided to connect the duct to the terminal diffusers, which are furnished with integral volume dampers. Since dampers generate noise when partially closed, the sound power levels of the units are a result of the air volume handled by the diffuser, and the magnitude of the pressure drop across the damper.

A misalignment or offset that exceeds approximately one-quarter diameter in a diffuser collar length of two diameters can cause a significant change in diffuser sound power level above that of the manufacturer's published data. The figure below shows an example of increased pressure drop and increased noise level for a flexible duct connection. When there is an offset of only 1/8 the diameter, there is no appreciable change in the diffuser performance.



Example of Increased Pressure Drop and Noise Level for Flexible Duct Connection

Duct Fittings

As conditioned air travels from a fan to an occupied room, it is subjected to acceleration, deceleration, changes in direction, division and a variety of surfaces and obstacles. Each of these effects disturbs the uniformity of the airflow and causes turbulence, which in turn creates noise. The phenomenon is more pronounced where rectangular ducts change size or direction.

1. Elbows, take off fittings, transitions, etc. help reduce low frequency sounds through the duct system especially in the 500 Hz range as generated by fans. These duct fittings tend to reflect sounds back upstream, back towards the fan, thus reducing the amount of sound energy moving forward towards the room.
2. For a square duct, the bend attenuation is a maximum of some 7 or 8 dB at the frequency (octave band) for which the wavelength of sound in air is twice the duct width. At higher frequencies, the attenuation drops to 3 or 4 dB and at lower frequencies it can fall to less than 1 dB.
3. If the bend contains turning vanes to help the airflow around the corner smoothly then the attenuation produced will be very much reduced (but so, of course, will the amount of noise regenerated by the bend).
4. Metal turning vanes reduce turbulence by smoothing airflow in the right angle, but they can create their own noise problems by vibrating and reflecting or regenerating sound. Choose rounded finger guards when possible.

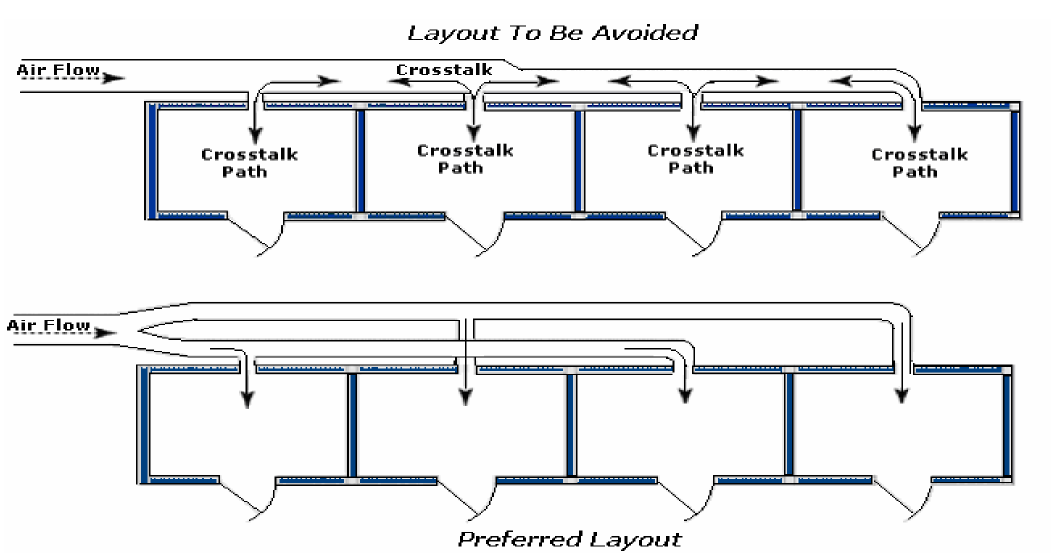
5. Turns and thin linings help reduce high-frequency noise but have little effect on low rumble. Thick linings, plenums (large in comparison to connected ducts), or wraps on rectangular duct provide the most reduction of low-frequency sound.

Duct Crosstalk

Lay ducts to prevent crosstalk and to attenuate the fan noise. The duct work that connects a fan or air handler to a room is a contained system that will also connect the equipment noise and vibration to the room unless adequate precautions are taken to attenuate the noise before it gets there. Without internal sound-absorptive duct liner or prefabricated sound attenuators, noise travels effectively down the duct system right along with the conditioned air.

Crosstalk occurs when noise from a space, e.g. talking, music, radiated noise, etc., enters the ductwork, propagates along the duct work, and ultimately impacts an adjacent space. A common example occurs in residential homes when on the third floor, you can hear the television on the first floor by listening at the supply diffuser or return grille. Crosstalk is typically composed of ductborne noise, break-in noise, and break-out noise. *The most common method of controlling cross talk is to avoid connecting rooms with short lengths of duct, by lining the ducts connecting these rooms with acoustical materials, and by installing silencers (sound absorbing devices) in the duct.* Crosstalk through ducts can also be attenuated by modifications to duct layouts (refer to the figure below) and increase in the amount of end reflection (higher number of smaller registers is preferable to fewer larger registers).

Ductwork Layout to Reduce Crosstalk



Although noise control issues through the supply air duct system are routinely considered in HVAC design, inexperienced mechanical engineers and contractors often forget that the *return*-air path is an equally important contributor to noise problems. In fact, because return-air systems sometimes employ common plenums above corridor ceilings, there may be less duct work in the return-air path to attenuate the noise, and the transfer of return air from one space to another may be a significant breach of the sound isolation between them. The best and really only recommended solution is to avoid short returns completely. The return intake should be based on a very low face velocity, return duct lengthened and lined, plus the air should be made to turn at least twice in its path toward the equipment.

ACOUSTIC LOUVRES

Similar to silencers, acoustic louvers are also commercially available devices that allow the passage of air while restricting the passage of sound generated from noisy spaces. They act much the same as ordinary louvers but consist of hollow acoustic vanes instead of flat sheet vanes or could be fabricated from fiberglass. Turning vanes made of fiber glass instead of metal eliminate the noise of turbulence and vibration at duct angles.

The acoustic performance of an acoustic louver is specified by the transmission loss in decibels (dB) in each octave band. This enables a direct comparison to be made between the performance of the louver and a solid wall/structure which it probably replaces. Since an acoustic louver is a very short attenuator, it is appropriate only where the length of space is restricted and the noise reduction requirement is low. Acoustic louvers are frequently installed in the facades of buildings where they are architecturally acceptable and provide a requisite amount of noise attenuation to prevent creating unacceptably high noise levels outside.

LARGE PLENUMS FOR SOUND ATTENUATION

Plenums can provide a substantial reduction in outlet noise that is critical for achieving low NC levels outside the MER. An acoustical plenum refers to either an air-handling unit enclosure or a large volume expansion in the ductwork. The plenum causes little static pressure drop and is effective in reducing low-frequency noise.

The methods for approximating the acoustic attenuation of a plenum depend on the sound absorption coefficients of the plenum lining, the exit area, wall surface area, and the distance between the entrance and exit. It is essential that a plenum with the following features be part of the overall acoustic solution:

1. 18-gauge outer casing to provide maximum stability and minimize vibration.
2. 3” insulation provides a significantly higher sound absorption coefficient for the critical lower octave bands versus 2” insulation.
3. 3 lb/ft^3 density insulation provides higher sound absorption coefficient for the critical lower octave bands versus 1½ lb/ft^3 density insulation.
4. Perforated sheet metal lining over all insulation to protect it from erosion and enhance sound attenuation.
5. Factory-supplied duct connections to provide a proper fit and avoid exposed insulation from field cut duct openings.
6. Internal baffles to minimize turbulence in the plenum, reducing the mechanical energy and noise.
7. The plenum can be fabricated out of fiberglass board that offers better low-frequency attenuation than lining. Duct board can “leak” some noise, but does not carry it along the length of the duct like bare sheet metal. This makes duct board popular for duct plenums or sections leading directly from the HVAC equipment. Any noise leaked by the duct board is diffused into unoccupied space and quiet air is sent on into the system. The use of fiberglass duct is limited by the levels of static pressure it can withstand and is not advised for high-pressure systems.

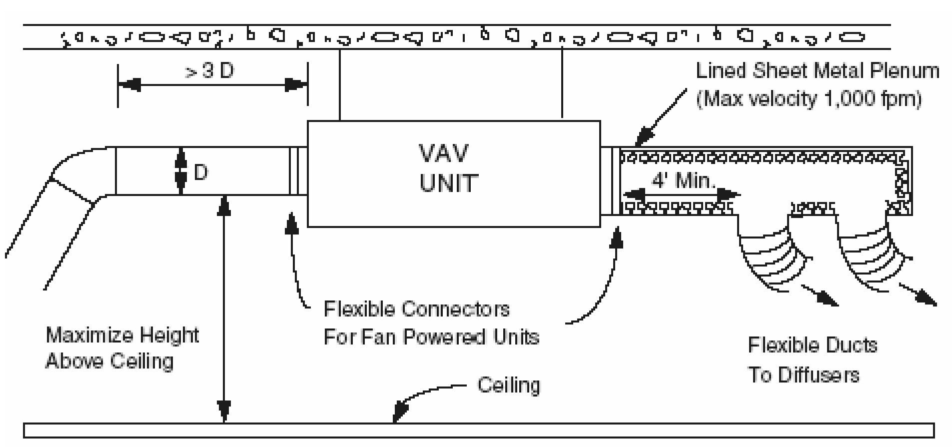
AIR TERMINAL BOXES

VAV (Variable Air Volume) air terminal units are components used in ducted air distribution systems to maintain the desired temperature in a space served by varying the volume of air. Basically these units consist of a sheet metal box containing a damper, controls and a sensor, and they are usually connected to a supply header via a flexible circular duct. They usually discharge air to one or more diffusers via rectangular sheet metal ducts. The noise of any air terminal unit can propagate through two prominent paths:

1. The units discharge ductwork to the connected outlet(s);

2. By direct sound radiated from the casing of the unit into a ceiling plenum and then through a ceiling (usually acoustical) into the space served.

Usually the VAV installations always require acoustical tile ceiling (NRC times % coverage > 0.65), but it is of little help if the design does not allow for the source noise control of VAV boxes. Noise emitted from the opening to the plenum, duct borne supply and return noise, cabinet radiation, and break-out noise must all be considered along with attenuation of ductwork.



Quiet VAV Terminal Installation Guideline

Engineers can minimize the sound contribution of air terminals to an occupied space through good design practice. Typical noise control measures for air terminal units are:

1. Locating units above spaces such as corridors, work rooms, or open plan office areas. Do not locate VAV units over spaces where the noise should not exceed an NC or RC 35.

2. The air terminal and the return air grille location should be separated as far as possible. Radiated sound can travel directly from the terminal through the return air grille without the benefit of ceiling attenuation. Locating the units at least 5 ft. away from an open return air grille located in the ceiling.

3. The use of lined duct work or manufacturers' attenuators downstream of air terminals can help attenuate higher frequency discharge sound. Flexible duct (used with moderation) is also an excellent attenuation element. Vinyl flexible

duct is magic for reducing noise. Practically, three or four feet of flex duct can cut noise as much as 10 or 12 NC points.

4. Designing systems to operate at low supply air static pressure will reduce the generated sound level. This will also provide more energy efficient operation and allow the central fan to be downsized.
5. Sharp edges and transitions in the duct design should be minimized to reduce turbulent airflow and its resulting sound contribution. Installing acoustically lined sheet metal elbows, at the induced (return) air openings in the casings or installing acoustically lined elbows above ceiling return air openings when they must be near, or directly below a terminal unit.
6. Do not place fan-powered boxes of variable volume systems over critical areas. Locate them over corridors or other non-critical spaces. Make sure there is lining downstream from such boxes. The same applies to unit heaters and fan coil units.
7. The air-handling system using VAV boxes must be designed with variable frequency drive primary fan to take advantage of energy conservation and noise issues.
8. Minimize inlet static pressure to VAV box; Select VAV box for minimum noise in 125 Hz band (< 65 dB radiated, <70 dB ducted).
9. Large static pressure drops (typically greater than 1 in.) across VAV boxes are noisy. If possible, design to avoid them. If necessary, provide lined ductwork and/or silencers downstream.
10. Consult ARI Standard 885, "Procedure For Estimating Occupied Space Sound Levels In The Application Of Air Terminals And Air Outlets." This standard provides current application factors for converting rated sound power to a predicted room sound pressure level. It also provides a repeatable and comparable method of both predicting and specifying sound levels.

VAV noise problems are also traced to improper air balancing. For example, air balance contractors commonly balance an air distribution system by setting all damper positions without considering the possibility of reducing the fan speed. The end result is a duct system in which no damper is completely open and the fan is delivering air at a higher static pressure than would otherwise be necessary. If the duct system is balanced with at least one balancing damper wide open, the fan speed could be reduced with a corresponding reduction in fan noise. Lower sound levels will occur if most balancing dampers are wide open or eliminated.

DIFFUSER, GRILLES AND CONTROL DAMPERS SOUND ISSUES

Diffusers and grilles are devices used to deliver to, or return air from, a building space. These are the most noise sensitive of all HVAC products since they are almost always mounted in or directly over occupied spaces. They usually determine the residual background noise level from 125 Hz to 2,000 Hz.

Generally these devices include vanes, bars, tins, and perforated plates to control the distribution of air into the space. When the air flows across these elements, unique noise is generated, that for a particular diffuser or grille design, varies by the 5th to the 6th power of the velocity. Because of the wide variety in diffuser design, and the sizes available, manufacturers publish sound level data in their catalogs. Most manufacturers only provide the NC level that the diffuser noise will reach with different quantities of air flow in a room. In using the manufacturer's data care should be taken to note the data usually applies only to diffusers in an ideal installation. For example placing a damper, even in an open position, behind a diffuser or grill may increase the noise generated by up to 15 dB. In addition, to achieve the catalog rating, the duct attached to the diffuser must be straight for at least 2.5 to 3 equivalent diameters, which is unachievable in many buildings due to inadequate plenum space.

<u>Good design practices:</u>

1. Use the neck size of the properly selected diffusers as a guide. Flow velocity within 10 feet of the neck should not exceed the velocity in the neck.
2. Perhaps the most common and potentially greatest source of noise coming from a diffuser or register is caused by the balancing dampers immediately behind the diffuser. Depending upon how severely the dampers must be closed to balance the system, noise levels can increase from 5 to 20 db above ratings.
3. Lining a duct cannot solve a noise problem if the register is the actual source of the noise, say if fins/bars are vibrating, damper is rattling or air is whistling

through the openings. Generally, if the noise is a rumble, it is the result of a fan or blower. A whistle on the other hand is a result of high frequency sound.

4. Provide 3 diameters straight flex at diffuser inlet or use minimum 6 feet of acoustical flex between low pressure duct and diffuser.
5. Select diffuser design and size based on NC rating and number of diffusers:
   - 2 diffusers: select NC minus 6
   - 4 diffusers: select NC minus 9
   - 8 diffusers: select NC minus 12

There is much confusion about diffuser NC ratings. The rating applies to one diffuser in a room with a "room factor" of usually 10 dB. Beware that some manufacturers use room factors more than 10 dB in rating their diffusers. Such diffusers are noisier than a diffuser with the same NC rating based on a 10 dB factor. The room factor is the difference between the sound power introduced and resulting sound level in the room. It depends on room geometry and absorption in the room. Most rooms have multiple diffusers and room factors different from 10 dB. Thus, the catalog rating must be adjusted to get the expected noise level.

Volume Dampers

An important feature in the air distribution system is its ability to control the amount of air that flows through each segment of the duct and to ensure that the volume of air supplied to each space is tailored to its conditioning needs. To accomplish this, volume dampers are installed to limit the amount of air that is allowed down the duct path. Unfortunately, dampers accomplish their volume control by pinching down the air stream, increasing the pressure and consequently the noise wherever they occur.

Ceiling supply diffusers are normally equipped with volume control dampers located right at the inlet to the diffuser. The airflow noise created by the dampers is essentially exposed directly into the room and the noise is audible even if the dampers are left wide open. In acoustically sensitive applications, locate the dampers well upstream of the outlet within the lined ductwork. The dampers should preferably be located where the diffuser duct branches off the header, or main duct.

MASKING

In buildings where background sound levels are low, it is possible to alleviate a noise problem by adding a more pleasant sound to the environment. A good example is open-plan offices where people can hear activities in other offices and areas. The new sound has the effect of covering up the noise or making it less noticeable. (An important caution is that the system be set to operate at not too loud a level or with a harsh frequency response.)

Many people think of masking as ‘white noise,’ which is characterized by an equal amount of energy in all audible frequencies. More often, however, masking systems stress lower frequency sounds, such as the noise produced by a normal HVAC unit. For example, loudspeakers emitting an HVAC-like sound might be placed between dropped ceilings and structural ceilings.

In fact, any desirable sound can provide masking. Of nature’s many soothing sounds, running water is the most popular. Fountains, in fact, have been found to provide both acoustical and aesthetic enhancements to residential and commercial environments.

CONTROLLING NOISE LEAKS

Penetrations

One of the most commonplace problems caused by HVAC systems has nothing to do with noise and vibration generated by the equipment or duct work. As it is distributed throughout the building spaces, duct work inevitably penetrates the wall, ceilings and floors that are responsible for a room's sound isolation capabilities. If these penetrations are not adequately treated, they allow sound leaks that can render these acoustical barriers completely ineffectual. Three treatments are necessary:

- First, the penetrating duct work should be supported independently from the partition. If a duct rests on the wall it penetrates, any vibration within the duct can be transmitted into the wall itself, which can then provide a large radiating surface to turn the vibration into airborne noise.
- Second, the duct should be resiliently isolated from the surrounding construction as it passes through the partition, to avoid any contact that might transmit the vibration into the wall.

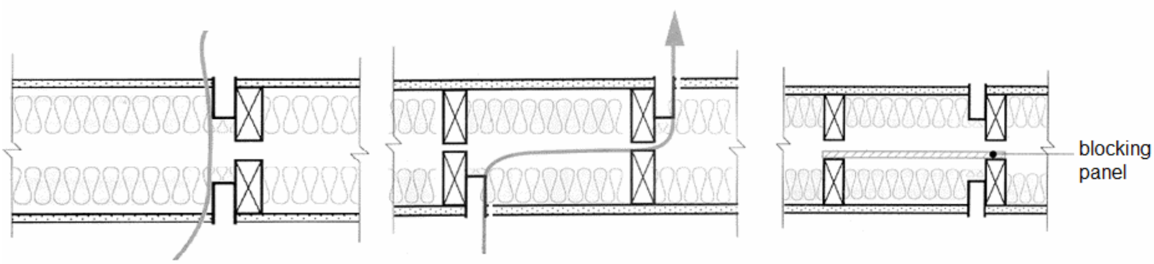
- Third, the area surrounding the penetration should be sealed airtight, using materials that won't allow noise from an adjacent space to leak through the gap.

Two other types of noise leaks are common problems in small buildings; leakage through electrical outlets and leakage around interior partitions can be major.

Electrical and Other Wiring Outlets

Leaks around electrical and other wiring outlets are a common problem, especially when the outlets are back-to-back. One recommended solution is to offset the boxes. This is especially effective when the wall is filled with sound absorbing material. Leaks usually permit only high-frequency sound to pass through. Forcing high-frequency sound to travel a long path through sound absorbing material is a good way to attenuate it, since sound absorbing material is most effective at high frequencies.

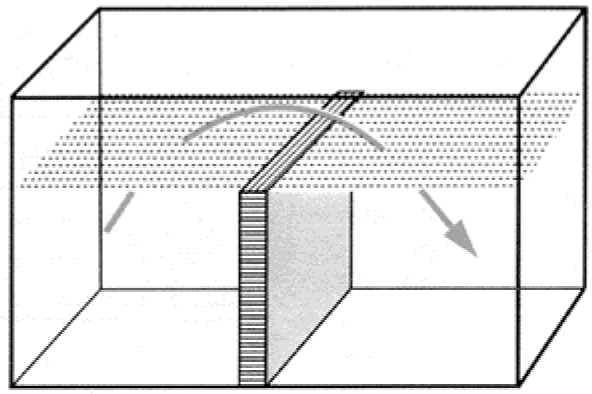
The use of blocking panels is another way of achieving good sound insulation while still having back-to-back outlets.



Electrical boxes back to back   Offset electrical boxes to make path longer   Use blocking panel

Partitions and Ceiling Spaces

This major leak is common with suspended ceilings. Sound is transmitted via the space above the ceiling where the common wall does not extend to the slab above. There are two approaches to dealing with this problem: either the partition is made full height or the attenuation of the path through the plenum and ceiling is increased. For each of these approaches there are variants.



Noise leak through plenum space

Sound leaks can be controlled using sound barrier materials, ceiling boards and other sound absorbing materials over the suspended ceiling.

Partition extension is another way but remember that extending the wall will interfere with airflow and ventilation when the space above the ceiling is used as a return air plenum. Additional ductwork penetrating the barrier will be necessary to restore the airflow. Since this ductwork introduces a path for sound, it should be lined with sound absorbing material.

USE OF ACTIVE NOISE CANCELLATION EQUIPMENT

As one of the noise control methods discussed so far, active noise control involves electronically altering the character of the sound wave in order to reduce its level. In sound cancellation, a microphone measures the noise and a processor generates a mirror image (180° out of phase) of its source. This mirror image is then broadcast in the path of the original sound. Depending on the circumstance, the new sound can cancel enough of the original signal to reduce noise levels up to 40 decibels.

Although sound cancellation is a very powerful noise control tool, it is only practical in confined environments where tonal frequencies are below 500 Hertz (cycles per second). Ventilation ducts are ideal candidates for active noise control because they are enclosed, with the dominant noise often consisting of low-frequency pure tones (associated with the fan characteristics).

Noise cancellation equipment requires 10 to 20 feet of space and can easily cost 10 times what a silencer costs. Nonetheless, in the right situation, it is less expensive than tearing out walls, ceilings, and plumbing to install soundproofing. The system is commonly used in efforts to silence return ducts. The system can be installed in tight places, offers superior silencing of low-frequency noise and causes no static pressure drop.

CONTROLLING NOISE FROM OUTDOOR EQUIPMENT

Air-borne Noise from Air-cooled Chillers

Noise generated from air-cooled chillers may cause noise disturbance to nearby residents. It mainly comes from the air flow noise resulting from air turbulence at condenser fans and compressor noise during running and on/off cycle of refrigerant.

Typical Sound Power Levels of Air-cooled Chillers

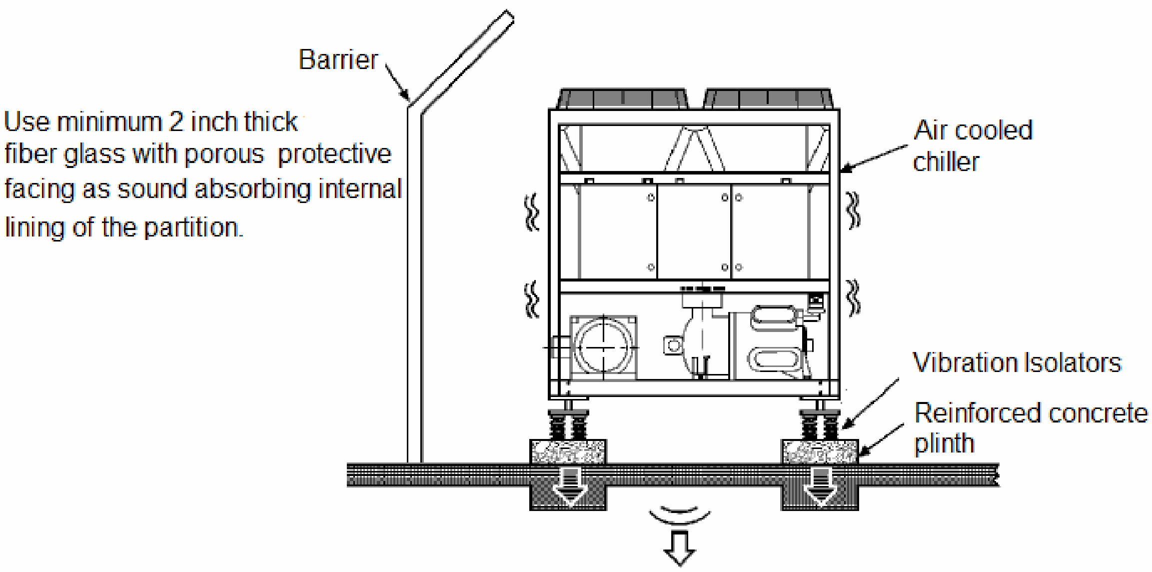
Cooling Capacity (Ton)	Sound Power Level, dB (A)
50	100
100	102
150	103
200	105
250	106
300	106
350	107

The other problem is the structure-borne noise. The vibrations due to chiller operation can be transmitted indoors through building structure at points where the chiller is rigidly fixed to the structure without proper isolation. The vibration transmitted may activate the building structure to generate noise which can be annoying to the occupants of the building.

<u>Remedies</u>

- Use quieter chiller models. If the total sound power level of the machine exceeds 100 dB(A), erect a barrier or partial enclosure between the plant and nearby surroundings to block the noise propagation path;
- Apply barrier or partial enclosure all around when there are noise sensitive receivers in the surroundings;

- Install floating floor so as to reduce air-borne noise transmission through floor slab when the floor underneath is a noise sensitive receiver;
- Provide vibration isolators to support an air-cooled chiller.



Noise Attenuation of Air-cooled Chillers

Air-borne Noise from Water Cooling Towers

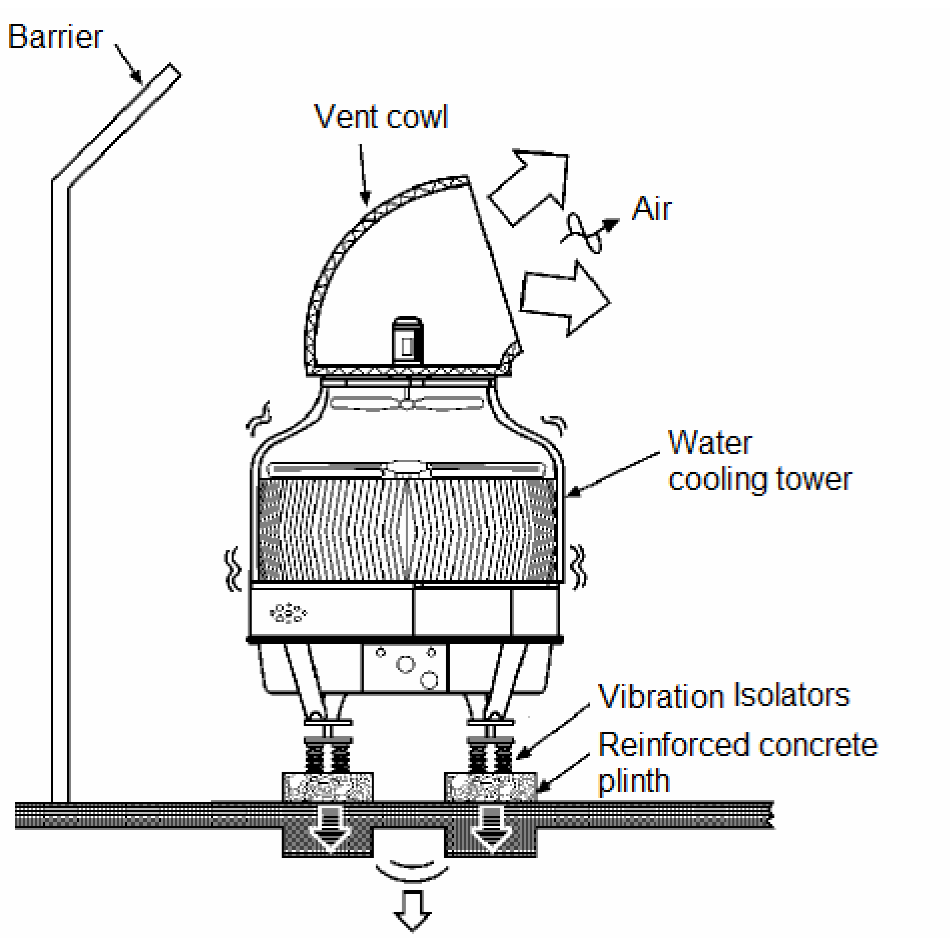
Noise from water cooling towers mainly comes from the air flow noise resulting from air turbulence at condenser fans, mechanical noise of fan(s) and motor(s) and water splashing noise due to water flowing through the tower into the collection basin.

Typical Sound Power Levels of Water Cooling Towers

Horsepower of Fan (hp)	Sound Power Level, dB (A)
10	96
20	99
30	101
40	102
50	103
60	104
80	105

Remedies

- Erect a barrier or partial enclosure between the plant and nearby surroundings to block the noise propagation path;
- Provide acoustic mat on the water surface so as to reduce the water splashing noise;
- Install acoustically lined vent cowl at fan discharge outlet;
- Apply partial enclosure all around when there are noise sensitive receivers all around;
- Provide vibration isolators to support water cooling tower, thereby isolating it from the building structure.



Noise Attenuation of Water Cooling Tower

REDUCING NOISE TRANSMISSION FROM OUTDOOR MACHINES

Sound enclosures, barriers and partitions reduce sound transmission between adjacent spaces.

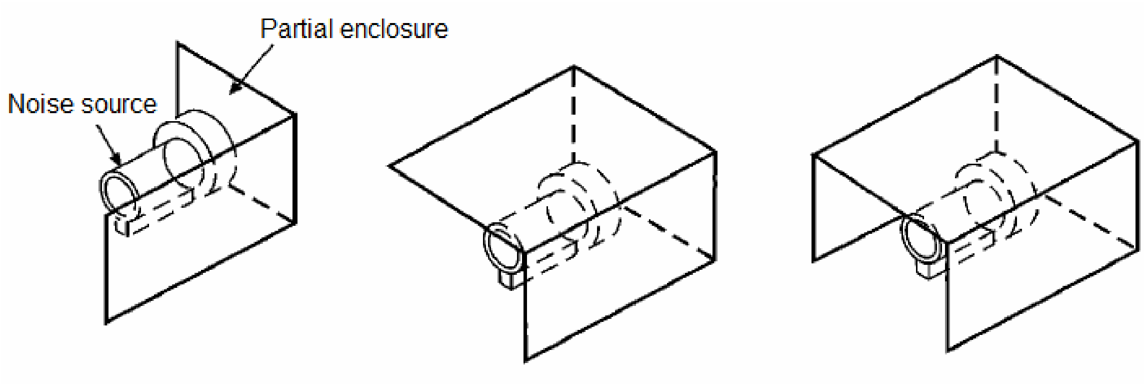
Complete Enclosures

When a noise reduction of 20dB (A) or more is required, it is generally necessary to use a complete enclosure especially if the noise problem is a result of air-borne noise transmission. The enclosure should be internally lined with 2 inch thick sound absorbing material (e.g. fiber glass).

Ventilation of enclosures should not be overlooked as most equipment, such as motors, air cooled condensers or cooling towers require an adequate air supply either to prevent overheating or to enable them to function efficiently.

Partial Enclosures

Partial enclosures are structures erected around a source of noise, but not fully enclosing the source and leaving space for natural ventilation, which will be effective only when there is no line of sight between the noise source and the receiver. The use of partial enclosures has advantages over complete enclosures in terms of cost, accessibility, and ventilation, but design and construction should be done carefully. Ideally, a reduction of up to 20dB (A) can be achieved.



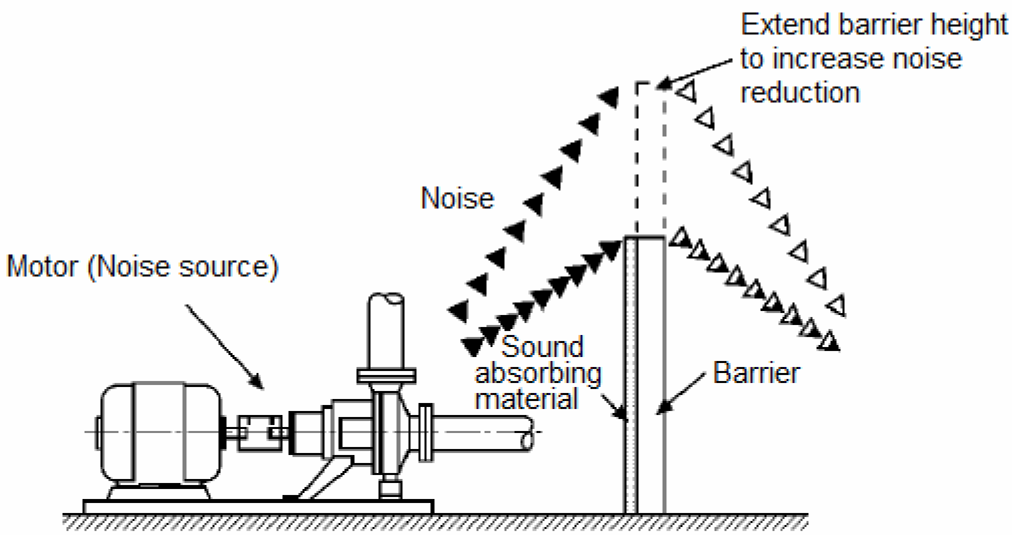
Partial Enclosures

Barriers

To be effective, an acoustic barrier needs to be placed as close as possible either to the noise source or the receiving position. There should be no gap or joint in the barrier through which noise will leak. The surface density of the barrier must be at least

10kg/m^2. Ideally, the length of the barrier should be at least 5 times its height. Line of sight between the source and the receiver must be cut off completely.

A reduction of noise level of between 5dB (A) to 10dB (A) can generally be resulted. Noise reduction will be greater if the barrier is lined with sound absorbing material at the surface of the barrier facing the noise source or is extended as high as possible above the line of sight.



Barriers

ADDITIONAL THOUGHTS

Beyond the initial design, mechanical engineers can try many things to reduce unwanted noise from HVAC systems. Thus the foremost thing is to downsize HVAC equipment and reduce airflow, if possible.

Correctly Estimating the Heat Loads

When designing an HVAC system, the first step is to accurately identify the heat loads generated by the occupants, equipment, lighting and surrounding environment. Although this isn't really an acoustical issue, it's an area that suffers from the "garbage in, garbage out" syndrome. If a project's mechanical and electrical engineers are given bad information about equipment loads or the heat dissipation, the HVAC systems will have a high mismatch between capacity and actual load.

Most of the time, rules of thumb data is used to size HVAC equipment. For instance, heat load is often made on sq-ft basis without considering the geographical location and volume of the space. Engineers often base HVAC sizing decisions on the full nameplate or “connected” load of computers, copiers, printers, and so on; and assume simultaneous operation of such equipment. In fact, most of this equipment operates at a fraction of the nameplate value, and rarely operates simultaneously. Many HVAC designs are based on plug load assumptions on the order of 5 W/sq-ft in office spaces. Invariably, these do not reflect the true operating environment and in actual practice bear little resemblance to their actual power consumption over time. According to an ASHRAE, one W/sq-ft is a reasonable upper bound when equipment diversity and reasonable estimates of the true running load are included.

Over and above, for conservation reasons, the HVAC designers keep very high level of safety margin.

Many factors affect heating or cooling load. A good estimator will measure walls, ceilings, floor space, and windows for the accurate determination of room volume and also accounts for the true R-value of the building insulation, windows, and building materials. A close estimate of the building's air leakage is necessary and a good estimate will also include an inspection of the size, condition (how well joints are sealed and the ducts are insulated), and location of the distribution ducts. Make sure the designer uses the correct design outdoor temperature and humidity for your area.

The following may be noted:

Insist upon a correct system sizing calculation before signing a contract. This service is often offered at little or no cost to homeowners by gas and electric utilities, major heating equipment manufacturers, and conscientious heating and air conditioning contractors. Manual J, published by the Air Conditioning Contractors of America (ACCA), is the most common method in use. Many user-friendly computer software packages or worksheets can simplify the calculation procedure.

Using the correct inputs shall result in right load estimation and optimum size of air handling & refrigeration equipment.

Reducing Airflow

The greater the airflow the greater is the noise. A 20% reduction in airflow will reduce the noise level by approximately 5 dB. Understanding some of the fan fundamentals provides clues to where to look for possible reduction of air flow and consequent reduction in equipment sizes. Fan airflow in CFM is a function of the heat load and also the temperature differential of the supply and return air temperature. For a given air-conditioning load, as the supply air temperature is reduced, the supply air volume is reduced proportionally. The sensible heat gain equation is $Q = 1.08 \times CFM \times \Delta T$

Where:

- Q is sensible heat in Btu/hr
- CFM is the air volume required
- ΔT is the temperature differential of the space setpoint minus the supply air temperature

Let's check this for 1 ton of air-conditioning load. *(Note that 1 ton of refrigeration, Q is equivalent to heat extraction rate of 12000 Btus per hour.)*

Consider two cases:

Case # 1: The room setpoint temperature is 75°F and the supply air temperature is 55°F

Case # 2: The room setpoint temperature is 75°F and the supply air temperature is 45°F

In case # 1, the ΔT is 20°F and therefore the air volume per ton of air-conditioning load shall be:

CFM = 12000 / (1.08 x 20) or = 555/ton

In case # 2, the ΔT is 30°F and therefore the air volume per ton of air-conditioning load shall be:

CFM = 12000 / (1.085 x 30) or = 370/ton

This shows that by lowering the supply-air temperature from the 55°F to 45°F reduces the supply-air volume by 33%. The air handling units, fan terminal units, etc., are reduced to approximately one half the size which means the fan/s and motor/s shall be smaller and therefore it lowers the mechanical sound power levels of the air handling units. Thus the cold air systems deliver much less air volume and are substantially quieter on air sound, often minimizing, or entirely eliminating, sound attenuating requirements.

Summarizing…..

In HVAC system design, controlling three types of sound propagation are important:

1. Air-borne noise is transmitted through both interior partitions and the exterior facade. Sources include mechanical rooms (interior), rooftop equipment, and fans exhausting to the exterior via louvers or stacks. Mechanical equipment noise can also come from neighboring buildings. Control of air-borne mechanical noise can be achieved using appropriate partition construction and detailing, building facade design, site planning, silencers, acoustic louvers, barriers, and selection of quieter equipment.
2. Duct-borne noise travels efficiently through ventilation ducts to any space that is serviced by the system. Duct-borne noise is best reduced at the source by selecting quieter fans or by adding silencers. Duct layouts should incorporate space for silencers. Duct layouts should enhance privacy and sound isolation.
3. Structure-borne noise is caused by vibration from mechanical equipment entering the structure where it can propagate efficiently and be re-radiated as air-borne noise. Structure borne noise is best controlled at the source using rubber or spring isolators and inertia bases.

Majority of the noise in HVAC systems is attributed to the air distribution and the fan systems. The noise control strategy for HVAC systems can be described by the following eight steps.

1. Select a suitable location for mechanical equipment rooms away from noise-sensitive areas of the building.
2. Select fans with low sound power output levels. Ideally the octave band sound power levels should be known for both the sound radiated into the duct and that from the fan casing into the fan room.
3. Vibrationally isolate the fan. This includes both isolating the fan and motor from the floor, and installing a vibration break in the duct immediately adjacent to the fan, to prevent the propagation of vibration energy through the duct walls.
4. Acoustically isolate the fan noise. The equipment room walls, floor, and ceiling must provide a high transmission loss to the airborne noise in the equipment room. One must attenuate the sound propagating down the duct by the

installation of a dissipative silencer immediately after the duct vibration break. Such a silencer is selected to provide adequate attenuation in each octave band and also to not produce excessive airflow noise.

5. Calculate the attenuation of the sound propagating through the duct system so that the sound power entering each room is known.
6. Estimate flow noise sound power levels entering each room. Sharp bends, control mechanisms, grills, diffusers, and combinations of flow-noise-producing devices placed too close together will lead to increased flow-induced noise. Flow-induced noise is very dependent on the velocity of the air flow and can thus be dramatically reduced by reducing this velocity.
7. Calculate the combined sound power entering the room and the expected octave band sound pressure levels in each room. From these, calculate an NC value and compare it with the established noise criteria for the room.
8. If the desired noise criteria are exceeded, further reductions are needed. If fan noise propagating through the duct is the dominant source of noise in the room, an improved dissipative silencer would help. If flow-induced noise is the dominant problem, lining the last few meters of the duct with sound-absorbing material would help. This would require a slightly larger duct to maintain the original internal dimensions and flow velocities. Reducing the air flow velocity or modifying particular flow-noise-producing devices could also be considered.

If the above guidelines are followed properly, it will reduce the noise level significantly. The target noise level from a HVAC system must be chosen with care after studying the scope and the area of application. Remember there is no justification in reducing noise levels below reasonable values. In fact, the total cost of a system increases exponentially as the NC rating is lowered. For this reason, it is recommended to consult an acoustic engineer for critical applications.

Annexure - 1

RULES OF THUMB

1. When specifying sound criteria for HVAC equipment “refer to sound power level, not sound pressure level.”
2. When comparing sound power levels, remember the lowest and highest octave bands are only accurate to about +/-4 dB.
3. Lower frequencies are the most difficult to attenuate.
4. 2 x sound pressure (single source) = +3 dB (sound pressure level)
5. 2 x distance from sound source = -6dB (sound pressure level)
6. +10 dB (sound pressure level) = 2 x original loudness perception
7. When trying to calculate the additive effect of two sound sources, use the approximation (note that logarithms cannot be added directly).

Difference between sound pressure levels	dB to add to highest sound pressure level
0	3.0
1	2.5
2	2.1
3	1.8
4	1.5
5	1.2
6	1.0
7	0.8
8	0.6
9	0.5
10+	0

If two sound levels are identical, the combined sound is three dB higher than either. If the difference is 10 dB, the highest sound level completely dominates and there is no contribution by the lower sound level.

<u>General rules of thumb for controlling noise within a space:</u>

1. You have to at least double the absorption in a space before there is a noticeable difference. Every time you double the absorption, the reverberant noise field is reduced by 3 dB, which is classified as “just perceptible”.

2. Adding absorption to a space can prove a clearly noticeable improvement, if the space is fairly reverberant to begin with. The practical limit for noise reduction from absorption is 10 dB, which sounds half as loud.
3. The improvement will not be as noticeable as you get closer to the noise source.
4. Carpet is not a cure. In fact, it is typically only 15-20% absorptive. It would take four times as much carpet to have the same impact as a typical acoustic material, which is about 80% absorptive.

<u>General rules of thumb for controlling noise between spaces:</u>

1. A wall must extend to the structural deck in order to achieve optimal isolation. Walls extending only to a dropped ceiling will result in inadequate isolation.
2. Sound will travel through the weakest structural elements, which, many times, are doors, windows or electrical outlets.
3. When the mass of a barrier is doubled, the isolation quality (or STC rating) increases by approximately 5 dB, which is clearly noticeable.
4. Installing insulation within a wall or floor/ceiling cavity will improve the STC rating by about 4-6 dB, which is clearly noticeable.
5. Often times, specialty insulations do not perform any better than standard batt insulation.
6. Metal studs perform better than wood studs. Staggering the studs or using dual studs can provide a substantial increase in isolation.
7. Increasing air space in a wall or window assembly will improve isolation.

Facts on dB

1. Human range of hearing starts at 0 dB and is considered safe up to 70 dB. Over and above this level, it is hazardous and can result in permanent hearing damage.
2. Auditory nerves can be permanently damaged from prolonged exposure at 90 dB.
3. 120 dB can cause pain and ringing in the ear.

4. A change in level of 10 dB corresponds approximately to a doubling of perceived loudness or a 5-decibel reduction can cut the risk of hearing loss in half.
5. On the decibel scale, a soft whisper at 2 m distance would have a sound level of about 35 dB (A) while average background sound levels in an office would be about 40 dB (A).
6. A jack hammer 15 m away can raise the sound level to 95 dB (A), while a discotheque can generate noise levels in excess of 110 dB (A).
7. The human ear is not equally sensitive at all frequencies; sounds of the same level but with different frequencies will not be considered equally loud. For example, a sound at 3 kHz and a level of 54 dB will sound about as loud as one at 50 Hz and a level of 79 dB.
8. Doubling the sound power raises the sound power level by 3 dB.
9. Doubling the sound pressure raises the sound pressure level by 6 dB.
10. Depending on the frequency there is a difference on how noise is perceived. For example, for low frequencies (10-100 Hz) the upper pain limit is 130-140 db, for 100-1000 Hz the limit is 120-125 db, and for high frequencies (1000-10000 Hz) the limit is 110-130 db.
11. The sound intensity is attenuated in relation to the distance from the source. For example, it is reduced by 10 db at a distance of 3 m and by 40 db at a distance of 100 m from the source.
12. A basic measure of sound, the sound pressure level (SPL) is expressed in decibels (dB). Sound pressure is the parameter that is normally measured in noise assessments. People's hearing mechanisms respond to pressures that represent the range from the threshold of hearing to the threshold of pain. When the SPL = 0 dB, the acoustic pressure is the same as the threshold of hearing.
13. OSHA Noise Standard indicates that for sound levels of:

- 84 dB A or less - No hearing protection required
- 85-89 dB A - Hearing protection highly recommended
- 90 dB A or greater - Hearing protection is required at the workplace

14. 20 dB is **not** twice as loud as 10 dB. Because a logarithmic scale is used, every 3 dB increase in sound level doubles the perceived sound level for humans. If the energy level is 20, the dB measure is 13 dB.

15. Not all sound pressures are equally loud*. The two identical sound pressures but at different frequencies shall be perceived to be at a different loudness.* This is because the human hearing process is not the same at all frequencies.

Annexure - 2

GLOSSARY OF NOISE TERMS

Absorption Coefficient: A measure of the quantity of sound lost on impinging on a surface. It can be defined as: {1 – (Reflected sound energy / Incident sound energy)}

It is a property of the material on which the sound impinges and is dependant on thickness of material and frequency of the sound.

Absorption: The properties of a material composition to convert sound energy into heat thereby reducing the amount of energy that can be reflected.

Absorptive Silencer: A silencer making use of the absorptive properties of materials to reduce the sound passing through it.

Acoustics: The science of sound; its production, transmission and effects.

Acoustical: The properties of a material to absorb or reflect sound.

Acoustical Analysis: A review of a space to determine the level of reverberation or reflected sound in the space (in seconds) as influenced by the building materials used to construct the space. Also, a study of the amount of acoustical absorption required to reduce reverberation and noise.

Acoustical Environment: The acoustical characteristics of a space or room influenced by the amount of acoustical absorption, or lack of it, in the space.

Ambient Noise: The existing background noise in an area can be sounds from many sources, near and far.

Anechoic Room: A specially constructed room in which as much sound as possible is absorbed at its boundaries. It is typically achieved by using sound absorbing wedges.

Architectural Acoustics: The control of noise in a building space to adequately support the communications function within the space and its effect on the occupants. The qualities of the building materials used determine its character with respect to distinct hearing.

Articulation Class (AC): A single number rating used for comparing acoustical ceilings and acoustical screens for speech privacy purposes. AC values increase with increasing privacy and range from approximately 100-250. This classification supersedes speech privacy Noise Isolation Class (NIC) rating method.

Articulation Index (AI): A measure of speech intelligibility influenced by acoustical environment rated from 0.01 to 1.00. The higher the number the higher the intelligibility of words and sentences understood from 0-100%.

Area Effect: This term suggests the efficiency of sound absorption. Acoustical materials spaced apart can have greater absorption than the same amount of material butted together. The increase in efficiency is due to absorption by soft exposed edges and also to diffraction of sound energy around panel perimeters.

Attenuation: The reduction of sound energy as a function of distance traveled.

Attenuator: It is a noise-reducing device **-** often colloquially known as a 'silencer'.

A-Weighing: An electronic filtering system in a sound meter that allows meter to largely ignore lower frequency sounds in a similar fashion to the way our ears do.

Ambient Noise/Sound: Noise level in a space from all sources such as HVAC or extraneous sounds from outside the space. Masking sound or low-level background music can contribute to ambient level of sound or noise.

Audiogram: Graph of hearing threshold level as a function of frequency (ANSI S3.20-1995: audiogram).

Audiometer: An instrument for measuring hearing acuity.

Barrier: A material that when placed around a source of noise inhibits the transmission of that noise. Also, anything physical or an environment that interferes with communication or listening e.g., a poor acoustical environment can be a barrier to good listening and especially so for persons with a hearing impairment.

Background Noise: The existing noise associated with a given environment, can be sounds from many sources, near and far. (See also Ambient Noise.)

Baffle: A free hanging acoustical sound absorbing unit. Normally it is suspended vertically in a variety of patterns to introduce absorption into a space to reduce reverberation and noise levels.

Baseline Audiogram: The audiogram obtained from an audiometric examination administered before employment or within the first 30 days of employment that is preceded by a period of at least 12 hr of quiet. The baseline audiogram is the audiogram against which subsequent audiograms will be compared for the calculation of significant threshold shift.

BEL: A measurement of sound intensity named in honor of Alexander Graham Bell. First used to relate intensity to a level corresponding to hearing sensation

Boominess: Low frequency reflections. In small rooms, acoustical panels with air space behind can better help control low frequency reflectivity.

Breakout: It is the escape of sound from any source enclosing structure, such as ductwork and metal casings.

Cloud: In acoustical industry terms, an acoustical panel suspended in a horizontal position from ceiling/roof structure; similar to baffle but in a horizontal position.

Cocktail Party Effect: Sound in a noisy crowded room generated mostly by conversation. Levels rise and fall as people compete with one another to be heard. Perception of speech can be nearly impossible in high levels of noise.

Cochlea: A snail shaped mechanism in the inner ear that contains hair cells of basilar membrane that vibrate to aid in frequency recognition.

Continuous Noise: Noise with negligible small fluctuations of level within the period of observation (ANSI S3.20-1995: stationary noise; steady noise).

Crest Factor: Ten times the logarithm to the base ten of the square of the wideband peak amplitude of a signal to the time-mean-square amplitude over a stated time period. Unit dB (ANSI S3.20-1995: crest factor)

Cross talk: It is the transfer of airborne noise from one area to another via secondary air paths, such as ventilation ductwork or ceiling voids.

Cut off Frequency: It is the frequency at which performance of an acoustic item or material starts to fall below normal or below criterion. Applied to anechoic wedge treatment it refers to the frequency below which the absorption coefficient is worse than 0.99.

Cycle: In acoustics, the cycle is the complete oscillation of pressure above arid below the atmospheric static pressure.

Cycles per second: The number of oscillations that occur in the time frame of one second. Low frequency sounds have fewer and longer oscillations.

Decibel (dB): One tenth of a Bel; a Bel being a unit of amplification corresponding to a tenfold increase. In terms of Sound Level Measurements it is related to datum levels as follows:

- For Sound Pressure Level (SPL) datum = 2 X 10^{-5} Pascal
- For Sound Power Level (SWL) datum = 1 X 10^{-12} Watts
- S.P.L. (dB) = 20 log [P / (1 x 10^{-5})], where P is the amplitude of the Sound Pressure Waves concerned measured in Pascal.
- S.W.L. (dB) = 10 log [W / (1 x 10^{-12})], where W is the sound power radiated by the source concerned measured in watts.

Decibel, A-Weighted (dBA): Unit representing the sound level measured with the A-weighting network on a sound level meter.

Decibel, C-Weighted (dBC): Unit representing the sound level measured with the C-weighting network on a sound level meter.

Deaf: Loss of auditory sensation with or without use of assistive listening device

Diffusion: The scattering or random reflection of a sound wave from a surface. The direction of reflected sound is changed so that listeners may have sensation of sound coming from all directions at equal levels.

Directivity Factor: When sound radiates from any source sound levels can be higher in certain directions than others. This is called 'Directivity'. Directivity Factor is the ratio of the increased level to the average value.

Directivity Index: Is directivity factor expressed in decibels (dB).

It is usually designated by DIO where O is the angle between the axis of the source and the direction of the measuring point.

Discrete Frequency: It is a single frequency signal or a single frequency noise sufficiently dominant over other frequencies to be distinctly audible.

Dose: The amount of actual exposure relative to the amount of allowable exposure, and for which 100% and above represents exposures that are hazardous. The noise dose is calculated according to the following formula:

$D = \{C_1/T_1 + C_2/T_2 + ... + C_n/T_n\}$ H 100 where Cn = total time of exposure at a specified noise level T_n = exposure time at which noise for this level becomes hazardous

Dynamic Insertion Loss (DIL): It is a measure of the acoustic performance of an attenuator when handling the rated flow. Not necessarily the same as Static Insertion Loss because it may include regeneration and / or other velocity effects and will account for the effects of the actual fluid and fluid conditions for which the silencer is designed.

Echo: Reflected sound producing a distinct repetition of the original sound. Echo in mountains is distinct by reason of distance of travel after original signal has ceased.

Echo Flutter: Short echoes in small reverberate spaces that produce a clicking, ringing or hissing sound after the original sound signal has ceased. Flutter echoes may be present in long narrow spaces with parallel walls.

Effective Noise Level: The estimated A-weighted noise level at the ear when wearing hearing protectors. Effective noise level is computed by (1) subtracting de-rated NRR from C-weighted noise exposure levels, or (2) subtracting de-rated NRR minus 7 dB from A-weighted noise exposure levels. Unit, dB

End Reflection: End reflection occurs when sound energy radiates from a hole. The sudden expansion to atmosphere causes some low frequency noise to be reflected back towards the source. Expressed in decibels (dB), the effect is dependent on hole size and frequency. Maximum at lowest frequency from smallest hole

Equal-Energy Hypothesis: A hypothesis stating that equal amounts of sound energy will produce equal amounts of hearing impairment, regardless of how the sound energy is distributed in time.

Equal Loudness Contours: Curves represented in graph form as a function of sound level and frequency which listeners perceive as being equally loud. High frequency sounds above 2000 Hz are more annoying. Human hearing is less sensitive to low frequency sound. (See also Phon).

Equivalent Continuous Sound Level: Ten times the logarithm to the base ten of the ratio of time-mean-square instantaneous A-weighted sound pressure, during a stated time interval T, to the square of the standard reference sound pressure. Unit, dB

Exchange Rate: An increment of decibels that requires the halving of exposure time or a decrement of decibels that requires the doubling of exposure time. For example, a 3-

dB exchange rate requires that noise exposure time be halved for each 3-dB increase in noise level; likewise, a 5-dB exchange rate requires that exposure time be halved for each 5-dB increase.

Free Field: It is a sound field, which is free from all reflective surfaces or where there are no obstructions. A simulated free field can be produced inside an anechoic room.

Frequency: For a function periodic in time, the reciprocal of the period. Unit, hertz (HZ) (ANSI S1.1-1994: frequency)

Frequency (Hz) – Sound: It is the number of sound waves to pass a point in one second.

Frequency – vibration: It is the number of complete vibrations in one second.

Frequency Analysis: An analysis of sound to determine the character of the sound by determining the amount of sounds of various frequencies that make up the overall sound spectrum i.e. higher frequency sound or pitch vs. low frequency.

Flanking transmission: It is the transfer of sound between any two areas by any indirect path, usually structural. It can also apply to noise transmitted along the casing of a silencer.

Hearing Impairment: Hearing impairment means a hearing loss of mild, moderate or severe degree as opposed to "deafness" which is generally described as little or no residual hearing with or without the aid of an assistive listening device

Hearing Range:

- 16 - 20000 Hz (Speech Intelligibility)
- 600 - 4800 Hz (Speech Privacy)

Hearing Threshold Level (HTL): For a specified signal, amount in decibels by which the hearing threshold for a listener, for one or both ears, exceeds a specified reference equivalent threshold level. Unit, dB (ANSI S1.1-1994: hearing level; hearing threshold level)

Helmholz Resonance: A resonance created by the mass of a "plug" of fluid acting on the resilience of "spring" of a volume of fluid; e.g. a "plug" of air in a bottleneck resonates on the volume when one blows across the neck. This principle can be used in silencers etc.

Hertz (Hz): Frequency of sound expressed by cycles per second.

Impulse: Product of a force and the time during which the force is applied; more specifically, impulse is the time integral of force from an initial time to a final time, the force being time-dependent and equal to zero before the initial time and after the final time (ANSI S1.1-1994: impulse).

Impulsive Noise: Impulsive noise is characterized by a sharp rise and rapid decay in sound levels and is less than 1 sec in duration. For the purposes of this document, it refers to impact or impulse noise.

Insertion Loss: The reduction of noise level by the introduction of noise control device established by the substitution method of test, or by "before and after" testing. The term can be applied to all forms of treatment including silencers and enclosures. (See also Dynamic Insertion Loss and Static Insertion Loss)

Insulation (Sound): It is the property of a material or partition to oppose sound transfer through its thickness.

Inverse Square Law: The reduction of noise with distance in terms of decibels, it means a decrease of 6dB for each doubling of distance from a point source when no reflective surfaces are apparent. This is only applied in free field conditions where the source is small in comparison with the distance.

Intensity: (See loudness).

Intermittent Noise: Noise levels that are interrupted by intervals of relatively low sound levels.

Inverse Square Law: Sound levels full off with distance traveled. Sound level drops off 6 dB from source point for every doubling of distance.

Loudness: A listener's auditory impression of the strength of a sound. The average deviation above and below the static value due to sound wave is called sound pressure. The energy expended during the sound wave vibration is called intensity and is measured in intensity units. Loudness is the physical resonance to sound pressure and intensity.

Laminar Flow: It is colloquially used to describe the preferred state of airflow. Strictly means undisturbed flow at very low flow-rates where the air moves in parallel paths.

Level Difference: The difference in Sound Pressure Levels between two positions, e.g. inside and outside an enclosure. (This is not the same as Insertion Loss, Transmission Loss or Sound Reduction Index although in some circumstances they may be similar.)

Masking: The process by which extra sound is introduced into an area to reduce the variability of fluctuating noise levels or the intelligibility of speech.

Mass Law: Heavy materials stop more noise passing through them than light materials. For any airtight material there will be an increase in its "noise stopping" ability of approximately 6dB for every doubling of mass per unit area.

Near Field: It is the area close to a large noise source where the inverse square law does not apply.

Noise: Undesired or unwanted sound. By extension, noise is any unwarranted disturbance within a useful frequency band, such as undesired electric waves in a transmission channel or device.

Noise Criterion Curves (NC): An American set of curves based on the sensitivity of the human ear. They give a single figure for broadband noise. It is used for indoor design criteria.

Noise Rating Curves (NR): A set of curves based on the sensitivity of the human ear. They are used to give a single figure rating for a broad band of frequencies. It is used in Europe for interior and exterior design criteria levels. They have a greater decibel range than NC curves.

Noise Isolation Class (NIC): A single number rating of the degree of speech privacy achieved through the use of an Acoustical Ceiling and sound absorbing screens in an open office. NIC has been replaced by the Articulation Class (AC) rating method.

Noise Reduction: The amount of noise that is reduced through the introduction of sound absorbing materials. This is established by measuring the difference in sound pressure levels adjacent to each surface

Noise Reduction Coefficient (NRC): The NRC of an acoustical material is the arithmetic average to the nearest multiple of 0.05 of its absorption coefficients at 4 one-third octave bands with center frequencies of 250, 500, 1000, 2000 Hertz.

Noise Reduction Rating (NRR): The NRR, which indicates a hearing protector's noise reduction capability, is a single-number rating that is required by law to be shown on the label of each hearing protector sold in the United States. Unit, dB

Octave: A pitch interval of 2:1. The tone whose frequency is twice that of the given tone

Octave Bands: It is a convenient division of the frequency scale. Sounds that contain energy over a wide range of frequencies are divided into sections called bands. Identified by their center frequency, typically 63 125 250 500 1000 2000 4000 8000 Hz.

PHON (Loudness contours): A subjective impression of equal loudness by listeners as a function of frequency and sound level (dB). An increase in low frequency sound will be perceived as being much louder than an equivalent high frequency increase.

Pink Noise: It is noise of a statistically random nature, having an equal energy per octave bandwidth throughout the audible range.

Pitch: The perceived auditory sensation of sounds expressed in terms of high or low frequency stimulus of the sound.

Presbycusis: The loss of hearing due primarily to the aging process. High frequency loss is frequently a result of early hearing loss.

Pressure Drop: The difference between the pressure upstream and down stream of a silencer at given flow conditions. If the silencer is to be installed other than in a duct system of constant cross-section care must be taken with regard to measuring positions and methods to allow for difference in velocity head.

Pulse Range: Difference in decibels between the peak level of an impulsive signal and the root-mean-square level of a continuous noise.

Pure Tone: It is a single frequency signal.

Random Noise: A confused noise comprised from large number of sound waves, all with unrelated frequencies and magnitudes.

Reactive Attenuator or Resonant Attenuator / Silencer: An attenuator, in which the noise reduction is brought about typically by changes in cross section, chambers and baffles sections.

Reflection: The amount of sound wave energy (sound) that is reflected off a surface. Hard non porous surfaces reflect more sound than soft porous surfaces. Some sound reflection can enhance quality of signal of speech and music.

Regeneration: The noise generated by airflow turbulence. The noise level usually increases with flow speed.

Resonance: The emphasis of sound of a particular frequency.

Resonant Frequency (Hz): It is the frequency at which resonance occurs in the resilient system.

Reverberation: Reflected sound in a room, which decays after the sound source has stopped. **Reverberation Room or Chamber:** A calibrated room specially constructed with sound reflective walls, e.g., plastered concrete. The result is a room with a "long smooth echo", in which a sound takes a long time to die away. The sound pressure levels in this room are very even.

Reverberation Time: The time taken for sound to decay 60 dB to 1 / 1,000,000 of its original sound level after the sound source has stopped. Sound after it has ended will continue to reflect off surfaces until the wave loses enough energy by absorption to eventually die out. Reverberation time is the basic acoustical property of a room, which depends only on its dimensions and the absorptive properties of its surfaces and contents. Reverberation has an important impact on speech intelligibility.

Room Constant: It is the sound absorbing capacity of a room, usually expressed in m^2.

Sabin: It is a unit of absorption comprising the sum of the products of absorption coefficients and areas of the materials of a room. It must be qualified by the units of area used e.g. Sq-m Sabines.

Sabine Formula: A formula developed by Wallace Clement Sabine that allows designers to plan reverberation time in a room in advance of construction and occupancy. Defined and improved empirically, the Sabine Formula is $T = 0.049(V/A)$ where T = Reverberation time or time required (for sound to decay 60 dB after source has slopped) in seconds. V = Volume of room in cubic feet. A = Total square footage of absorption in sabins. It becomes inaccurate when absorption is high.

Septum: A thin layer of material between 2 layers of absorptive material i.e. Foil, lead, steel, etc. that prevents sound wave from passing through absorptive material.

Signal to Noise Ratio: The sound level at the listener's ear of a speaker above the background noise level. The inverse square low impacts on the S/N ratio. Signal to Noise Ratios are important in classrooms and should be in range of + 15 to +20 dB.

Significant Threshold Shift: A shift in hearing threshold, outside the range of audiometric testing variability (5 dB), that warrants follow-up action to prevent further hearing loss. NIOSH defines significant threshold shift as an increase in the HTL of 15 dB or more at any frequency (500, 1000, 2000, 3000, 4000, or 6000 Hz) in either ear that is confirmed for the same ear and frequency by a second test within 30 days of the first test.

Silencer: It is colloquialism for attenuator.

Sound: Oscillation in pressure, stress, particle displacement, particle velocity, etc. in a medium with internal forces (e.g., elastic or viscous), or the superposition of such propagated oscillations.

Sound Absorption: The property possessed by materials, objects and air to convert sound energy into heat. Sound waves reflected by a surface cause a loss of energy. The energy not reflected is called its absorption coefficient.

Sound Absorption Coefficient: The fraction of energy striking a material or object that is not reflected. For instance, if a material reflects 70% of the sound energy incident upon its surface, then its Sound Absorption Coefficient would be 0.30. SAC = absorption/area - sabins per sq. ft.

Sound Insulation: It is the property of a material or partition to oppose sound transfer through its thickness.

Sound Intensity: Average rate of sound energy transmitted in a specified direction at a point through a unit area normal to this direction at the point considered. Unit, watt per square meter (W/m^2); symbol, I (ANSI S1.1-1994: sound intensity; sound-energy flux density; sound-power density)

Sound Intensity Level: Ten times the logarithm to the base ten of the ratio of the intensity of a given sound in a stated direction to the reference sound intensity of 1 picoWatt per square meter (pW/m^2). Unit, dB

Sound Level: A subjective measure of sound expressed in decibels as a comparison corresponding to familiar sounds experienced in a variety of situations.

Sound Level Meter (Noise Meter): It is an instrument for measuring sound pressure levels. It can be fitted with electrically weighting networks for direct read-off in dBA, dBB, dBC and octave or third octave bands.

Sound Power: It is a measure of sound energy in watts. It is a fixed property of a machine, irrespective of environment.

Sound Power Level (SWL or PWL): It is the amount of sound output from a machine, etc., cannot be measured directly. It is expressed in decibels of SWL.

Sound Pressure: Root-mean-square instantaneous sound pressure at a point during a given time interval. Unit, Pascal (Pa); (ANSI S1.1-1994: sound pressure; effective sound pressure)

Sound Pressure Level (SPL): It is a measurable sound level that depends upon environment. It is a measure of the sound pressure at a point in N/m^2. It is expressed in decibels of SPL at a specified distance and position. It can also be considered as a measure of intensity of terms of Sound Energy per unit area at the point considered, but is not a vector (i.e. directional) as Sound Intensity strictly is.

SPL Direct Field: It is the sound radiating directly from the source(s) to the receiver without reflection.

SPL Reverberant Field: It is the sound reaching the receiver after one or more reflections.

Sound Level Meter: A device that converts sound pressure variations in air into corresponding electronic signals. The signals are filtered to exclude signals outside frequencies desired.

Sound Reduction Index (SRI): A set of values measured by a specific test method to establish the actual amount of sound that will be stopped by the material, partition or panel, when located between two reverberation rooms. Average SRI can be calculated by averaging the set of values in the sixteen third octave bands from 100 Hz to 3150 Hz. It is a property of the material(s) or construction, not directly measurable in the field.

Sound Transmission Class (STC): The preferred single class rating system designed to give the sound insulation properties of a structure for the rank ordering of a series of structures

Spectrum: The description of a sound wave's components of frequency and amplitude.

Sound Spectrum: It is the separation of sound into its frequency components across the audible range of the human ear.

Speech: The act of speaking. A child learns to speak by imitating those people around him. It is important that a child can hear proper speech. 'We speak what we hear.'

Speech Intelligibility: The ability of a listener to hear and correctly interpret verbal messages. In a classroom with high ceilings and hard parallel surfaces such as glass and tile, speech intelligibility is a particular problem. Sound bounces off walls, ceilings and floor, distorting the teacher's instructions and interfering with students' ability to comprehend.

Speech Privacy: The degree to which speech is unintelligible between offices. Three ratings are used: Confidential, Normal (Non Obtrusive), Minimal.

Standing Waves: These occur due to room geometry. Sound levels at some locations in the room at certain frequencies will be intensified by additive interference of successive waves, and in other locations reduced by cancellation.

Static Insertion Loss (SIL): The Insertion Loss of an attenuator under static (no flow) conditions (c.f. Insertion Loss, Dynamic Insertion Loss).

Threshold Shift: A partial loss of hearing caused by excessive noise, either temporary or permanent, in a person's threshold of audibility.

Threshold of Audibility or Hearing: The minimum sound levels at each frequency that a person can just hear.

Threshold of pain: The sound level at which a person experiences physical pain. (Typically 120 dB)

Third Octave Bands: It is a small division of the frequency scale, three to each octave. It enables more accurate noise analysis.

Time Weighted Average (TWA): The averaging of different exposure levels during an exposure period. For noise, given an 85-dBA exposure limit and a 3-dB exchange rate, the TWA is calculated according to the following formula: TWA = 10.0 H Log(D/100) + 85 where D = dose.

Tinnitus: 'Ringing in the ears' of which there is no observable cause.

Transmission Loss: American preferred description for sound reduction index. A set of values measured by a specific test method to establish the actual amount of noise that will be stopped by the material, partition or panel when placed between two reverberation rooms.

Turbulent Flow: A confused state of airflow that may cause noise to be generated inside, for example, a ductwork system.

Varying Noise: Noise, with or without audible tones, for which the level varies substantially during the period of observation

Velocity Head or Velocity Pressure (Pv): It is a measure of the inertia of a flowing fluid used in assessing pressure losses in duct systems and/or silencers for air at atmospheric conditions.

Pv = $v^2/4$; where, Pv is in millimeters of water and v is the velocity in meters/seconds. Or

Pv = v^2 / 3970; where, Pv is in inches of water and v is the velocity in feet/minute. **Volume:** The cubic space of a room bounded by walls, floors, and ceilings determined by Volume = length x Width x Height of space. Volume influences reverberation lime.

Wavelength: Sound that passes through air produces a wavelike motion of compression and rarefaction. Wavelength is the distance between two like points on a wave shape, e.g. distance from crest to crest. Length of sound wave varies with frequency. Low frequency equals longer wavelengths.

White Noise: It is noise of a statistically random nature having an equal energy level per Hertz throughout the audible range.

*****

Annexure – 3

Sound Power & Sound Power Levels

The table below indicates the Sound Power and the Sound Power Level from some common (and some not so common) sources.

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
25 to 40,000,000	195	Saturn Rocket
100,000	170	Jet engine
10,000	160	Turbojet engine 3200kg thrust
1000	150	4 propeller airliner
100	140	Large centrifugal fan, 800000 m^3/h
10	130	• 75 piece orchestra • Axial fan, 100000 m^3/h
1	120	• Large chipping hammer • Human pain limit
0.1	110	• Large aircraft 150 m over head • Centrifugal fan, 25000 m^3/h • Blaring radio

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
0.01	100	• Large air compressor • Air chisel • Magnetic drill press • High pressure gas leak • Banging of steel plate • Drive gear • Car on highway • Normal fan
0.001	90	• Axial ventilating fan (2500 m^3/h) • Cut-off saw • Hammer mill • Small air compressor • Grinder • Heavy diesel vehicle • Heavy city traffic • Lawn mover • Maximum sound up to 8 hour • OSHA1) criteria - engineering or administrative noise controls)

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
		• Jackhammer at 15 m • Bulldozer at 15 m
0.0001	80	• Voice - shouting • Maximum sound up to 8 hour (OSHA criteria - hearing conservation program) • Pneumatic tools at 15 m • Alarm clock • Buses, trucks, motorcycles at 15 m • Dishwasher
0.00001	70	• Voice - conversational level • Car at 15 m • Vacuum cleaner at 3 m
0.000001	60	• Large department store • Busy restaurant or canteen
0.0000001	50	Room with window air conditioner
0.00000001	40	Voice, low
0.000000001	30	• Voice - very soft whisper

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
		• Room in a quiet dwelling at midnight
0.0000000001	20	Noise at ear from rustling leaves
0.00000000001	10	Quietest audible sound for persons under normal conditions
0.000000000001	0	Quietest audible sound for persons with excellent hearing under laboratory conditions