APPLICATION AND INSTALLATION GUIDE

NOISE

CATERPILLAR®

Contents

Foreword

This section of the Application and Installation Guide generally describes Noise, its causes and suggested corrections on Caterpillar® engines listed on the cover of this section. Additional engine systems, components and dynamics are addressed in other sections of this Application and Installation Guide.

Engine-specific information and data are available from a variety of sources. Refer to the Introduction section of this guide for additional references.

Systems and components described in this guide may not be available or applicable for every engine.

Noise

There is increasing world-wide pressure to reduce noise in our environment. This is evidenced by a pattern of legislation that imposes an obligation on manufacturers and suppliers to produce machinery that does not generate noise levels that induce hearing loss.

Caterpillar has played a leading part in the field of noise control on engines and engine installations. This effort spans many years and continues to benefit both engine users and the communities in which the engines are used.

The intent of this guide is to provide a greater understanding of noise and noise control.

This is achieved in three major parts:

- Review the basic theory and nomenclature of noise.
- Define the sources of engine-related noise.
- Outline appropriate noise control methods.

SECTION CONTENTS

Sound

Noise is sound. Specifically, any unwanted sound. Any discussion of noise reduction or noise control is, essentially, sound control. So to describe noise, we must first understand sound.

Sound is a pressure which makes the membrane in the human ear deflect. Sounds begin as a vibration like the ringing of a bell. As the bell vibrates, it disturbs the air around the bell. This disturbance radiates through the air and is sensed by the ear. The fluctuation in pressure is interpreted by the brain as sound.

Other terms to describe sound pressure are sound level, strength, power, amplitude and loudness.

Sound pressure is a pressure and it can be reported in units on the following scales:

- 20 μPa (micropascal)
- 1 Atmospheric Pressure
- 1 bar
- 100 kPa
- 14.5 psi
- Threshold of Hearing
- 0 dB (decibels relative to 20 micropascal)

The ear can also tolerate pressures a million times higher. This extensive, non-linear, range requires a logarithmic scale to measure the sound pressure. The scale is divided in units called decibels (dB). A decibel is the relative measurement of amplitude of sound. **Figure 1** illustrates sound pressure in dB and psi for some common sounds.

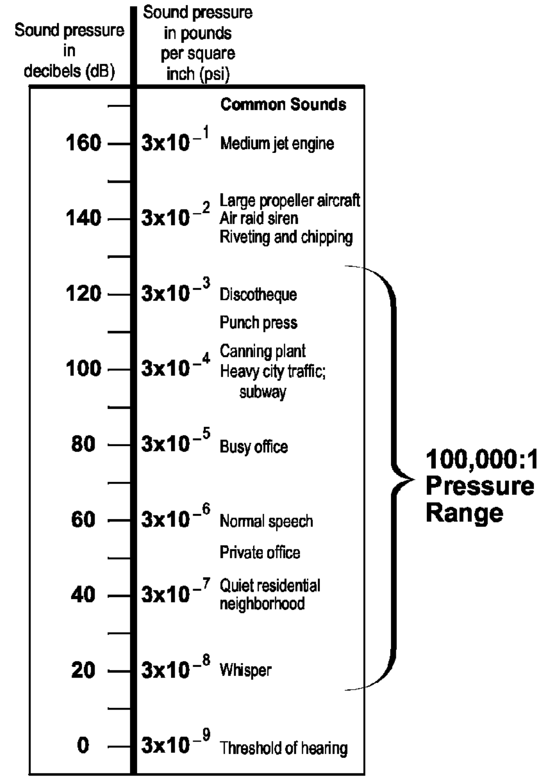

Figure 1

It is important to understand that an increase of 10 units anywhere on the decibel scale is not just “a little” louder, it is 10 times louder in terms of sound intensity, representing a factor of 10 times the power of generated sound, than the initial value. On the other hand, the human ear is a non-linear instrument, and this increase of dB is sensed as a doubling of loudness (based on statistical testing of audiences exposed to sound of varying loudness).

In addition to being non-linear, human hearing is also frequency sensitive.

The frequency of sound refers to the rapidity or cycles of an oscillation in a unit of time. The conventional unit is Hertz (Hz). One Hz is equal to one cycle per second. Also referred to as pitch, rapidly oscillating sounds have a higher pitch than lower oscillating sounds.

Frequency sensitivity is illustrated in **Figure 2**. The graph shows three sounds of equal loudness. Note that lower frequencies must be produced at higher decibel levels to be heard equally with higher frequencies. In general, sounds with frequencies in the 1,000-4,000 Hz range are the easiest to hear; sounds with very low frequencies are the hardest to hear.

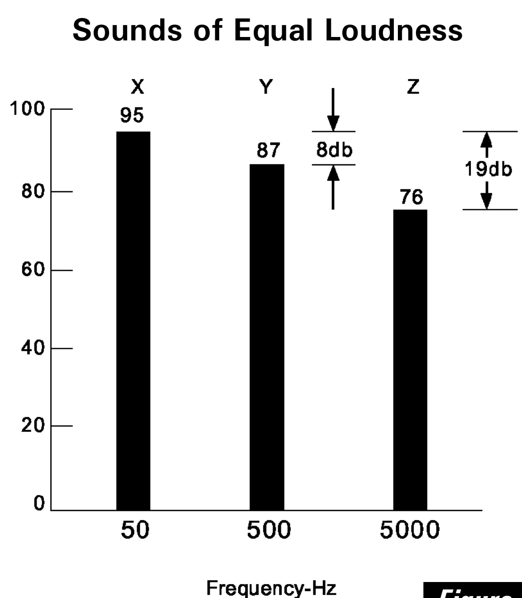

Figure 2

The range of frequency dependant hearing sensitivity in humans, as Illustrated in the figure above, is encoded in the commonly used A-scale, as shown in the figure, below.

Xxxxxxxxxxxx A-scale

Sound Waves and Measurement

As sound waves radiate from a source, their strength decreases as the distance increases.

For every doubling of the measuring distance from the center of radiating machine, its noise reduces by 6 dB. This rule is generally valid for measurements taken at a distance from the center greater than the largest dimension of the device.

Refer to **Figure 3**. For example, the first sound reading for an installation whose greatest dimension is "X" should be taken at a distance from the center of the noise source equal to X. (correct illustration change 117 to 114, and show distances from the center of the device.)

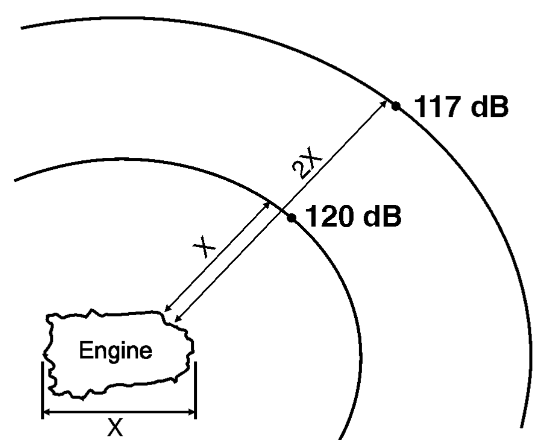

Figure 3

At that point, the following rule applies:

For each doubling of the distance from the first measuring point, sound strength is reduced by 6dB.

The following table illustrates this relationship another way.

Distance	Strength
x	120
2x	114
4x	108

Sound is mostly directional, meaning that the sound tends to move more in one direction than another. High frequency sound is more directional than low frequency sound.

The contour of the sound wave can be complex as shown in **Figure 4**. By measuring the sound pressure level around the engine, the contour of the sound wave can be determined.

It is not only the source of the sound which will give the direction, but also any kind of reflective surface in the area of the engine, i.e. floor, walls or ceiling.

Contour of a Sound Wave

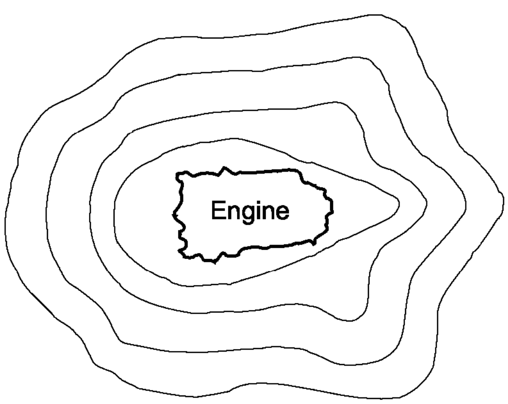

Figure 4

Amplitude & Frequency

Amplitude and frequency are the only sound properties that can be measured using ordinary techniques.

Microphones produce voltage proportional to the sound waves that act upon it. The voltages indicate the amplitude, of the sound pressure waves. The number of waves, or cycles, in a given amount of time indicate the frequency.

Sound Pressure Level

Sound Pressure Level (SPL) is a measure of the energy of a particular sound relative to a reference sound. In almost all situations SPL is expressed in decibels and compared to a standardized value of 20 μPa (micropascal) or 2 x 10^{-4} microbars which is equal to 0 dB.

Sound Pressure Level (SPL) in decibels equals:

$$20 \text{ Log}_{10} \left[\frac{\text{Measured Pressure}}{\text{Reference Pressure}} \right]$$

The relationship between μPa and dB is such that multiplying the sound pressure (μPa) by 10 will add 20 dB to the dB level.

Because of their logarithmic nature, differences in two decibel ratings will indicate the wave strength ratio between two measured sound levels.

While the non-linearity of the scale may be difficult at first, it will become useful once the following relationships are learned.

Difference In Two Signal Levels In Decibels	Pressure Level Ratio
1	1.12 to 1
3	1.41 to 1
6	2.00 to 1
10	3.16 to 1
12*	4.00 to 1
20	10.00 to 1
40	100.00 to 1

** Illustrated in Figure 5.*

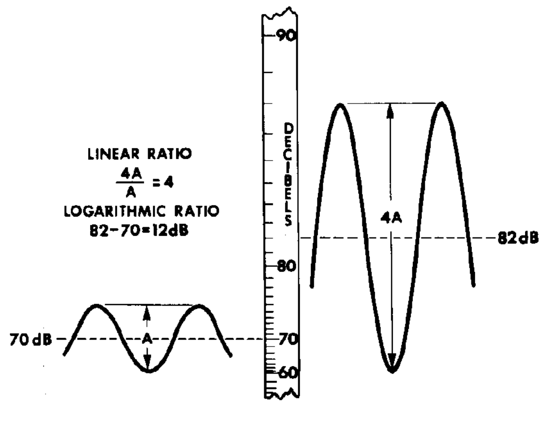

Figure 5

Weighting Networks

Since human hearing is nonlinear and more sensitive to high frequencies than low frequencies, measurements are adjusted, or weighted, to match the sensitivity of the ear, or other function.

Sound pressure is measured by 1 microphone.

The signal from the measuring microphone is then passed to an amplifier and an attenuator which is calibrated in decibels.

Next, the signal is passed to one of four weighting networks referred to as A, B, C and D. Each network modifies, or filters, the input signal accordingly.

The most commonly used network weighting is A. The A-scale is intended to mimic the response of the human ear. It is also known as (A-scale), dB(A). Characteristics of weighting network A is shown in **Figure 6**. Note that the A-network is most sensitive at 2,000 Hz, the point where humans typically have the highest hearing sensitivity, and reduces rather dramatically at lower frequencies, to the point that at 63 Hz the A-filter corrects the measured level downward by 26 dB.

Response Characteristics of Weighting Network Filter "A"

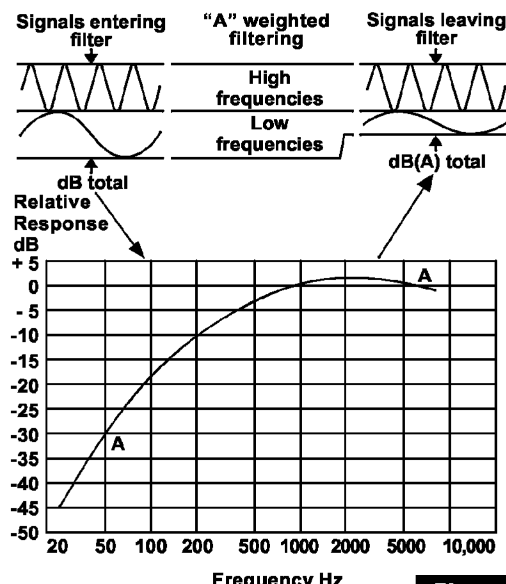

Figure 6

Octave Band Levels

Measuring noise by spectral plots of the sound pressure in various frequency bands is more descriptive of sounds than just the single values from the A, C, or other networks. Generally the simplest frequency division is by Octave frequency bands.

Measurements are made with filters subdividing sounds over the entire audible range into standardized frequency bands, permitting the pressure levels of only the sound within each subdivision to be measured. In the case of the Octave Spectrum, each filter spans one octave. **Figure 7** shows octave bands having center frequencies from 63 Hz to 8000 Hz.

Standard Octave Bands
(ANSI Standard S1.11 IEC 225)

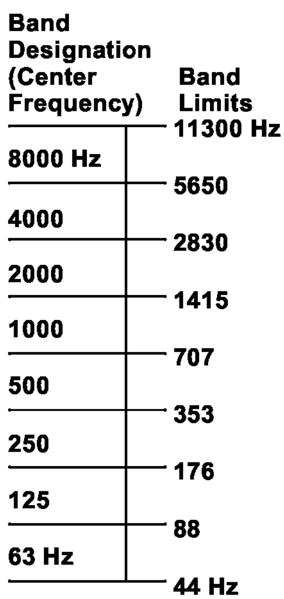

Figure 7

Sound Power

While sound pressure is the actual pressure measured at a point including direct and indirect (reflected) sounds, and is what the ear hears. Sound Power is used to measure the total noise emission from a device, and its value is not affected by distance from the device or by reflections. Sound Power ratings have no information regarding the directivity of radiation from a device.

This is analogous to a light bulb; wattage (power) is a constant, while intensity (pressure) will vary with distance, and the lens directivity.

Sound power is the starting point for consideration of the sound pressure which will be measured at a given location, from one or many noise emitting machines.

Sound Power, is expressed as Octave Band spectrum values, and sometimes Sound Power is given with A-network filtered values. The Sound Power emissions of a device are determined by measurements made around the operating device in free field conditions or inside a reverberation chamber. Certifialbel measurements are performed in comformance with ASTM or ISO standards.

The normal expression of Sound Power Level (PWL) is given is decibel units
PWL = 10*LOG10(Acoustic Watts Emitted / 10^{-12} Watts)

With sound power, 80 dB expresses an acoustic radiation of 0.0001 watts. In this scale, a difference of 3 dB is a ratio of 2:1; 10 dB a ratio of 10:1.

The chart in Figure 8 illustrates differences in decibels and ratios in sound pressure and power. Sound power, in decibels, is a measure of the total sound radiation from a unit; while sound pressure, also in decibels, is the strength of a sound wave after it travels a specified distance from the unit.

The two decibel scales are related despite the discussed differences. The change in one will produce the same numerical change in the other.

For example, if the sound power of an engine was increased by 10 dB, the sound pressure of that noise at any given point would also increase 10 dB.

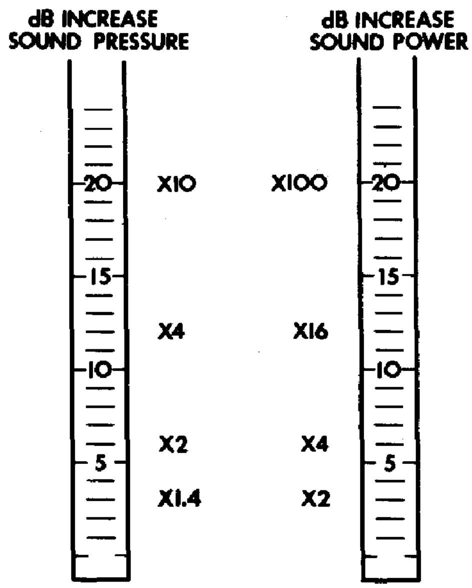

Figure 8

Sound Addition

Sound from multiple sources will have an additive effect on the overall sound level heard by the human ear.

For example, when standing by an engine, the sound heard from other engines operating in the same area will be increased, but the extent of the perceived increase will depend on the spacing of the engines and where the person is in relation to the spacing.

In addition, it must be noted that overall sound levels in dB(A) cannot be determined by simple addition. It must also include the addition of acoustic energy. This is done by taking the logarithm of the algebraic sum of the acoustical powers. Therefore, adding dB(A) levels requires some logarithmic manipulation; the determination of their antilogs, then their addition, and finally the determination of the total's logarithm.

A graph showing the combined effect of up to ten equal sound sources is shown in **Figure 9**. As illustrated in the graph, two engines, each individually producing 100 dB(A), would effectively produce 103 dB(A) when operating together.

These two sets of dB addition apply to Sound Pressure at a particular measuring position, in comparison to the sound pressure at that location from the individual devices. They also apply to Sound Power of more than one sound emitting device on a single site.

Addition of Equal Sounds

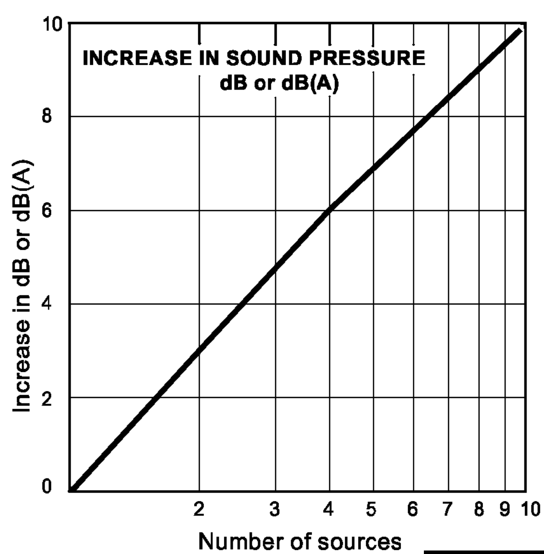

Figure 9

Figure 10 illustrates the combined effect of two unequal sound sources.

For example, the combination of a 100 dB(A) sound source and a 106 dB(A) sound source would result in a 107 dB(A) sound level. To add a third sound source, use the same process to combine it with the total of the first two.

Addition of two Unequal Sounds

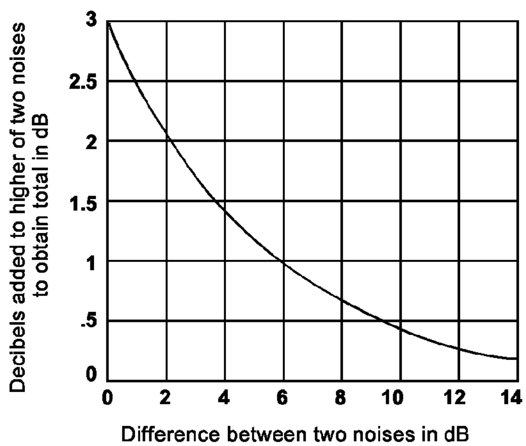

Figure 10

Sound Power and Pressure Conversions

Sound level information is presented both in terms of sound power level, L_W (dB(A)), and sound pressure level, L_P (dB(A)). L_W is the total sound power being radiated from a source and its magnitude is independent of the distance from the source. Relative loudness comparison between engines is simply a comparison of their sound power levels at equivalent operating conditions.

Note: Some Caterpillar Technical Data Sheets will refer to sound power level as SWL and sound pressure level as SPL. These terms are equivalent to the terms L_W and L_P used in this section of the guide.

When the sound power level (L_W) is known, the sound pressure level (L_P) at any distance from a point source (such as exhaust noise) can be calculated.

The equation for determining the sound pressure level of exhaust noise without any correction for ambient temperature and pressure is:

$$L_P \text{ in dB(A)} = L_W \text{ in dB(A)} - 10\text{Log}_{10}(C\pi D^2)$$

Where:

C = 2 for exhaust source adjacent to a flat surface, such as a horizontal exhaust pipe adjacent to a flat roof

or

C = 4 for exhaust source some distance from surrounding surfaces, such as a vertical exhaust stack some distance above roof

D = Distance from exhaust noise source (m)

For C = 4

$$L_P = L_W - 20\ \text{Log}_{10}\ D - 11$$

L_P measurement requires only a simple sound level meter. However, this measurement is the sum of sound waves arriving from every direction; It is dependent on the acoustic characteristics of the environment and varies with position relative to the noise source. L_P

cannot be used to describe the strength of a noise source without specifying relative position and room acoustic properties of the test environment. A disadvantage is that sound pressure level conversion is valid for a point source or at large distances from a distributed source, such as a machine. In the case of large machines, the conversion is valid to one dB or less so long as the measurement distance is equal at leat four times the maximum lenth of the machine.

If the sound pressure level of a point source at some distance is known, the sound pressure level at another distance can be calculated using this formula:

$$L_P2 = L_P1 - 20 \times \mathrm{Log}_{10} (D_2 \div D_1)$$

Where:

L_P1 = known sound pressure level, dB(A)

L_P2 = desired sound pressure level, dB(A)

D_1 = known distance, m (ft)

D_2 = desired distance, m (ft)

Noise

Noise, or unwanted sound, can disrupt verbal communication and produce adverse psychological effects. Excessive noise can even cause physical damage.

Psychological effects include irritability, anxiety and difficulty concentrating.

The physiological impact of noise can include short-term hearing loss, possible long-term hearing loss and headaches. Hearing loss from exposure to noise is frequency sensitive.

As mentioned before, exposure to excessive noise can cause permanent hearing damage and adversely affects working efficiency and comfort. Recognizing this, the U.S. Government created the Occupational Safety and Health Act (OSHA) which established limits for industrial environments.

When an individual's daily noise exposure, designated D(8), is composed of two or more periods of noise at different levels, the combined effect is calculated by:

$$D(8) = (C_1/T_1) + (C_2/T_2) + \ldots + (C_N/T_N).$$

Where:

C_N = Duration of exposure at a specified sound level

T_N = Total time of exposure permitted at a specified sound level (Refer to the following table)

The noise exposure is acceptable when D(8) is equal to or less than 1.

Duration of Daily Exposure (hours)	Allowable Level dB(A)
8	90
6	92
4	95
3	97
2	100
1.5	102
1	105
0.5	110
0.25	115

Caterpillar Noise Data

Caterpillar provides noise values (L_P) for engines at different ratings. The noise levels are:

Free Field
Free field means that it is a 100% open area with no sound reflections or other modifying factors. It is important to recognize that most engines or packaged units are installed in a building, vessel or other location, where sound does reflect off surrounding surfaces, so on-site sound measurements will be higher than the published free field noise levels.

Sound Pressure Level (L_P) — Mechanical or Inlet & Exhaust
Sound pressure level is presented under two index headings, mechanical and exhaust.

Mechanical
Sound pressure level data is obtained by operating the engine in an open "free" field and recording sound pressure levels at a given distance. The data is recorded with the exhaust sound source isolated.

Inlet & Exhaust
Sound pressure level data is recorded with the mechanical sound source isolated.

Measurements
All measurements are for "without" radiator fan arrangements.

Engine Installations

Some installations require very little noise abatement, for example, a remote facility far from people. Very sensitive installations, on the other hand, may require extensive noise abatement measures. Because of the variety of noise criteria that may apply to a given site, it is impossible to provide a description of abatement measures meeting all site criteria. It is the responsibility of the facility designer to ensure that the specific criteria of the site are met.

It is strongly advised that a noise control expert be involved in the facility design process from the beginning if the engine unit is to be installed in a building or area that is noise sensitive. Since internal combustion engines produce high noise levels at low frequencies, many traditional noise control approaches are relatively ineffective. Every aspect of facility design must be reviewed with special emphasis on low-frequency attenuation characteristics in order to meet site criteria.

This section provides information for designing Caterpillar large-engine unit installations to meet site noise criteria. Large-engine units include an engine and some piece of driven equipment, such as a generator or a compressor. Guidelines for installation design are provided, along with information on using noise data on Caterpillar units.

SECTION CONTENTS

Typical Design Approach

Considerations

A typical approach to designing an engine installation is:

- Recognize the special requirements of engine installations. The first step is to become aware of the special noise characteristics of engine installations. Possible sources, paths, and receivers of large-engine noise are reviewed.
- Identify site noise criteria. For example, is the installation in a remote or a populated area? Is it within a building sensitive to noise (for example, a laboratory or a hospital)? What regulations, standards, or restrictions apply to noise? The noise criteria form an essential part of the design goals. Since criteria vary from site to site, this guide cannot identify all the criteria that apply to a particular site. However, some guidelines for site noise criteria are provided.
- Obtain noise data on the engine unit. Noise data provided by Caterpillar is available in the Technical Marketing Information (TMI) system. Note: Data is free field and will be affected by structures, walls and enclosures. Acoustical design engineer needs to take this into account.
- Identify and select appropriate noise abatement measures. Guidelines for attenuation of noise, both through commercially available equipment and through facility construction, are provided.

Source/Path/Receiver Model

A simple source/path/receiver model is useful in noise control programs for basic understanding of the problem. Such a model is illustrated in **Figure 11**.

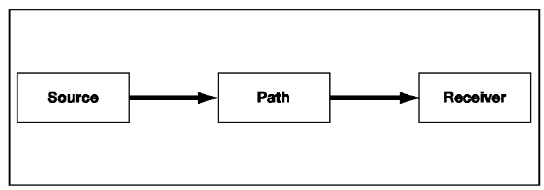

Figure 11

Following are some general observations regarding this simple model.

- Sources generally emit both airborne and structure-borne noise (the latter form of noise also commonly referred to as vibration). Each form of noise may result in undesirable airborne noise at the receiver, which is the primary concern here.

- Noise generally travels from the source to the receiver along several paths simultaneously. Every significant noise path must be treated in order to successfully reduce levels at the receiver. This is analogous to electrical switches in parallel: all switches must be open to stop current flow.
- A path may involve a series of structural or acoustic elements. Any element in the path can be controlled in order to attenuate noise along that path. This is analogous to electrical switches in series, where any open switch can stop the current flow.
- It is essential to identify the receivers and to determine what noise limits apply to them. This dictates the attenuation required along each path.
- Sources, paths, and receivers of noise all have frequency-dependent response properties. Sources such as large internal combustion engines typically emit high-amplitude, low-frequency noise, while most path attenuators (e.g., walls or silencers) are more effective at high frequencies than at low frequencies. Finally, receivers such as building structures or the human ear are more sensitive at some frequencies than at others.
- The role of the facility designer usually is to control the paths of noise, since the source generally cannot be altered and the noise restrictions at the receiver are often fixed.

Large-Engine Installation

For facility design purposes, the large-engine unit may be modeled as a number of different sub-sources, each having one or more different paths to possible receivers, as shown in **Figure 12.** This particular illustration relates to external noise, but the external environment in this problem could also be an interior space in another part of the facility. It permits the introduction of some important concepts of the noise problem.

Airborne mechanical noise (1), (4) and (5) is radiated from the engine unit; some surfaces radiate far more noise than others. Receivers of airborne mechanical noise may be located in the same room as the unit - for example, an operator, a person outside the engine room, a person elsewhere in the building or a community resident. Noise paths to receivers outside the room include walls, ceilings, tunnels, and ventilation system ducts.

The acoustic environment inside the room consists of a direct sound field, sound radiated straight from the source to the receiver, and a reverberant sound field, sound reflected from room boundaries.

The reverberant sound field is affected by the room acoustics of the space, principally, the sound absorptive properties of the room boundaries. The direct sound field, however, is independent of the room acoustics.

Large Engine Unit Modeled as Sub-Sources

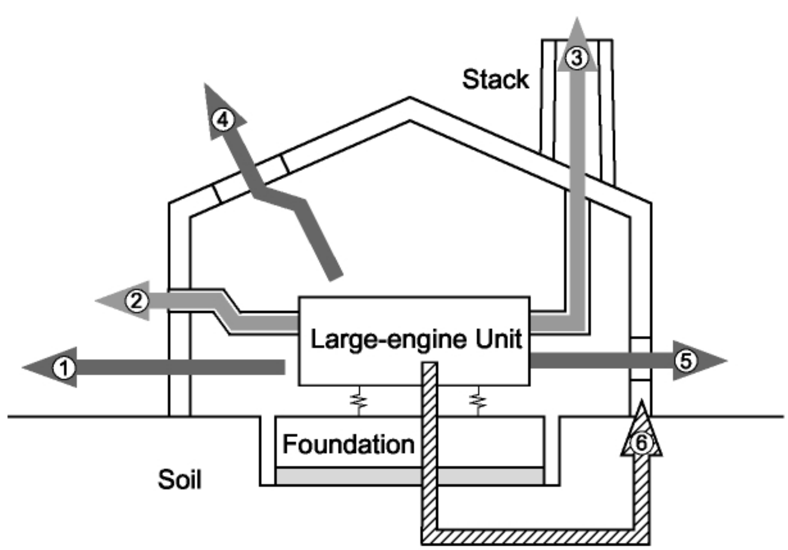

Figure 12

Inlet and exhaust noise (2) and (3) are also airborne components, and generally are significant noise sources on large internal combustion engines. In addition to the airborne noise transmitted along the inlet and exhaust piping, noise may be radiated from the piping, or from structures connected to the piping.

Structure-borne noise (6) is emitted from the source and propagates through connecting structures to airborne sound radiating surfaces. In the case of the engine unit, structure-borne noise travels first through the spring mounts to the foundation. Next, the noise energy may travel to the building structure, either directly if the foundation is supported directly on the building structure or after traversing resilient material or soil if the foundation is isolated from the building structure. At that point the structure-borne energy is radiated as airborne noise.

Site Criteria

The site criteria determine the extent of noise control measures and treatments required for a given installation. The criteria may be defined by law or by acceptability standards.

Generally, for large-engine units, the criteria addresses the comfort of exposed people. These may be in adjacent residential areas or in occupied spaces in other parts of the facility. However, criteria may also address safety issues, either for people, as in hearing protection or for equipment, as in protection from vibration.

It is imperative that the site criteria be defined before facility design is undertaken. Either under- or over-designing the facility can be unnecessarily expensive if excessive abatement equipment is specified or if construction has to be redone in order to meet site criteria. The key is to determine the requirements of the site and design toward that goal.

Because of the extensive variation in the form and the level of applicable restrictions and standards around the world, it is beyond the scope of this document to provide a detailed coverage of this area. It is the responsibility of the facility designer to identify the criteria that apply to a particular installation. The facility must meet regulations governing the installation site, and may also voluntarily meet criteria and guidelines promoting safety and comfort of the receivers. A poorly designed facility meeting local ordinances while disregarding standards of comfort and safety for residents can still be the target of community complaint.

Noise Control

A properly designed facility provides adequate attenuation for all possible paths of noise in order to meet the site noise criteria. It has been emphasized throughout this discussion that this requires attention to both airborne and structure-borne noise. Either form can result in excessive airborne noise within the facility or in the exterior environment. The following paragraphs are intended to provide some basic guidelines and cautions regarding facility design for noise control.

SECTION CONTENTS

Airborne Noise Control

Noise Criteria

Octave Bands in Cycles Per Second	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*
Highly Critical Hospital or Residential Zone	71	63	44	37	35	34	33	33	33
Night, Residential	73	69	52	44	39	38	38	38	38
Day, Residential	76	71	59	50	44	43	43	43	43
Commercial	81	75	65	58	54	50	47	44	43
Industrial-Commercial	81	77	71	64	60	58	56	55	54
Industrial	87	85	81	75	71	70	68	66	66
Ear Damage Risk	112	108	100	95	94	94	94	94	94

Airborne noise control is a straightforward and well-developed area compared with structure-borne noise control. There is abundant information available on sound absorption and transmission properties of common construction materials, and there are accepted and proven procedures for applying that information.

However, it is important to recognize that much of the conventional information and procedures were developed for higher-frequency noise, and thus may not be appropriate for engine units, which produce strong low-frequency acoustic energy. For example, structural and acoustic resonances (conditions of minimum dynamic stiffness) may coincide with pure-tone frequency components of the engine noise, resulting in very efficient transfer of energy. Conventional building acoustics generally is based on statistical descriptions of noise, and therefore does not address resonance effects.

For some installations, airborne noise must be controlled at several receiver points: inside the engine room, in other rooms in the building and outside the building. The simplest way to reduce airborne noise within a building is through good building layout. Equipment rooms should be situated far from sensitive receiver locations in the building. This takes advantage of the fact that propagating sound energy diminishes with distance from the source. In addition, there are two other methods of controlling airborne noise: with high transmission loss walls and with absorption.

It is helpful to review some terminology before discussing the sound transmission characteristics of walls. The transmission loss (TL) of a partition is a measure of the ratio of energy incident on the wall to that transmitted through the wall, expressed in dB. The less relative sound transmitted through the wall, the higher the TL of the wall. TL is a function of frequency.

The sound transmission class (STC) of a partition is a single-number rating calculated from the partition TL. A reference contour is adjusted against the measured TL data, and the STC rating equals the value of the adjusted contour at 500 Hz. The STC rating does not include information in frequency bands below 125 Hz. This rating is useful for designing walls that provide insulation against the sounds of speech and music. It is inappropriate for industrial machinery with low-frequency energy such as engine units. TL data should be used instead, whenever possible.

In typical partitions, sounds at higher frequencies are attenuated more than sounds at lower frequencies. The highest transmission loss values are found in cavity wall (two-leaf) constructions, where the two separate wall layers are well isolated. The transmission loss values increase with the masses of the individual leafs, the depth of the airspace, and the characteristics of any sound-absorptive material in the airspace.

It should be noted that noise leaks can severely degrade the performance of a partition. Materials are tested for their transmission loss characteristics in a controlled laboratory setting, with all edges sealed. But in typical construction, sound leaks may occur at the edges of the wall, at openings for pipes or electrical outlets, and across shared ceilings (so-called flanking paths). A wall with a leakage area equal to 0.01% of that of the wall area cannot exceed STC = 40, no matter how high the STC of the wall construction.

A partition may include elements with various transmission loss characteristics, for example, windows and doors. The transmission loss of the partition must be calculated taking all elements into consideration.

To estimate the total airborne noise transmission loss of a facility, subtract the noise value for each receiver from the estimated room-average sound pressure level. If there is more than one space, the sum of the individual contributions must not exceed the criterion.

Mechanical Noise

Many techniques for isolating generator set vibrations are applicable to mechanical noise isolation.

Modest noise reductions result from attention to noise sources, i.e., reducing fan speeds, coating casting areas, and ducting air flows. But for attenuation over 10 dB(A), units must be totally isolated.

One effective method utilizes concrete blocks filled with sand to house the generator set. In addition, the unit must incorporate vibration isolation techniques.

A rough guide comparing various isolation methods is illustrated in **Figure 13**.

Completely enclosed engines are impractical due to openings required for pipes, ducts, and ventilation. Enclosures with numerous openings rarely attain over 20 dB(A) attenuation.

Isolation Methods

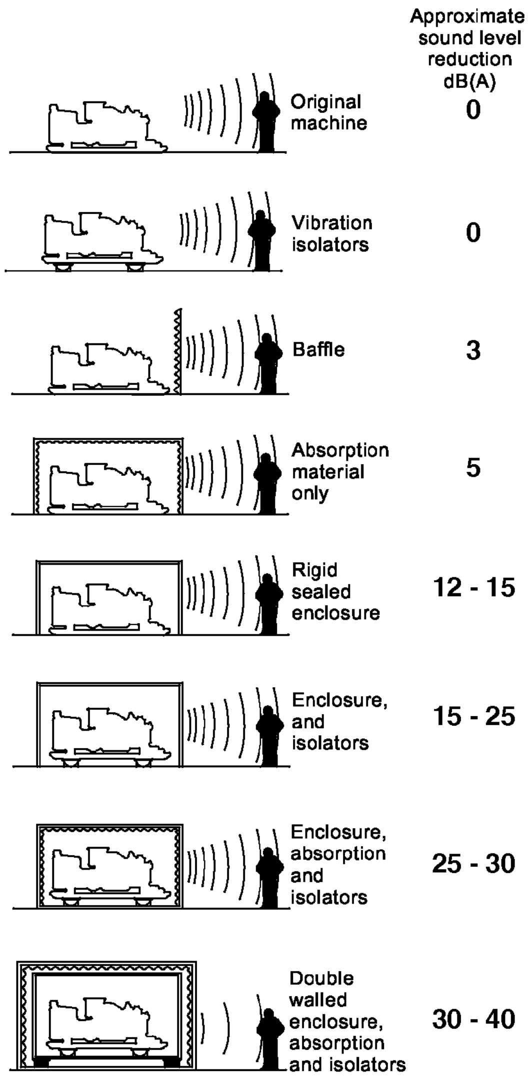

Figure 13

Intake Noise

Intake noise attenuation is achieved through either air cleaner elements or intake silencers.

Noise attenuation due to various air cleaners and silencers can be supplied by the component manufacturer.

Air intake silencers, particularly glass-filled absorbers, must be located upstream of the air cleaners to prevent glass or debris from entering the engine.

Careful consideration must be given to air intake silencers to ensure they do not excessively restrict air flow.

Exhaust Noise

Exhaust noise is typically airborne. Exhaust noise attenuation is commonly achieved with a silencer typically capable of reducing exhaust noise 15 dB(A) when measured 3.3 m (10 ft) perpendicular to the exhaust outlet. Locating the silencer near the engine minimizes transmission of sound to the exhaust piping.

Since the number of cylinders and engine speeds result in varied exhaust frequencies, specific effects of silencers must be predicted by the silencer manufacturer.

Silencers

Silencers are used to attenuate airborne noise in piping and duct systems. Their effectiveness generally is frequency sensitive, so it is essential that they be matched to the frequency content of the noise.

There are two major categories of silencers: dissipative and reactive.

Dissipative

Dissipative silencers use absorptive, fibrous material to dissipate energy as heat. They are effective only for high frequency applications, such as 500 to 8000 Hz.

Reactive

Reactive silencers, on the other hand, use a change in cross-sectional area to reflect noise back to the source. They are typically used for low-frequency applications, such as internal combustion engines, and they may incorporate perforated tubes to increase broadband performance.

The effectiveness of a reactive silencer depends on its diameter, volume, and overall design. Some specific designs include:

- Multi-chamber silencers to provide maximum sound attenuation with some flow restriction.
- Straight-through silencers provide negligible flow restriction with slightly lower sound attenuation.
- Stack silencers to be inserted directly into a stack and withstand a harsh environment.
- Combination heat-recovery silencers are designed for hot gas exhaust.

Most manufacturers offer silencer dynamic insertion loss (DIL) information in octave bands from 63 to 8000 Hz, tested in accordance with ASTM E-477. DIL is the difference in sound level with and without a silencer installed in pipe or duct with air flow. Some manufacturers rate silencers as being “industrial”, “commercial”, or “residential” grade. In such a cases, the DIL of the silencer should still be requested in order to determine the grade of silencer most suitable for the installation.

To determine the DIL required by a particular application, information is required on the actual (unsilenced) and desired noise levels at the emission point. The difference between these values is the silencer DIL. The desired source level is determined from the criteria governing the site.

When used to attenuate exhaust noise, the silencer must be sized to accommodate the specified volume of flow without imposing excessive backpressure. The flow area for a given backpressure can be calculated from the engine exhaust flow (CFM) and the exhaust temperature. The pressure drop will determine the required size of the silencer.

Sound Absorption Treatments

Acoustically absorptive surfaces convert acoustic energy into heat, and are generally described by sound absorption coefficients in octave bands. Absorptive surfaces may be used to reduce the reverberant (reflected) sound field within a room. As mentioned above, reducing the reverberant field within a room can also reduce the noise field outside the room. It should be noted that absorptive materials do not attenuate the direct sound field.

The absorption of a room may be estimated on an octave-band basis from the absorption coefficients and the area of each room surface, such as ceilings, walls and floors. Alternatively, the room absorption may be determined through reverberation time measurements. Using this information and the source sound power data, the noise reduction that can be obtained by adding absorption to a room may be determined. Information on the absorption coefficients of a material or element may be obtained from the manufacturer.

A wide variety of commercially available sound absorbing elements are available for almost every application. Ceiling treatments include lay-in tiles or boards for suspended ceilings, tiles that can be directly affixed to the ceiling surface and suspended absorbers.

Acoustic wall panels range from "architectural" panels with attractive finishes to perforated metal panels filled with absorbing materials. Concrete blocks with slotted faces and acoustical fill may be used to add sound absorption to normal concrete block wall construction.

Sound absorbing elements are selected on the basis of their sound absorption coefficient in the octave bands of interest. In addition, the elements must survive their environment, be easy to maintain, and offer acceptable flame spread properties.

Barriers and Enclosures

Barriers and enclosures block and reflect direct-radiated sound from a noise source. A barrier provides a "shadow zone" of sound attenuation between the source and the receiver, much as light casts a shadow behind a wall.

Full enclosures may be used around the source (the engine) or around the receiver (the operator or personnel in affected areas). Partial barriers may be used to protect noise sensitive areas by locating receivers in the shadow zone.

The effectiveness of a barrier in blocking noise transmitted through it is a function of its sound transmission characteristics. Both enclosures and barriers should be lined with absorptive material to be fully effective.

In the case of an enclosure without absorption, the reverberant field inside the enclosure can greatly increase the interior sound pressure, so that noise outside the enclosure is also increased.

In the case of a barrier without absorption, the noise is simply reflected elsewhere. Transmission loss and absorption are the main selection criteria for barriers and enclosures, and each is a function of frequency.

Opening in enclosures should be acoustically treated, for maximum effectiveness. Also, when using sound barriers it is important to control "flanking paths" (sound paths around the barrier).

There are many types of commercially available enclosures and barriers. Complete enclosures for specific types of mechanical equipment are available, some of which include silenced air inlets/exits and a reactive silencer for exhaust noise. Several types of modular panels are available that may include sound absorbing material on one or both sides of the panel. Outdoor barriers, designed to resist wind and seismic forces, are also available to block or reflect noise outdoors.

CAUTION: Enclosure manufacturers may publish "dB drop data." However, this may not take into account the room acoustic effects. Room acoustic effects will make the installation louder when an enclosure is placed over a free field noise source.

For example, a 120 dB(A) free field engine, if placed inside a "20 dB(A) drop" enclosure, will not be reduced to 100 dB(A). The room acoustic effect can raise the level inside to 123 dB(A) or more. This could result in 103 dB(A) or higher outside. It is important to specify actual sound levels "outside" the enclosure, not the enclosure's sound reduction capability.

Along with acoustical performance, practical issues must be considered in using barriers or enclosures. Engine enclosures require ventilation to dissipate the heat that builds up within the enclosure. The enclosure must be accessible for maintenance and inspection, and may require panic latches on doors. Acoustic materials within the enclosure must be fire-resistant.

Structure-Borne Noise Control

The purpose of a vibration isolation system (whether simple or compound), or a wave barrier, is to control the transmission of structure-borne noise from the engine unit to the building structure, either directly or through the ground.

Those measures are intended to control noise close to the source, where control measures generally are most effective. However, even with effective isolation mounting of the engine unit it still may be necessary to provide additional structure-borne noise attenuation in the building construction.

Conceptually, the simplest way to attenuate structure-borne noise along a path is to increase the distance between the source and receiver, since the amplitude of structure-borne noise decreases with increasing distance from the vibration source. The attenuation of noise in concrete-frame buildings has been found to be about 5 dB per floor for frequencies up to 1000 Hz. Attenuation for vibrations traveling along continuous concrete floor slabs typically range from 1.5 to 2 dB/meter. In general, there is less attenuation along horizontal building structures.

Another way to attenuate structure-borne noise is through structural discontinuities. A discontinuity, or impedance mismatch, causes a reflection of energy back toward the source, thereby controlling noise transmission. Such discontinuities are usually filled with a resilient material to prevent debris falling into and "shorting out" the gap. Semi-rigid fiberglass board is normally used to fill wall gaps, while asphalt-impregnated fiberglass board is normally used between on-grade slabs, foundations, and footings. Many times, large buildings already incorporate expansion joints to allow for thermal expansion and contraction. These may be used to attenuate structure-borne noise by placing the source and receivers on opposite sides of the expansion joint. It is essential that construction elements, pipes, or any other rigid connections do not bridge these discontinuities.

In addition to the source and the path, receiver locations can also be treated to control structure-borne noise in some situations. For example, a "floating floor" construction may be used to isolate the receiver (e.g., a person or some piece of vibration-sensitive equipment) from building vibration.

Foundation

Foundation Design is a very important and often overlooked aspect of large-engine unit facility design. Large-engine units, as noted above, emit relatively strong low frequency energy — structure-borne as well as airborne. If the facility design does not account for both forms of noise, it is likely that site noise criteria will not be met.

Note: Foundation design for installations where noise is not an issue is discussed in the Mounting Application and Installation Guide.

Unfortunately, structure-borne transmission and radiation is much more difficult to analyze than airborne noise. Whereas it may be relatively straightforward to estimate the airborne noise transmission loss of the building structure and various types of noise control systems, and thereby assess the adequacy of a facility design, reliable quantitative estimates of structure-borne noise transmission may be extremely difficult or impossible to obtain with current technology, particularly at low frequencies. Thus, the usual approach for noise-sensitive installations is to over-design for structure-borne noise, to ensure that it is not a problem. This means taking care to control every possible structure-borne noise path. Especially in this area, designers are strongly urged to consult qualified professional noise control engineers for noise-sensitive installations.

Engine units usually are mounted on concrete pad or metal deck foundations, using spring mounts between the unit base and the foundation. Some of the smaller engine units come with isolators between the engine/generator and base and do not require additional spring mounts for the unit base. Since the unit base provides sufficient stiffness for alignment and relative deflection of the engine and the driven equipment, there is no need to rely on the foundation for additional stiffness. Thus a foundation that is adequate for supporting the static load of the unit will be satisfactory for many installations where noise is not a critical concern.

In installations where noise is a major concern, attention must be directed toward all elements of the isolation system and to the structural paths between the foundation and the rest of the building structure. Adequate isolation often can be achieved with a simple system, but some installations may require a compound isolation system. Both types are discussed briefly in the following paragraphs.

Simple Isolation System

An isolation system with one dynamic mass and one set of isolation mounts is termed a simple isolation system. The transmissibility function for an ideal simple isolation system is shown in **Figure 14**. Transmissibility describes the ratio of force transmitted to the foundation (assumed rigid) to the force generated in the excitation source. Thus low transmissibility is desirable.

At low frequencies the transmissibility has a value of unity—that is, force is transmitted across the isolator without a change in amplitude. Around the resonant frequency of the system, the transmissibility reaches a maximum; that is, the transmitted force is substantially greater than the applied force, depending on the amount of damping in the mounts. The resonance frequency is determined by the ratio of mount stiffness to dynamic mass.

At a point above the resonance frequency, the transmissibility function drops below unity and isolation benefit begins to be realized. (The frequency at which the function crosses unity is 1.4 times the resonance frequency.) From that point the transmissibility diminishes at a rate of -2 decades per decade of frequency. Clearly, the isolation system must be designed so that the frequency of the rigid-body mode is much lower than the lowest frequency of significant structure-borne noise from the source, in order to realize isolation benefit.

Actual installations differ from the simple, ideal system in several important respects. First, the isolated mass actually has six rigid-body degrees of freedom, rather than a single one as in the system described previously. That means that an actual system has six resonance modes defining the lower-frequency range where no isolation benefit is provided.

The second important difference between actual systems and the ideal system is that the structures on either side of the isolators are not perfectly rigid. The effectiveness of the system depends on the dynamic compliance (inverse of dynamic stiffness) of the resilient elements relative to the compliance of the attached structures. Thus, an isolation system may not perform satisfactorily, even with very soft resilient elements, unless the connected structures are relatively stiff.

The principal source of dynamic weakness in structures is resonance. Resonance is a dynamic effect where the structure may be hundreds or thousands of times weaker than it is statically (i.e., at zero Hz). Therefore, isolation effectiveness will suffer around resonance frequencies of the structures on either side of the isolation elements. It is important to ensure that there are no uncontrolled resonances in the critical frequency range of the isolation system.

Figure 15 shows a measured transmissibility function on an actual simple mounting system. Several of the six rigid-body modes are evident below 3 Hz. In the frequency range above those modes, the transmissibility function falls off with a -2 decade per decade slope, as with the ideal system. However, note the peaks in the function starting at around 20 Hz. Those peaks are associated with structural resonances in one or both of the structures on either side of the isolation elements.

Compound Isolation System

Increased structure-borne noise isolation can be realized with a compound isolation system. Such a system has an intermediate mass with an additional set of isolation mounts, thus doubling the number of rigid-body modes of the isolation system. Above the rigid-body mode range, the transmissibility function has a -4 decade per decade slope—

twice the slope of the simple system. This is illustrated in **Figure 16** for an ideal system with only vertical translation response (the response of the simple model is also plotted, with a grey line, for comparison).

Note that there are two rigid-body modes in this function, one at about 3.5 Hz and one at about 7 Hz. Above the second mode the greater negative slope in the function is clearly evident.

An actual compound isolation system would have a total of 12 rigid-body modes. An actual system also would have reduced isolation effectiveness associated with resonances in the structural elements on either side of the isolation elements, just as in a simple isolation system.

A common question from facility designers is, "How large should the foundation mass be to prevent excessive structure-borne noise?" For a simple isolation system, where the foundation is rigidly supported on the building structure or embedded in the ground, the above discussion indicates that the mass of the foundation is not in itself important. Rather, it is the dynamic stiffness of the foundation that affects the isolation effectiveness.

Another critical factor is the nature of the structural connection between the foundation and the rest of the building structure. For example, with a soil-embedded foundation, where there is direct ground contact all around the foundation, substantial structure-borne noise transmission may occur from the foundation to the soil and from the soil into the building structure, regardless of the mass of the foundation. This is because wave motion occurs in any elastic medium, and wave motion will be transmitted into any other elastic medium in contact with the first. Thus, for a simple isolation system, attention must be focused on foundation stiffness as well as the interface between the foundation and its supporting ground or structure.

Another concern with any type of isolation system is to ensure that there are no rigid structural connections across the isolation system, which can in effect short-circuit the isolation system.

Comparison of Isolation Systems

To illustrate different isolation systems, consider the case of a G3600 generator set package. The total weight of the package is 113,000 lb (51,300 kg). The standard mount system for the package consists of 8 spring mounts, each with a spring rate of 34,640 lb/in (6.08e6 N/m) and a capacity of 22,900 lb (10,400 kg). This standard system has a fundamental vertical bounce mode frequency of 4.9 Hz (for simplicity, ignore the other five rigid-body modes of the system). The transmissibility function for this system is shown in **Figure 14**. The function has a slope of -2 decades per decade of frequency above the resonance frequency.

Now, consider a modified simple isolation system where the standard spring mounts are replaced with air mounts with a total effective spring rate of 26,000 lb/in (4.56e6 N/m). This modified system has a fundamental vertical bounce mode frequency of 1.5 Hz, as illustrated in the transmissibility function in **Figure 18**, which also shows the transmissibility function for the standard system for comparison. Note that above 4 Hz the air-mount system transmissibility is an order of magnitude lower than that of the standard spring-mount system.

Finally, consider a compound isolation system having an intermediate block with a mass equal to that of the generator set package, with 8 standard spring mounts between the generator set base and the block and 10 standard spring mounts between the block and ground. The total weight on the lower mounts is slightly below the total rated capacity of those mounts. This compound system has two vertical bounce modes, one at 3.1 Hz (where the generator set and the block vibrate in phase) and one at 8.6 Hz (where the generator set and the block vibrate out of phase).

The transmissibility function for the compound system is plotted in **Figure 19**, along with the functions for the two simple systems for comparison.

Above the higher mode, the transmissibility function has a slope of -4 decades per decade of frequency - twice as great as that of a simple system. Thus, above 10 Hz the transmissibility of the compound system is much better than that of the standard system, and the difference continues to increase with frequency. Similarly, the performance of the compound system is better than that of the simple air-mount system above 20 Hz. However, below 10 Hz the transmissibility function for the compound system is substantially worse than that of either of the simple systems.

Note that while the standard simple spring-mount system is quite stable by itself, some additional isolation elements might be required for adequate lateral stability in either the simple air-mount system or the compound system.

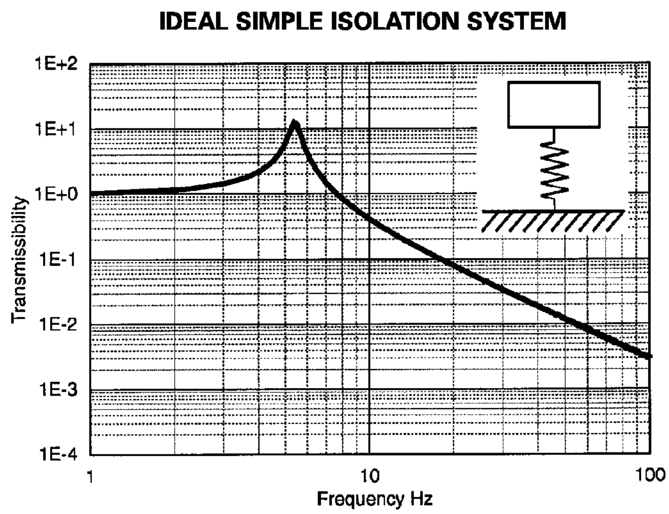

Figure 14

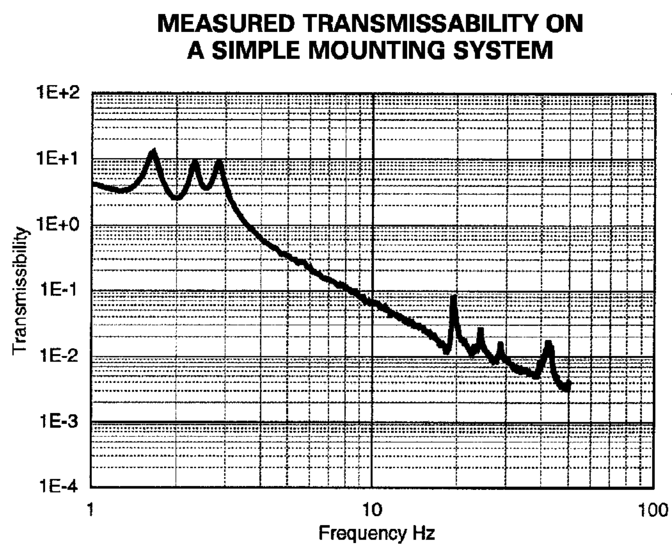

Figure 15

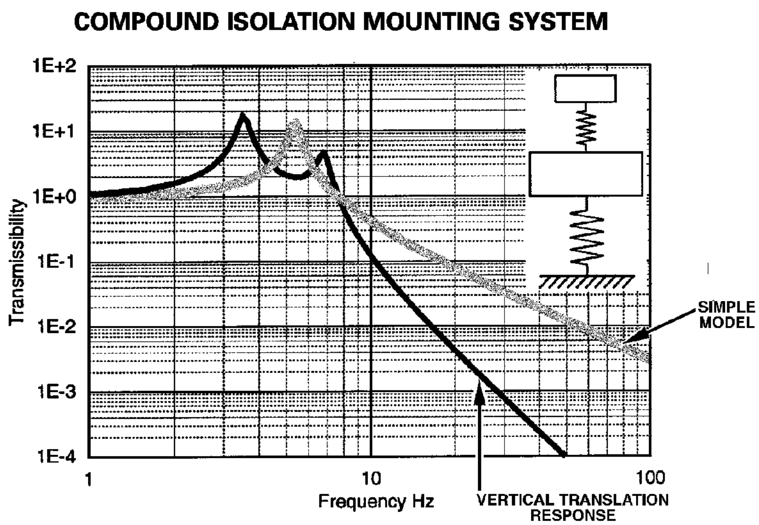

Figure 16

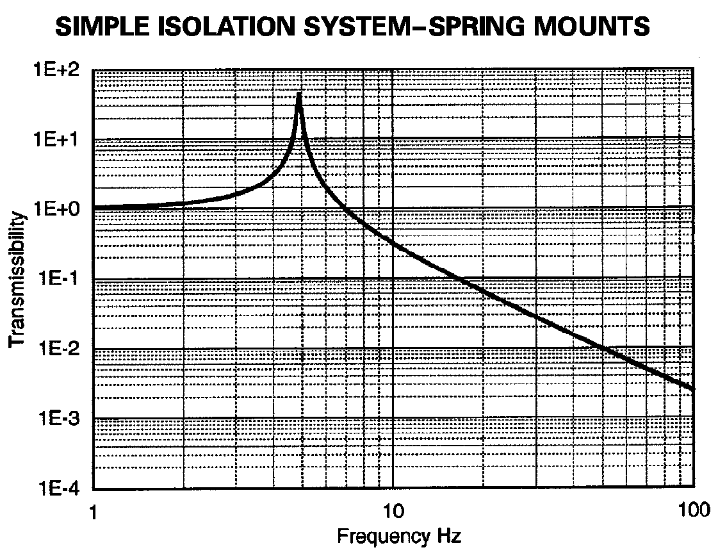

Figure 17

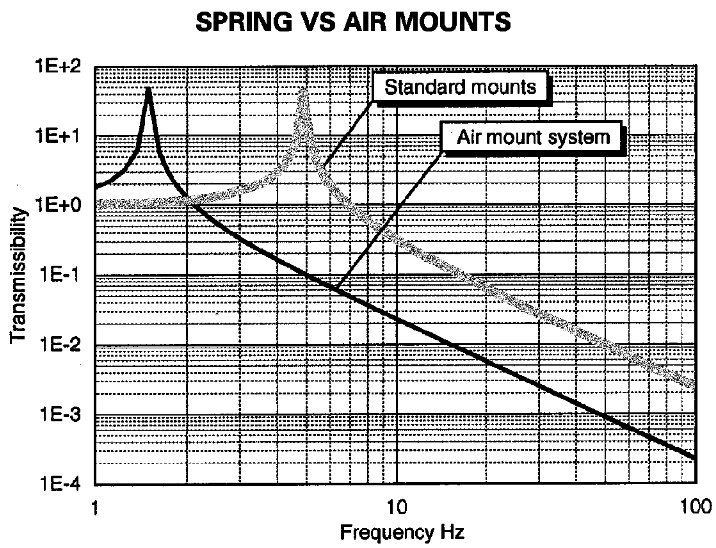

Figure 18

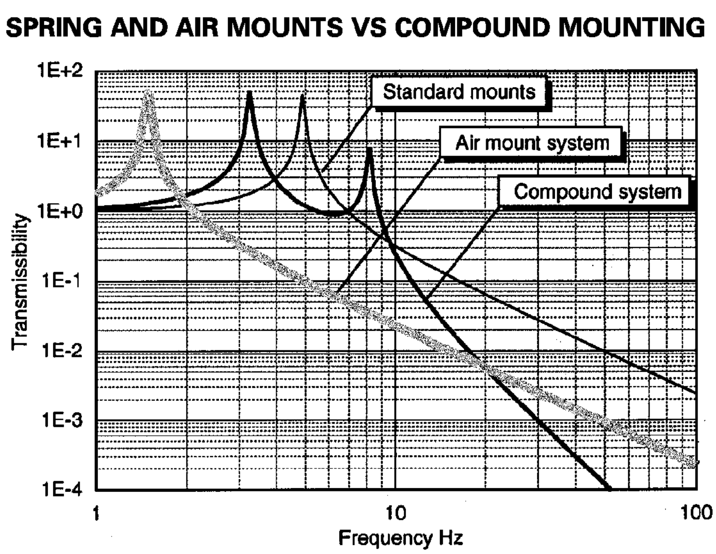

Figure 19