MINISTRY OF HUMAN RESOURCES
DEPARTMENT OF OCCUPATIONAL SAFETY AND HEALTH

GUIDANCE ON OCCUPATIONAL NOISE CONTROL

2024

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PREFACE

These guidance may be cited as Guidance on Occupational Noise Control 2024 and replaces the Guidelines for Control of Occupational Noise 2005.

The purpose of the guidance is to provide basic guidance on the implementation of engineering control pertaining to noise emission generated in the workplace. It is employer's responsibility to develop a noise control plan and action program in the workplace as outlined in the Occupational Safety and Health (Noise Exposure) Regulation 2019 and Industrial Code of Practice for Management of Occupational Noise Exposure and Hearing Conservation 2019.

Engineering control at sources, path and receiver as a significant means to reduce the excessive noise in the workplace is to be emphasized in these guidance instead of administrative and personal hearing protection.

It is hoped that all employers and hearing conservation administrators will refer to these guidance in determining an appropriate and effective engineering control so that sustainable noise reduction can be achieved. Thus, the occurrence of Occupational Noise Related Hearing Disorder (ONRHD) cases can be prevented.

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2024

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These guidance have been approved in the Department's Policy Review Meeting chaired by the Director General of the Department of Occupational Safety and Health Malaysia.

1.0 INTRODUCTION

These Guidance have been drawn up to cater to the noise control needs of all workplaces falling within the purview of the Occupational Safety and Health Act 1994 (Amendment) 2022 Act A1648. These Guidance may help employers and hearing conservation administrator to implement effective control measures in order to prevent occurrences of occupational noise related hearing disorder in all workplaces through the application of engineering noise control principles. In addition, these guidance will assist noise risk assessor in providing recommendations, so far as is practicable, noise control measures at a place of work.

2.0 OBJECTIVE

These Guidance are primarily intended to be used by employers, noise risk assessor, designers, manufacturers, importers and suppliers to comply with the Regulation 6 of Occupational Safety and Health (Noise Exposure) Regulations 2019 and Industry Code of Practice (ICOP) for Management of Occupational Noise Exposure and Hearing Conservation 2019. The objectives of these guidance are to:

a) Explain the conceptual understanding of occupational noise control; and
b) Provide practical guidance in developing and implementing engineering noise control measures at the workplace.

3.0 CONCEPT AND BASIC PRINCIPLE

3.1 Introduction To Sound

Sound is a science of wave motion. Waves change or disturbance in some physical property of a medium is transmitted through a medium. The essential features of a medium which can transmit sound waves must possess elasticity and inertia (mass). Sound waves can travel through solids, liquids, and gasses but not through vacuum medium.

Technically, the sensation of sound results from oscillations in pressure, stress, particle displacement and particle velocity in any elastic medium that connects the sound source with the ear. When sound is transmit through air, it is usually describe in terms of changes in pressure that alternate between above and below atmospheric pressure. These pressure changes are produced when a vibrating object (i.e. sound source) causes alternating regions of high pressure (compression) and low pressure (rarefaction) that propagate from the sound source.

The sound characteristics of a particular sound depend on the rate at which the sound source vibrates, the amplitude of the vibration, and the characteristics of the conducting medium. A sound may have a single rate of compression-rarefaction alternation (frequency), but most sounds have many frequency components. Each of these frequency components or bands of sound may have a different amplitude.

3.1.1 Frequency of Sound

Frequency is defined as the rate at which the waves of a sound are emitted from its source. Physically, it is measure by the number of times per second that the pressure oscillates between levels above and below atmospheric pressure **(Figure 3.1)**. Frequency is denote by the symbol f and is measure in Hz (Hertz), where 1 Hz = 1 cycle / second. Frequency is inversely relate to the period (T), which is the time (in second) a sound wave requires to complete one cycle. The range of sound frequency over which normal young human adults are capable of hearing sound at moderate levels is 20 to 20,000 Hz. 'Pitch' or 'tone' is the sensation closely associated with frequency.

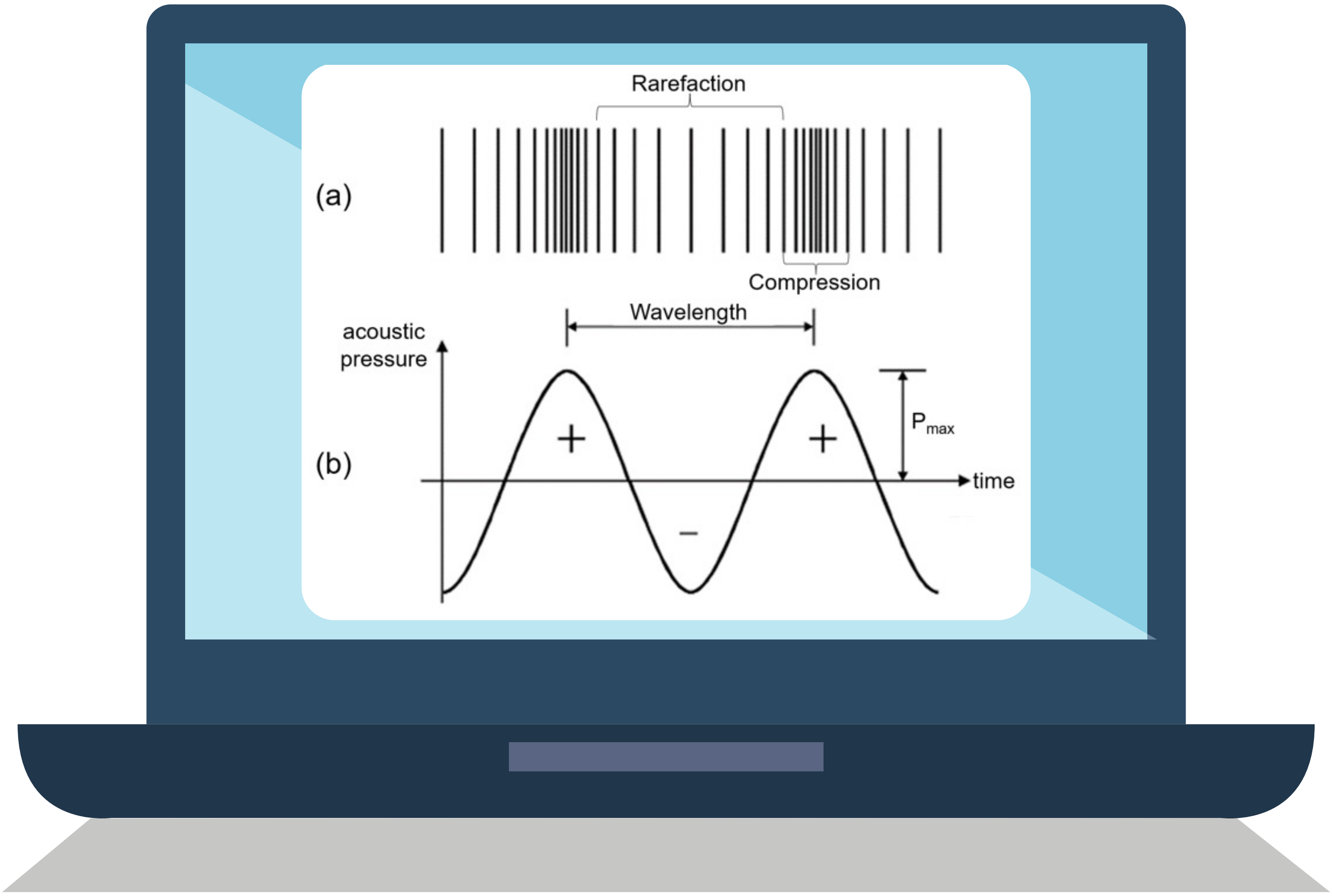


Figure 3.1: Representation of a sound wave, (a) Sound wave compression and rarefactions in air; (b) graphic representation of pressure variations above and below atmospheric pressure.

A sound may consist of a single frequency (i.e. a pure tone). However, most common sounds comprise many frequency components. It is generally impossible to report the characteristics of all the frequencies emit from noise sources. Therefore, measurements are made that include the sound energy from a broad range of frequencies. The sound wavelength is inversely proportional to the sound frequency **(Figure 3.2)**. High frequency defines a small wavelength, whereas low frequency defines a huge wavelength.

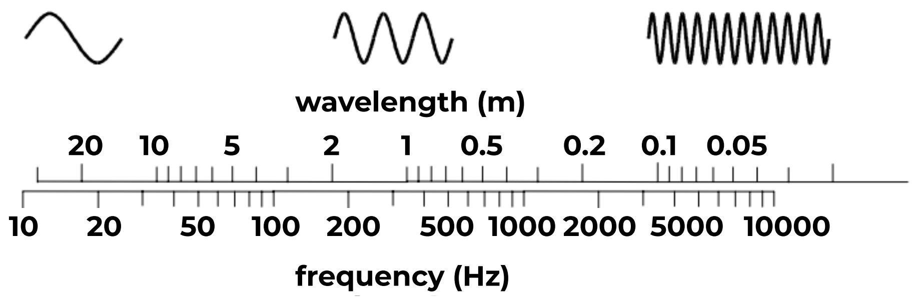


Figure 3.2: Air wavelength versus frequency.

A pure sine wave is produced if the sound wave moves with simple harmonic motion. Such propagation is characterized by a single frequency. The motion and corresponding spectrum are illustrated in **Figure 3.3 (a)** and **Figure 3.3 (b)**.

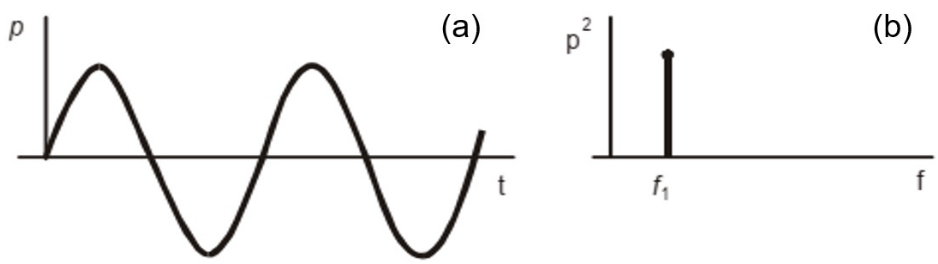


Figure 3.3: (a) Pressure wave, *p* varies sinusoidal with time; **(b)** *t* at a single frequency f_1.

If the sound wave moves irregularly but cyclically, for example, it produces the waveform shown in **Figure 3.4 (a)**, the resulting sound field will consist of a combination of sinusoids of several frequencies. The spectral (frequency) distribution of the energy in this particular sound wave is represent by the frequency spectrum of **Figure 3.4 (b)**. As the wave is cyclic, the spectrum consists of a set of discrete frequencies.

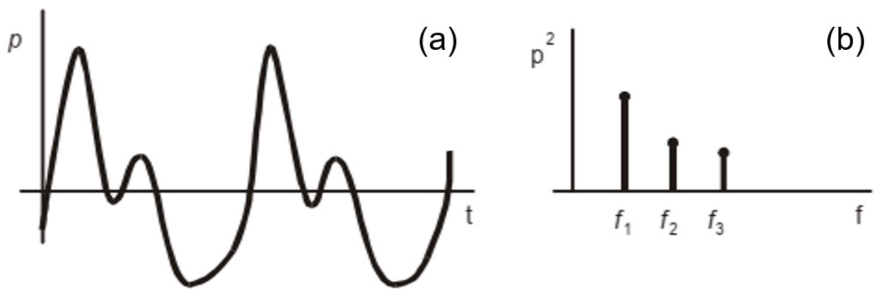


Figure 3.4: (a) Pressure wave, *p* varies cyclically with time; **(b)** t as a combination of three sinusoidal wave of fixed relative amplitudes and phases; the associated spectrum has three single-frequency components f_1, f_2 and f_3.

Although some sound sources have single frequency components, most sound sources produce a very disorder and random waveform of pressure versus time, as illustrated in **Figure 3.5 (a)**. Such a wave has no periodic component, but Fourier analysis may be shown that the resulting waveform may be represent as a collection of waves of all frequencies. For a random type of sound pressure wave, squared in a band of frequencies are plotted as shown in the frequency spectrum of **Figure 3.5 (b)**.

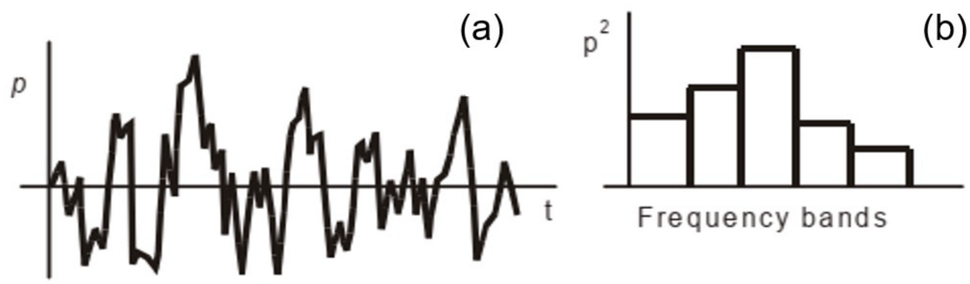


Figure 3.5: (a) Pressure wave, *p* varies erratically with time; **(b)** *t* with a frequency band spectrum.

Frequency weighting networks have been standardizing for single number assessments of sounds having properties similar to the response of the human ear. In addition to these broadband weightings, the frequency ranges in acoustics are frequently divided into smaller ranges. The standard range of frequencies is the one-one octave band, where the upper edge of the band is twice the frequency of the lower edge **Figure 3.6 (a-c)**. One-third octave bands (i.e. where three bands are used to measure 1 octave) and narrow bands are used.

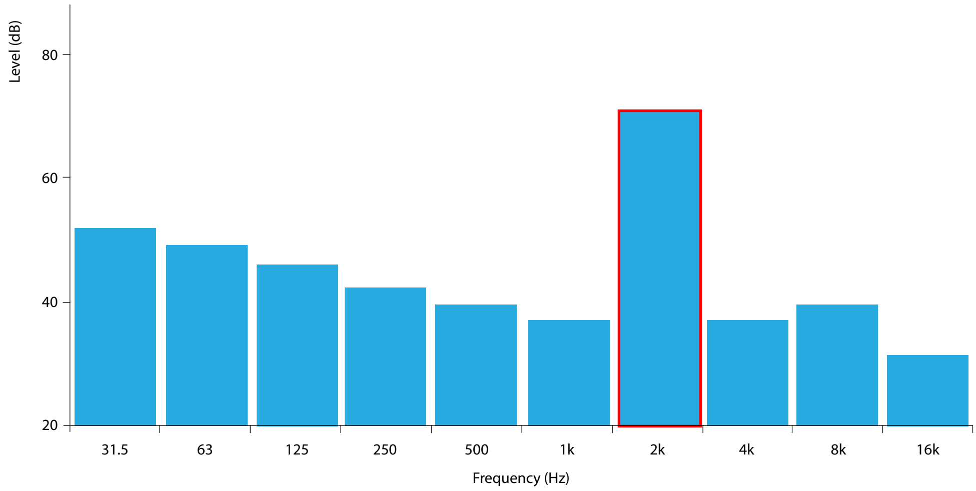


Figure 3.6 (a): An example of a 1/1 octave band starts at 31.5 Hz and goes to 16 kHz with a different noise level at each octave band. The 2 kHz octave band indicates the dominant noise level amplitude.

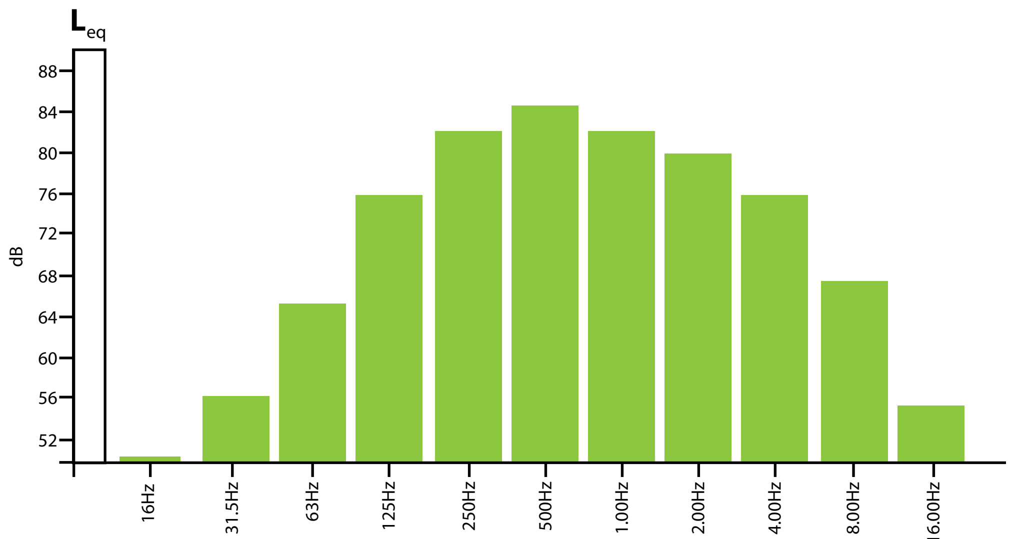


Figure 3.6 (b): An example of a 1/1 octave band of diesel generator.

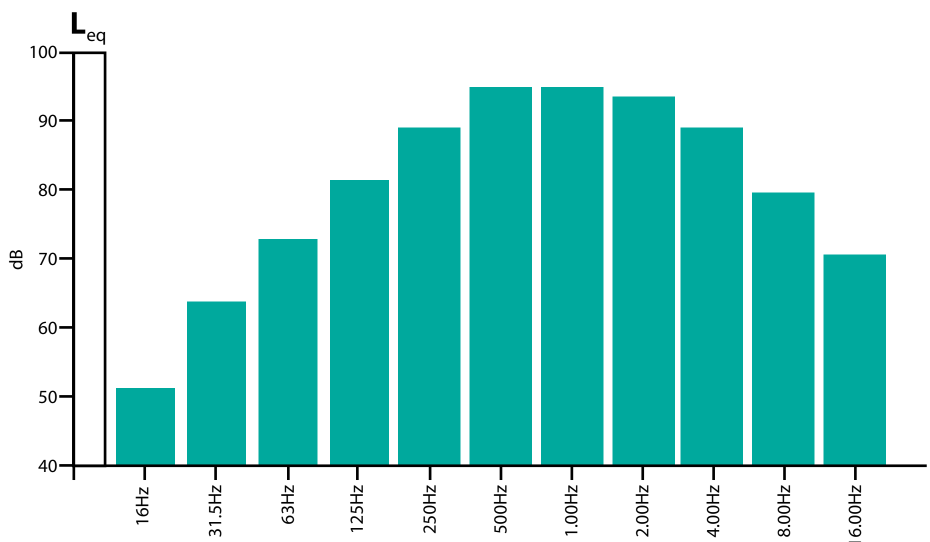


Figure 3.6 (c): An example of a 1/1 octave band air cooler.

3.1.2 Wavelength of Sound

The distance of sound wave travels during one sound pressure cycle is called the wavelength (λ). The wavelength is relate to the speed of sound (c) and its frequency (f) by:

$$Wavelength, \lambda = \frac{c}{f}$$

where;

λ : the wavelength in meter (m),
c : the sound speed in meters per second (m/s)
f : the frequency in Hertz (Hz)

For most applications in noise control for hearing conservation, it is sufficient to note that sound travels in the air at an approximate 344 m/s at normal atmospheric pressure and room temperatures.

Example 1:

Calculate the wavelengths of sound waves of frequencies 20 Hz, 1,000 Hz and 20,000 Hz.

Answer:

1. $20\ Hz, \lambda = \frac{c}{f} = \frac{344\ m/s}{20/s} = 17.2\ m$
2. $1{,}000\ Hz, \lambda = \frac{c}{f} = \frac{344\ m/s}{1{,}000/s} = 0.344\ m$
3. $20{,}000\ Hz, \lambda = \frac{c}{f} = \frac{344\ m/s}{20{,}000/s} = 0.017\ m$

When considering the behavior of sound, one important characteristic is the ratio of the length of the item affected by the noise to the wavelength of the sound. The effectiveness of a barrier in shielding one side from a noise source on the other depends on the path differences between going around the barrier and the straight-line distance in terms of the number of wavelengths.

3.1.3 Sound Pressure And Sound Pressure Level

When a sound source vibrates, it causes very small variations in air pressure around (i.e. above and below) the ambient atmospheric pressure. It is these variations in air pressure that are referred to as the 'sound pressure'. Atmospheric pressure (i.e. 1 atmosphere) = 1.013 x 10^5 pascals (Pa) or 1.013 x 10^5 N/m^2. In direct comparison, the magnitude of sound pressure relative to air pressure is very small indeed that is so small as to be practically negligible. For instance, the sound pressure resulting from normal speech at 1 meter distance from a talker would average about 0.1 Pa, i.e. 1 millionth of an atmosphere.

The range of sound pressure relevant to the human ear and therefore commonly measured is very wide. Sound pressure well above the ear pain threshold is about 20 Pa can be found in some working environments. Meanwhile, sound pressure down to the threshold of hearing is about 20 µPa (micropascals). The range of common sound pressure exposures therefore exceeds 10^6 Pa. This range cannot be scaled linearly with a practical instrument while maintaining the desired accuracy at the low and high ends. This is due to being able to show "just noticeable differences" in hearing sound pressure in noisy environments would require a long scale.

Therefore, in order to accommodate this wide range of sound pressure units, a logarithmic number of scale divisions that respond more closely to the response of the human ear is used. The abbreviation of dB is used with upper and lower case whereby "d" for deci and "B" for Bell in honor of Alexander Graham Bell. This measurement unit was first conceived at Bell Laboratories (USA) to facilitate calculations involving the loss of signals in long lengths of telephone lines.

By definition, the decibel is a unit without dimensions. It is the logarithm to the base 10 of the ratio of a measured quantity to a reference quantity when the quantities are proportional to power. The decibel is sometimes difficult to use and to understand because it is often used with different reference quantities. Acoustic intensity, acoustic power, hearing thresholds, electrical voltage, electrical current, electrical power and sound power level may all be expressed in decibels (each having a different reference). Obviously, the decibel has no meaning unless a specific reference quantity is specified or understood. Any time the term 'level' is referred to in acoustics, decibel notation is implied.

The majority of sound-measuring instruments are calibrated to provide a reading known as the root mean square or RMS of sound pressures on a logarithmic scale in decibels. The decibel reading taken from such an instrument is called the sound pressure level (Lp). The term 'level' is used because the measured pressure is at a particular level above a given pressure reference.

For sound measurements in air, 0.00002 Pa commonly serves as the reference sound pressure. This reference is an arbitrary pressure chosen many years ago because it is approximately the normal threshold of young human hearing at 1,000 Hz. Since the eardrum responds proportionally to the intensity (i.e. energy per unit time per unit area) of the sound wave, and sound intensity is proportional to sound pressure squared, sound pressure level is calculated from the square of sound pressure. Mathematically, sound pressure level, ***Lp*** is written as follows:

$$L_P = 10 \log \left(\frac{P}{P_r}\right)^2$$

$$= 20 \log \left(\frac{P}{P_r}\right)$$

where;

P : the measured actual sound pressure
Pr : is the reference sound pressure

In using this equation, *P* and *Pr* must be in the same units.

The use of this form to specify sound pressure levels obscures the fact that levels are ratios of quantities equivalent to power. For technical purposes, ***Lp*** should always be written in terms of decibels relative to the recommended reference sound pressure of 20 μPa.

Figure 3.7 shows the relationship between sound pressure (in pascals) and sound pressure level (in dB re 20 μPa). This diagram illustrates the advantage of using the decibel scale rather than the wide range of direct pressure measurements. It is of interest to note that any doubling of sound pressure is equivalent to a 6 dB change in sound pressure level. For example, a range of 20 to 40 μPa or a range of 1 to 2 Pa which might be found in hearing measurements, are both ranges of 6 dB. Measuring in decibels allows reasonable accuracy for both low and high sound pressure levels.

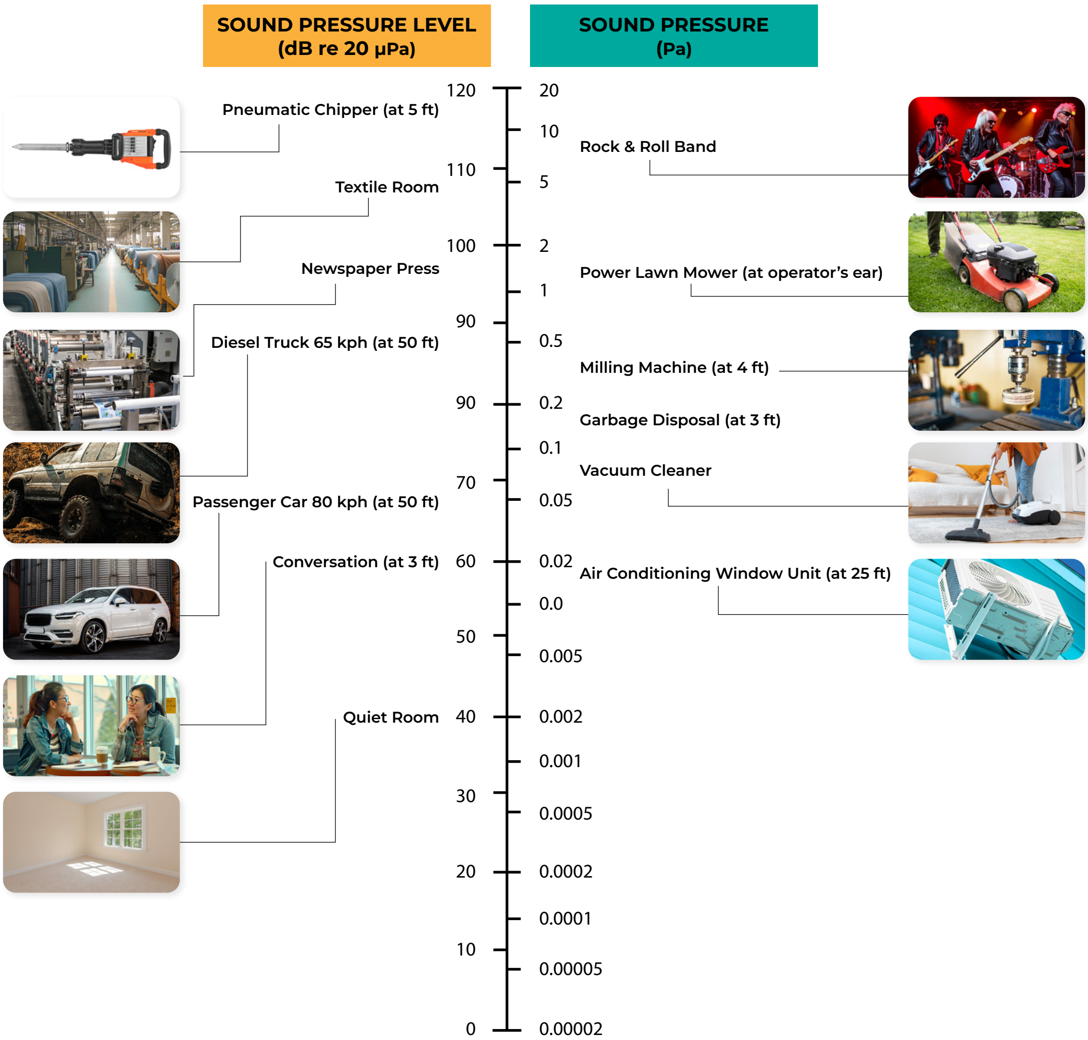


Figure 3.7: Relationship between sound pressure (in pascals) and sound pressure level (in dB re 20 μPa).

3.1.4 Sound Intensity And Sound Intensity Level

Sound intensity at any specified location may be defined as the average acoustic energy per unit time passing through a unit area that is normal to the direction of propagation. For a spherical or free-progressive sound wave, the sound intensity may be express by:

$$Sound\ Intensity, I = \frac{p^2}{\rho c}$$

where;

p : RMS sound pressure
ρ : density of the medium
c : speed of sound in the medium.

Sound intensity units cover a wide range, and it is often desirable to use decibel levels to compress the measuring scale. To be consistent, sound intensity level, L_{Int} is define as

$$Sound\ Intensity\ Level, L_I = 10\ log\left(\frac{I}{I_r}\right)$$

where;

I : the measure actual sound intensity at some given distance from tahe source
Ir : a reference sound intensity, 10^{-12} Watt/m²

In air, this reference sound intensity closely corresponds to the reference sound pressure of 20 μ Pa used for sound pressure level. Therefore, L_I equal to L_p .

3.1.5 Sound Power And Sound Power Level

Consider a vibrating object suspended in the air as shown in **Figure 3.8**. The vibrations will create sound pressure waves that travel away from the source which decreasing by 6 dB as the distance from the source doubles. The sound power of this source is independent of its environment, but the sound pressure around the source is not.

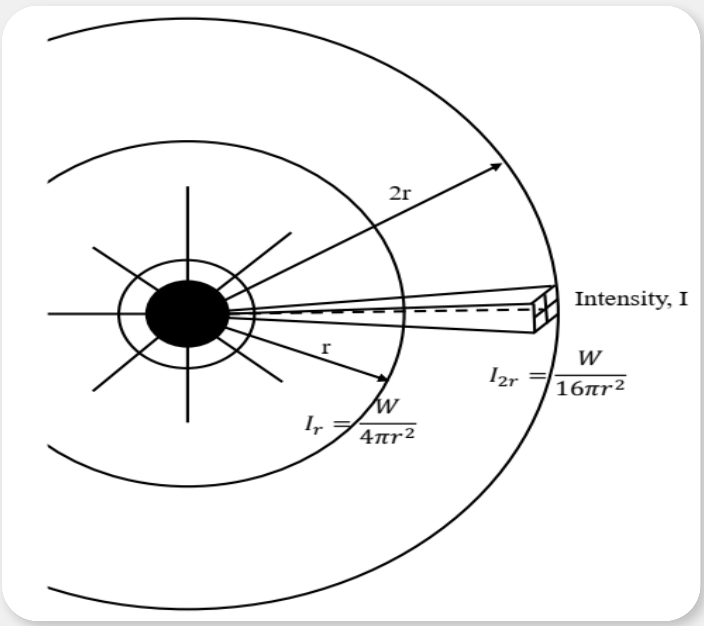


Figure 3.8: Pulsating object suspended in air (i.e. free field).

Sound power represented by *W* is use to describe the sound source in terms of the amount of acoustic energy that is produced per unit time (Watts). Sound power is relate to the average sound intensity produced in free-field conditions at a distance, *r* from a point source by:

$$Sound\ Power, W = I_{avg} 4\pi r^2$$

where;

Iavg : the average sound intensity at a distance r from a sound source whose sound power is W.

The quantity $4\pi r^2$ is the area of a sphere surrounding the source over which the sound intensity is average. The sound intensity decreases with the square of the distance from the source, hence the well-known inverse square law.

Units for sound power are also usually given in terms of decibel levels because of the wide range of power covered in practical applications. For consistency, sound power level, L_W is define as

$$Sound\ Power\ Level, L_W = 10 \log\left(\frac{W}{W_r}\right)$$

Choosing a reference surface area, S_r of 1 m² derives a convenient expression for sound power level, L_W as follows:

$$\frac{W}{W_r} = \frac{I_{avg} 4\pi r^2}{I_r S_r} = \frac{I_{avg}}{I_r} \times \frac{4\pi r^2}{S_r}$$

$$10 \log\left(\frac{W}{W_r}\right) = 10 \log\left(\frac{I_{avg}}{I_r}\right) + 10 \log\left(\frac{4\pi r^2}{S_r}\right)$$

$$L_W = L_I + 10 \log \frac{4\pi r^2}{S_r}$$

where;

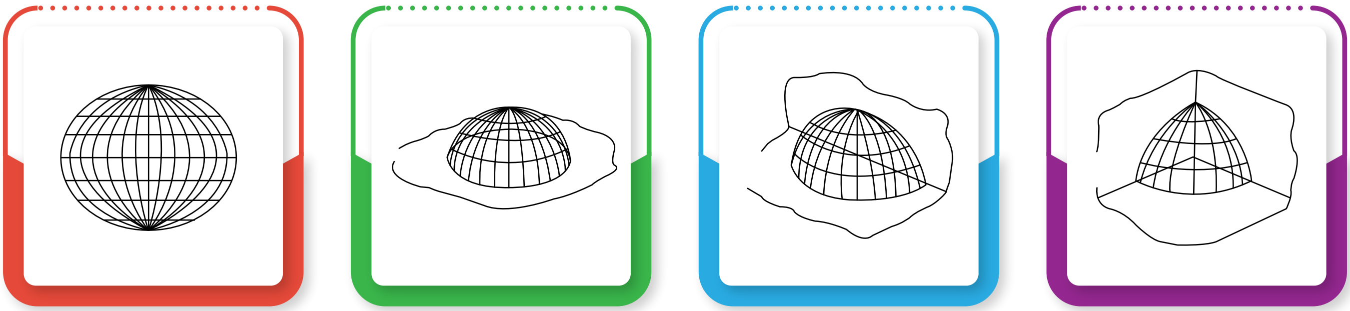
W : the sound power of the source in Watts (1 Watt = 1 N m/s)
W_r: is the reference sound power of 10^{-12} Watt.

Figure 3.9 shows the relation between sound power in Watts and sound power level in dB reference 10^{-12} Watt. Note that the distance must be specified or inferred to determine sound pressure level from the sound power. Equation above is used to predict sound pressure levels if the sound power level of a source is known, and the acoustic environment is known or can be estimated.

Figure 3.9: Relation between sound power levels and sound power.

In a free field (no surfaces to reflect sound waves), sound spreads out from the source, losing power as the square of the distance. Normally, reflecting surfaces are present or the sound source is not omnidirectional. A reflecting surface will increase the sound intensity since the volume in which the sound radiates is reduced.

The directivity factor (Q) is a dimensionless quantity used to describe the ratio of volume to which a sound is emitted relative to the volume of a sphere with the same radius shown in **Figure 3.10.**



Q = 1 (Spherical Radiation)
Q = 2 (1/2 Spherical Radiation)
Q = 4 (Radiation from floor and wall)
Q = 8 (Radiation into 1/8 sphere)

Figure 3.10: Directivity factor (Q)

Location of sound source will contribute to different level of sound pressure level as shown in **Table 3.1.**

Table 3.1: Relationship between directivity factor (Q) and sound pressure level (L_p).

Source Position	Q	L_p at 1 meter
In Air	1	X dBA
On Hard Floor	2	X + 3 dBA
Near Hard Wall	3	X + 6 dBA
At Corner	4	X + 9 dBA

In addition, applying the inverse square law to predict the sound pressure level, L_P at the different distances, *r* from a point source of sound power, *W* (or sound power level, L_W) in a free field without considering in air directivity factor, Q = 1 i.e. no reflecting surfaces using Equation below:

$$Sound\ Pressure\ Level, L_P = L_W - 20 \log r - 11$$

Example 2:

Measurements are made of the sound pressure levels at positions around an electrical motor in an anechoic room (a special test room with completely sound absorbing surfaces, which provide free field test conditions). Six measurement positions are selected, each at 2m from the motor: four in the horizontal plane of the motor (positioned in the center of the room) at the compass points (N, S, E and W) and directly above and below the motor (A and B).

The sound pressure levels (all in the same octave band) are: N 80 dB, S 78 dB, E 76 dB, W 82 dB, A 74 dB and B 84 dB. Calculate the sound power level of the machine.

Answer:

First it is necessary to calculate the logarithmic average of the six pressure levels:

$$L_{eq} = 10 \log \left[\frac{(10^{7.4} + 10^{7.6} + 10^{7.8} + 10^{8.0} + 10^{8.2} + 10^{8.4})}{6} \right] = 80.3\ dB$$

The sound power level may be found using the average sound pressure at 2m and the inverse square law formula:

$$80.3 = L_W - 20 \log (2) - 11$$
$$L_W = 97.3\ dB$$

3.1.6 The Relationship Between Sound Power, Sound Intensity And Sound Pressure

The resolution of noise control problems requires a practical knowledge of the relationship between sound pressure, sound intensity and sound power. For example, consider the prediction of sound pressure levels that would be produced around a proposed machine location from the sound power level provided by the machine.

Example 2:

The manufacturer of certain machine states that this machine has an acoustic power output of 1 Watt. The machine is expected to be mounted on the floor of a factory workroom. Predict the sound pressure level at a location 10 m from the machine. Density of air medium is 1.18 kg/m³ and speed of sound in air medium is 344 m/s.

Answer:

For an omnidirectional source in a free field, putting $I = I_{avg}$ gives the sound pressure as:

$$p = \left(I_{avg}\rho c\right)^{\frac{1}{2}} = \left(\frac{W\rho c}{4\pi r^2}\right)^{\frac{1}{2}}$$

$$p = \left(\frac{(1\ N\ m/s)\times(1.18\ kg/m^3)\times(344\ m/s)}{4\pi\times(10\ m)^2}\right)^{\frac{1}{2}} = 0.57\ N/m^2$$

However, since the source will actually be sitting on the floor (Q = 2), reflections will double the sound pressure giving an actual sound pressure of 1.14 N/m². The estimated sound pressure level, will thus be

$$L_P = 10\log\left(\frac{1.14}{0.00002}\right)^2 = 95.1\ dB$$

Admittedly, there are few truly free-field situations and few omnidirectional sources.

So, the above calculation can only give rough estimate of the absolute value of the sound pressure level. However, comparison or rank ordering of different machines can be made and at least a rough estimate of the sound pressure level is available. Noise levels in locations that are reverberant or where there are many reflecting surfaces can be expected to be higher than that predicted because noise is reflected back to the point of measurement.

3.1.7 Frequency Weighting

The human ear is not equally sensitive to sound at different frequency. Therefore, the sound measuring equipment shall consider this difference in sensitivities over the audible range. For this purpose, frequency weighting networks by means frequency weighting filters have been introduced. The sound pressure levels are reduced or increased as a function of frequency before being added together to give an overall sound level. The two standardized weighting network is common use are the “A” and “C”, which have been proposed to correlate to the frequency response of human ear for different sound levels as per specified in IEC 60651. **Table 3.2** shown the attenuation provided by the A and C weighting.

Table 3.2: Frequency weighting characteristic for A and C networks.

Frequency Hz	Weighting, dB	
	A	C
31.5	-39	-3
63	-26	-1
125	-19	0
250	-9	0
500	-3	0
1,000	0	0
2,000	1	0
4,000	1	-1
8,000	-1	-3

When frequency weighting networks are used, the measured noise levels are indicated specifically by dB(A) or dB(C). If the noise level is measured without a “frequency-weighting” network, then the sound levels corresponding to all frequencies contribute to the total noise energy occur. This physical measurement without modification is not particularly useful for noise risk assessment and is refer to as the linear or unweighted sound pressure level.

The overall sound pressure level is obtained by adding the individual unweighted octave band sound pressure levels given in **Table 3.3**, using decibel addition equation as below:

Table 3.3: Sound level meter data example

	Octave band center frequency, Hz								
	31.5	**63**	**125**	**250**	**500**	**1,000**	**2,000**	**4,000**	**8,000**
L_P , dB	70	81	89	101	103	93	83	77	74
A weighted	-39	-26	-16	-9	-3	0	1	1	-1
L_{PA} , dB(A)	31	55	73	92	100	93	84	78	73

Therefore;

$$L_{PA} = 10 \log \left[\sum_{i-2}^{n} 10^{\frac{L_{PA}}{10}} \right]$$

$$L_{PA} = 10 \log [10^{\frac{31}{10}} + 10^{\frac{55}{10}} + 10^{\frac{73}{10}} + 10^{\frac{92}{10}} + 10^{\frac{100}{10}} + 10^{\frac{93}{10}} + 10^{\frac{84}{10}} + 10^{\frac{78}{10}} + 10^{\frac{73}{10}}]$$

$$L_{PA} = 101.4\ dB(A)$$

3.2 Noise Control – Concept And Basic Principle

As in all hazard control, noise control efforts should be approached according to the hierarchy of control strategies using the triangle paradigm in **Figure 3.11.**

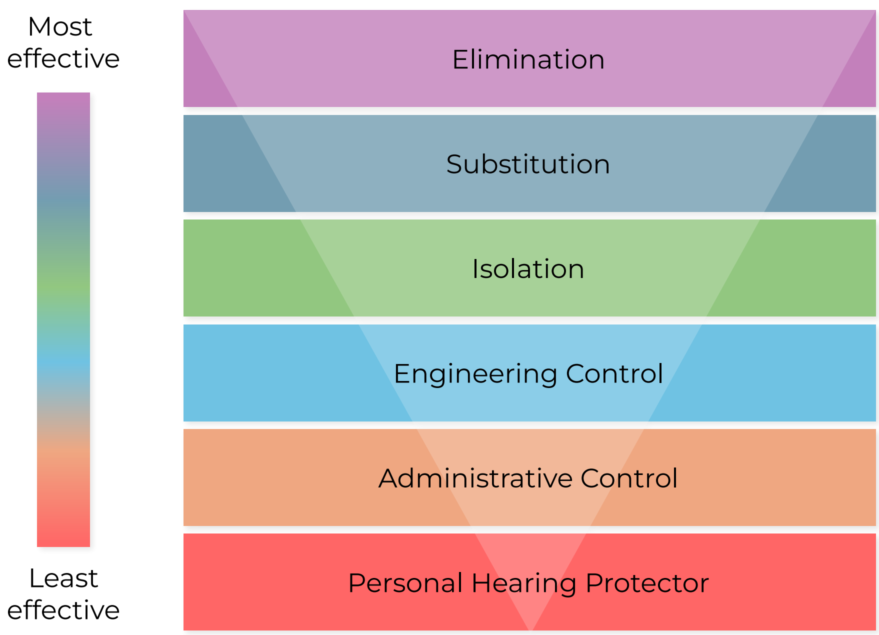


Figure 3.11: Hierarchy of Noise Control.

Noise from most equipment comprises mainly waste energy. For this reason, as well as others related to efficiency, the best way to reduce noise is to tackle the problem at the source. Generally, reducing the noise at the source also offers the most options. Reduction at path would generally involve adding barriers or enclosing the equipment but may also involve adding sound-absorbent materials. Reductions of more than a few decibels are difficult to achieve by these modifications.

At the other end, reduction at receiver that affected employee is achieving by either removing the employee from the sound field, limiting his working time in the area, or using personal hearing protector (PHP). PHP is highly dependent on consistency appropriate human behavior to work adequately, requires very robust management commitment and enforcement. Therefore, PHP tends to be less effective.

Furthermore, PHP ranks lowest in the hierarchy of noise control strategies and should not be relied upon as the primary means of noise control, but rather be treated as a means of last resort for controlling noise exposure which only to be used as a temporary or supplementary measure. The usage of PHP rely on where other higher hierarchy of noise control measures strategies are pending or been attempted and proven inadequate, ineffective, or impracticable.

3.3 Noise Control Strategies

Possible strategies for noise control are always numerous for new facilities and products than for existing facilities and products. Consequently, it is always more cost effective to implement noise control at the design stage than wait for complaints or noise risk assessments about a finished facility or product. For both existing and proposed new facilities and products, an important part of the process is to identify noise sources and rank order them in terms of contribution to excessive noise. When the requirements for noise control have been quantified and sources identified and ranked, it is possible to consider various options for control and finally to determine the cost effectiveness of the various options.

Noise level prediction and calculation of the effect of noise control carried out in octave frequency bands. Generally, octave band analysis provides a satisfactory compromise between too much or too little detail. Where greater spectrum detail is required, 1/3 octave band analysis is often sufficient, although narrower band analysis (1 Hz bandwidth for example) is useful for identifying tones and associate noise sources.

Any noise problem may be described in terms of a sound source, a transmission path and a receiver as shown in **Figure 3.12**. Noise control may take the form of altering any one or all these elements.

Figure 3.12: Element in noise problem description

A simple approach would be where a source such as a machine produces noise that reaches a receiver in the same room by three different paths: direct airborne transmission, airborne transmission after reflection from room surfaces and as a result of structure borne sound via transmission of vibration from the machine through floor or walls.

When faced with an industrial noise problem, reducing its hazard can be achieved in several ways and these are listed below in order of effectiveness:

1. At Source
    - Eliminate the hazard, which means physically removing it
    - Substitute the noisy process with a quitter one
    - Reduce the hazard by quiet technology design

2. At Transmission Path
    - Isolate personnel from the hazard via physical barrier or mufflers

3. At Receiver
    - Change the way employee work by rotating them out of noisy areas or by introducing quieter ways of doing things
    - Provide earplugs or earmuffs

When considered in term of cost effectiveness and acceptability, modification at source is well ahead of either modification of the transmission path or the receiver. On the other point of view for existing facilities, the last two may be the only feasible options.

3.4 Overall Noise Control Procedure

The mode of tackling an occupational noise problem is somewhat similar to any method of controlling hazard at workplace. Appropriate control measures include such things as change in plant design and layout, substitution of a less hazardous work method, reduction of the hazard at source and reduction of the hazard at its transmission path. It has been proven helpful in the past to follow a structured method of analysis so that no assumption of control measure is implemented. Thus, the recommended method of approach according to the order outlines below.

1. Plant planning
   - Design
   - Layout

2. Substitution
   - Equipment
   - Process
   - Material

3. Engineering control
   - Absorption
   - Insulation
   - Distance
   - Silencer
   - Vibration Isolation
   - Damper

4.0 PLANT PLANNING CONTROL

4.1 Introduction

As with any occupational hazard, control technology should aim at reducing noise to acceptable levels by action on the work environment. Such action involves the implementation of any measure that will reduce noise being generated, and/or will reduce the noise transmission through the air or through the structure of the workplace. Such measures include modifications of the machinery, the workplace operations, and the layout of the workroom. In fact, the best approach for noise hazard control in the work environment, is to eliminate or reduce the hazard at its source of generation, either by direct action on the source or by its confinement.

4.2 Existing Installations And Facilities

In existing facilities, quantification of the noise problem involves identification of the source or sources, determination of the transmission paths from the sources to the receivers, rank ordering of the various contributors to the problem and finally determination of acceptable solutions.

To begin, noise levels must be determined at the locations from which the complaints arise. Once levels have been determined, the next step is to reduce noise below noise exposure limit to each location and thus to determine the required noise reductions, generally as a function of one-one or one-third octave frequency bands.

Once the noise levels have been measured through noise risk assessment and the required reductions determined, the next step is to identify and rank order the noise sources responsible for the excessive noise. The sources may be subtle or alternatively many, in which case rank ordering may be as important as identification.

Often noise sources are either vibrating surfaces or unsteady fluid flow (air, gas or steam). The latter are referred to as aerodynamic sources and often associated with exhausts. In most cases, it is worthwhile to determine the source of the energy which is causing the structure or the aerodynamic source to radiate sound, as control may best start there.

Once the noise sources have been ranked in order of importance in terms of their contribution to the overall noise problem, it is often also useful to rank them in terms of which are easiest to do something about and which affect most people and take this into account when deciding which sources to treat first of all.

Employer should follow the recommended work process for noise control implementation of existing facilities as specified in **Figure 4.1** to ensure the practicability and adequacy .

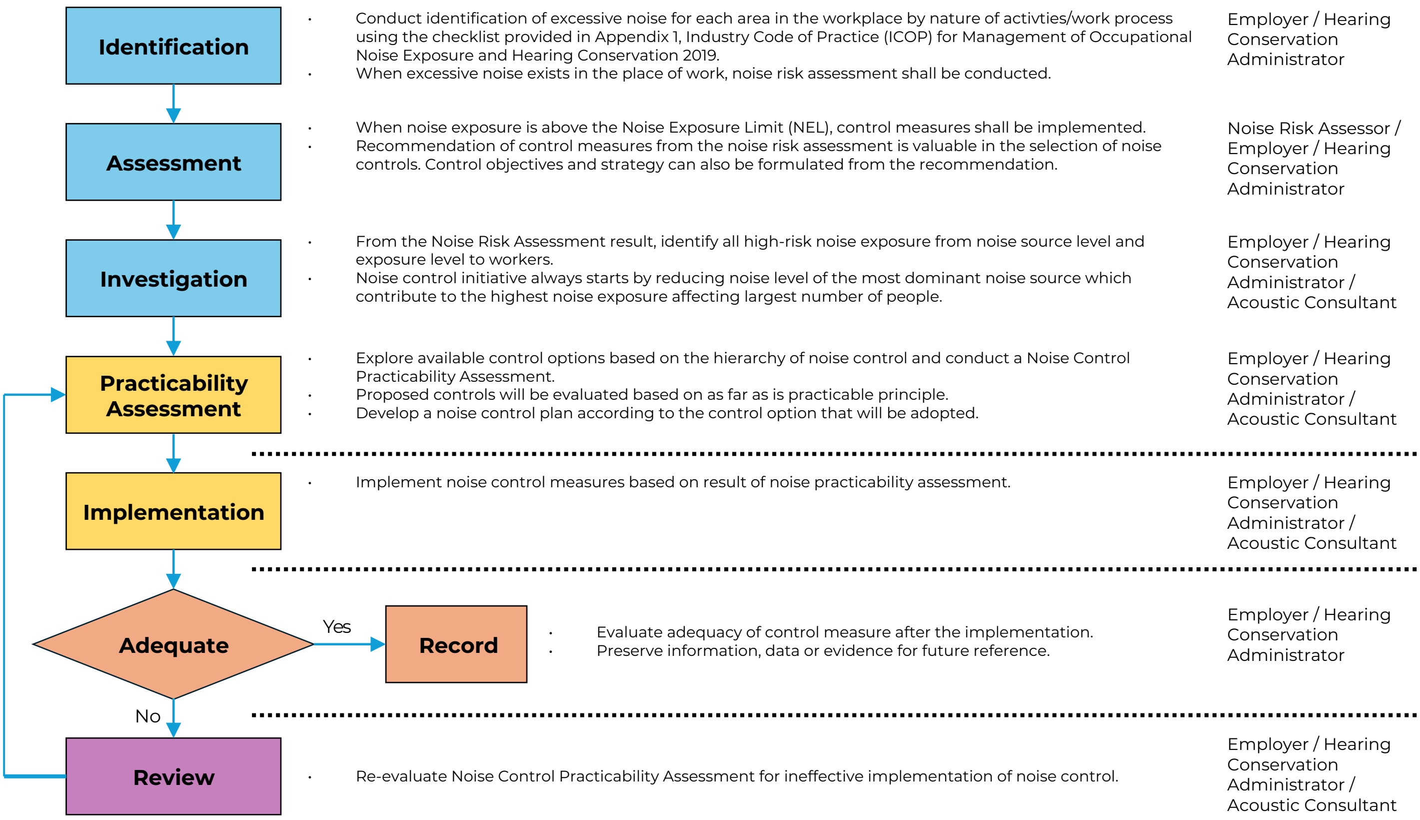


Figure 4.1: Recommended work process for noise control implementation of existing facilities

4.3 Noise Control Practicability Assessment

Based on noise risk assessment conducted in workplace, if any of the employees is exposed to noise exceeding noise exposure limit (NEL), employer shall take measures to reduce the noise. The employer shall make an assessment whether the measure is practicable to reduce excessive noise by way of engineering control or administrative control according to the sequence determined under Regulation 6 (4) of Noise Regulation 2019 and prepare a justification report.

4.3.1 Assessment Report

The report should consider the following factors:

a) The scale of the noise problem and its impact on the business (including workers);
b) Cost and effort required to reduce noise exposure;
c) Effectiveness of planned control measures; and
d) The number of individuals who would benefit from those control measures.

The report should include introduction, scope of assessment, control options practicability assessment, summary of noise control measures to be implemented and gantt chart for action plan. Details explanation of each parts of the report are as follows;

1. Introduction

a) Particular of the workplace:
    i. Name
    ii. Address
    iii. DOSH registration number

b) Date of assessment

2. Scope of assessment

a) Similar exposure group (SEG) involved in the assessment

b) Related Noise Risk Assessment Report
    i. Name of Noise Risk Assessor
    ii. Date of assessment by Noise Risk Assessor
    iii. Date of report received

3. Scope of assessment

a) Similar Exposure Group (SEG)

Table 1: Information of SEG

SEG	Noise Source	$L_{EX,8h}$ dB(A)	Max Level, dB(A)	Peak Level, dB(C)	Number Of Workers Exposed
SEG 1					
SEG 2					
SEG n					

Table 2: Assessment of Control Practicability for SEG 1

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes / No)	If Control option cannot be adapted, please justify.	Predicted Noise Exposure level after implementation[2], dB(A)
Engineering	NRA's Recommendation - 1							
	NRA's Recommendation - n[3]							
	Other options[4] - 1							
	Other options[4] - n[3]							

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Engineering control solely possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected control options (can be one control option or combination of any control options stated)
- If No, proceed with assessment of combination of engineering control and administrative control approach

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes / No)	If Control option cannot be adapted, please justify.	Predicted Noise Exposure level after implementation[2], dB(A)
Administrative	NRA's Recommendation - 1							
	NRA's Recommendation - n[3]							
	Other options[4] - 1							
	Other options[4] - n[3]							

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Combination of engineering control and administrative control possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected combination of engineering and administrative control options.
- If No, consider to implement administrative control only (can be one control option or combination of any control options stated)

Administrative control solely possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected control options (can be one control option or combination of any control options stated)
- If No, proceed with assessment of personal hearing protection usage

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes / No)	If Control option cannot be adapted, please justify.	Predicted Noise Exposure level after implementation[2], dB(A)
PHP	NRA's Recommendation - 1							
	Other options[4] - n[3]							

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Combination of engineering control, administrative control and PHP possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the combination approach.
- If No, consider to implement PHP only

PHP usage solely possible to reduce exposure below NEL? **(Yes / No / Not Applicable)**

Note:
- If Yes, stop assessment and implement the usage of PHP
- If No, reassess with other control options.

Table 2: Assessment of Control Practicability for SEG 1

Control Hierarchy	Control Option[1]	Details of control specifica-tions	Cost, RM	Potential Noise Reduction, dB(A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes / No)	If Control option cannot be adapted, please justify.	Predicted Noise Exposure level after implementation[2], dB(A)
Others	NRA's Recommendation - 1							
	Other options[4] - n[3]							

Note:

[1] Attach relevant information gathered such as quotation and brochure from supplier, source of reference etc.

[2] Overall predicted noise exposure level after implementation of selected control options based on noise decibel addition equation.

[3] Add if have more than one control options recommended for this SEG.

[4] Based on employer's survey, consultation with acoustic consultant or noise control supplier.

Note: Repeat all steps stated in SEG 1 to assess SEG 2, SEG 3 onwards.

4. Summary of Noise Control Measures to be Implemented

No	SEG	Please tick (√) measures to be implemented			
		Engineering	Administrative	PHP	Others
1	SEG 1				
2	SEG 2				
n	SEG n				

5. Gantt Chart for Action Plan

No	SEG	Control Measures To be Implemented	PIC	Year _____ / Month											
				1	2	3	4	5	6	7	8	9	10	11	12
1	SEG 1														
2	SEG 2														
3	SEG n														

4.3.2 Example of Assessment Report

1. Introduction

XYZ Sdn. Bhd is located at Lot 9, Jalan 9, 99000 Johor. This company manufacture children's bed. DOSH registration number of the factory is JH/09/9999.

The Noise Control Practicability Assessment Report was performed by Mr. Ahmad on 9 September 2023.

The purpose of the report is to carry out practicability assessment of noise reduction measures and prepare the assessment report as required by Regulation 6(3), 6(4) and 6(5) of OSH (Noise Exposure) Regulations 2019.

2. Scope of Assessment

Particular of Noise Risk Assessment Report

- Name of noise risk assessor: Mr. Ali (DOSH Reg. No: HQ/19/PEB/00/999)
- Date of assessment by noise risk assessor: 9 July 2023
- Date of report received: 9 August 2023

Similar exposure group (SEG) involved in assessment

- Lamination operator
- Cutting operator

3. Control Options Practicability Assessment

Table 1: Information of SEG

SEG	Noise Source	Noise Level dB(A)	$L_{EX,8h}$ dB(A)	Max Level, dB(A)	Peak Level, dB(C)	Number Of Workers Exposed
Lamination Operator	S1 - Product Impact	94.3	91.3	110.5	130.3	12
	S2 - Dust Collector	88.2				
	S3 - Motor	86.7				
Cutting Operator	S1 - Cutting Saw	98.1	94.2	113.3	138.5	8
	S2 - Air Gun	95.6				
	S3 - Motor	87.1				

Table 2: Assessment of Control Practicability for Lamination Operator

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Predicted Noise Level After Implementation, dB (A)	Potential number of workers that will be benefit	Control option will be adapted? (Yes/No)	If Control option cannot be adapted, please justify.	Overall Predicted Noise Exposure level after implementation[2], dB(A)
Engineering	(S1) Automated board flipping machine	To replace manual board flipping process with automated machines.	40,000	15	79.3	12	Yes	Nil	90.8
	(S2) Partition layer	To install partition layer to segregate the dust collector from the lamination line.	3,000	6	82.2	12	Yes	Nil	88.6
	(S3) Acoustic motor enclosure	To install acoustic enclosure at motor machine complete with duct silencer attached to the ventilation fan.	15,000	10	76.6	12	Yes	Nil	84.7

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Engineering control solely possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected control options (can be one control option or combination of any control options stated)
- If No, proceed with assessment of combination of engineering control and administrative control approach

Table 3: Assessment of Control Practicability for Cutting Operator

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Predicted Noise Level After Implementation, dB (A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes/No)	If Control option cannot be adapted, please justify.	Overall Predicted Noise Exposure level after implementation[2], dB(A)
Engineering	(S1) Replace cutting saw blade	To replace silent cutting saw blade.	2,500	10	88.1	8	Yes	Nil	96.8
	(S2) Quiet air gun	Replace to quieter air gun.	700	8	87.6	8	Yes	Nil	92.4
	(S3) Acoustic motor enclosure	To install acoustic enclosure at motor machine complete with duct silencer attached to the ventilation fan.	15,000	10	77.1	8	Yes	Nil	91.0

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Engineering control solely possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected control options (can be one control option or combination of any control options stated)
- If No, proceed with assessment of combination of engineering control and administrative control approach

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Predicted Noise Level After Implementation, dB (A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes/No)	If Control option cannot be adapted, please justify.	Overall Predicted Noise Exposure level after implementation[2], dB(A)
Administrative	Job rotation	To limit worker's exposure at less than 2 hours per shift.	Nil	Nil	91.0	Nil	No	Not practical to implement due to work must conduct continuously and limitation of workers (other workers also work at noise risk area).	91.0

Predicted noise exposure level after implementation is below NEL? **(Yes / No)**
Combination of engineering control and administrative control possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected combination of engineering and administrative control options.
- If No, consider to implement administrative control only (can be one control option or combination of any control options stated)

Administrative control solely possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the selected control options (can be one control option or combination of any control options stated)
- If No, proceed with assessment of personal hearing protection usage

Control Hierarchy	Control Option[1]	Details of control specifications	Cost, RM	Potential Noise Reduction, dB(A)	Predicted Noise Level After Implementation, dB (A)	Potential Number of workers that will be benefit	Control option will be adapted? (Yes/No)	If Control option cannot be adapted, please justify.	Overall Predicted Noise Exposure level after implementation[2], dB(A)
Personal Hearing Protection (PHP)	PHP Program	Continue to supply ear muff with minimum NRR of 19 dB until effective engineering and administrative control implemented.	RM 2,000/year inluding training, supervision & maintenance program	11.5 (Based on actual NRR of 30 dB)	79.5	8	Yes	Nil	79.5 (For the purpose of prediction, noise reduction calculation is based on level of noise source)

Predicted Noise exposure level after implementation is below NEL? **(Yes / No)**
Combination of engineering control, administrative control and PHP possible to reduce exposure below NEL? **(Yes / No)**

Note:
- If Yes, stop assessment and implement the combination approach.
- If No, consider to implement PHP only

PHP usage solely possible to reduce exposure below NEL? **(Yes / No / Not Applicable)**

Note:
- If Yes, stop assessment and implement the usage of PHP
- If No, reassess with other control options

4. Summary of Noise Control Measures to be Implemented

No	SEG	Please tick (√) measures to be implemented			
		Engineering	Administrative	PHP	Others
1	Lamination Operator	√			
2	Cutting Operator	√		√	

5. Gantt Chart for Action Plan

No	SEG	Control Measures To be Implemented	PIC	Year 2024 / Month				Year 2025 / Month					
				9	10	11	12	1	2	3	4	5	6
1	Lamination Operator	(S1) Automated board flipping machine	En. Ali										
		(S2) Partition layer	En. Ali										
		(S3) Acoustic motor enclosure	En. Ali										
2	Cutting Operator	(S1) Replace cutting saw blade	Mr. Ang										
		(S2) Quiet air gun	Mr. Ang										
		(S3) Acoustic motor enclosure	Mr. Ang										
		PHP Program	Pn. Ana										

4.4 Installations And Facilities In The Design Stage

In new installations, quantification of the noise problem at the design stage may range from simple to difficult but never impossible. At the design stage the problems are the same as for existing installations which are identification of the source or sources, determination of the transmission paths of the noise from the sources to the receivers, rank ordering of the various contributors to the problem and finally determination of acceptable solutions. Most importantly, at the design stage the options for noise control are generally many and may include rejection of the proposed design.

The first step for new installations is to determine the acceptable noise level for sensitive locations which may typically include locations of operators of noisy machinery. If the estimated noise levels at any sensitive location exceed the established criteria, then the equipment contributing most to the excess levels should be targeted for noise control, which could take the form of:

- Specifying lower equipment noise levels to the equipment manufacturer as a maximum 85 dB(A) shall be the design noise limit used for the work area (care must be taken whenever importing equipment, particularly second hand which can be very noisy and hence no longer acceptable in the country of origin);

- Including noise control fixtures (mufflers, barriers, vibration isolation systems, enclosures, or factory walls with a higher sound transmission loss) in the factory design; or

- Rearrangement and careful planning of buildings and equipment within them. The essence of the discussion is that sources placed near hard reflective surfaces will result in higher sound levels at the approximate rate of 3 dB for each large surface. Note that the shape of the building space generally is not important, as a reverberant field can build-up in spaces of any shape. Care should be taken to organize production lines so that noisy equipment is separated from workers as much as possible. Sufficient noise control should be specified to leave no doubt that the acceptable noise level will be met at every sensitive location. Saving money at this stage is not cost effective in the long term.

4.5 Control By Plant Planning

One of the greatest opportunities for the industrial hygiene practitioner in the field of noise control is to guide the design of new plants and the modernization of existing ones using available acoustics knowledge and technology such as predictive modeling software. In this manner noise problems can be anticipated and prevented before arise. Successful planning for noise control involves:

- Knowledge of the noise characteristics of each machine and process;
- Proposed location of each noise source, operator, and maintenance worker; and
- election of design criteria based on employee exposure time

Engineering specifications for design and selection of equipment should incorporate a requirement for noise information data. In most cases, the machinery manufacturer is in the best position to reduce the noise of the machine at the source with built-in designs. It is expected that many such designs will not substantially increase the cost of the machine. The acoustically important details of the building's load-bearing structure and work areas should be calculated and fixed early in the planning stage. The need for noise control depends first and foremost on the way the plant is designed and laid out. The structural design of the building often depends on where the machinery is placed and the need for insulation against both airborne and structure-borne sound. Towards this end, it is important to consider the following:

1. The building's load-bearing structure, floors and machine foundation should be chosen so that all noise sources can be effectively vibration isolated. Heavy equipment demands stiff and heavy foundations, which must not be in direct contact with other parts of the building structure.

2. Powerful noise sources should be enclosed by structures which give adequate airborne-sound insulation. Doors, inspection windows and other building elements where there is a risk of sound leakage require special attention.

3. Rooms where there are sound sources and where personnel are present continuously should be provided with ceiling cladding (also wall cladding where high ceilings are concerned) which absorb the incident sound. Sound absorption characteristics vary widely for different materials which must therefore be chosen with regard to characteristics of the noise. Good sound absorption characteristics can often be combined with good thermal insulation.

4. Office areas should be separated from building elements where vibrating equipment is installed by a joint of elastic material.

5. Walls and ceiling construction, windows, doors, etc. should be chosen so as to achieve the required sound isolation.

6. Mounting noisy equipment on light or movable partitions should always be avoided. If ventilation for cooling systems must be mounted on such a light foundation in any case, e.g. a false ceiling, special effort must be made to obtain sufficient vibration isolation.

7. In open-plan offices and large rooms where there are several office functions carried out in the same room, there must be a ceiling with high sound absorption; and soft carpeting on the floor is also beneficial. It should be noted that it is especially important to ensure that sound absorption is effective not only at the high frequencies, but also at the low and medium frequencies.

The most effective and economical approach to noise control is to include noise control features as an integral part of the plant design. Such an approach is most efficiently handled by proper use of equipment performance and design specifications. Performance specifications require that the proposed equipment will satisfy the selected criterion; design specifications indicate to the supplier specific noise control features known to be effective and compatible with plant operations.

A sample equipment noise specification schedule is given as below:

1. Equipment noise level limitations, noise testing procedures, and noise data documentation requirements. These limitations and requirements should apply to all stationary and mobile equipment and machinery that produce steady continuous, fluctuating, and impulse noise.
2. Provisions for a uniform method of conducting and recording noise tests to be made on equipment. Requirements that the equipment manufacturer and the engineering contractor guarantee to meet the noise limits set forth in the specification.
3. Statements indicating that if the noise survey of a completed plant indicates that an item of equipment is producing noise levels that exceed equipment specifications, the equipment manufacturer, subcontractor, or engineering contractor will be responsible for the extra cost of treating the equipment to bring noise levels within the equipment and plant specification requirements.
4. Agreements that all equipment manufacturers and engineering companies will be penalized if the equipment and plant and/or site noise specifications are not met.
5. Pre-bid and final test noise measurements shall be made on the purchased equipment and test data will be made available and determined acceptable by the buyer or their representative prior to shipment with authorized signatures.
6. Reservation of the right to send qualified representatives to the equipment manufacturing plant to observe or conduct noise tests.
7. Maximum acceptable noise levels for the plant site perimeter, plant area perimeters, and interior plant areas including production areas, control rooms, offices, laboratories, etc.
8. Maximum acceptable vibration levels for all equipment, and noise reverberation levels for all work and process areas.
9. Instrumentation and measurement techniques.
10. Pre-bid equipment noise level data sheets requiring equipment noise specification guarantee signature and buyer approval signature.
11. Final test noise level data sheets requiring equipment noise specification guarantee signatures and witness/or noise data acceptance signatures.

In selecting equipment for a new plant or for equipment replacement or additions to an existing facility, satisfactory noise limits often can be obtained by proper attention to specific design features of the equipment. The prescribed equipment specifications may sound stringent. However, successful activities in all plants that have followed the criteria have not only been rewarding but also cost-effective. A summary of some of these design features are shown in **Table 4.1**.

Table 4.1: Desirable features of equipment design for noise reduction.

Equipment	Source of Noise	Design Features
Heaters	Combustion at burners	Acoustic air intake plenum
	Inspiring of premix air at burners	Inspiriting air intake silencer Acoustic air intake plenum
	Draft fans	Air intake silencer or acoustic plenum lagging
	Ducts	Lagging
Motors	TEFC cooling air fan WP II cooling air openings	Acoustic fan shroud, unidirectional fan, and/or intake silencer
	Mechanical and electrical	Enclosure
Air fin coolers	Fan	Lower RPM (increased pitch). Tip and hub seals. Increased number of blades. Decreased static pressure drop. More fin tubes.
	Speed changer	Belts in place of gears.
	Fan shroud	Streamlined air flow. Damping and stiffening.
Centrifugal compressors	Discharge piping and expansion joints	In-line silencer and/or lagging.
	Anti-surge bypass system	Quiet valves, reduced velocity, and streamlining. Lagged valves and piping. In-line silencers.
	Intake piping and suction drum	Lagging.
	Air intake/air discharge	Silencer.
Screw compressors (axial)	Intake and discharge piping	Silencers and lagging
Speed changers	Gear meshing	Enclosure, constrained damping on case, or lagging
Engines	Exhaust	Silencer (muffler)
	Air intake	Silencer
	Cooling fan	Enclosed intake and/or quieter discharge
Condensing turbine	Expansion joint on steam discharge line	Lagging
	Discharge jet	Discharge silencer

Equipment	Source of Noise	Design Features
Atmospheric exhausts and intakes	Upstream valves	Quiet valve or silencer
Piping	Leading pipe	Lagging
	Excess velocities	Limited velocities
		Smooth, gradual changes in size and direction
		Lagging
	Valves	Limited velocities
		Constant velocity or a quiet valve
		Divided pressure drop
Pumps	Cavitation of fluid	Enclosure
Flares	Steam jets	Multiport on air injectors

4.6 Equipment Noise Emission Level Data Sheet

Equipment noise emission level data sheet shall be prepared for all relevant items of equipment or an equipment train provided by a manufacturer or supplier specifying the noise exposure limit of 85 dB(A). If the components of a train will be provided by different suppliers, separate equipment noise data sheets shall be prepared. The equipment noise limits shall be given as an overall A-weighted value in decibels corresponding 63 Hz to 8 kHz octave-band spectrum may be specified in addition.

Guidance for designers, manufacturers, importers and suppliers on the presentation of information about noise levels generated by plant is provided in Appendix 8, The Industry Code of Practice (ICOP) for Management of Occupational Noise Exposure and Hearing Conservation 2019.

The format of an equipment noise emission level data sheet shown in **Table 4.2** shall be used to specify a sound pressure level at a specified distance, usually 1 m, from the equipment surface. The supplier is obliged to state noise guarantees for the equipment to be provided, for any of the conditions of operation for which the equipment can be expected to be used. The equipment noise emission level data sheet shall contain all the information required to understand, comply with and measure compliance with and the noise limits for the equipment to be supplied.

The equipment noise emission level data sheet with its guarantee section completed by the supplier, giving the following information:

01 Unsilenced sound pressure levels in octave bands, and the A-weighted overall levels of the equipment.

02 Sound pressure levels in octave bands, and the A-weighted overall levels of the equipment, together with details of any silencing measures that are necessary to meet the specified noise limit.

03 The minimum attainable sound pressure levels in octave bands and the A-weighted overall levels, if the specified limits cannot be met.

04 Details of any tonal, impulsive, intermittent, or fluctuating noise levels.

05 The information shall be given where applicable such as complete data / requisition sheets of any reduction measures.

Table 4.2: Equipment Noise Emission Level Data Sheet

<table>
<tr><th colspan="13">EQUIPMENT NOISE EMISSION LEVEL DATA SHEET</th></tr>
<tr><td colspan="7">Type of equipment:</td><td colspan="6">Mechanical power (kW):</td></tr>
<tr><td colspan="7">Supplier / manufacturer:</td><td colspan="6">Speed (RPM):</td></tr>
<tr><td colspan="7">Type No:</td><td colspan="6">Size l x b x h (m):</td></tr>
<tr><td colspan="13">1 GENERAL
This data sheet covers the noise limits of the equipment, given below.</td></tr>
<tr><td colspan="13">2 NOISE LIMITS TO BE MET BY EQUIPMENT
The noise generated by the equipment shall not exceed the more stringent of the noise limits given in the table below, for any of the conditions of operation for which the equipment may normally be expected to be used.

L_p is the maximum sound pressure level, re 20 µPa in dB, for the mode of operation indicated at any location at 1 m from the equipment surface.

If the equipment generates noise with tonal or impulsive components, this shall be indicated.</td></tr>
<tr><th rowspan="3">Equipment items/Locations</th><th rowspan="3">a</th><th colspan="9">Sound pressure level guaranteed by supplier dB</th><th rowspan="3">b

Exceeding Noise Exposure Limit, 85 dB(A)

Yes / No</th><th rowspan="3">Remarks

Noise Reduction Measures</th></tr>
<tr><th colspan="8">Unweighted octave-band levels</th><th rowspan="2">Total A-weighted</th></tr>
<tr><th>63 Hz</th><th>125 Hz</th><th>250 Hz</th><th>500 Hz</th><th>1 kHz</th><th>2 kHz</th><th>4 kHz</th><th>8 kHz</th></tr>
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<tr><td colspan="13">Purchaser shall indicate:
In column “b”, the A-weighted sound pressure limit i.e., 85 dB(A)
In column “a” using the appropriate number, which of the following applies to the required noise levels:
1) Without acoustic provisions
2) With acoustic provisions
3) Special low-noise design
4) Supplier best estimate, not necessarily guaranteed
5) Others ....</td></tr>
<tr><td colspan="13">Information on special operating conditions:</td></tr>
</table>

5.0 SUBSTITUTION CONTROL

5.1 Use Quieter Equipment

The first step in providing quiet workplace equipment is to make a strong effort to have equipment purchase specifications include noise emission limits. Even if the desired specifications may not always be available, or even achievable, at least these specifications will provide an incentive for the design and development of quieter products in the future. In selecting equipment for replacement or additions to an existing facility, a satisfactory noise limit often should be obtained by proper attention to specific design features of the equipment.

Successful activities in all plants that have followed the criteria have not only been rewarding but also cost-effective. Modification to equipment to reduce noise includes closer tolerances, better assembly, balancing of rotating machinery, redesign of components and other quality control measures. Usually, these changes must be left to the manufacturer of the equipment. However, users of equipment can specify the noise level that will be tolerated in new equipment purchases.

It would be reasonable compliance to the Industry Code of Practice for Management of Occupational Noise Exposure and Hearing Conservation 2019 that include provision for declaration of equipment noise emission levels by manufacturers. Many manufacturers worldwide already attempt to distinguish their products by emphasizing their lowered noise emission level. Purchasers of equipment need to be aware of the cost benefit if quiet equipment is purchased such as avoiding the need for a hearing conservation program and preventing hearing loss among employees. The hearing conservation administrator should assure that both overall management and the purchasing department are aware of the value and importance of buying quiet equipment.

Quiet often and impossible to substitute a quieter machine from a noisy machinery. Existing machines were probably selected based on the most economical and efficient means of producing a desired product or service. Noise level was very likely not to be considered as an important factor during purchasing time. However, it may be more economical to pay more for quieter equipment rather than to purchase cheaper noisy equipment that requires additional expenditure for noise control measures. **Figure 5.1** presents a comparison of noise levels produced by a normal air gun versus a silent air gun. The left image shows the noise level generated by the normal air gun, which is measured at 91.2 dB (LAeq). The right image displays the noise level of the silent air gun, measured at a lower level of 81.5 dB (LAeq). This indicates that the silent air gun produces significantly less noise compared to the normal air gun, demonstrating its effectiveness in reducing noise exposure.

New equipment type and speed should be selected on the basis of the applicable noise criteria. Some substitution or replacement examples are given below:

1. Replace high-frequency propeller or axial flow fan with squirrel cage type blower, which emits noise of lower frequency.
2. Replace high speed fans or blowers with low-speed types.
3. Replace gear increasers with turbine drives which are usually less noisy where high speeds are concerned.
4. Replace air ejector for removing parts from punch presses with a mechanical ejector.
5. Replace a mechanical hopper vibrator with an internal pulsating diaphragm type pneumatic feeder. The ultimate goal is to take only the part of the material handling system which impedes the flow. Avoid vibrators employing metal to metal impact.
6. Replace pneumatic portable tools with electric tools.
7. Replace a pneumatic riveting hammer with a hydraulic press. The exhaust of compressed air from the operation cylinder of pneumatic presses must be muffled.
8. If an impact tool must be used, choose the smallest tool that is capable of performing the job.
9. Replace mechanical presses with hydraulic presses.

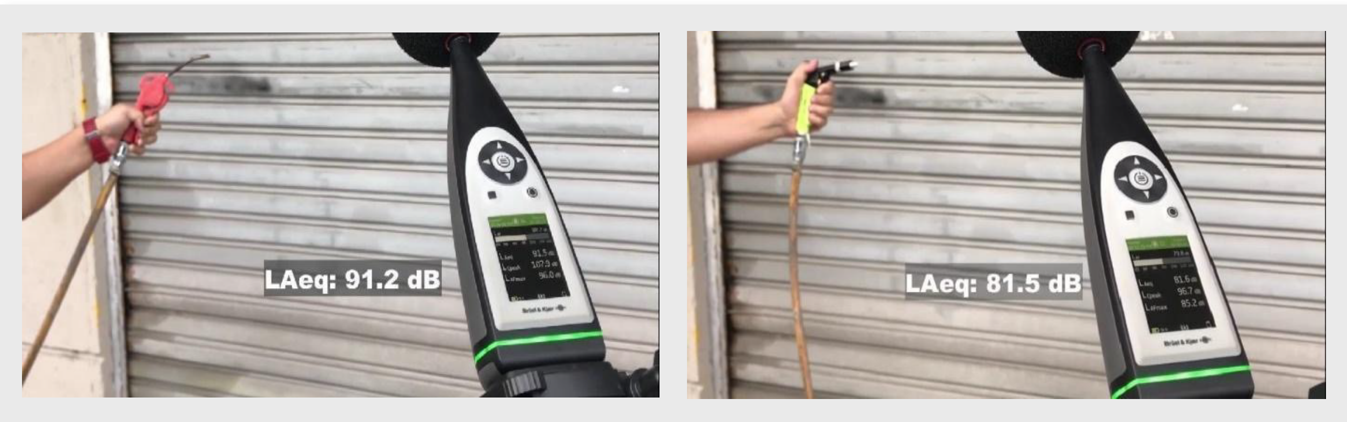


Figure 5.1: Comparison of noise level between normal air gun and silent air gun.

5.2 Use Quieter Processes

In many cases, changing the process can be one way of getting to grips with noise generation. This would require one to be aware of the availability of quitter processes for both actual production work as well as material handling. This would in turn involve cooperation between the employer, supplier, process designer and OSH practitioner. The following are several common examples.

i. Replace percussion or impact riveting with: (i) welding, unless chipping is also required in the weld preparation; (ii) high strength bolts; or (iii) compression riveting as shown in **Figure 5.2**.

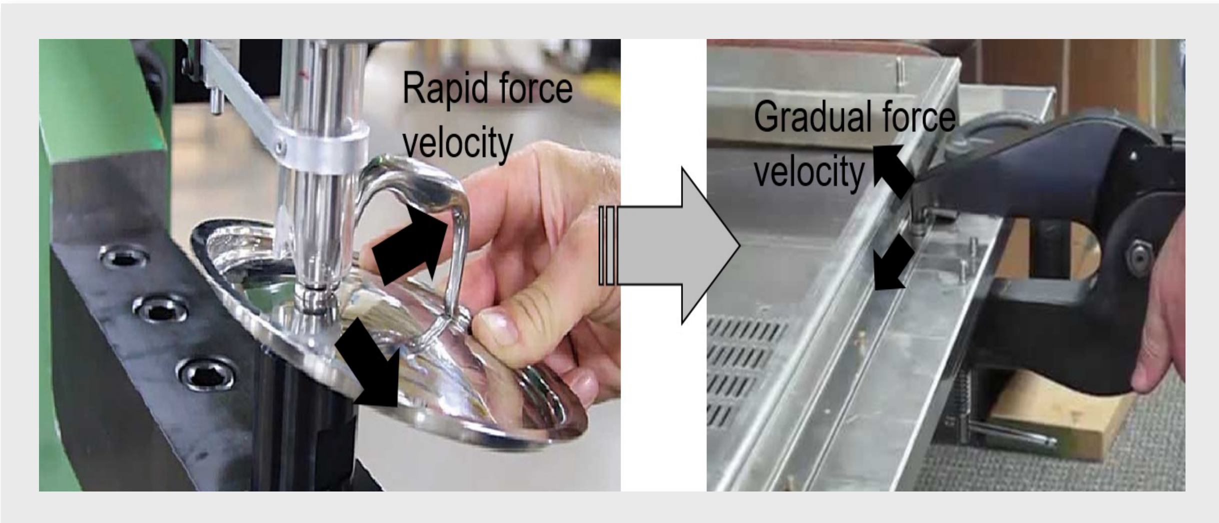


Figure 5.2: Eliminating rapid force velocity by substituting gradual force velocity using compression riveting.

ii. Replace chipping with grinding; Arc-air process of flame gouging as shown in **Figure 5.3**.

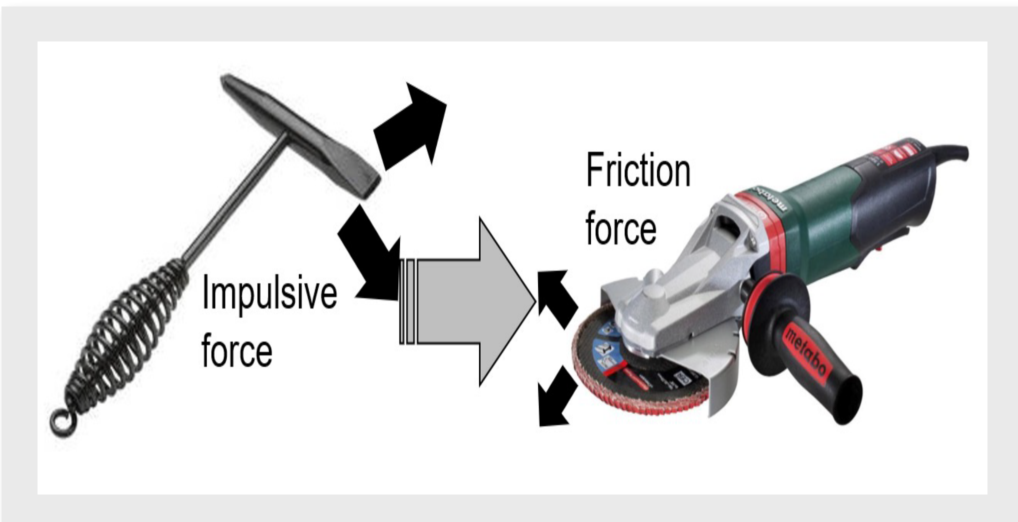


Figure 5.3: Substitute impulsive force from chipping with friction force from grinding.

iii Replace cold working with hot working as shown in **Figure 5.4**.

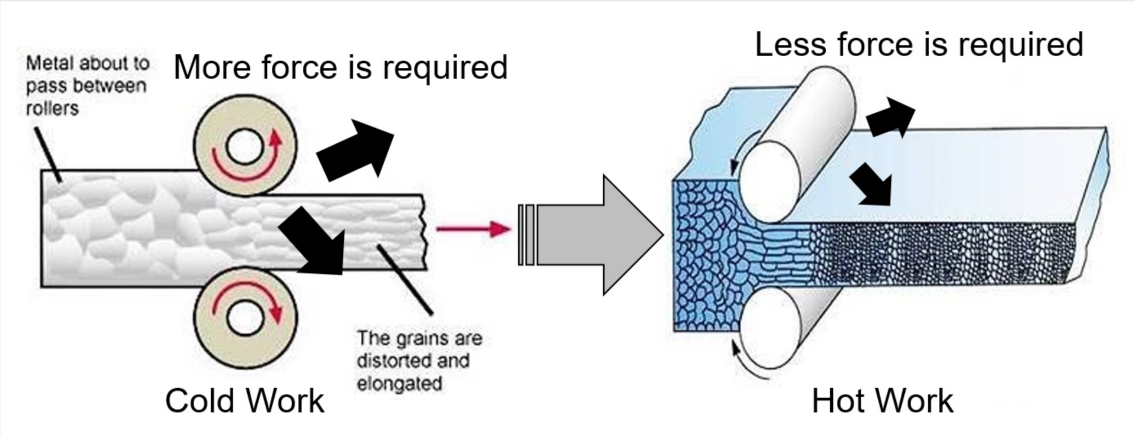


Figure 5.4: Replace metal cold work with hot work process.

iv Replace high impact impulsive force (i.e., mechanical power) used for pile driving in building and construction with hydraulic power as shown in **Figure 5.5**.

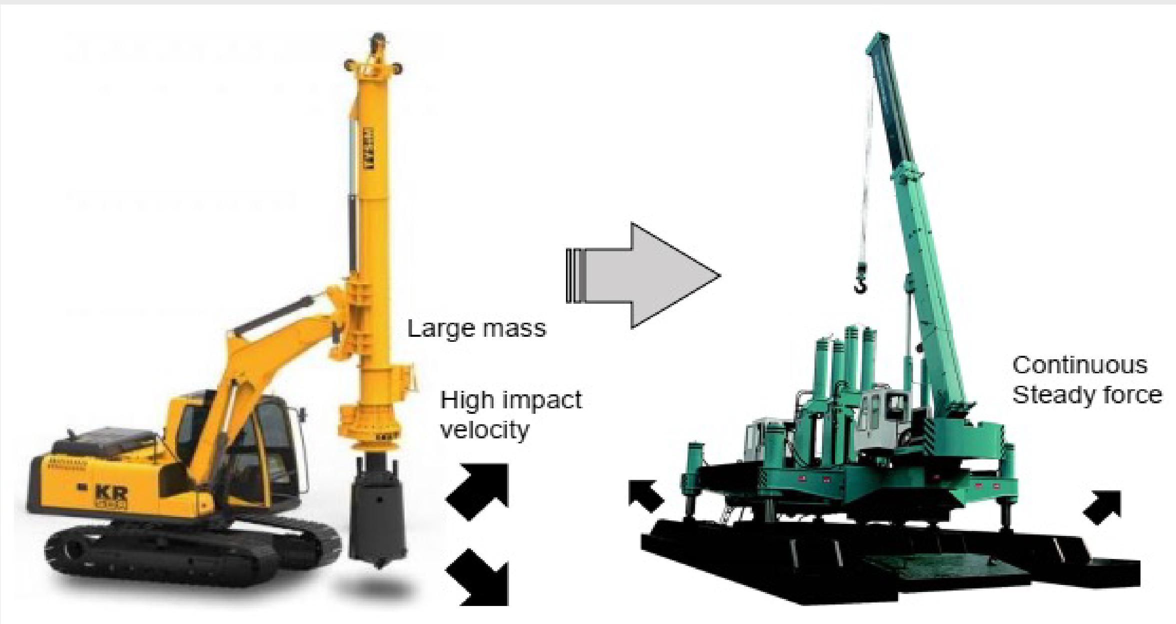


Figure 5.5: Eliminating impact noise by substituting hydraulic power for impulsive force.

In most building and construction work, sheet piles are normally drive into the ground via the impact of a heavy mass (i.e., the pile driver) dropped from a great height, often powered up again by exploding a diesel charge. Excessive noise levels are generated both by the impact on the pile and from the explosion, and annoyance may be caused at distances of up to a few miles. In many situations, it is possible to use an entirely different technique as shown in **Figure 5.5**, with avoids impact noise. A set of hydraulically operated rams is used to grip a number of sheet piles simultaneously. One pile is force down at a time while the machine pulls upwards on all the rest, which anchor it to the ground. Vibration of the ram holding the pile being drive assists its progress. Impact is avoided completely, and noise level can fall to as low as the hydraulic equipment allows.

5.3 Use Quieter Material

Material from which building, machinery, piping, and containers are constructed have a vital relation to noise control. Some materials have high internal damping and are call 'dead' material, while others call 'live' material, have low internal damping, and cause a ringing sound when struck. Some ways of using quitter materials are as follows:

i. Replace acoustically 'live' materials with acoustically 'dead' (highly internal damped) materials. For instance, elastomers are good materials to use for bumpers. Good examples are gaskets, seals industrial truck tires and rubber caps for hammerhead as shown in **Figure 5.6**.

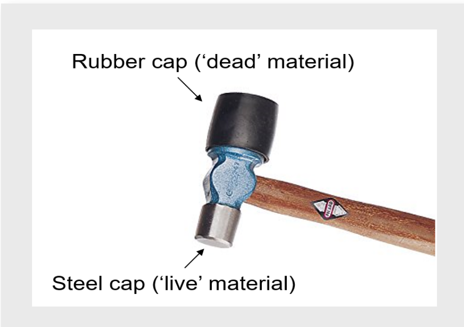


Figure 5.6: Putting high internal damped material at hammerhead.

ii. Replace steel wheels on hand trucks with rubber or plastic tires as shown in **Figure 5.7**.

Figure 5.7: Example of rubber-type material for trolly tires.

6.0 ENGINEERING NOISE CONTROL

6.1 Hierarchy Of Noise Control Strategy

6.1.1 Source

Engineering control at the source is the preferred method of permanently removing the problem of noise exposure due to machinery or processes at the workplace. Since all noise-emitting objects generate airborne energy (noise) and structure-borne energy (vibration), the control of these noise problems may require modification, partial redesign or replacement of the noise-emitting object. Noise risk assessment conducted can identify how and where the noise is generated. Some problems can be solved by relatively inexpensive and simple procedures, although some are difficult. The advice from specialists such as acoustic experts and Noise Risk Assessor may be beneficial in providing best results. These guidance include references to some of the simpler methods of noise control that might be achieved.

When seeking a solution to a noise problem, an understanding of the operation of the machine or process is necessary in considering the possible control of the noise at source. Engineering noise control measures can be specifically targeted at the machine and its parts, or towards the actual processes, including material handling systems.

General noise control solutions, and examples of specific engineering noise control measures which can be carried out on machines, are provided below:

a. Eliminate or replace the machine or its operation by a quieter operation with equal or better efficiency, for example, by replacing rivets with welds.

b. Replace the noisy machinery by installing newer equipment designed for operating at lower noise levels. Machinery power sources and transmissions can be designed to give quiet speed regulation, for example, by using stepless electric motors. Vibration sources can be isolated and treated within the machine. Cover panels and inspection hatches on machines should be stiff and well damped. Cooling fins can be designed to reduce the need for forced airflow and hence fan noise.

c. Correct the specific noise source by minor design changes. For example, avoid metal-to-metal contact by the use of plastic bumpers, or replace noisy drives with quieter types or use improved gears.

d. A high standard of equipment maintenance should be provided to facilitate compliance with the Occupational Safety and Health (Noise Exposure) Regulations 2019 and, going a step further, reduce noise levels to as low as practicable. Badly worn bearings and gears, poor lubrication, loose parts, slapping belts, unbalanced rotating parts and steam or air leaks all create noise which can be reduced by good maintenance. Equipment resulting in excessive noise levels should be repaired immediately.

e. Correct the specific machine elements causing the noise by a local source approach, rather than by consideration of the entire machine as a noise source. For example, adding noise barriers, noise enclosures, and vibration isolation mountings, lagging to dampen vibrating surfaces, mufflers or silencers for air and gas flows, or reducing air velocity of free jets. These may be considered as a solution for the individual noise-producing elements of the total operation.

f. Separate the noisy elements that need not be an integral part of the basic machine. For example, move pumps, fans and air compressors that service the basic machine.

g. Isolate the vibrating machine parts to reduce noise from vibrating panels or guards.

In addition to engineering changes to machinery and parts, processes can be modified to reduce noise. Specific means of modification include the use of processes that are inherently quieter than the alternatives, for example, mechanical pressing rather than drop forging. Metal-to-metal impact should be avoided or reduced, where possible, and vibration of the surfaces of the machine or the material being processed should be suppressed. This can be achieved, for example, by the choice of suitable materials, by adequate stiffness and damping or by careful dynamic balancing where high-speed rotation is used.

Material handling processes, in particular, can also be modified to ensure that impact and shock during handling and transport are minimized as far as possible. This may be achieved by:

a. Minimizing the fall height onto hard surfaces of items collected by tables and containers;

b. Fixing damping materials to, or stiffening, tables, walls, panels or containers where struck by materials or items during processing;

c. Absorbing shocks through the provision of wear resistant rubber or plastic coatings;

d. Using conveyor belts rather than rollers, which are more likely to rattle; and

e. Controlling the speed of processes to match the desired production rates, thereby obtaining a much smoother workflow and less likelihood of noise generation due to stop-start impact noise.

6.1.2 Path

If it is not possible to change or modify the noise-generating equipment or processes by engineering control of the source, engineering control of the noise transmission path between the source and the recipient, in this case the employee, should be investigated.

Engineering control of the noise transmission path includes isolating the noise-emitting object(s) in an enclosure or placing them in a room or building away from the largest number of employees, and acoustically treating the area to reduce noise to the lowest practicable levels.

Alternatively, it may be desirable to protect the operator(s) instead of enclosing the sound sources. In this case, design of the soundproof room or sound-reducing enclosures should still follow the same principles.

The principles to be observed in carrying out engineering control of the noise transmission path are listed below:

a. Distance is often the cheapest solution, but it may not be effective in reverberant conditions. **Figure 6.1** shows illustration of sound attenuation over distance.

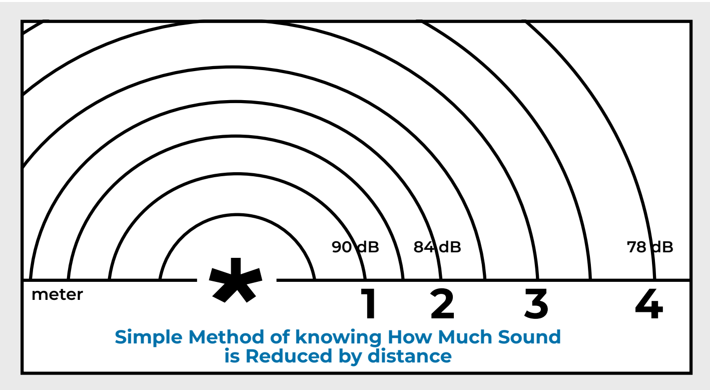


Figure 6.1: Illustration of Sound Attenuation Over Distance.

b. Erect a noise barrier between the noise source and the listener, in some instances a partial barrier can be used to advantage. In cases where either area has a false ceiling, care should be taken to ensure that the dividing wall extends to the true ceiling and that all air gaps in the wall are closed and airtight. **Figure 6.2** shows sound frequency attenuation and shadow zone diagram relative to the height of the barrier.

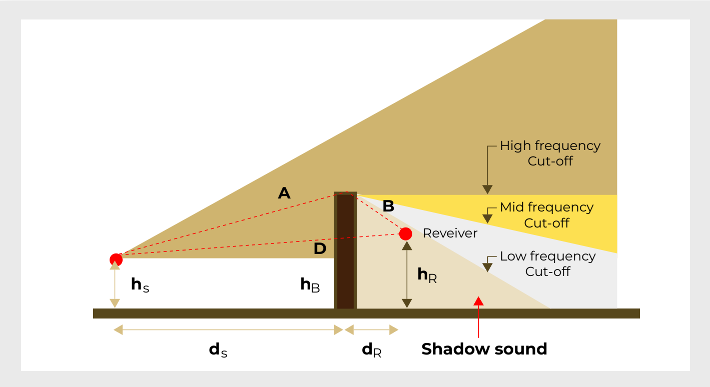


Figure 6.2: Sound Frequency Attenuation and Shadow Zone Diagram Relative to the Height of the Barrier.

c

Once the acoustical barrier is erected, further treatment, such as the addition of absorbing material on surfaces facing the noise source and may be necessary. **Figure 6.3** shows transmission of airborne and structure-borne sound from machinery.

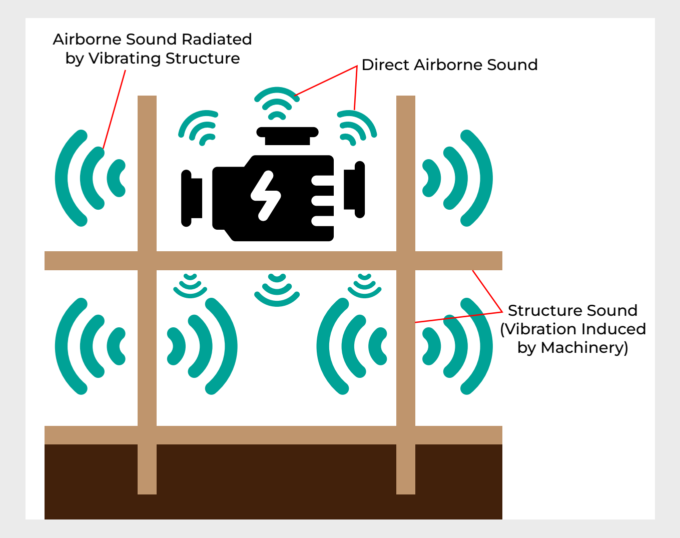


Figure 6.3: Transmission of Airborne and Structure-borne Sound from Machinery.

d

Materials that are good noise barriers, for example, lead, steel, brick, and concrete, are poor absorbers of sound. The denser and heavier the material, the better the noise barrier.

e

Good sound absorbers, for example, certain polyurethane foams, fiberglass, rockwool and thick pile carpet, are very poor barriers to the transmission of sound.

f

Walls and machine enclosures must be designed to minimize resonances which will transmit acoustical energy at the resonant frequency to the protected area. This can be achieved by placing reinforcement or bracing in strategic areas during construction or modification. **Figure 6.4** shows representation of sound wave propagation create resonance.

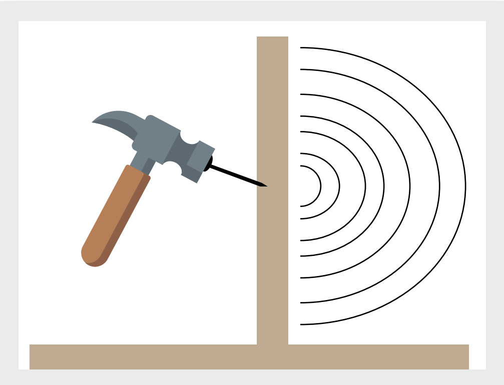


Figure 6.4: Representation of Sound Wave Propagation Create Resonance.

g. Reduce, as far as possible, the reverberation of the room where noise is generated by the introduction of acoustically absorbent material(s). The presence of reverberation in a room shows the need for absorbing material. Excessive reverberation produces unpleasant and noisy conditions, which can interfere with speech communication. **Figure 6.5** shows acoustic reflection in an auditorium create reverberation.

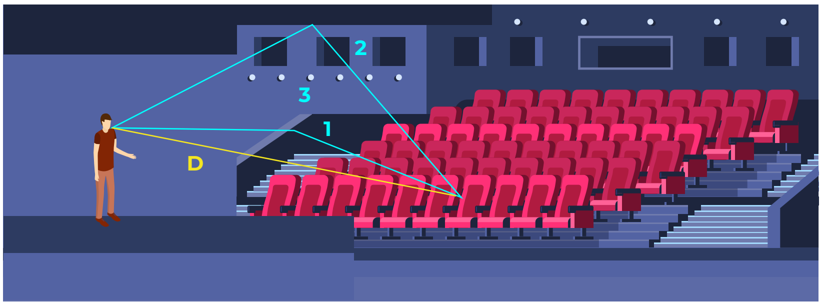


Figure 6.5: Acoustic Reflection in an Auditorium Create Reverberation.

These principles can be utilized in the following way:

a. Using a sound-reducing enclosure which fully encloses the machine(s);
b. Separating the noisy area and the area to be quietened by a sound-reducing partition;
c. Using sound-absorbing material on floors, ceiling and/or walls to reduce the sound level due to reverberation; and
d. Using acoustical silencers in intake and exhaust systems associated with gaseous flow activity, for example, internal combustion engine exhaust systems or air conditioning systems.

6.1.3 Receiver

Receiver control in an industrial situation is generally restricted to providing headsets and/or ear plugs for the exposed workers. It must be emphasized that this is a last resort control and requires close supervision to ensure long term protection of workers' hearing. The main problems lie in ensuring that the devices fit adequately to provide the rated sound attenuation and that the devices are properly worn. Extensive education programs are needed in this regard. Hearing protection is also uncomfortable for a large proportion of the workforce; it can lead to headaches, fungus infections in the ear canal, a higher rate of absenteeism and reduced work efficiency. It is worth remembering that the most protection that a properly fitted headset/earplug combination will provide is 30 dB, due to conduction through the bone structure of the head. In most cases, the noise reduction obtained is much less than this.

Another option which is sometimes practical for receiver control is to enclose personnel in a sound reducing enclosure. This is often the preferred option in facilities where there are many noisy machines, many of which can be operated remotely. In this case, the enclosure design may be calculated to obtain performance and the appropriate material and construction selected after the required noise reduction has been established. Guidance which should be followed during design and construction are:

a. Doors, windows and wall panels should be well sealed at edges;
b. Interior surfaces of enclosure should be covered with sound absorptive material; and
c. All ventilation system should provide with acoustic silencer

6.2 Absorption

Consideration will now be given to determining when it is appropriate to treat surfaces with acoustically absorbing material. The first part of the procedure is to determine whether the reverberant sound field dominates the direct sound field at the point where it is desired to reduce the overall sound pressure level, because treating the reflecting surface with acoustically absorbing material can only affect the reverberant sound field. At locations close to the sound source for example a machine operator position, it is likely that the direct field of the source will dominate, so there may be little point in treating a workplace with sound absorbing material to protect operators from noise level produced by their own machines. However, if an employee is affected by noise produced by other machine some distance away, then treatment may be appropriate.

With absorption, small amounts of sound energy are changed into correspondingly small amounts of heat energy. Suitable materials are usually fibrous, lightweight, and porous. The fibers should be relatively rigid. If a cellular material is used, the cells must intercommunicate. Foams should be reticulated to the proper degree. Examples of absorbent materials are: acoustical ceiling tile, glass fibre and foamed elastomers. Physically, the flow resistance of fibrous materials is the most important characteristic. For optimum results, the flow resistance must usually be increased as the thickness of the absorbent decreases, to maintain peak absorption. Absorbent materials are employed in several applications including muffler linings, wall, ceiling and enclosure linings, wall fill and absorbent baffle construction.

The effectiveness of an acoustically absorbent material is measured by the absorption coefficient. A material with an absorption coefficient of 1 (100%) would "soak up" the entire sound incident on it. Industrially useful acoustically absorbent materials have coefficients above 60% in the frequency range from 500 Hz and up. The absorption coefficient is the ratio of absorbed sound intensity in an actual material to the incident sound intensity and can be expressed as:

$$a = \frac{l_a}{l_i}$$

where;

α = sound absorption coefficient
I_a = sound intensity absorbed (W/m²)
I_i = incident sound intensity (W/m²)

It is relatively simple, then, to estimate the new sound level from the new sound pressure levels. **Table 6.1** shows Sabine Absorption Coefficients of various absorbent materials. Sabine absorption coefficients for material are generally measured in a laboratory using a reverberant test chamber. Procedures and test chamber specifications are described in various standard ASTM C423-17, ISO 354:2003 The material to be tested is placed in a reverberant room and the reverberation time T_{60}, is measured.

Table 6.1: Sabine absorption coefficients of common acoustic materials.

Material*	Frequency (Hz)					
	125	**250**	**500**	**1,000**	**2,000**	**4,000**
Fibrous Glass 4 lb/ft^3, hard backing						
1 inch thick	0.07	0.23	0.48	0.83	0.88	0.80
2 inches thick	0.20	0.55	0.89	0.97	0.83	0.79
4 inches thick	0.39	0.91	0.99	0.97	0.94	0.89
Polyurethane foam (open cell)						
1/4-inch thick	0.05	0.07	0.10	0.20	0.45	0.81
1/2-inch thick	0.05	0.12	0.25	0.57	0.89	0.98
1 inch thick	0.14	0.30	0.63	0.91	0.98	0.91
2 inches thick	0.35	0.51	0.82	0.98	0.97	0.95
Hairfelt						
1/2-inch thick	0.05	0.07	0.29	0.63	0.83	0.87
1 inch thick	0.06	0.31	0.80	0.88	0.87	0.87

* For specific grades, see manufacturer's data; note that the term Noise Reduction Coefficient (NRC) when used is a single-term rating that is the arithmetic average of the absorption coefficients at 250 Hz, 500 Hz, 1000 Hz and 2000 Hz.

Absorbent materials on room surfaces reduce the amount of reverberant sound in a working space and thus reduce the effects of reflected sounds. It is very important to recognize that absorbents materially affect the transmission of sound, thus, the absorbent material should never be used as shields, barriers or enclosure walls. The reduction of reverberant sound pressure levels that could be expected by addition of an absorbent material is given as approximately 10 times the logarithm of the ratio of the room constant obtained after adding the absorbent material, divided by the original room constant.

NR= 10 log (A2/A1)

$$NR = 10 \log \left(\frac{A2}{A1}\right)$$

where;

NR = Noise reduction (NR) a reverberant sound level between two different conditions of room absorption dB
A1 = Total absorption of original room condition, m^2 sabine
A2 = Total absorption after adding the absorbent of room condition, m^2 sabine

The total absorption in a room can be expressed as;

$$A = S1\ a1 + S2\ a2 + \ldots + Sn\ an$$

$$A = \sum Si\ ai$$

where,

A = the absorption of the room (m^2 Sabine)
Si = area of the actual surface (m^2)
αi = absorption coefficient of the actual surface

Example 4:

For a room 8 m long x 6 m wide x 4 m high, determine the noise reduction using the Sabine approach when the absorption coefficient before treatment for the floor is 0.02, for the walls 0.04, and for the ceiling 0.01, and after treatment with an acoustic panel which is attached to the walls using 0.08 absorption coefficient.

Answer:

Area of walls = 2 x [(8 x 4) + (6 x 4)] = 112 m^2
Area of floor = 48 m^2
Area of ceiling = 48 m^2
Total area = 208 m^2

Item	Surface Area, S	Before Treatment		After Treatment	
		α	A1	α	A2
Walls	112	0.04	4.48	0.08	8.96
Floor	48	0.02	0.96	0.02	0.96
Ceiling	48	0.01	0.48	0.01	0.48
Total			5.92		10.40

Noise reduction, NR = 10 log (10.4 / 5.92)
= 2.45 dB

Note that for each doubling in the amount of absorption, you can expect a 3 dB noise reduction in reverberant levels. The first 3 dB reductions are therefore relatively cheap to obtain, you must add twice as much material to obtain a second 3 dB reduction. Note also that the ultimate noise reduction potential would be limited. You would not be able to reduce the sound level to below that which would be obtained if there were confining walls present in the workspace.

Absorbent materials may have special facings. For resistance to grease and water that would clog pores, a thin plastic film covering is often used. Such films, as well as perforates vinyl or sheet metal facings, tend to produce a maximum in the mid frequency absorption coefficient. Absorbents protected by a film still have exposed edges. These may be sealed by a latex paint that anchors itself to the pores of the absorbent and closes the edges. Some thin construction materials, notably plywood, can show increased low-frequency absorption by panel resonance, if not securely fastened down.

6.2.1 Noise Reduction Coefficient

Noise Reduction Coefficient (NRC) is a single number to describe the absorption characteristics of a material. This is particularly useful when a comparison of relative benefits of a number of different materials has to be made quickly. For this purpose, the frequency-average Noise Reduction Coefficient (NRC) has been introduced. It is defined as:

$$NRC = \frac{(\alpha_{250} + \alpha_{500} + \alpha_{1000} + \alpha_{2000})}{4}$$

where;

α_{250}, α_{500}, α_{1000}, and α_{2000} are the sound absorption coefficients at 250 Hz, 500 Hz, 1000 Hz, and 2000 Hz, respectively.

Example 5:

By referring to **Table 6.1**, find the value of noise reduction coefficient (NRC) for 1-inch-thick Polyurethane foam material.

Answer:

$$NRC = (0.30 + 0.63 + 0.91 + 0.98)/4 = 0.705$$

NRC provides the standard rating for how well a material absorbs sound. Different materials have different NRC ratings that range from 0.00 - 1.00. Typically, the NRC rating of a material is viewed as a percentage. For example, an NRC rating of 0.75 means 75% of the sound energy coming in contact with that specific material is absorbed. Simply put, the sound is not reflected back into the room to create noise. A material with an NRC of 0.75 would also be considered 25% reflective. **Table 6.2** shows example of available absorption material with NRC rating as below.

Table 6.2: Noise Reduction Coefficient of common acoustic materials.

Sample	Density (kg/m³)	Thickness (mm)	Noise Reduction Coefficient
Feather Fibre Mat	32	50	0.70
Kenaf	100	60	0.70
Wood Fibre	100	60	0.60
Sheep Wool	40	60	0.70
Coconut	60	50	0.50
Cellulose Fibre Mat	56	50	0.65
Mineral Wool Mat	41	50	0.65

6.3 Insulation

Sound insulation refers to the practice of reducing the transmission of sound between different areas, typically through the use of materials and construction techniques that block or absorb sound waves. Effective sound insulation is crucial in industrial environment settings to create functional acoustic conditions. By incorporating sound-insulating materials like dense barriers, acoustic panels, and specialized windows and doors, it is possible to minimize the intrusion of external noise and prevent the escape of internal noise.

6.3.1 Sound Transmission Loss

The sound isolation properties of materials are stated in term of sound transmission loss (STL). As with absorption, the concept of energy flow is used, here it is the energy transmitted through the material, relative to that flowing toward it. Transmission loss is ideally increased with frequency at the rate of about 5 to 6 dB per doubling of frequency.

The standard measurement for determining transmission loss is made in accordance with ASTM E90:99 - Standard Test Method For Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions and Elements or ISO 140-l:1997 - Acoustics - Measurement of Sound Insulation In Buildings and of Building Elements - Part 1: Requirements For Laboratory Test Facilities with Suppressed Flanking Transmission and ISO 140-3:1995 - Acoustics - Measurement of Sound Insulation in Buildings and of Building Elements - Part 3: Laboratory Measurements of Airborne Sound Insulation of Building Elements.

$$STL - 10\log\left(\frac{1}{\tau}\right)$$

where;

STL : Sound Transmission Loss in decibel (dB)
τ : Transmission loss factor

$$\tau = \frac{I_T}{I_i}$$

where;

transmitted sound intensity (*I Transmitted*)
incident sound intensity (*I Incident*)

As a result of the search for a single number to indicate the average full transmission loss, the concept of sound transmission class (STC) was developed. It is useful specifically in assessing the degree to which intelligible speech is prevented from being transmitted through a wall. Use the STC with caution in industrial work, however, because the noise spectrum can be much different from that of speech. Therefore, the transmission loss in each octave band for the proper application of isolating materials is illustrated in **Table 6.3**.

Table 6.3: Transmission Loss for Typical Building Material.

Material *	Frequency (Hz)					
	125	250	500	1000	2000	4000
Brick, 4”	30	36	37	37	37	43
Block, 7 5/8” hollow cinder	33	33	33	39	45	51
Block, 6” concrete lightweight painted	38	36	40	45	50	56
Curtains, lead/vinyl 1.5 lb/ft^2	22	23	25	31	35	42
Door 2 5/8” hardwood	26	33	40	43	48	51
Fiber Tile, filled mineral 5/8”	30	32	39	43	53	60
Glass plate, 1/4"	25	29	33	36	26	35
Glass laminated, 1/2"	23	31	38	40	47	52
Panels, perforated metal with mineral 4”	28	34	40	48	56	62
Plywood, 1/4", 0.7 lb/ft^2	17	15	20	24	28	27
Plywood, 3/4", 2 lb/ft^2	24	22	27	28	25	27
Steel, 18 gauge, 2 lb/ft^2	15	19	31	32	35	48
Steel, 16 gauge, 2.5 lb/ft^2	21	30	34	37	40	47
Sheet metal laminate, 2 lb/ft^2	15	25	28	32	39	42

Note: *For specific grades, see manufacturer's data.

A wall with a rated Sound Transmission Class (STC) is designed to provide a specific level of sound insulation, which can significantly affect the hearing quality within a building. STC ratings are a measure of how well a building partition attenuates airborne sound. For example, a wall with an STC rating of 30 allows most normal speech to be heard and understood through it, while a wall with an STC rating of 50 will block most loud speech, rendering it barely audible. Higher STC ratings, such as 60, indicate superior sound insulation, where even very loud sounds are not easily heard. The choice of STC rating is crucial for creating spaces that meet specific acoustic requirements and provide an optimal auditory environment, as illustrated in **Table 6.4**.

Table 6.4: Typical hearing quality for a wall of rated sound transmission class (STC).

STC	Partition type
33	Single layer of 1/2” drywall on each side, wood studs, no insulation (typical interior wall)
39	Single layer of 1/2” drywall on each side, wood studs, fiberglass insulation [2]
44	4” Hollow CMU (Concreate Masonry Unit) [3]
45	Double layer of 1/2” drywall on each side, wood studs, batt insulation in wall
46	Single layer of 1/2” drywall, glued to 6” lightweight concrete block wall, painted both sides
46	6” Hollow CMU (Concreate Masonry Unit) [3]
48	8” Hollow CMU (Concreate Masonry Unit) [3]
50	10” Hollow CMU (Concreate Masonry Unit) [3]
52	8” Hollow CMU (Concreate Masonry Unit) with 2” Z-Bars and 1/2” Drywall on each side[4]
54	Single layer of 1/2” drywall, glued to 8” dense concrete block wall, painted both sides
54	8” Hollow CMU (Concreate Masonry Unit) with 1 1/2” Wood Furring, 1 1/2” Fiberglass Insulation and 1/2” Drywall on each side[4]
55	Double layer of 1/2” drywall on each side, on staggered wood studs wall, batt insulation in wall
59	Double layer of 1/2” drywall on each side, on wood studs wall, resilient channerls on one side, batt insulation
63	Double layer of 1/2” drywall on each side, on double wood/metal stud walls (spaced 1” apart), double batt insulation
64	8” Hollow CMU (Concreate Masonry Unit) with 3” Steel Studs, Fiberglass Insulation and 1/2” Drywall on each side[4]
72	8” concrete block wall, painted, with 1/2” drywall on independent steel stud walls, each side, insulation in cavities

STC partition ratings taken from: “Noise Control in Buildings: A Practical Guide for Architect and Engineers”, Cyril M. Harris, 1994

6.3.2 Enclosure

The use of an enclosure for noise control will produce a reverberant sound field within it, in addition to the existing direct sound field of the source. Both the reverberant and direct fields will contribute to the sound radiated by the enclosure walls as well as the sound field within the enclosure.

The sound field immediately outside of an enclosure consists of two components. One component is due to the internal reverberant field and the other is due to the direct field of the source.

Noise from certain sources may be most effectively reduced by acoustically enclosing the source. Sound transmission loss characteristics of a composite partition composed of panels of different materials is a function of the total percentage of area occupied by each material and its transmission loss factor. Transmission loss characteristics of various commonly used materials for enclosures are available in most noise control handbooks.

Leaks or open areas in the enclosure create significant reduction in the overall transmission loss of a panel, as illustrated, in **Figure 6.6**. For example, if an enclosure with a 60 dB expected noise transmission loss or noise reduction has an opening that represents 1.0% of the total area, the effective net noise transmission loss is reduced to 20 dB.

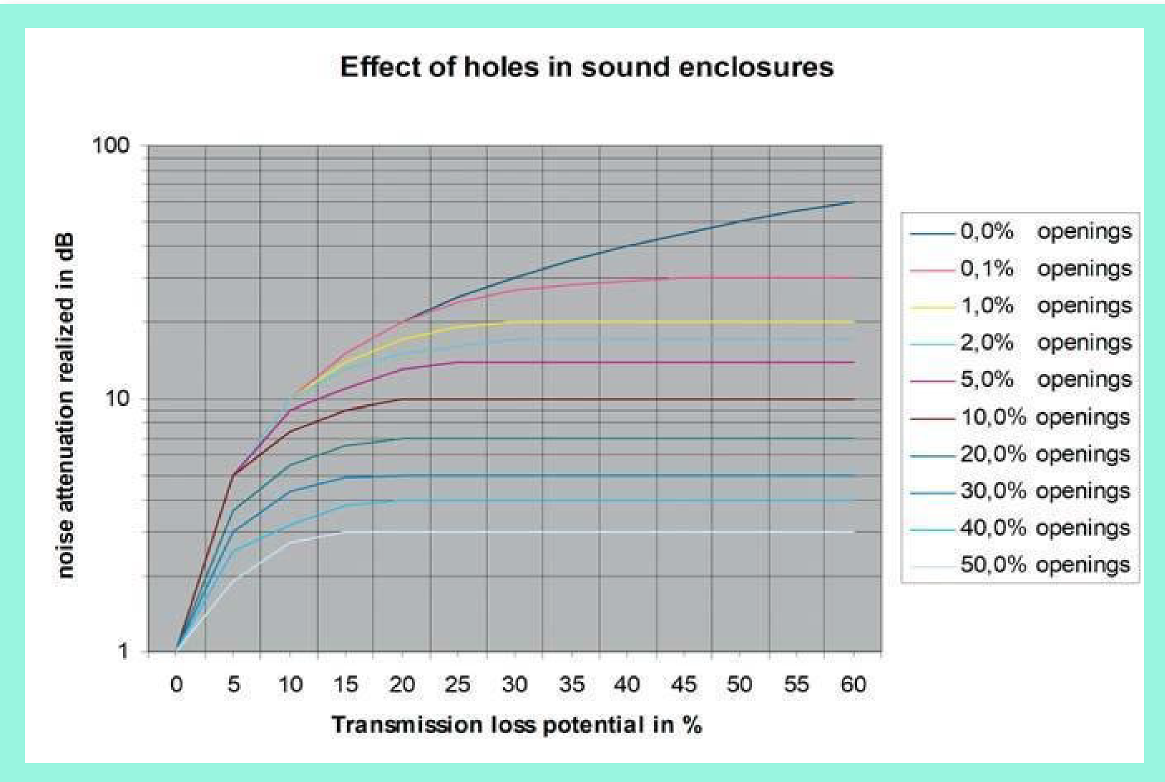


Figure 6.6: Effects of opening holes area on potential noise reduction of an enclosure.

To contain equipment requiring cooling air, an enclosure must have openings for air intake and discharge. Such openings must be acoustically treated, as shown in the example of **Figure 6.7**. Typical noise reduction achieved for the same machines with different enclosure designs shows different potential sound pressure level reductions for each 1/1 octave band frequency.

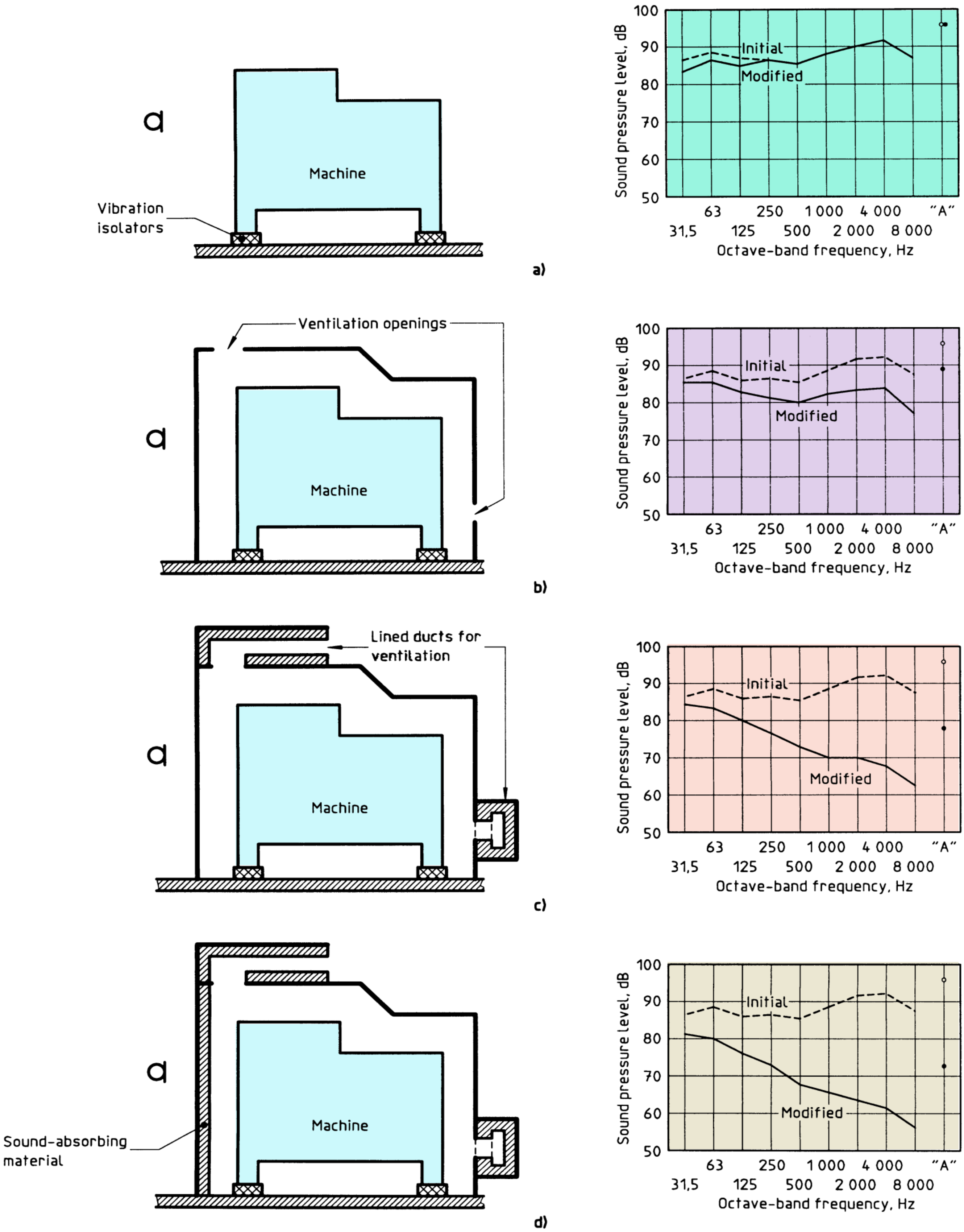


Figure 6.7: Typical noise reduction achieved for different machines and enclosures.

For maintenance purposes disassembly of an enclosure should be given appropriate design consideration. Lift-out panels, panels mounted on overhead tracks, or merely large, well-positioned enclosure doors with chain hoist accessibility can often solve equipment removal and maintenance problems.

Normally, thinner panels may be employed to absorb high frequency noises. However, for low frequencies, thicker panels with a heavy type material partition is recommended. Panels should also be mounted with vibration isolation to avoid the transmission of low frequency to the floor or attachment structures.

6.3.3 Acoustic Lagging

Radiation from the walls of pipes or air conditioning ducts is a common source of noise. The excitation usually arises from disturbed flow through valves or dampers, in which case it is preferable to reduce the excitation by treatment or modification of the source. However, as treatment of the source is not always possible, an alternative is to acoustically treat the walls of the pipe or duct to reduce the transmitted noise.

The effect of wrapping a pipe with a layer of porous absorbent material may be calculated by taking into account sound energy loss due to reflection at the porous material surfaces and loss due to transmission through the material.

Acoustical lagging of noise sources, principally pipelines, valves, and ducts, consists of encapsulating the source with treatments that provide a combination of sound barrier and sound absorption mechanisms to obtain the maximum noise reduction. The optimum lagging design is dependent upon the spectral content of the radiated noise as well as the level of noise reduction required. All designs should avoid any mechanical coupling between pipe or duct surface and outer shell treatment and should have a layer of resilient absorptive material between the pipe, duct, or valve surface and the outer shell treatment. Increased thickness of material or a heavier composite material is required for low frequency performance. For thin-shelled pipes or ducts, a vibration damping material with adhesive backing should be directly applied on the surface. **Figure 6.8** illustrates typical acoustical lagging designs. For a 20 dB reduction, 2-inch layers of 6 lb/ft^3 absorptive materials with at least a 1.0 lb/ft^3 septum interposed between layers is an average requirement.

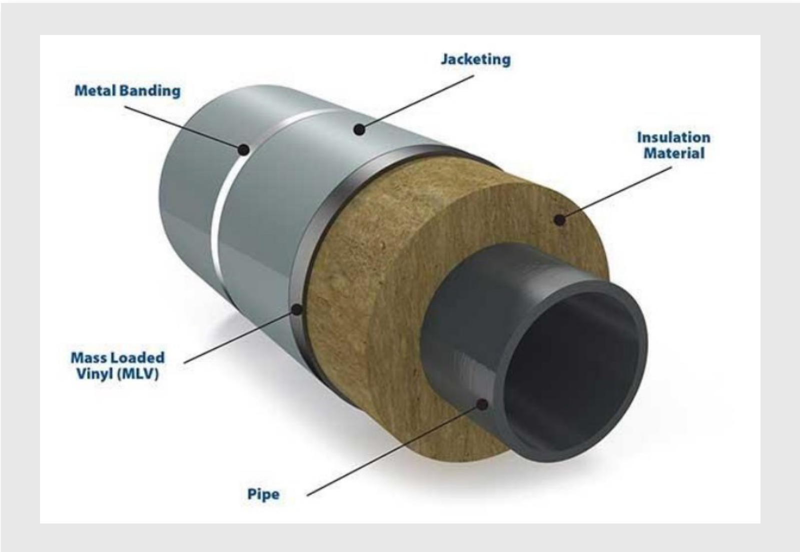


Figure 6.8: Typical acoustical lagging designs for noise radiating pipelines.

6.4 Distance

6.4.1 Drop Height

Mechanical and material handling devices commonly produce impact noise. This type of noise can be reduced by:

1. Reducing the dropping height of goods collected in boxes or bins, as illustrated in **Figure 6.9**.

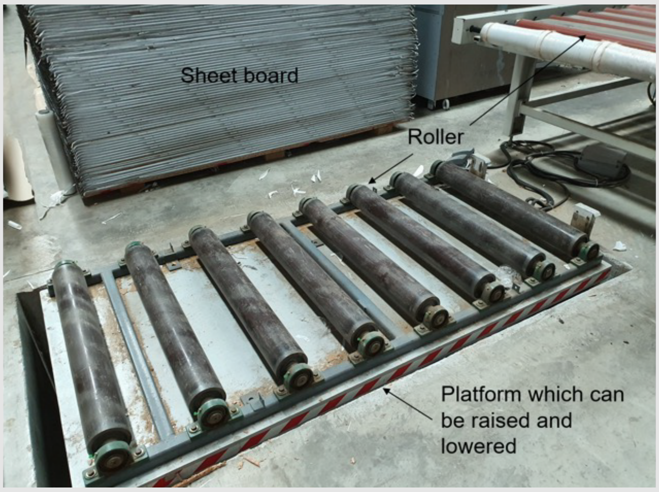


Figure 6.9: Reducing drop height using adjustable platform.

2. Using soft rubber or plastic to receive and absorb hard impacts, such as were 2 panels are likely to be struck by materials during processing, as illustrated in **Figure 6.10**.

Figure 6.10: Reducing drop height and using rubber flaps to slow down fall speed and absorb impact.

3 Increasing the rigidity of containers receiving impact goods and adding damping material - especially to large surfaces.

4 Regulating the speed or cycle time of conveyors to prevent collisions and excessive noise.

6.4.2 Turbulence Noise

Smooth laminar flow in ducts or pipes does not generate noise. Fluid noise is due to turbulence. The more turbulent flow, the greater would be the noise. Vapour bubbles can be created by abrupt changes in the flow of fluids. Providing gradual transition in a cross-sectional area reduces the likelihood of these bubbles forming, as illustrated in **Figure 6.11 (a) & (b)**.

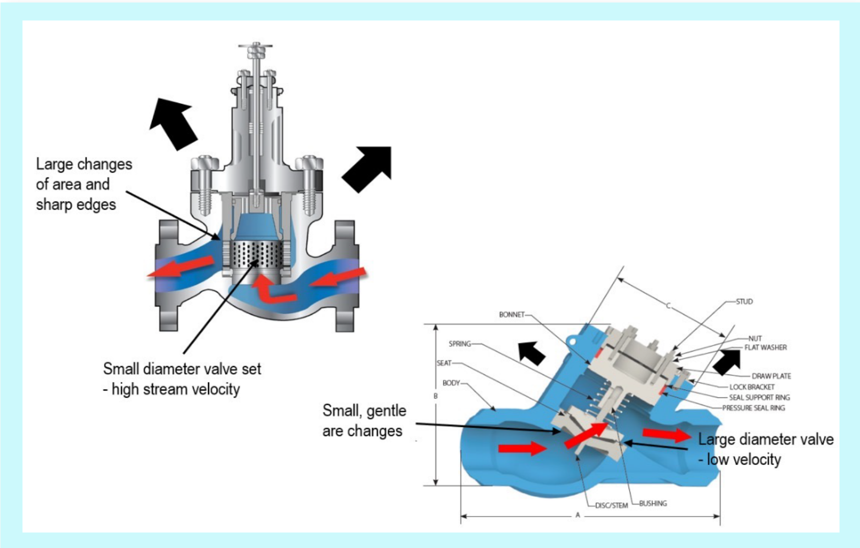


Figure 6.11 (a): Reducing fluid turbulence by straightening flow pathways.

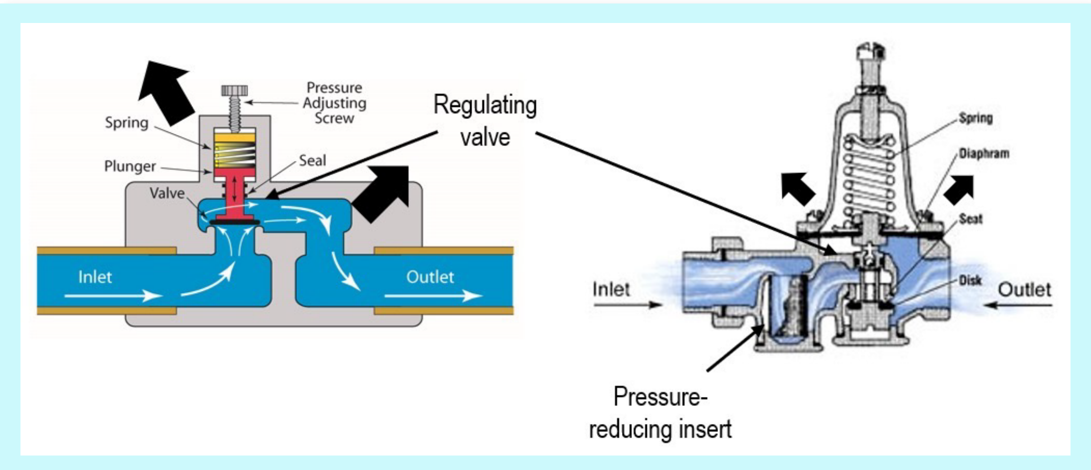


Figure 6.11 (b): Preventing vapour bubble implosion (cavitation) by reducing pressure in several smaller steps.

Turbulence at the walls of ducts or pipes is always present. To reduce noise, interior walls should be smooth, free of protrusions at joints, and sharp bends at T junctions and Y junctions should be avoided. Turning vanes can be placed inside ductwork when construction methods utilize sharp bends. Straightening vanes can be used to smoothen the flow downstream of any change in direction, diameter, or branching, as illustrated in **Figure 6.12**.

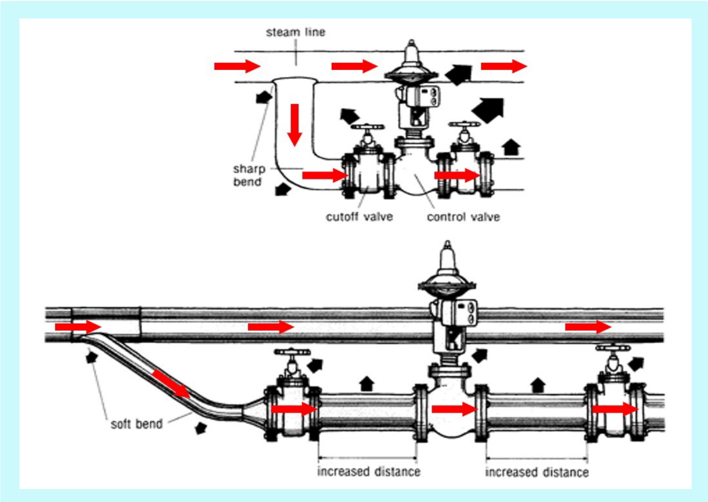


Figure 6.12: Reducing fluid turbulence by straightening bends and transitioning diameter changes in fluid flow.

Fans produce large amounts of turbulence, which in turn produces noise. Since the sound power generated by a fan varies with the fifth power of its rotational speed, the most cost-effective method of fan noise control is to reduce this speed wherever possible.

Wherever possible, when purchasing new fans, choose quieter designs. For instance, backward-curved blades on squirrel-cage fans are quieter than straight blades or forward-curved blades. Fans mounted inside ductwork create significant noise, especially when mounted in regions where a great deal of turbulence is present. In-line duct fans should be mounted in low-turbulence regions of ductwork **(Figure 6.13)**.

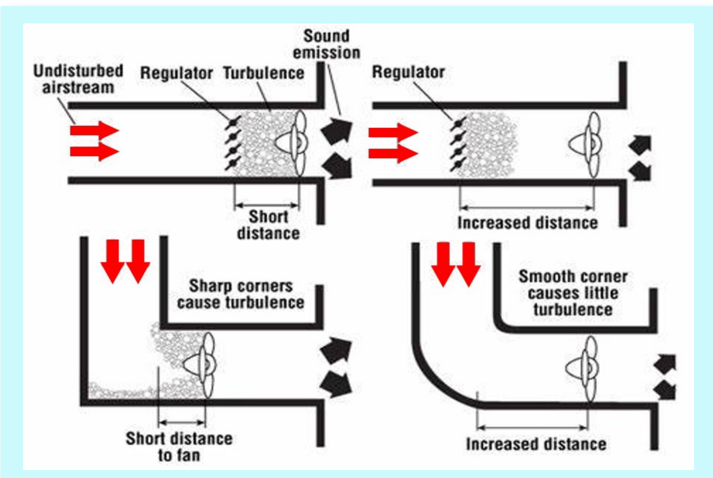


Figure 6.13: shows an airflow effect of in-line duct fans should be mounted in low-turbulence regions of ductwork.

6.5 Silencer

Silencers or mufflers can be classified into two fundamental types which are absorptive (sometimes also called dissipative) or reactive. Both types are made to reduce noise while permitting the flow of air or gas. Absorptive silencers contain porous or fibrous material and use absorption to reduce noise. The basic mechanism for reactive silencers is expansion or reflection of sound waves, leading to noise cancellation.

Silencer or mufflers are commonly used to reduce noise associated with internal combustion engine exhaust, high pressure gas or steam vents, compressors, and fans. These examples lead to the conclusion that a silencer device allows the passage of fluid while at the same time restricting the free passage of sound. Silencer might also be used where direct access to the interior of the enclosure is required, but through which no steady flow of gas is necessarily to be maintained.

Silencers or mufflers can be considered simply as a duct or pipe that has been acoustically treated or shaped specifically to reduce noise transmission in the contained medium. The noise may originate from a machine source or it may be flow generated. Sources include blow downs to atmosphere, draft fans, vacuum pumps, pelletizers, chillers, blowers, compressors, piping systems, pressure reduction valves, turbines, reciprocating engines, and other equipment.

The characteristics of gas flow noise vary widely. Therefore, a thorough analysis of the spectral content of gas flow noise is an important first step in the choice and application of a silencing mechanism.

6.5.1 Reactive Silencer

The simplest form of a reactive silencer is a single expansion chamber. As the air enters and leaves the chamber, the expansion and contraction in pressure cause reflection of sound waves. The reflected wave added to the incoming sound wave results in destructive interference, leading to noise reduction.

This equation can be used to calculate the length needed for a reactive silencer:

$$L = \frac{n\lambda}{4} \quad where, n = 1, 3, 5 \ldots$$

where;

L = is the length of the silencer
λ = is the wavelength of the tone
n = is a positive odd integer, representing the harmonic series for this type of resonator. The 1/4 fraction indicates that the silencer length is 1/4 of the wavelength of the fundamental frequency (where n=1).

The acoustical properties of a reactive silencer are governed primarily by its internal configuration and the reduction of flow velocity by providing an expansion chamber. Reactive silencers are designed to take advantage of sound reflections from abrupt changes in shape and resonances of added branches or cavities to a pipe or duct. These reactive mechanisms obstruct the acoustical passage by impedance mismatch of acoustic energy flow within the pipe or duct. Reactive silencers are most effective at low frequency and limited spectral bandwidth applications.

6.5.2 Absorptive Or Dissipative Silencers

The simplest form of absorptive silencer is a lined duct. Generally, long sections of ducts are lined with absorptive material, but lining is particularly effective along duct bends. Typically, 2 - 5 cm acoustical grade fibrous glass is used.

Another form of absorptive silencer comprises parallel baffles. Good design includes aerodynamically streamlined entrance and exit ends with perforated spaces filled with highly absorbent acoustic materials. The first few feet of length are highly absorbent, so the attenuation is not linear with length. Thick absorbent material with wide spaces between absorbers is effective for low frequencies, while thin material with narrow spaces is effective for higher frequencies. This design should be considered for the entrant and exhaust air whenever sources are enclosed, as shown in **Figure 6.14**.

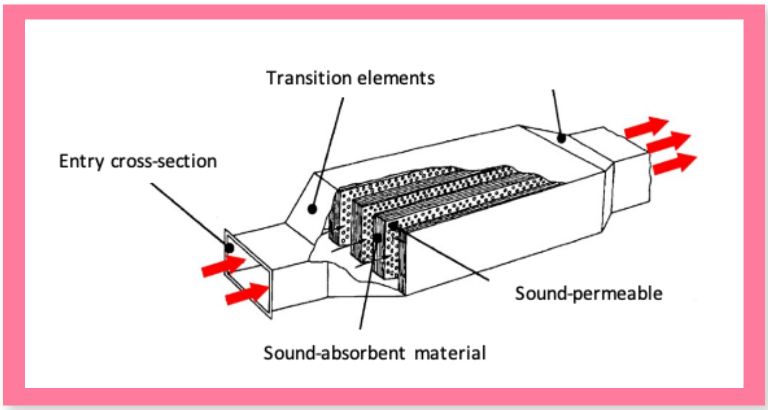


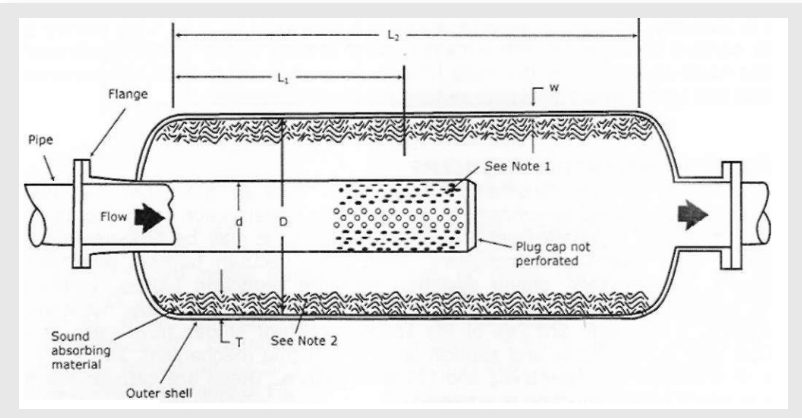
Figure 6.14: Absorbent splitter silencer sized for dominant frequencies.

Its acoustical properties are governed primarily by the presence of sound absorbing material that dissipates acoustic energy. Materials such as rock wool, fiberglass, and felt, when deployed within a duct, form a dissipative silencer. Maximum absorption of such materials usually occurs at the higher frequencies, yet dissipative silencers usually have a relatively wideband noise reduction capability. A potentially undesirable feature of this type of silencer is that if improperly designed, bits of the absorptive material may be drawn into the gas stream and eventually degrade silencer performance.

The distinction between reactive and dissipative silencers is conceptual. In practice, all silencers achieve some noise reduction by both reactive and dissipative means. Certain silencers, however, are designed as combination reactive-dissipative devices for a specialized application. In all silencer employment considerations, the back pressure increase in the duct must be considered and its effect on the total plant equipment design specifications.

6.5.3 Piping Noise Silencers

Pulsating flow created by the intake and discharge of reciprocating compressors and pumps is a frequent and serious source of noise and vibration. Devices called "snubbers" or "pulse traps" are used to buffer this pulsating flow by providing both an expansion chamber and dissipative elements in the associated piping system. The performance of such installed devices is a function of the system. Therefore, an analysis of a complete piping system should be performed to ensure operational compatibility and acoustical performance. **Figure 6.15** shows the general design feature of in-line silencers.



Note 1

Hole diameter should be spaced as closely as possible and yield a total open area greater than twice the piping cross section area.

Note 2

Sixteen (16) gauge (or heavier) non-corrosive steel with at least 50% perforated open area and backed by 16 x 16 or 18 x 14 non-corrosive wire mesh.

Figure 6.15: General design features of an in-line silencer.

Primary design characteristics of an in-line silencer as following:

- **a** The most important characteristic is the absorptive material should cover as much area as practical.
- **b** Limitation of turbulence re-excitation as the gas leaves the silencer that can be achieved by avoiding sharp edges or area restrictions.
- **c** Suppression or avoidance of resonances.
- **d** Wide band attenuation.
- **e** Construction suitable to avoid external radiation of sound.
- **f** No serious pressure loss in net flow.
- **g** Construction suitable to withstand operating pressures.
- **h** Simplicity of installation.

6.5.4 Silencer And Its Application

1 Exhaust Silencer

Exhaust silencers come in standard, and custom made to fit various types of internal combustion engines. Exhaust silencer is installed after the engine exhaust outlet with manifold or pipe connection in order to reduce noise level emitted by high flow rate exhaust gas. Commonly known as a muffler and designed with multiple chambers and perforated tubes. Easy to clean since it does not contain fibrous materials and it creates negligible pressure loss. Small in size which makes it suitable for engines and exhaust systems. Commonly used for fast speed machinery, such as generators and blowers. Best used for machinery that produces pure tones and noises that range from low to medium frequency. **Figure 6.16** shows the example of exhaust silencer.

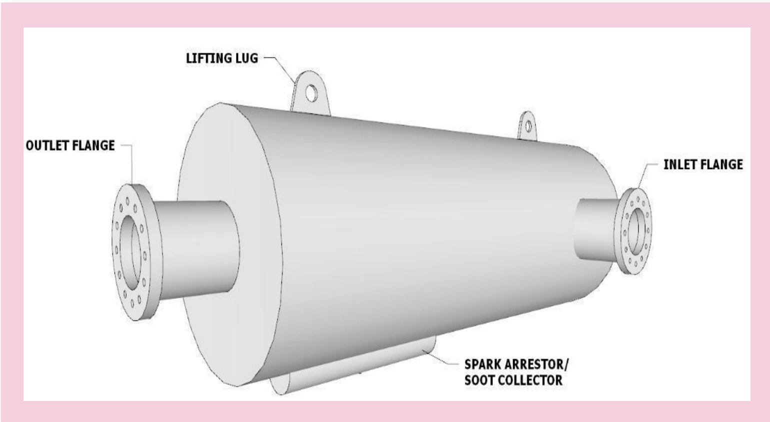


Figure 6.16: Example of an exhaust silencer.

2 Duct Silencer

Duct silencers are designed to efficiently reduce noise for generator sets, compressors, pumps, blowers and all systems that require intake and discharge for this equipment to operate. A duct silencer needs to be installed in the plant room and as close to the noise source as possible. Normally for single equipment, two units of duct silencers - intake and discharge is required to effectively reduce the noise to a desired level. **Figure 6.17** shows an example of duct silencer

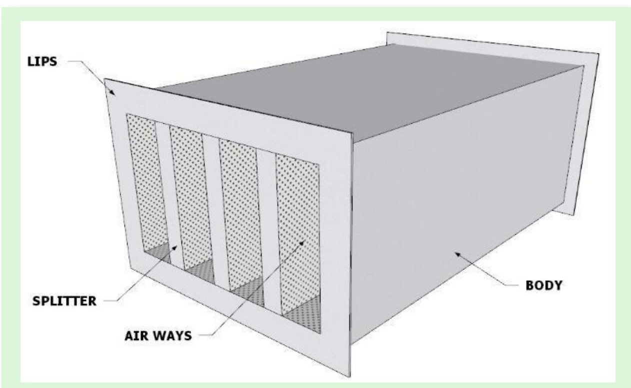


Figure 6.17: Example of a duct silencer.

3 Steam Vent Silencer

Steam vent silencer reduces the noise produced by the expansion of gas or steam from elevated pressures to atmospheric pressure. These absorptive silencers are used to suppress noise generated by high velocity gas streams such as steam vents, safety relief valve outlets, system blow downs and purge outlets. Each vent silencer is designed to attenuate the noise level to the required sound pressure level criteria at a given distance from the silencer. Typically used in oil and gas processing, chemical processing and heat recovery steam generators or boilers. **Figure 6.18** shows an example of vent silencer.

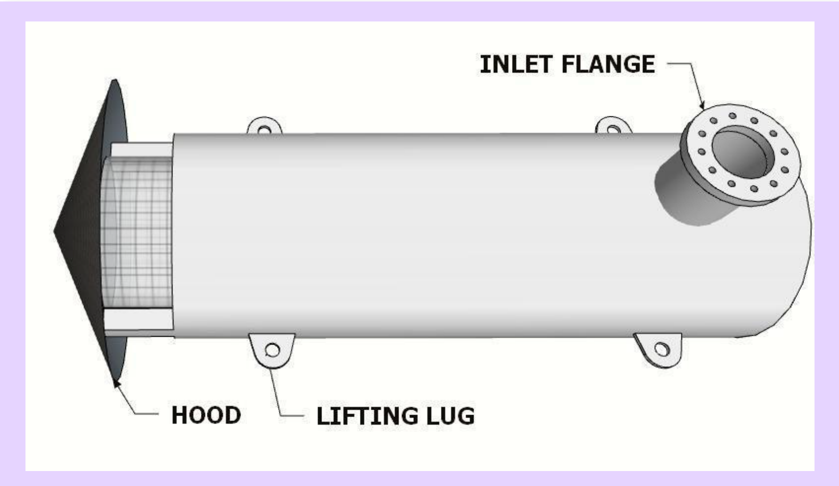


Figure 6.18: Example of a vent silencer.

4 Absorptive Silencer

Absorptive silencers primarily absorb sound energy by using fibrous and porous packing material to attenuate sound, rather than reflecting it. As the sound waves travel through the insulative materials the sound energy is partially transformed to heat energy, which is then dispersed through the air. The thickness of the acoustical lining is often dependent on the predominant frequency of the noise. This kind of silencer is effective for high noise frequencies between a range of 500 to 8000 Hz. **Figure 6.19** shows an example of absorptive silencer.

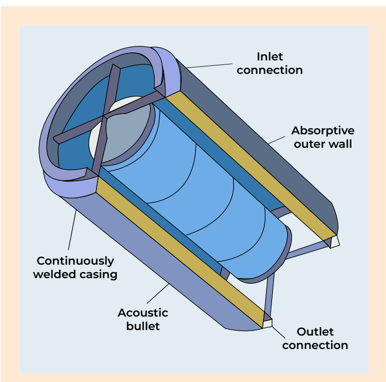


Figure 6.19: Absorptive Silencer.

5 Combination Silencers

Combination silencer incorporates elements from several different types of industrial silencers for both scattering / reflecting and absorption which are able to create a custom noise solution. This kind of silencer is tuned to target specific problematic tonal noise frequencies and can reduce tonal spike, which also reduces the overall noise created by the fan exhaust. Installed in the gas path of a fan and are often used in conjunction with an absorptive silencer to provide additional attenuation. Figure 6.20 shows an example of combination silencer.

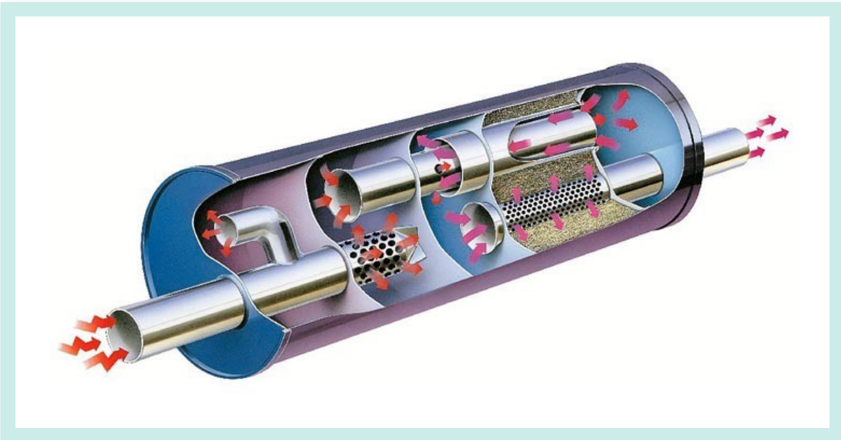


Figure 6.20: Example of a combination silencer.

6.6 Vibration Isolation

The application of vibration isolation is generally required to silence structure-borne noise. A relatively small vibrating machine, pipe, or other mechanism, when closely coupled to a floor or panel and then radiating the vibration acoustically, can often produce objectionable noise levels. Structure-borne noise refers to the transmission through structures of mechanical vibrations that produce airborne noise when a panel or other structure is set into motion and radiates sound.

Figure 6.21 (a) displays a motor connected directly to the floor. Because the rotating parts of the motor can never in practice be perfectly balanced, the rotation results in a vertical component of an out of balance force shown as F_0. This vibratory force is efficiently transmitted to the floor shown as F_t, where it can cause structure borne noise transmission.

Figure 6.21 (b) displays the same motor had been placed on a spring called vibration isolators. This has the effect of creating a mass-spring system, with a natural frequency in which the motor acts as the mass and the isolator as springs.

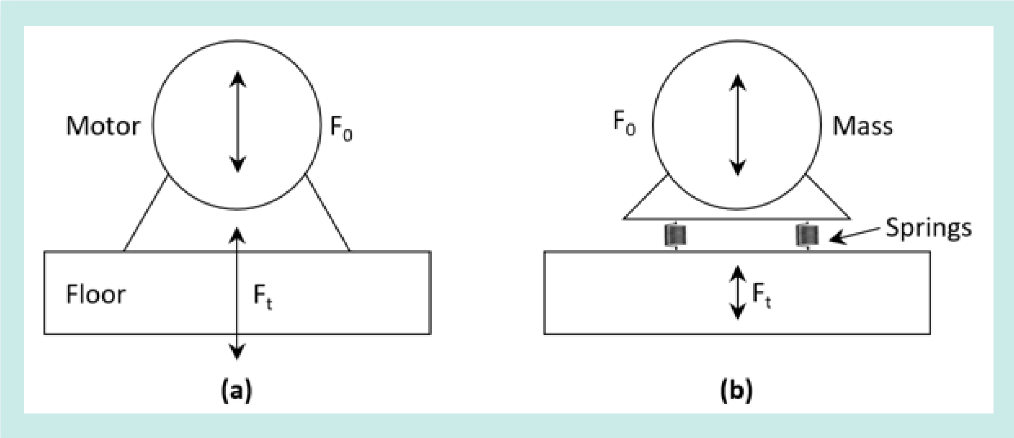


Figure 6.21: The fundamental of vibration isolation.

The transmission of vibratory motions or forces, T_r from one structure to another may be reduced by interposing relatively flexible isolating elements between the two structures. This is called vibration isolation and when properly designed, the vibration amplitude of the driven structure is largely controlled by its inertia. An important design consideration is the resonance frequency of the isolated structure on its vibration isolation mount. To achieve isolation, the vibratory motions or forces transmitted, F_t to the foundation should be less than excitation force, F_0. Therefore, the equation of vibration transmissibility, T_r as below:

$$T_r = \frac{F_t}{F_0} = \frac{force\ transmitted\ to\ the\ floor}{force\ at\ the\ vibrating\ machine}$$

where;

$T_r > 1$ means amplification of vibration across the isolator (amplification region)
$T_r < 1$ means reduction of vibration (isolation region)

Resonance causes large displacement, strains and stresses induced failure of the system. Often, the operational frequency, *f* cannot be controlled because it is imposed by the functional requirements of the system. Hence, shall control natural frequency by varying mass, *m* with kg unit or stiffness, *k* with N/m unit to avoid resonance. Practically, mass cannot be changed easily. Therefore, change stiffness, *k* by altering the material or number and location of support points. In a real situation, the natural frequency of a system can be determined by using free vibration modal analysis.

Equation of natural frequency, f_n with Hz unit as below:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where;

k : stiffness
m : mass

Forcing frequency, f defined as the frequency of an oscillating force applied to a system. When the forcing frequency is well above the natural frequency, thus a vibrating system reduces to low values. In best practice, the value of forcing frequency should be at least three times the value of natural frequency ($f/f_n > 3$) in order to achieve a significant amount of isolation.

The equation of forcing frequency, f with Hz unit as below:

$$f = \frac{Revolution\ Per\ Minute}{60}$$

For example, the motor is rotating at a speed of 3600 RPM (revolutions per minute). This corresponds to 3600/60 = 60 revolutions per second and so results in a vibratory driving force with a forcing frequency, f of 60 Hz. In order that the mass-spring system is driven well above resonance, using the rule of thumb, the natural frequency f_n of the system should be arranged to be no more than 60/3 = 20 rad/s. This is arranged by selecting the stiffness of the spring in conjunction with the mass of the motor using the natural frequency equation.

Vibration suppression or reduction is generally accomplished by the installation of vibration mounts that combine the properties of resilience and vibration damping to provide two fundamental mechanisms for control:

1. Dissipation and reduction of vibrational energy generated within the system by conversion of that energy into heat.

2. Mechanical decoupling or removing the vibration paths of the system from its mounting structure and floor.

The process for selecting a particular vibration isolation or damping design is described in detail in the technical data sheet supplied by manufacturers of such vibration isolation devices. Vibration isolation is a procedure by which the undesirable effects of vibration are reduced. It involves the insertion of an isolator between vibrating mass and the source of vibration.

Vibration can be summed up into a single number and compared to alarm limits. Vibration meter will give a single value of acceleration, velocity, or displacement unit. These readings are an overall level reading – the RMS value. Over time the level may trend upwards if faults develop. **Figure 6.22** is an example of vibration meter.

Figure 6.22: Example of vibration meter.

To determine the effectiveness of vibration isolators placed in between machine structure and rigid foundation, the vibration measurement shall be measured at rotating machine, F_o and foundation base, F_t location. During onsite measurement, the vibration meter will display velocity, v values with mm/s unit at both machine structure and rigid foundation surface. The transmissibility, T_r value can be measured by calculating ratio of F_t/F_o.

Basic equation for calculating transmissibility vibration isolation undamped system at rigid foundation as below:

$$T_r = \frac{1}{(r^2 - 1)}$$

where;

r : ratio of forcing frequency to natural frequency of the system (f/f_n)

Following **Figure 6.23** shows the graph variation of transmissibility, T_r with frequency ratio, *r*. The graph illustrates typical features of the response of a mechanical system mounted on viscous dampers. The vertical axis represents the transmissibility of the mounting *(T_r)* which is the ratio of transmitted force, F_t to excitation force, F_O. Meanwhile, the horizontal axis represents the frequency ratio, which is the ratio of *ω* to *ωn* of the system. The magnitude of transmissibility in the graph at the ratio of *ω/ωn* = 1 approach T_r equal to infinity, which resulted in resonance. Meanwhile, the horizontal axis represents the frequency ratio, which is the ratio of *ω* to *ωn* of the system. The magnitude of transmissibility in the graph at the ratio of *ω/ωn* = 1 approaches T_r = infinity which resulted resonance to happen.

Note that if the frequency ratio *ω/ωn* is less than $\sqrt{2}$, amplification of the transmitted force will result, and the mounts will deteriorate. The ξ symbol define as damping ratio consist of ξ value from 0 to 1 which indicates an underdamped vibrating system. In summary, transmissibility is reduced when the forcing frequency, *ω* exceeds the natural frequency, ωn in the isolation region. The relationship between frequency symbol of ω and f is *ω=2πf*.

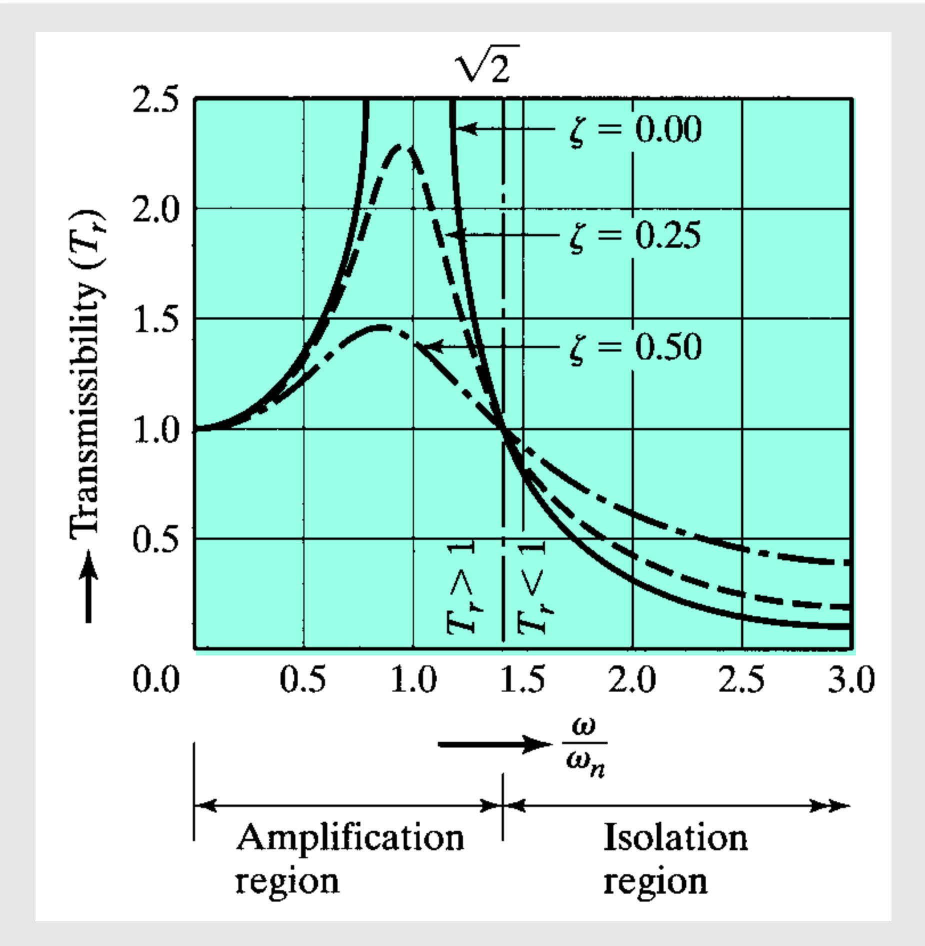


Figure 6.23: Typical vibration response of a single degree of freedom mechanical system mounted on viscous vibration isolators.

Coil springs are another form of vibration isolation. Coil springs are employed primarily for the isolation of vibrating motion that have a low forcing frequency and where elastomeric mounting pads are not very effective. Coil springs are a transmission path for high-frequency vibrations and have large static deflections, therefore, design precautions must be taken in the proper selection of spring-type vibration isolation equipment mounts.

A simple vibration isolation selection design procedure:

1. Determine the lowest possible driving frequency, f. For a rotating machine, where vibration is produced by out of balance forces, where RPM is the lowest possible rotational speed in revolution per minute.

2. Determine the required isolator performance such as the amount of vibration required. This amounts to determining the reduction required transmissibility, T_r .

3. From the equation or graphs of T_r against $\omega/\omega n$, knowing f (Step 1) and $\omega/\omega n$ (Step 3), calculate the required natural frequency, f_n of the machine or isolator system.

4. Knowing f_n and m (mass of machine), and number of isolators, determine the required stiffness, k of the isolators.

5. Select springs from the manufacturer technical data sheet with suitable stiffness.

Example 6:

If the total mass of the fan and its baseplate is 500 kg and the fan rotates at 1200 RPM, calculate the total stiffness required by the springs in order to reduce the vibratory force transmitted to the floor by transmissibility of 0.3. Assume that the damping ratio is zero.

Answer:

Determine driving frequency, ;*f*

$f = \frac{1200\ RPM}{60};$
$f = 20\ Hz$

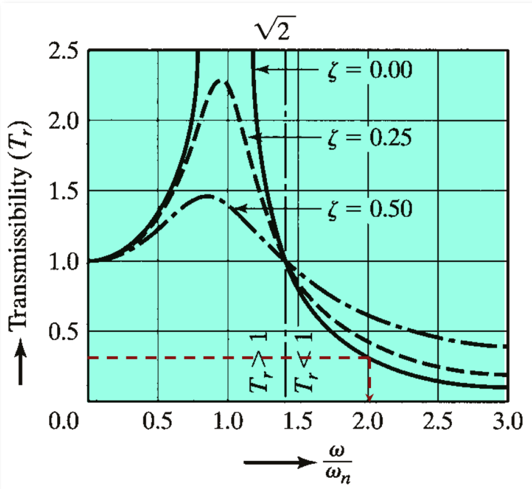
Determine driving frequency, *r*;

1. Use formula $T_r = \frac{1}{(r^2-1)};$

$0.3 = \frac{1}{(r^2-1)};$

$r = 2$

2. Use vibration response graph; ***r* = 2**



Determine natural frequency, f_n;

$r = \frac{f}{f_n};$

$f_n = \frac{20}{2}$

$f_n = 10\ Hz$

Determine required stiffness, *k*;

$k = 4\pi^2 {f_n}^2 m$; (where mass, m = 500 kg)
$k = 4\pi^2 \times 10^2 \times 500$

$k = 1.97\ \times 10^6 \frac{N}{m}$

6.6.1 Type of Isolator And Application

There are four resilient materials that are most used as vibration isolators:

1. Rubber

Rubber in the form of compression pads is generally used for the support of large loads and for higher frequency application. The stiffness of a compressed rubber pad is generally dependent on its size, and the end restraint against lateral bulging.

2. Metal

Metal springs can be designed to provide isolation virtually at any frequency. However, when designed for low frequency isolation, it has the practical disadvantage of readily transmitting high frequency. Higher frequency transmission can be minimized by inserting rubber or felt pads between the ends of the spring and the mounting points and ensuring that there is no metal to metal contact between the spring and the support structure.

3. Cork

Cork is one of the oldest materials used for vibration isolation. It is generally used in compression and sometimes in a combination of compression and shear. The dynamic stiffness and damping of cork are very much dependent on frequency.

4. Felt

To optimize the vibration isolation effectiveness of felt, the smallest possible area of the softest felt should be used, but in such a way that there is no loss of structural stability or excessive compression under static loading conditions.

When equipment is attached to the building structure, care must be taken to provide vibration isolation, or the noise may be transmitted throughout the building by vibrating the building structural members. The equipment may be mounted on concrete inertia blocks or directly to steel frames. Regardless of the mounting, some form of vibration isolator is usually used, as shown in **Figure 6.24**. The degree of isolation achieved with vibration isolators depends on the frequency of vibrations relative to the natural frequency of the system and the amount of damping built into the isolator.

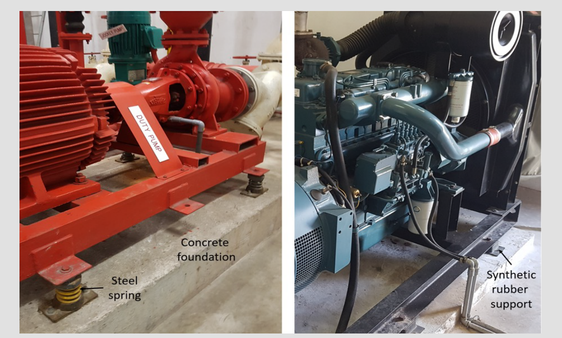


Figure 6.24: Isolating vibration by placing heavy vibrating equipment on an inertial block with vibration isolators and dampers.

Vibration isolation may not be completely effective when noise is transferred through piping or conduits from the equipment. Flexible connectors to mount the tubing to the building must also be considered as shown in **Figure 6.25**.

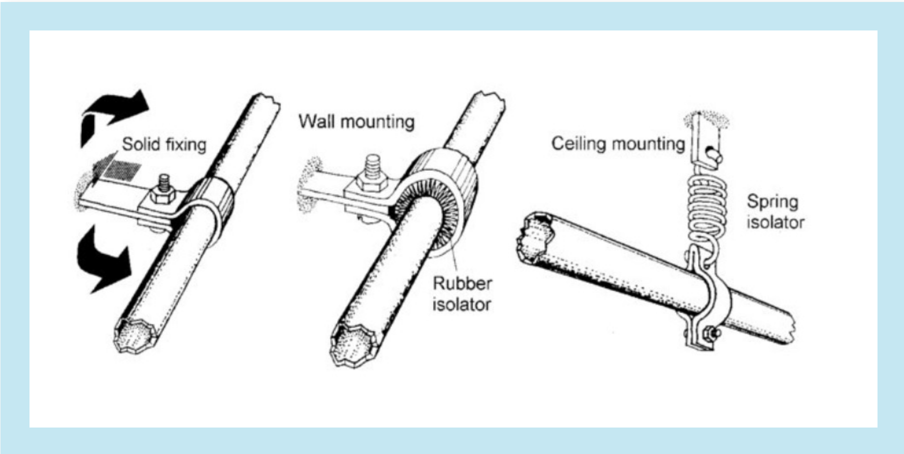


Figure 6.25: Flexible connectors for preventing vibration transmission to building structure.

When vibrating equipment is permanently mounted to a slab, wall, or ceiling, the equipment vibration can be transmitted into the mounting surface and generate intolerable levels of structure borne noise throughout the building. Likewise, piping, conduits, and ducting can operate as transmission pathways of structure borne sound if establishing inflexible connection paths between the building and the vibrating equipment. Isolating equipment vibration from building slabs, walls, and ceilings is vital for controlling structure borne sound propagation.

6.7 Damper

Damping is the term given to any mechanism, occurring either within or between the components in a vibrating system which lead to the conversion of vibrational energy into heat energy and reduction in vibration amplitude. The four different cases of undamped, overdamping, underdamping and critically damping are illustrated in **Figure 6.26**.

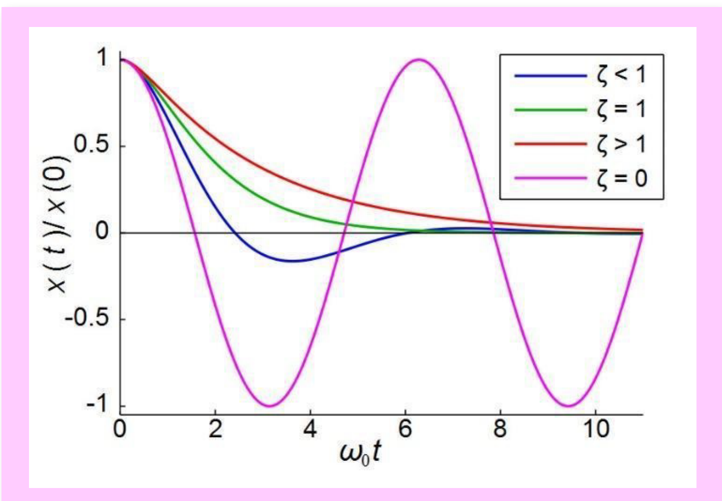


Figure 6.26: Undamped, overdamping, underdamping and critically damping oscillation shape.

The amount of damping in vibrating system may be described by its damping ratio, ξ.

a Undamped (ξ = 0) situations occur when no damping at all.

b Overdamping (ξ > 1) situation occurs when the damping force is so large that vibration does not occur, and the mass gradually moves back towards the rest position without overshoot.

c Underdamping (ξ < 1) situation occurs, when the amount of damping is small, and the mass oscillates about the rest position but with an amplitude decrease over time until it eventually comes to rest.

d Critically damping (ξ = 1) situation, the mass returns to its rest position in the minimum time without overshoot.

However, underdamping case is usually most concerned in structure borne noise control.

Values of ξ can be as low as 0.01 for lightly damped materials such as mild steel and as high as 0.2 for highly damped materials such as plastics. Therefore, the case of underdamping motion depends on the value of damping ratio as shown in **Figure 6.27**.

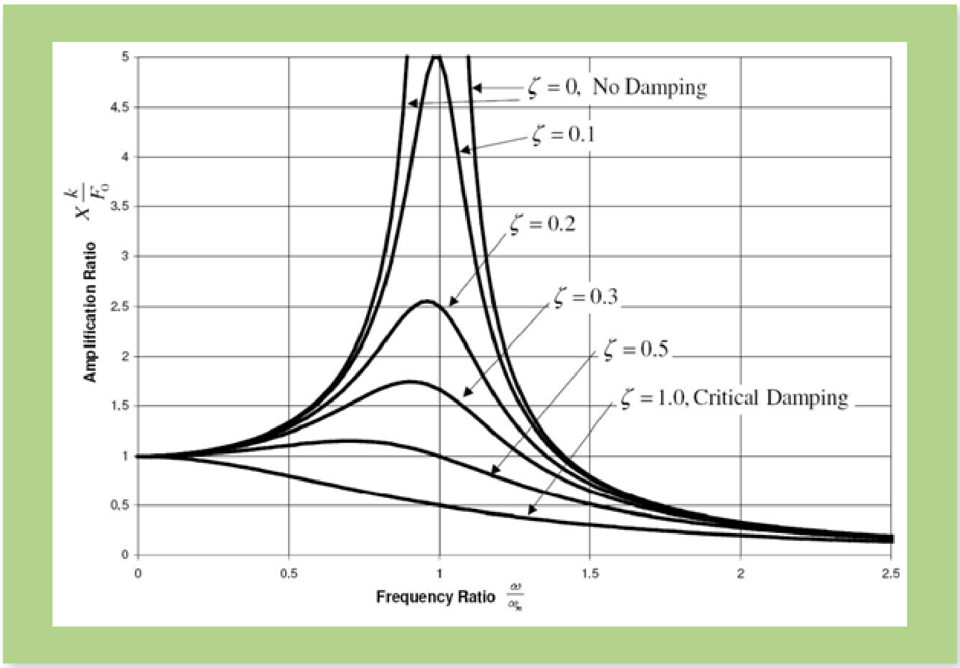


Figure 6.27: Graphs showing amplitude ratio against frequency ratio of damped vibration for various values of damping ratio, ξ.

Figure 6.27 also shows how the amplitude of forced vibration varies with the forcing frequency for various degrees of damping, with the force amplitude remaining constant. The vibration amplitude increases sharply when the driving frequency, *f* approaches the natural frequency of the system, f_n.

This phenomenon is called resonance, and the force vibration amplitude is maximum at the resonance frequency at a similar position for a lightly damped system. Therefore, the amplitude at and near to resonance is controlled mainly by damping. Low damping produces a very sharp resonant peak and very large amplitude of vibration at resonance. While more damping reduces the amplitude at resonance and produces a broader response curve.

There are two types of damping materials which are homogeneous layers and constrained layers. A homogenous layer material is sprayed or trawled on in a relatively thick coat, depending on the thickness and type of metal to be damped. A constrained layer material consists of a thin layer of the actual damping material with a backing of thin metal or stiff plastic. The mechanical action is one of making the damping layer much more effective than if it were homogenous. Constrained layer damping materials can be purchased as an adhesive/metal-foil tape combination, where the adhesive is selected for its energy loss properties as well as its adhesion. These damping tapes are especially useful on thin panels (1/16 in. steel or less).

Damping of sheet metal structures can be accomplished by the application of damping material to the metal sheet, such as is used on car bodies. Types of damping are available from various manufacturers' technical data sheets for this purpose. The form of tapes, sheets or sprays which may be applied like paint and make use of some non-hardening, viscoelastic material. For optimum results, the weight of the layer of damping material should be at least equal to that of the base panel.

Damping materials can be applied more efficiently and effectively using a laminated construction shown in **Figure 6.28** of one or more thin sheet metal layers, each separated by a viscoelastic layer, the whole being bonded together. Very thin layers, approximately 0.4 mm of viscoelastic material are satisfactory in these constrained layers. The optimum vibration reduction of the base structure occurs when the sheet metal constraining layer is equal in thickness to the base structure. For damping high frequency vibration, the viscoelastic damping layer should be stiffer than for damping low frequency vibration.

Figure 6.28: Constrained layer damping construction.

Viscoelastic materials typically are the most versatile and effective materials in providing the desired vibration damping and ultimate noise reduction, especially for thin structures. To determine what surfaces may require treatment by application of viscoelastic materials, it is necessary to measure vibration levels.

For those surfaces having high vibration levels relative to other surfaces, decisions should be made as to the type(s) of appropriate treatment. If the surfaces are easily accessible and continuous, a lagging treatment may be employed. If the surfaces are irregular or not easily accessible, the application of damping materials may be the most practical noise reduction approach. The characteristics of applied damping materials vary significantly with temperature and frequency. Therefore, the manufacturer's data sheet should be consulted prior to selection and application.

A mass, stiffness and damping may be associated with vibration, which has a corresponding resonance frequency at which only a small excitation force is required to make the structure vibrate strongly. For example, if a panel is excited by an incident sound field, forced vibration will strongly drive and will contribute most to the radiated sound, although resonance may dominate the apparent vibration response. This is because at frequency below the surface critical frequency, the sound radiating efficiency of the forced vibration will be unity, and thus much greater than the efficiency of the large amplitude resonant vibration. In this case, the addition of damping material to the structure may well reduce the overall structural vibration amplitude without reducing the sound radiation amplitude as shown in **Figure 6.29**.

Figure 6.29: Damping material located between two steel pipe connections.

If a machine panel such as a belt guard is subjected to vibration, it will radiate sound strongly at its resonant frequencies. Damping the panels or guards can reduce this radiated sound. In another application, parts that fall into and are carried along metal chutes can excite the chute panels by repeated impact. Installing damping materials along the chute surfaces will reduce the noise, but these materials must be selected with heat resistance and mechanical integrity. Damped stock tubes are available for quieting screw operation. Panels for isolating enclosures can transmit large amounts of sound in certain frequency regions. Damping can help retain transmission loss in those regions. Examples of existing manufacturer damping material shown in **Figure 6.30**.

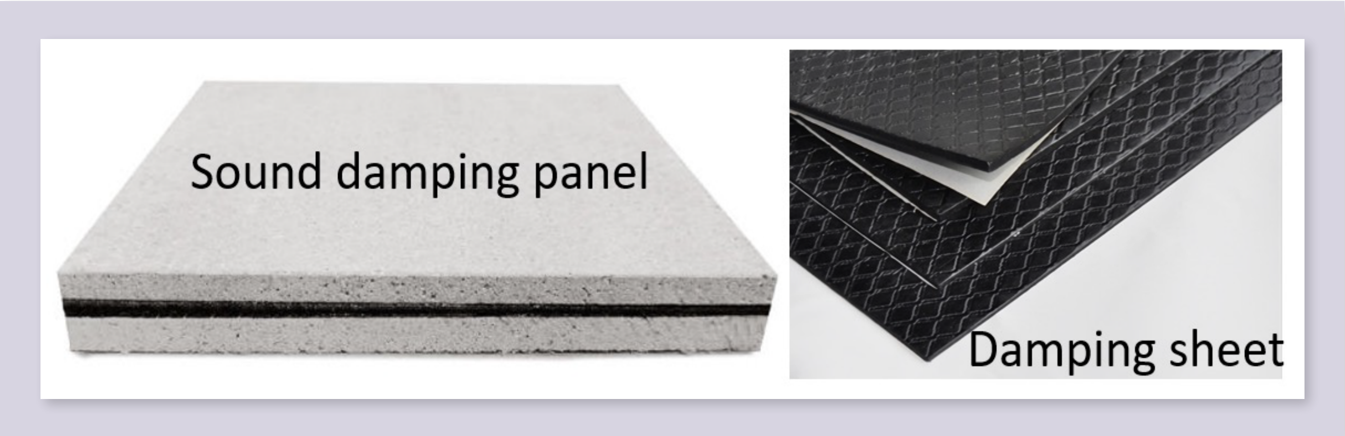


Figure 6.30: Example of sound damping panel and sheet.

7.0 REFERENCES

1. R.J. Peters et al, 2011. Acoustics and Noise Control. United Kingdom: Routledge.

2. BS EN ISO 11200:2014. Acoustics – Noise emitted by machinery and equipment – Guidelines for the use of basic standards for the determination of emission sound pressure levels at a work station and at other specified positions.

3. BS ISO 15664:2001. Acoustics – Noise control design procedures for open plant.

4. EN ISO 11690-1:1996. Acoustics – Recommended practice for the design of low-noise workplaces containing machinery – Part 1: Noise control strategies.

5. BS EN ISO 11690-2:1997. Acoustics – Recommended practice for the design of low-noise work places containing machinery – Part 2: Noise control measures.

6. McGuire, J. (1991) Industrial Plant Noise Abatement in The Work Environment (Volume One) – Occupational Health Fundamentals. 1st Ed. Hasnon, D.J., Lewis Publisher

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