NOISE CANCELLATION FOR SMALL TYPE GENERATORS

SUBMITTED IN PARTIAL FULFILLMENT OF
THE REQUIREMENT FOR THE DEGREE OF

BACHELOR OF SCIENCE
IN
MECHANICAL ENGINEERING

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DECLARATION

This is hereby declare that thesis entitled "**NOISE CANCELLATION FOR SMALL TYPE GENERATORS**" is an authentic report of my study carried out as requirement for the award of degree B.Sc. (Mechanical Engineering) at Islamic University of Technology, Gazipur, Dhaka, under the supervision of **PROF. DR. MD. A.R.M HARUNUR RASHID**, assistant professor, Department of Mechanical and Chemical Engineering, IUT during January 2018 to October 2018.

The matter embodied in this thesis has not been submitted in part or full to any other institute for award of any degree.

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Although we have given our best to complete this thesis flawlessly, we sincerely apologize for any mistakes that may have been made.

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Abstract

A simple, effective design for enclosing portable generators to reduce the radiated noise is an idea that seems to be desired by the consumers in this market. This investigation is to determine the feasibility of producing such an enclosure for a generator.

Several engineering aspects are incorporated in the design of the enclosure. The first, and probably the most paramount, are the acoustical effects of the enclosure itself.

The investigation follows the theories for insertion loss of a close fitting enclosure. The thesis examines the system behaviour of a close fitting enclosure that most acoustic text books ignore and how the material stiffness, density and source-to-enclosure distance affect the insertion loss and effectiveness of the enclosure. Measured and theoretical sound pressure level around the generator before and after the application of the enclosure are presented using standards described by ISO standard 1344.

The second important consideration for the enclosure design involves the heat transfer characteristics. The requirements of cooling air to the generator are discussed.

Also presented are some acoustic design considerations to prevent any “direct line of sight” to any of the necessary openings which will help in the overall insertion loss.

The use of an optimal engineering design technique is presented, demonstrating its strengths and weakness in this application.

CHAPTER 1:

Introduction

An engine is a typical noise source radiating multiple tones at the fundamental ring frequency. Acoustical enclosures constitute the most frequently used measures to reduce the noise radiated by engines [2]. The performance of acoustical enclosures is based upon the principle of trapping the noise radiated by the sound source, by massive, impervious layers, and dissipating the retained sound energy, by porous sound-absorbing lining. The efficiency of this passive technique depends on the ratio of the sound wavelength to the thickness of the blocking and absorbing layers. The larger is the sound wavelength (the lower is the frequency), the thicker must be the walls of the enclosure. Co-generation plants in hospitals and generators to supply electrical power to mobile TV units are examples of heavy, bulky enclosures designed to provide high attenuation at the whole frequency range.

More often, space is limited and enclosures are close-ratting and leaky [2]. Close-ratting enclosures are those in which most of the enclosed volume is occupied by the noise source. Enclosed engines must have openings for air intake and gas exhaust. Sound leaks through these openings, especially at low frequencies. It is well known that traditional passive techniques are less effective at lower frequencies.

Active noise control (ANC) techniques rely on the principle of destructive interference between the existing, or primary, noise and the anti-phase, or secondary, noise set up by an electronic controller [3]. The simpler adaptive ANC system processes a reference signal to generate the control signal that drives the secondary source. An error signal monitors the control performance. The adaptive filter must extrapolate the acoustic field from the reference sensor to the error sensor (primary path) and compensate for the transfer function between the control source and the error sensor(secondary path) [4,5]. Since ANC works best on low frequency noises, the active approach complements the traditional passive control methods.

A hybrid passive/active system to control the noise radiated by a small generator was reported by Cobo et al [6]. Passive control was provided by enclosing the generator within a rectangular, steel box, lined with absorbing materials. Low frequency noise escaped from the enclosure through air intake, air refreshing, and gas exhaust openings. Exhaust noise dominated the noise radiated by the enclosed generator.

The exhaust noise spectrum consisted on a series of single tones harmonically related, whose fundamental frequency, concerned with the ®ring frequency of the engine, varied slightly depending on the electrical load connected to the generator. A SISO ANC system was designed to attenuate the periodic exhaust noise. The reference signal was supplied by an accelerometer over the air filter case of the engine [7]. A cheap microphone along the exhaust pipe, conveniently housed to avoid direct contact with the exhaust gas, picked up the error signal [8]. A special loudspeaker, located along the exhaust pipe in a side branch configuration [9], withstanding operating temperatures of over 260_C, was used as secondary source.

The adaptive alter to tailor the cancellation process was supplied by a programmable, commercial ANC system.

This ANC system successfully attenuated harmonics 2nd, 3rd, 4th, 7th, and 8th, barely reduced harmonics 5th and 6th, but failed to cancel harmonics 9th, 10th, and 11th. The first harmonic was deliberately not cancelled to avoid instability problems with the loudspeaker at this rather low frequency (about 30 Hz). The aim of this paper is to elucidate the origin of this lack of attenuation at some harmonics and to alleviate it. Two possible causes are exhaustively analysed:

The transfer function between the control source and the error sensor (the secondary path). The coherence between the reference and error signals. Theoretical principles of active and passive control can be found elsewhere [2,4,5,10]. Here, the emphasis is on practical and technical aspects of the implementation of a hybrid passive/active control system. Firstly, the experimental set-up is described. Then, the influence of the exhaust system on the control performance is discussed. After that, the reference signal is optimised based upon its coherence with the error signal.

1.1 The Effects of Noise

The effects of noise on human beings are summarised in *Figure 1*. This illustrates how noise complaints and annoyance can be caused through various effects. When considering a permit level, the potential for complaints should be considered. When complaints about an existing facility that is licensed have been received, the reason for the complaints should be investigated.

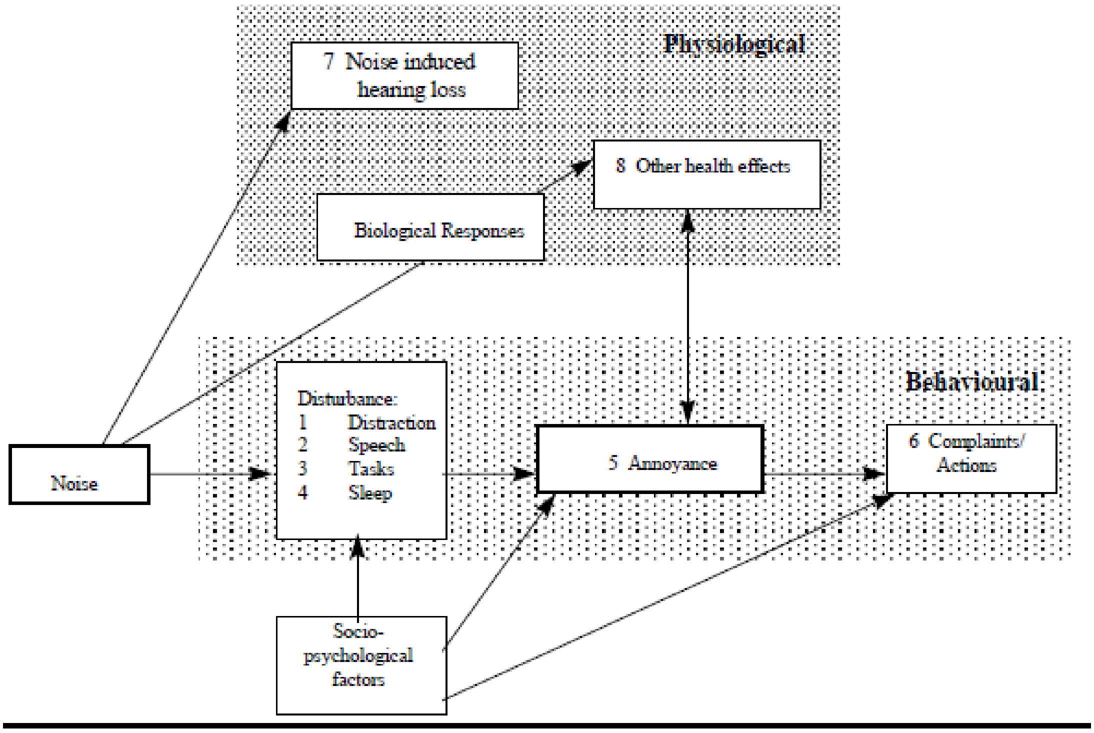


Figure No.01: Cause and Effect relationship of noise.

Noise induced hearing loss is not a concern at the levels of noise experienced by neighbours of noise emitting facilities. It is only a potential hazard above noise levels of at least 80 dB(A) and where exposure is over very long periods of time. The potential risk to workers is dealt with by the Health and Safety Executive and in some cases by Local Authorities.

A distinction is made between disturbance and annoyance. Someone is disturbed by noise when it prevents or inhibits them from undertaking an everyday activity such as concentrating while reading (distraction), hearing spoken conversation, listening to the radio or sleeping. The feeling of displeasure caused by noise is annoyance. Annoyance is often a result of disturbance but not necessarily, and it can be influenced by socio-psychological factors, such as a bias for or against the facility or person making the noise, of the environmental expectations of an individual. Whereas disturbance can be assessed analytically, annoyance is measured by social survey questioning, and over the years has been used as a common indicator of overall community noise impact. Other effects may occur as a result of vibration. These include: perceptible vibration, secondary rattling of windows, items on shelves and pictures hanging on walls. In addition, the sound radiated from vibrating walls may give rise to indirect effects. These effects may contribute to annoyance.

1.2 Measurement of Noise

Noise Measurement Equipment

Various types of noise measurement equipment are commercially available. The range includes equipment that is capable of measuring basic time varying sound pressure level and equipment that is capable of calculating statistical noise indices over time. The first level of sophistication is an *integrating* or *integrating averaging sound level meter* that is capable of measuring the 'A' weighted equivalent sound level, LAeq. *Appendix A* gives a glossary of acoustic terminology. *Appendix B* describes the decibel scale and provides further details on noise measurements techniques and calculations. The next level of sophistication is a meter that can calculate statistical noise measurement parameters such as LA90, LA10, LA01, the levels exceeded 90, 10 and 1 percent of the measurement period. This type of noise meter is called a *statistical analyser* or a *statistical sound level meter*.

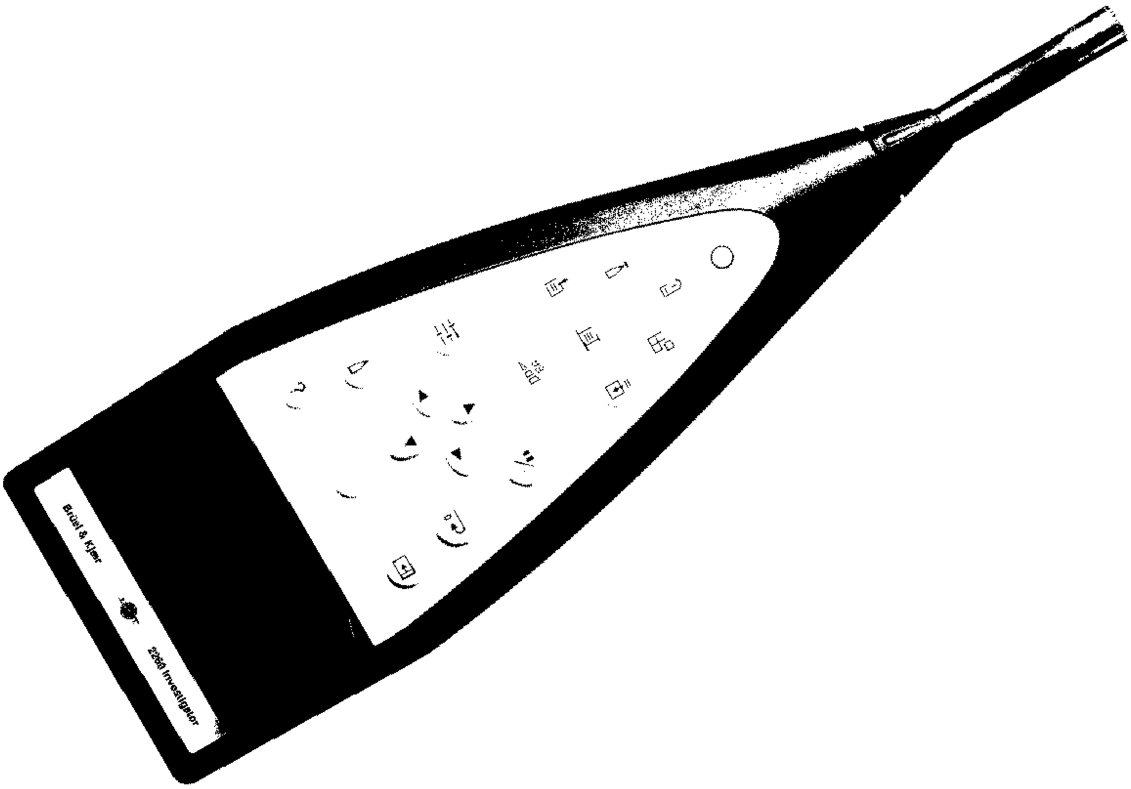


Figure No.2 : Statistical sound level meter.

The equipment may be switched on and off manually when sufficient data has been collected, or may be equipped with a *noise logging* facility. This will allow the meter to be set-up to take one sample over a pre-defined period, store the result in its memory, start another measurement, and repeat the process continuously.

The above meters will all be able to measure in terms of dB(A) noise levels. This is the most commonly used scale for environmental noise studies. More sophisticated noise equipment has additional internal frequency filters that enable measurements to be made in *frequency bands*. The filters can measure the *octave band* components of the sound, and in some cases, the narrower *1/3rd octave bands*. Measurements in these bands can generally be used to determine if noise contains distinctive *tonal* components which can make the noise more annoying and require special attention in one of the most commonly used assessment methods. *Appendix B* gives examples. It is also useful to know the frequency content of a noise source when calculating noise attenuation from screens and enclosures and when considering ground absorption in predictions as these effects are all frequency dependent. In general it is harder to abate lower frequency noise. The human ear is an extremely sensitive organ with a very large dynamic range over a wide range of frequencies. To cater for this dynamic range the decibel (dB) scale is logarithmic allowing the full range of audible sound levels to be expressed as a number between 1 dB (the threshold of perception) and about 140 dB (the threshold of pain). The frequency range that most of us can hear falls within the range 20 to 20,000 Hz. Because the ear is much less sensitive to lower frequencies than to higher ones sound level meters are adjusted to mimic this response. This is done by the inclusion of a sound filter, called the *A-weighting*. Using the A-weighting the levels measured by a sound level meter are comparable with the levels we perceive.

In some circumstances, tape recorders provide a useful means of capturing an event for later analysis. This will be useful when the event is rare or when it is expensive to repeat a certain operation for measurement purposes. An audio recording must be calibrated if absolute levels are required. Audio recording for analytical purposes is probably best left to an experienced consultant. Even an un-calibrated recording can be a useful memory aid when writing a report following a site visit. Generally, DAT recorders have now replaced traditional tape machines. The use of mini-disk recorders is becoming more common in the acoustics industry, although care may be required if the equipment uses a signal compression system. Some meters are able to switch a tape recorder into record mode, and to start internal analysis software, during noise events above a specified noise level. This again allows measurements to be made of occasional events without an operator present during the measurements.

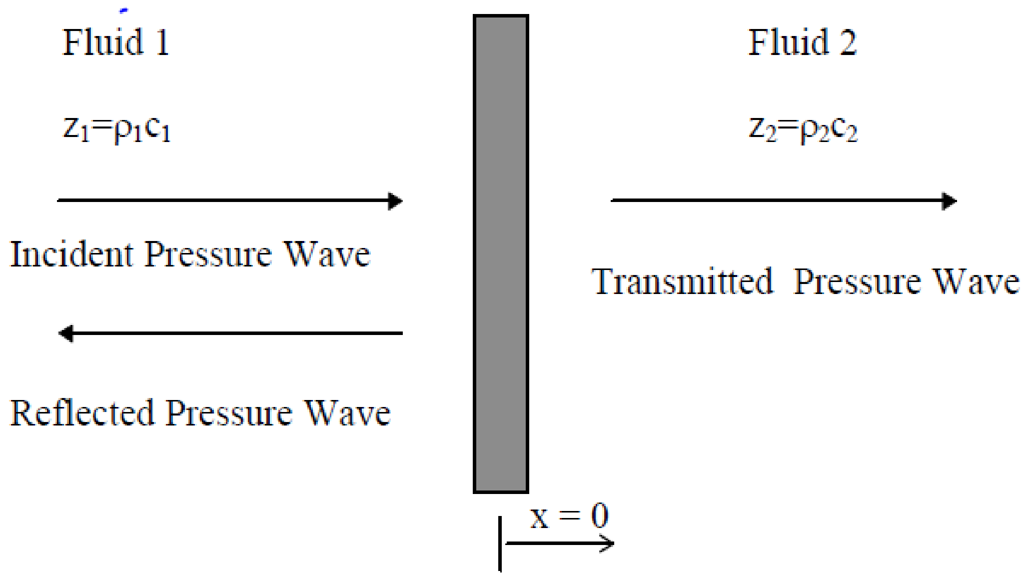
It is often useful to record the variation in noise over time in a graphical form. This has been traditionally done by calibrating a paper trace so that it displays a time history of the sound pressure level. However, with advances in electronics, noise graphs showing noise variation over time can be made electronically for presentation in reports (see *Appendix B*). The most advanced of the electronic systems allow individual events to be marked to identify the noise source responsible for a particular noise level. When it is important to isolate the contribution from a particular plant item to the total noise output more complex measurements may be required. Sometimes a combination of measurements with a sound level meter made close to different plant and calculations, can be used to predict the contributions. In a crowded plant area, an *intensity meter* may be used.

An example of a situation in which this technique may be useful would be to estimate the effect of silencing a particular machine. The advantage of this type of meter in this situation is that noise in only a particular direction can be measured. The noise contributions from other plant in the area can be virtually eliminated. This type of meter is now generally available but is expensive and complex to use. However, in the hands of a skilful user intensity meters can be used to identify the noise output of plant items in terms of their individual sound power level and to demonstrate that it would be useful to focus noise control effort on certain key sources. The accuracy of results in all cases will also depend on practical measurement constraints such as access limitations to key plant that could make it difficult to take appropriate measurements.

CHAPTER 2:

A Review of Sound Transmission Theories

Much of the interest in the study of transmission loss through panels began in the early 1900's. An elementary theory in transmission loss was published by Buckingham, (1924) where



he defined transmission loss as the $\log_{10}$ of the ratio of transmitted to incident pressure waves on a surface. Figure 2.1 is a representation of incident and transmitted pressure on a surface. This representation is only for normal incidence even though a noise source can produce random incidence on a surface. The equation for pressure waves on a surface is complex and varies over frequency and position. It is further complicated by the effects of the multiple degrees of incidence on the surface. Since computations were done by hand, the early models that were developed are quite simplistic and do not incorporate boundary conditions, nor the effects of random incidence of noise on a surface or its directivity (Crocker, 1994).

Several more studies were conducted over the next twenty five years which further developed the transmission loss theory. These studies included theories of infinite panel dimensions which eliminated the accounting of the boundary constraints. Further investigation during this time lead to the development of theories for the transmission of sound through partitions which incorporated the now well known mass law term, iwm, of the panel. This accounting of the mass effect of the panel was developed by Beranak (Beranak et al., 1949).

After Beranak's developments in transmission loss theory, London advanced the theory one step further by including the mechanical impedance of the pane (London, 1950). This is one of the first accounts of incorporating the vibration of the panel in predicting the transmission loss. Since then, there has been more development in the details of transmission loss theory which have incorporated the effects of resonant and non-resonant transmission through panels (Crocker, 1994). More recently, the effects of panel modes have also been incorporated into the science of predicting sound transmission (Oldham et al., 1991).

With the invention of computers which are capable of multiple computations, the ability to simulate acoustic behavior through mathematical analysis has become more successful and subsequently more useful. Methods like mode simulation analysis, statistical energy analysis, finite element analysis and boundary element analysis have become some of the more popular techniques used in recent years (Crocker, 1994). Since it is the goal of this investigation is an optimal design of an enclosure, which requires a prediction of the insertion loss, some details of the mathematical models will be reviewed.

CHAPTER 3:

BEST AVAILABLE TECHNIQUES FOR NOISE CONTROL

3.1 Introduction

This chapter summarises the multitude of techniques that are available to control noise emissions from the range of noise sources that are likely to be encountered at the type of facilities that will require licensing under the IPPC and Waste Management Regulations. Many of these noise sources can be controlled using common techniques such as noise barriers or enclosures, others require unique solutions. This chapter gives an overview of the most commonly used techniques, many of which are applicable to numerous situations. Active noise control, the latest hi-tech noise control technique, is described separately. This chapter also provides a discussion of administrative noise control options under the headings *Noise Management Practices and Alternatives to Noise Control.* In order to understand what constitutes the best noise control technique in a given situation, and the shortcomings of some techniques, it is helpful to have a general understanding of the mechanisms through which noise can be attenuated. The following sections offer an introduction to the general principles of noise control.

3.2 How Sound is Attenuated

Sound passes through the air as a pressure wave. The amplitude of that pressure wave can be reduced by several means. In open space a sound wave will naturally reduce in amplitude as it moves from source to receiver through two principal mechanisms; spherical spreading – the natural dilution of the sound energy as it is spread over a widening area, and absorption – due to frictional forces in the air. Spherical spreading reduces the sound level radiating away from a point source at the rate of 6 dB for every doubling of distance from the source. Over small distances (up to a few hundred meters) absorption can generally be ignored as its effect is minor compared to that of spherical spreading. If the level of noise generation at source cannot be reduced, two methods of noise control are commonly used; increasing distance or increasing attenuation. Increasing distance from source to receiver is usually only possible at the planning stage, and can be extremely effective, but in many cases the sound wave must be attenuated through additional means. This usually involves blocking the sound path with a solid structure of some kind.

3.3 Sound Attenuation Through Structures

Sound incident on a solid structure, such as a wall, has one of four paths to follow. It is either reflected back away from the wall, it passes directly through the wall or it passes along the wall and may be radiated away from its structure somewhere else (the so called *re-radiated* noise path). Noise may also pass around the end of the wall or over the top to form another *flanking path*. Let us consider each of these paths in turn, and how they effect the attenuation that is achieved by the structure.

3.3.1 Reflected Sound

The reflection of sound back from a structure is useful in reducing the level of sound that passes through, but there is a potential disadvantage too. If the noise source is enclosed, reflected sound will be trapped within the enclosure and will result in *reverberation* of sound This *reverberant* sound may build up within the enclosed space and increase the apparent strength of the sound source. This can be combated by adding acoustic absorption to the enclosed space, for example with mineral fibre quilting in acoustic panels. Barriers may also be covered in absorptive material on the side facing the source to increase their effectiveness.

3.3.2 Flanking Paths and Re-radiated Noise

There are various types of flanking paths. Sound may pass along a structure for some distance and radiate away from it somewhere else, for example as a result of vibration being transmitted through the plant supports into the floor, along the floor into the walls and radiating away from the walls to the outside of a building. Or sound may simply diffract around the edge of the structure, for example under the eaves of a roof. Flanking paths can reduce the acoustic performance of the structure and should not be ignored when a high performance is needed.

3.3.3 Transmitted Sound

The portion of the incident sound wave that passes through the structure is first converted into vibrational energy in the structure and then back to air-borne vibrational energy as it is radiated away from the far side. The extent to which this process attenuates the sound depends on two key characteristics of the structure; its mass and its stiffness.

3.3.4 The Effect of Mass

It is mass that attenuates sound through frictional losses, the greater the mass the greater the attenuation. This attenuation depends on the frequency of the sound, higher frequencies are absorbed more easily than low frequencies. As a result, a common feature of noise control is that it is nearly always more difficult to attenuate low frequency noise than high frequencies. This problem is accentuated when a long propagation distance is involved because air absorption will effect higher frequencies far more than lower frequencies. The result is that low frequency noise tends to 'travel' more than high frequency noise. Mathematically, the relationship between sound reduction, mass (M), and frequency (f) are given in the 'Mass Law' of sound insulation as follows:

· Sound Reduction = 20 log Mf - 43

Or, stated in words:

· At a given frequency, sound reduction increases by about 6 dB for each doubling of mass; and

· For a given mass, the sound reduction increases by about 6 dB for each doubling of frequency.

These are idealised relationships that ignore other mechanisms that may effect attenuation in given materials, such as the effect of stiffness, discussed below.

3.3.5The Effect of Stiffness

The stiffness of the structure will control the extent to which it flexes under the influence of the incident sound wave. A structure that is flexing will radiate sound as it oscillates at the frequency of that oscillation. If a structure has very low stiffness this can form a very efficient route for sound to pass through. Even heavy walls can allow sound to pass through them in this way because flat structures such as walls or windows have a natural frequency at which they will oscillate easily if energy of that frequency if put into them. Hence, only very specific frequencies of sound are transferred through structures in this way. Nonetheless because the process transfers noise very efficiently it can lead to substantial losses of attenuation performance at given frequencies.

3.3.6 Holes, Openings, and Barrier's

For a noise attenuating structure to be effective it must completely block the sound transmission path. Any hole or opening in the structure will greatly reduce its acoustic performance. For example, if a wall or partition capable of attenuating sound by 25 dB has an opening in it with an area that is 10% of the area if the wall, its performance will be reduced from 25 dB to about 10 dB. Sound can be thought to behave rather like a liquid in such cases, it will simply pour through the easiest route it can find. Similarly sound will flow round a structure if allowed to do so. More correctly speaking sound will diffract around the edge of a structure. It is the ease with which sound diffracts around structures that limits the performance of noise barriers which only partly block the sound transmission path. This is
also the reason why tree planting is not usually an effective noise control technique. Sound does not move in straight lines

3.4 METHODS:

There are several methods to reduce the noise produced from generators:

- To reduce the production of noise sources:

The inner structure of a generator set is optimized and new types of combustion chambers are developed.

- To control local noise:

The key parts that radiate higher noise are treated. The control methods include installing a silencer for exhaust noise and attaching the absorption material to the outside of the equipment.
Sound pressure can only be reduced to a limited level with these methods

- \To reduce noise through the generator room:

Insulating, absorbing, vibration-isolated, and damping techniques are synthetically applied to attenuate noise.

- To control noise in a sound-attenuated enclosure:

Insulating, absorbing, vibration-isolated, and damping techniques are synthetically applied to design a convenient and flexible enclosure. The generator with an enclosure can be conveniently moved, and lower noise can be obtained by the enclosure and secondary noise control in the generator room.

3.5 Two types of loss causes noise cancellation:

- Transmission Loss
- Insertion Loss

Transmission Loss

- When a propagating pressure wave is interrupted by an infinite barrier, the incident pressure wave is dispersed into two new waves. The infinite barrier is used because it eliminates the need to account for diffraction of the sound around the edges of the barrier. Some of the wave is reflected back toward the source and some of the wave is allowed to pass through the barrier. The part of the wave that is reflected back is referred to as the reflected pressure wave and the part of the wave that is allowed to pass through the barrier is called the transmitted pressure wave.
- The transmission loss of double partitions varies from 6 dB greater for a tightly coupled walls to 20 dB greater for loosely coupled partitions as compared to single walled partitions. Closely coupled partitions are double walls that are separated by a few inches, while loosely coupled partitions are separated by several feet.

Insertion Loss

Insertion loss is the reduction, in dB, of the sound pressure level or sound power level of a noise source. It is more effective, and measurable, than the transmission loss of an enclosure.

The amount of attenuation that foam will provide is related to how the foam is attached to the enclosure panel. Observations of introducing an air gap between the acoustic foam and the panel. The performance of the foam is five times better when the foam is separated from the panel by a 1 mm air gap compared to the foam being directly attached to the panel itself.

The use of acoustic foam or perforated panels would enhance the acoustic attenuation capabilities of the enclosure.

CHAPTER 4:

4.1 Experimental set-up

PROPOSED:

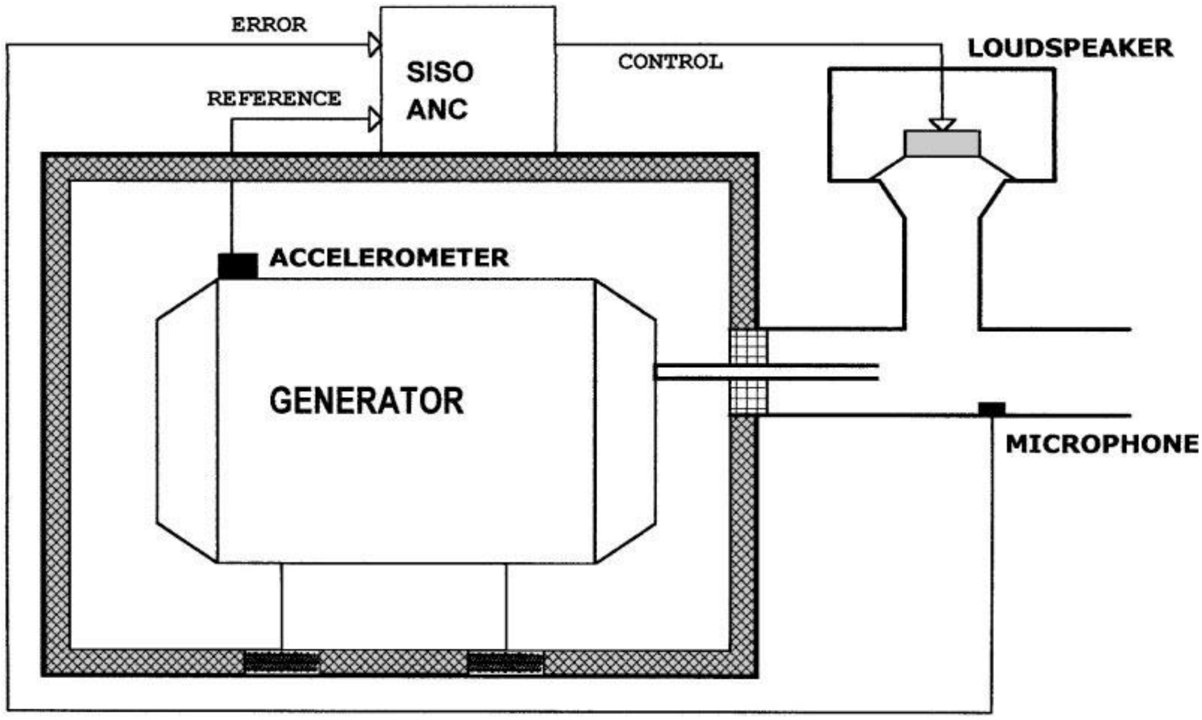
- The rectangular (100 _ 71 _ 53 cm) enclosure, built with 2 mm thick steel panels, lined with 3 cm thick layer of absorbing material.
- The air intake opening, consisting on 8 mm diameter holes evenly distributed on a (29 _ 18 cm) window.
- The gas exhaust system, consisting in two concentric tubes. Through the inner 3 cm diameter steel tube blows the hot gas. The outer 8 cm diameter sheet metal tube supports the ANC system. The inner tube penetrates 25 cm in the outer tube. To avoid gas leaks the outer tube is sealed at the enclosure opening with rockwool. A more detailed description of the exhaust geometry is given in Section 3.
- Four passive mounts, consisting on three elastomeric layers of different hardness matching each other in a sine pattern, inserted between the engine and the enclosure for. Each passive mount has dimensions (10 _ 10 _ 2.7 cm).
- A centrifugal fan inside the enclosure, in the opposite wall of the intake opening, to extract the hot air through a (13 _ 10 cm) window. The fan runs at 1410 rpm, supplies a volume of 240 l/s, and its sound power level is 60 dBA.
- .A temperature regulator, consisting of a temperature probe and an automatic controller, is installed for security reasons. If the temperature exceeds some programmed value, the engine is switched o€ A single-input single-output (SISO) ANC system has been implemented on a commercial and conquerable controller, following the recommendations of Oka- moto et al. [9]. The ANC system includes:
- The reference signal, picked up by one of two accelerometers on the air filter case of the engine (Fig. 2). More details of the reference signal are given in Section 4.
- The control source, provided by a high temperature loudspeaker, in side- branch configuration (Fig. 3). More details of the control source are given in Section 3.
- The error sign al, picked up by one electret microphone on the junction between the outer exhaust tube and the side-branch tube. To avoid direct contact with the exhaust gas the microphone is housed with a brass cap. A Thin ring isolates thermally the microphone from the brass cap.

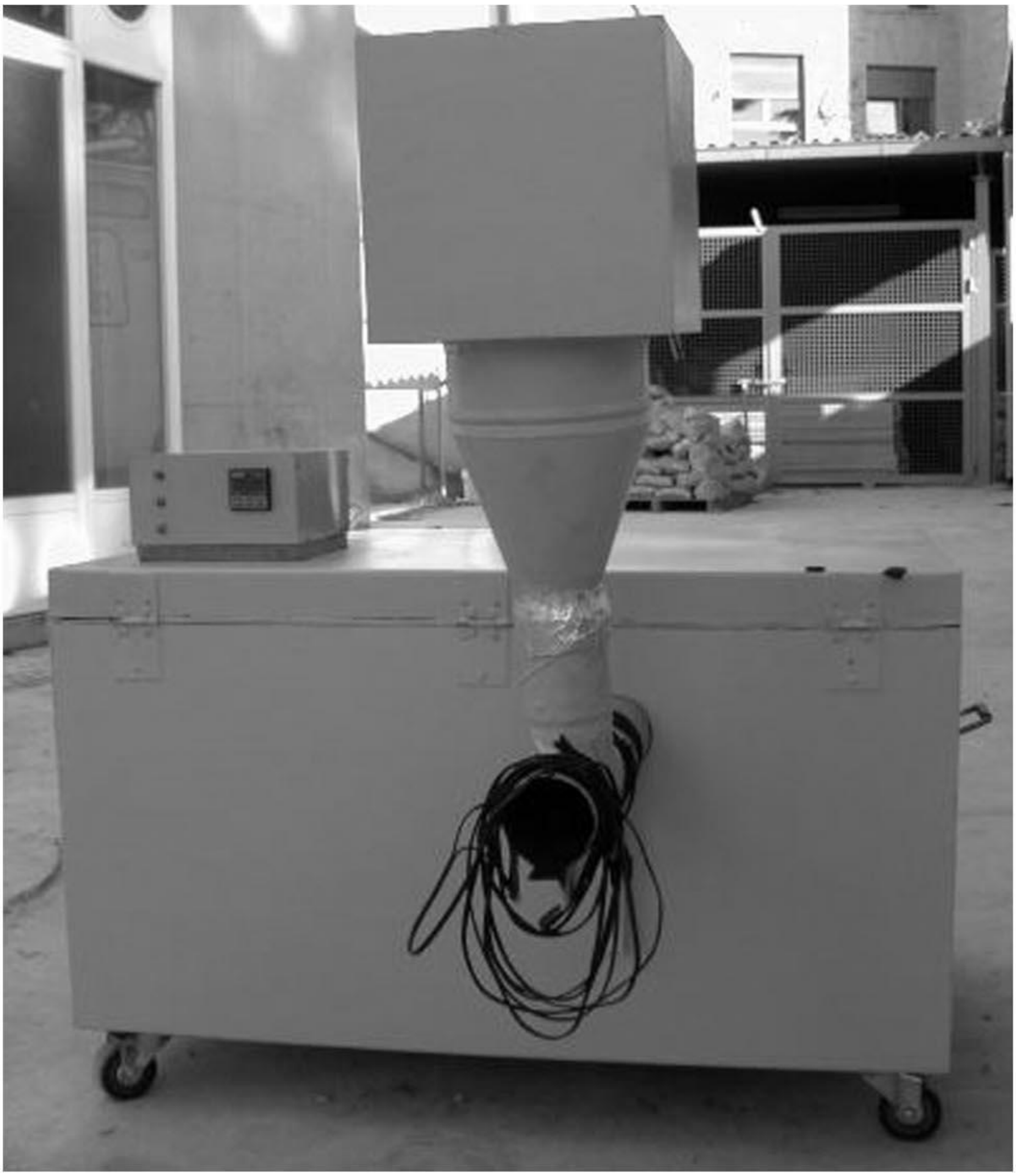
4.2 Influence of the exhaust system

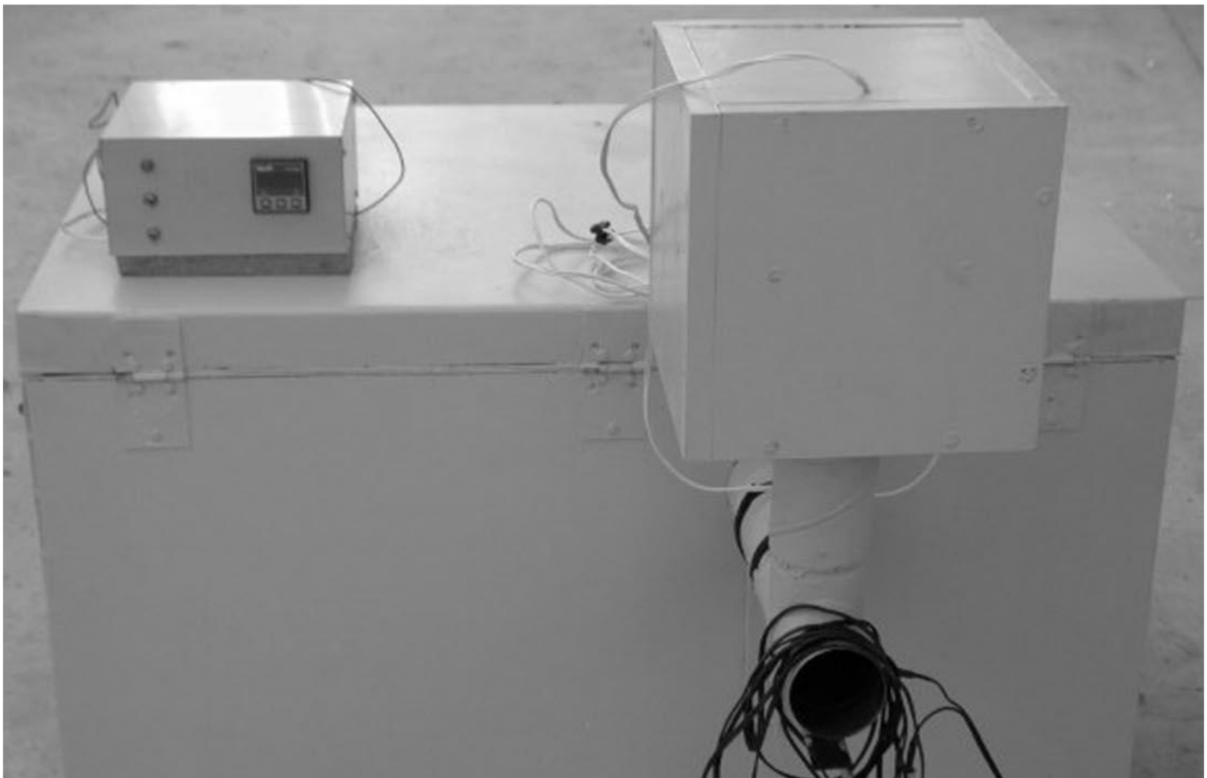
Engines usually generate high sound pressure levels in the exhaust system. In a typical SISO ANC system the secondary source (the loudspeaker in Fig. 4) must record at the error microphone location an acoustic pressure equal (and opposite in phase) to that produced by the engine. If the amplitude of the signal yielded by the ANC system and the

power amplifier is not enough to drive the loudspeaker to such high pressure levels the ANC system will overflow. Furthermore, loudspeakers operated at higher levels have shorter life and may distort. A combination of a passive mu.er with the ANC system can indeed lower the acoustic pressure, and hence, the power requirement of the secondary sources [11]. Okamoto et al. [9] demonstrated that a side-branch resonator with an end-mounted loudspeaker is the best position for the secondary source because:
. An end-mounted high-impedance loudspeaker always requires less source strength than a loudspeaker directly mounted at the duct wall.

From a practical point of view, the loudspeaker can be better protected from hostile (hot and dirty) environments. A side-branch resonator will provide high transmission loss peaks at kL=(n+0.5) p, where k is the wavenumber and L is the side-branch length (quarter wavelength filter). Thus, an active side-branch end-mounted controller will allow a large reduction in the necessary secondary source strength around frequencies multiple of the resonance frequency.
Fig. shows the two exhaust systems designed for this research. In Case I, the loudspeaker box connects directly with the 37 cm side-branch tube. In Case II, the loudspeaker connects to the 29 cm side-branch tube through a small Helmholtz







resonator. According to Okamoto et al. [9] a loudspeaker in front of a Helmholtz resonator driven at frequencies much higher than the Helmholtz resonance frequency is a close approach to a high-impedance source. Fig. 6 displays the exhaust log-spectral noise level, measured at the error microphone, in both cases. The quarter wavelength frequency (f=c/4L) is around 230 Hz for Case I and 293 Hz for Case II.
Therefore, the ®rst exhaust system provides higher transmission loss at harmonics 4th to 9th. From Fig. 6 it is clear that the ®rst exhaust system provides higher passive performance. The ANC system must compensate for the transfer function between the loud- speaker and the microphone. Thus, the simpler is this transfer function, the better performs the active filter. Fig. shows the Bode plot of the secondary path in Cases I (dark line) and II (light line). It can be seen that the secondary source in the first configuration supplies higher pressure levels at the error microphone in the whole frequency band of interest (50±300 Hz). Thus, the first exhaust system also yields higher active performance.

4.3 Features:

- A multi-pass dissipative silencer is mounted on the air outlet region, a two-stage expansion chamber silencer is applied to control exhaust noise, and a multi-pass silencer is installed on the outlet control region at the end of the enclosure to reduce outlet noise.
- The main wall panels of the enclosure are made of the sound boarding. A cooling air inlet on one side wall of the enclosure admits the external air into the enclosure, and a cooling air outlet on the wall of the enclosure expels the air from the interior of the enclosure.

4.4 Sound insulation

The main wall panels of an enclosure are made of sound boarding to prohibit noise from radiating. It can be classified into the following types:

- The sound boarding consisting of a layer of steel plate and two layers of different absorption materials. Thus, the absorption region of the enclosure is enlarged because of the different absorption coefficients of the two materials in the different frequency domains.
- The sound boarding consisting of two layers of sound boarding embedding the absorption material. The two layers of sound boarding are made of different materials and with different thicknesses. The material insulates noise and the sound

insulation property of the enclosure is enhanced because of the different insulation coefficients in the different frequency domains

- The sound boarding consisting of two layers of sound boarding with different materials and thicknesses. There is an air gap between the two layers, and the inner surface of the boarding is lined with the absorption material. In the structure, the air gap works as an added layer to enhance insulation without additional costs.

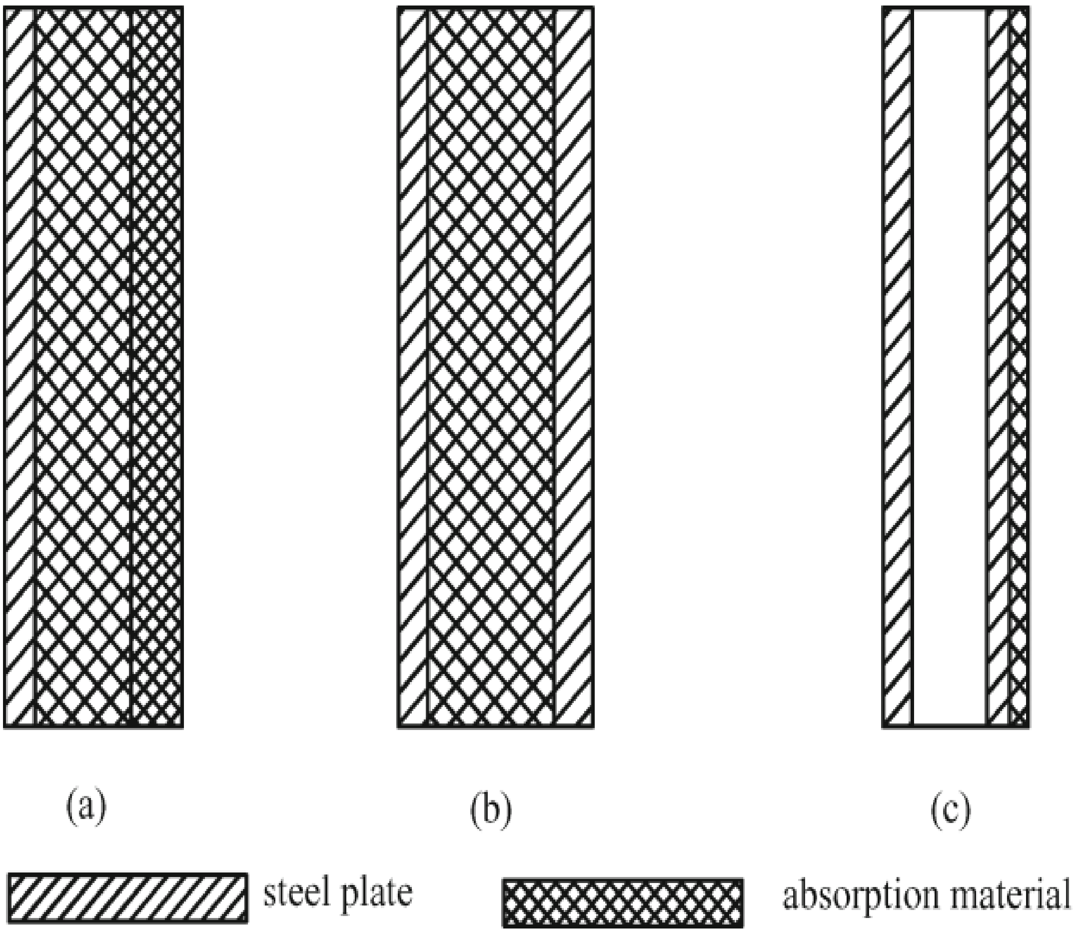


Fig. 2 Sound boarding

4.5 Acoustic Absorbing Materials and Perforated Panels

Although it is not the focus of this investigation of acoustic enclosures and insertion loss, it is necessary to discuss the importance of absorbing acoustical material because it can help attenuate the lower frequencies that a single paneled enclosure cannot. This material is usually a porous material, like foam, which is used to insulate a surface from transmitting sound waves.

The performance of the acoustic material is determined by knowing the absorption coefficient of the material. The absorption of the material depends mainly on the impedance of the material. There is also some dependency on how the noise strikes the material's surface. Some materials are not as effective when trying to attenuate noise that strikes the surface at oblique incidence angles. The absorption coefficient is frequency dependent. A material can have a high absorption in a given frequency band width, thus producing a great deal of insertion loss. Outside of this band width, the absorption coefficient might be negligible (Walker, 1971). By adding absorbing material the enclosure could be tuned to attenuate a more specific frequency range.

Determining the absorption coefficient of acoustic material is probably more difficult than determining the insertion loss of a panel. The performance of an acoustic material liner is based on the impedance of the material. It is difficult to predict this with out actually testing the material. From research observations, semi-empirical formulae have been developed (Bolton etal., 1993). These models have limited success in predicting the actual insertion loss that the material will provide.

The vibration of the structure also effects the absorption of the material. The amount of attenuation that foam will provide is related to how the foam is attached to the enclosure panel.

Bolton discusses his observations of introducing an air gap between the acoustic foam and the panel. The performance of the foam is five times better when the foam is separated from the panel by a 1 mm air gap compared to the foam being directly attached to the panel itself (Bolton et al., 1993).

The use of perforated panels backed by an air space to attenuate low frequency noise has also been investigated. The perforated panel and the air space become a resonator type muffler system. The perforated panels have small diameter holes that are spaced far apart relative to the hole diameters. Maa develops mathematical models to predict the performance of perforated and double perforated panels (Maa, 1987). His model determines the acoustic impedance of the panel based on material properties and perforation density.

`Jinko and Swenson show experimental results using a perforated wall for sound reduction of noise in a room (Jinko et al., 1992). Their model is based on the model developed by Maa. They found that this type of noise attenuator is more resilient to a harsh environment than

acoustic foam. They also showed that the perforated panel has an absorption coefficient close to 1 for low frequencies with normal incidence. Their work showed that even in room acoustics, the panel vibration effects the performance of the acoustic absorber.
The use of acoustic foam or perforated panels would enhance the acoustic attenuation capabilities of the enclosure. However it is the purpose of this investigation to determine the interaction between the noise source and the enclosure to predict the insertion loss and design an optimal enclosure using a simple panel. Once an optimal design is predicted, techniques developed to determine the optimal acoustical lining could be used to improve the performance of the enclosure. To achieve this, more extensive modeling and testing will have to be performed to develop a model for predicting the insertion loss of a complex panel consisting of a base material,acoustic liners and perforations.

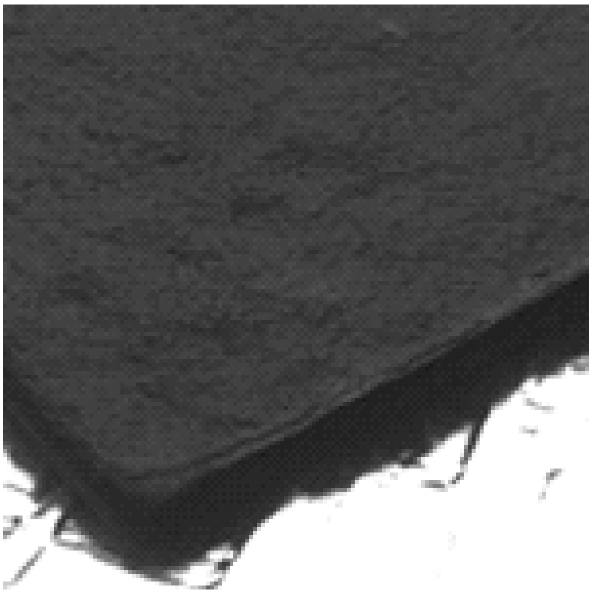
4.6 ACOUSTIC MATERIALS:

The following acoustic materials are used in most cases in providing a total solution for industrial noise control:

- **Akotherm** – Moisture-resistant wool plate with good noise-absorption performance and moisture suppression.
- **Merfocell PU** – Good noise-absorption combined with a moisture-repellent top layer.
- **Flamex PU** – Absorption foam equipped with a top layer with moisture suppression and heat resistance.
- **Isomat** – Pliant insulation material.
- **Vibraflex** – Anti-drumming material; strong in insulating low frequencies in combination with steel walls.
- **Regufoam** – Vibration insulation material; due to various possibilities always an ideal frequency damper.

LIQUID VIBRATION DAMPING COMPOUND

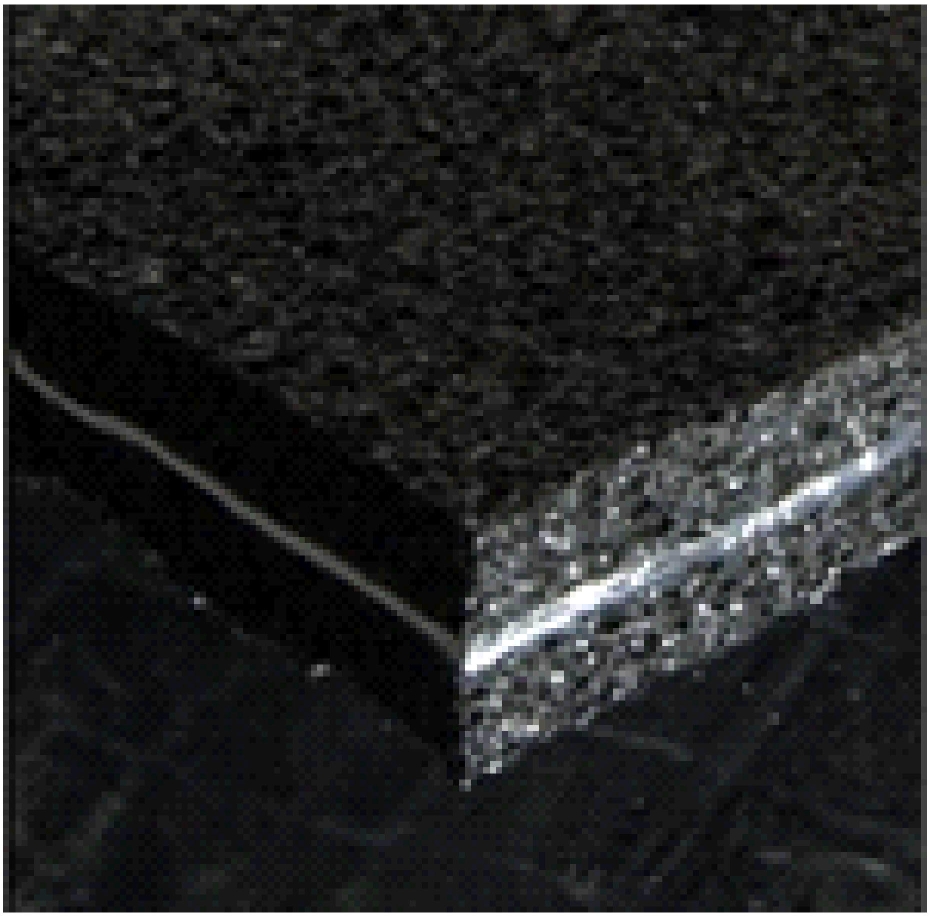
This is an air curing, liquid vibration damping compound (fig.1).Its combination of silica-mica, ceramics and an advanced chemical binder greatly reduces structural resonance by converting vibration into “viscous” friction or low level heat. This will also increase speaker output by 0.5 to 1 db when applied onto the inside of a speaker enclosure.



This is a water based, sprayable vibration damping compound that has been engineered to damp vibration by converting it into low level heat. This is loaded with silica-mica and ceramic components that create friction when vibration occurs. It is commonly used on the inside of outer body panels including the floor pan. One benefit of using this is that it covers large surface areas very quickly when using the spray gun. When sprayed, it is purple and as it cures it will change to black. It is also frequently used to improve the performance of speaker enclosures of all types. In wood enclosures, this will soak into the wood and seal up the pores, seams and gaps between the panels. This will also damp fiberglass and plastic enclosures extremely well. When cured, creates an impedance mismatch that results in an increase in overall output by 1 to 1.5 db. Spray gun will save a massive amount of install time. This may also be applied with a paint brush or roller. This cleans up easily with warm water. This is water resistant, not water proof.

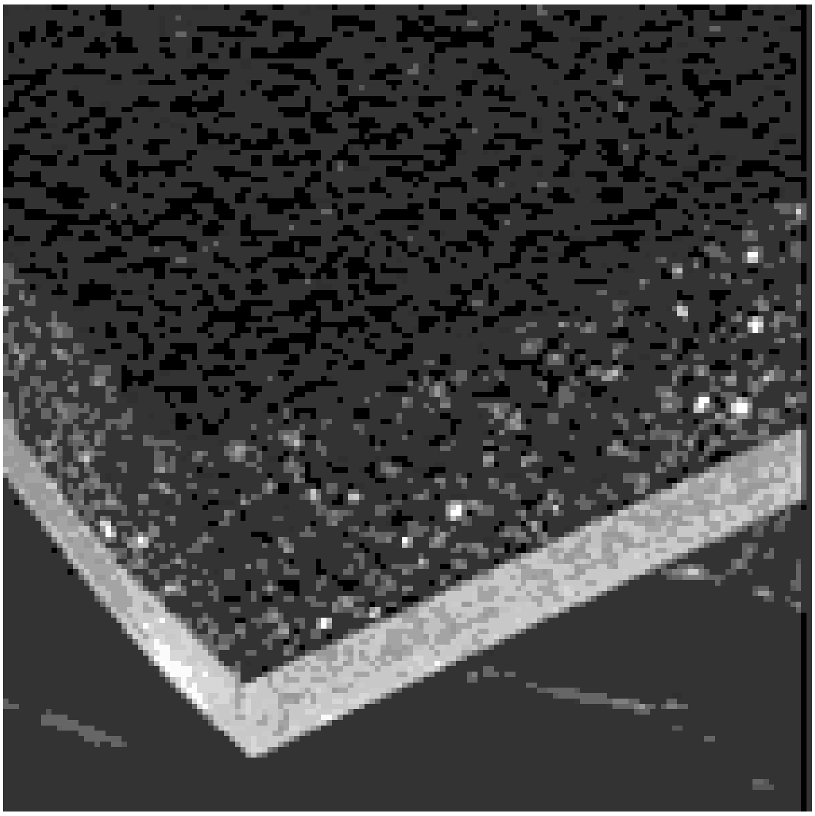
Formable Floor Barrier Material

This is designed to be used over highly irregular areas of the floor pan such as the transmission tunnel and over the wheel arches. It consists of two layers of 1/8" neoprene foam that sandwich a lead core. One lightweight layer of Formable Floor Barrier Material is equivalent to 4 or 6 layers of a heavy vibration damping sheet.



Flexible Floor Barrier:

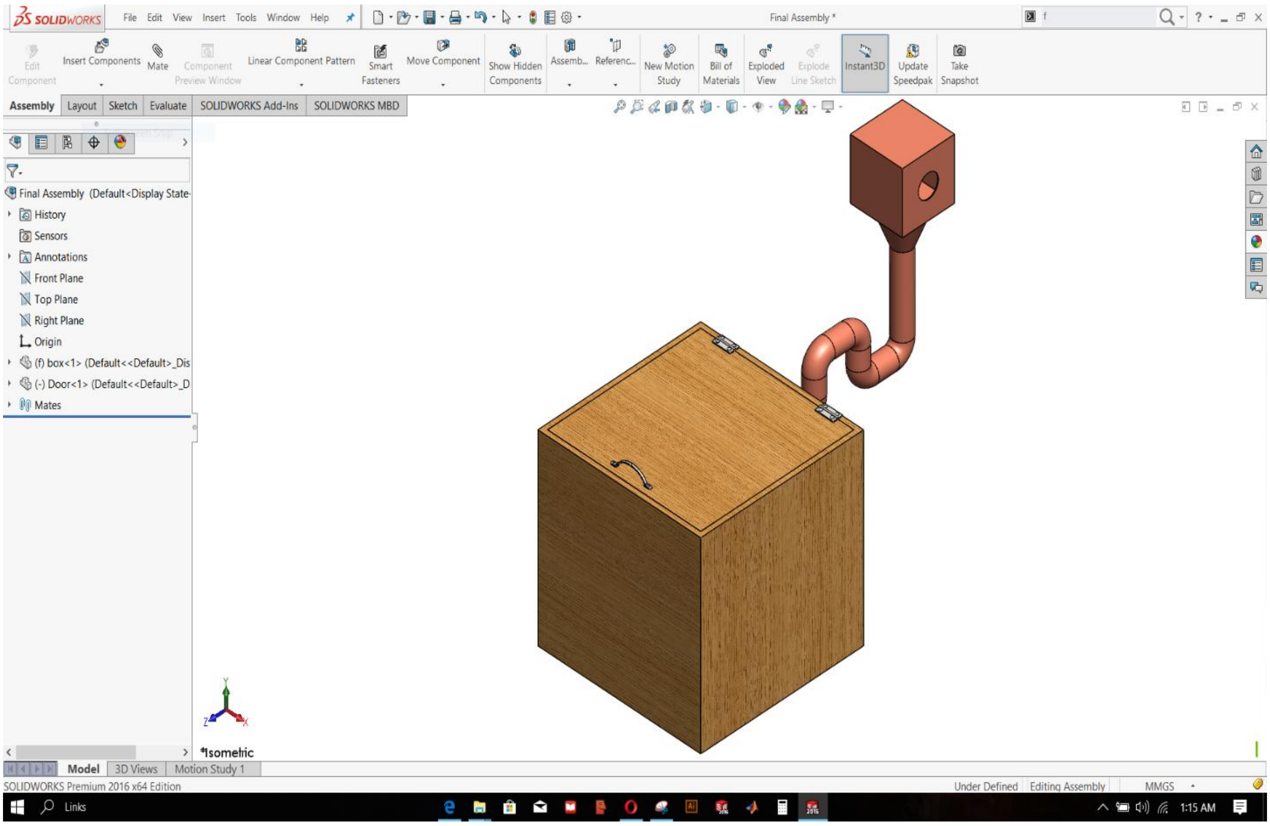
This is a composite material with a flexible, loaded vinyl mass barrier, and a foam decoupling layer which reduces road noise by impeding the passage of sound waves.

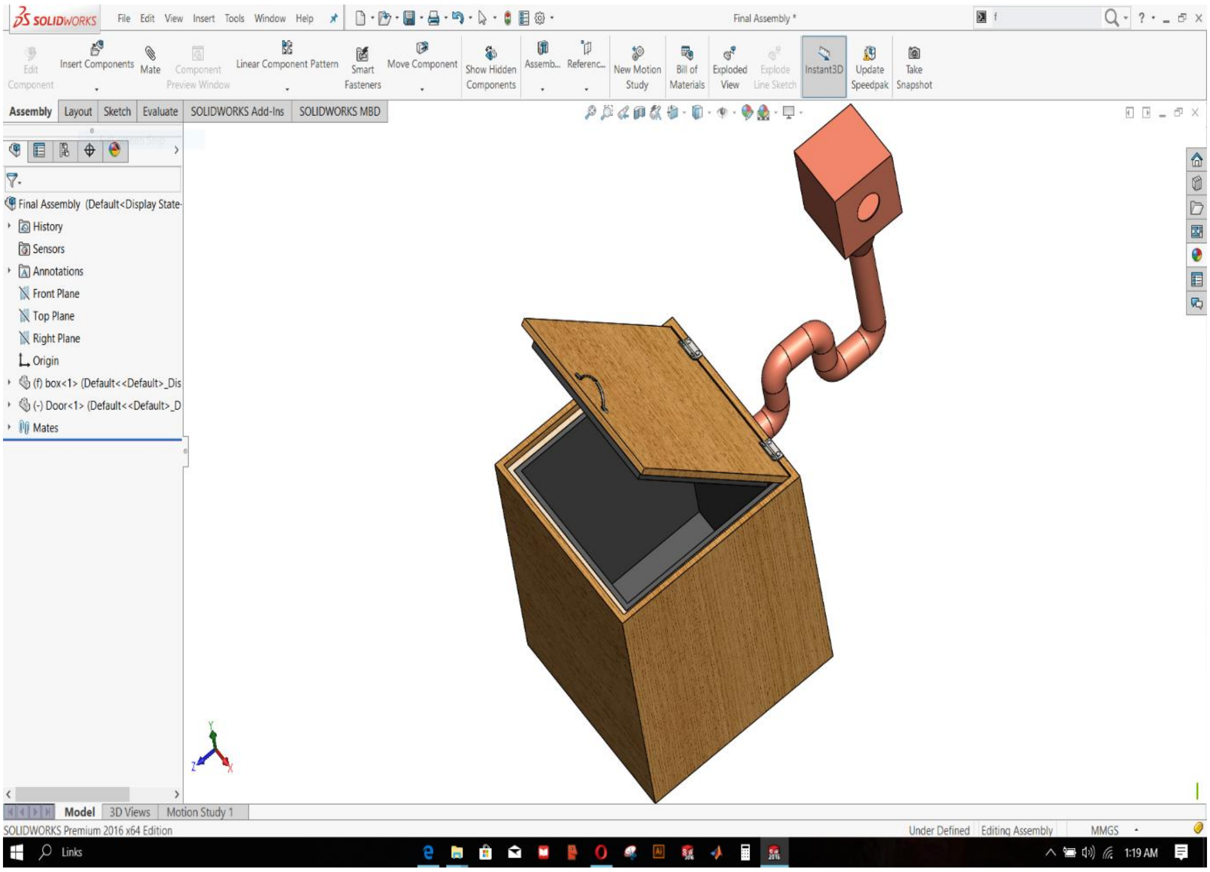


CHAPTER 05:

RESULT AND CONCLUSION:

5.1 MODELLED DESIGN:

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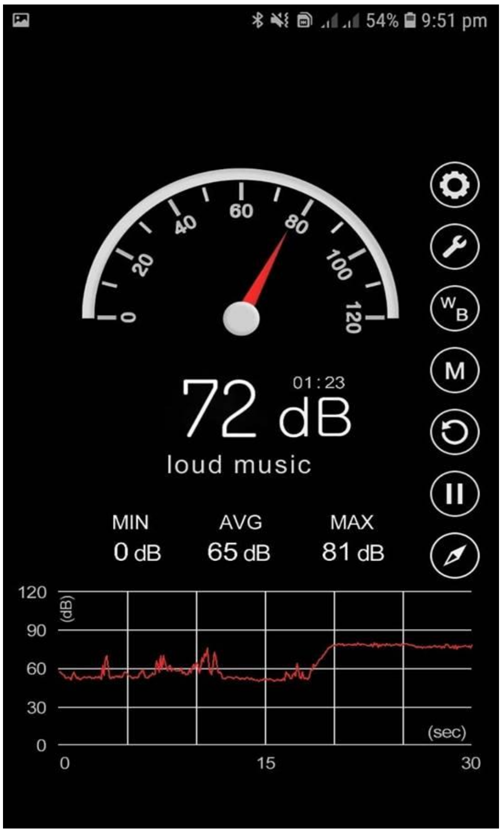
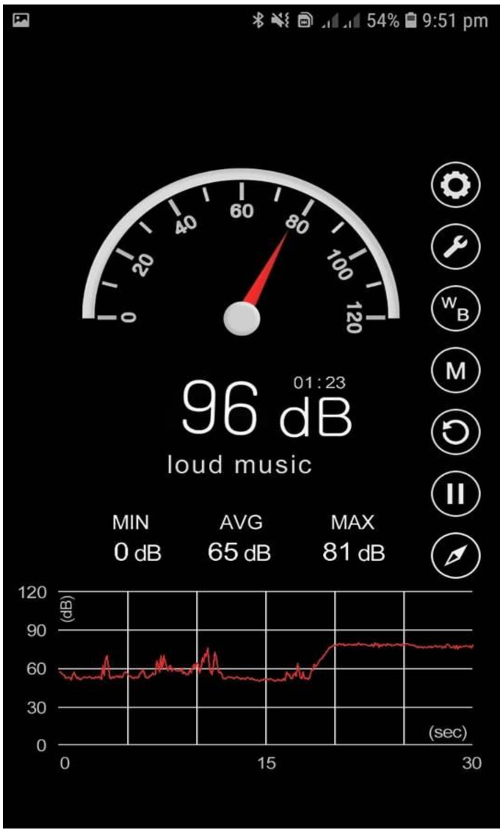
5.2 Result:

Sound Range (Normal Condition) dB	Sound Range (Within Enclosure) dB	Percantage of reduction
96	72	25%
82	60	27.55%
76	53	30%
52	38	26.92%
46	33	28.20%



WITHIN ENCLOUSURE



IN NORMAL CONDITION

5.3 CONCLUSION:

5.3.1 Impacts

The noise level of the generators in a high-density area is presented in. The minimum and maximum measurements obtained were 90.50 dB and 117.40 dB, respectively, while an average/mean noise measurement of 97.5967 dB was obtained. This is at variance with a noise level of 89 dB obtained. Also, the value obtained is above the World Health Organization (WHO) standard for residential areas. This could be detrimental to the socioeconomic and health of the residents. So its getting compulsory for us to be aware of the every way we can reduce the amount of noise pollution.

5.3.2Target

We have worked on the basic construction of our sound attenuated enclosure and its working principle. Our next target is to attain an optimized process to manufacture this enclosure in a wide range.

Our environment has already suffered greatly due to noise pollution. In every residential area, they have generators to maintain the continuous supply of electricity. And this is a must feature for these areas to mitigate the noise pollution.

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[13]Optimisation of an active control system to reduce the exhaust noise radiated by asmall generator MarõÂa Cuesta *, Pedro Cobo Instituto de AcuÂstica, CSIC, Serrano 144, 28006 Madrid, Spain