PDHonline Course M206 (4 PDH)

Overview of Noise Control and HVAC Acoustics in Buildings

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Course Content

Sound may be defined as vibrations or pressure changes in an 'elastic' medium, which are capable of being detected by the ear. Elastic means that the particles of the medium return to their original position after disturbance by the vibrational wave.

These vibrations travel through solids, liquids and gases but the normal process of hearing depends on their ultimate transmission through air so that the ear drum is set in vibration and a sequence of events we call hearing begins.

The terms noise and sound are often used interchangeably, but the term sound is generally used to describe useful communication or a pleasant sound; whereas the term noise is generally used to describe an unwanted sound which interferes with speech.

In examining building acoustics we are mainly concerned with sound vibrations through the air, whereas in the subject of noise control we are equally concerned with vibrations transmitted through solid materials such as pipe work, concrete plinths and building structures.

This course provides a brief introduction to the ways the noise is characterized, measured and predicted. This is followed by discussion on some key measures to control HVAC noise in the buildings. To emphasize the simplicity of the approach, equations are kept to a minimum.

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Source of Sound

Noise/sound is created when the air is mechanically vibrated. The origin of sound is usually the vibrations of a solid body such as a rotating machine causing vibrations in its associated parts.

A pump motor for example rotates inside a casing and this rotates the impeller also inside a casing. These two rotating items run on bearings which transfer vibrations to the whole pump unit which in turn vibrates the pump mounting bracket(s).

If part of the rotating machine is out of balance or damaged it is quite likely that the noise will be increased as the intensity of vibration increases.

Sound waves can also be caused by air turbulence, by an explosive expansion of air or by a combination of vibration and air turbulence.

Noise/sound (the vibrating air) is radiated in all directions from the source producing it. This means that all people at the same distance radially from the source can hear the sound equally. If there was no air for the sound to travel through there would be no noise, for instance if a bell is rung in a vacuum then there would be no sound.

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How Is Sound Described?

1) The term used to describe sound is the **decibel (dB).** The higher the decibel number, the greater the sound pressure on the ear drum thus the louder the sound.

2) Pitch - Hertz (Hz) - a unit of measure for sound frequency. One Hz = one cycle per second. The more cycles per second, the faster the eardrum vibrates, the higher the sound pitch. The human ear can perceive sounds with frequencies ranging from 20 Hz up to 20000 Hz.

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.

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Sound Frequency

The frequency 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)

   = 0.002 s

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

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Wavelength

The wavelength of sound is the distance between analogous points of two successive waves or is the distance the sound travels in one cycle time. *The higher the frequency the shorter shall be the wavelength.*

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 diagram below illustrates the difference between loud and soft sounds.
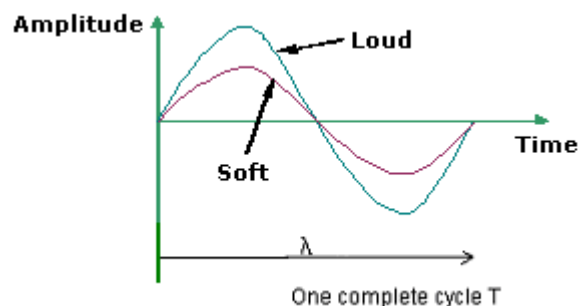
λ = c / f , where

λ = wavelength (m)

c = speed of sound (m/s) (*in air at normal atmosphere and 0°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.

$λ = (331.2$ m/s$) / (500$ Hz$)$

= 0.662 m

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

Sound Power and Sound Pressure

The difference between sound power and sound pressure is critical to the understanding of acoustics. To understand the concept of sound power and sound pressure, an analogy can be made between a noise source and a light bulb. A light bulb is rated to dissipate a particular number of watts of power. The bulb will always dissipate the same amount of power independent of its surroundings or the environments in which it is located. As an example, a 60W light bulb consumes 60W no matter where it is screwed in. However, the same bulb may appear to brighten a room covered with shiny reflecting walls more than another with dull black walls. The effects of distance, volume of space, absorbing and reflecting surfaces etc. will combine to determine the resulting lighting level at any point.

This is an exact parallel to the acoustical situation. *Sound Power is the amount of acoustical power a source radiates in a non-reflective environment and is measured in Watts. Sound power is a fixed property of a machine irrespective of the distance and environment. On the other hand, Sound pressure is related to how loud the sound is perceived to be, and 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.

What is threshold of hearing?

The threshold of hearing is the lowest sound the human ear can perceive.

- The lowest sound that can be perceived by the human ear is 10^{-12} Watt and this corresponds to 0 dB.

- The lowest sound pressure possible to hear is approximately $2 * 10^{-5}$ Pa (20 micro Pascal) and this corresponds to 0 dB.

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

Measure of Sound – Decibel (dB)

The ear is capable of hearing a very large range of sounds: from a soft whisper at 0.000000001 Watts to a NASA rocket at 40,000,000 Watts. To deal with such a vast range, "decibel" (dB) notation is used to represent sound levels along a logarithmic scale in which 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.

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Sound Power Level (Lw)

Sound Power Level is a logarithmic comparison of sound power output by a source to a reference sound source and is expresses in dB as a ratio relative to some reference level which is threshold of hearing –

10^{-12} W

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

Where:

1) Lw is sound power level in decibel (dB)

2) W $_{ref}$ is 10^{-12} W;

3) W $_{source}$ is sound power in W

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

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Sound Pressure Level (Lp)

The threshold of hearing is assumed to correspond to pressure fluctuations of 2 *10^{-5} Pa (20 micro Pascal's) and is given the value of 0 dB; (Note that the lowest sound pressure possible to hear is approximately 2 *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

1) Lp = Sound Pressure Level (dB)

2) P_{ref} = 2 x 10^{-5} – Reference Sound Pressure (Pa);

3) P = sound pressure (Pa)

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

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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.

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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 (in the absence of any reflecting surfaces), 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.

A vibrating surface, such as the one shown below will emit sound waves not only in front and behind the surface but also in all directions. Near to the surface, which is emitting the sound, the shape of the sound wave approximates to the shape of the surface but as the pressure waves expand they will become virtually spherical.
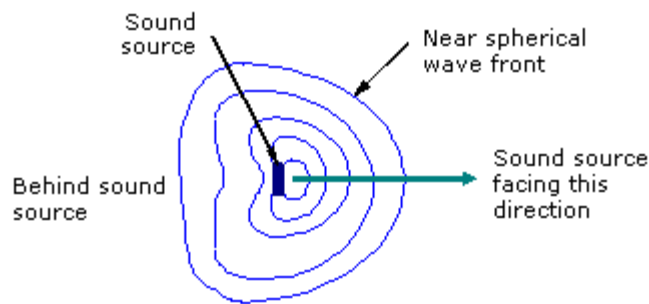
Sound Waves from a Source

Most of the individual sources of sound emit sound in all directions but may emit sound with greater intensity in one direction - examples being the human voice and wind instruments. This means that the expanding sound waves, though spherical, may vary in intensity in different parts of their frontal area.

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What measurements are useful?

Fluctuating pressures that are in the frequency range of 20-20,000 Hz can be classified as audible sound for the human ear. The human ear is more 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 and sound pressure 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.

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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.

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A-weighted Measurement, dB A

Human ear is extremely clever but it does have its limitations. If the ear were to be presented with identical sound pressures but at different frequencies it would perceive them to be at a different loudness. This is because the human hearing process is not the same at all frequencies and therefore the noise measurements are often adjusted or weighted, as a function of frequency to account for human perception and sensitivities. Sound Level Meters are programmed to take this into account. The meter, measures the sound pressures in all the different frequencies, corrects them for the human ears limitations 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

Octave bands

The range of sound that can be heard by humans is very large, from the fluttering of a butterfly's wings near your ear to a jet fighter's painful roar at 200 ft away. However, the human hearing system is NOT able to distinguish between two separate sounds with frequencies too close to each other. A slight change in the frequency of the tone will not be audible unless the frequency change is greater than a defined value. To better describe the intensity and quality of a noise, the audible frequencies are divided into 220 ranges and a sound power level quoted for each; the higher the frequency the wider the range.

Octave Bands is a convenient division of the frequency scale. Sounds that contain energy over a wide range of frequencies are divided into sections called bands. This frequency range is usually separated into eight unequal segments commonly referred to 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz. When a greater frequency resolution is needed, a unit with less value such as 1/3 octave can be used.

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.
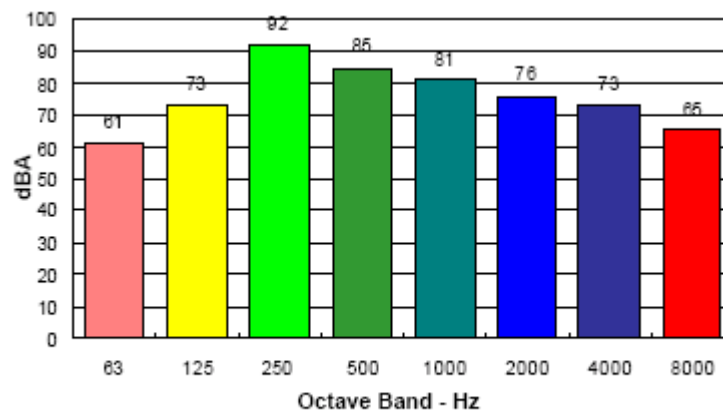

SOUND PRESSURE LEVELS BY OCTAVE BANDS

Typical sources in buildings emit sound at many frequencies. A graph showing the energy at each frequency is called a spectrum. 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 metre 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 Machine (dB)
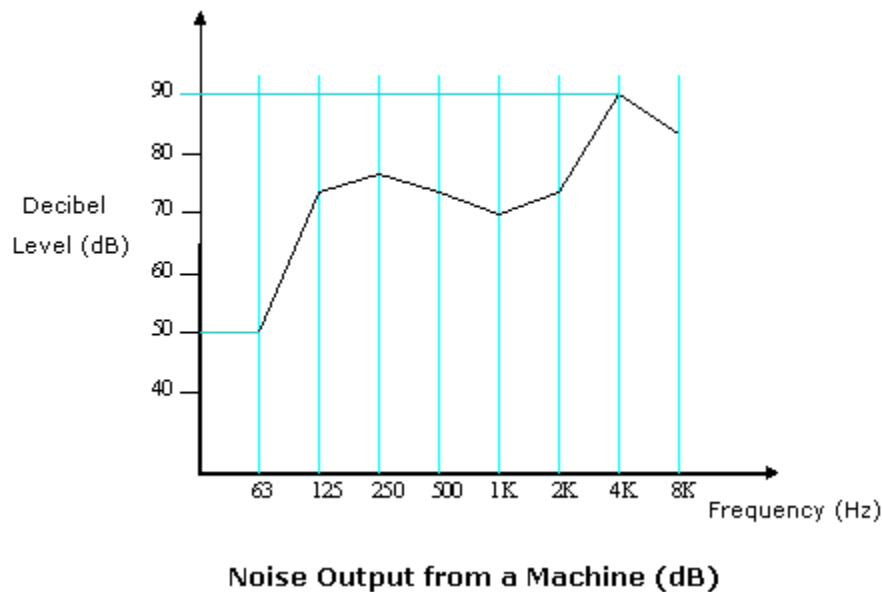
It will be seen from the above graph that the maximum noise output of 90 dB occurs at 4K (4000) Hertz. Also the lowest noise level of 50 dB occurs at a frequency of 63 Hz.

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.
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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

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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.

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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.

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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

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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

1) L_I = sound intensity level (dB)

2) I = sound intensity (W/m^2)

3) $I_{ref} = 10^{-12}$ ----- 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

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

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

$= \underline{40} \text{ dB}$

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.*

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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)

$\rho = 1.2$ = density of air (kg/m^3) at 20^oC

c = 340 – speed of sound (m/s)

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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 π 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.

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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.

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Section-2   NOISE LEVELS IN BUILDINGS

Background sound in a space has three basic sources: 1) building services, primarily the HVAC systems; 2) intrusive noise from outside the space; and 3) the activities within the space.

The acceptability of background sound in a space depends on two factors.

1) This first is the loudness of the sound compared with that of activities in the space. If it is clearly discernible above normal activity, then it can make speech communication or concentration difficult.

2) The second factor is the quality of the background sound. A background that is perceived as rumbly, roaring, hissy, or tonal is likely to result in complaints of discomfort due to annoyance and stress. The background sound spectrum in this situation is said to be unbalanced. Intrusive noise can be particularly annoying, even if it is not very loud, when it has some message associated with it, or if it is distinctive. Speech heard through a wall, a dripping faucet, buzzing from lights or the whine from a pump are examples of such noise.

It is very important to keep background noise down to a level that ensures a congenial living or working environment for building occupants. The degree of occupant satisfaction is determined by many factors. For example, large conference rooms, auditoriums, and recording studios can tolerate only a low level of background sound. On the other hand, higher levels of background sound are acceptable and even desirable in certain situations, such as open-plan offices where a certain amount of speech and activity masking is essential. Therefore, the system sound control requirements vary depending on the use of the space.

The ideal background sound should be:

✓ Balanced distribution of sound energy over a broad frequency range with no dominant band of noise

✓ Smooth, with no audible tones such as whine, whistle, hum, or rumble

✓ Steady, with few fluctuations in level such as throbbing or pulsing

At present no acceptable process easily characterizes the effects of audible tones and level fluctuations. The preferred sound rating methods generally include the A-weighted sound pressure level dBA and noise criteria NC curves (or European predominant Noise rating NR curves), and the more recent room criteria RC and balanced noise criteria NCB.

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A-weighted dB (A), (L_A)

The sound level in an occupied space can be measured directly with a sound level meter which measures the sound pressure level at the microphone location. The db (A), is a weighted method compensating for the human sense of sound pressure at different frequencies. The main advantages with the A -weighting are:

1) it is adapted to the response of the human ear to sound

2) it is possible to measure easily with low cost instruments

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 tell the whole story and the data is meaningless to anyone trying to solve the problem. If you want to know what's going on, you have to take sound level as a function of frequency or octave band, and that takes a spectrum analyzer. Then you can do diagnostics.

Where a more detailed procedure is required, the noise criteria (NC) rating procedure, requiring measurements in octave bands, is normally used.

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What is Noise Criterion – NC values?

These are single figure expressions of specific pressure levels at the frequencies making up the octave band. NC level is a standard that describes the relative loudness of a space, examining a range of frequencies (rather than simply recording the decibel level).

NC-levels refer to the constant, continuous background noise perceived inside the space as opposed to any intermittent noise from activities occurring within the space. NC should be considered for any project where excessive noise would be irritating to the users, especially where speech intelligibility is important. There are a few spaces where speech intelligibility is absolutely crucial, includes recording studios, lecture halls, performance halls, courtrooms, libraries and educational facilities. For other areas, such as machine shops or kitchens, it is not essential to maintain a particularly low NC level.

Noise Criterion (NC) is the most common method used in <u>North America</u> to specify maximum sound pressure levels in rooms. NC is particularly used in Heating, Ventilating and Air Conditioning (HVAC) work, with an NC-35 being the most common requirement. In Europe NR level is used.

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NC Level – Methodology

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
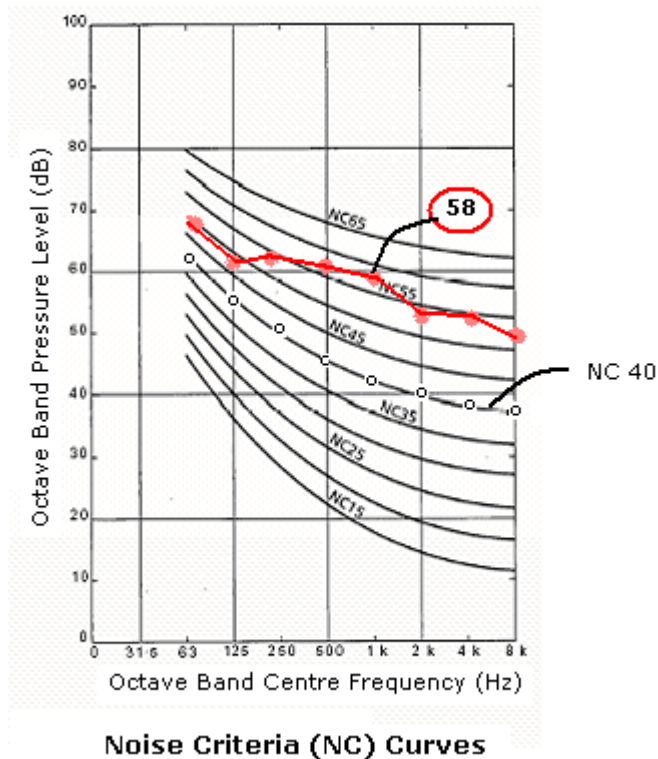
For general office application, the recommended NC values is 35-45, therefore if say NC 40 is chosen, then the amount of insulation at each frequency can be calculated.

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

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NC Level – Limitations

The NC method of rating a spectrum does not address two important questions:

- Is the spectrum balanced so as to avoid being annoying?
- Is the spectrum low enough in the speech bands for proper speech communication?

It must be noted that NC contours 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.

In addition, NC does not account for sound at very low frequencies. The NC curves are only defined to 63 Hz but HVAC systems often have noise in the 16 and 31.5 Hz bands. While NC is a great improvement over previous single number dBA ratings, it gives little indication of the "quality" of the sound. These drawbacks lead to the development of other rating methods such as the RC, NCB, and RC Mark II.

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Room Criteria (RC)

The room criterion (RC) is the most widely used method to evaluate existing mechanical systems and to establish design goals for new HVAC systems. The RC method measures background sound in a building over the frequency range of 16 Hz to 4000 Hz and requires two steps: determining the mid-frequency average level and determining the perceived balance between high and low frequency sound. The measurement values are taken in *an unoccupied room*.

The RC rating system employs two descriptors; a number descriptor which represents the speech interference level of the spectrum and a letter descriptor, which represents the subjective quality of the sound to a typical listener. Four "Quality" letter designations currently in use are:1) "R" - Rumble; 2) "H" - Hiss; 3) "V" -Vibration (acoustically induced) and 4) "N" - Neutral

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. The

The letter description is determined by analyzing the low and high frequency spectra compared to a line drawn with a -5 dB slope per band through the numerical RC point @ 1,000 Hz. This establishes the SIL (Speech Interference Level) line. Lines of -5 dB slope create the RC chart.

Essentially, the RC curves use a three-pronged method for rating a sound:

a) The arithmetic average of the levels in the 500-, 1,000- and 2,000-Hz bands is the RC rating.

b) 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.

c) 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.

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Balanced Noise Criteria (NCB)

Balanced Noise Criteria (NCB) is based on the ANSI threshold of audibility for pure tones and is defined as the range of audibility for continuous sound in a specified field from 16 Hz to 8000 Hz. This system had three purposes: a) to extend the rating system down to 16 Hz; b) to update the NC curves to more modern equal loudness contours, and c) to get away from the tangential method of the NC curves. These criteria are all specified in ANSI Standard S12.2-1995 (R1999), Criteria for Evaluating Room Noise. The sound level measurements for NCB should be taken in *an occupied room*.

The RC and NCB ratings include procedures for checking different factors such as the rumble compliance (excessive noise at frequencies below 500 Hz) and the hiss compliance (excessive noise at frequencies above 1000 Hz). With NCB curves, recommended levels of background noise for any given space are based on the level of speech interference that can be tolerated for designated activities within that area.

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What is Noise Rating- NR values?

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. This method is used in Europe for interior and exterior design criteria levels.

The Noise Rating (NR) – curves were developed by the International Organization for Standardization (ISO) to determine the acceptable indoor environment for hearing preservation, speech communication and annoyance. The measurement is similar to NC curves where noise rating graphs for different sound pressure levels are plotted at acceptable sound pressure levels at different frequencies. They have a greater decibel range than NC curves.

Choosing the appropriate noise criteria is important when specifying what level of noise is acceptable. Most organizations use a particular index based upon practical experience.

A comparison of different indices and recommended maximum levels of Noise Criterion limits for different types of rooms use is listed below:

Type of Room - Occupancy		Noise Criterion - NC	Noise Rating - NR	db(A)*
Very quiet	Concert and opera halls, recording studios, theaters, etc.	10 - 20	20	25 - 30
	Private bedrooms, live theaters, television and radio studios, conference and lecture rooms, cathedrals and large churches, libraries, etc.	20 - 25	25	25 - 30
	Private living rooms, board rooms, conference and lecture rooms, hotel bedrooms	30 - 40	30	30 - 35
Quiet	Public rooms in hotels, small offices classrooms, courtrooms	30 - 40	35	40 - 45
Moderate noisy	Drawing offices, toilets, bathrooms, reception areas, lobbies, corridors, department stores, etc.	35 - 45	40	45 - 55
Noisy	Kitchens in hospitals and hotels, laundry rooms, computer rooms, canteens, supermarkets, office landscape, etc.	40 - 50	45	45 - 55

Organizations that Set the Guidelines

There are three primary organizations that recommend guidelines for background sound levels for interior spaces: the American National Standards Institute (ANSI), ASHRAE and the International Building Code (IBC 2000).

The American National Standards Institute (ANSI)

ANSI is a private, nonprofit organization that administers and coordinates the voluntary standardization and conformity assessment system in the United States. ANSI has established a two part standard specification and design guideline for maximum sound levels in classrooms referred by (K-12), ANSI S12.60-2002, Acoustical Performance Criteria, Design Requirements, and Guidelines for Schools. This standard can help eliminate acoustical problems in the design stage and provides a resource for architects, school superintendents, audiologists, and even parents by identifying the minimum requirements for an effective learning environment. ANSI background noise limits for classrooms are as follows:

1) For learning spaces with volumes less than 20,000 ft^3, one-hour average noise levels should not exceed 35 dBA.

2) For learning spaces with volumes more than 20,000 ft^3, one-hour average noise levels should not exceed 40 dBA.

The maximum reverberation time* is as follows:

1) For learning spaces with volumes less than 10,000 ft^3, the reverberation time should not exceed 0.6 seconds.

2) For learning spaces with volumes more than 10,000 ft3 but less than 20,000 ft^3, the reverberation time should not exceed 0.7 seconds.

These are the primary factors of the ANSI standard S12.60-2002. By meeting these guidelines a good classroom acoustical environment can be provided and the learning process improved.

(Note * - The reverberation time is the amount of time required for sound to decay 60 decibels from its initial level, this is discussed further in next section).

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ASHRAE

ASHRAE has been recommending design noise level criteria for more than 40 years in the areas of heating, ventilation, air-conditioning, refrigeration and related areas. The RC rating system has become the preferred rating system by the ASHRAE, and is based on the level of the mechanical system background noise in octave band frequencies. The RC method measures background sound in a building over the frequency range of 16 Hz to 4000 Hz. This rating system requires two steps: determining the mid-frequency average level and determining the perceived balance between high and low frequency sound. The room criterion (RC) is mostly used for acoustical design of HVAC systems.

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International Building Codes (IBC)

IBC is being adopted across the U.S. Under the leadership of the Federal Emergency Management Agency, the major code organizations have banded together to form the International Code Council (ICC) to generate the IBC, which is being adopted across the country. Building codes generally specify noise levels inside residences in terms of dB (A) of NC. Typical code values are 40 to 45 dB (A) and NC-35.

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European Standards

The European Commission has published since 1986 a directive (86/188/EEC, Official Newspaper of the EC L137/28/24.5.86) for the protection of workers from the dangers resulting from the exposure to noise in their working environment. The most important International and European Standards include the **EN 12354**

(Building Acoustics - Estimation of acoustic performance of buildings from the performance of elements), **EN ISO 717** (Acoustics - Rating of sound insulation in buildings and of building elements - Part 1: Airborne sound insulation), **ISO 1996** (Acoustics - Description and measurement of environmental noise), **EN ISO 11654** (Acoustics - Sound absorbers for use in buildings - Rating of sound absorption), **ISO 140** and **EN 20140** (Acoustics -- Measurement of sound insulation in buildings and of building elements), **ISO 3744** and **EN 23744** (Acoustics - Determinations of sound power levels of noise sources, - Engineering methods for free field conditions over a reflecting plane), **ISO 11201** (Acoustics -- Noise emitted by machinery and equipment). The characteristics of building materials and the minimum requirements for buildings and indoor spaces depending on their function, construction examples and calculation methods, guidelines for planning and execution, proposals for increased sound insulation, recommendations for sound insulation in personal living and working areas, are specified by the German Standard **DIN 4109** and British Standard Code of Practice **BS8233:1987 '**Sound insulation and noise reduction for buildings'.

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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 "sound transmission loss". For machines to be used in buildings the important characteristic is the sound power.

The section below provides information on the common noise descriptors.

Reverberation

The sound reverberation time 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.

A space with a long reverberation time is referred to as a "live" environment. When sound dies out quickly within a space it is referred to as being an acoustically "dead" environment.

- In a more reflective room, sound continues to reflect or reverberate and it will take longer for the sound to die away. The effect of this condition can be described as 'live' space with long reverberation time.

- In a very absorbent room, the sound will die away quickly and the room will be described as acoustically 'dead'.

An optimum reverberation time depends highly on the use of the space. For example, speech is best understood within a "dead" environment. Music can be enhanced within a "live" environment as the notes blend together. Different styles of music will also require different reverberation times.

	Reverberation Time			
	0.8 - 1.3	1.4 - 2.0	2.1 - 3.0	Optimum**
Speech	Good	Fair - Poor	Unacceptable*	0.8 - 1.1
Contemporary music	Fair - Good	Fair	Poor	1.2 - 1.4
Choral music	Poor - Fair	Fair - Good	Good - Fair	1.8 - 2.0+

* With an adequately designed and installed sound system, speech intelligibility concerns can be mitigated.
** The optimum reverberation time can be somewhat subjective and can shift based on numerous variables.

It is difficult to choose an optimum reverberation time in a multi-function space, as different uses require different reverberation times. A reverberation time that is optimum for a music program could be disastrous to the intelligibility of the spoken word. Conversely, a reverberation time that is excellent for speech can cause music to sound dry and flat.

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 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.
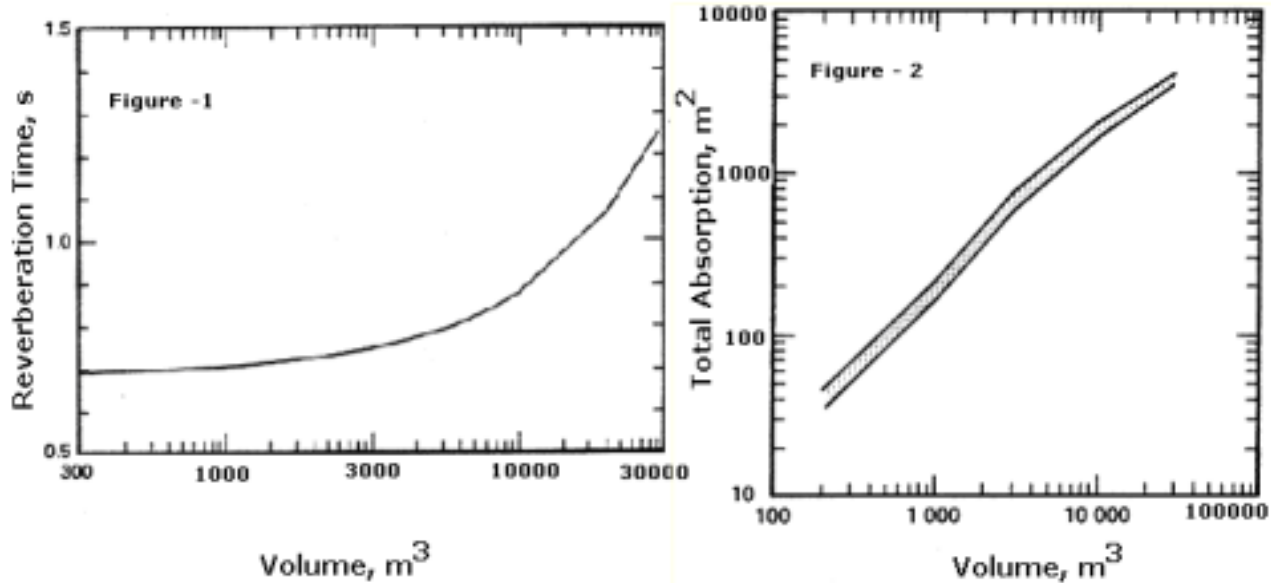

Figure -2 above shows the resulting range of acceptable total room absorption (in metric sabins) versus room volume (in cubic metros) that would lead to the desired optimum reverberation time within +0.1 second. *Designing for a total room absorption as close as possible to the middle of this range should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

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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 <u>sabins</u>.

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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.

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Sound Absorbing Materials

Sound absorbing materials control sound within spaces and function by allowing sound to pass through them relatively easily. They are generally porous and absorb sound as a result of many interactions. Porous sound-absorbing materials can be made of glass fiber, rock wool, open cell urethane foam, cloth or other materials that are porous to air flow. *These are characterized by high absorption coefficients at high frequencies, decreasing at lower frequencies depending on the type and thickness of the material.* Conversely, a material or system that provides a good sound transmission loss is usually non-porous and a good reflector of sound, although double-layer partitions often contain sound absorbing materials to reduce the reflection of sound inside the partition.

Sound Absorption Coefficients for some common Materials

Material	Sound Absorption Coefficient - α
Plaster walls	0.01 - 0.03
Unpainted brickwork	0.02 - 0.05
Painted brickwork	0.01 - 0.02
3 mm plywood panel	0.01 - 0.02
6 mm cork sheet	0.1 - 0.2
6 mm porous rubber sheet	0.1 - 0.2
12 mm fiberboard on battens	0.3 - 0.4
25 mm wood wool cement on battens	0.6 - 0.07
50 mm slag wool or glass silk	0.8 - 0.9
12 mm acoustic belt	0.5 - 0.5
Hardwood	0.3
25 mm sprayed asbestos	0.6 - 0.7
Persons, each	2.0 - 5.0
Acoustic tiles	0.4 - 0.8

For architectural purposes, sound absorbing materials and constructions can be divided into four types of materials depending on the way the absorption is mainly performed:

1) Turning the sound energy into heat such as fiberglass and carpet.

2) Vibrating with a specific frequency when the sound hits the surface such as lightweight panels and 5/8" gypsum board. (These materials absorb the sound effectively on a narrow band of frequencies)

3) Turning the sound energy into heat in the neck of the cavities (Helmholtz resonator) such as sound blocks. (This construction has a good absorption on low frequencies)

4) Allowing the sound to go through such as some types of grid systems and lay-in ceiling with sound leakage above it.

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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.
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.
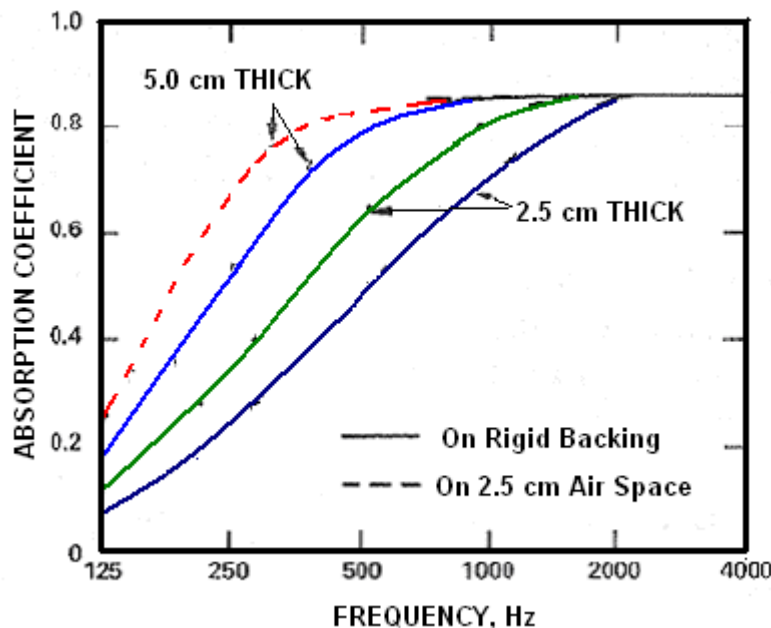
There are three main standard methods used to test materials for absorption. Two of them are reverberation chamber methods – ASTM C423 in the U.S.A. and ISO 354 in Europe. These two methods are quite similar, but the ISO method – in general – will produce slightly lower overall numbers than the ASTM 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.

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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 expresses 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), although 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.

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Mean Absorption Coefficient

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

$a_m = A / S$

Where

1) a_m = mean absorption coefficient

2) A = the absorption of the room (m^2 sabin)

3) 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.

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Noise Reduction Coefficient (NRC)

The NRC is a single number index of rating how absorptive a material is.

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 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.*

In general, the higher the number, the better shall be the absorption. NRC is useful for a general comparison of materials. However, for materials with very similar NRCs, it is more important to compare absorption coefficients.

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NRC Limitations – What you should know

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 gives no information as to how absorptive material is on low and high frequencies.

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.

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Sound Transmission Loss (STL)

When sound waves strike a partition, the pressure variations cause the partition to vibrate. A portion of the vibration energy on the sound source side will be transferred through the partition where it is re-radiated as airborne sound on the other side. *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 shall be the amount of sound energy passing through the partition.

The transmission loss (TL) usually increases with the frequency of the incident sound and also varies with the direction of the sound waves. Laboratory measurements of the TL of an assembly are made in one-third

octave bands from 125 to 4000 Hz. It is convenient, however, to have a single number to compare building assemblies. The designation most commonly used in North America is the sound transmission class (STC).

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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.

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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).

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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.

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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 <u>whereas</u> 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 <u>whereas</u> 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.

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Section - 4   **HVAC 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.

With building structures getting lighter, duct air velocities increasing, mechanical equipment rooms (MER's) getting smaller, and equipment running at higher speeds, mechanical and HVAC noise problems are on the rise. Attempts to reduce energy use often aggravate the acoustical problems. While these may be very common citations, there is usually no excuse for it because the HVAC noise problems appear to arise more from ignoring the proper design process than from inadequate technology. The best way to avert noise problems is to review the mechanical system plans and layout with a qualified acoustical consultant during the early design phases. This would help spot trouble and may alleviate potential problems much more cost effectively than retrofitting systems after construction.

The ASHRAE Handbook, Chapter 43: "Sound and Vibration Control provides some guidelines for noise control in HVAC systems.

Noise from Ventilation and AC Systems

The HVAC noise that ultimately reach living/ working quarters are made up of

a. Low-frequency fan noise - fans generally produce sound in 16-Hz through 250-Hz octave bands; Variable-air-volume (VAV) boxes noise is usually in the 125-Hz through 500-Hz octave frequency bands.

b. Mid-frequency airflow or turbulence-generated noise - Velocity noise from airflow and turbulence in a duct ranges from 31.5 Hz through 1,000 Hz.

c. High-frequency damper and diffuser noises - Diffuser noise usually contributes to the overall noise in the 1,000-Hz through 4,000-Hz octave bands.

Typical mechanical system noise is made up of a variety of noise components. Therefore the HVAC designer has to deal with noise control at various frequencies. Noise in any HVAC system can be measured as a function of frequency in decibels, typically divided into eight octave bands. That result is then compared against a criteria curve, such as the noise criterion (NC) family of curves. "Private offices are typically designed for NC 30 to 35, while open offices are designed for NC 40 to 45. Experience has shown that sound levels in offices above NC 45 can start to create complaints.

Noise Points in HVAC Systems

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

1. Ductborne noise

2. Radiated equipment noise

3. Break-in noise

4. Break-out noise

5. Terminal end noise

Each of these issues must be addressed, or else the design will fail.

Figure below shows an illustration of how each of these noise sources could be generated.

TYPICAL MECHANICAL SYSTEM NOISE COMPONENTS
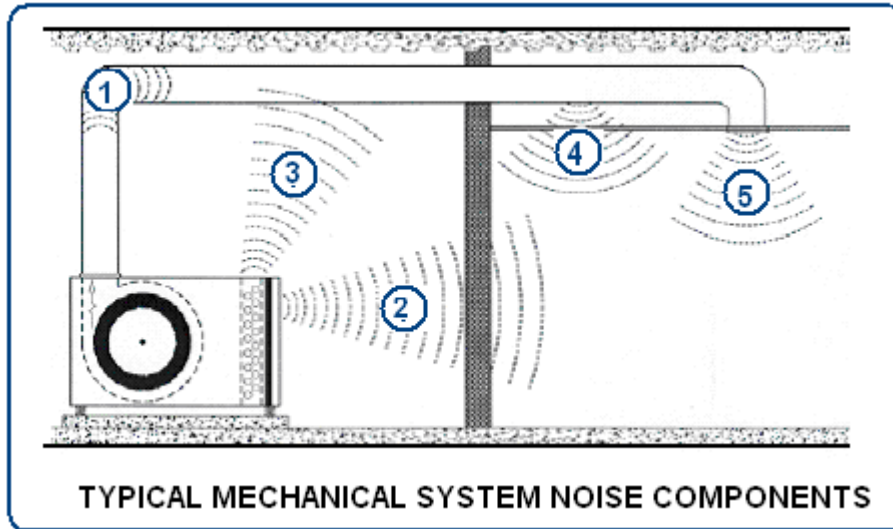
1) **Ductborne Noise**

Ductborne noise propagates along the ductwork, follows all transitions and takeoffs, and ultimately exits at the diffuser or grille, thus, impacting the space being served. The sound generated by the fan will travel along the ductwork both upstream and downstream of the fan easily because the velocity of sound is much greater than the velocity of air in ducts. The noise is generated aerodynamically by the flow of air through the duct system. This component of the noise (sometimes called regenerated noise) is very much dependent on the velocity of the air and the smoothness of the flow. It increases very rapidly as the flow velocity is increased and/or by changes in cross-section in the duct. Turbulent noise in ducts is generated from the following:

1) Objects such as dampers, grilles, rods, etc

2) Constrictions in duct cross sectional area, silencer splitters etc.

3) Jet noise, inlet or discharge noise flowing through orifices

4) Boundary layer turbulence, air passing over the inner surface of the duct

5) Flow around bends and duct take offs (branches)

The best way to control ductborne fan noise is to not create it. To avoid ductborne noise, note the following:

1) Select quiet fans based on sound power data. Do not buy noisy fans and try to "fix" them. Provide good fan outlet conditions in accordance with ASHRAE guidelines.

2) Do not turn the air in "the wrong direction," or the ducts will rumble.

3) In VAV systems, do not use inlet vanes to modulate air flow, particularly if the fan motors are 10 hp or greater. Inlet vanes cause rumble.

4) Use variable frequency drives on air handling unit's fans and pumps.

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2) **Radiated Equipment Noise**

Radiated equipment noise transmits through the wall or floor into the adjacent space or, in the case of roof-top equipment, to the environment. Radiated equipment noise is generated by vibration of the fan casing and motor. All structures have multiple frequencies of vibration resonance that result from factors related to their construction. Such factors include design, type of material(s), building mass, and comprise the vibration characteristics of a specific structure. 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 from

1) Damaged or unbalanced fan wheel

2) Belts too loose; worn or oily belts

3) Speed too high

4) Incorrect direction of rotation

5) Bearings need lubrication or replacement

6) Fan surge

To reduce vibration noise paths, note the following:

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) Use of resilient suspended plaster ceiling below the floor level

6) Sound absorption treatment of the mechanical room walls and roof

7) Routine maintenance of fan drive and motor

8) Make sure the fan rotates in same direction as the arrows on the motor or belt drive assembly

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3) **Duct Break-In Noise**

Break-in noise is radiated equipment noise that enters the ductwork and propagates down the duct system.

Noise inside ceiling plenums or from air conditioning equipment, plant rooms etc, can break into ducts, particularly flexible ducts and then be carried into rooms or spaces downstream. Flexible ducts, due to their light weight, flexibility, speed and ease of installation, are commonly used in air conditioning systems. Noise can more easily penetrate flexible ducts because of their lightweight nature.

To avoid break-in noise, note the following:

1) Where possible, avoid ducts passing through noisy areas as this can significantly increase noise through the air conditioning system.

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

3) The flexible ducts can be enclosed in a solid enclosure constructed from timber, plasterboard or sheet steel, etc.

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.

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4) **Duct Break-Out Noise**

Duct Break-out noise also propagates along the ductwork, however, 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 know 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

To reduce noise breakout from ducts, note the following:

1)   Like fan noise, the best way to control air flow noise is to not create it. It's best to design low-pressure, low-velocity systems. From a noise control standpoint, 1000 fpm is low velocity, 2000 fpm is medium velocity, and 3000 fpm is high velocity. Design proper duct fittings for smooth flow and gradual velocity changes.

2)   Make ducts stiffer. External bracing of ducts can also increase stiffness.

3)   Use heavier material for duct walls and increasing damping (i.e. thicker steel sheeting)

4)   Adding damping (spray on or self adhesive compounds)

5)   Acoustic lagging, preferably with a heavy limp impervious layer isolated or decoupled from the duct with either glasswool or rockwool. Use 1 in. thick duct liner instead of ½ in, which is too thin to be useful

6)   Provide acoustic silencers or long runs of internally lined rectangular duct to reduce fan noise prior to allowing a duct to run above the acoustical tile ceiling of an occupied space.

The solutions to reduce noise breaking out from ducts can be expensive. Therefore it is more cost effective to avoid noise break out problems than to try to correct them later.

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5)   **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. Air noise from the diffusers and from transitions can cause additional noise in the receiving room.

When considering devices like these – for supply or return air – you should try to find out the "NC" ("Noise Criteria") rating for them from their respective manufacturers. These are noise ratings that the device manufacturer must provide for all possible airflow rates. For a critical space, you should choose a device that has NC-30 or lower for the designed airflow rate. Actually, NC-30 is the highest you should consider. It should be quite easy to find a device that is "off the charts" i.e., it doesn't have a rating because it did not produce any noise when tested at your airflow rate.

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Noise Reduction Strategies in HVAC Systems

Unwanted sound – noise that is – can generally be controlled at one of three points: at its source, at the listener, or in the path between the source and the listener.

Noise control at the source can be achieved by maintaining equipment, relocating noise sources, removing unnecessary noise sources, using quiet models, or redesigning noisy equipment. If a quiet, well-balanced machine is specified and installed, there will be less need for the addition of silencers or acoustic treatment. A variety of laboratory and field test procedures can be used to measure the sound power of machines.

When noise reduction cannot be achieved at the source, it may be useful to modify the noise propagation path. Measures include full or partial enclosures; sound barriers; room absorption; isolation; sound cancellation; masking; changes in duct geometry, fan type and flow characteristics etc.

HVAC systems noise control requires attention to many, many details. This section attempts to summarize here the many concepts covered by various handbooks.

1)   **Location**

Do **not** locate any of your heating/cooling equipment anywhere near the acoustically sensitive rooms. If an air handler is too close to the noise sensitive area, risk of excessive vibration and direct air flow noise is greatly magnified.

Always put the air handler in a room that is physically as far away as possible from anywhere you will be doing any critical listening. The more distance (and walls and ceilings and floors) you can put between you and any of the equipment, the better off you'll be.

Each space is different. Take an example of a recording studio where noise is very critical has to be dealt differently with respect to other structures. Perhaps due to their superficial resemblance to warehouse space, studios may appear to inexperienced mechanical engineers to be perfect candidates for rooftop units located directly overhead. This is almost always a big mistake, because exposing a studio to noise and vibration by poking a hole in its roof and placing rotating machinery there makes it virtually impossible to achieve industry-standard background noise levels.

The following may be noted:

- Locate all fans and noisy equipment as far away from critical spaces as practical, and never on a roof directly over a critical space.

- Make sure the roof under rooftop units is as rigid as possible. Consider flexibility of roof or upper floor structures in selecting vibration isolation.

- Locate the mechanical equipment room on grade, in a basement, or in a sub-basement.

- Use non-critical spaces to buffer critical spaces from equipment room noise. These buffer spaces include corridors, elevator cores, toilet rooms, storage rooms, telephone equipment rooms, etc.

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2) **Vibration Isolation**

Rotating or motor-driven machinery generates vibration energy 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 prudent to isolate any vibrating equipment from the surrounding structure. This can be accomplished by mounting the equipment on springs, pads, or inertia blocks. Floating floors may also be considered particularly if the mechanical room is located on different floor.

How much isolation?

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. The following key points should be noted when applying isolation:

- The principal strategy for controlling equipment vibration is location. Place the rotating equipment on grade, in the basement, or in the sub-basement. Most pumps need isolators even in these desirable locations.

- Do not place equipment on limber long-span floors or roofs. The structure must be at least 10 times as stiff as the equipment vibration isolators.

- Speaking of vibration isolators, specify isolators for equal static deflection. If the weight at each of the four corners of a machine base is significantly different, then four different isolators with four different stiff nesses are required.

- Always use total weight of equipment when selecting isolation. Always consider weight distribution of equipment in selection.

- Specify isolators with adequate static deflection and flex connections of adequate length. Use thrust restraint isolators when necessary.

- A quiet motor bearing system, a well balanced motor and fan impeller as well as providing the required air movement at as low a RPM as possible shall result in lower vibrations.

- Selecting specific isolating equipment should be left to a specialist trained in vibration analysis, who tunes each vibration isolation device to a specific frequency range. If not matched properly with the

treated equipment, the devices can amplify the vibrations and cause more of a problem than would have occurred without any treatment.

A common rule of thumb for selection of vibration isolation is as follows:

Equipment RPM	Static Deflection of Isolation	
	Critical Installation	Non-critical Installation
1200+	1.0 in	0.5 in
600+	1.0 in	1.0 in
400+	2.0 in	1.0 in
300+	3.0 in	2.0 in

Note in above table

- Critical installations are upper floor or roof mounted equipment

- Non-critical installations are grade level or basement floor.

At the very least, be sure the mechanical equipment rests on some sort of vibration damping pad. You can also "float" the unit on some rubber or mechanical springs. Just be aware that the proper type of springs must be used. The wrong type of springs can do nothing or even amplify vibrations from equipment. Get an HVAC or noise and vibration control expert in the loop.

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3) **Airflow noise**

Air can only be heard when it is moving. Still air is quiet. The faster the air moves the louder the noise. Often, the easiest way to minimize airflow noise ("whoosh" or "hiss") is to decrease the air velocity *throughout the duct systems*.

Increasing duct or opening size to lower air velocities effectively reduces noise in the right applications.

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. It should not be more than about 25% higher in the next 10 feet. A common problem is a quiet diffuser directly on the side of an exposed duct with high velocity flow in the duct. The duct velocity should not exceed the diffuser neck velocity.

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4) **Noise due to Ductwork**

The supply duct is generally the limiting element in the acoustical design of an equipment room. With a given duct layout, the supply duct construction, aspect ratio of the duct, internal lining, and duct stiffness can all affect the end results. Bare sheet metal air ducts provide a limited noise barrier and have no sound absorptive properties. On the contrary, these act as conduits for equipment noise. Sheet metal also clicks and pops due to expansion and contraction from air temperature changes, and may even vibrate from airflow or fan action. The attenuation will depend on the mass per unit area of the duct walls, upon the frequency and the duct size. At low frequencies the stiffness of the duct walls will be more important than the mass and as result low frequency attenuation will be different for round and circular cross-section ducts.

For any ducted system, some key things to keep in mind:

- With centrifugal-type fans, the low frequency content of the fan noise is the most prevalent and difficult to attenuate. Mass best attenuates low frequency noise. Therefore, by increasing the duct gauge from SMACNA standards, one can increase the mass and enhance the sound barrier.

- 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.

- Increasing both the angle bracing and duct gauge has been successful in reducing the fan noise breakout in occupied spaces.

- Internal lining provides attenuation for downstream/upstream noise issues as well as damping element to the duct construction. The selection of duct lining should be done understanding the aspect ratio, overall fan sound, and attenuation requirements.

- The attenuation of unlined sheet metal ducts is very small, ranging from 0·6 dB/meter for small rectangular ductwork at low frequencies to less than 0·1 dB/meter at high frequencies.

- With all ducts, care must be taken that the ducts are large enough to deliver the needed volume of air. Smaller ducts tend to be noisier and leakier than larger ducts due to higher air speeds and pressures.

- 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.

- 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 (ft²). Example: 500 CFM requiring 12" round duct yields 0.785 ft² 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.

- When the airflow generated noise is unacceptably high, engineers should always first attempt to reduce the air velocities (airflow generated noise is proportional to the sixth power of the velocity). 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.

- Use round duct near fans when the duct must pass over critical areas. Use rectangular duct near fans over non-critical areas.

- If the ducts are sheet metal, you may need to isolate them from the building using spring isolating hangers.

- Use round, insulated, flexible ducts when joining to the terminal registers. Do keep in mind that since there is no sheet metal, sounds could "break-in" to the ducts from spaces they are passing over or through. Size flex to match diffuser neck and locate balancing damper at flex inlet.

- 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.

- Any new ductwork should be sized according to recognized industry standards such as Manual-D, published by the Air Conditioning Contractors of America (ACCA).

- Use sheet metal ducts with 2" internal lining for first 10 feet (1000 ft/min).

- Locate the dampers well upstream of the outlet within the lined ductwork.

- Use minimum 6 feet of acoustical flex between low pressure duct and diffuser.

- The more (preferably lined) duct length you run between the equipment and your room, the less noise you'll have to worry about.

- Avoid putting ducts in walls shared with other noise sensitive or noise producing spaces. This will create a weak link in your wall construction.

- In order to deliver the same volume of air, flex duct and duct board systems must usually be sized larger than metal ducts as their interior surface is much rougher, leading to more restrictive air flow

- Connect the air handler to the duct work using flexible connectors.

- 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. It should not be more than about 25% higher in the next 10 feet. A common problem is a quiet diffuser directly on the side of an exposed duct with high velocity flow in the duct.

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5) Acoustic Duct Lining

Fan noise must be silenced before entering the ducts. *The addition of acoustic lining results in significant attenuation of high frequency sound*. The ARI Standard (ARI 885), as well as the ASHRAE Handbook, provides guidance on insertion loss per foot of duct based on inside duct dimensions. Two reference tables are provided here for rectangular and round lined duct from the ARI Standard.

Round 1" Spiral Duct (Lined) (dB / ft.)

Duct Diameter	Octave Band					
	2	3	4	5	6	7
6"	0.59	0.93	1.53	2.17	2.31	2.04
12"	0.46	0.81	1.45	2.18	1.91	1.48
24"	0.25	0.57	1.28	1.71	1.24	0.85
48"	0	0.18	0.63	0.26	0.34	0.45

(ARI 885-98)

Rectangular 1" Lined Duct (dB / ft.)

Duct Dimension	Octave Band					
	2	3	4	5	6	7
6" x 6"	0.6	1.5	2.7	5.8	7.4	4.3
12" x 12"	0.4	0.8	1.9	4.0	4.1	2.8
24" x 24"	0.2	0.5	1.4	2.8	2.2	1.8
48" x 48"	0.1	0.3	1.0	2.0	1.2	1.2

Lining sheet metal with fiber glass duct liner insulation can deliver significant HVAC noise control for the dollar. The following facts may be noted:

- The amount of attenuation varies with duct size and lining thickness. The thicker *the internal duct liner, the better the sound absorption.* A common reference for measuring sound absorption of duct liner is the Noise Reduction Coefficient (NRC), an average of the sound absorbed at four different frequencies. For example, a one-inch-thick fiber glass duct insulation from 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 noise.

- 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.

- 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.

- If the bend is lined with acoustically absorbing material or fabricated out of fiberglass then the attenuation will be greatly increased. Adding insulation helps reduce high frequency sounds.
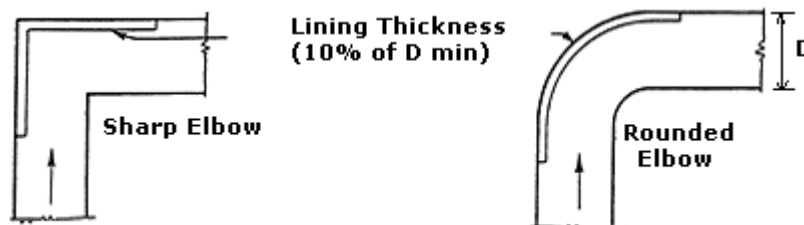
- Square elbows are preferred to radius bends (for acoustics though pressure drop shall be higher).The lining should have a thickness at least 10% of D, the clear width between the two linings and the length of lining should extend a distance not less than 2D before and after the bend.

- Attenuation of rectangular tees is determined by treating the tee as two elbows placed side by side.

- The benefits of lining air ducts go beyond acoustical improvements. The thermal insulating properties of fiber glass help reduce energy consumption and promote comfortable temperatures. R-value exceeds 4 per inch of most fiber glass duct liner products. Thermal and acoustical performance is documented using industry standard test methods specified under ASTM C 1071.

- The insulation properties of the lining material also prevent condensation from forming on the outside of metal ducts, eliminating one possible source of mold growth.

- It is less expensive than other noise control solutions and requires no additional space. It is effective in reducing the transmission of equipment noise and the noise of cross-talk between the rooms.

- 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.

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6) External Duct Lagging

External lining (lagging) of ducts with foil faced rockwool or glasswool reduces duct breakout noise by damping the duct. The sound attenuation achieved inside the duct is also enhanced by duct lagging particularly at low frequencies, up to about 500Hz.

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7) 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. 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.
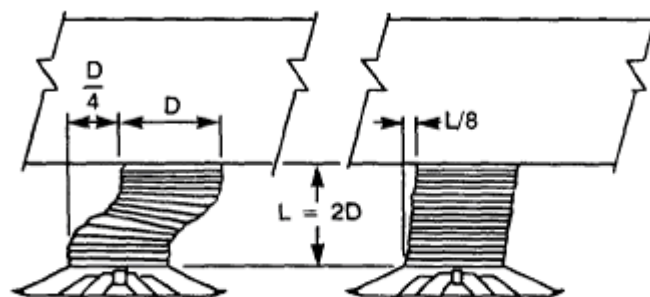
Example of Increased Pressure Drop and Noise Level for Flexible Duct Connection

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8) Sound Attenuation by 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.

- 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.

- 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.

- 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).

- 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.

- 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.

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8) Noise due to Duct Branches

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.

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9) Volume Dampers

An important feature in the proper design of a duct work system is the ability to control the amount of air that flows through each segment of the duct to ensure that the volume of air supplied to each space is tailored to its conditioning needs and that each supply diffuser in a given room is balanced with the others. To accomplish this, volume dampers are needed 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.

- For office spaces, ceiling supply diffusers routinely are installed with face dampers, which are volume control dampers located right at the inlet to the diffuser. The airflow noise created by the face dampers is essentially exposed directly into the room. In acoustically sensitive spaces, even if face dampers are left wide open they can generate audible noise. Do not use face dampers on the

diffusers to adjust the air volume. Adjust the volume upstream at the start of branch duct using opposed-blade-type dampers.

- Do not use opposed blade dampers in noise sensitive applications.

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10) **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 figure below) and Increase the amount of end reflection (higher number of smaller registers is preferable to fewer larger registers).

Ductwork Layout to Reduce Crosstalk
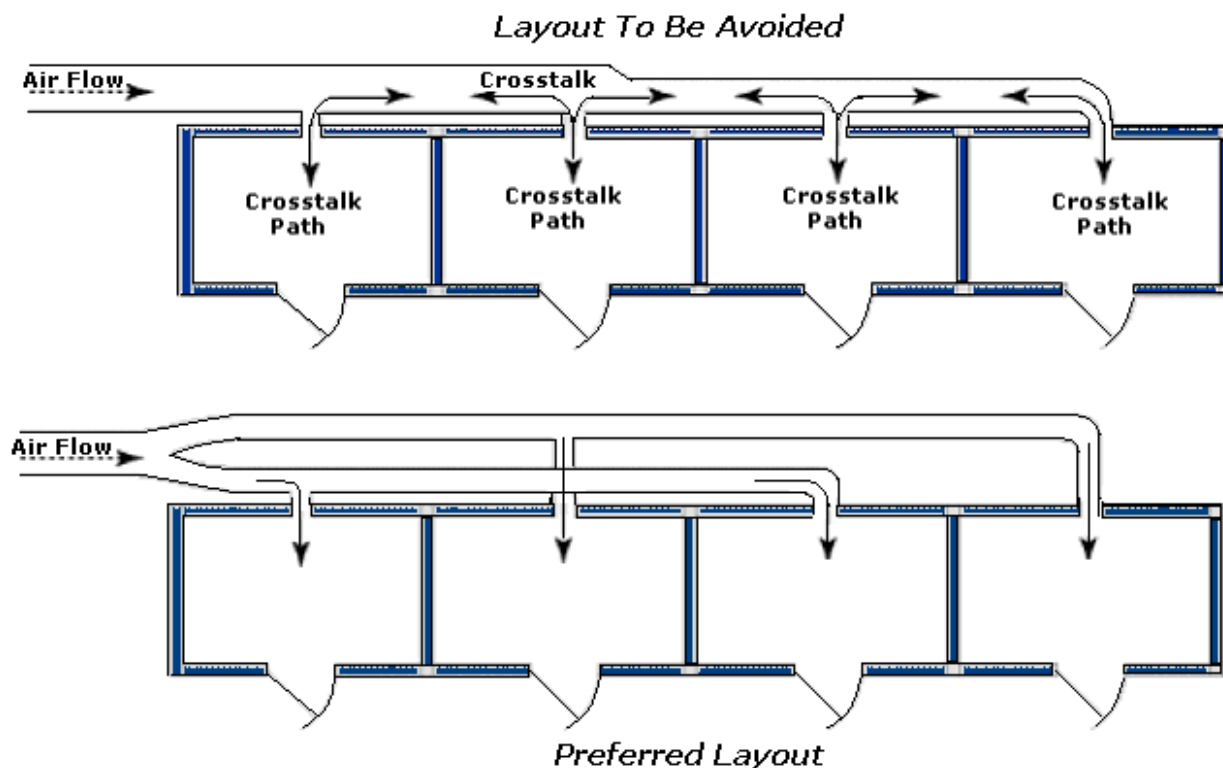
Although noise control issues through the supply air duct system are routinely considered in HVAC design, inexperienced mechanical engineers and contractors often forget that the *return*-air path is an equally important contributor to noise problems. In fact, because return-air systems sometimes employ 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.

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11) Mechanical Room Sizing

Make mechanical rooms large enough to allow some silencing before ducts exit.

- Defy alleged floor space constraints, and do not place air handling unit cabinets or built-up air handler enclosures or transformer cabinets close to 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 ft usually suffices.

- Provide a nominal 4 in. concrete housekeeping pad beneath equipment cabinets to minimize the effects of close coupling to the floor.

- Use round duct near fans when the duct must pass over critical areas.

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12) Mechanical Room Treatment

Equipment noise travels up, down, and sideways. This means that equipment room walls must be selected as a function of the equipment housed therein and the noise sensitivity of the adjacencies. The standard spec building steel stud and drywall partition constructions usually demonstrate very poor low frequency sound insulation performance. Some examples of concepts of building treatment include:

- 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.*

- The best way to stop sound transmission through a building structure is to isolate the sound source from the structure before the structure has a chance to vibrate.

- Walls need to be isolated from ceilings and floors, usually by means of dense, pliable rubber.

- The main ways to minimize sound transmission from one space to another are adding mass and decoupling. 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.

- Equipment room floors and ceilings must provide adequate sound insulation to protect adjacent spaces. Unfortunately, modern lightweight floor constructions typically consist of a fluted metal pan with a nominal 5 in. concrete topping, which is often not adequate.

- Limp mass is most often better than rigid mass (actually, a combination of the two is really is better).

- Every object, every construction material has a resonant frequency at which it is virtually an open window to sound - kind of like a tuning fork that "sings" at its particular resonant frequency.

- Trapped air (a.k.a., air spaces and air gaps) is a very good decoupler.

- Airtight construction is a key concept. Sound, like air and water, will get through any small gap. (Sound can leak through openings as small as 1/32" – in some cases even smaller.)

- Sound bounces back and forth between hard, parallel surfaces.

- The sound insulation of any single-leaf wall or floor built without gaps depends mainly on its 'mass'. According to the "mass law", there will be an increase in sound insulation of about 5dB if the mass/unit area is doubled. The insulation also increases by about 6dB for a doubling of frequency.

- Double leaf walls give good insulation if they are completely decoupled. For example, 2 sheets of plasterboard bonded together will give 30dB attenuation; this would increase to 50dB, if they were perfectly isolated. High levels of insulation can be achieved with care and expense - for example, separation of multiplex cinema auditoria of weighted standardized level difference of 65dB to over 70dB can be achieved by using 2 layers of 15mm plasterboard on separate studs, a large cavity with 100mm quilt inlay and careful head, base and edge detailing.

12 - B) Wall construction

The acoustical capabilities of the mechanical equipment should match the acoustical capabilities of the equipment room wall and door 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.

The same guidelines are true of emergency generators, boiler house and large chillers, which may cause noise complaints at properties hundred of feet away.

12-C) Duct penetrations

How the ducts penetrate the wall will affect the net performances of the MER wall. If a concrete masonry unit (CMU) wall is selected and the equipment room layout has a "gang" penetration of the wall, the ability of the contractor to seal off the areas between each duct will be difficult at best. Therefore, the resultant sound barrier of the installed wall with duct penetration will be significantly different from the laboratory rating of the CMU.
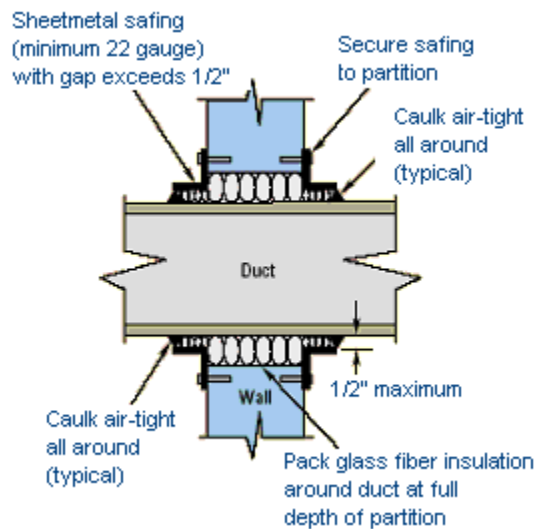
Duct penetration detail at wall

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.

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13) Acoustic Louvers & Turning Vanes

The intake and exhaust air louvers can cause property line noise complaints. Use of acoustic louvers at the air intake and outlets greatly reduces the air flow noise. The acoustic louvers could be fabricated from fiberglass. There are other fiberglass products that contribute to quiet, comfortable air-handling.

Turning vanes made of fiber glass instead of metal eliminate the noise of turbulence and vibration at duct angles.

Dampers, and other adjusting and balancing devices, should be relocated away from outlets, and secondary attenuation inserted between their new positions and the outlets.

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14) **Large Plenums provide sound attenuation**

When large values of sound attenuation are required, a sound absorption plenum will also be advantageous. 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. For sound frequencies sufficiently high, the wavelength of sound is less than any of the plenum dimensions (width, height, or length); the methods are accurate to a few decibels. At lower frequencies, when the wavelength of sound becomes greater than the plenum dimensions, the attenuation approximations are conservative and the actual sound attenuation exceeds the approximation.

- 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.

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15) **Duct Silencers**

Duct attenuators or silencers are used where high attenuation is required. Provide attenuator between fan and supply duct connection to ensure sound power is less than: 80 dB at 63 Hz, 73 dB at 125 Hz, and 70 dB at 250 Hz. The following facts on silencers are worth noting:

- Silencer is an effective means of reducing broadband noise as it travels down a duct system, and have the advantage of predictable performance. These silencers usually consist of sheet steel duct housing containing sound absorbent 'splitters' usually made of rockwool or glasswool.

- Duct silencers operate by restricting the airflow through a system of baffles exposing as much of the air stream as possible to sound-absorptive filler materials and/or resonant cavities

- The silencer's attenuation is normally quoted as an insertion loss in octave frequency bands. Silencers cause a pressure drop across them and also regenerated noise through the splitters, which increases with the air velocity through the ducts. Silencers perform well in reducing mid- and high-frequency noise.

- They are also much larger and heavier than the air ducts, often requiring six to 10 feet of unoccupied space. Silencers are available in numerous sizes and shapes to fit both in round or rectangular cross-sectional ducts.

- Silencers restrict airflow, and may require more powerful fans to maintain the desired air volume. If air flow is restricted to 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.

- Recognize that duct linings, plenums, or silencers will usually be needed to silence fans. Less-expensive, low-frequency tuned silencers are often the best silencer choice.

- Since attenuation values are generally higher in the first five octave bands in the reverse flow mode compared to the forward flow mode, more economical silencer selections can often be made on return-air systems. Forward flow occurs when air and sound-waves travel in the same direction, as in an HVAC supply system or a fan discharge. Reverse flow occurs when sound-waves and air travel in opposing directions, as in a typical return-air system.

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20 A) Type of Silencers

There are two different types: dissipative (or absorptive) silencers and reactive silencers. In absorptive silencers acoustic energy is converted to heat by the sound-absorbing processes which take place in the small interconnected air passages of fibrous or open-celled foam plastic materials. They are used to provide attenuation, of fan noise for example, over a broad band of frequencies. Because of the frequency characteristics of the absorbing materials they employ, this type of silencer is much more effective at medium and high than at low frequencies.

Reactive silencers reflect sound waves toward the source. Reactive silencers have tuned cavities or membranes that are designed to attenuate low frequency noise from machines where the absorptive type of silencer is ineffective. The attenuation produced depends on the dimensions of the pipes and chambers of the silencer. They are generally used to reduce noise from pulsating gas flows such as the air inlet and exhaust systems of internal combustion engines. The reactive silencer is non-fibrous, cleanable and has small or negligible pressure loss. The simplest kind of reactive muffler is the expansion chamber.

Many silencers use both reactive and absorptive mechanisms to achieve their effect.

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20 B) Performance factors of Silencers

The absorptive silencer is most commonly used in HVAC applications. The attenuation produced, in dB per meter run of duct, depends on four factors:

- Sound absorption coefficient of the duct lining material -The attenuation produced by absorptive silencer is at greatest at medium frequencies, dropping (at the high and low frequency ends of the spectrum). *The poor low frequency performance is simply because the sound absorption coefficient of the lining material is poor at low frequencies.* The poor high frequency performance occurs because much of the high frequency sound energy tends to be 'beamed' down the centre of the duct airways, and is unaffected by the sound absorbent duct linings. It is seen that reducing the airway width will improve the silencer performance considerably particularly at high frequencies.

- Thickness of the absorbing lining - an increase in thickness produce improved attenuation particularly at low frequencies.

- Width of airway- Reducing the airway width will increase this resistance to the flow of air and can be measured as a pressure drop across the silencer. Excessive restriction of air flow will also have an effect on another important silencer parameter, the noise generated by the flow of air through the silencer. Forcing the air through narrow airways will obviously cause an increase in flow velocity, and therefore in the amount of this self- generated noise. One way of obtaining the benefits of narrow airways without incurring the penalties of high flow resistance is to increase the number of airways, which increases the overall width of the silencer. Increasing the height has a similar effect. The alternative is to use a larger airway width, and increase the length of the silencer.

- 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. It follows that the optimum shape for the cross-section of a rectangular duct is a long thin one. In commercial splitter silencers a rectangular duct is split into several such sections.

Silencer design is a compromise between acoustic performance, silencer size (height, width and length), acceptable flow resistance and material costs. A common reference for evaluating the performance of silencer is by the term "Dynamic Insertion Loss" (DIL), which is the difference between two sound power levels or intensity before and after the silencer has been inserted. The other frequently used term the "Self Noise" - SN - is the sound power level in decibels generated by the silencer by a given air flow.

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20 C) 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.
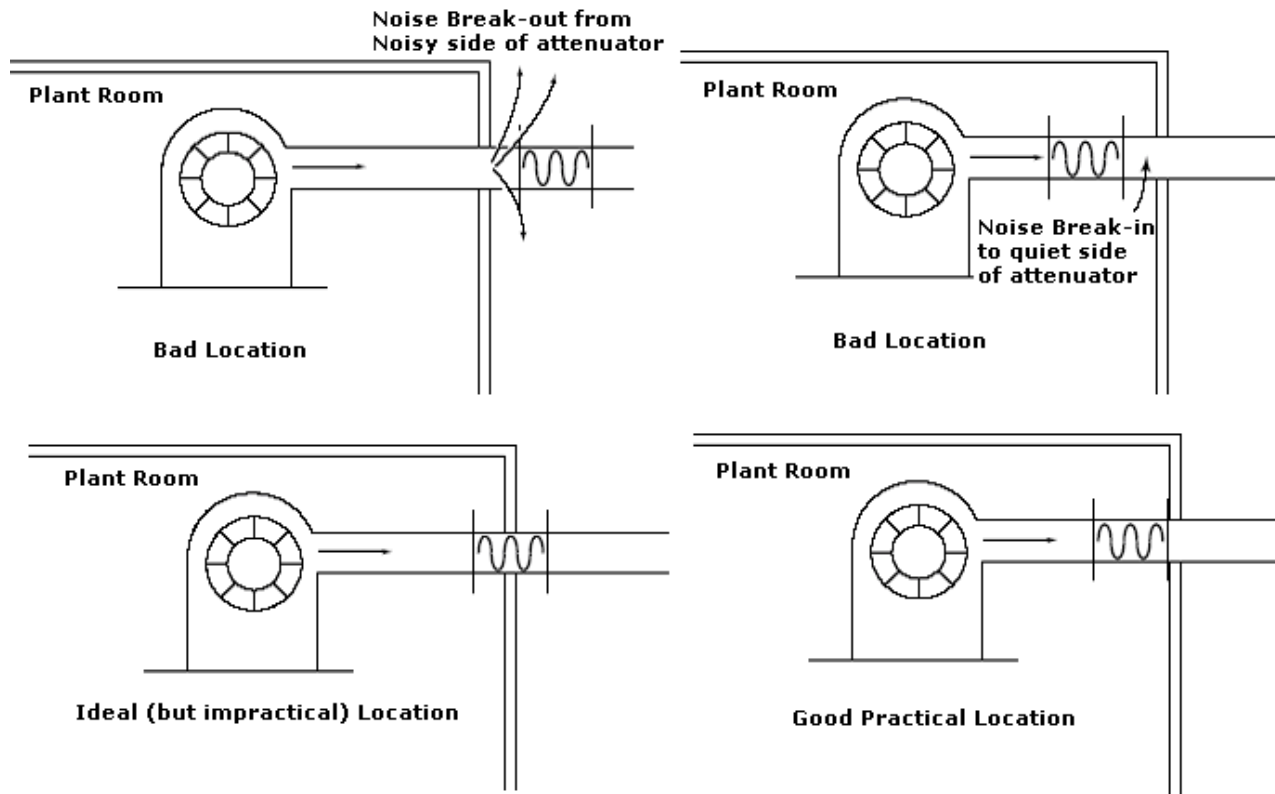
If a silencer is located within a mechanical room, noise may enter the system through the sheet metal duct after the silencer. If a silencer is located away from the mechanical room walls, noise may escape the system through the sheet metal duct before it is attenuated by the silencer. Ideally, a sound 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.

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16) Use of Active Noise Cancellation Equipment

Unlike 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 [degree] 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.

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17) Equipment Selection

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.

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17 A) Rooftop units (RTU's)

The most frequent complaints about rooftop unit noise occur at night when the ambient noise is lower than the daytime hours. The noise of a rooftop installation may propagate to other nearby buildings. In such instances, the addition of sound attenuators on the intake and discharge or the installation of barriers in the source-receiver path may be required.

Roof top HVAC Noise can be solved in many ways:

- Locate RTU's with extreme care over toilet rooms, storage rooms, or other non-critical spaces.

- Do not place rooftop units on limber, long-span roofs. If the roof is not stiff at the mounting location, provide a structural steel frame to transfer the weight to bearing walls or columns.

- When locating HVAC units on the top of a building, it is best to position the unit over the least noise sensitive area.

- The need for vibration isolation requires evaluation early in the design process or else the noise criterion for the spaces directly below may not be achieved. Usually OEM internal isolation of the fan and motor is adequate.

- If a vibration isolation curb is provided, it should be the type that permits visual inspection of the springs.

- Provide roof openings only for the supply and return ducts—not one large opening for the entire unit. Beware the fan discharge condition in a rooftop unit. A downblast configuration is not advisable and rather horizontal discharge into a lined plenum section is better solution. Horizontal discharge into internally lined and externally insulated ductwork is best.

- If the rooftop unit has compressors and/or an economizer section, its noise may antagonize the neighbors. Obtain noise emission data and compare it to applicable local ordinances.

- Specially constructed screens and barriers can be used to control noise. Also, buildings or parts of buildings, earth berms or other formations can be used as attenuation devices.

- Some of the noise problems can be reduced by the way the unit is designed such as 1) eliminating belt squeal, 2) use of soft starters, 3) reducing fan speed and 4) the use of variable frequency drives.

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18) Noise from Fan Systems

Indoor noise related problems that come from central ventilation or/and the air-conditioning installations, are usually linked to the operation of fans. The noise resulting from a fan system is caused in several ways and can enter an area by more than one route.

- The noise energy generated by a fan will be fed into the duct system, forced to pass along its length, and a proportion will enter the area being served by the system. The balance of original noise energy not reaching the ventilated area will be absorbed by the system. One major cause of fan noise is surge. This is the result of periodic vibrations of the fan and ducts connected to it. It is caused by unstable operation of the fan. Surge commonly occurs when the actual static pressure is high, compared to the static pressure that the fan can reach at the particular speed at which it is operating. One way to check for surge is to relieve fan static pressure.

- Noise will also be generated by the airflow as it is obstructed and turned in its passage along the duct system. Sound energy will therefore, be introduced into the system at bends, dampers, heater batteries, etc., and again a proportion will pass into the ventilated area.

- Some of the noise absorbed by the duct system is in fact lost through the duct walls and can be a nuisance in those areas through which the ducting passes. This break-out noise can even reach the ventilated area itself. Airborne noise from plant room and vibration energy from the fan can also be transmitted through walls, floors and ceilings into adjacent areas.

Another cause of fan noise is resonance. The transmission of vibration energy depends on the resonance frequency of the machine on the resilient supports and on the energy damping in the system. The resonance will result in one or more sections of the duct system vibrating at the same frequency as a vibration produced by the fan. This can be checked by changing the fan speed by ±10 percent and noting whether the vibration stops. If the fan performance is not matched to the duct system, fan noise will increase. One possibility is that the fan may be handling more air than required. Reducing the fan speed would reduce noise.

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18-A) Choice of Fan

The choice of fan for particular application will depend on the volume airflow rate, static pressure needs and on cost.

- The types of fans used in HVAC application are either axial type or centrifugal type fans. Axial fans generate a higher proportion of high frequency noise but less low frequency noise than centrifugal fans of similar duty. Centrifugal fans produce most of their noise in the low frequencies. In general centrifugal fan is quieter. If the fan is close to a critical space, consider an airfoil fan and silencers since the low rumble of forward-curved fans can be difficult to silence.

- There are a number of types of fan selections available in floor-by-floor air handling units. With given equipment room layout, the types of fans, in terms of acoustical preference, are:

  ➢ Mixed flow

  ➢ Vaneaxial

  ➢ Plug

  ➢ Standard centrifugal

Generally, the price per cfm varies as a function of the acoustical performance—*i.e.*, a standard centrifugal unit costs approximately $0.50 per cfm while a mixed flow unit costs about $1.50 per cfm.

- Carefully select the most efficient fan for the application, as it will be the quietest. Poorly chosen inefficient fans can produce more noise than an efficient fan that moves ten times more air. If the fan is close to a critical space, consider an airfoil fan and silencers since the low rumble of forward-curved fans can be difficult to silence.

- Some high frequency noise will be removed by the 'natural attenuation' of the system (ducts, bends, etc.) and if necessary extra attenuation can be added in the form of a silencer. However, low frequency noise (63, 125 and 250 Hz octave bands) is much more difficult to remove. Therefore when choosing between two different fans of suitable duty, the one with the smaller sound output at the lower frequencies will be preferable from the noise point of view, even if the total sound powers of the two are the same.

- The lowest possible rotation per minute (rpm) is usually best for maintaining low noise levels. For least noise, an air mover speed must be set such that to provide the required airflow and pressure at about 70% of free airflow on its performance curve. Consider allowing maximum flow somewhat greater than for a fixed speed design.

- Care should also be taken to ensure that the chosen fan will always be operating at or near its maximum aerodynamic efficiency, i.e., at the recommended static pressure for the volume flow rate required. 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.

- Maximum noise usually occurs from the blade tip frequency of the fan. This is determined from the number of blades on the fan rotor multiplied by the number of revolutions per second. The octave band in which the blade tip frequency falls will have the highest sound power level.

- It is necessary to know the sound power level in each octave band. This information is supplied by the fan manufacturer. Fan sound power is determined through tests conducted in accordance with AMCA Standard 300, "Reverberant Room Method for Sound Testing of Fans." The alternative is to use one of the empirically based formulae which relate the sound power level of the fan to other of its properties such as the power of the fan motor, the static pressure developed and the volume flow rate of air delivered.

Estimating the sound power level from fans:

The following empirical formulae may be used approximate estimations:

$SWL = 90 + 10\log_{10}s + 10\log_{10}h$

$SWL = 55 + 10\log_{10}q + 20\log_{10}h$

$SWL = 125 + 20\log_{10}s - 10\log_{10}q$

Where

SWL = overall fan sound power level, dB

s = rated motor power   (hp)

h = fan static head   (in water gauge)

q = volume discharged   (ft^3/min)

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. Correction factors for different types of fan allow the octave-band levels to be obtained from the overall level. These formulae will only give approximate levels and data from test measurements should always be used in preference, if available.

18-B) Sound Power vs. Sound Pressure

Fan sound is a very important consideration in the selection and application of fans.

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

Most industrial fan catalog data is presented in sound power levels in each of eight octave bands. However, some commercial and residential non-ducted catalog sound data is presented on a sound pressure basis using a single number rating system such as dBA or sones.

It has been repeatedly emphasized that sound power level values are independent of the distance and acoustical environment. On the other hand, sound pressure levels are a function of the location of the source as well as the listener. To estimate sound pressure levels requires a detailed knowledge of many different parameters. Since fan manufacturers have no idea where and how their fans or the ultimate listener are located, they are not in a position to calculate sound pressure levels for most applications.

However, over the years, in order to provide users with some idea of what sound pressure levels might be expected, a "default set of assumptions" have been made which may or may not match the actual application. This default set of assumptions has been well accepted by users to the point where many catalogs contain sound levels based upon sound pressure. The following Para outlines the default assumptions and the calculation process used to obtain various catalog sound pressure levels.

- Point source- This assumes that the listener is far enough away from the fan to consider it a point source. This is consistent with most theoretical calculations made in acoustics. The key emphasis is that the listener is not in what is called the near field.

- Directivity factor- It is assumed that the fan is mounted on a floor, ceiling, or wall. Therefore, it can also be assumed that there is one reflecting surface that bounces the sound waves back toward the listener. This is referred to a directivity factor of two.

- Hemispherical radiation pattern- Consistent with the directivity factor, it is assumed that the sound radiates from the fan in a hemispherical radiation pattern. A spherical radiation pattern would mean that the sound radiates equally in all directions from the fan and does not have a reflecting surface.

- Straight line distance- It is assumed that the sound travels in an uninterrupted straight line from the fan to the listener. In other words, the listener can look directly at the fan and see it. No ductwork is between the fan and listener. *Typically, a distance of five feet is selected as being reasonable.*

- Free field conditions- It is assumed that the sound is free to radiate outwardly in an uninterrupted manner and is not reflected from any other surface other than the floor or ceiling it is mounted upon. The sound is free to just go and go.

- Constant difference between sound power and sound pressure- Based upon all of the previous assumptions, (five feet away from a point source in a hemispherical free field) it is possible to calculate the difference between sound power and sound pressure. This means that the sound pressure level is 11.5 dB lower than the sound power level regardless of octave band.

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18-C) Relationships – A-Level, Sound Power and Sound Pressures

Sound pressure levels which are heard by the human ear are based upon the "A" scale. There is a constant set of weighting factors illustrated in the following examples to approximate the response to the human ear.

It is important to realize that "A" weighting numbers are fully meaningful only when applied to sound pressure values. "A" weighting factors are sometimes applied to sound power levels which are then combined into a single number. This provides a single number for comparison between fans when sound power spectrums are provided.

The PWL number cannot be verified by measurements in the field.

Example # 1

Single Number Sound Power Level (PWL)

Band No.	1	2	3	4	5	6	7	8
Sound Power Level	75.5	78.5	74.5	74	70	65	61.5	57
A- Weighing	-25	-15	-8	-3	0	+1	+1	-1
PWL by Octave Band	50.5	63.5	66.5	71	70	66	62.5	56

Combining into a single number, PWL = 75.5 dB

Example # 2

Calculating Sound Pressure Levels

Band No.	1	2	3	4	5	6	7	8
Sound Power Level	75.5	78.5	74.5	74	70	65	61.5	57
Delta Power Pressure	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
Sound Pressure Level	64	67	63	62.5	58.5	53.5	50	45.5

Example # 3

Calculating Single Number Sound Pressure Level (dBA)

Band No.	1	2	3	4	5	6	7	8
Sound Power Level	75.5	78.5	74.5	74	70	65	61.5	57
Delta Power Pressure	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
Sound Pressure Level	64	67	63	62.5	58.5	53.5	50	45.5
A- Weighing	-25	-15	-8	-3	0	+1	+1	-1
A – Weighing Pressure	39	52	56	59.5	58.5	54.5	51	44.5

Combining into a single number (dBA) = 64 dB

Several sound level quantities can be calculated using combinations of the information as shown in examples above.

Knowing values of fan sound is vitally important. They should be obtained for every fan selection and compared to available acceptance criteria up front in the design stages. This will help provide the assurance necessary for a successful application.

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18-D) What is Sone?

An alternative loudness description is sones. A sone is a term of loudness perceived by the ear related to a frequency of 1,000 Hz. The sone is a sound pressure term at a distance of five feet from the fan and is linear to the human ear.

Sones follow a linear scale, that is, 10 sones are twice as loud as 5 sones. Use the following formula to convert sones to decibels:

dBA = 33.2 Log 10 (sones) + 28, Accuracy +/- 2dBA

PWL (LwA) and dBA respectively are sound power and sound pressure rating systems for most industrial and commercial fan equipment. A third single-number system, is used for non-ducted propeller fans and power roof ventilators, is called the sone.

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18-E) Fan Characteristics

It is often mistakenly assumed that the largest diameter fan wheel operating at the lowest rpm will always generate the least sound power. In fact, large, slow fans can generate higher levels of low-frequency noise than smaller, faster fans. Depending on the duct system's air-flow and static pressure drop, a smaller, high-speed fan operated near peak mechanical efficiency may produce lower overall sound power levels. At the very least, such a fan may generate sound power in higher-frequency bands, which is easier to control than low-frequency sound.

It is advisable to consider variety of fan sizes that comply with air flow and static pressure requirements. Selection can then be based on the unit that either produces the lowest octave band sound power levels or requires the least overall cost of silence.

Fan selection should be based on mechanical efficiency. Sound power levels produced by a fan increase as the fan operating point shifts from the point of peak mechanical efficiency. Remember, poorly chosen inefficient fans can produce more noise than an efficient fan that moves ten times more air.

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18-F) Fan discharge velocities for quiet operation

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

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18-G) Fan Controls

After a fan selection is made, the type of fan controls affect the acoustical performance. Variable speed control and inlet vanes are control methods commonly used in variable air volume systems.

The acoustically preferred control method is speed control. Varying the speed of the fan also varies the acoustical output of the fan. At 100 percent design air volume, the method of control does not impact the

acoustical performance. However, systems operate at 80 percent or less of design air volume most of the time. At reduced air volumes, speed control not only saves energy but reduces the overall acoustical impact of the unit on occupied spaces. Noise levels can be reduced by 3 to 5 dB at reduced air volumes (a 3 dB change in noise level is barely perceptible, and a 5 dB change in noise level is clearly perceptible).
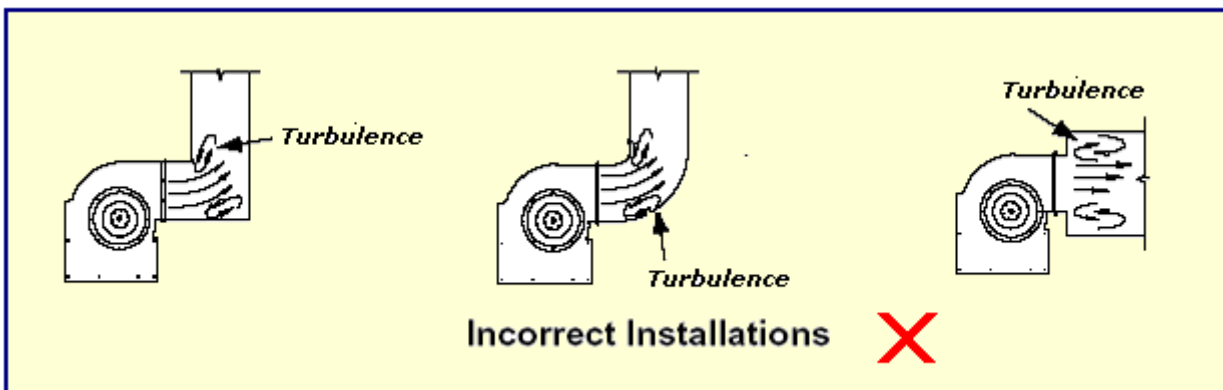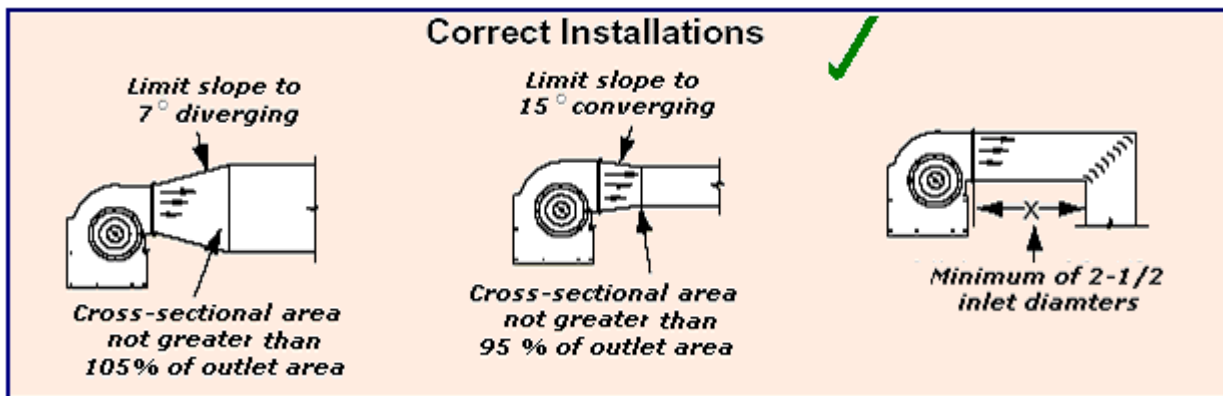
18-H) Fan Arrangement & Installation

The sound power levels specified by the manufacturer will be the minimum noise level, which it will emit, under the most favorable installation conditions. These occur when the flow of air into the fan is smooth and uniform. If as a result of poor installation design or practice the airflow is non-uniform or turbulent, then the fan will create a considerable amount of extra broadband noise. Such conditions can be caused by obstacles in ductwork which are close to and upstream of the fan, or if the fan is downstream of and too close to a bend. Obstacles and bends should be at least one fan diameter upstream of the fan to allow the airflow to smooth out before entering the fan.
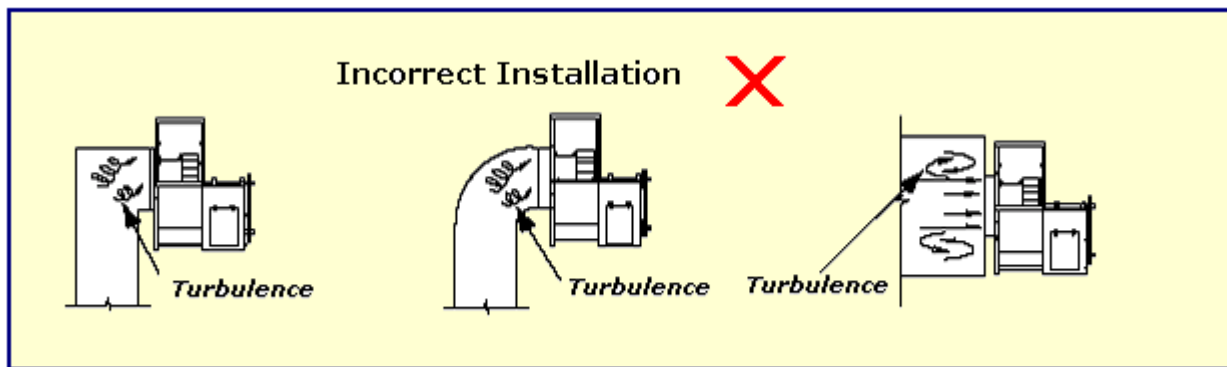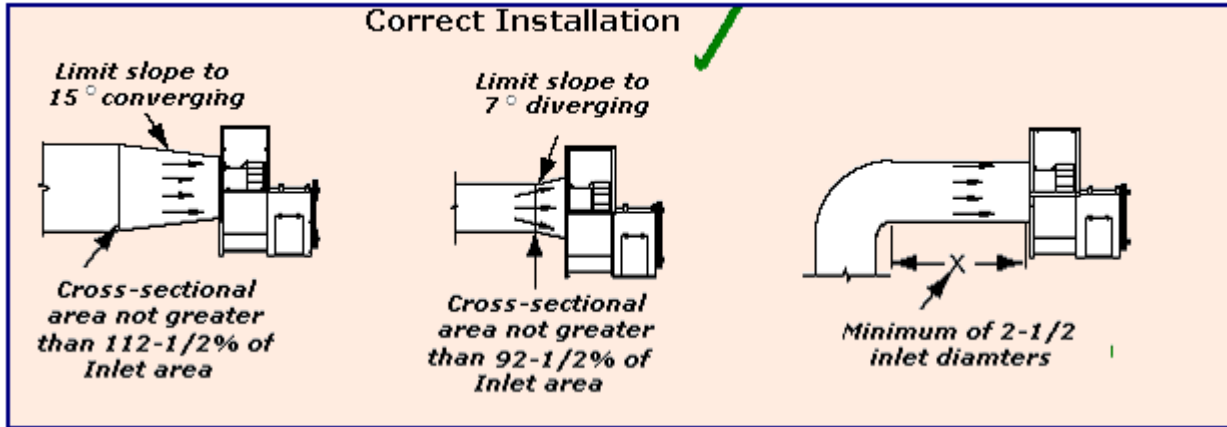
Air which has to 'turn sharp corners' into the fan will also cause extra noise, which may be reduced by using a tapered or a bell-mouth inlet to the fan. Obstacles such as struts, guide vanes and instrumentation probes which are close to the fan blade can also cause a very great increase in the pure tone component of the fan noise, as a result of the periodic chopping action on the air in the small gap between obstacle and blade.

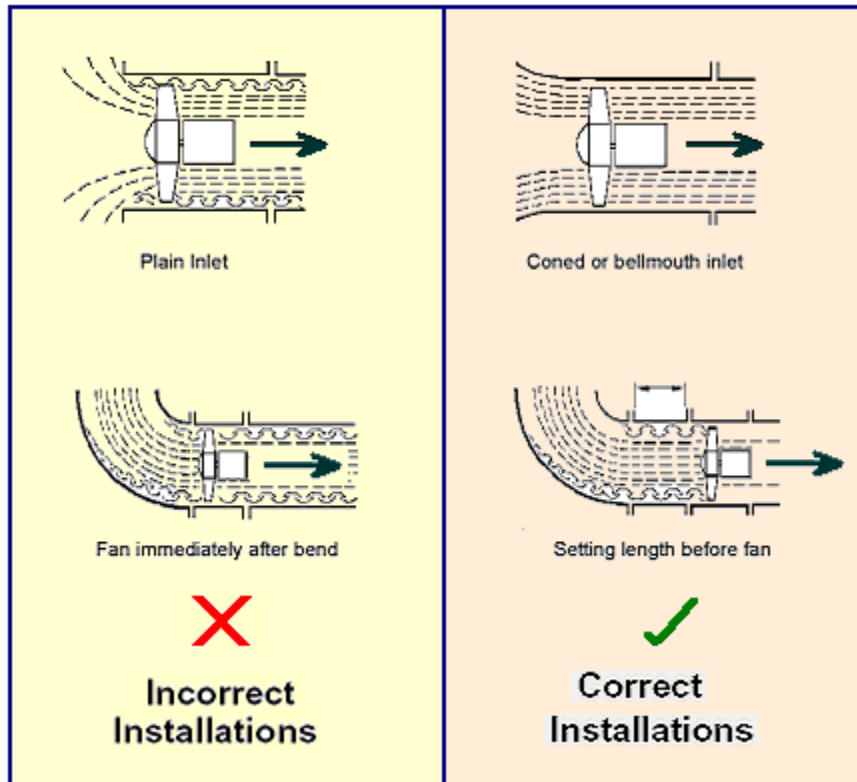Figure below shows designs which will cause turbulence (and hence noise) and how they can be improved.

Centrifugal Fan –Typical Outlet Connections

Centrifugal Fan –Typical Inlet Connections

Correct Installation ✓

Limit slope to 15° converging

Cross-sectional area not greater than 112-1/2% of Inlet area

Limit slope to 7° diverging

Cross-sectional area not greater than 92-1/2% of Inlet area

Minimum of 2-1/2 inlet diamters

Incorrect Installation ✗

Turbulence

Turbulence

Turbulence

Axial Fan –Typical Inlet Connections

Plain Inlet

Coned or bellmouth inlet

Fan immediately after bend

Setting length before fan

✗

✓

Incorrect
Installations

Correct
Installations

Air flow at the entrance and exit of a fan should be as smooth as possible to minimize the generation of turbulence. Conditions that produce turbulent air flow usually result in greater noise generation and increase static pressure drop in the system. The air flow on the outlet side of a fan is always turbulent for at least 3 to 6 duct diameters downstream. Fittings (such as elbows or sudden transitions) placed closer to the fan than this distance may result in noise problems.

The engineer should remember that noise generation within an air distribution system is caused by aerodynamic turbulence. If, therefore, he conforms to the codes of recommended design practice, paying special attention to those areas where turbulence is like, both aerodynamic and acoustic efficiencies will improve.

18-I) 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.

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19) Air Terminal Sound Issues

Air terminals 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. Room air devices such as diffusers, grills, light fixtures, and air handling ceiling suspension bars are always rated for noise generation. The room air terminal unit is selected to meet the noise criterion (N.C.) required or specified for the room, bearing in mind that the manufacturer's sound power rating is

obtained using a uniform velocity distribution throughout the diffuser neck or grill collar. When balancing dampers are installed immediately before the diffuser, or if a duct turn precedes the entrance to the diffuser, airflow will be turbulent and the noise generated by the device will be substantially higher than the manufacturers published data.

Good design practice dictates that a designer establish the acceptable noise for the occupied space and then determine the selection criteria for both radiated and discharge sound.

Radiated sound "breaks out" from the terminal casing or induction port and travels through the plenum and ceiling to enter the occupied space. Discharge sound travels out the discharge of the terminal through the duct work and outlet to enter the occupied space. Make note of the following:

- 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 grille. A poorly selected grille can cause all kinds of noise – whistles, rattles, etc. Depending upon how severely the dampers must be closed to balance the system, noise levels can increase from 5 to 20 db above ratings. In critical applications, it is preferable to use dampers placed well upstream in the supply duct to balance the air flow.

- While velocity is perhaps the most important criterion governing terminal noise, size is also a factor. Given two identical diffuser models, experiencing the same air velocity through the outlet, the smaller outlet will generate more noise. This is also true for return air intakes. If you're working with an HVAC expert, let him/her know that you require "air devices" with "NC" ratings of "30 or less" *for the designed airflow*.

- 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. If the noise disappears when the outlet or inlet is removed, obviously the register is to blame.

- Provide 3 diameters straight flex at diffuser inlet

- 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. 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. A common problem is buildings designed with inadequate plenum space to allow this. A turn near the diffuser results in uneven distribution of air over the diffuser and strongly increased noise.

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20) **Use of Variable Air Volume (VAV) Terminal Boxes**

VAV systems are commonly used in a wide variety of commercial and institutional buildings, including hotels, hospitals, office buildings, and retail applications. These systems are usually seen as being beneficial to building owners and occupants, in that they both save energy and can provide a high level of comfort. VAV systems are different from constant volume systems in that they adjust the airflow to the occupied space based on the building's heating and cooling requirements. When less heating or cooling is needed, the terminal box reduces the amount of air that is supplied to the space.

VAV systems rely on control dampers that can create airflow-generated noise, which can travel into the occupied space. The terminal units are typically located above the occupied space and their casing-radiated and discharge sound levels can affect the amount of noise in the occupied space.

There will almost always be noise in the space when the terminal unit is sitting above the ceiling. The intensity of noise depends on numerous factors, including the application involved, ceiling height, room sound levels, occupants, and time of day. Noise is definitely subjective: What's objectionable to one person may be just fine to another. Occupants may be more aware of the noise in a VAV system simply because it causes changes in the airflow, and people notice that. Constant volume systems can be just as noisy, though their sound levels are constant.

Air terminals are commonly specified and reported in NC. More importantly, most people cannot differentiate between two sources which differ by less than 3 dB. If the background sound is an NC–35, and the device in question is predicted at an NC–35, it is likely that the space will be at an NC–38 (although this is dependent on which octave bands are critical), but most people cannot hear the difference.

In a similar manner, sound from VAV boxes and diffusers combine to create the room sound pressure level. Since they peak in different bands, however, they often complement each other. In many cases, a Series Fan Terminal will be predicted to have an NC–30 in a space, but when combined with an NC–40 diffuser, will result in a room sound pressure level of NC–40, which is optimum for providing speech privacy in open plan spaces.
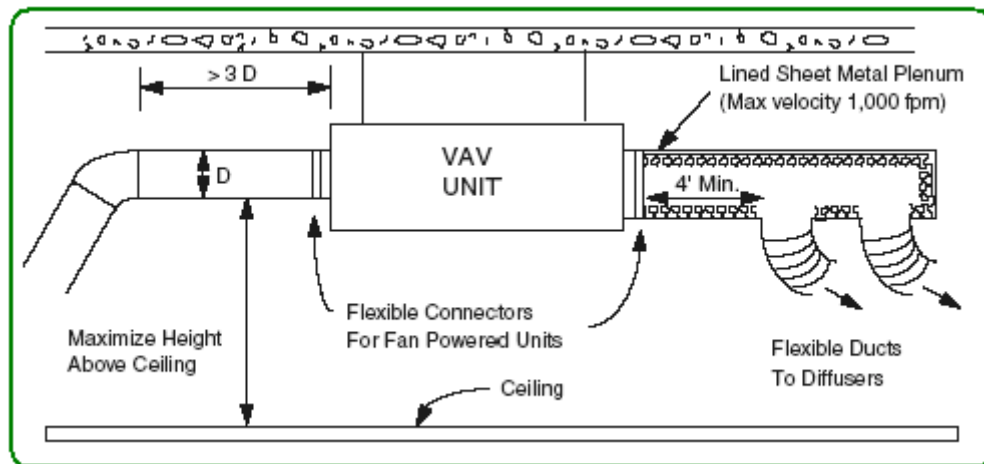
Quiet VAV Terminal Installation Guideline

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, ductborne supply and return noise, cabinet radiation, and break-out noise must all be considered along with attenuation of ductwork.

Sound Design Guidelines

Engineers can minimize the sound contribution of air terminals to an occupied space through good design practice.

- If the box is located above a critical space, and separated from the space by a suspended acoustical ceiling that has little or no transmission loss at low frequencies, the radiated noise from the box may exceed the noise criterion for the room below. In such a case, relocate the box over areas less sensitive to noise. This includes corridors, copy rooms, storage rooms, etc. Quiet air terminals facilitate the location of terminals over unoccupied space as with these units larger zones are possible resulting in fewer terminals. This also reduces first cost and improves energy efficiency.

- 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. Frankly, three or four feet of flex duct can cut noise as much as 10 or 12 NC points.

- Duct liners can reduce mid- and high-frequency noise levels but need substantial length to be effective and do very little in the low frequency region where most HVAC noise complaints are focused. Silencers can be supplied in lieu of duct liner at a cost savings and to meet noise criteria that may otherwise be unachievable.

- Sound will be reduced when appropriate fan speed controllers are used to reduce fan rpm rather than using mechanical devices to restrict airflow. This form of motor control is often more energy efficient.

- 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.

- 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.

- Sharp edges and transitions in the duct design should be minimized to reduce turbulent airflow and its resulting sound contribution.

- While noisy VAV systems are a common problem, another issue is when they are too quiet. This is especially true in open-plan offices, where to achieve speech privacy, a system should be at around NC 40, but in a VAV system, that noise isn't constant. If we know the client wants background sound masking, you should not design a VAV system to meet an NC 35, because it is not needed. The sound masking system will increase that level up to NC 40 anyway.

- 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.

- 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.

- Minimize inlet static pressure to VAV box; Select VAV box for minimum noise in 125 Hz band (< 65 dB radiated, <70 dB ducted); Do not use flex connection at VAV inlet; Vibration isolators are not required

- 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.

- For variable volume applications, use variable speed controls, not inlet vanes. Recognize that the NC noise ratings of these devices assume certain conditions that may be different from actual system and room design. Read all footnotes explaining the ratings carefully.

- 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.

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21) Downsizing the equipment during design

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.

21-A) 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. 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.

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21-B) 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.

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22) 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.

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23) Controlling Noise from Air Handling Units

Let's look at the air handler, in which fan efficiency, static pressure, motors, and drives can be optimized, as well as the AHU interface to the air-distribution system and the utility systems supporting it.

The overall design procedure is to some extent iterative; if, after an initial calculation, noise criteria are exceeded, changes must be made and a further set of calculations must be carried out. The procedure can be described in overview 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. This will 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. 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.

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24) <u>Controlling Noise from Mechanical Room</u>

Figure below shows a source (air-handling unit) placed on a resilient support inside the mechanical room. The noise control from mechanical room involves:
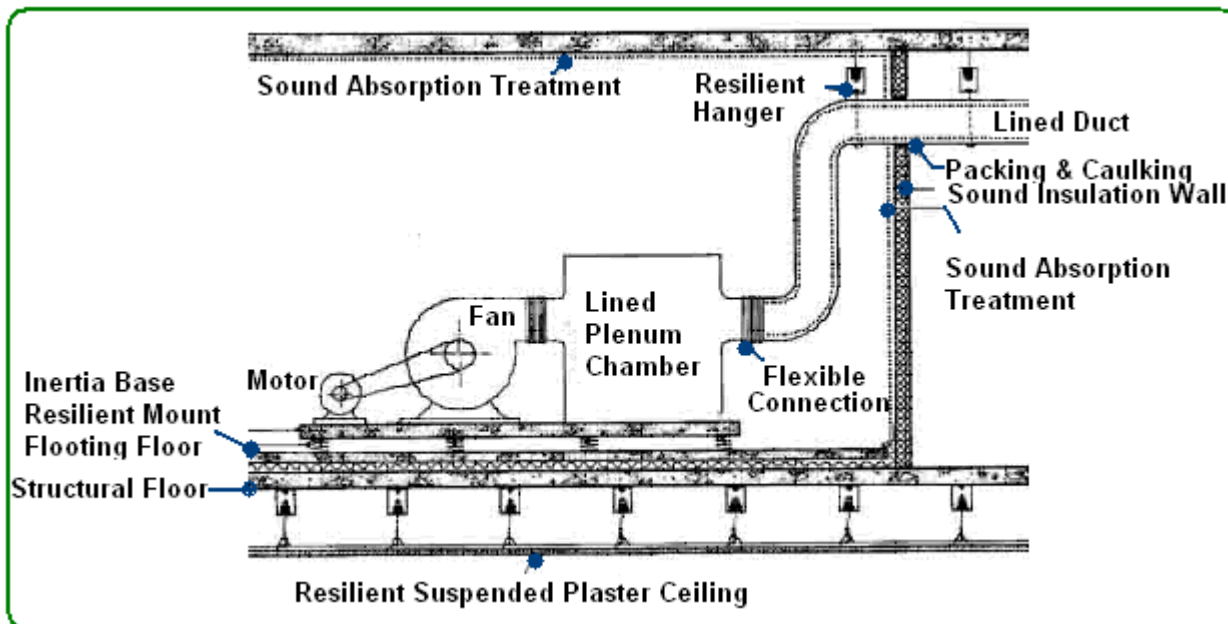
Two important techniques used in almost all noise control situations involve:

a.   **Sound Absorption and Reverberation:** One method for controlling the sound level within a room is through dissipation of the sound energy in absorptive materials. Generally, higher frequencies are more easily absorbed than low frequencies. *Materials that are good absorbers permit sound to pass through them relatively easily; this is why sound absorbers are generally not good sound barriers.* They reduce the level of noise inside an enclosure, because while the sound waves are being reflected from the surfaces in the room, they interact with the sound-absorbing materials and lose some energy each time.

b.   **Resilient Mountings:** Most machines generate vibration, which can be transferred to the structure of a building through the machine supports. The vibration level can be reduced by balancing the machines to reduce the forces at the source. Transmission into the building structure can be reduced by 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. Concrete bases are also used with fans because they can reduce the amplitude of oscillation of the equipment. This reduction, in turn, reduces the transmission of vibration through connected piping and duct. Flexible connectors should be used 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.

The extent to which these measures are necessary will depend on a number of factors. If quiet machines are selected for use in a building and are placed as far as possible from areas where quiet is required, fewer noise control procedures will be necessary. Criteria should be set during the design stage and inspection or measurements made during or after construction to ensure that the criteria are being met.

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Concluding, the planning stage is the best time to consider these problems and address them cost-effectively. It is fairly economical to design an air-handling system to minimize unwanted noise, but once the space is built, options are much more limited and solutions more costly. Adding sound control measures such as those described above may add one to five percent to initial cost, but trying to solve the problem after construction is completed can easily drive up costs as much as 50 percent.

If none of these measures reduce the noise level sufficiently, then it may well prove necessary to attempt to alter the acoustic characteristics of the area being served in an effort to absorb more of the sound energy discharging from the HVAC system. However, depending on circumstances the reduction achieved by this method may be somewhat limited. An acoustic engineer should be consulted if such a stage is reached.

However, be aware that reducing noise levels below reasonable values is incredibly expensive. In fact, the total cost of a system increases exponentially as the NC/NR rating is lowered. For this reason, the target noise level resulting from a HVAC system must be chosen with care after studying the guides and the conditions special to the contract.

Remember - there is no justification in paying for a HVAC system which would generate a background level of only NR25, if the noise from an adjacent road would make NR35 more appropriate!

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<div align="right">**Annexure - 1**</div>

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.

General rules of thumb for controlling noise within a space:

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.

General rules of thumb for controlling noise between spaces:

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

16) 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 = 2×10^{-5} Pascal

- For Sound Power Level (SWL) datum   = 1×10^{-12} Watts

- S.P.L. (dB) = $20 \log [P / (1 \times 10^{-5})]$, 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.

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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
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 m³/h) • Cut-off saw

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
		• 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) • 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 • Room in a quiet dwelling at midnight
0.0000000001	20	Noise at ear from rustling leaves

Sound Power (Watts)	Sound Power Level (dB) (re 10^{-12} W)	Source
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