

STUDY AND ANALYSIS OF GENERATOR NOISE AND ITS CONTROL

**A Dissertation
Submitted in partial fulfillment of the requirements for
the award of degree of**

**MASTER OF ENGINEERING
IN
CAD/CAM & ROBOTICS**

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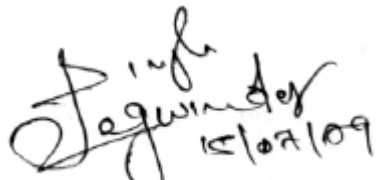


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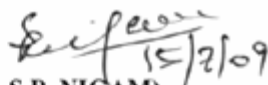
CERTIFICATE

I hereby certify that the work which is being presented in this dissertation entitled as **“Study And Analysis Of Generator Noise And Its Control”** in partial fulfillment of the requirement for the award of degree of **Master of Engineering in CAD/CAM & ROBOTICS** in the Mechanical Engineering Department, **Thapar University, Patiala** is an authentic record of my own work carried out under the supervision and guidance of **Dr. S.P Nigam** and refers other researcher’s work which are duly listed in the reference section.


The matter embodied in this dissertation has not formed the basis for the award of any other degree, associate ship, fellowship or any other similar title to any other university or institute.


(JAGWINDER SINGH)

This is to certify that above declaration made by the student concerned is correct to the best of my knowledge & belief.


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ABSTRACT

Noise levels associated with diesel generators often exceed local codes and city ordinances, not to mention personal comfort levels making noise study a necessity. Noise propagating from the engine, generator, radiator fan, turbo charger and even from the muffler casing produces a combined effect. Diesel generators are used very commonly in shops, offices and homes today in order to supply power during power shutdowns. These generators emit very high levels of noise, in addition to noxious air emissions. Exposure to noise causes detrimental effects on neuroendocrine, cardiovascular, respiratory and digestive systems. Chronic exposure to noise causes fatigue and interferes with concentration, thus reducing work efficiency.

All the effects produced by generator noise depends on acoustic power of the generator .Acoustic power gives the information that how large the source of noise is and also helps to compare the noise radiated by generators of same type and size, whether a machine complies with a specified upper limit of sound emission.

In this measure the Acoustic Power of Diesel Generator was measured with the help of Parallelepiped method after measuring the Sound Pressure Level at 17 Grid points. Further frequency Spectrum was also obtained at three typical locations to obtain the peak frequencies at different load conditions and locations. An attempt has been made in this work to study the effect of various noise controls measures in reducing the Generator Noise.

In this regard an acoustic enclosure was functionally designed and polyurethane form was use to line the surfaces of the enclosure. The effects of adding only silencer, only enclosure, enclosure with silencer, enclosure

with silencer and inlet outlet duct were studied. Thermal consideration taken into account by using a cooling fan and its noise was reduced with help of partial barriers.

Finally noise contour also obtained around the generator with enclosure at six different heights to get an idea of directivity of the noise pattern around the Diesel Generator Set and effectiveness of different noise control measures.

CONTENTS

Description	Page No.
❖ CERTIFICATE	i
❖ ACKNOWLEDGEMENT	ii
❖ ABSTRACT	iii-iv
❖ LIST OF CONTENTS	v-viii
❖ LIST OF FIGURES	ix-x
❖ LIST OF TABLES	xi
❖ CHAPTER 1 INTRODUCTION	1-19
1.1 Sound	1
1.2 Noise	1
1.3 Environmental Noise Pollution	1
1.4 Harmful Effects Of Noise On Human Beings	1-2
1.5 Audible Sound	2
1.6 Characteristics of sound	2-5
1.6.1 Amplitudes	2
1.6.2 Frequency	2
1.6.3 Loudness level	3
1.6.4 Frequency –weighted contours	3-4
1.6.5 Change in noise	4
1.6.6 Background noise	5
1.7 Types of Sound Sources	5-7
1.7.1 Point source	5-6
1.7.2 Line source	6
1.7.3 Plane source	6-7
1.8 Major Sources of Noise	7-8
1.9 Measuring Sound level	9
1.9.1 Steps For Measurement Of Noise	9
1.9.2 Noise Measurement Procedure	9
1.10 Frequency Analyzer	10-12
1.11 Noise Control	12-14
1.12 Noise Control Technique	14-16
1.12.1 Noise control at the source	14

1.12.2	Control of the transmission path	14-15
1.12.3	Protective –measures at the receiver	16
1.13	NC curves	16-17
1.14	General Properties of Sound Absorptive Material	17-18
1.15	Sound Absorption Coefficient	18
1.16	Selection of Acoustical Materials	18-19
❖	CHAPTER 2 LITERATURE REVIEW	20-31
❖	CHAPTER 3 INTRODUCTION TO DIESEL GENERATORS NOISE	32-40
3.1	Parts of diesel generator set	32
3.2	Types of generators	33
3.3	Engine noise	33
	3.3.1 Combustion noise	34
	3.3.2 Mechanical noise	34-35
	3.3.3 Bearing noise	36
	3.3.4 Exhaust noise	37
3.4	Relation between noise ,engine design and parameter	37-38
	3.4.1 Engine speed	37
	3.4.2 Engine size	38
	3.4.3 Engine load	38
3.5	Generator noise	38-39
3.6	Radiator fan noise	39-40
❖	CHAPTER 4 TECHNIQUES FOR CONTROL OF DIESEL GENERATOR NOISE	41-65
4.1	Different component of noise control	41
	4.1.1 Source	41
	4.1.2 Path.	42
	4.1.3 Receiver	42
4.2	Noise Control Procedure For Archiving An economical Solution	42-45
4.3	Use Of Acoustical Barriers	46-47
4.4	Partial Enclosures	47-49
4.5	Check List Of Noise Control Techniques	49-51
❖	CHAPTER 5 DESIGNING OF ACOUSTIC ENCLOSURE FOR 5KW GENERATOR	52-65

5.1	The use of sound barriers in acoustic enclosure	52
5.2	Rules for use of barriers inside acoustic enclosure	53
5.3	Complete enclosures around a source	53-55
5.4	Reduction of leakage path	55-56
5.5	Use of lined ducts for ventilator opening	56-57
5.6	Material use for fabrication of acoustic enclosure	58
5.7	Specification of polyurethane foam	58-59
5.8	Parts of acoustic enclosure	59-62
5.9	Assembly of designed acoustic enclosure	63-65

**CHAPTER 6 GENERAL PROCEDURE AND
EXPERIMENTAL SETUP 66-74**

6.1	General Procedure For The Measurement of Sound Power	67
6.2	Additional Aspect Of Measurement Of Acoustic Power	68-69
6.3	Appropriate Steps For Sound power Measurements	69
6.4	Advantage Of Measure the Sound Power Level	69
6.5	Measurement Of Sound Power	70
6.6	Measurement Of sound pressure level For Frequency Spectrum	71
6.7	Procedure For Generate Contour Diagrams	72-74

❖ CHAPTER 7 RESULTS AND DISCUSSION 75-119
**❖ CHAPTER 8 CONCLUSIONS AND SCOPE
FOR FUTURE WORK 120-121**

8.1	Conclusion	120
8.2	Scope for Future Work	121

❖ REFERENCES & BIBLIOGRAPHY	122-124
❖ APPENDIXES	125-161
❖ APPENDIX A: Tables for Acoustic Power	126-140
❖ APPENDIX B: Tables for 1/1 octave band frequency spectrum	141-155
❖ APPENDIX C: Tables for Contour	156-161

LIST OF FIGURES

Description	Page No.
Fig. 1.1 Weighting curves	4
Fig. 1.2 Relation between sound level & loudness	4
Fig. 1.3 Back ground noise correction	5
Fig. 1.4 Point source.....	6
Fig. 1.5 line source.....	6
Fig. 1.6 Plane source.....	7
Fig. 1.7 Sound level meter.....	8
Fig. 1.8 Accurate method of sound measurement.....	9
Fig. 1.9 1/1 octave band level.....	11
Fig. 1.10 1/3 octave band level	12
Fig. 1.11 N.C Curves.....	17
Fig. 3.1 Comparison of generator noise with fuel type.....	33
Fig. 3.2 Two pole ac generator.....	39
Fig. 3.3 Comparison of generator noise with & without radiator.....	40
Fig. 4.1 Component of noise control.....	41
Fig. 4.2 Noise control system.....	41
Fig. 4.3 Octave band spectra for noise control system.....	43
Fig. 4.4 Graphical representation of different types of barriers.....	46
Fig. 4.5 Partial enclosures used in machinery noise control	48
Fig. 4.6 Check list of component.....	49
Fig. 4.7 Comparison between right and wrong fitment of components.....	50
Fig. 5.1 Acoustic enclosure with air ventilation.....	52
Fig. 5.2 Complete enclosure around a source.....	54
Fig. 5.3 Graph of comparison in between without lined and lined duct.....	55
Fig. 5.4 Comparison between treated and untreated inlet and outlet duct.....	57
Fig. 5.5 Specification's of polyurethane foam.....	58
Fig. 5.6 Top view of acoustic enclosure with generator.....	59
Fig. 5.7 Front side of Acoustic enclosure.....	60

Fig. 5.8	Left side of Acoustic enclosure.....	60
Fig. 5.9	Right side of Acoustic enclosure.....	61
Fig. 5.10	Front side of Acoustic enclosure.....	62
Fig. 5.11	Normal and Sectional View of Silencer.....	62
Fig. 5.12	Ventilation duct of Acoustic enclosure.....	62
Fig. 5.13-5.17	Complete assembly of Design enclosure.....	63
Fig. 6.1	Location of A,B,C points.....	66
Fig. 6.2	Imaginary hemispherical surface around a noise source.....	68
Fig. 6.3	Imaginary parallelepiped surface around a noise source.....	69
Fig. 6.4	Location of grid point.....	70
Fig. 6.5-6.6	Location of SLM at different locations.....	71
Fig. 6.7	Grid formation for contour generation.....	72
Fig. 6.8	Different planes in Z-direction.....	73
Fig. 6.9	Isimetric view of hypothetical grid for contour generation.....	74.
Fig.7.1- Fig.7.6	graphs of acoustic power at different.....	75-78
Fig.7.7-Fig.7.12	graphs for 1/1 octave band frequency spectrum at location (A).....	82-87
Fig. 7.13	Locations of A,B,C.....	89
Fig.7.14 -Fig.7.20	graphs for 1/1 octave band frequency spectrum at location (B).....	91-98
Fig.7.21- Fig.7.26	graphs for 1/1 octave band frequency spectrum at location(C).....	99-104
Fig.7.28 to Fig 7.39	Contour Graphs.....	107-118

LIST OF TABLES

Description	Page No.
Table – 1.1 Relative frequency of Sound Level Meter.....	11
Table – 4.1 NC curves.....	44
Table – 5.1 Effectiveness of partial enclosure.....	56
Table – 5.2-Fig. 5.3 Specific data of Generator use in experiment.....	57
Table - 5.4 Specifications of Polyurethane.....	58
Table - 7.1 Results of acoustic vs load.....	79
Table - 7.2 Results of 1-1 octave band at location A.....	89
Table - 7.3 Results of 1-1 octave band at location B.....	96
Table - 7.4 Results of 1-1 octave band at location C.....	104

CHAPTER – 1 INTRODUCTION

This chapter discusses the basic details and concepts of related topics on noise associated with the thesis work.

1.1 SOUND [28]

Sound is produced as result of some mechanical disturbance creating pressure variations in an environment such as air or water, or in fact any medium which can transmit a pressure wave. To be able to hear the sound there must always be air or other medium at the ear. The magnitude of the pressure variations (the amplitude of the pressure oscillation) is proportional to the loudness of the sound.

1.2 NOISE

Noise is sound, while under some circumstances sound is noise. The sound from our record player or TV set made for our own pleasure appears on our neighbor's side of the wall as noise. Noise is conveniently and concisely defined as "unwanted sound.

1.3 ENVIRONMENTAL NOISE POLLUTION

- Serious Threat to Quality of Life.
- Permanent Part of Human Environment.
- Increasing Noise Levels in Industries & Urban Environment cause
- Severe Nuisance.
- Annoyance.
- Occupational Health Hazard to Industrial Worker.

1.4 HARMFUL EFFECTS OF NOISE ON HUMAN BEINGS

- Reduces work efficiency.
- Affects the speech communication.
- May cause temporary threshold shift (TTS)/permanent threshold shift (PTS)

- Induces loss of hearing ability.
- Causes psychological strain and mental fatigue.
- May damage the heart.
- Increases the cholesterol level in the blood.
- Dilates the blood vessels of the brain.
- Upsets the chemical balance of the body.
- Causes headache, nausea and general feeling of uneasiness.
- Induces errors in 'motor' performance, in visual perception and in distance and size evaluations.
- Induces psychosis and acute mental agony.

1.5 AUDIBLE SOUND

The minimum audible sound is referred to as the threshold of hearing of the individual subject, and this varies considerably for individuals according to age and past exposure to noise. Although it is difficult to define normal hearing, the best realizable definition is provided by the average over a large group of people for whom no defect is expected.

1.6 CHARACTERISTICS OF SOUND

1.6.1 AMPLITUDE

Amplitude is a direct measure of the magnitude, or loudness, of a sound without consideration for other factors that may influence its perception. The ranges of sound Pressures that occur in the environment are so large that they are expressed on a Logarithmic scale. The standard unit of measurement of sound is the decibel (dB).

1.6.2 FREQUENCY

The frequency of sound is expressed as Hertz (Hz) or cycles per second. The normal audible frequency range for young adults is 20 Hz to 20,000 Hz. The

prominent frequency range for community noise, including aircraft and motor vehicles, is between 50 Hz and 5,000 Hz.

1.6.3 LOUDNESS LEVEL

This scale has been devised to approximate the human subjective assessment to the “Loudness” of a sound. Loudness is the subjective judgment of an individual as to how Loud or quiet a particular sound is perceived. This sensitivity difference varies for different sound pressure levels.

1.6.4 FREQUENCY-WEIGHTED CONTOURS (dB(A), dB(B), dB(C))

In order to simplify the measurement and computation of sound loudness levels, Frequency-weighted networks have obtained wide acceptance. The equal loudness level Contours for 40 dB, 70 dB, and 100 dB have been selected to represent human frequency Response to low, medium, and loud sound levels. By inverting these equal loudness level contours, the A-weighted, B-weighted, and C-weighted frequency weightings were developed. Figure presents these frequency-weighted contours. The most common weighting is the A-weighted noise curve. The A-weighted decibel scale dB(A) performs this compensation by discriminating against frequencies in a manner approximating the sensitivity of the human ear. In the A-weighted decibel, everyday sounds normally range from 30 dB(A) (very quiet) to 100 dB(A) (very loud). Most community noise analyses are based upon the A-weighted decibel scale. Figure presents examples of various sound environments expressed in dB(A). Some interest has developed by communities close to some airports in utilizing a noise curve other than A-weighting for lower frequency noise sources. For example, the C weighted curve is used for the analysis of the noise impacts from artillery noise. For evaluation of aircraft noise, A-weighting is used because the majority of noise associated with aircraft operations is better suited to the A-weighting; no mitigation methods have been proven to be effective for C-weighted noise (i.e., sound insulation), which is the minority portion of the noise associated with aircraft operations.

Basic Sound Analysis – Frequency Filtering

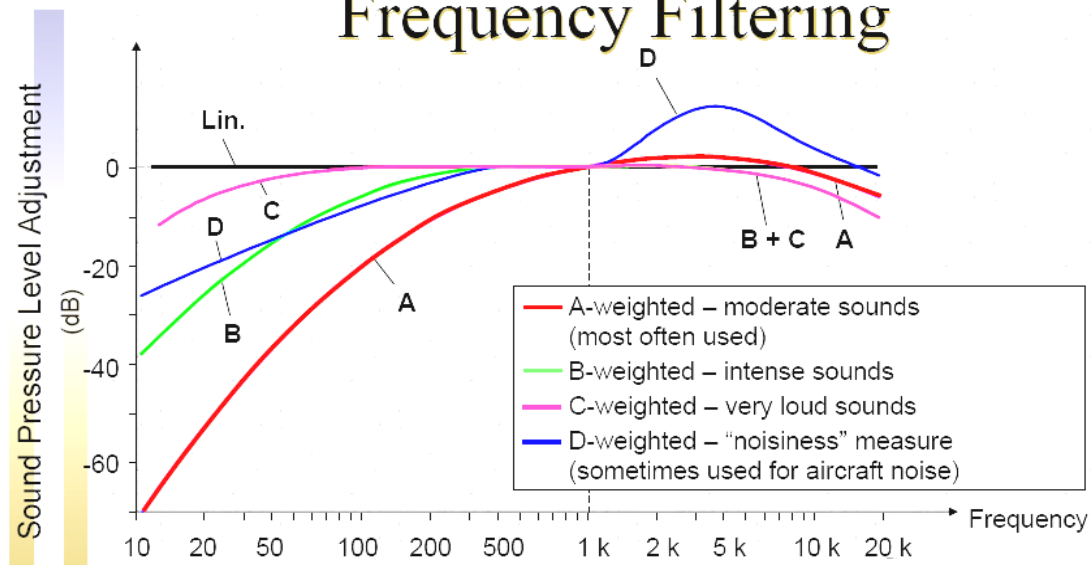


FIG 1.1 FREQUENCY-WEIGHTED CONTOURS (dB(A), dB(B), dB(C))

1.6.5 CHANGE IN NOISE

The concept of change in ambient sound levels can be understood with an explanation of the hearing mechanism’s reaction to sound. Under controlled laboratory conditions, listening to a steady unwavering pure tone sound that can be changed to slightly different sound levels, a person can just barely detect a sound-level change of approximately 1 dB for sounds in the mid-frequency range. When ordinary noises are heard, a young healthy ear can detect changes of 2 to 3 dB. A 5 dB change is readily noticeable, while a 10 dB change is judged by most people as a doubling or halving of the loudness of sound.

Sound Amplitude – Loudness

Change in Sound Level (Δ dB)	Change in Loudness
1 to 3 dB	“Just perceptible”
5 dB	“Noticeable” change
10 dB	“Twice” (or $\frac{1}{2}$) as loud
15 dB	“Large” change
20 dB	“Four times” (or $\frac{1}{4}$) as loud

FIG 1.2 RELATION BETWEEN SOUND LEVEL & LOUDNESS

1.6.6 BACKGROUND NOISE

When sound measurement on for instance a machine is carried out, it is important that the background noise level is so low, that it does not have any influence on the result. This can be tested in the following manner. Measure the sound at the position where it should be measured with the source (machine) running. Switch off the machine and measure the sound level without the machine running. If the difference is less than 3dB measurements should be stopped until the background noise has been reduced. If the difference is between 3 and 10 dB use the curve to correct the measured value. If the difference is more than 10 dB, the background noise may be ignored.

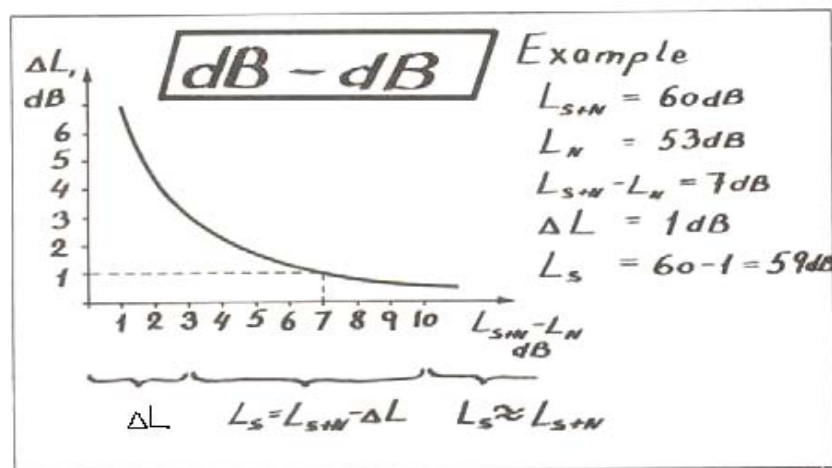


FIG 1.3 REPRESENTING METHOD OF BACKGROUND NOISE CORRECTION

1.7 TYPES OF SOUND SOURCES

POINT SOURCE

LINE SOURCE

PLANE SOURCE

1.7.1 POINT SOURCE

A sound source can be considered as a point source, if its dimensions are small in relation to the distance to the receiver and it radiates an equal amount of energy in all directions. Typical point sources are industrial plants, aircraft and individual road vehicles. The sound pressure level decreases 6 dB whenever the distance to a point source is doubled.

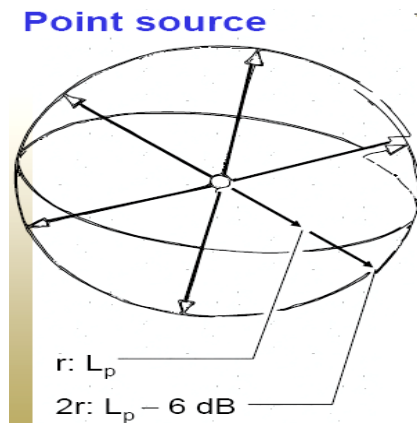


FIG 1.4 POINT SOURCE

1.7.2 LINE SOURCE

A line source may be continuous radiation, such as from a pipe carrying a turbulent fluid, or may be composed of a large number of point sources so closely spaced that their emission may be considered as emanating from a notional line connecting them. The sound pressure level decreases 3 dB, whenever the distance to a line source is doubled.

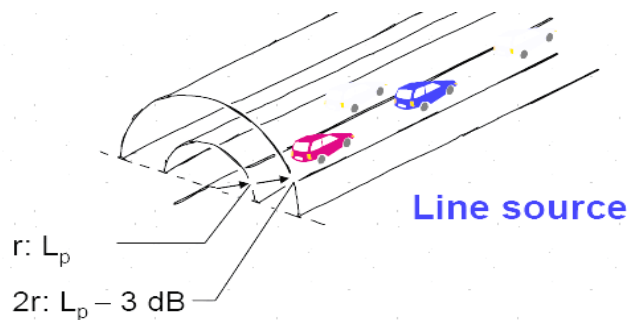


FIG 1.5 LINE SOURCE

1.7.3 PLANE SOURCE

A plane source can be described as follows. If a piston source is constrained by hard walls to radiate all its power into an elemental tube to produce a plane wave, the tube will contain a quantity of energy numerically equal to the power output of the source. In the ideal situation there will be no attenuation along the tube. Plane sources are very rare and only found in e.g. duct systems. When 2 sources radiates sound energy, they will both contribute to the sound pressure level a distance away from the sources. If they radiate the same amount of energy and the distance from the point of

measurement to the sources is the same, the level will increase by 3 dB compared with the level created by one source alone.

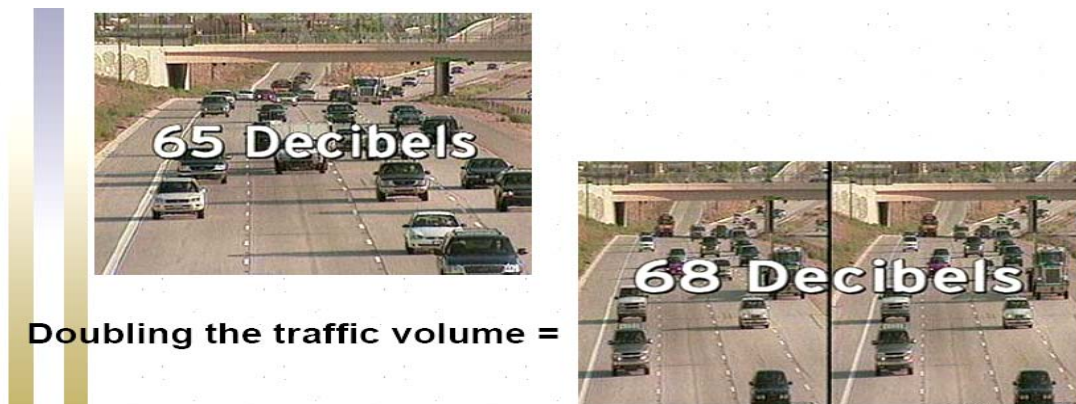


FIG 1.6 PLANE SOURCE

1.8 MAJOR SOURCES OF NOISE

(a) ROAD TRAFFIC NOISE

In the city, the main sources of traffic noise are the motors and exhaust system of autos, smaller trucks, buses, and motorcycles. This type of noise can be augmented by narrow streets and tall buildings, which produce a canyon in which traffic noise reverberates.

(b) AIR CRAFT NOISE

Now-a-days, the problem of low flying military aircraft has added a new dimension to community annoyance, as the nation seeks to improve its nap-of-the-earth aircraft operations over national parks, wilderness areas, and other areas previously unaffected by aircraft noise has claimed national attention over recent years.

(c) NOISE FROM RAILROADS

The noise from locomotive engines, horns and whistles, and switching and shunting operation in rail yards can impact neighboring communities and railroad workers. For example, rail car retarders can produce a high frequency, high level screech that can reach peak levels of 120 dB at a distance of 100 feet, which translates to levels as high as 138, or 140 dB at the railroad worker's ear.

(d) CONSTRUCTION NOISE

The noise from the construction of highways, city streets, and buildings is a major contributor to the urban scene. Construction noise sources include pneumatic hammers, air compressors, bulldozers, loaders, dump trucks (and their back-up signals), and pavement breakers.

(e) NOISE IN INDUSTRY

Although industrial noise is one of the less prevalent community noise problems, neighbors of noisy manufacturing plants can be disturbed by sources such as fans, motors, and compressors mounted on the outside of buildings. Interior noise can also be transmitted to the community through open windows and doors, and even through building walls. These interior noise sources have significant impacts on industrial workers, among whom noise-induced hearing loss is unfortunately common.

(f) NOISE IN BUILDING

Apartment dwellers are often annoyed by noise in their homes, especially when the building is not well designed and constructed. In this case, internal building noise from plumbing, boilers, generators, air conditioners, and fans, can be audible and annoying. Improperly insulated walls and ceilings can reveal the sound of amplified music, voices, footfalls and noisy activities from neighboring units. External noise from emergency vehicles, traffic, refuse collection, and other city noise can be a problem for urban residents, especially when windows are open or insufficiently glazed.

1.9 MEASURING SOUND LEVEL

The sound pressure level can be measured with a Sound Level Meter, which basically contains the following 5 blocks.

1. Microphone 2. Amplifier 3. Rectifier 4. Smoothing circuit 5. Meter



FIG 1.7 SOUND LEVEL METER

1.9.1 STEPS FOR MEASUREMENT OF NOISE

1. Check the Sensitivity (Calibration) Of the Measurement System
 - Before and after each measurement.
 - It is a legal requirement.
2. Measure the acoustical noise level
3. Apply all necessary correction to the observed measurement
 - Correction for Back Ground Noise.
 - Correction for reflection of nearby surfaces
 - Correction for ambient pressure

1.9.2 NOISE MEASUREMENT PROCEDURE

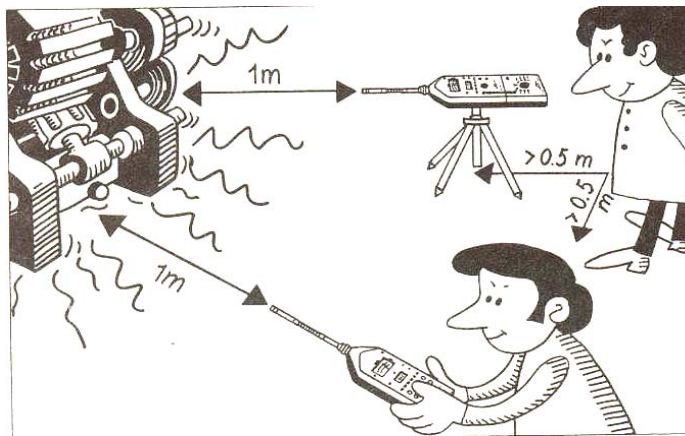


FIG .1.8 REPRESENTING ACCURATE METHOD OF MEASUREMENT THE SOUND LEVEL

- SLM should be at least at a distance of 0.5 m from the body of the observer.
- Reflections from the body of the observer can cause an error of up to 6 dB at frequencies around 400 Hz.
- SLM should be at a height of 1.2 –1.5 m from the floor level.
- Position from near buildings and windows is 1 –2 m away.
- Outdoor measurements to be made at least 3.5 m away from other reflecting structure.
- Within the room measurement should be made in the Free Field zone.

1.10 FREQUENCY ANALYZER

All non-sinusoidal signals are composed of two or more sinusoidal signals. The non-sinusoidal signal can be represented in either the time domain as a function of time or in the frequency domain, where the individual frequency components are represented on a frequency scale. A noise signal will contain signals of all frequencies, or at least a broad spectrum of frequencies.

When a sound signal is investigated, it is often desirable to investigate a limited part of the frequency spectrum. This can be done with the aid of a filter which will allow passage of only that part of the spectrum which lies inside the bandwidth (Δf) of the filter. A practical filter however will not have such a steep cut-off and the usual filter characteristic is together with the characteristic for an ideal filter.

The bandwidth (Δf) of the filter can be defined as the frequency range between the points, where the filter characteristic shows a reduction of 3dB, or the frequency range of an ideal filter, which would allow the same amount of power of a signal containing all frequencies to pass. The difference between the bandwidth found using these two definitions is for most filters very small. It is common to classify a filter according to its bandwidth, and there are two classes of filters which may be encountered i.e. constant bandwidth filters and filters with a constant percentage bandwidth.

(1) OCTAVE-BAND LEVEL

An important measure of noise is its frequency distribution. Instruments used to measure the distribution of sound over the audible frequency range, called spectrum analyzers. The spectrum analyzer which is in most common use divides the audible frequency range into bands one octave wide (An octave is a frequency interval between two sounds whose frequency ratio is 2, e.g., from 707 to 1414 Hz.) Such an analyzer is called an octave-band analyzer. The sound pressure level within a band which is one octave wide is called the octave band sound pressure level, or simply octave-band level. (In the frequency column of Table 1.1, the numbers shown in boldface are the center frequencies of One-third-bands.)

Frequency, Hz	A-weighting, dB
25	-44.7
31.5	-39.4
40	-34.6
50	-30.2
63	-26.2
80	-22.5
100	-19.1
125	-16.1
160	-13.4
200	-10.9
250	-8.6
315	-6.6
400	-4.8
500	-3.2
630	-1.9
800	-0.8
1000	0.0
1250	+0.6
1600	+1.0
2000	+1.2
2500	+1.3
3150	+1.2
4000	+1.0
5000	+0.5
6300	-0.1
8000	-1.1
10,000	-2.5

TABLE 1.1 RELATIVE FREQUENCY RESPONSE OF A SOUND LEVEL METER WITH A-WEIGHTING

Above table shows an example of a plot of octave-band level as a function of frequency. These data represent measurements on a blower driven by a motor. Such a plot of the different octave-band levels vs. frequency is called an octave-band spectrum. The actual experimental data are represented by the plotted point. Connecting lines are useful to indicate the general shape of the spectrum.

(2) ONE-THIRD-OCTAVE-BAND LEVEL

When more detailed information is required than that provided by an octave-band analysis, a one-third-octave-band analysis may be employed. Figure 1.10 shows an example of a plot of noise measurements by one-third-octave bands. These data are for the same blower and motor whose octave-band spectrum is shown in Fig. 1.9 the measurement conditions are the same.

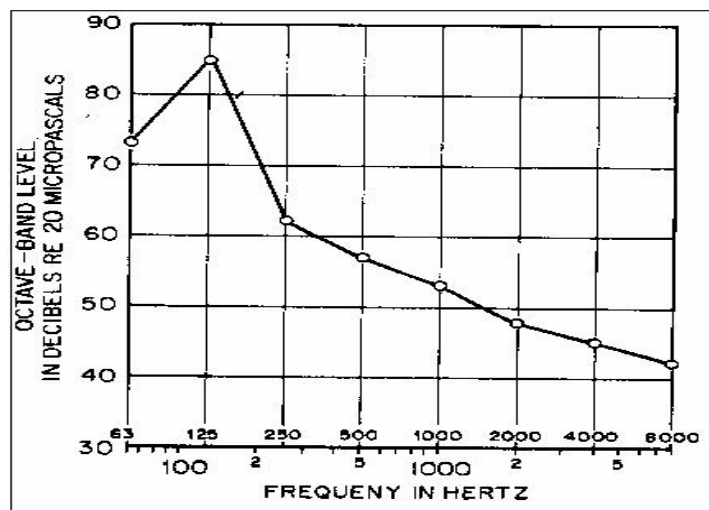


Fig 1.9

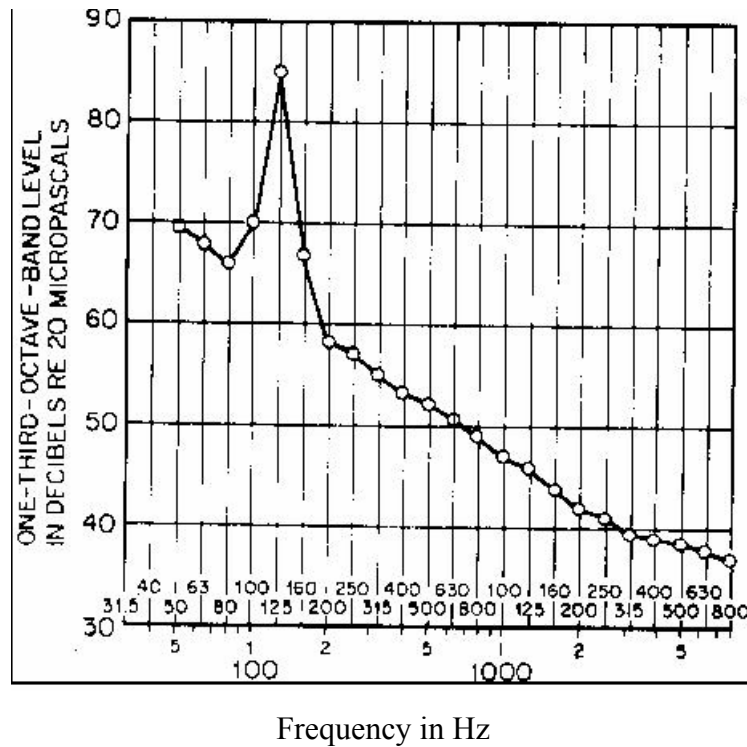


Fig 1.10 1/3 octave band frequency level

COMBINING LEVELS

Often it is necessary to combine levels, for example:

1. To calculate the sound level that results from a combination of sources of noise.
2. To determine the combined sound level of a source plus background noise.
3. To calculate the overall sound pressure level from octave-band levels or one-third-octave-band levels.
4. To combine the A-weighted sound level for a given octave-band spectrum.
5. To combine the sound power level of two or more sound sources.

1.11 NOISE CONTROL

Noise control is the technology of obtaining an acceptable noise environment, consistent with economic and operational considerations. The acceptable environment may be required for an individual, a group of people, an entire community, or a piece of equipment whose operation is affected by noise. Noise control is not the same as noise reduction. In a specific problem, the amount of noise reduction required to achieve acceptable results sometimes may be obtained simply by applying all the various noise-reduction techniques listed in a following section. The problem should be analyzed systematically to determine.

How acceptable conditions might be achieved in the most economical way. In unusual cases the solution to some noise control problems may even suggest a noise increase, rather than a noise reduction. Consider, for example, the waiting room in a physician's office that is separated from the consultation room by a partition which provides so little sound insulation that private conversation can be overheard in the waiting room. Acceptable conditions in the waiting room could be achieved by the construction of a partition providing greater airborne sound

Insulation. A possible alternative solution is to increase the noise level in the waiting room by installing another noise source there (for example, a fan) so as to mask the conversation that would otherwise be overheard. While this latter solution has its disadvantages, it is much more economical and therefore may be more desirable under some circumstances. It illustrates once again that noise control and noise reduction are not always synonymous.

2 HOW MUCH NOISE REDUCTION IS REQUIRED?

The following steps are taken to determine the amount of noise reduction required for a specific problem:

1. EVALUATE THE NOISE ENVIRONMENT, UNDER EXISTING OR EXPECTED CONDITIONS.

Existing conditions may be evaluated from noise measurements which furnish data that are statistically significant. This process requires the appropriate selection and use of measurement equipment, accurate calibration, the taking of data under properly controlled conditions, and the evaluation of any environmental factors which affect the measurements. Under some conditions it is impractical or impossible to evaluate existing conditions. In such cases, or where the noise environment must be estimated for expected or future conditions, an estimate must be made either from empirical engineering formulas or from existing data.

2. DETERMINE WHAT NOISE LEVEL IS ACCEPTABLE

This information is provided by an appropriate criterion. A criterion may be defined as a standard or rule for judging, such a standard may be used, for example, for establishing an acceptable limit or restriction that is to be imposed. Noise control

criteria provide standards for judging the acceptability of noise levels under various conditions and for various purposes.

3. The difference, between the levels in Steps 1 and 2 represents the noise reduction that must be provided to obtain an acceptable environment. This difference usually is determined as a function of frequency.

1.12 NOISE-CONTROL TECHNIQUES

Noise control measures may be classified in three categories:

- (1) noise control at the source
- (2) Noise control of the transmission path.
- (3) The use of noise protective measures at the receiver

1.12.1 NOISE CONTROL AT THE SOURCE

One important method of controlling noise at the source is to reduce the amplitude of the forces which result in the generation of noise, for example, by balancing rotating masses or by isolating vibrating components of the source. Another method is to reduce the motion of the components which are set into vibration; for example, the vibration of panels which are set into vibration maybe reduced by application of vibration-damping materials or by alteration of the resonance frequencies of the panels.

Changes in the usual procedure of operation may be an effective noise control technique. Thus some factories, adjacent to residential areas, suspend or reduce noise operations at night, when the normal activity in a community diminishes and the ambient noise level in the community is decreased. Without this ambient noise to mask it, the factory noise becomes more noticeable. Because of this and possible interference with sleep, factories that would otherwise operate on a 24 hour-a-day basis may curtail their operations at night.

1.12.2 Control of the Transmission Path

Another general technique of noise reduction is that of controlling the transmission path so as to reduce the energy that is communicated to the receiver. This may be done in a number of ways:

SITING

In the open air, maximum attenuation should be provided by increasing insofar as possible as possible—the distance between the source and the receiver. Since many noise sources do not radiate uniformly in all directions, by altering the relative orientation of the source and receiver a considerable reduction in noise level at the receiver may be possible. Thus the orientation of an airport runway may be an important consideration in reducing noise in an adjacent community. Where possible, a site should be chosen that will take advantage of the natural terrain to provide additional shielding of the receiver from the source.

BUILDING LAYOUT

The careful planning of the location of rooms within a building, with respect to the relative position of the noise sources and those areas in which quiet conditions are desired, may result in considerable economy by reducing the extent of the noise control measures that otherwise would be required.

BARRIERS

Barriers in the open air can be effective when they are large in size compared with the wavelength of the sound to be deflected. For example, barriers which make an angle of 45° with respect to the horizontal have been used in the noise field of jet aircraft engines to reflect the high frequencies toward the sky.

ENCLOSURES

Considerable attenuation may be provided by the use of a properly designed enclosure around the noise source or around the receiver.

ABSORPTION

One of the most effective means of attenuating sound in its transmission path is by means of absorption. Suppose a number of machines are in operation in a large office. Most of the noise from these sources that reaches workers on the opposite side of the room is reflected by the ceiling, walls, and floor.

Such absorption also reduces the level of the sound which reaches the workers after a multiplicity of reflections from the walls, ceiling, and floor.

1.12.3 PROTECTIVE MEASURES AT THE RECEIVER

The following noise-control techniques may be employed where the noise level at the receiver is excessive:

EAR PROTECTION DEVICES

Ear plugs, ear muffs, and helmets provide an economical means of reducing the noise exposure of industrial workers.

BOOTHS

In many cases it is impractical or uneconomical to reduce the noise level to which a worker is exposed: it is better to provide a booth or partial or partial enclosure to the worker.

1.13 NC CURVES

The family of NC curves¹ illustrated in Fig. 1.11 is widely used, both as a means of evaluating noise in an existing HVAC system and for defining design goals for a system to be built- NC curves specify the octave-band limits of the permissible noise spectrum that the system may produce.

In evaluating noise problems, NC curves are used as a reference point from which noise reduction requirements can be determined. This rating system is based on the premise that two factors have the greatest influence on how people respond to HVAC background noise in their own environment: the loudness of this background noise and its spectrum shape and level in the speech frequency region. This rating system assumes that the potential for complaints to occur is minimized in a speech communication environment acceptable to the occupant, if the numerical loudness level of the noise does not exceed the numerical speech interference level.

The choice of which NC curve to use as a design criterion for background noise control depends on the type of space use being considered. In an office environment, the desired speech communication (or speech masking) potential is a primary consideration. In contrast, in a concert hall environment the quality of speech communication is not a governing factor; instead, the objective is to avoid masking the faint; pianissimo passages that occur in most musical scores.

The octave-band values for specified NC curves are sometimes considered as levels not to be exceeded, irrespective of the shape of the actual background

spectrum. It has even become common practice to assign an NC rating to a noise, in accordance with the highest NC curve tangent to the spectrum- There are two steps in this tangent-contour method of noise rating:

Step I: Plot the noise spectrum to be rated on a "field" of NC curves similar to that illustrated in Fig. 1.11.

Step 2: Determine the highest NC curve tangent to the noise spectrum, and use this NC curve number to assign the noise rating.

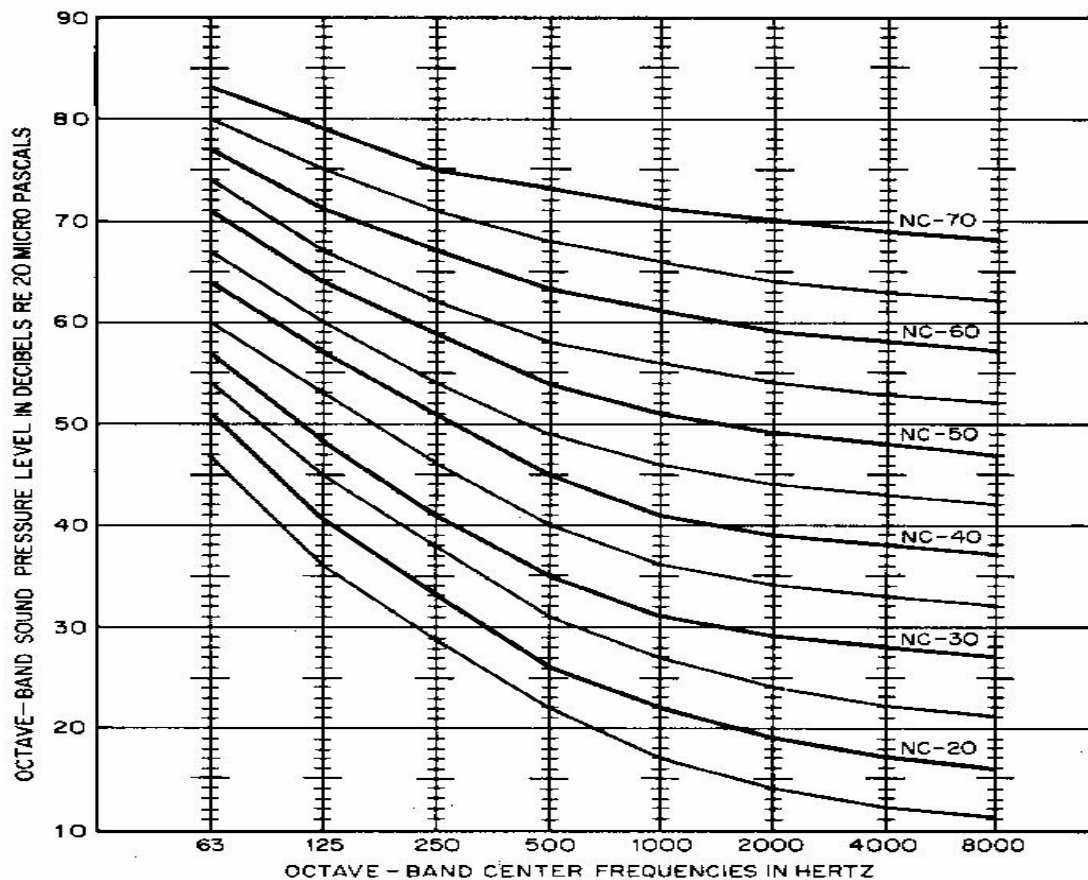


FIG 1.11

1.14 GENERAL PROPERTIES OF SOUND-ABSORPTIVE

(1) MATERIALS

When sound waves strike the surface of a material, a fraction of the incident energy is absorbed by conversion to heat. All materials absorb sound to some extent acoustical materials are those materials whose primary function is to absorb sound. Therefore, they absorb a large fraction of the acoustical energy which strikes them.

The element which accounts for the dissipation of sound energy in most acoustical materials is a layer of highly porous material [at least $1.27 \text{ cm } C^{-2}$ in) thick] in which the pores intercommunicate throughout. The pores may be formed by felted mineral or fiberglass, by the interstices between small granules, or by a foamed composition in which the solidified bubbles interconnect throughout the material. When a sound wave enters a porous material, the amplitude of vibration of the air molecules is progressively damped out by friction against the surfaces of the fibers or particles forming the porous structure. This friction acts as an acoustical resistance whose value depends on the resistance of the material to direct airflow; such friction depends only slightly on the frequency.

Another factor which affects sound absorption, principally in the low frequency range, is the depth of the space between the face of the material and a rigid backing surface behind it. The volume of air between these two surfaces includes both the air in the pores of the material and any airspace between the material and its backing. The latter may vary from zero, when the material is secured directly to a rigid backing, to 1 m (3 ft) or more in the case of suspended acoustical ceilings. When the total depth is less than about one-fourth wavelength, the low-frequency absorption coefficient of the material decreases with decreasing frequency.

1.15 SOUND ABSORPTION COEFFICIENTS

The sound absorption coefficient of a material is a measure of the sound absorptive property of a material. It is the fraction (expressed as a decimal number) of the randomly incident sound power which is absorbed or otherwise not reflected by the material. For example, a sound absorption coefficient of 0.65 indicates that 65 percent of the incident acoustical energy which strikes a material is absorbed. The sound absorption coefficient of every material varies with frequency. It is common practice to list the coefficients of a material at six frequencies: 125, 250, 500, 1000, 2000, and 4000 Hz.

1.16 SELECTION OF ACOUSTICAL MATERIALS

A number of other properties or considerations must be considered in the selection of an acoustical material, including:

1. Flame spread and fire endurance

2. Mechanical strength, abuse resistance
3. Dimensional stability
4. Light reflectance
5. Sound attenuation
6. Maintenance, cleanability, paintability
7. Appearance
8. Cost
9. Ease of installation, method of mounting
10. Space availability for acoustical installation
11. Weight of acoustical installation
12. Compatibility with other materials and components

CHAPTER-2 LITERATURE REVIEW

W. D. Bryce et al [1] To assist in the identification and understanding of the noise sources that contribute to the exhaust noise of aircraft gas turbine engines, controlled experiments have been carried out to study the noise characteristics of a model turbo-jet exhaust system. The noise data have related to measurements of the aerodynamic conditions in the model and, with the aid of specific diagnostic tests, the predominant noise mechanisms are considered to have been recognized. The noise radiation, above that of the jet, is attributed primarily to dipole sources generated by the turbine outlet struts, the transmission of this noise being modified by duct propagation and nozzle impedance effects.

J. Stephenson and H A. Hassan [2] calculated the energy release rate resulting from the combustion of propane-air mixtures is presented and the result is used to calculate the far field noise spectrum for an open flame by using appropriate Fourier transform techniques. The results illustrate the broad band nature of combustion noise and show that, for the range of parameters indicated, the peak frequency in the f octave band is in the range 400-1 000 Hz. The results also indicate that the shape of spectrum is influenced by the time history of the heat release rate and the turbulence intensity and length scales; on the other hand, the peak frequency is a function of the heat release per unit mass of fuel which is essentially the same for hydrocarbon fuels.

Andrian David Jones [3] experimental study of radiated exhaust noise from a single cylinder, piston ported 2- stroke engine. Part 2 consists of a study of noise sources on a rotary 2-stroke lawnmower.

In part 1, a detailed study of the gas dynamics of the exhausting process in a 2-stroke and the associated sound power radiated by the exhaust of the engine has been made. The exhaust systems considered include straight pipes of lengths 0.4m and 1.3m and a tuned expansion chamber of length 1.38m. Measurements show the significance of non-linear behavior which results in wave steepening and shock wave formation. A series of measurements of far field SPL and associated pressure at several locations in the exhaust pipe for a different exhaust systems. These results then compared with theoretical predictions obtained by calculating, using the method

of characteristics, the detailed unsteady flow in the exhaust pipe matched to the flow out of the engine cylinder, for several engine cylinders. The calculations include entropy characteristics and therefore allows for the significant variations in the entropy, arising mainly from the variable shock strength at the exhaust port, which occur in the system. Of particular interest is the close agreement between the calculated third-octave radiated sound pressure spectra and the measured spectra, for both the straight pipes and the tuned expansion chamber exhaust systems.

In part 2, the problem of rotary 2-stroke lawnmower noise is initially considered in terms of the four components parts: inlet, exhaust and mechanical and blade noise. Of these, all but inlet noise are found to be significant at moderate engine speeds, with blade noise dominating at high speeds. Detailed investigations, experimental and theoretical, into the nature of exhaust, mechanical and blade noise are then described. In particular the mechanical noise is investigated quite thoroughly with the analysis made of piston slap. Finally, recommendations are made for design changes which would lead to the reduction of the major components of the total noise.

Tetsuru Oguchi [4] studied the piston slap noise, which is generated by the sudden impact of the piston on the cylinder wall, is the most predominant mechanical noise emitted from the engine. It is studied here in terms of its transmission paths through the engine elements. Two engines with different cylinder liners types are tested in a non-running condition. The transmission paths are measured and evaluated in terms of the mechanical mobility, the ratio of the velocity response on the cylinder wall to the force applied to the external surface of the engine, which represents, by reciprocity, the ratio of the velocity response on the external surface of the engine to the force applied to the cylinder wall. The work concentrates on the major transmission paths of the piston slap noise and the factors which affect that transfer mobility in the frequency range of 1.5 to 4 kHz.

The major transmission path of the piston slap noise is experimentally determined to be the path through the cylinder wall and the upper deck of the cylinder block to the external surface of the cylinder block.

E. J. Richards, M. E. Westcitt and R. K. Jeyapalan [5] studied the noise generation by impacting bodies due to the high surface accelerations during the contact period. An account is presented of the theoretical development and experimental validation of

curves for the prediction of peak sound pressure and radiated energy for collisions of compact bodies which are incapable of flexural motions. It is shown that acceleration noise energy is of the same order of magnitude as that due to ringing, that it cannot be greater than 1.5×10^4 times the kinetic energy input at impact and that it falls off rapidly as the normalized contact time increases above a critical value.

A.D. Jones and G.L. Brown [6] used a computational technique for the method of the characteristics solution of 1-D compressible, unsteady flow in the duct to the wave action in engine exhaust system. By using the method it was possible to compute the detailed flow in both straight pipe and tuned expansion chamber exhaust system as matched to flow the cylinder of a small two stroke engine. The radiated exhaust noise was then determined by assuming monopole radiation from tailpipe outlet. Experiment on an operating engine has been achieved with the calculation of both the third octave radiated noise and associated pressure cycles at several locations in the different exhaust systems. Its essential feature is the computation of the precise paths on the x-t plane of a finite number of C+, C- and P characteristics, to provide high accuracy in determining the tailpipe outlet velocity and hence radiated noise.

J.M. Cuschierit and E.J. Richards [7] studied the noise radiated from I.C. engine due to combustion and piston slap excitation is investigated by considering single impacts. From the results obtained, possible methods of noise control are studied, and the expected results due to changes in the liner mounting to the engine frame, and the bearings of the camshaft for an injected engine, are compared to measured noise levels. This proves to be very successful and radical modifications in the engine for noise control can be investigated in this way prior to full development of the prototype engine.

V. Tandara [8] studied the radiator fan noise. The combustion engine is only one of many vehicle noise sources. Every combustion engine has inner and external noise sources. The cooling fans can be important noise sources. They are installed to cool the engine, encasement and the inside of the car. The influence of fans is great in case of high ambient temperature, low traveling speed and frequent stoppages.

P. Schachner, W. Reisinger [9] studied the concept of exhaust gas emissions. The design and development of modern internal combustion engines is marked by a reduction in exhaust gas emissions and increase in specific power and torque.

Correspondingly, combustion noise excitation and fuel consumption also have to be reduced. These objectives can be achieved through the development of advanced combustion systems, the increased flexibility of fuel injection systems. However, development of modern combustion systems and vehicle applications has become increasingly complex. Creating an exact, yet straightforward description of combustion noise is a very important task. The customer's subjective impression of the entire vehicle, regarding items such as diesel knock sensitivity, provides evidence to support its value.

L. Desmons and J. Kergomard [10] calculated the noise radiated by the exhaust system of a four cylinder engine analytically and numerically and compared to experiment. The basic idea is as follows: the effect of the exact shape of the volume velocity signal produced by a cylinder during opening can be considered as a second order effect on the result. The amplitude of the every harmonic of the radiated noise is shown to be proportional to the first order of certain quantities, the zeroth order be zero. The analysis of the amplitude of the radiated noise exhibits the role of the resonance of the whole exhaust as viewed from outside when the cylinders are closed, and of the resonances of the manifold. Comparison with experiment confirms the basic hypothesis and the result of the theory which are related to the interference with the parallel tube.

L. Desmons, J. Hardy and Y. Auregan [11] applied a least squares method to characterize an internal combustion engine considered as a noise source. It is shown that, although extremely severe conditions exist (high sound pressure level, high temperatures, turbulent flow, etc.), a linear theory can predict the noise level at the output of the exhaust systems with a surprisingly good accuracy when the transfer matrix is known. The measurements avoid the use of calibrated internal microphones; only one external microphone and a set of calibrated loads are needed. An indirect validation of the linearity hypothesis is achieved and the predicted exhaust noise level is compared with experimental results in the case of a tube and in the case of a silencer composed of three expansion chambers.

Torben Astrup[12] gives an overview of observations from almost 1.5 years of practical experience with acoustic intensity measurements. The often difficult working conditions of acoustic consultants mean that we tend to make demands on

the measuring methods and the instrumentation that presently cannot easily be met. Therefore the instrumentation should be improved in order to increase both the dynamic and the frequency range. Further development of 'real-time' control of measurement quality is also desirable, i.e. the possibility of surveying the quality of the result during or immediately after the measurement is performed.

M.L. Munjal [13] presented an overview of the research findings of the author and his students in different aspects of active as well as passive mufflers. Mufflers have been developed over the last seventy years based on electro-acoustic analogies and experimental trial and error. Passive mufflers based on impedance mismatch, called reflective or reactive mufflers, have been most common in the automobile industry. Mufflers based on the principle of conversion of acoustic energy into heat by means of highly porous fibrous linings, called dissipative mufflers or silencers, are generally used in heating, ventilation and air-conditioning systems. The author has been working in this area for over 16 years.

Tandon N. et al [14] Identified major noise sources in a noisy portable I.C. engine driven generator set. The generator set engine is petrol start and kerosene run. The exhaust silencer of the engine was providing substantial insertion loss. The separation of engine combustion and mechanical noise indicated that their contribution was almost equal. Since generally combustion noise is more than mechanical noise, the main noise sources were considered to be mechanical in nature. Sound intensity measurements were performed to identify major noise sources in the generator set when the exhaust was ducted away. The results of these measurements indicated that the main sources of noise in the generator set are: cooling fan cover, silencer shell, silencer cover and the engine crankcase. Noise control measures were applied to these parts. Sound pressure and power levels were measured before and after the application of noise control measures. Constrained layer damping treatment and stiffening of the cooling fan cover had a combined effect of reducing the sound pressure level by about 3 dB(A). Rigid clamping of the silencer also reduced the noise level. A partial enclosure was designed for the generator set. The partial enclosure reduced the sound pressure and power levels by about 4 and 3.7 dB(A) respectively. There was an increase in the engine cylinder head temperature due to the enclosure, but the increase in temperature was considered to be safe. An overall noise reduction of 8.5 dB(A) was obtained on side 4 of the generator set as a result of the implementation of all the noise

control measures. The noise reduction on the other sides of the generator set was also substantial.

MarôÃa Cuesta. et al [15] Attempted to design and implement a hybrid passive/active system to control the noise radiated by a small generator. Passive control is adored by enclosing the generator with a close, leaky, rectangular box. The measured Insertion Loss is higher than 20 dB above 500 Hz. Special attention is paid to technical aspects such as air refreshing and temperature inside the enclosure. Low frequency noise escapes from the enclosure through air intake and gas exhaust openings. A single-input single-output (SISO) feed forward active noise control (ANC) system, implemented in a commercially available device, is used to reduce the radiated exhaust noise below 500 Hz. The reference signal to the SISO ANC system is supplied by an accelerometer located on the air ®lter case of the generator. The error signal is provided by an electrets microphone along the exhaust pipe. The control source consists of a high temperature loudspeaker, positioned in a side-branch conguration to avoid direct con-tact with the exhaust gas. Some harmonics were attenuated more than 30 dB.

J.O. Odigure. et al [16] Attempt to quiet the engine Noise levels associated with diesel generators often exceed local codes and city ordinances, not to mention personal comfort levels making noise control a necessity. In many cases, an engine exhaust muffler is the only form of noise control utilized in an. However, engine exhaust is only one small component of the overall noise caused by the generator system. Noise propagates from the engine, generator, radiator fan, turbo charger and even from the muffler casing. Numerous problems also arise when a large generator must be mobile. North American Power and Controls is a company known in the industry as a "generator packager." NAP&C brings together and assembles all the components that make up a generator system including the engine, generator, fuel system, enclosure, controls, ventilation, and acoustics in this case, transport. One of NAP&C's customer's is Pacific Bell, a regional telecommunications company, which uses standby power supply systems for their switching facilities. The need arose for a portable generator to be used to maintain power while the primary building power was down for construction or in an emergency power outage. The generator needed to be both mobile and quiet since it would often be used near commercial or residential areas. As a frequent and satisfied customer of Ruskin Sound Control, NAP&C turned to Ruskin

for a solution to their portable generator problem. Working closely with Ruskin engineers, a design, capable of meeting and exceeding the end customers' performance requirements, was engineered and a final design was approved. Ruskin 2" thick acoustical panels were chosen for the enclosure walls and roof. Due to the dynamic nature of placing an engine in a housing mounted on a trailer the design team was faced with several problems not the least of which was the acoustical attenuation requirements. Weight and strength constraints were the most limiting and acoustics became a secondary concern. However, based on manufacturer's sound power ratings for the un-enclosed generator compared to acoustical measurements taken in the field using the Ruskin enclosure under actual operating conditions sound levels were decreased significantly

MarôAa Cuesta et al [17] Attempt is made to control the exhaust noise radiated by an enclosed generator. Applied Acoustics 2000;61(1):83±94) the authors reported a passive/active system to control the exhaust noise radiated by a small generator. Passive control was adored by a steel rectangular enclosure lined with a layer of absorbing material. The enclosure, designed to provide the higher Insertion Loss, supplied attenuation higher than 20 dB above 500 Hz. To reduce the noise below 400 Hz, an active control system was designed with one reference, one error input and one control output. Whilst many harmonics in this frequency band were attenuated, the ANC system was unable to reduce some of them. The aim of this paper is to identify the origin of this lack of attenuation and to alleviate it. An exhaustive analysis of both the transfer function between the secondary source and the error sensor (the plant) and the reference signal has been carried out. A new ANC system with improved performance is described which controls all the harmonics until 300 Hz. # 2001 Elsevier Science Ltd. All rights reserved.

C. Polacsek et al [18] Aimed at reducing rotor–stator interaction modes in fan engines when mounted upstream of the rotor has been studied here. This device complements other active noise control systems currently proposed. The compressor model of the instrumented ONERA CERF-rig is used to simulate suitable conditions. The design of the grid is drafted out using semi-empirical models for wake and potential flow, and experimentally achieved. Cylindrical rods are able to generate a spinning mode of the same order and similar level as the interaction mode. Mounting the rods on a rotating ring allows for adjusting the phase of the control mode so that an 8 dB sound pressure

level (SPL) reduction at the blade passing frequency is achieved when the two modes are out of phase. Experimental results are assessed by a numerical approach using computational fluid dynamics (CFD). A Reynolds averaged Navier–Stokes 2-D solver, developed at ONERA, is used to provide the unsteady force components on blades and vanes required for acoustics. The loading noise source term of the Ffowcs Williams and Hawking equation is used to model the interaction noise between the sources, and an original coupling to a boundary element method (BEM) code is realized to take account of the inlet geometry effects on acoustic in-duct propagation. Calculations using the classical analytical the Green function of an infinite annular duct is also addressed. Simple formulations written in the frequency domain and expanded into modes are addressed and used to compute an in-duct interaction mode and to compare with the noise reduction Obtained during the tests. A fairly good agreement between predicted and measured SPL is found when the inlet geometry effects are part of the solution (by coupling with the BEM). Furthermore, computed aerodynamic penalties due to the rods are found to be negligible. These results partly validate the computation chain and highlight the potential of the wake generator system proposed.

Parvathi, K. et al [19] Experimental studies on the assessment and control of indoor and near-field noise levels due to the operation of a portable power generator were undertaken at the Centre for Environmental Studies (CES), Anna University Portable power generators are used very commonly in shops, offices and homes today I order to supply power during power shutdowns. These generators emit very high levels of noise, in addition to noxious air emissions. Exposure to noise causes detrimental effects on neuro-endocrine, cardiovascular, respiratory and digestive systems. Chronic exposure to noise causes fatigue and interferes with concentration, thus reducing work efficiency.. Noise control was studied employing anti-vibration mounts (made up of rubber, coir, polyurethane foam, thermocole, wool–felt and sand bed) and enclosures (made up of cardboard, thermocole and a sandwich of cardboard and thermocole). The sand beds of 32mm thickness (containing sand particle size 0.5 mm) with an air gap of 5 cm between sandwich enclosure walls, among anti-vibration mounts and enclosures, respectively, were found to be most effective in controlling noise due to the generator operation.

J.O. Odigure et al [20] Attempt is to be made to determine unsafe zone of generation operation.he objects that constitute our living environment have one thing in common

–they vibrate. In some cases such as the ground, vibration is of low frequency, seldom exceeding 100 Hz. On the other hand, machinery can vibrate in excess of 20 KHz. These vibrations give rise to sound – audible or inaudible – depending on the frequency, and the sound becomes noise at a level. Noise radiations from the generator are associated with exploitation and exploration of oil in the Niger-Delta area that resulted public annoyance, and it is one of the major causes of unrest in the area. Analyses of experimental results of noise radiation dispersion from generators in five flow stations and two gas plants were carried out using Q-basic program. It was observed that experimental and simulated model values conform to a large extent to the conceptualized pollutant migration pattern. Simulation results of the developed model showed that the higher the power rating of the generator, the more the intensity of noise generated or produced. Also, the farther away from the generator, the less the effects of radiated noise. Residential areas should therefore be located outside the

Hyeon-Don Ju. et al [21] Experimental study is made for design of the Acoustic enclosure, which is initially layout by rule of thumb, is evolved systematically through numerical reanalysis procedure, based on indirect boundary element method (IBFM) with a commercial acoustic analysis code. Diesel engine generator sets in heavy industry plants and residential/official buildings can cause serious noise problems. In

this paper, a low noise diesel engine generator set is developed through constructing an acoustic enclosure with ventilation duct silencers that electively block the acoustic flow but guarantee good thermal flow. The cooling performance of the acoustically determined enclosing structure is checked and confirmed through numerical heat flow analysis. The acoustic and cooling performances of the developed low noise diesel engine generator set are confirmed by the experiment.

W. Eversman et al [22] To identify and reduce noise radiation mechanisms in claw pole (Lundell) alternators used in automotive applications. Noise of claw pole alternators, generated electromagnetically and structurally radiated, has been the subject of an extensive research program. The goal has been to identify and reduce noise radiation mechanisms in claw pole (Lundell) alternators used in automotive

applications. Two approaches have been followed. In the first, electromagnetic sources of noise have been investigated by lumped parameter and magnetically equivalent circuit modeling and simulation, and by related experimentation. This is the subject of separate papers. The second, concurrent study reported here has investigated machine and mount responses to an electromagnetically generated torque ripple. Modeling and experimentation has led to the conclusion that there exists a high correlation between electromagnetic sources, torque ripple, and radiated noise. Experimentation also has led to the conclusion that noise characteristics of a given machine are substantially altered by modification of the mounting configuration. The work reported here involves modeling, simulation, and experiment to isolate machine dynamic characteristics and mounting geometries which contribute to strong coupling between torque ripple and machine/mount dynamic response. A low- order model of the alternator which includes shaft flexibility, gyroscopic effects, shaft bearing asymmetry, mounting lug geometry, and mounting structure dynamics has been created. The model provides a rapid simulation of dynamic response in the form of a transfer function between torque ripple and mounting forces. Generic studies of a simplified mounting structure coupled to the machine model are presented here. Acoustic testing of several machine configurations on a production mount has been carried out to investigate 36th order noise in three phase machines and 72nd order noise in six-phase machines. Electromagnetic modeling and dynamic response simulations suggest that the six-phase machine is inherently quieter. This is supported by experimental results. A test fixture for the measurement of torque ripple has been developed. Experimentation shows that the relative amplitudes of torque ripple in three- and six-phase machines correlates with relative noise levels. In addition, in torque ripple measurements, machine mounting characteristics are substantially altered and a predictably large reduction in radiated noise is realized.

M. Carfagni et al [23]. This work provides an integrated approach developed with the aim of reducing the noise of generating sets, and the results obtained. The methodology and the procedures employed, with the indications obtained from theoretical analysis, are extendable to a vast range of industrial machines. The approach described has been applied to some PRAMAC Lifter generating set, presenting a high value of sound power, in order to reduce it accordingly with the value provided by the recent normative. Two production series generating sets were

examined. The first, named S 12000 is a petrol engine generating set, with a 10.8 KVA alternator power and 614 cm³ of cubic capacity; the second, named S 6000, is a diesel engine, with a 5.5 KVA alternator power and 406 cm³ of cubic capacity. The implementation of the method described in the paper concerns the design of a new frame for the two machines and the development of new acoustical muffler.

V. Kota et al [24] Experimental study of undesirable tonal noise generated by aircraft engines. The secondary field can either be generated by loudspeakers or by the fan itself if secondary non-uniformities are deliberately introduced into the flow. In the research reported here rods inserted radially into the duct were used to generate the secondary field. The distance by which each rod protrudes into the duct was adaptively adjusted in response to an array of in-duct microphones so as to minimize the radiated sound power, whereas previously only fixed rods have been considered. The ability of the steepest- descent algorithm to minimise in-duct sound power under suitable conditions, and hence reduce radiated sound power is demonstrated in both simulations and low Mach number experiments. It is shown how the ability of such a system to control noise depends on the number and position of the controller rods, and the number of acoustic duct modes to be controlled. Thus at low fan speed, when only one mode was present just two controllers achieved an in-duct noise reduction of 25 dB at the blade passing frequency, whereas at a higher fan speed with three modes present six controllers only achieved 2 dB. To implement such a scheme in practice, where large numbers of modes are typically present, it would be necessary to develop controller arrays with many actuators, but with low aerodynamic penalty. Such a system might also be useful in HVAC applications, or in wind-tunnel testing.

W.T. Kung et al [25] This study presents a full-scale experiment to identify the sound leakages on a wall of an enclosed room using the probabilistic approach. Distinguishing between the interior pressures induced from two independent sources within a room using the probabilistic approach, *Applied Acoustics* (2007), accepted and forthcoming; T.C. Hsia, *System Identification: Least-Squares Methods*, Lexington Books, D.C., Heath and Company, Lexington, MA, 1977]. A model class selection index is developed and used to identify the number of leakages, which is an unknown parameter in the identification process. The experiment was conducted to prove the

validity of the probabilistic

Paolo Pennacchi^À et al [26] It consists of modeling of the unbalanced magnetic pull (UMP) in generators and the experimental validation of the proposed method are presented in this paper. The UMP is one of the most remarkable effects of electromechanical interactions in rotating machinery. As a consequence of the rotor eccentricity, the imbalance of the electromagnetic forces acting between rotor and stator generates a net radial force. This phenomenon can be avoided by means of a careful assembly and manufacture in small and stiff machines, like electrical motors. On the contrary, the eccentricity of the active part of the rotor with respect to the stator is unavoidable in big generators of power plants, because they operate above their first critical speed and are supported by oil-film bearings. In the first part of the paper, a method aimed to calculate the UMP force is described. This model is more general than those available in literature, which are limited to circular orbits. The model is based on the actual position of the rotor inside the stator, therefore on the actual air-gap distribution, regardless of the orbit type. The closed form of the nonlinear UMP force components is presented. In the second part, the experimental validation of the proposed model is presented. The dynamical behavior in the time domain of a steam turbo-generator of a power plant is considered and it is shown that the model is able to reproduce the dynamical effects due to the excitation of the magnetic field in the generator

CHAPTER – 3 INTRODUCTION TO DIESEL GENERATORS NOISE

Diesel Engine Generator Sets are used primarily for emergency power generation or stand-by power in case of power failure. In the case of emergency power the generator sets are extremely important piece of equipment for hospitals, utilities, and government buildings in case of an emergency. Stand-by generator sets provide back-up power for many financial and data companies that rely on power to facilitate their day to day operations. Many of these facilities are familiar and located in our neighborhood or near noise sensitive areas. Here lies the importance of generator set noise control and acoustical enclosures. The diesel power generators provide insurance and peace of mind in an emergency.

3.1

P

ARTS OF DIESEL GENERATOR SET:

- 3.1.1 Diesel engine
- 3.1.2 Generator.
- 3.1.3 Radiator fan

THE PRIMARY CONCEPTS IN A DIESEL ENGINE ARE AS FOLLOWS:

1. The fuel and air mixture is ignited by the heat generated by the compression stroke in a diesel engine
2. The fuel and air mixture in a diesel engine is compressed to about one twentieth of its original volume. The diesel engine must compress the mixture more tightly to generate enough heat to ignite the fuel and air mixture.
3. A diesel engine takes in only air through the intake port. Fuel is put into the combustion chamber directly through an injection system. The air and fuel then mix in the combustion chamber.
4. The engine speed and the power output of a diesel engine are controlled by the quantity of fuel admitted to the combustion chamber. The amount of air is constant.

3.2 TYPES OF GENERATORS

There are two categories of generators: Diesel and spark ignited. Spark ignited types include propane and natural gas engines. The right hand graph bellow in fig 3.1 shows the sound power level difference between the two types of standard generators. Y-axis is sound power level, x-axis is sound pressure level. As can be seen in the graph, the sound power level for natural gas and Propane are the same.

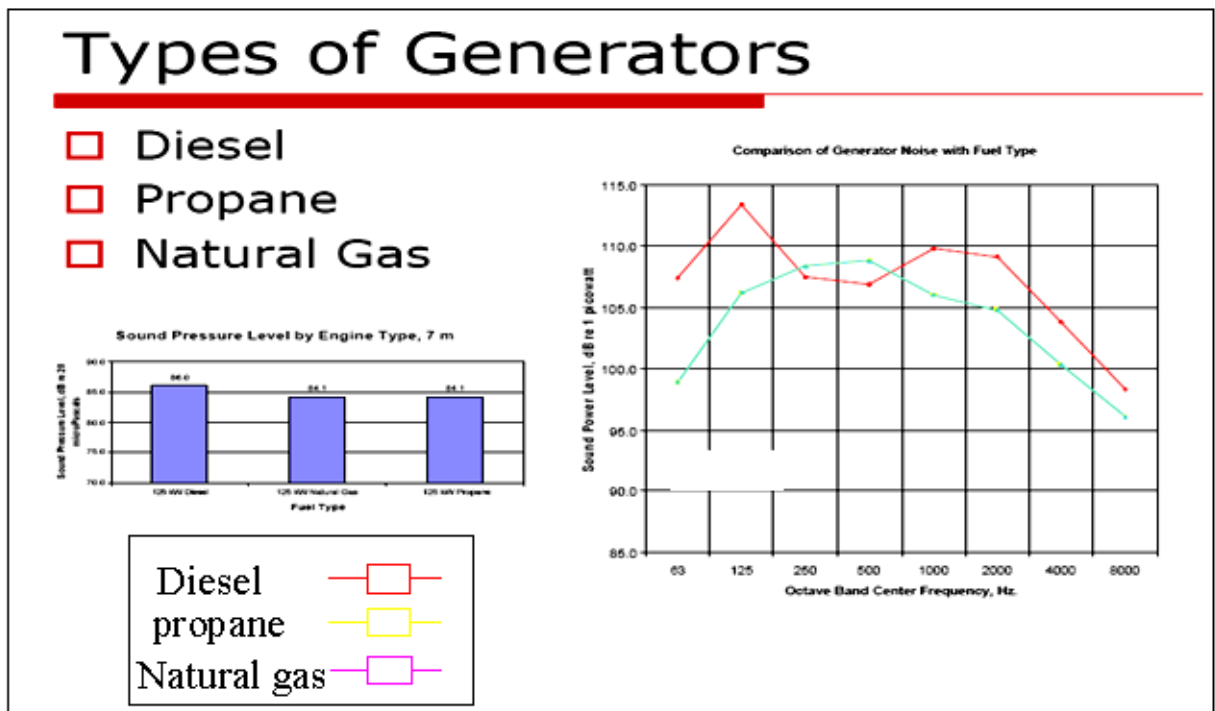


FIG 3.1 COMPARISON OF GENERATOR NOISE WITH FUEL TYPE

The left hand graph illustrates the variation of sound level with fuel type. This time the values are Sound Pressure Levels, dB(A). Sound Pressure Level varies with distance and location from the source.

3.3 ENGINE NOISE:

An engine is a mechanical device that produces some form of output from a given input. An engine whose purpose is to produce kinetic energy output from a fuel source is called a prime mover, alternatively, a motor is a device which produces kinetic energy from a preprocessed "fuel" (such as electricity, a flow of hydraulic fluid or compressed air). The various factors that contribute to the noise in engine are:

3.3.1 COMBUSTION NOISE

Combustion noise is produced because of unsteady combustion of fluid and is of two types: Turbulent combustion noise and periodic combustion oscillation. The turbulent combustion noise or combustion roar has no specific frequency but is composed of broad-band frequency spectrum. This noise is amplified if the flame is enclosed with the system resonance frequencies dominating. The requirements for reduction of this noise tend to be opposition to those for efficient combustion. Combustion oscillations involve a feedback cycle that converts chemical energy into oscillatory energy in the gas flow to the combustion region. The mechanism is such that the pressure waves generated are so phased to the velocity fluctuations. The noise spectrum involves one specific frequency and its harmonics and that frequency is related to the resonant modes of the combustion chamber. Some of the possible cures are:

1. Modification of Combustion chamber geometry
2. Change of air-fuel ratio, burner type etc.
3. Change of burning rate

It should be noted that Combustion roar in reciprocating engines which has frequency of the firing rate is not related to the combustion noise, but is due to the gross fluctuation in the flow rate produced by periodic action.

3.3.2 MECHANICAL NOISE

Mechanical noise is the noise which is generated by various impacts between the engine parts. This noise source is more important in the higher frequency range rather than in lower frequency range where combustion noise is important. There are lots of moving parts, for example, gear, valves, and rocker arms, piston and cylinder liner.

Some are as follows:

1. ENGINE CLICKING NOISE:

A clicking or tapping noise that gets louder when you rev the engine is probably "tappet" or upper valve-train noise caused by one of several things: low oil pressure, excessive valve lash, or worn or damaged parts.

2. COLLAPSED LIFTER NOISE:

Worn, leaky or dirty lifters can also cause valve-train noise. If oil delivery is restricted to the lifters (plugged oil galley or low oil pressure), the lifters won't "pump up" to take up the normal slack in the valve-train. A "collapsed" lifter will then allow excessive valve lash and noise.

3. VALVE LASH NOISE:

Too much space between the tips of the rocker arms and valve stems can make the valve-train noisy, and possibly cause accelerated wear of both parts.

4. DAMAGED ENGINE PARTS NOISE:

Excessive wear on the ends of the rocker arms, cam followers (overhead cam engines) and/or valve stems can open up the valve lash and cause noise.

5. PISTON SLAP NOISE

Piston slap noise is generated by the sudden impact of the piston to the cylinder wall is considered to be predominant due to the higher amount of energy released. In the compression stroke, the connecting rod pushes the piston upwards overcoming the gas force. The force acting on the piston has a lateral component and the piston slides upwards on the minor thrust side of the cylinder wall. As the piston moves through T.D.C. the gas forces dominate the internal forces and keep the connecting rod in compression. Thus, as the crank pin passes through the cylinder center line, the lateral component of force on the piston pin changes direction, causing the piston to accelerate through the clearance and slap against the major thrust side of the cylinder wall.

There are at least two piston slaps per revolution, but the major impact occurs at T.D.C. before the power stroke. These simple models do not take into account others factors which may affect the piston motion such as:

1. Piston pin offset.
2. Rocking motion of piston.
3. Frictions at piston pin as well as piston's outer surface.
4. Piston configuration, especially under operation.

5. Pressure distribution around piston due to the squeezing motion of oil film.
6. Compliance of cylinder liner wall.
7. Cylinder liner deformation.

3.3.3 BEARING NOISE

Crankshaft bearings are always replaced when rebuilding an engine because they are a wear component. Heat, pressure, chemical attack, abrasion and loss of lubrication can all contribute to deterioration of the bearings. The above features give rise to the noise. Some of the factors that cause bearing noise are as follows:

1. DIRT:

Dirt contamination often causes premature bearing failure. When dirt or other abrasives find their way between the crankshaft journal and bearing, it can become embedded in the soft bearing material. The softer the bearing material, the greater the embed ability, which may or may not be a good thing depending on the size of the abrasive particles and the thickness of the bearing material.

2. HEAT:

Heat is another factor that accelerates bearing wear and may lead to failure if the bearings get hot enough. Bearings are primarily cooled by oil flow between the bearing and journal. Anything that disrupts or reduces the flow of oil not only raises bearing temperatures but also increases the risk of scoring or wiping the bearing.

3. MISALIGNMENT:

Misalignment is another condition that can accelerate bearing wear. If the center main bearings are worn more than the ones towards either end of the crankshaft, the crankshaft may be bent or the main bores may be out of alignment.

4. DISASSEMBLY:

Disassembly can be another cause of premature bearing failure. Common mistakes include installing the wrong sized bearings, installing the wrong half of a split bearing as an upper, getting too much or not enough crush because main and/or rod caps are too tight or loose, forgetting to tighten a main cap or rod bolt to specs, failing to clean parts thoroughly and getting dirt behind the bearing shell when the bearing is installed.

5. CORROSION:

Corrosion can also play a role in bearing failure. Corrosion results when acids accumulate in the crankcase and attack the bearings causing pitting in the bearing surface. This is more of a problem with heavy-duty diesel engines that use high sulfur fuel rather than gasoline engines, but it can also happen in gasoline engines if the oil is not changed often enough and acids are allowed to accumulate in the crankcase.

3.3.4 EXHAUST NOISE

The engine exhaust noise originates at the exhaust tailpipe openings and is transmitted through the cabin walls, firewall, and nose gear bay. This is the loudest and most objectionable noise heard.

3.4 RELATION BETWEEN NOISE, ENGINE DESIGN AND PARAMETERS

Despite the numerous exciting forces which almost simultaneously excite the engine structure. Since the gas force resulting from combustion tends to be the predominant force in most of the engines, the relationship between the gas force characteristics and emitted noise can be used to establish a basic model to identify the effects of fundamental engine design and operating parameters.

The three basic parameters of an engine are

1. Speed
2. Size
3. Load

3.4.1. ENGINE SPEED

The engine structure characteristics can be defined by use of electro-dynamic vibration generators, and the broad response readily established line. It will be seen that when the structure is subjected to a constant sinusoidal force it exhibits maximum response in the high – frequency range from 800-2000 Hz.

Electronic analysis show is some detail the existence of numerous natural frequencies at which the structure can vibrate. It also indicates that it is reasonably heavily damped and thus any better resolution of various modes of vibration by any instrument is impossible. The gas force which again in it is very complex can be subjected to frequency analyses to quantify its exciting propensities. Analyses of the

gas forces shows that in the low-frequency range the magnitude of the harmonics in a maximum, gradually decreasing with increasing frequency at higher orders. Comparing this force spectrum with the response of the structure one can see that only the high order harmonics (frequency range 800 – 2000 Hz) are responsible for the predominant noise of the engine.

If the engine speed is doubled, the engine structure is now excited with lower order harmonics which have higher amplitudes. Since the general slope of the force spectrum is about 30 dB/decade an increase of excitation by 9 dB will be obtained with further speed the same pattern is followed. It can be concluded that the characteristics of force determine the rate of increase with engine speed which in this instance, for a naturally aspirated diesel engine, is 30 dB(A) per tenfold increase of speed.

3.4.2. ENGINE SIZE

Measurement carried out on a large number of engines with engine size is considerably less. An increase of size to ten times gives an increase of noise of 17.5 dB(A). The detailed investigations now indicate that vibration levels of the engine surfaces are about the same irrespective of their size, thus the increase of noise with size is simply due to larger radiating surface area.

3.4.3. ENGINE LOAD

Engine load has effect on noise. This occurs at the same intensity whether the engine is running at no load at all or full load. It can be concluded that:

1. The form of the exciting gas force determines the rate of increase of noise with engine speed.
2. At high engine speeds the form of the gas force has a less significant effect on noise.

3.5 GENERATOR NOISE

The predominant noise in generators results from magnetic forces in the air gap. Generator rotors have two or more magnetic poles. The pull of these magnetic forces may deform the stator laminations and deform the stator shell. This deformation rotates with the salient magnetic poles and causes vibration (see Fig.3.2). Noise of aerodynamic origin usually is less important in small generators but may be less

pronounced in large generators. These aerodynamic noise sources may be attributed to gas discharge from fans, gas discharge through the vent tunnels in the rotor, and gas turbulence caused by the surfaces moving at high speed. For example, the outer rotor surface may move as fast as 150 to 215 m/second. Other causes of vibration are mechanical unbalance and a difference in stiffness of the rotor in two planes perpendicular to each other. FIG. 3.2 Magnetic pull in a typical two-pole ac generator. Large generators (above about 15,000 kVA) are, in general, quieter than small ones. This is because they are hydrogen-cooled and hence hermetically sealed in a heavy shell which provides good sound insulation. Noise is due to vibrational excitation of generator parts and adjacent structures. Medium-sized generators (between 10,000 and 15,000 kVA) are generally air-cooled, their shells are thinner, and the air seal is not complete. Noise from the inside is transmitted through the shell and also through the incomplete seal. As in the case of larger machines, generator parts and adjacent structures are vibration-excited, some vibration being conducted through the vibration isolators. Smaller generators (below 7500kVA) are usually not provided with vibration isolators.

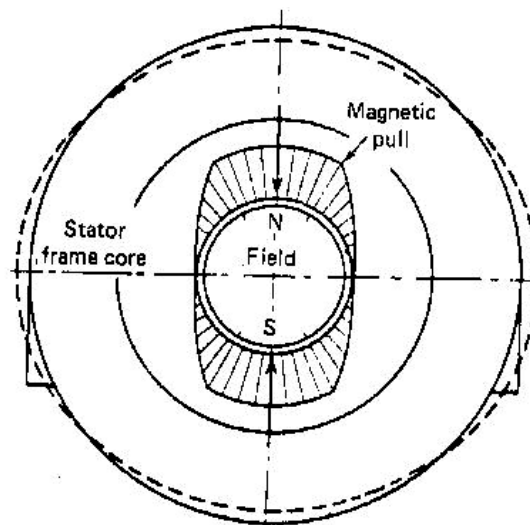


Fig 3.2 MAGNETIC PULL IN A TYPICAL TWO-POLE AC GENERATOR.

3.6 RADIATOR FAN NOISE

Radiator fan noise combines with the engine block to be the dominant sources of noise radiating out of discharge air opening in rooms containing generators, as we will see in the next slides, with larger engines the noise from the radiator fan can dominate

at least some portions of the frequency spectrum produced by the generator. Which shows engine sound levels with and without radiator?

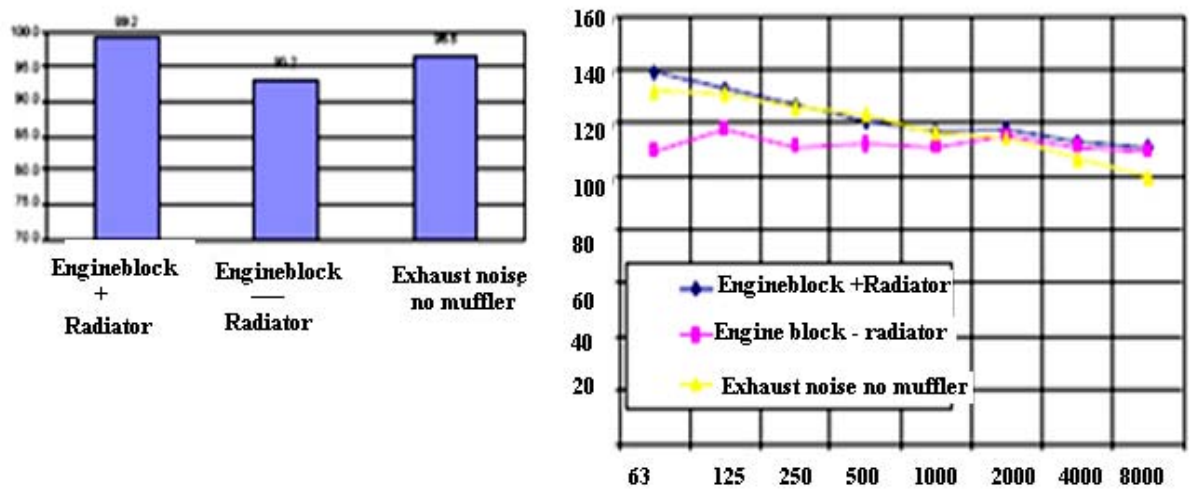


FIG 3.3 COMPARISON OF GENERATOR NOISE WITH & WITHOUT RADIATOR

CHAPTER-4 TECHNIQUES FOR CONTROL OF DIESEL GENERATOR NOISE

Noise Control is a system problem in which different components may be manipulated to achieve a particular and result. Problem in noise control usually can be represented by a system to the type shown in fig. 4.1, which contains three major parts. The source, path and receiver.

4.1 DIFFERENT COMPONENT OF NOISE CONTROL



FIG 4.1

4.1.1 SOURCE

Consider the relatively simple system depicted in figure 4.2 .Which Represents a noise source in a typical industrial environment source radiates air borne sound both through the intake and exhaust ventilation openings and through the housing of the machine, the source also radiates structure born e energy through its supports.

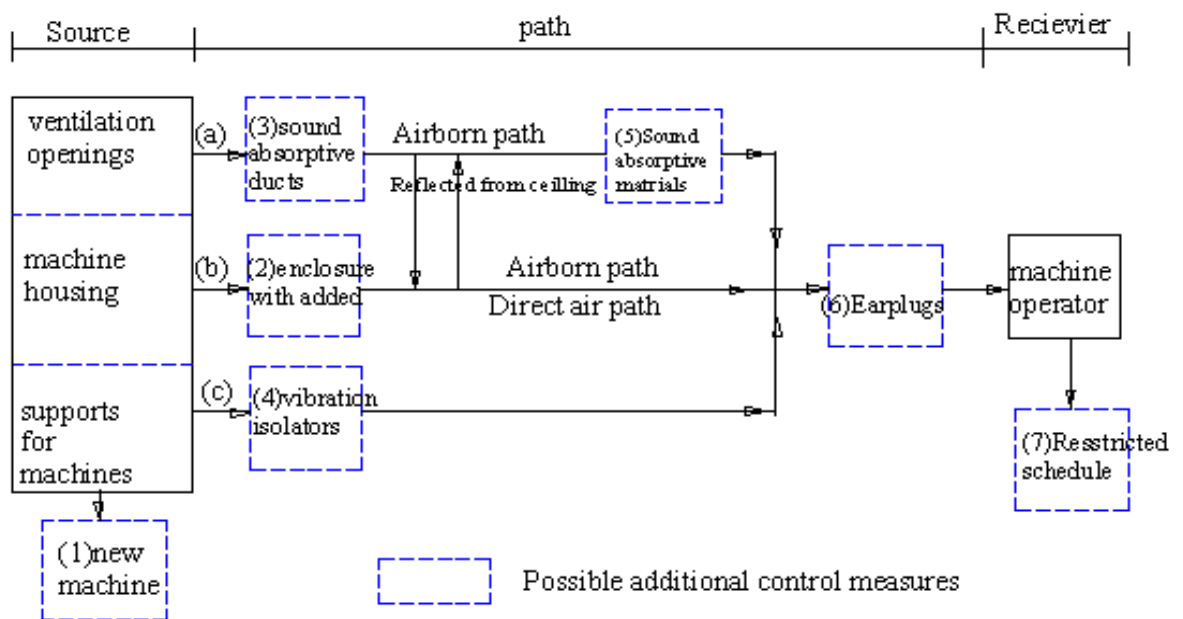


FIG 4.2 BLOCK DIAGRAM OF NOISE CONTROL SYSTEM DISCUSSED IN TEXT, SHOWING SOURCES, PATHS OF ENERGY PROPAGATION A, (B) AND (C), AND POSSIBLE METHODS OF NOISE CONTROL

4.1.2 PATH

Airborne sound radiated from the ventilation openings and from the machine housing travels along paths depicted as a and b in figure 4.2. Structure borne noise may follow a path shown as c, travelling through the floor until it reaches relatively light weight partitions, setting them in vibration and radiating noise that finally reach the receiver via an air borne path.

The noise that is propagated along the different paths has different frequency characteristics; each usually must be reduced by a different amount to achieve an economical solution to the noise control problem.

4.1.3 RECEIVER

The receiver may be a person. Example the operator of a machine, an office or factory worker or it may be a delicate item of equipment whose operation can be affected adversely by noise and vibration.

4.2 NOISE CONTROL PROCEDURE FOR ACHIEVING AN ECONOMICAL SOLUTION

To determine an appropriate and economical noise control treatment for noise problem such as one illustrated in figure 4.1. The following steps should be taken.

STEP 1 - DETERMINE THE CONTRIBUTION OF NOISE THAT IS PROPAGATED ALONG EACH OF THE PATHS BETWEEN THE SOURCE AND RECEIVER:

This may be done in terms of octave –band sound pressure levels as illustrated in figure 4.3 , In this example it is assumed that ambient noise is sufficiently low to be disregarded .The octave-band spectrum labeled “ total noise” represents the spectrum at the receiver when the equipment is in operation before any noise control measure are introduced.

The octave–Band spectrum labeled “air born reflected” represents the result of measurements in which energy transfers along both paths b and c has been eliminated. For example, energy transfer along path b may be eliminated by the temporary insulation of a barrier which effectively shields the source from the receiver. Energy transfer along path c may be eliminated by isolating the equipment from the floor on which it rests.

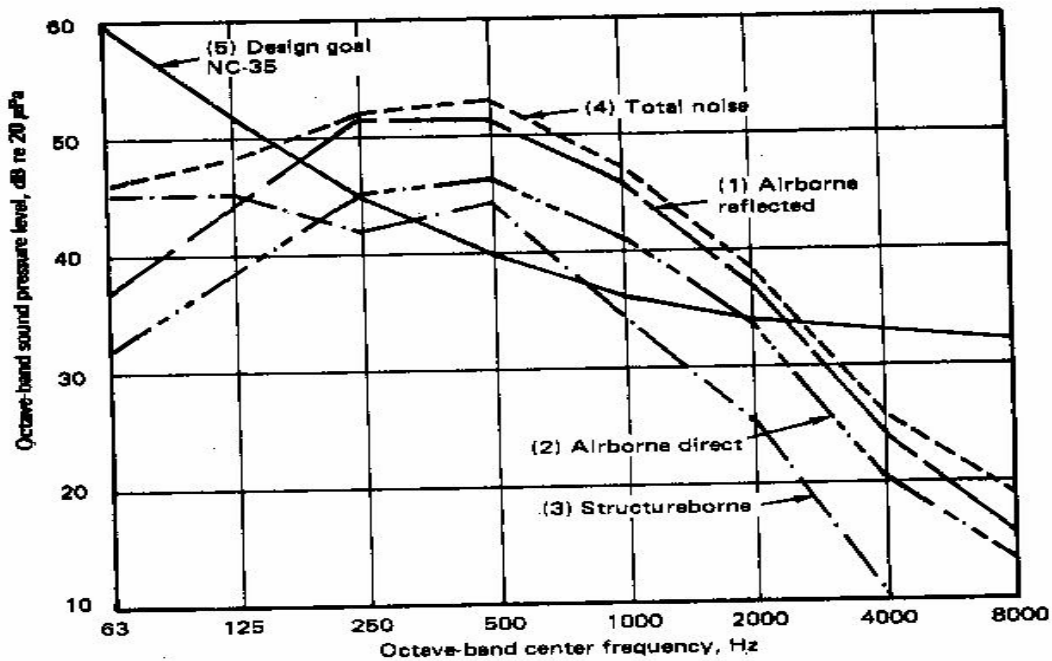


FIG. 4.3 Octave-band spectra for the system of Fig. 4.2 showing:-

- (1) The spectrum of the airborne sound that reaches the receiver by reflections path a.
- (2) The spectrum of the airborne sound that reaches the receiver directly, with no reflection path b.
- (3) The spectrum of the structure borne sound that reaches the receiver by path c.
- (4) The spectrum of the total sound that reaches the receiver by both airborne and structure borne paths.
- (5) The design goal.

The octave-band spectrum labeled “air borne direct” represents the result of measurements in which energy transfer along paths a and c has been eliminated, so that the only sound that reaches the receiver is propagated along a direct air path between the source and receiver. Such a spectrum is obtained by carrying out measurements on the machine in free field. It can be approximated by covering all surfaces of the room with a highly absorptive materials, thereby eliminating propagation along path a. Energy transfer along path c can be eliminated, as above, by isolating the equipment from the floor.

The octave band spectrum labeled structure borne represents the result of measurement taken in which the source of noise is completely surrounded by highly affective enclosure that eliminates the air borne contributions transmitted along with both paths a and b .

STEP 2- SELECT A DESIGN GOAL:

The design goal (example the octave–band spectrum to be achieved) depends on the nature of the equipment, the environment in which it is located and the nature of the requirements. For example, in an industrial environment , the designed goal may be to achieve a noise spectrum that compiles with government requirements ; in an outdoor environment, the design goal may be a noise spectrum that satisfies a local noise ordinance ; in an office environment, the design goal usually a noise spectrum that provides accepted standards for comfort and speech communication . Typical design goals for various occupancies are given in table 4.1 which may be used with the noise control curves given in figure 1.3. In this example, a design goal of NC 35 was selected, whose octave–band spectrum is plotted in figure 4.3

Type of room	Recommended		Approximate A-weighted sound level dB
	Preferred	Alternative	
Recording studios	RC 10-20(N)	NC 10-20	18-28
Concert and recital halls	RC 15-20(N)	NC 15-20	23-28
TV studios, music rooms	RC 20-25(N)	NC 20-25	28-33
Legitimate theaters	RC 20-25(N)	NC 20-25	28-33
Private residences	RC 25-30(N)	NC 25-30	33-38
Conference rooms	RC 25-30(N)	NC 25-30	33-38
Lecture rooms, classrooms	RC 25-30(N)	NC 25-30	33-38
Executive offices	RC 25-30(N)	NC 25-30	33-38
Private offices	RC 30-35(N)	NC 30-35	38-43
Churches	RC 30-35(N)	NC 30-35	38 ³
Cinemas	RC 30-35(N)	NC 30-35	38-43
Apartments, hotel bedrooms	RC 30-35(N)	NC 30-35	38-43
Courtrooms	RC 35-40(N)	NC 35 ⁰	43-48
Open-plan offices and	RC 35-40(N)	NC 35-40	43-8
Libraries	RC 35-40(N)	NC 35-40	43-8
Lobbies, public areas	RC 35 ⁰ (N)	NC 35 ¹⁰	43-48
Restaurants	RC 40-45(N)	NC 40-45	48-53
Public offices (large)	RC 40-45(N)	NC 40-45	48-53

TABLE 4.1 ACCEPTABILITY CRITERIA FOR STEADY BACKGROUND NOISE IN UNOCCUPIED ROOMS

STEP 3 - DETERMINE THE NOISE REQUIRED IN EACH OCTAVE-BAND IN ORDER TO ACHIEVE THE DESIGN GOAL:

The required noise reduction is given by the difference between (1) The octave band spectrums for the total noise at the receiver and no (2) the octave band spectrum for the design goal.

STEP 4 - EVALUATE THE VARIOUS OPTIONS THAT ARE AVAILABLE FOR ACHIEVING THE REQUIRED NOISE REDUCTION DETERMINED IN STEP 3:

The design goal should be achieved as economically as possible, both the direct costs and the indirect costs should be considered in evaluating each possible solutions. In addition to the cost of a solution, its possible adverse effects of in terms of operational restrictions must be considered.

4.3 USE OF ACOUSTICAL BARRIERS:

Consider a barrier which is placed in the direct path between a noisy piece of equipment in a room and the receiver, assume that the noise sources are properly isolated so that structure borne noise transmission can be neglected, i.e., the transmission is entirely airborne. The amount of noise reduction at the receiver provided by the barrier depends on:

- (1) The material of barrier is made of.
- (2) The dimensions of the barrier
- (3) The distance between the barrier and the source
- (4) The spectrum of the noise sources.
- (5) The sound-absorptive characteristics of other surfaces in the vicinity of the source that may redirect energy into the shadow zone created by the

barrier.

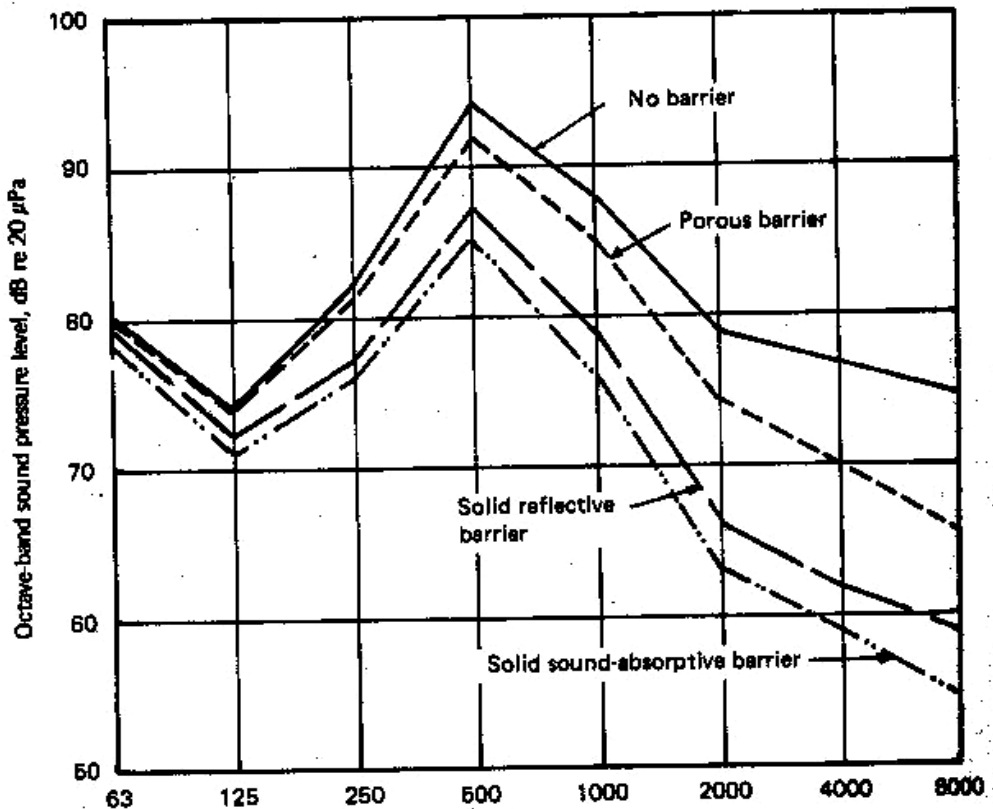


Fig 4.4 octave-band sound pressure level spectra in which different types of barriers are placed: porous barrier, solid reflected barrier, solid sound absorptive barrier

Figure 4.4 illustrates the effects of barrier material on the barrier's effectiveness in providing noise reduction where only the direct sound is significant. The uppermost curve shows the octave-band spectrum at the receiver in the absence of a barrier. The other three curves show the octave-band spectra for three barrier materials. The difference between the octave band spectrum without the barrier in place and the spectrum with the barrier represents the attenuation provided by the barrier and is a measure of its effectiveness. Three types of barriers are considered here. Porous barrier composed of a sound -absorptive material such as fiberglass Solid reflective barrier composed of nonporous material such as masonry or concrete. Solid sound absorptive barrier composed of a solid reflective barrier covered with a sound absorptive material on the side facing the source.

The effectiveness of the porous barrier in reducing noise level is poor because porous materials have low values of transmission loss. The solid barrier is better,

particularly at higher frequencies. Little sound can penetrate the barrier; the sound reaching the receiver must bend over the top of the barrier. Lining the barrier with sound absorptive material on the side facing the source provides some improvement, but only a few decibels.

4.4 PARTIAL ENCLOSURES

Partial enclosures, illustrated in Fig. 4.5, represent an intermediate step between the simple one-side barrier and the total enclosure of the equipment noise. Suppose that a source is indoors and that the receiver is far enough from the source that reflected sound predominates. In this environment, the noise reduction provided by a partial enclosure that is completely absorptive depends on the extent to which it surrounds the source. If the enclosure surrounds 50 percent of the source, a reduction of 3dB is achieved, if the enclosure surrounds 75 percent of the source, a 6-dB reduction is achieved.

If the source and the receiver are both outdoors and the receiver lies so close to the source that it is in the direct field of the source, the performance of a partial enclosure usually is similar to that of a barrier. The partial enclosure produces shadow zones within which attenuations in the range 5 to 20 dB can be achieved, depending on the frequency of the source and the construction and configuration of the partial enclosure.

Consider the performance of the partial enclosures shown in Fig. 4.5. Enclosures a, b, and c provide a noise reduction over a wider range of receiver locations than the solid absorptive barrier considered in Fig. 4.4. Enclosure c in combination with an absorptive ceiling provides a noise reduction similar to that provided by lined barriers in an open plan office.

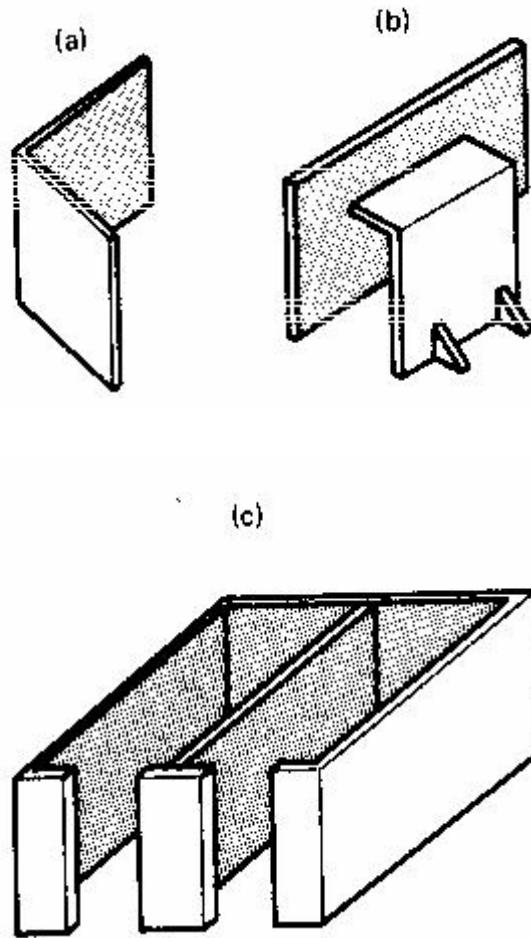
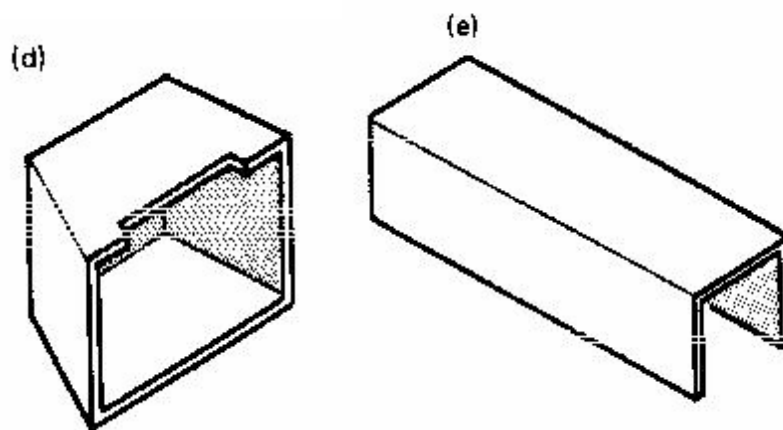


FIG 4.5 PARTIAL ENCLOSURES USED IN MACHINERY NOISE CONTROL

Partial enclosure in fig 4.5 d, e, and f, in general, provide more noise reduction than the simple barrier because the treatment extends over the top of the source. This reduces the extent to which diffractive scattering limits the available attenuation in the shadow zone.



(f)

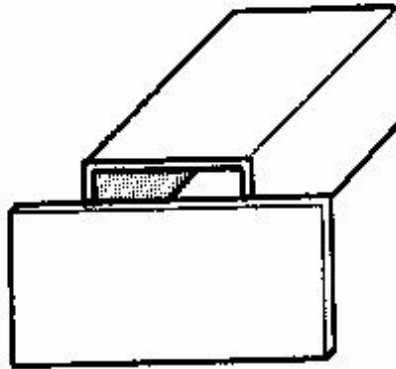


FIG 4.5 PARTIAL ENCLOSURES USED IN MACHINERY NOISE CONTROL

4.5 CHECKLIST OF NOISE CONTROL TECHNIQUES

Often the most effective method of solving a machinery or equipment noise problem is to use an alternative (less noisy) method of performing the same function. Where this is not practical, the following noise control techniques should be considered.

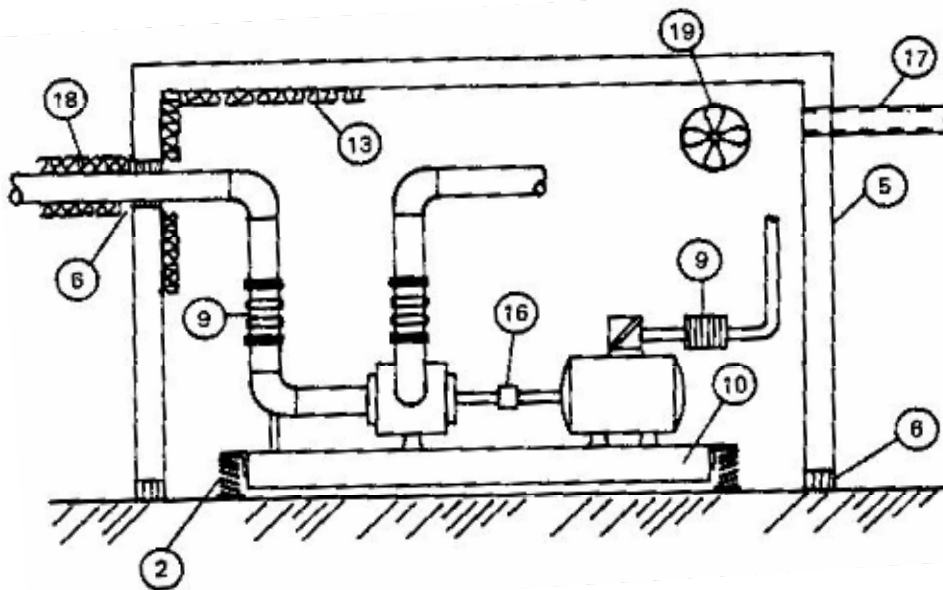


FIG 4.6 COMPONENTS MARKED THOSE TO BE CHECKED AS PER CHECK LIST

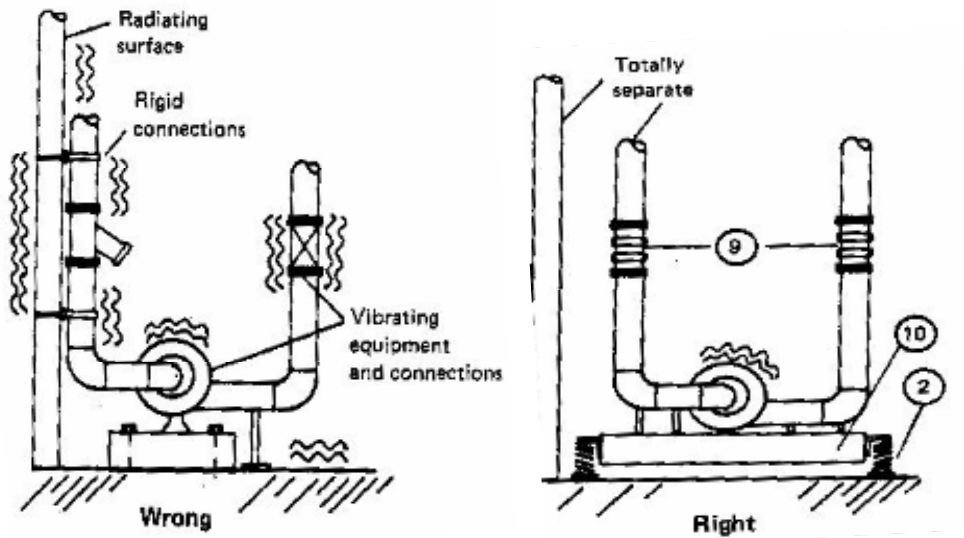


FIG 4.7 WRONG AND RIGHT INSTALLATION FOR A NOISY, VIBRATING PIECE OF EQUIPMENT

1. Move the machinery to a new location, more distant from the area where quiet is required.
2. Provide vibration isolation to reduce the radiation of noise from the surface on which the machinery is mounted.
3. Use an acoustical barrier to shield, deflect, and/or absorb the energy.
4. Use a partial enclosure around the machine.
5. Use a complete enclosure around the machine, or use a booth to house the operator when it is impractical to quiet the machine.
6. Reduce the leakage paths that permit noise to lead through openings in the enclosure.
7. Reduce impact forces.
8. Apply vibration damping materials to the housing of the machine.
9. Insert flexible connectors between the machine and conduit, cable, piping, or ductwork connected to it.
10. Use an inertia block, where appropriate.
11. Reduce or modify surfaces that radiate noise.
12. Reduce resonance effects in mechanical and acoustical systems.
13. Use sound absorptive material.
14. Modify or replace gears.
15. Modify or replace bearings.
16. Reduce unbalance in rotating systems.

17. Use ducts lined with sound absorptive material.
18. Use wrapping or lagging on pipes or ducts to increase their sound insulation.
19. Reduce or eliminate aerodynamically generated noise.

CHAPTER-5 DESIGNING OF ACOUSTIC ENCLOSURE FOR 5 KW DIESEL GENERATOR SET

ACOUSTIC ENCLOSURE:

As discussed in previous chapter to control the noise of a generator completely there is needed to design an acoustic enclosure. There is need to provide ventilation to prevent the overheating of the engine. Before to fabricate a functional design is to be made on the bases of following fundamentals

5.1 THE USE OF SOUND BARRIERS IN ACOUSTIC ENCLOSURE:

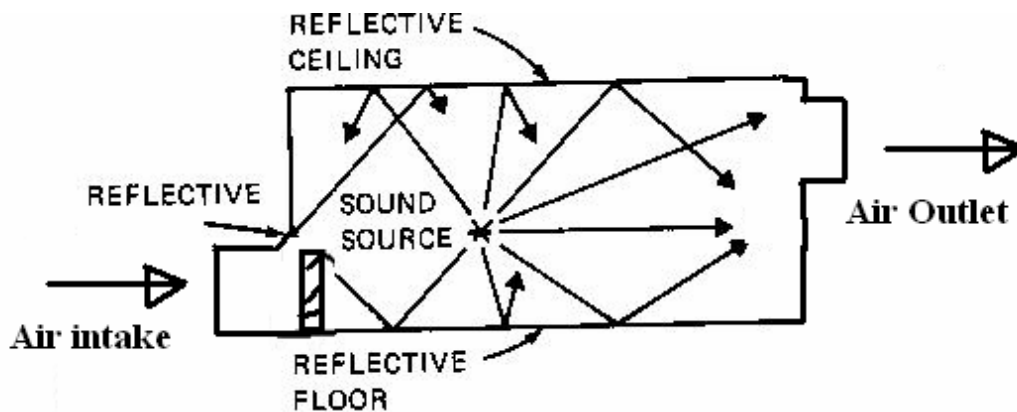


FIG 5.1(A).INSIDE AN ACOUSTIC ENCLOSURE SOUND WILL TRAVEL THROUGH AIR VENTILATION

Which sound from a noise sources travels to the receiver by a direct path and also by a multiplicity of reflections from the sides of enclosure, ceiling, and floor.

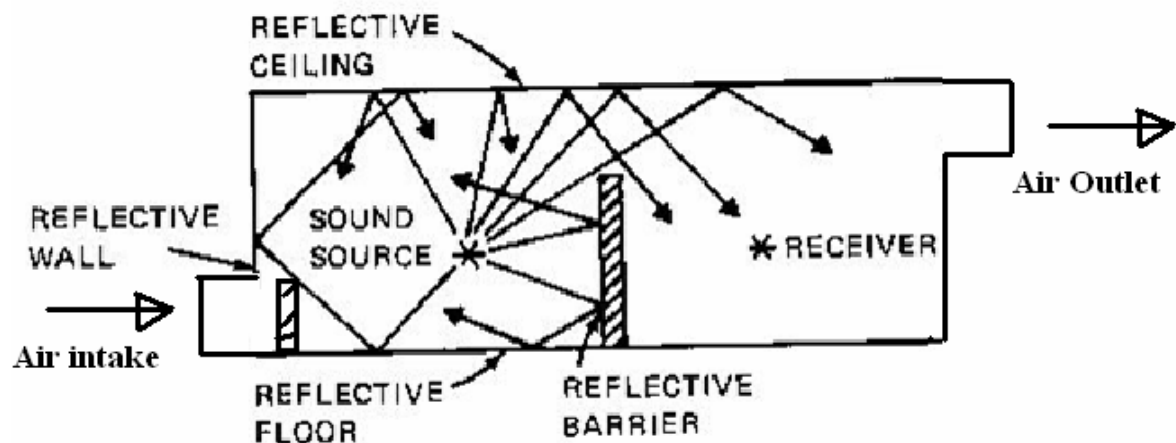


Fig 5.1 (b)

In Fig 5.1 (b) a solid reflective barrier is installed, which is effective in eliminating the direct sound; however, it increases (by reflections) the sound reaching the receiver that is not shielded by the barrier. The effectiveness of the barrier thus is greatly compromised by the reflected sound.

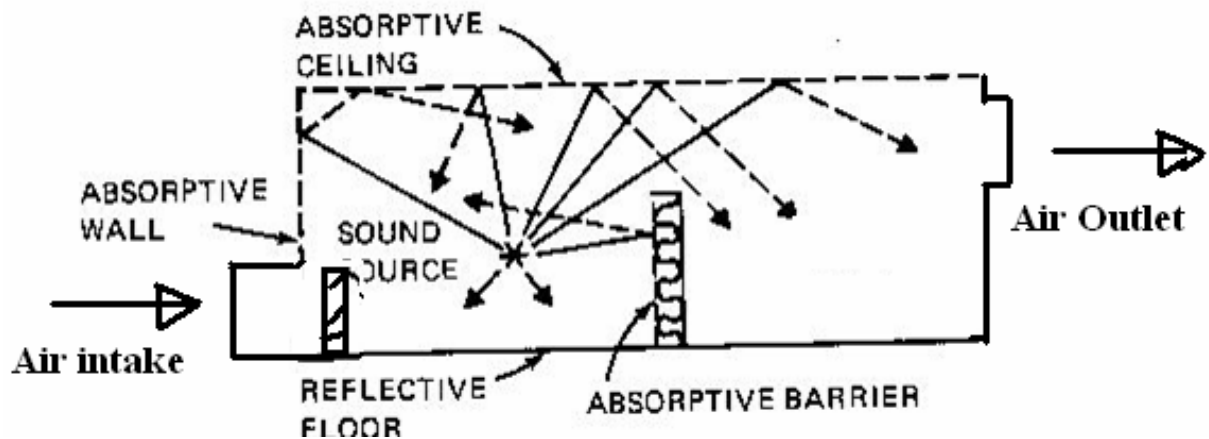


Fig 5.1(c) WITH ABSORPTIVE MATERIAL

In Fig 5.1 (c), sound absorptive material has been installed on the barrier. Under these conditions, the reflected paths are greatly attenuated and the maximum noise reduction is achieved.

5.2 RULES FOR USE OF BARRIERS INSIDE ACOUSTIC ENCLOSURE

1. Place the barriers as close to the source as possible - but not in contact with the source.
2. Extend the barrier beyond the line of sight of the source (both vertically and horizontally) by one-quarter wavelength of the lowest frequency for which significant attenuation is required.
3. Select a solid barrier material (with no openings or holes) having a sound transmission loss at least 10dB higher than the required attenuation.

5.3 COMPLETE ENCLOSURES AROUND A SOURCE

The purpose of a complete enclosure is to contain and absorb the acoustic energy radiated by the source as shown in fig 5.2 This reduces the sound pressure

levels at all distances from the source. The amount of noise reduction may approach the transmission loss of the material from which the enclosure is fabricated.

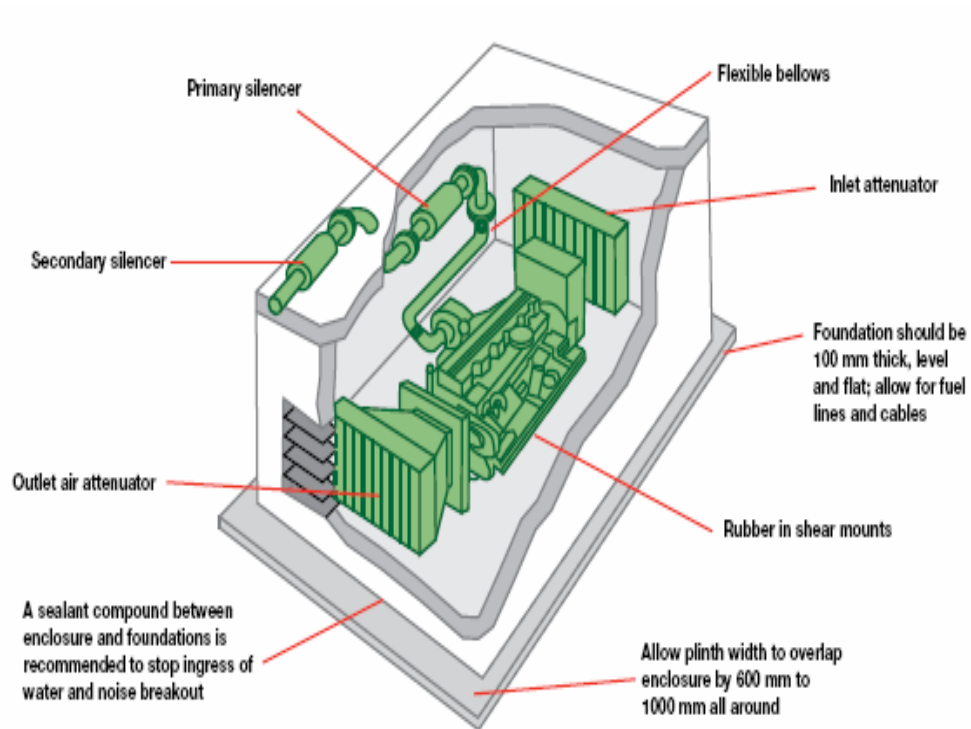


Fig 5.2 Complete enclosure around a source

The importance of using a solid rather than a porous material for the walls of the enclosure is illustrated in Fig. 5.3 Although the performance of the solid enclosure is much better than that of the porous enclosure, it is limited by multiple reflections within the enclosure. This process causes the internal sound pressure levels to build up until a steady state is achieved within the enclosure. The amount of buildup depends on the dimensions of the enclosure and the sound absorptive characteristics of the enclosure, but a buildup of 10 to 20dB is not unusual. This greatly reduces the effectiveness of the enclosure. For this reason an enclosure should be lined to the maximum extent possible with sound absorptive material. Because of this buildup, the transmission loss (TL) of the wall material must be greater than the noise reduction (NR) required at each frequency of interest..

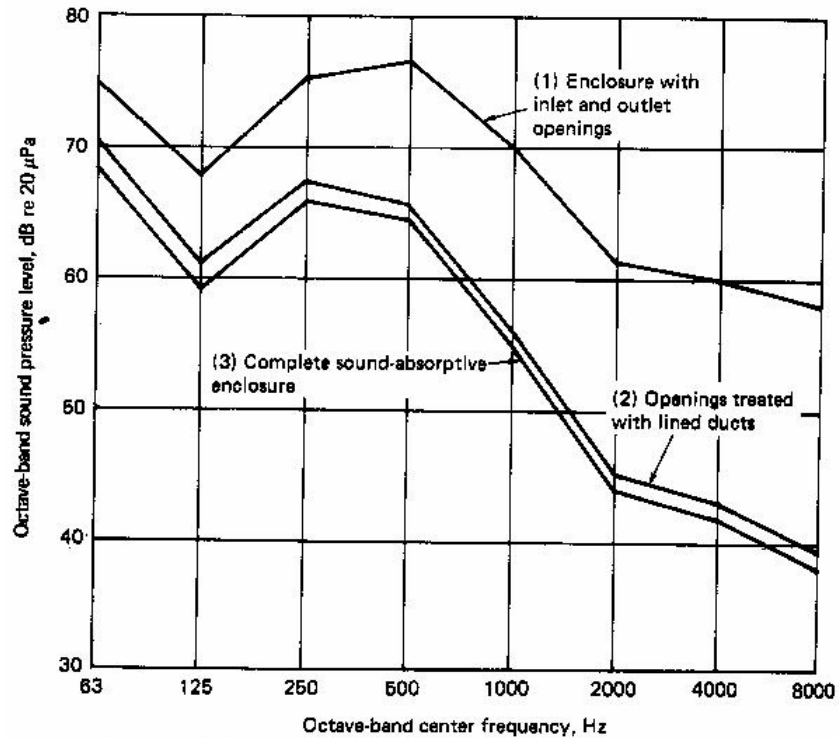


Fig 5.3

In some cases, the housing (or cabinet) of a machine. May serve as an effective enclosure for noise sources that are contained within the equipment. In this case the addition of a sound absorptive lining (and the sealing of leakage paths) may provide significant noise reduction.

Rules of Thumb

1. If enclosure has no internal absorption, then

$$TL > NR + 20$$

2. If enclosure has partial internal absorption, then

$$TL > NR + 15$$

3. If enclosure has complete internal absorption, then

$$TL > NR + 10$$

5.4 REDUCTION OF LEAKAGE PATHS

The attenuation of an enclosure is highly dependent on leakage paths that allow the sound energy to bypass the walls of the enclosure. The importance of leakage is illustrated in Table 5.1, which describes the extent to which a fully

absorptive partial enclosure must encompass a uniformly radiating point source to achieve different degrees of attenuation.

Sound energy enclosed and absorbed, %	Maximum achievable noise reduction, dB
50	3
75	6
90	10
95	13
98	17
99	20

TABLE 5.1 EFFECTIVENESS OF A PARTIAL ENCLOSURE

A leakage of only 1 percent of the enclosure surface area limits the available attenuation to 20dB. The attenuation of an unlined enclosure is even more dependent on leakage, since buildup already detracts from the performance.

There must be no leaks if noise is to be contained at its source:

- (1) all joints, seams, and penetrations of enclosures by ducts, pipes, conduits, etc., must be sealed so the acoustical integrity of the enclosure is not compromised
- (2) Access doors to enclosures must be tightly fitted and gasket, locking handles should be provided that draw the door tight to its gasketed surface so as to provide an airtight seal.

5.5 USE OF LINED DUCTS FOR VENTILATION OPENINGS

Complete enclosures around noisy equipment may require ventilation to prevent overheating of the equipment. The effect of inlet and outlet openings (for ventilation of the cabinet shown in Fig. 5.4 (a) is illustrated in Fig. 5.4 (b). The openings substantially reduce the attenuation provided by the complete enclosure with absorption.

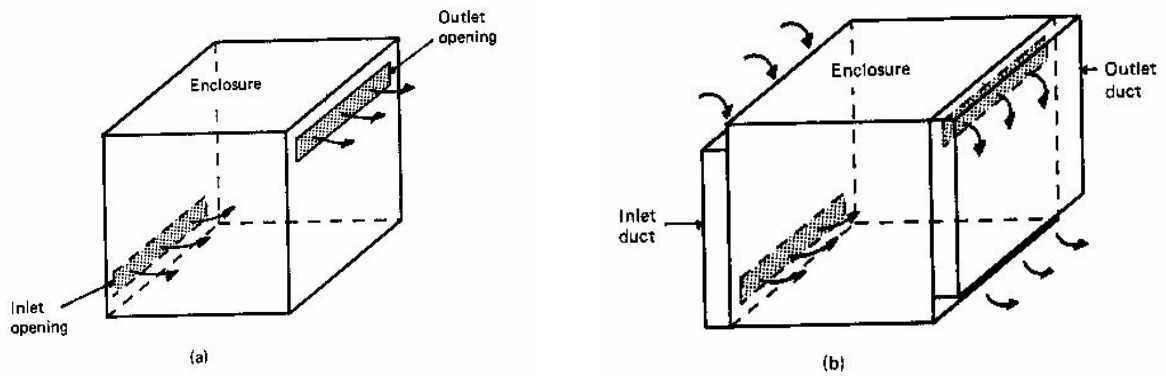


FIG 5.4 (A) UNTREATED INLET OUT LET OPENING.(B)OPENING WITH DUCTS LINED WITH SOUND ABSORPTIVE MATERIALS

SPECIFIC DATA, LOCATION OF GENERATOR USED IN EXPERIMENT

1.Engine Detail	Location	Manufacturer	Model no	Engine H.P	R.P.M	Lub.oil used	Piston bore
	Thapar university Patiala. (steam lab.)	Alamgir Generator manufacturers Ludhiana.	SVA-2	8 H.P	1500	20W40	87.5 × 110

TABLE 5.2

1.Generator Detail	Location	Manufacturer	Frequency	power	R.P.M	VOLAGE	CURRENT
	Thapar university Patiala (steam lab.)	Alamgir Generator manufacturers Ludhiana.	51Hz	5kVA	1500	240V	20 AMP.

TABLE 5.3

5.6 MATERIAL USED FOR FABRICATION OF ACOUSTIC ENCLOSURE:

5.1 M.S. bars

5.2 Steel angles & nut, bolts (for temporary fastening)

5.3 G.P. sheet.

5.4 Polyurethane foam.

5.5 Flexible pipe, elbows.

5.6 Silencer.

Major sound absorption is done by polyurethane foam and silencer.

5.7 SPECIFICATION OF POLYURETHANE FOAM:

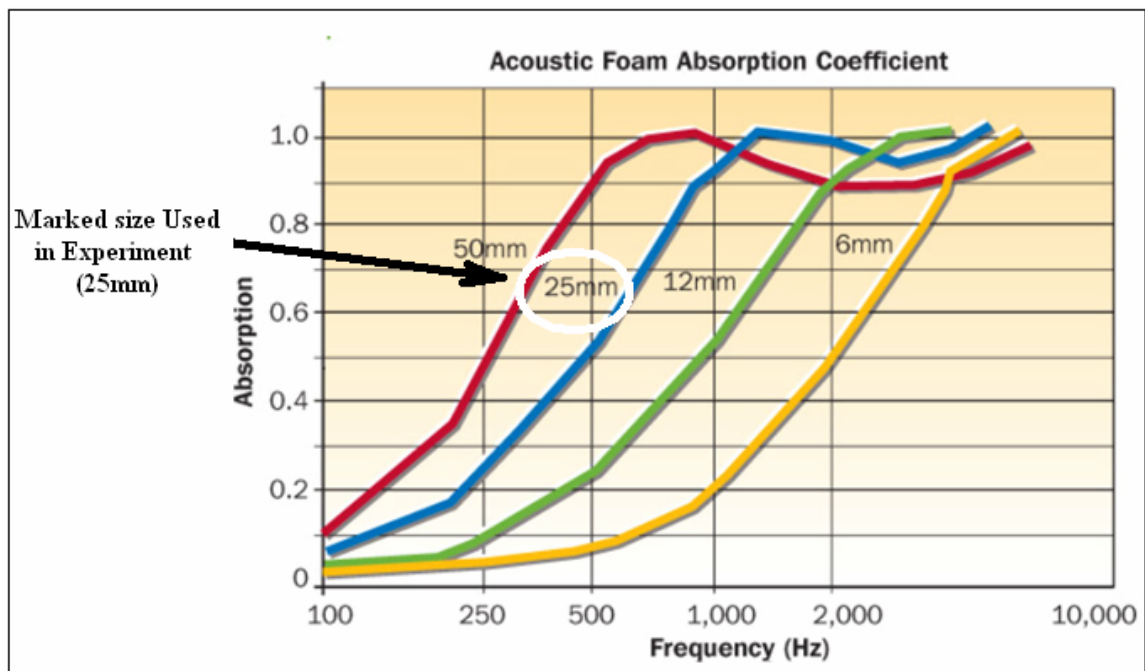


FIG 5.4 SPECIFICATION'S OF POLYURETHANE FOAM

1. Specification's of poly-urethane foam	Sheet Size	Piece Density	Tensile Strength	Flammability
	1500 x 1000mm	27-30 kg/m ³ .	100 kn/m ²	FMVSS 302 Self-Extinguishing

TABLE- 5.4

MAJOR DIMENSION'S OF ENCLOSURE & DIESEL GENERATOR SET:

Equipment	Length	Breadth	Height
Enclosure	2m	0.70 m	1.30 m
5kW Diesel Generator	1.30m.	0.60 m	0.90 m

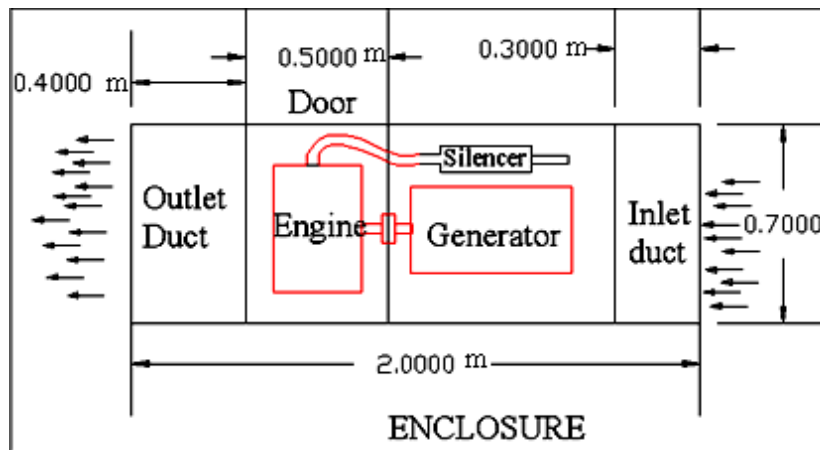


FIG 5.6 COMBINED TOP VIEW OF ACOUSTIC ENCLOSURE AND DIESEL GENERATOR

As per above discussion following parts of acoustic enclosure are fabricated.

5.8 PARTS OF ACOUSTIC ENCLOSURE:

1. Silencer.
2. Front side
3. Intake duct.
4. Outlet duct.
5. Long side.

6. Cooling fan.
7. Left & right covers.

FRONT SIDE:

There is no self-start system in 5kW diesel generator. Therefore, it starts with handle. For the free movement of human being to start the engine with handle door should be large, as shown in fig5.7 and covered with polyurethane foam. To reduce the noise leakage from door.

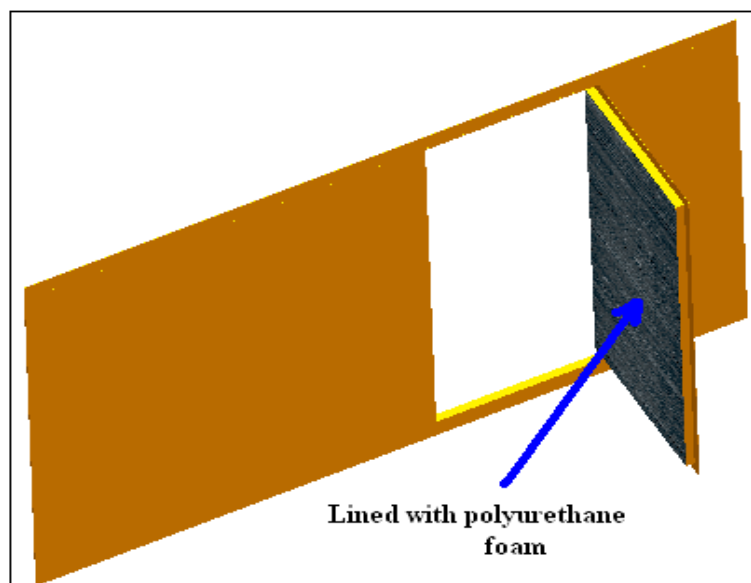


Fig 5.7

LEFT & RIGHT COVERS:

As discussed in previous chapter apply partial barriers in between the source and receiver, here in fabrication of enclosure providing barriers

at inlet & outlet of air as illustrated in fig 5.8 & 5.9 also covered with polyurethane foam.

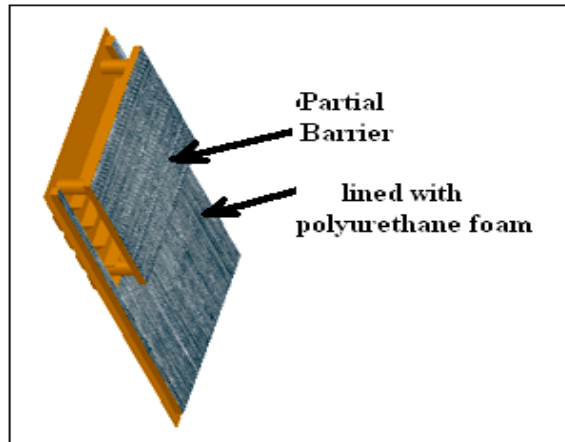


FIG 5.8 LEFT SIDE COVERS

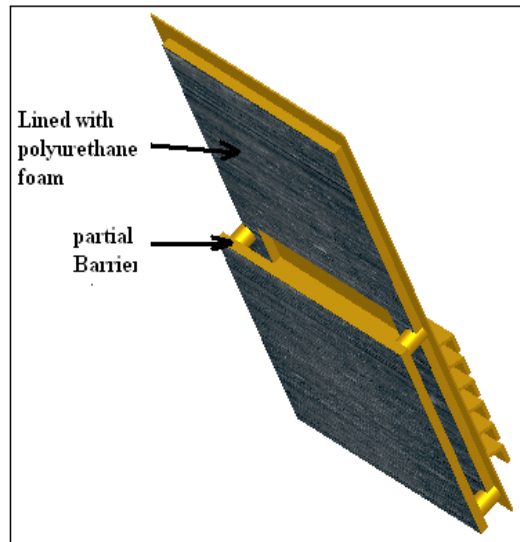


FIG 5.9 RIGHT SIDE COVERS

TOP COVER:

There is need to change the position of exhaust to minimize the heat production inside the enclosure .which reduces the problem of over heat of diesel engine. It is recommended that keep the exhaust away trough the top cover of the enclosure .there is need to make very precise hole to pass through the exhaust pipe from top cover illustrate in fig 5.10 There is need to provide insulating material to avoid make the point source of noise near the hole.

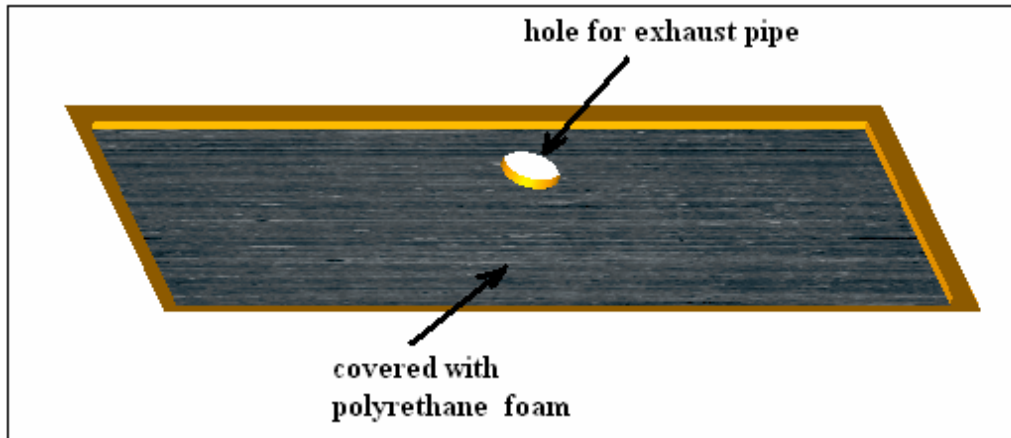


FIG 5.10 TOP COVER

SILENCER:

Exhaust noise is the biggest source of noise production as discussed in chapter. 4. To reduce the noise of exhaust there is need of effective silencer. Silencers are having different types. But their working principle is same. To provide the divergent to the sound and pass through the sound absorptive material as illustrated in fig 5.11 (b) cross-sectional area of silencer there are holed plates , but holes are not in one line end these plates are coated with sound absorptive material.

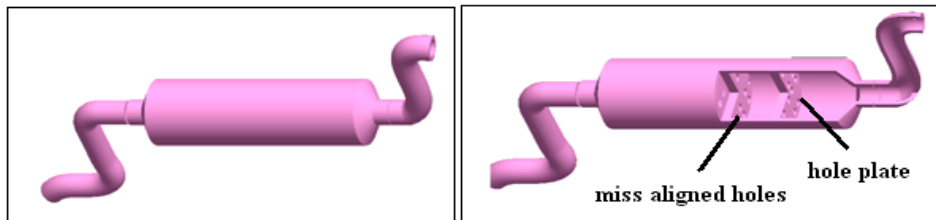


FIG 5.11 (A) & (B) WITH & WITHOUT SECTIONAL SILENCERS

INLET & OUTLET DUCT:

Noise reduction may be minimized by the use of lined ducts that provide attenuation without impeding the airflow, as illustrated in Fig. 5.12 (B) Here, the opening is fitted with sheet-metal ducts lined with polyurethane foam. The noise must negotiate a right angle turn and pass through a lined duct to reach the environment outside the enclosure as shown in fig 5.12 and enlarged view of portion which has partial barrier also.

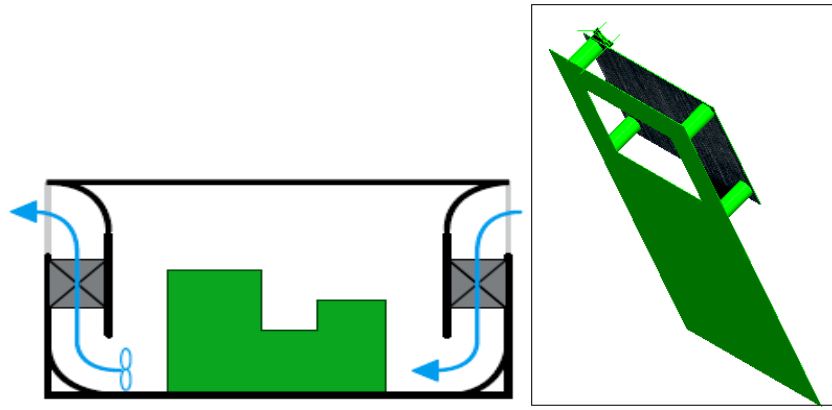


FIG 5.12 VENTILATION DUCT TURNED AT 90° ANGLE

5.9 COMPLETE ASSEMBLY OF DESIGNED ACOUSTIC ENCLOSURE:

From the above discussion by keeping in consideration all major factors of noise reduction acoustic enclosure will look like as illustrated in

fig 5 .13.

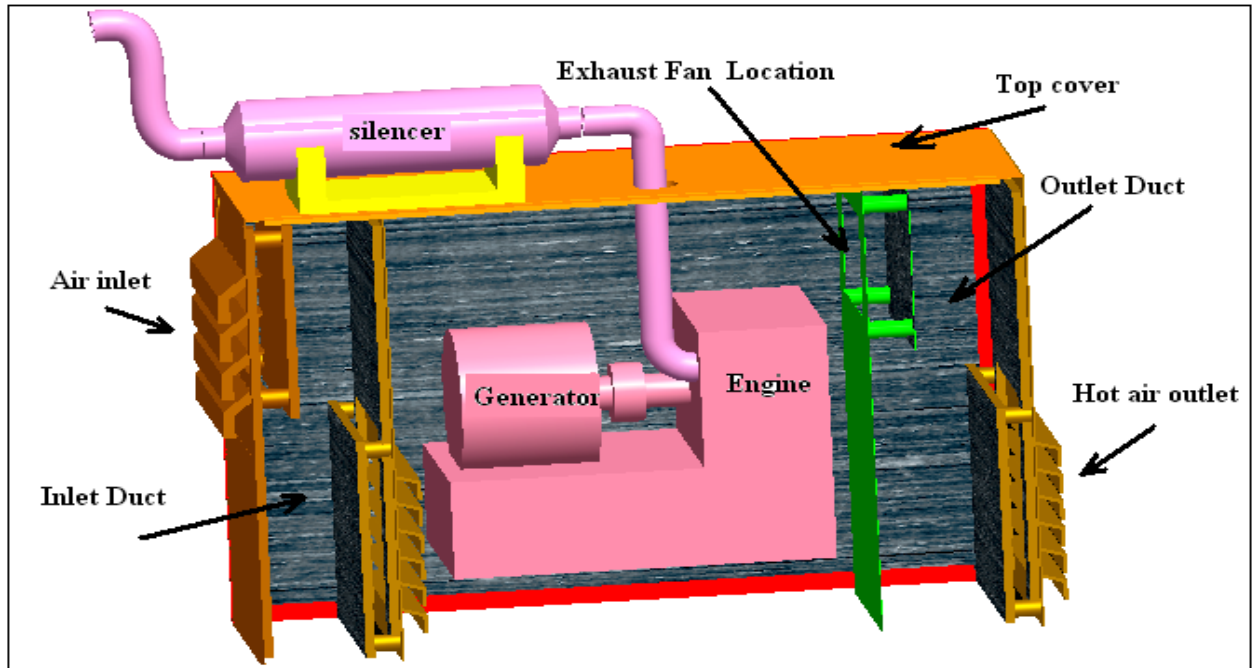


FIG 5.13 COMPLETE ASSEMBLY

5.10 DESIGNED AND FABRICATED VIEW OF ACOUSTIC ENCLOSURE

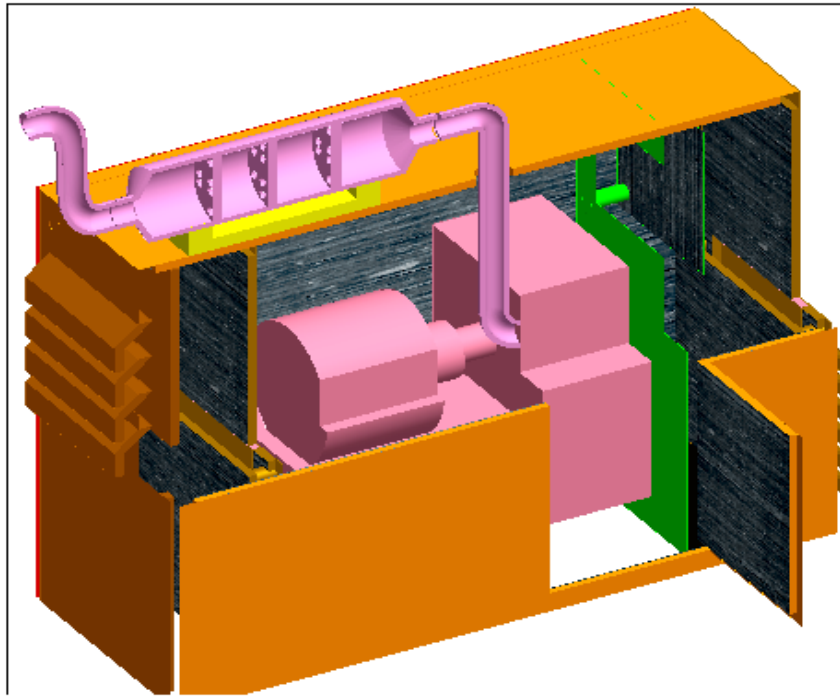


FIG 5.14 SECTIONAL DESIGN VIEW



FIG 5.15 FABRICATED VIEW



FIG 5.16 DESIGN VIEW



FIG 5.17 FABRICATED VIEW

CHAPTER – 6

GENERAL PROCEDURE AND EXPERIMENTAL SETUP

To check the effectiveness of the acoustic enclosure following experimental work is decided to be carried out .

1) Measure the acoustic power at different condition to check the individuals role to reduce the noise and to check that how much total reduction in noise is done by using an acoustic enclosure at different loads.the different conditions are as follow:

- Only generator.
- With Only silencer
- With only Enclosure
- enclosure with silencer
- Enclosure with silencer & inlet ,outlet duct.

2) To determine the peak sound level frequencies from analysis the frequency spectrum in 1/1 Octave-band and the reduction in dB(A) in those peaks at different conditions at different points A,B,C(their location is shown in fig 6.1) and at different loads. load will varies from 0kW to 3kW.Different conditions are as follow.

- 1) Only generator.
- 2) With Only silencer
- 3) With only Enclosure
- 4) enclosure with silencer
- 5) Enclosure with silencer & inlet ,outlet duct

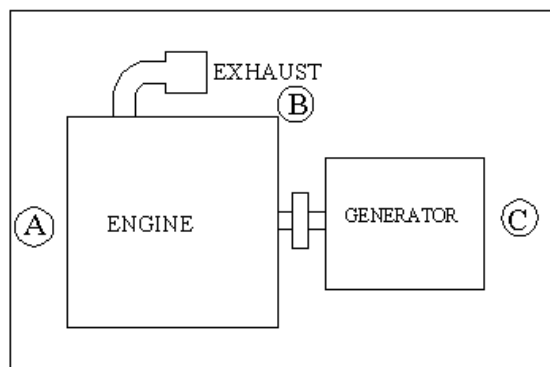


FIG 6.1 LOCATION OF A, B, C POINTS

3) There is change in the position of exhaust, silencer is placed outside the enclosure air circulation is restricted to steady the change in the impact of sound in different direction it is decided that contour formation is carried out at different condition and to find out still the areas where sound level is more. During the measurement of acoustic power averaged of all the point is taken .so it is difficult in which direction still sound impact is more.

So it is decided that contour graphs has to be made at six levels, explained in article 6.7 of this chaptering following conditions.

- Only generator
- With complete enclosure.

6.1 GENERAL PROCEDURE FOR THE MEASUREMENT OF SOUND

POWER[28]

SOUND POWER LEVEL MEASUREMENT WITH SOUND PRESSURE LEVEL :

The sound power level of noise sources can be measure with the help of sound pressure level with following steps:-

1. Surround the source with hypothetical surface of area S (either a hemisphere or a rectangular parallelepiped).
2. Calculate the area of this hypothetical surface if it is hemisphere, S is given by $2\pi r^2$ where r is radius of the hemisphere.
3. If it is rectangular, S is given by $ab+2(ac + bc)$, where a, b, c are its length, width and height.
4. Measure the sound pressure level at designated point on the hypothetical surface.
5. Obtain the average L_p of sound pressure level measured in the step 3.
6. Finally calculate the sound power level from the following equation.

$$L_W = L_p + 10 \log_{10} (S/S_o) \quad \text{dB}$$

Where S_o is reference area

S is hypothetical surface area

6.2 ADDITIONAL ASPECT OF MEASUREMENT CORRESPOND TO THE STEPS IN THE ABOVE PROCEDURE

1. For the small sources those whose largest dimension is significantly less than one meter. It is usually more convenient to use hemisphere than rectangular parallelepiped as a hypothetical measurement surface for large rectangular sources the rectangular parallelepiped surface is usually preferred.
2. The radius of hypothetical hemisphere should be equal to or greater than twice the major source dimension and not less than 1m for the rectangular parallelepiped, the measurement distance “d”, the perpendicular distance between the source and the measurement surface has a preferred value of 1m.
3. For hemisphere the designated point of the microphone locations are shown in figure 6.2. The corresponding point for the rectangular parallelepiped is shown in figure 6.3. The sound pressure level at designated point is measured with A-weighting or in octave or in one-third octave bands.
4. The average sound pressure level over the measurement surface, L_p is calculated from the measured sound pressure level L_{pi} , after correction for background noise.

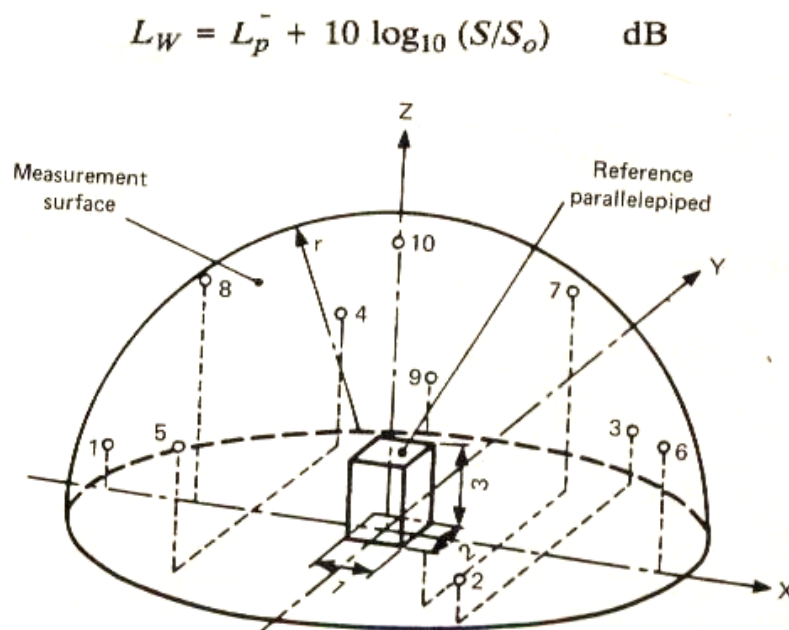


FIG 6.2 GRAPHICAL REPRESENTATION OF MICRO PHONES POSITION ON AN IMAGINARY HEMISPHERICAL SURFACE SURROUNDING A SOURCE.

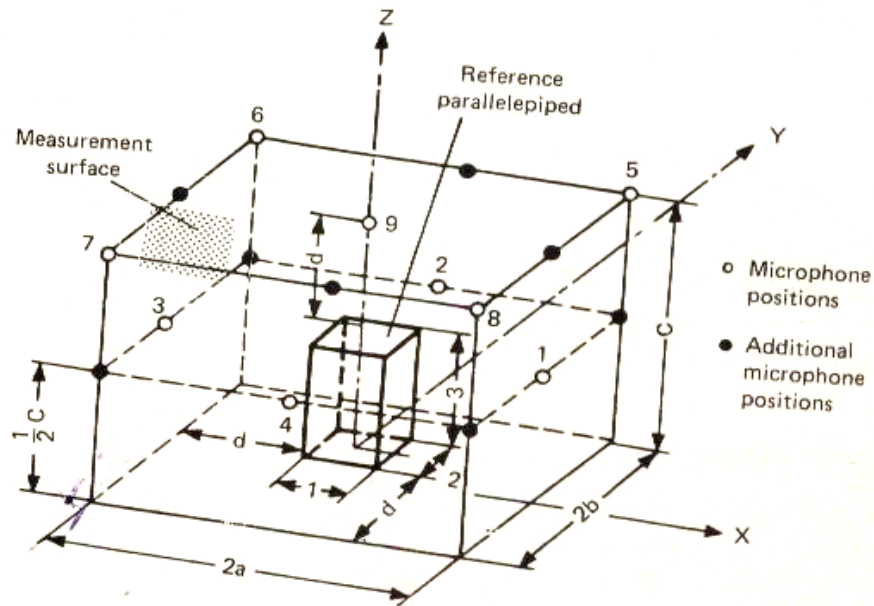


FIG. 6.3 ARRAY OF MICROPHONE POSITIONS ON AN IMAGINARY PARALLELEPIPED SURFACE SURROUNDING A SOURCE WHOSE SOUND POWER IS TO BE MEASURED.

6.3 APPROPRIATE METHOD OF MEASUREMENT DEPENDS ON FOLLOWING STEPS:

1. Mobility and location of the noise sources.
2. Nature and the configuration of the test environment.
3. Intended use of the measurement results.
4. Characteristics of the noise source.
5. Time and equipment available for the measurement.
6. Skill of the individual conducting the measurements.
7. Required accuracy of the measurements.

6.4 ADVANTAGES OF MEASUREMENT OF SOUND POWER LEVEL

1. Calculating the sound pressure level at a given distance from a machine.
2. Comparing the noise radiated by machines of the same type and size.
3. Comparing the noise radiated by machines of different types and sizes.
4. Determine whether a machine complies with a specified upper limit of sound emission.
5. Determine the amount of noise reduction required under certain circumstances.

6. Developing quiet machinery and equipment.

6.5 MEASUREMENT OF SOUND POWER:

Measurement of sound power gives the clear picture about the difference in an Engine having muffler or without mufflers. Measurement can be done according to the dimensions of an engine. In this case, rectangular parallelepiped method is used because maximum dimension is greater than 1m .In this method, the first step is to make a grid according to the dimensions of length, breadth, height. The grid is made by placing an engine at centre position and with the help of wire at required positions mark the different points. According to article 6.1, grid points formed are 17 points as shown in fig 6.4 now it is necessary to place the microphone at every grid points to take the readings at different points parameters as illustrated in fig 6.4.

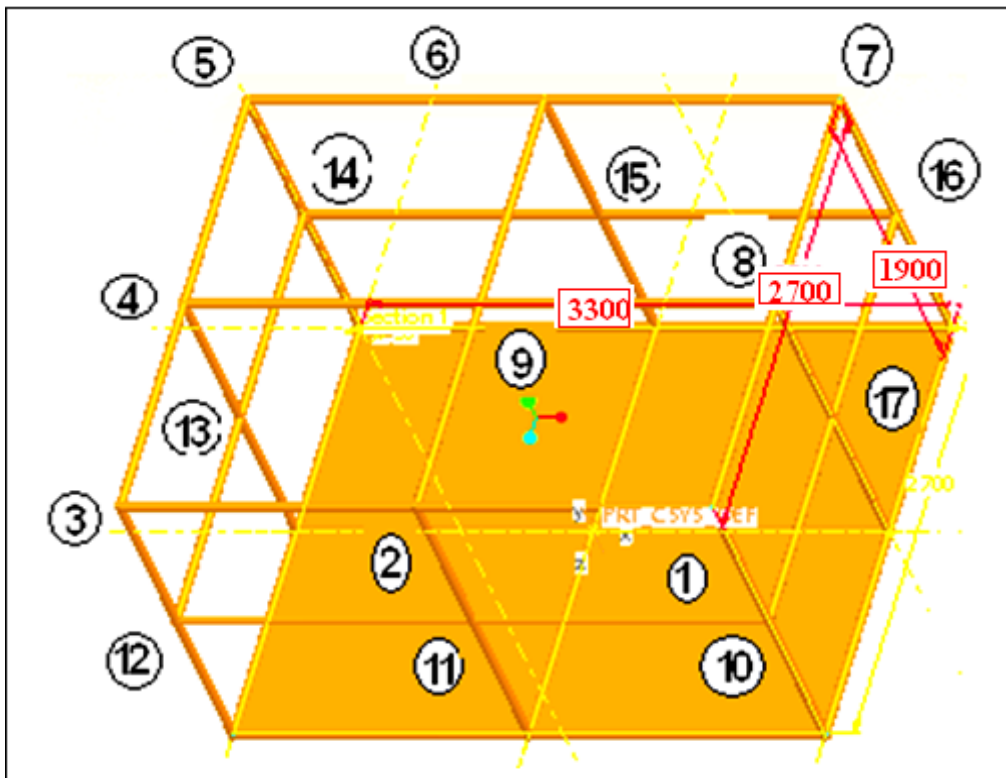


FIG 6.4

In this experiment load is the only parameter that can be changed. Here load is changed with the help of halogen lights. Load varied from 0 to 3kW.

$$L_W = L_P + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$L_W = L_P + 11.3 \text{ dB(A)} \text{ (ref. } 10^{-12} \text{ W)}$$

6.6 MEASUREMENT OF SOUND PRESSURE LEVEL FOR FREQUENCY SPECTRUM:--

The value of sound pressure level at 1-1 octave band frequency gives an idea of the maximum and minimum value at what frequency. In this, the point selected where frequency spectrum is formed is that point where sound pressure is more. Take the readings at that point by changing the parameters. This instrument gives the data of all frequencies in software "CESVA CAPTURE STUDIO". Measurement of sound pressure level at 1-1 octave band is taken at different positions shown in following

fig 6.5 & fig 6.6



FIG 6.5



FIG 6.6

6.7 PROCEDURE FOR GENERATE CONTOUR DIAGRAMS:

To generate contour graph as illustrated in fig 6.7 divided three dimensional areas into hypothetical grid form. Each square in grid having 30×30 Cm dimensions.

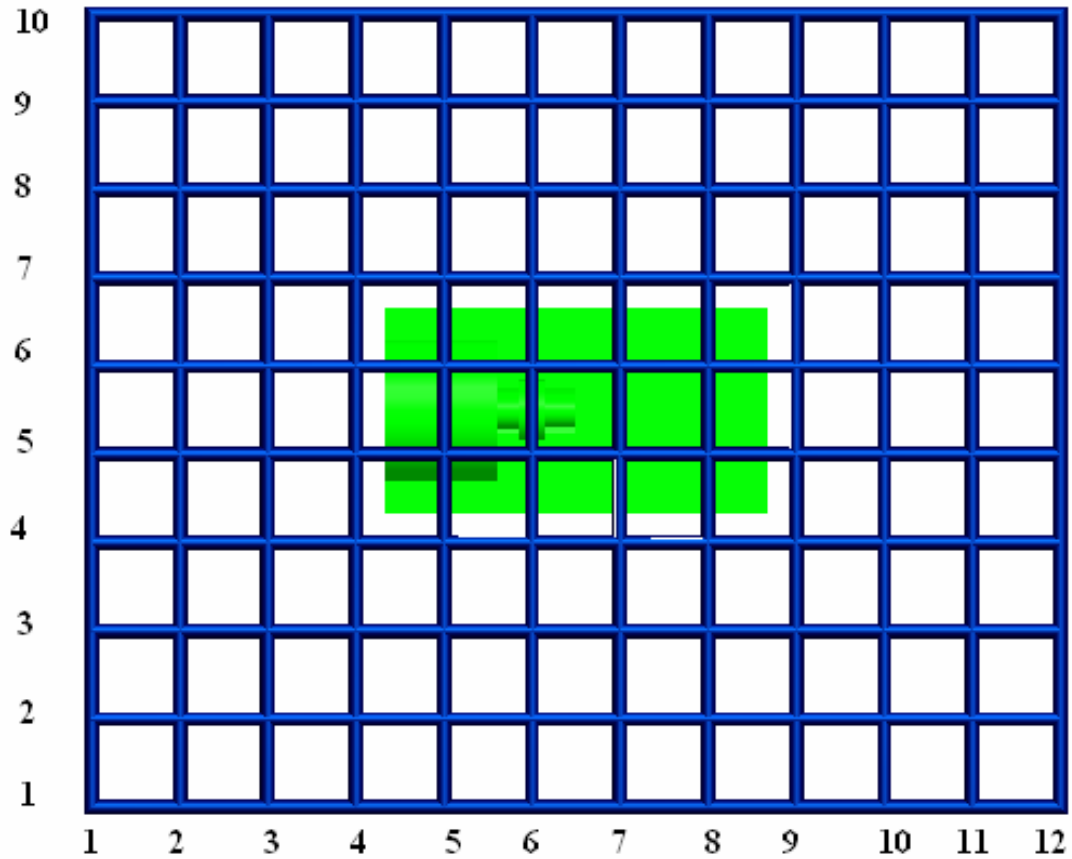


FIG 6.7 GRID FORMATION FOR CONTOUR GENERATION

There are twelve points in x-direction its parallel to the largest dimension of diesel generator set & acoustic enclosure. There are ten points in y direction its parallel to the breadth of generator & acoustic enclosure. Z-direction is parallel to the height.

To find the variation in sound level at different positions around the generator divided height into six planes from z-1(30 Cm) to z-6(180 Cm) as shown in fig 6.8.



FIG 6.8 DIFFERENT PLANES IN Z DIRECTION

There are few point comes under the diesel generator and acoustic enclosure are shown in fig 6.9. Toward the source of noise sound level will increases. Consider that there will be maximum value of sound level on those points.

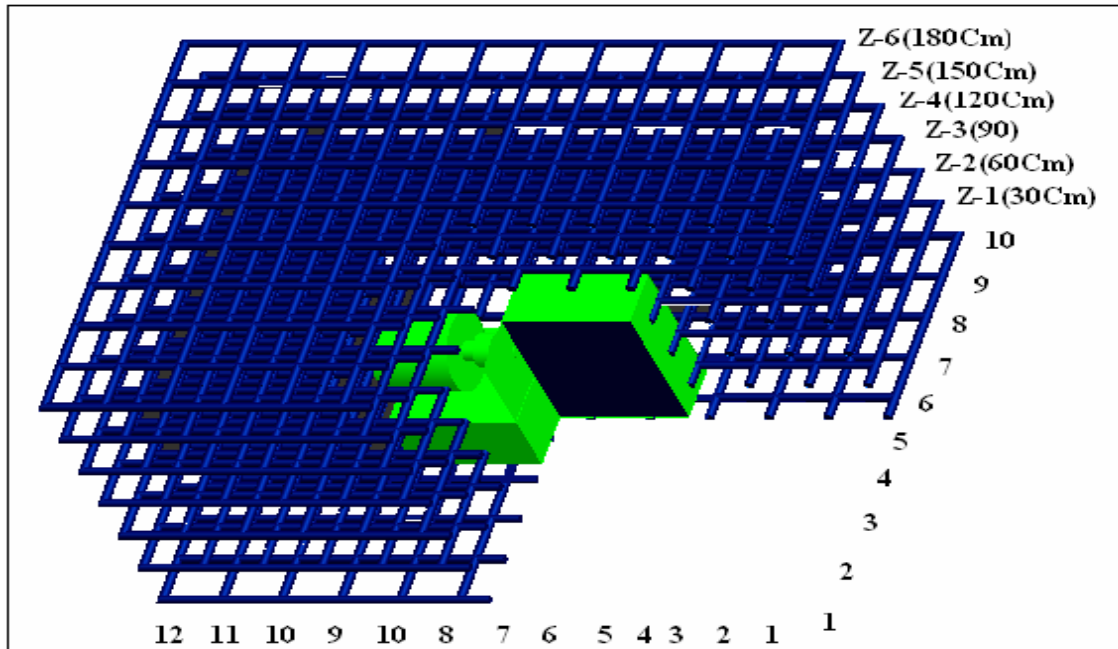


FIG 6.9 ISOMETRIC VIEW OF HYPOTHETICAL GRID FOR CONTOUR GENERATION

CHAPTER – 7

RESULTS & DISCUSSIONS

To check the effectiveness of the acoustic enclosure experimental work is carried out as per the procedure discussed in previous chapter. Results from the collected data are given below.

First experiment to check the role of particular change in reduction of noise production and to check that how much total reduction in noise is done by using an acoustic enclosure at different loads. Particular measuring parameter as follow:

1. Only Generator.
2. Only Silencer.
3. Only Enclosure.
4. Enclosure with silencer
5. Enclosure with Silencer & Inlet, outlet Duct.

1. ONLY GENERATOR:

Related data is given in appendix A-1

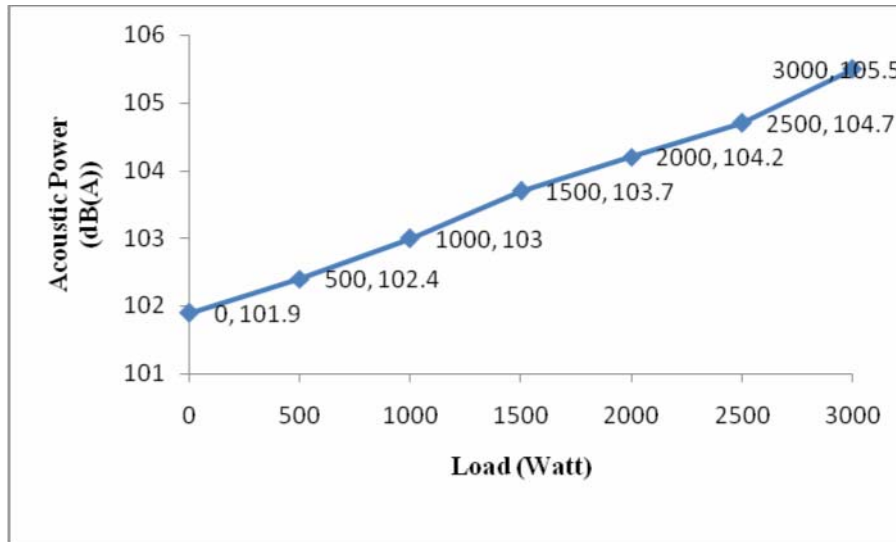


FIG .7.1 GRAPH OF ACOUSTIC POWER AT DIFFERENT LOAD WITH ONLY GENERATOR

2. ONLY SILENCER

Related data is given in appendix A-2

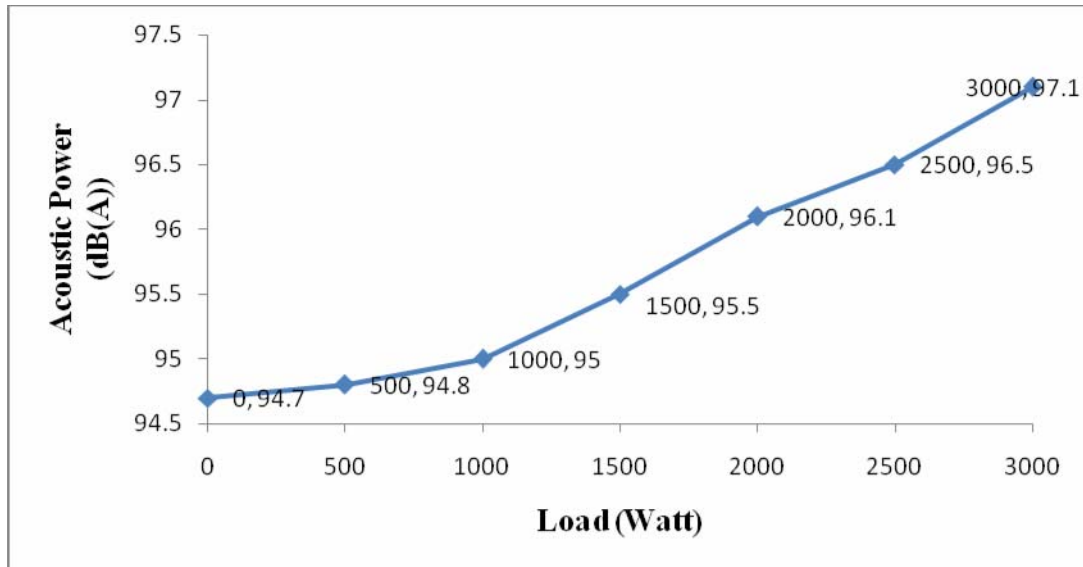


FIG .7.2 GRAPH OF ACOUSTIC POWER AT DIFFERENT LOAD WITH ONLY SILENCER

3. ONLY ENCLOSURE

Related data is given in appendix A-3

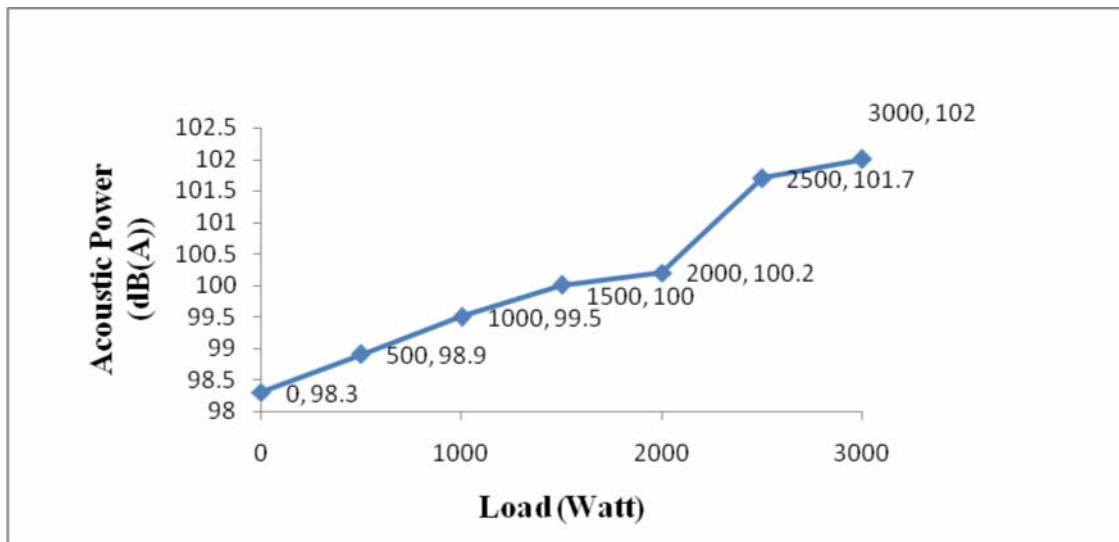


FIG .7.3 GRAPH OF ACOUSTIC POWER AT DIFFERENT LOAD WITH ONLY ENCLOSURE

4. ENCLOSURE WITH SILENCER

Related data is given in appendix A-4

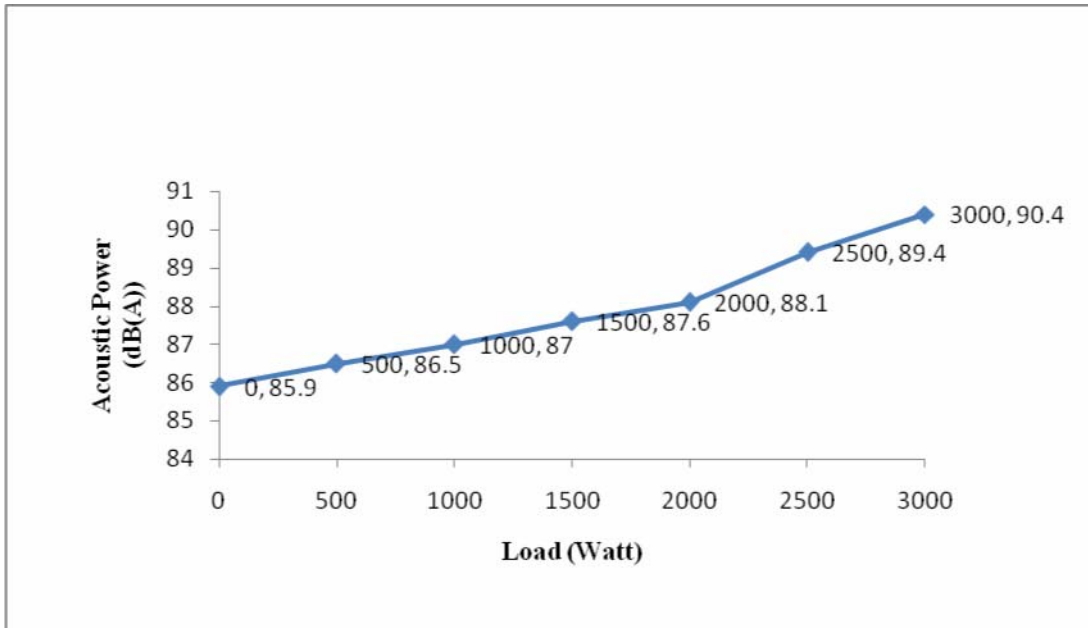


FIG .7.4 GRAPH OF ACOUSTIC POWER AT DIFFERENT LOADS WITH ENCLOSURE , SILENCER

5. ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT:

Related data is given in appendix A-5

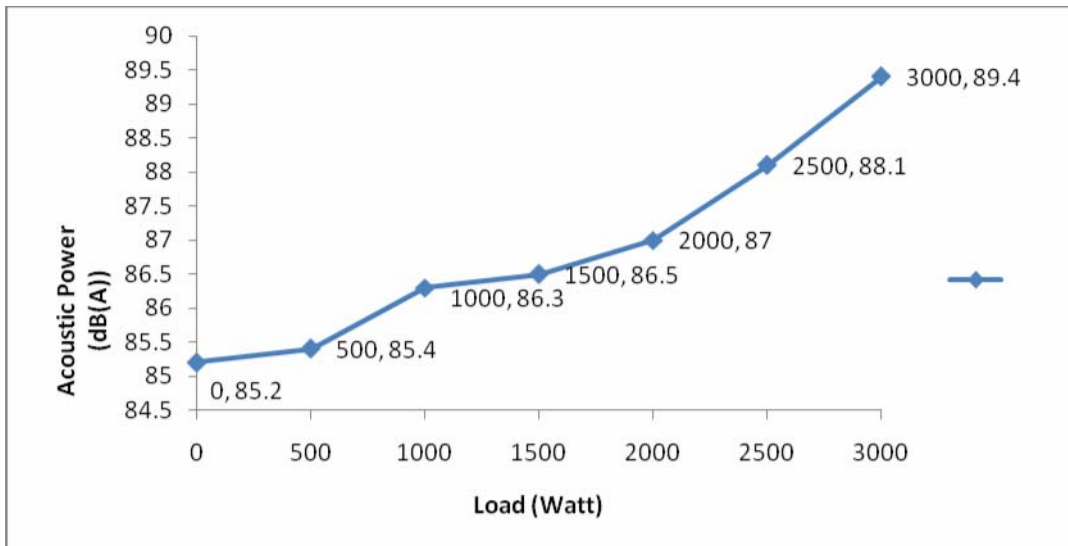


FIG .7.5 GRAPH OF ACOUSTIC POWER AT DIFFERENT LOAD WITH COMPLETE ENCLOSURE

GRAPH FOR DIFFERENT CONTROL MEASURE WITH LOAD VARIATION FROM 0kW TO 3kW

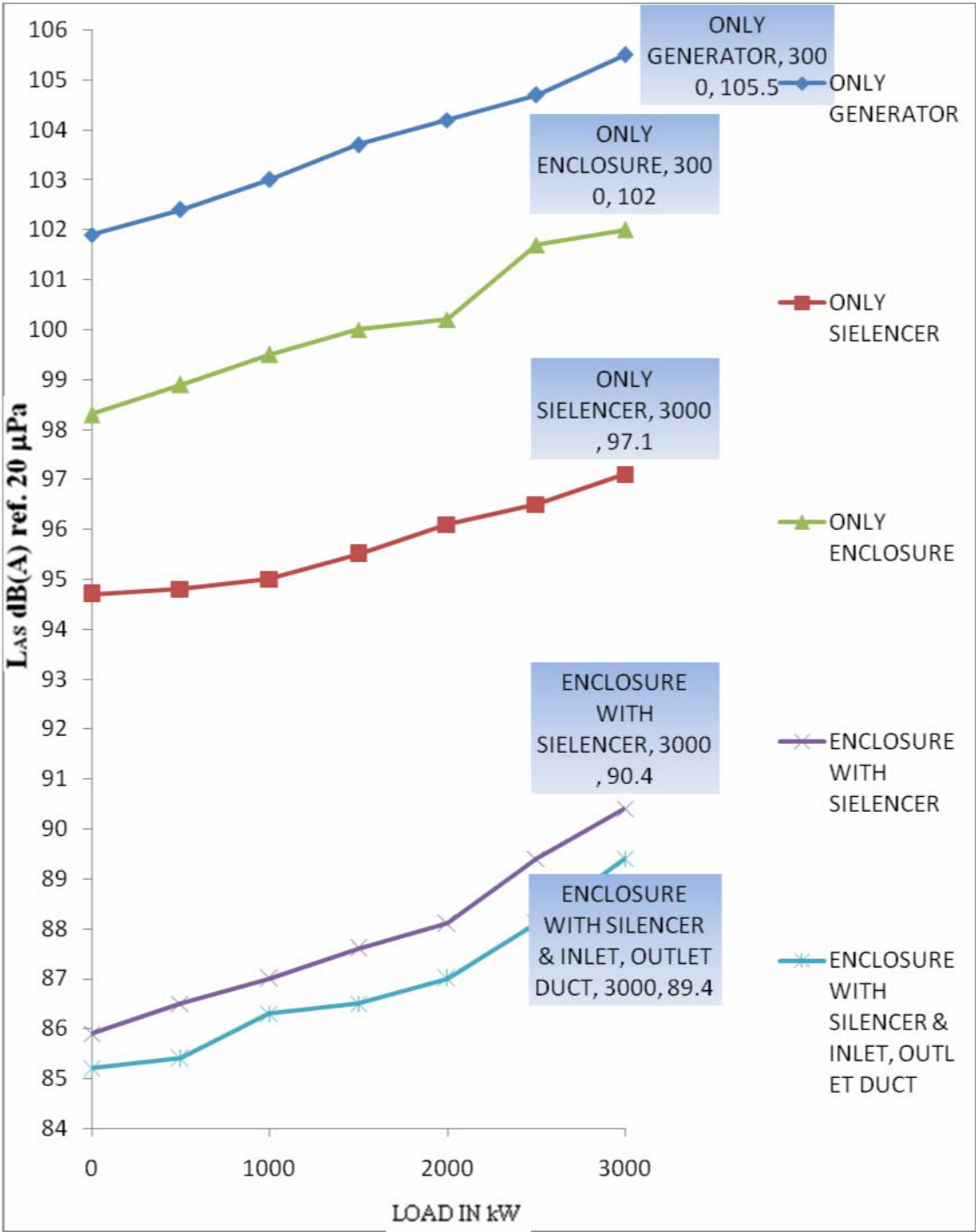


FIG .7.6 GRAPH FOR ACOUSTIC POWER /LOAD

RESULTS:--

S.NO	TYPE	Max. SOUND POWER LEVEL IN dB(A)	NOISE REDUCTION IN dB(A)
1	ONLY GENERATOR	105.2	
2	ONLY SILENCER	97.1	8.1
3	ONLY ENCLOSURE	102	3.1
4	ENCLOSURE WITH SILENCER	90.4	14.8
5	ENCLOSURE +SIL. + INLET &OUT LET DUCT	89.4	15.8

TABLE 7.1

DISCUSSION:

As shown in above table complete enclosure is working effectively. Various techniques discussed in chapter 4 applied in this experiment to control the noise of diesel generator. Their effectiveness to reduce noise describes bellow.

ONLY SILENCER:

It works very effectively as discussed in chapter-3. Maximum noise is produced by exhaust of the engine. Here results showing there is 8.1 dB (A). Sound power level reduces at 3kW load.

ONLY ENCLOSURE:

There is need to provide air ventilation inside the enclosure as discussed in chapter -4, partial barriers are provided to check there effectiveness. There is no silencer placed. From this experiment reduction in sound power level is 3.1 dB(A).

ENCLOSURE WITH SILENCER:

Both the silencer & enclosure together creating very good effect to control the noise. Silencer reduces the exhaust noise & enclosure stop the airborne transmission of noise. Together controlling sound power level 14.8 dB(A).

ENCLOSURE WITH SILENCER &INLET , OUTLET DUCT:

Technique discussed in chapter -4 fabricate air ventilation at right angle to avoid the direct air born transmission in source and receiver.

INLET &OUT LET DUCT:

Inlet and out let duct reduces sound power level (Here comparison scale is sound power level not sound pressure level).

COMBINED EFFECT OF SILENCER, INLET &OUTLET DUCT, AND PARTIAL BARRIERS:

Complete enclosure working effectively .Reducing 16 dB (A) sound power level at 3kW load. So it comes out from above discussion that when all the techniques work together reduced noise effectively.

1/1 OCTAVE BAND FREQUENCY SPECTRUM:

There are two major aspects to carried out this experiment

1. To determine those frequencies at which maximum sound pressure level is achieving at different loads and at different points.
2. Second aspect of this experiment is to find out the effect on those peak frequencies in following conditions.
 1. Only Generator.
 2. Only Silencer.
 3. Only Enclosure.
 4. Enclosure with silencer
 5. Enclosure with Silencer & Inlet, outlet Duct.

So to find the peak frequencies experiment is carried out as given in previous chapter. Results are displayed in the foam of graphs. Related data collected and written in appendix –B

There is a wide range of frequencies in 1/1Octave band from 32 to 8000 Hz. It is decided that data should be represent in two graphs.

- First graph to find rang of peak frequencies and to determine the behavior of sound pressure level with the increasing level of frequencies.
- Second enlarged scale graphs to determine the accurate value of frequency at particular load.

ONLY GENERATOR

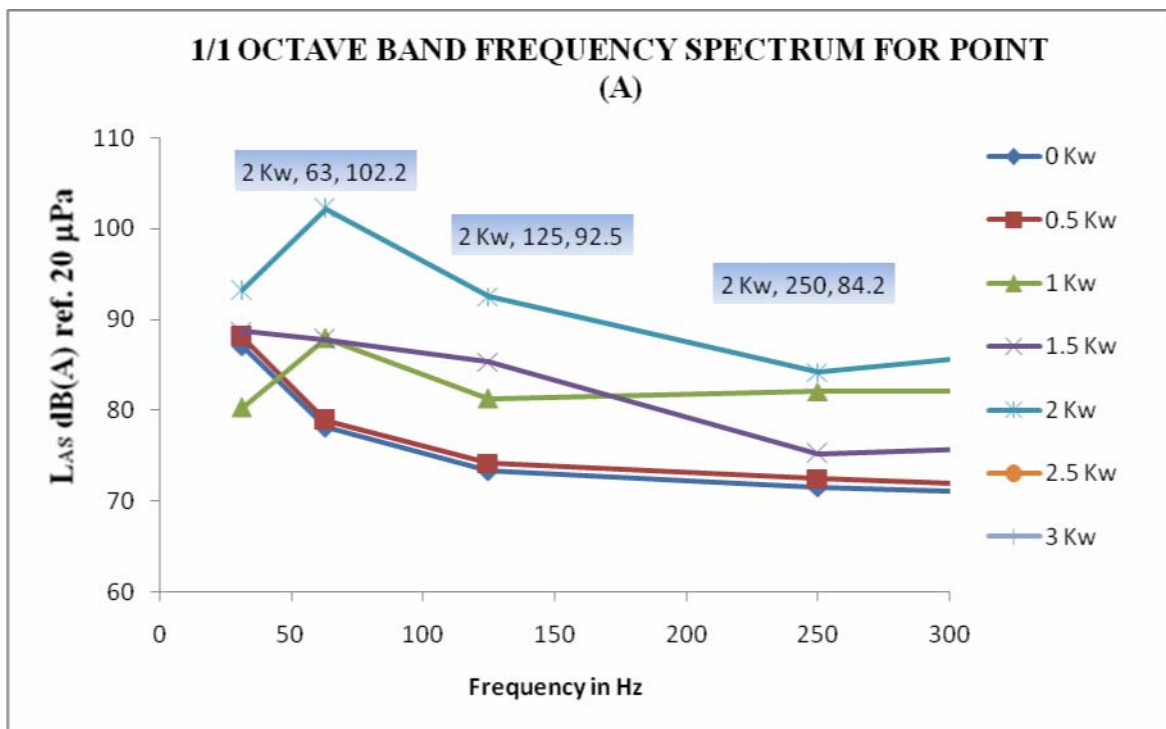
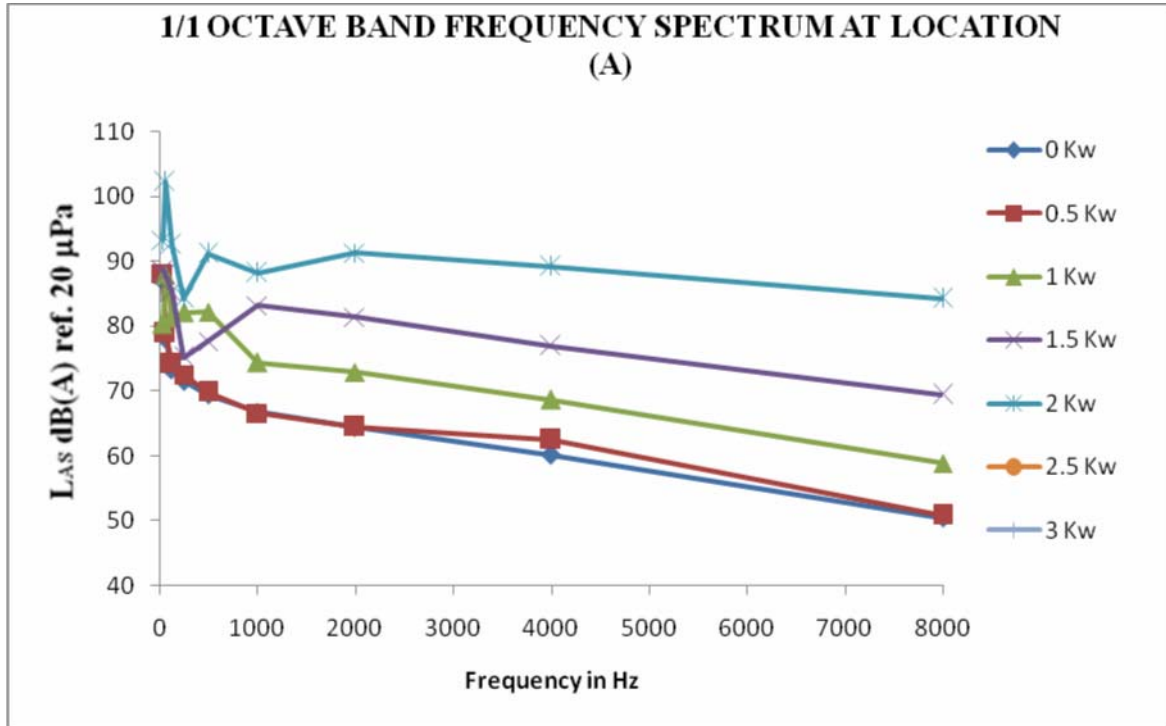


FIG 7.7 NOISE LEVEL GRAPH AT LOCATION (A)

ONLY SILENCER

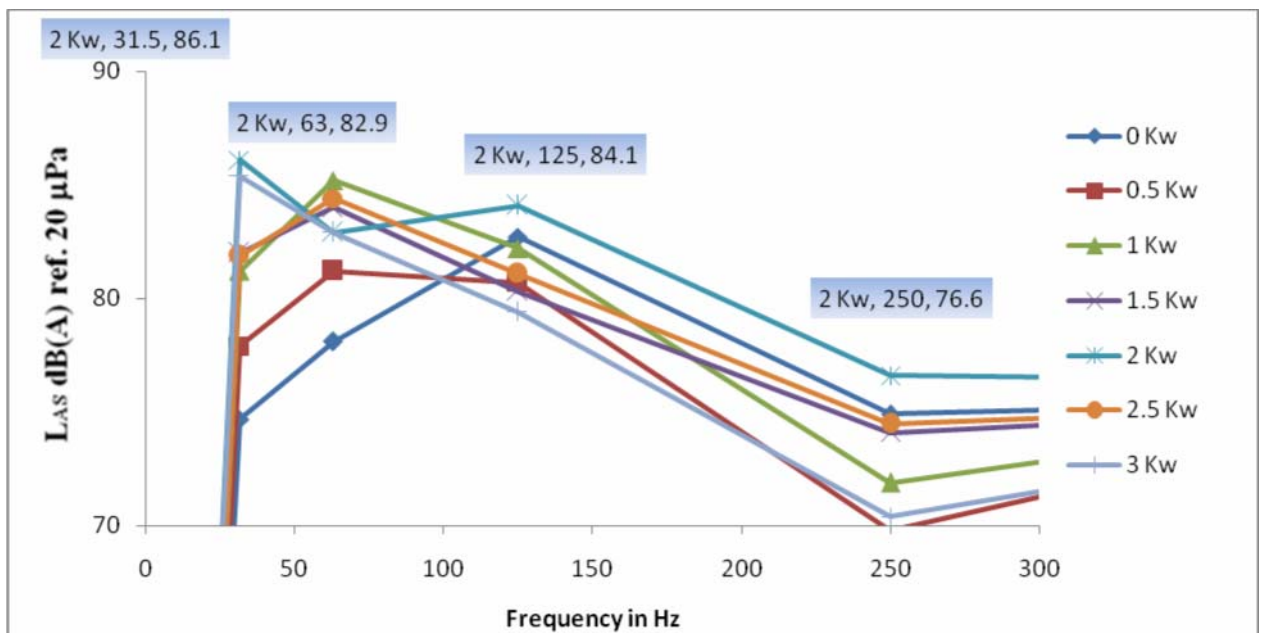
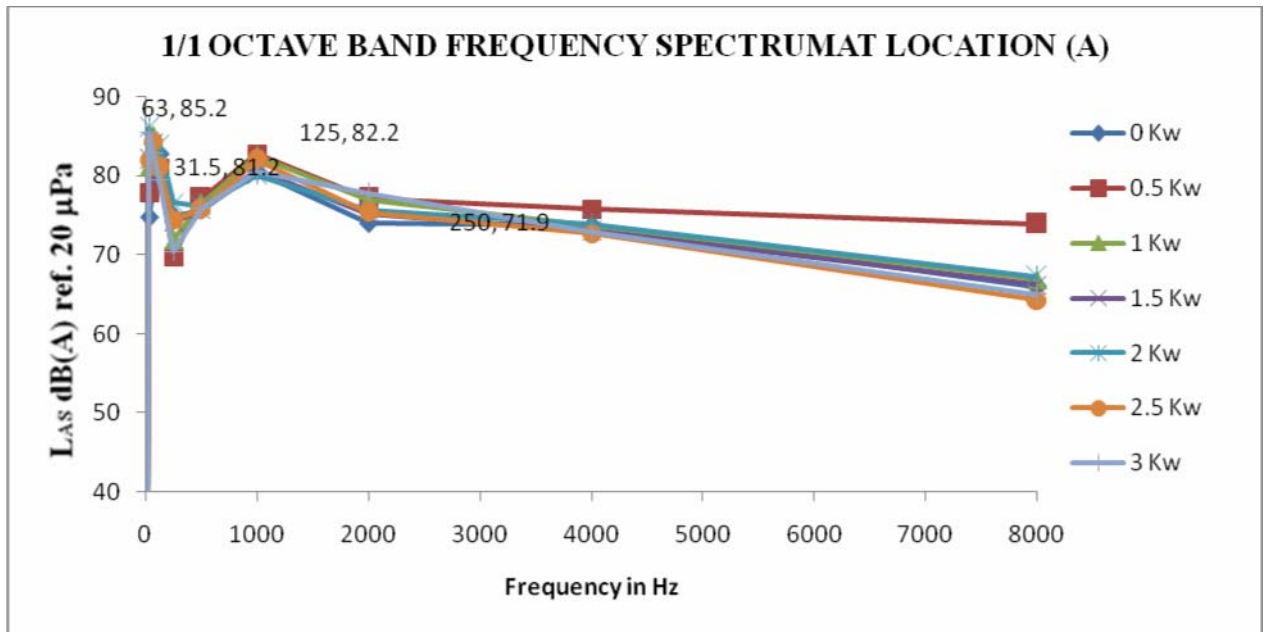


FIG 7.8 NOISE LEVEL GRAPHS AT LOCATION (A)

ONLY ENCLOSURE

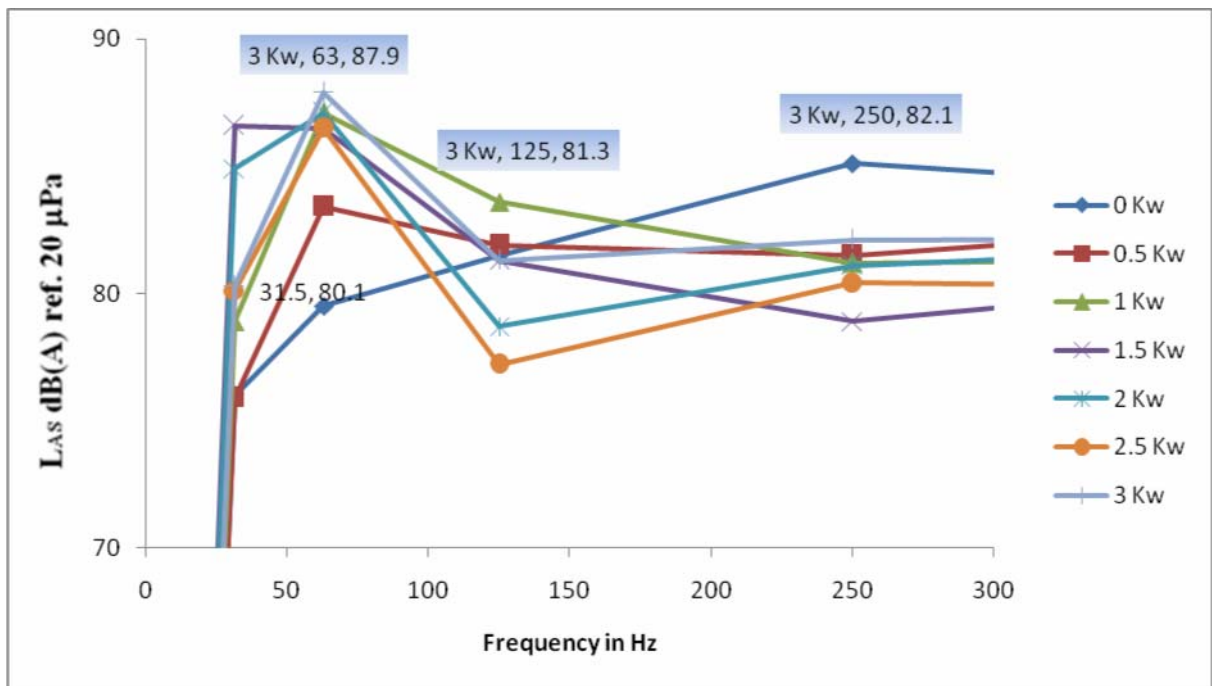
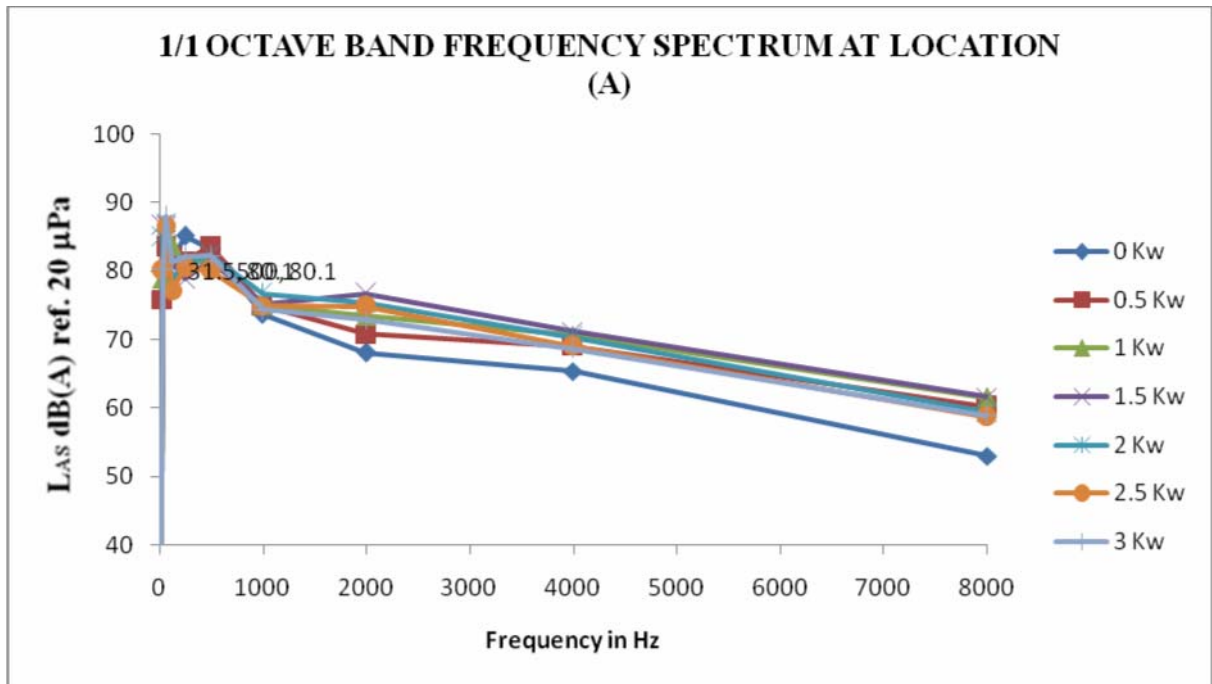


FIG 7.9 NOISE LEVEL GRAPH ONLY WITH ENCLOSURE

ENCLOSURE WITH SILENCER

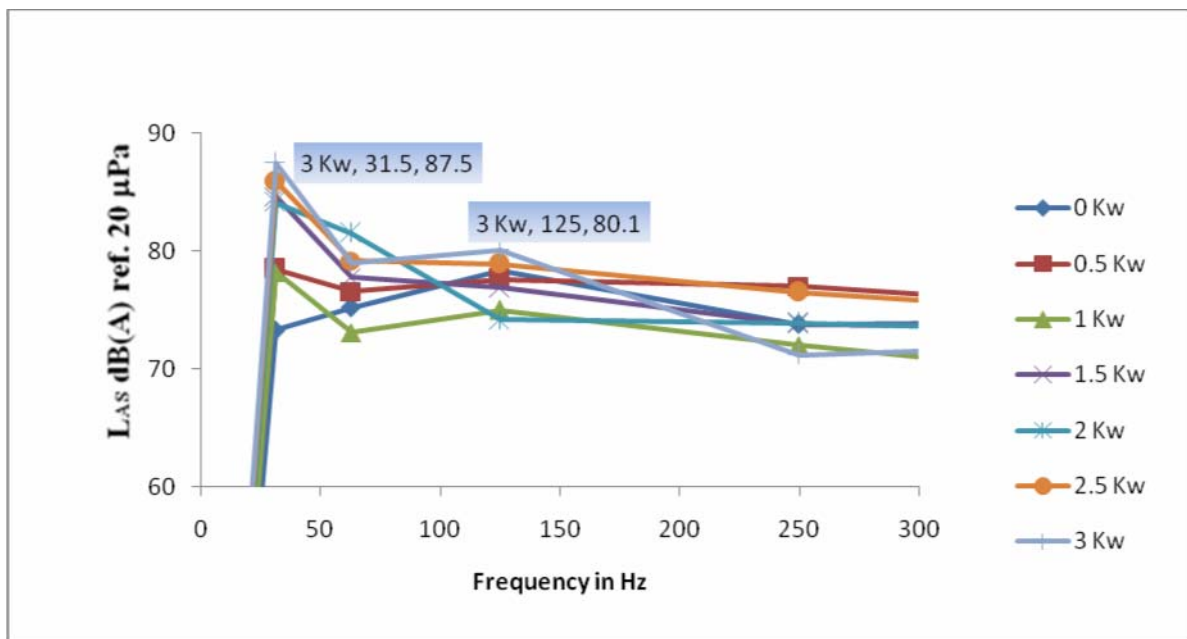
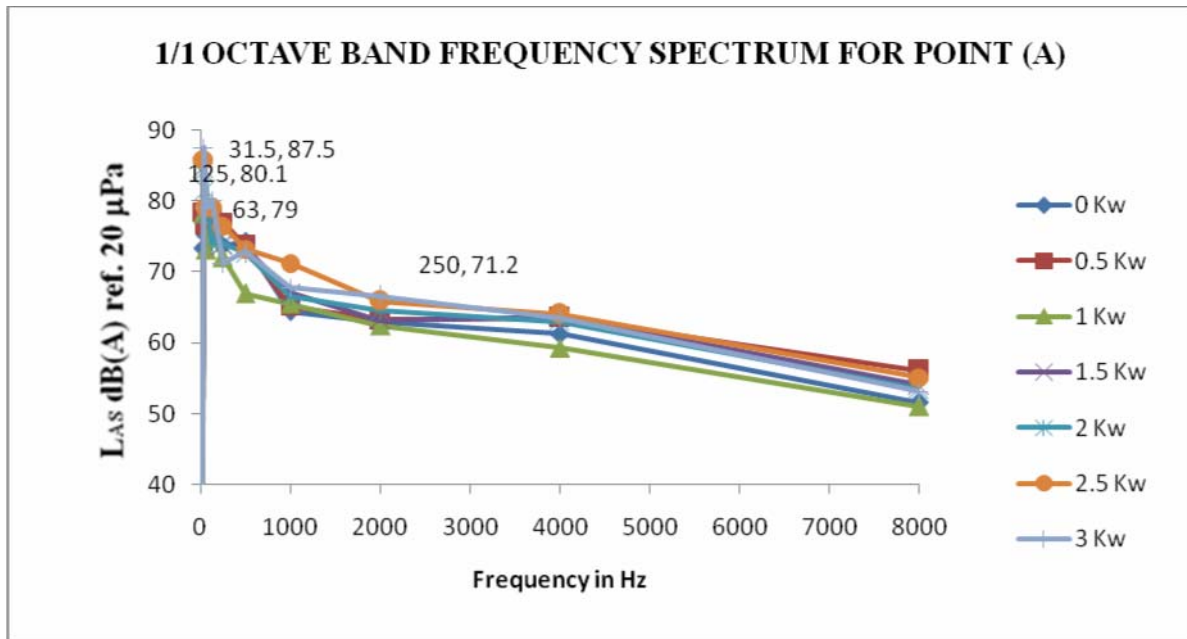


FIG 7.10 NOISE LEVEL GRAPH WITH ENCLOSURE+SILENCER

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT

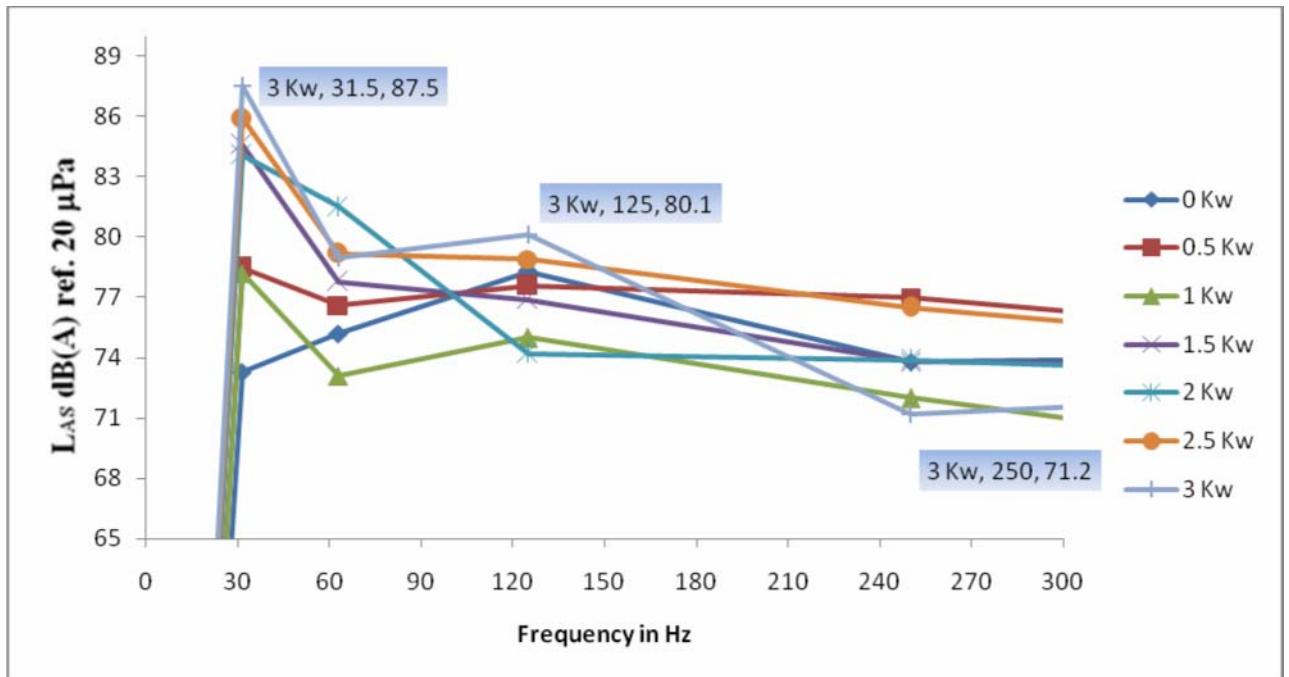
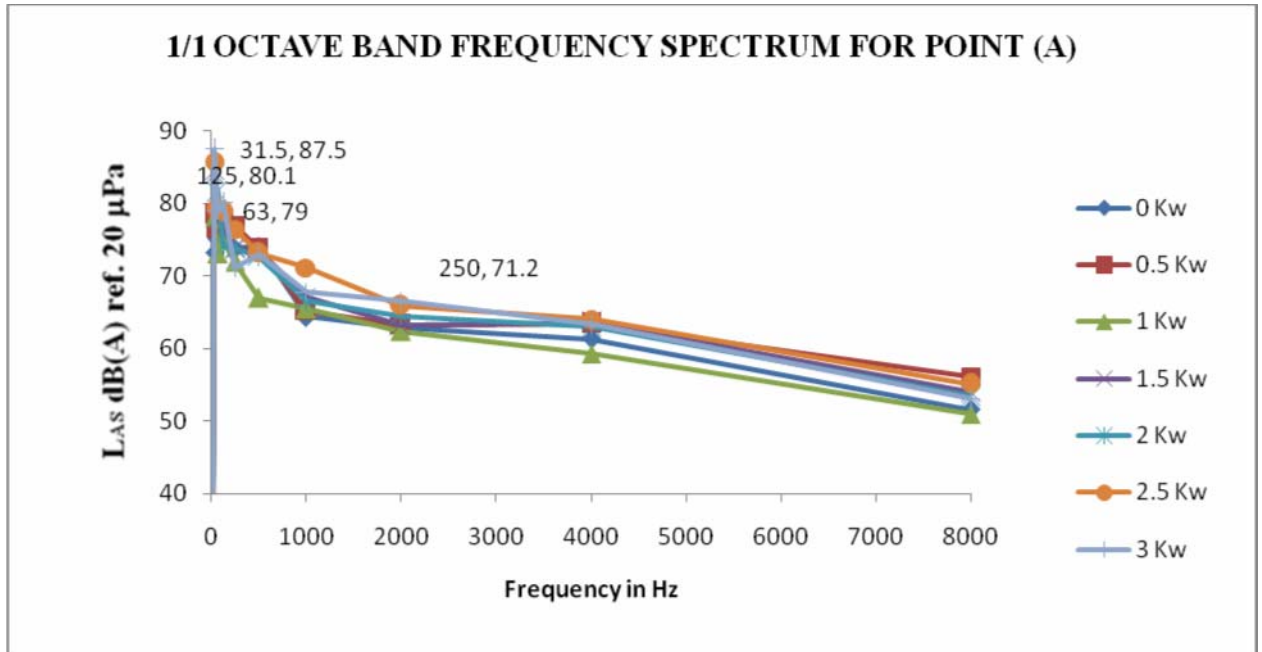


FIG 7.11 NOISE LEVEL GRAPH WITH ENCLOSURE+SILENCER

FOR DIFFERENT CONTROL MEASURES 1/OCTAVE BAND FREQUENCY SPECTRUM

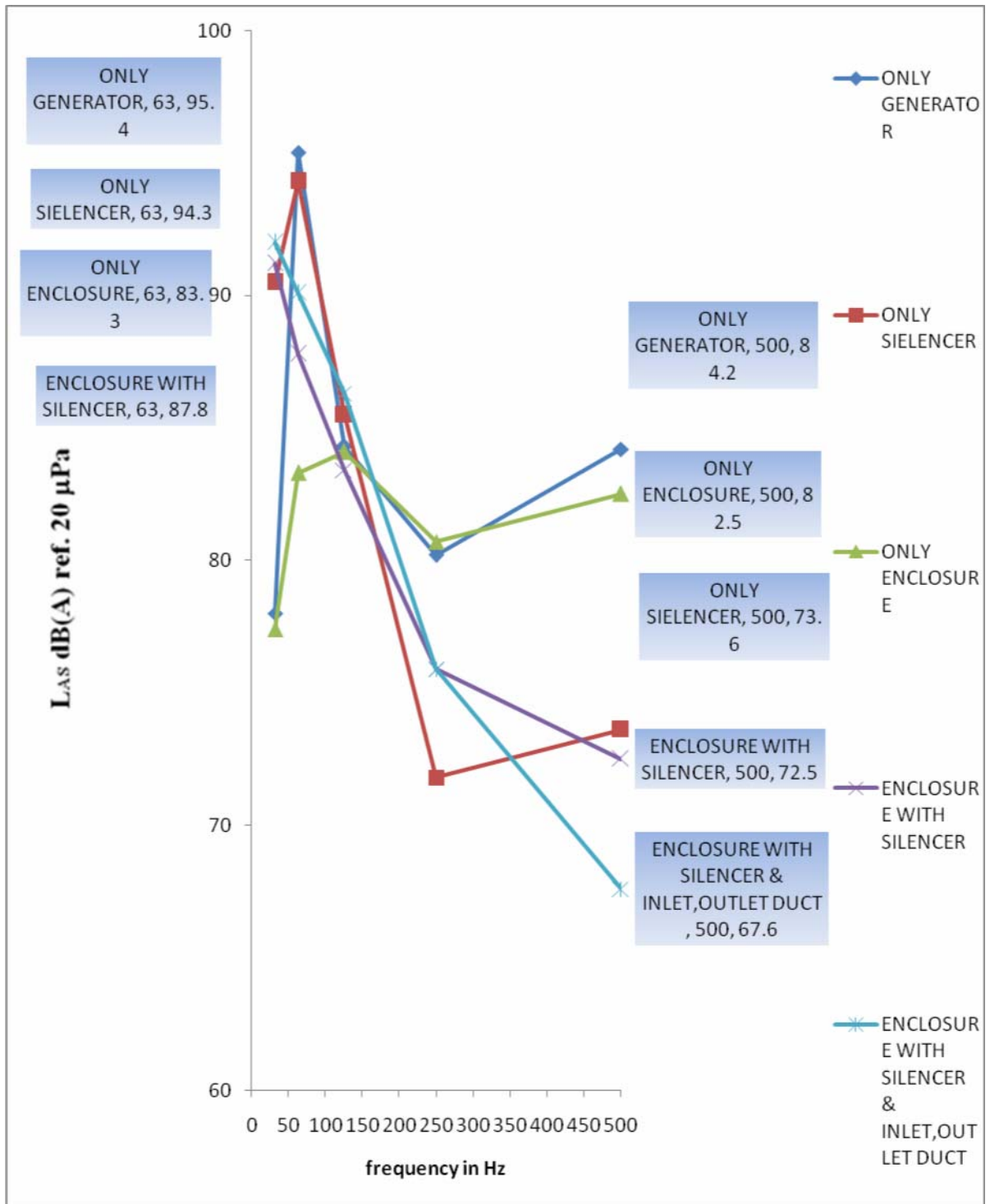


FIG 7.12

RESULT AT LOCATION (A):

S.NO.	TYPE	MAX. Las in dB(A)	PEAK FREQUENCY IN Hz	OBSERVED AT LOAD IN kW
1	ONLY GENERATOR	102.2	63	2
2	ONLY SILENCER	90.9	63	2
3	ONLY ENCLOSURE	92.9	63	3
4	ENCLOSURE WITH SILENCER	88.5	31.5	3
5	ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT	87.5	31.5	2

TABLE 7.2

DISCUSSION:

As discussed in the beginning of this experiment first aspect from this experiment was to find out the peak sound pressure level frequencies. Results from table 7.2 are as follow.

1. Peak sound pressure levels occur at lower frequencies at 63Hz, 31.5Hz.

It discussed in Chapter-5 that sound absorptive material are capable to reduce the peak frequencies. It observed from fig 7.12 that:

1. Complete enclosure with sound absorptive material can effectively reduce the peak sound pressure levels.
2. It observed that sound pressure levels are continuously reduces during the use of enclosure with silencer & inlet outlet duct.

Linear values of Sound Pressure Level at Location A at Different Operating Conditions at Load 3 kW			
Sr. No	Operating Conditions	Sound Pressure Levels dB(A)	Effectiveness of Individuals dB(A)
1	Only Generator	98.0	
2	Only Silencer	90.0	8.0
3	Only Enclosure	88.1	9.9
4	Enclosure with Silencer	84.1	13.9
5	Enclosure with Silencer and Outlet Duct	76.3	21.7

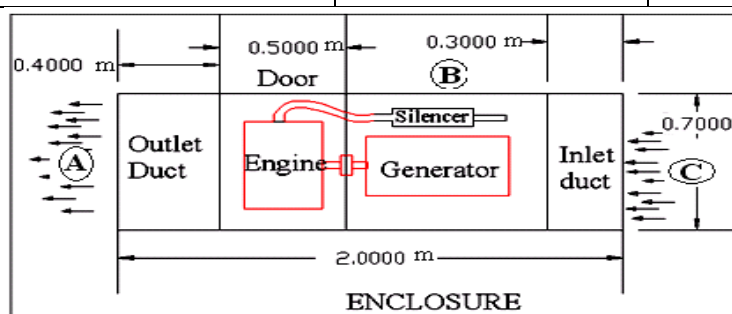


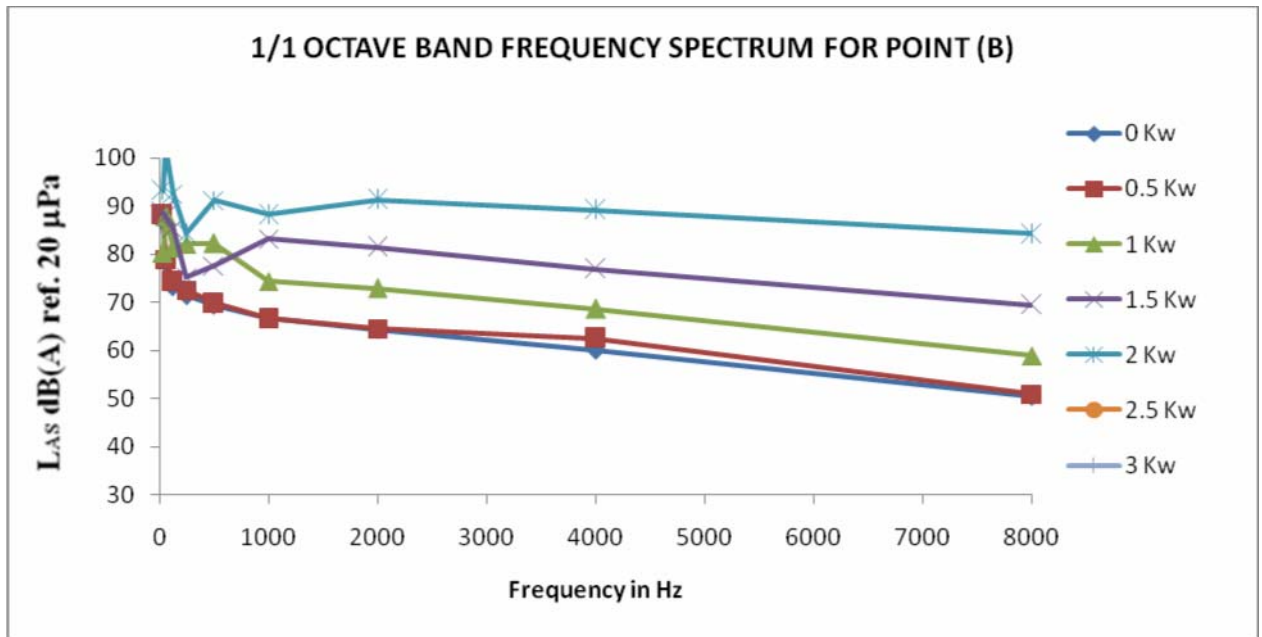
FIG NO7.13 LOCATION OF A, B, C

It has been observed that

1. With the use of only silencer, Sound Pressure Level at location A reduces by 8 dB(A).

2. With the enclosure the Sound Pressure Level reduces by 9.9 dB(A). It has been analyzed that the use of partial barriers and poly urethane form worked effectively.
3. The Sound Pressure Level reduces 8 dB(A) with the addition of Silencer. It results that well designed silencer could be much effective.
4. It has been very clearly observed that Sound Pressure Level reduced by 7.8 dB(A) at location A as shown in Fig.7.13 with the use of only outlet duct.

ONLY GENERATOR



NORMAL GRAPH

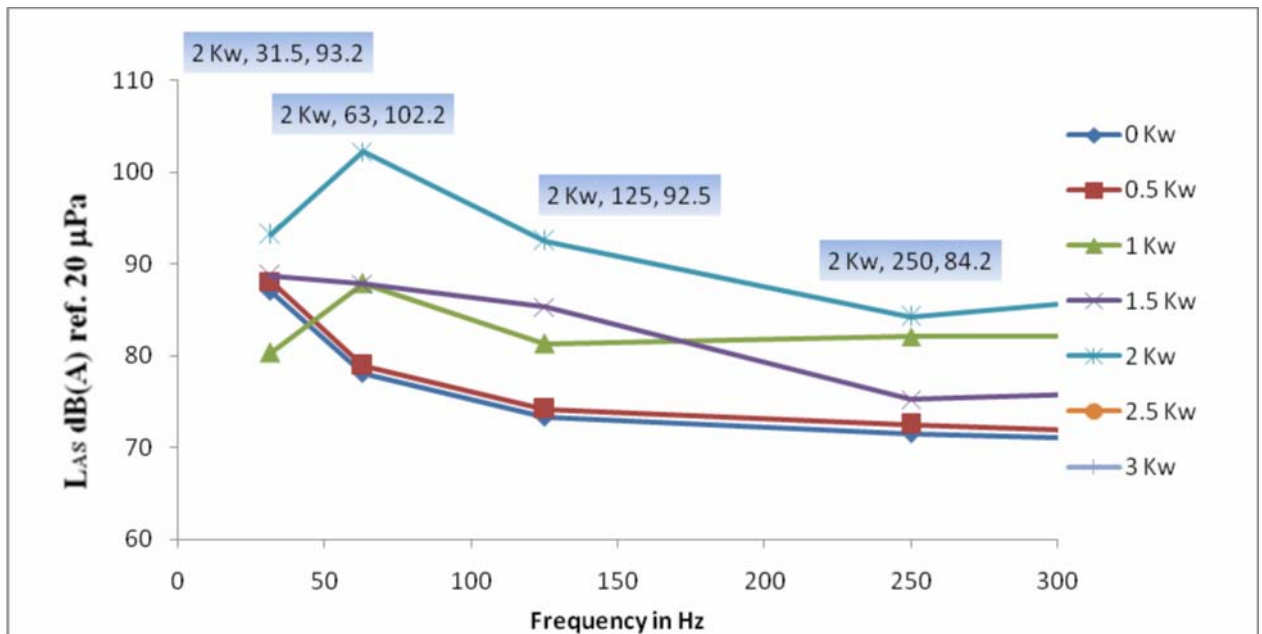
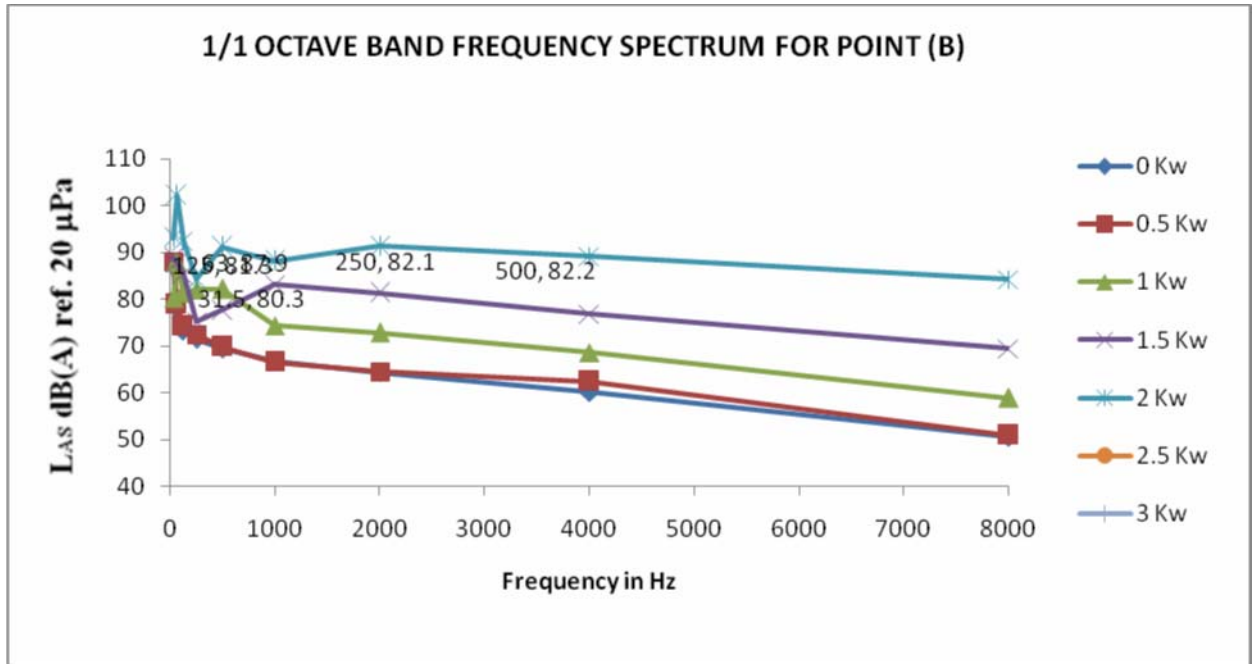


FIG 7.14 ENLARGED GRAPH

ONLY SILENCER



NORMAL GRAPH

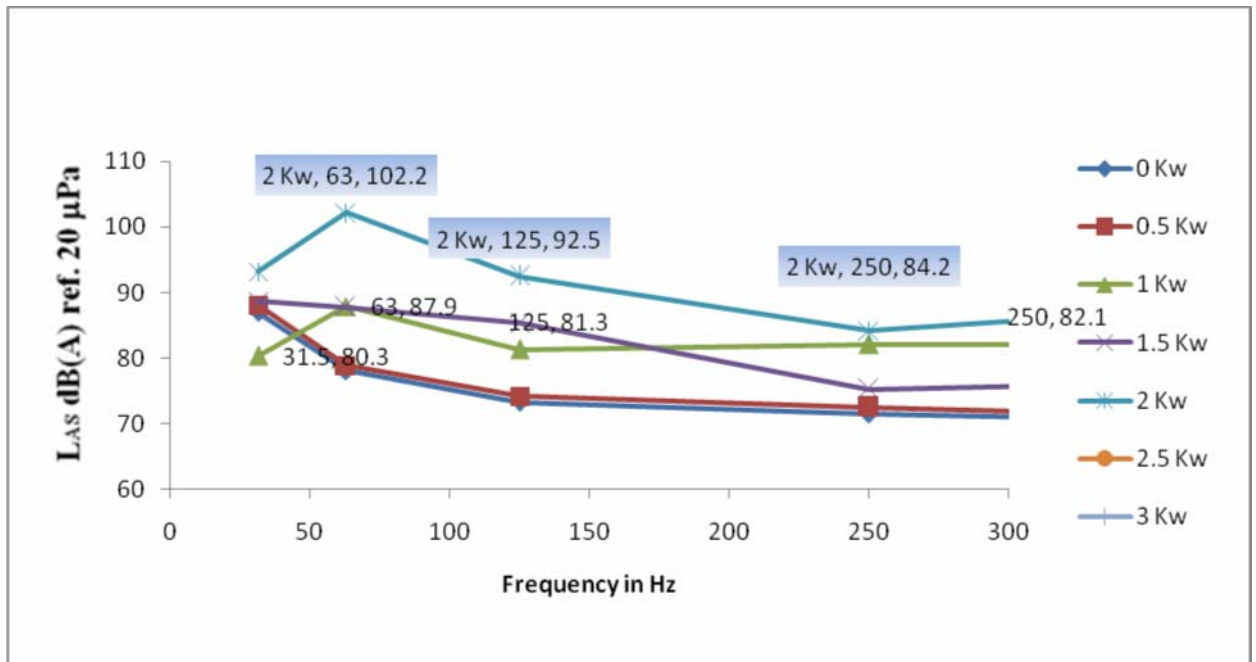
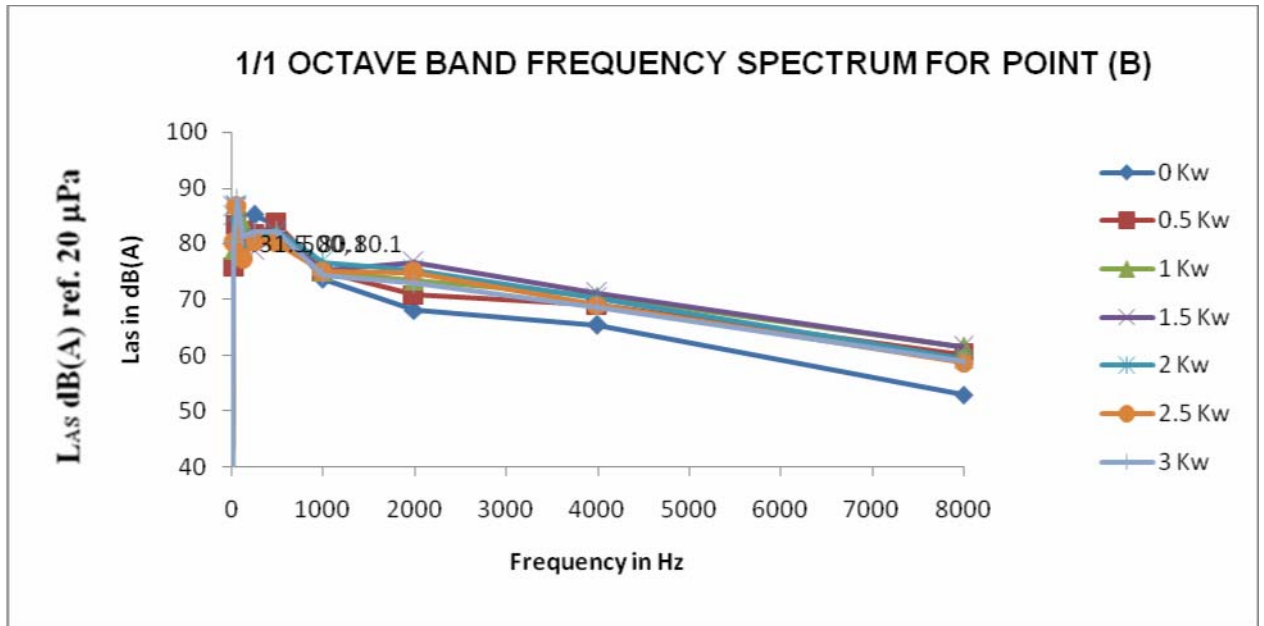


FIG 7.15 ENLARGED GRAPH

ONLY ENCLOSURE



NORMAL GRAPH

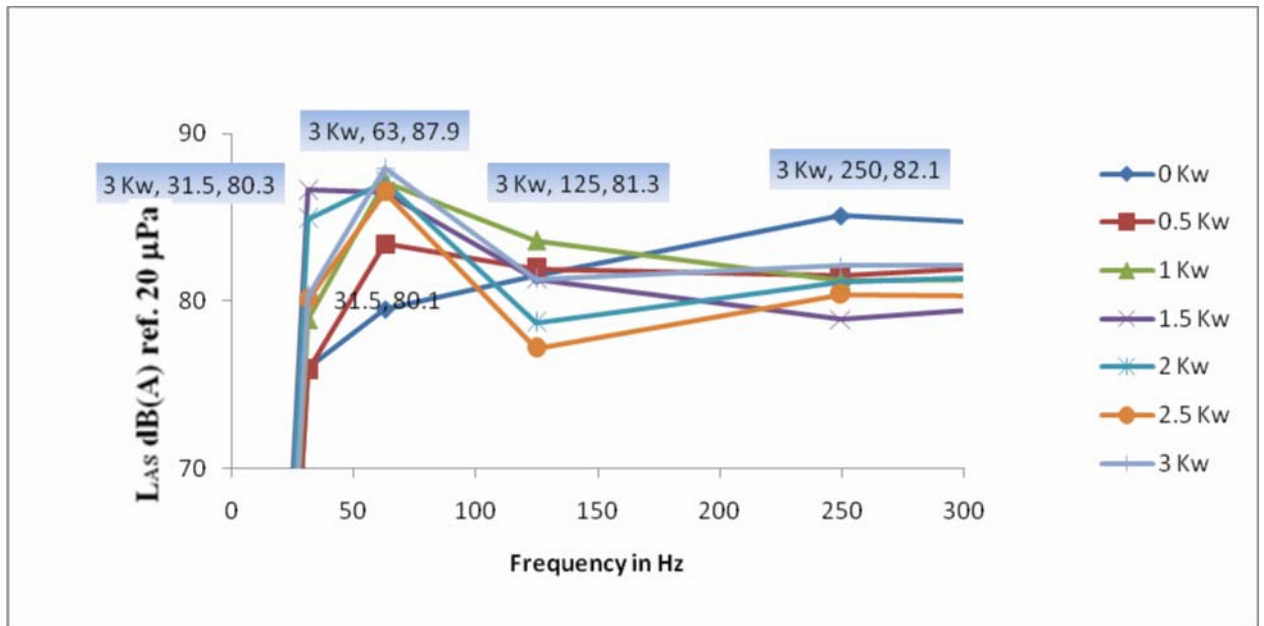
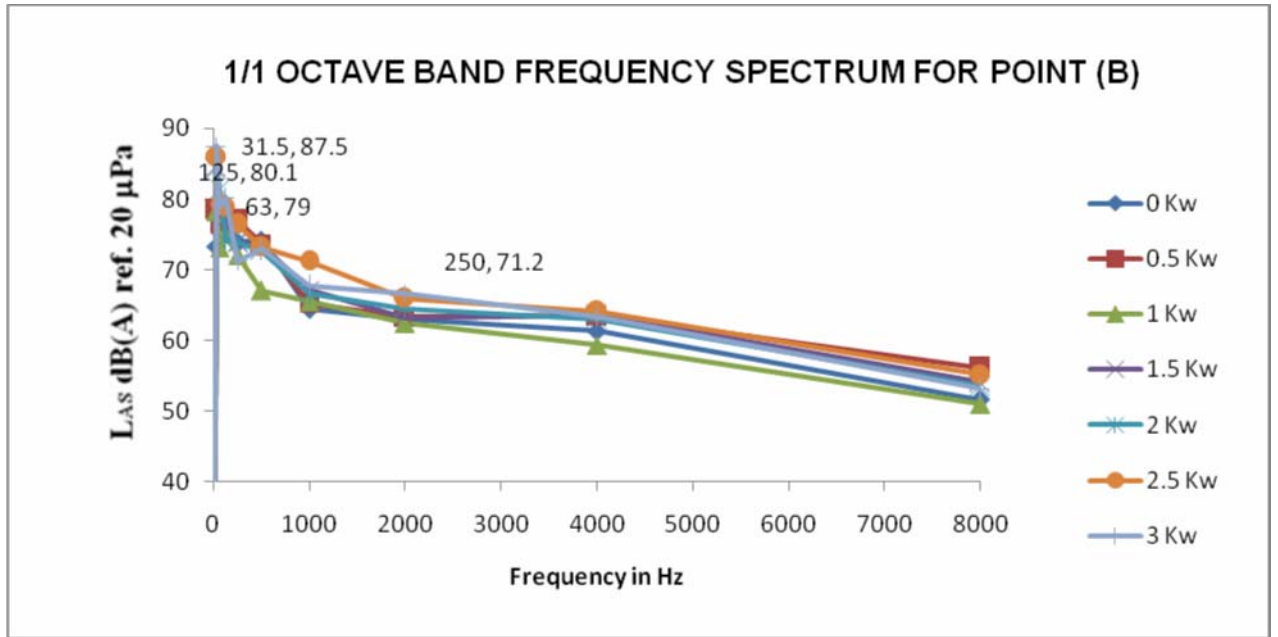


FIG 7.16ENLARGED GRAPH

ENCLOSURE WITH SILENCER



NORMAL GRAPH

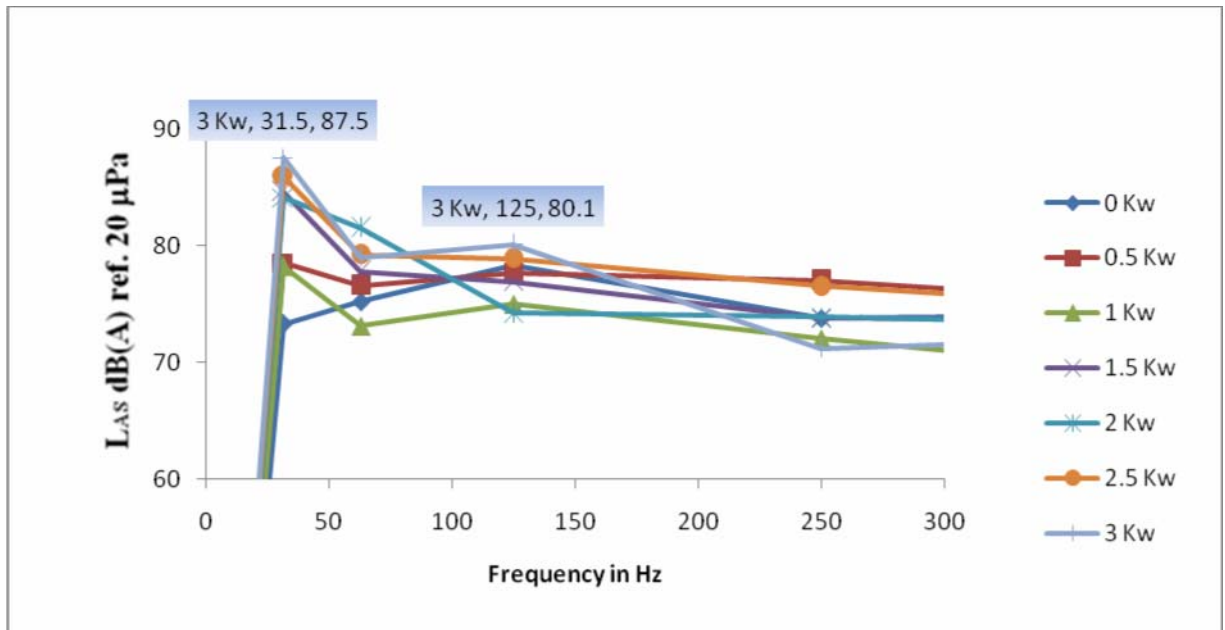
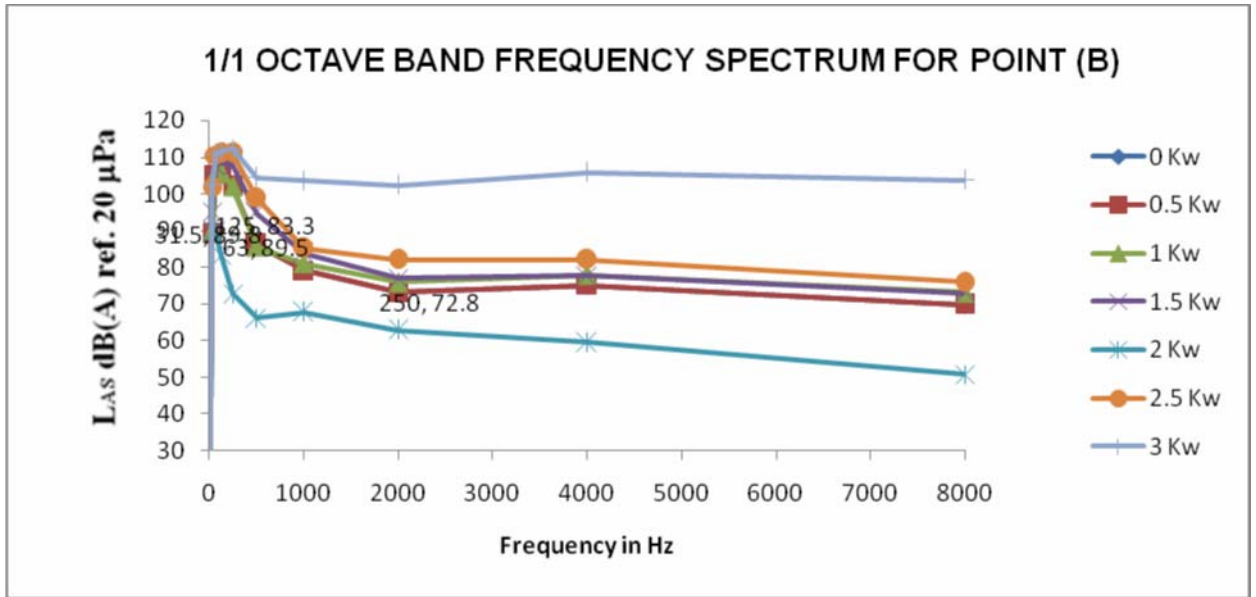


FIG 7.17ENLARGED GRAPH

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT



NORMAL GRAPH

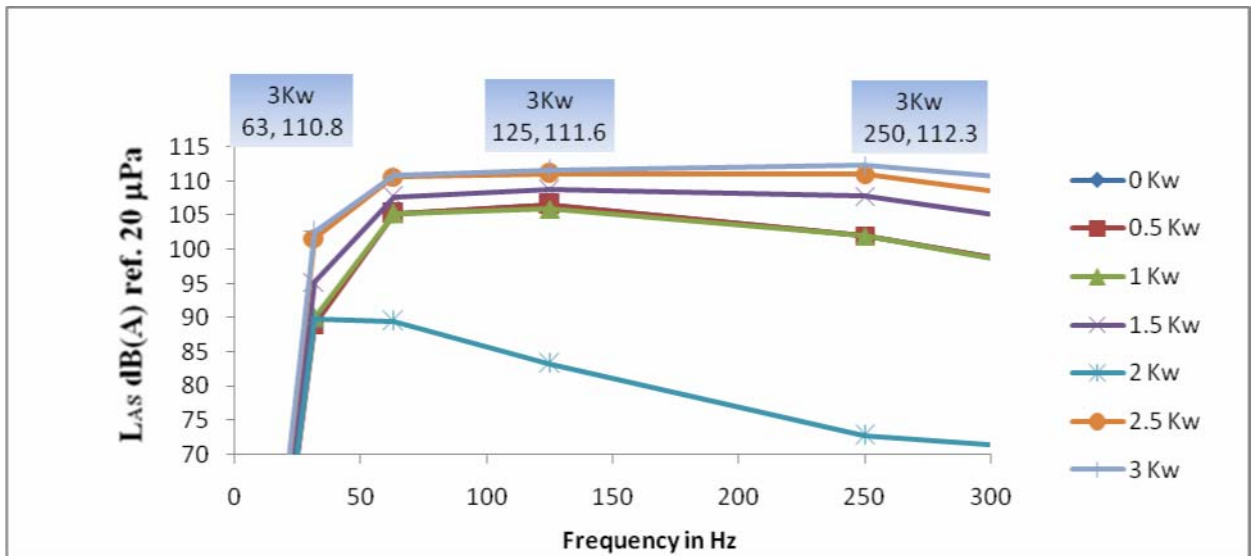


FIG 7.18 ENLARGED GRAPH

GRAPH FOR DIFFERENT CONTROL MEASURES AT LOCATION (B)

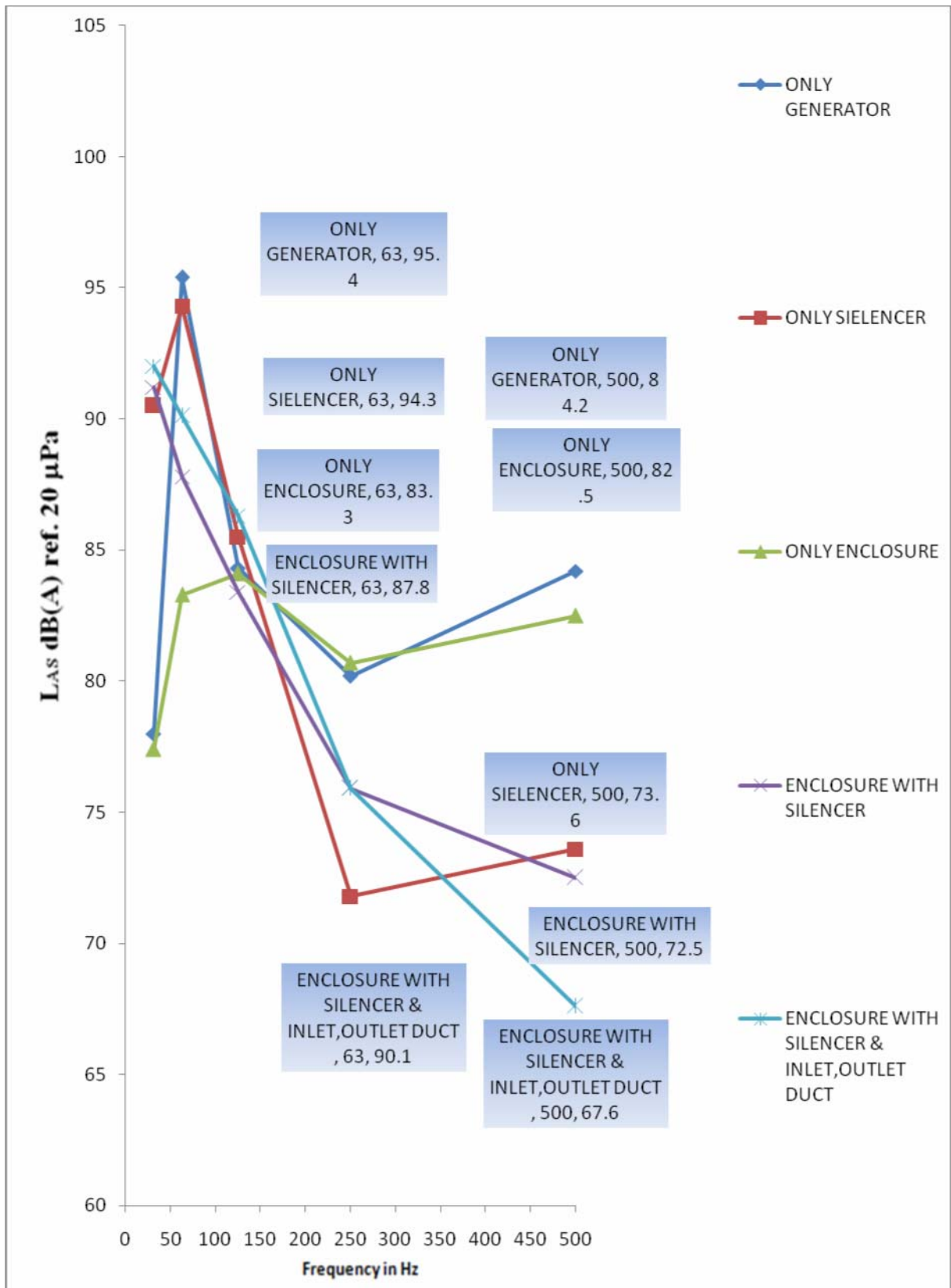


FIG 7.19

RESULTS FROM POINT (B):

S.NO.	TYPE	MAX. Las in dB(A)	PEAK FREQUENCY IN Hz	OBSERVED AT LOAD IN kW
1	ONLY GENERATOR	93.2	31.5	2
2	ONLY SILENCER	102.2	63	2
3	ONLY ENCLOSURE	87.9	63	3
4	ENCLOSURE WITH SILENCER	87.5	31.5	3
5	ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT	86.5	63	3

TABLE 7.3

1. Peak sound pressure levels occur at lower frequencies at 63Hz, 31.5Hz.
2. Peak sound pressure levels are achieving at 2kW,3Kw loads.

It discussed in Chapter-5 that sound absorptive material are capable to reduce the peak frequencies. It observed from fig 7.19 that:

3. Complete enclosure with sound absorptive material can effectively reduce the peak sound pressure levels.
4. Sound pressure level decreases continuously with respect to increase in lower frequencies from 31.5 to 500Hz. (shown in fig 7.19) when enclosure with silencer & inlet outlet is used.

Linear values of Sound Pressure Level at Location B at Different Operating Conditions at Load 3 kW			
Sr. No	Operating Conditions	Sound Pressure Levels dB(A)	Effectiveness of Individuals dB(A)
1	Only Generator	92.1	
2	Only Silencer	88.2	3.9
3	Only Enclosure	89.4	2.7
4	Enclosure with Silencer	82.5	9.6
5	Enclosure , Silencer, Inlet Duct	79.3	12.8

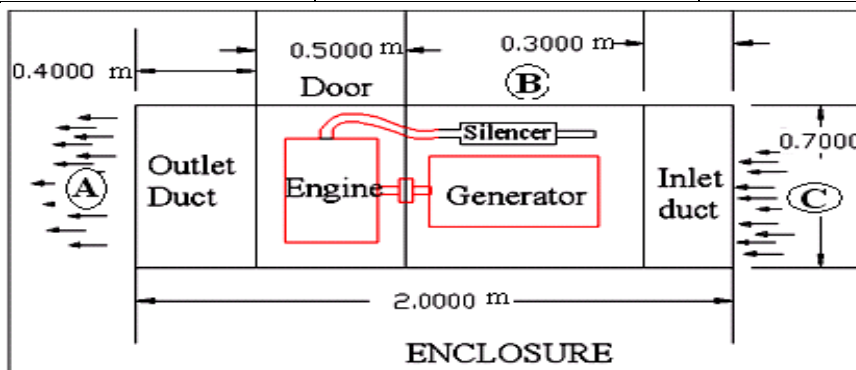


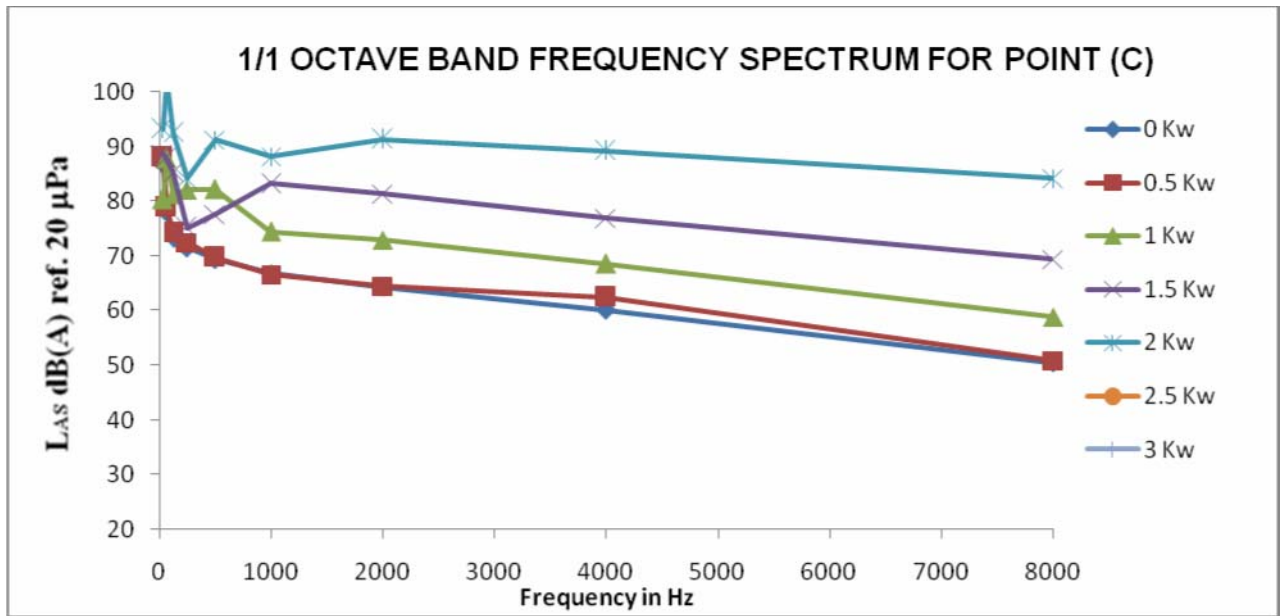
FIG 7.20 LOCATION OF A, B, C POINTS

It has been observed that

1. With the use of only silencer, Sound Pressure Level at location A reduces by 3.9 dB(A).
2. With the enclosure the Sound Pressure Level reduces by 2.7 dB(A). It has been analyzed that the use of partial barriers and poly urethane form worked effectively.
3. The Sound Pressure Level more reduces with the addition of Silencer.

4. It has been very clearly observed that Sound Pressure Level reduced by 3.2 dB(A) only by the the use of only inlet duct.

ONLY GENERATOR



NORMAL GRAPH

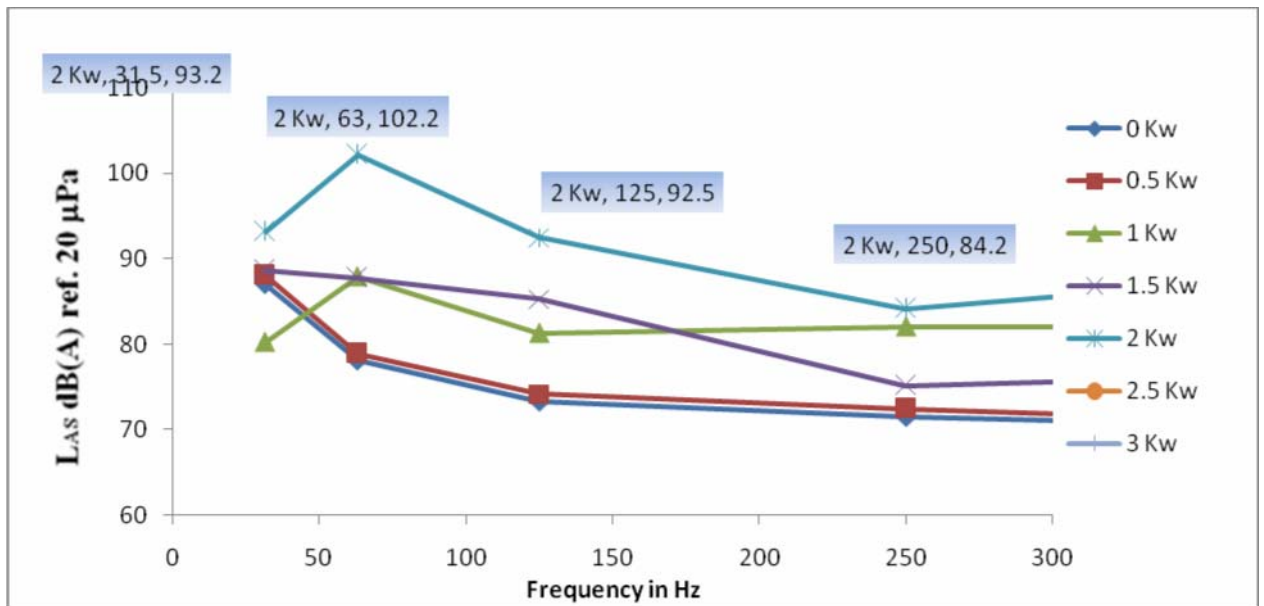
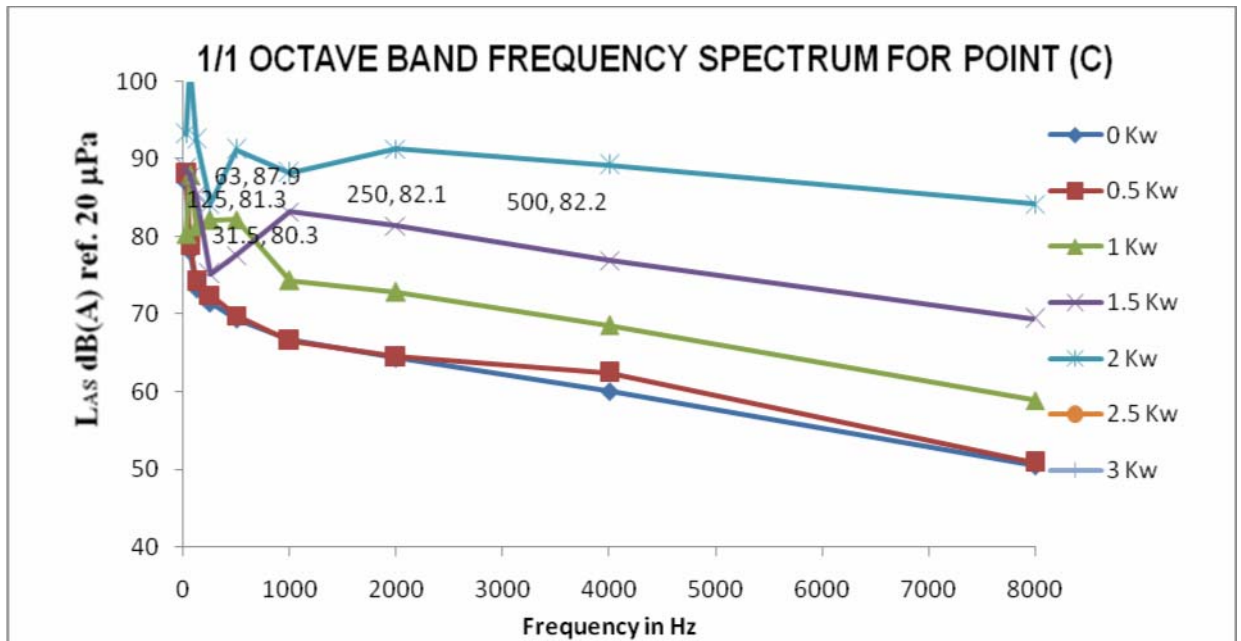


FIG 7.21 ENLARGED GRAPH

ONLY SILENCER



NORMAL GRAPH

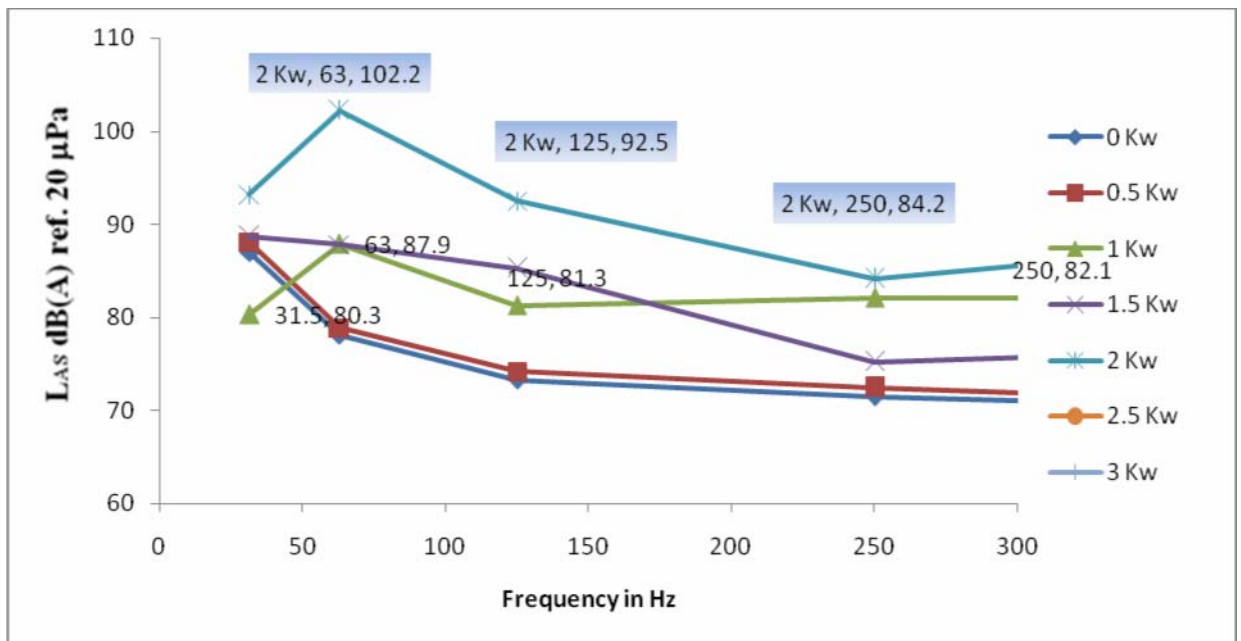
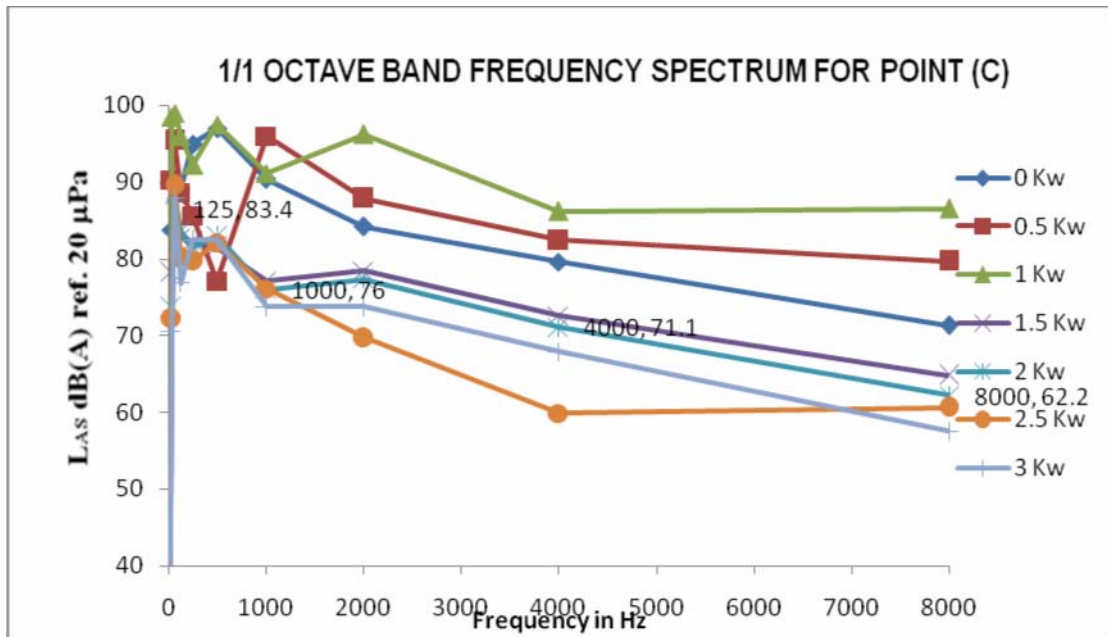


FIG 7.22 ENLARGED GRAPH

ONLY ENCLOSURE



NORMAL GRAPH

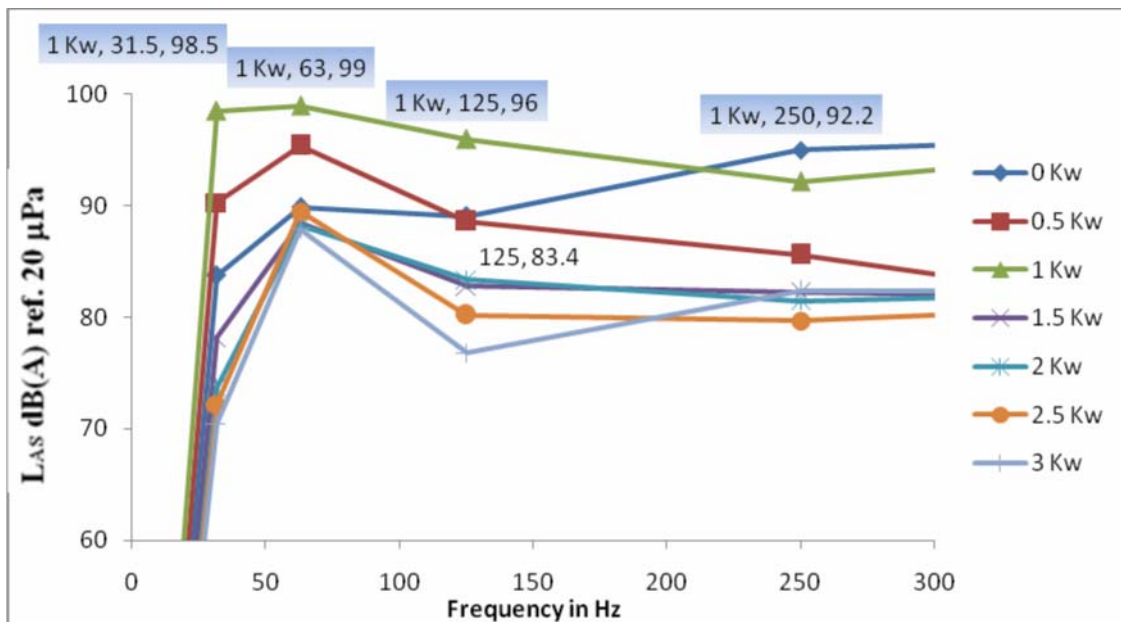
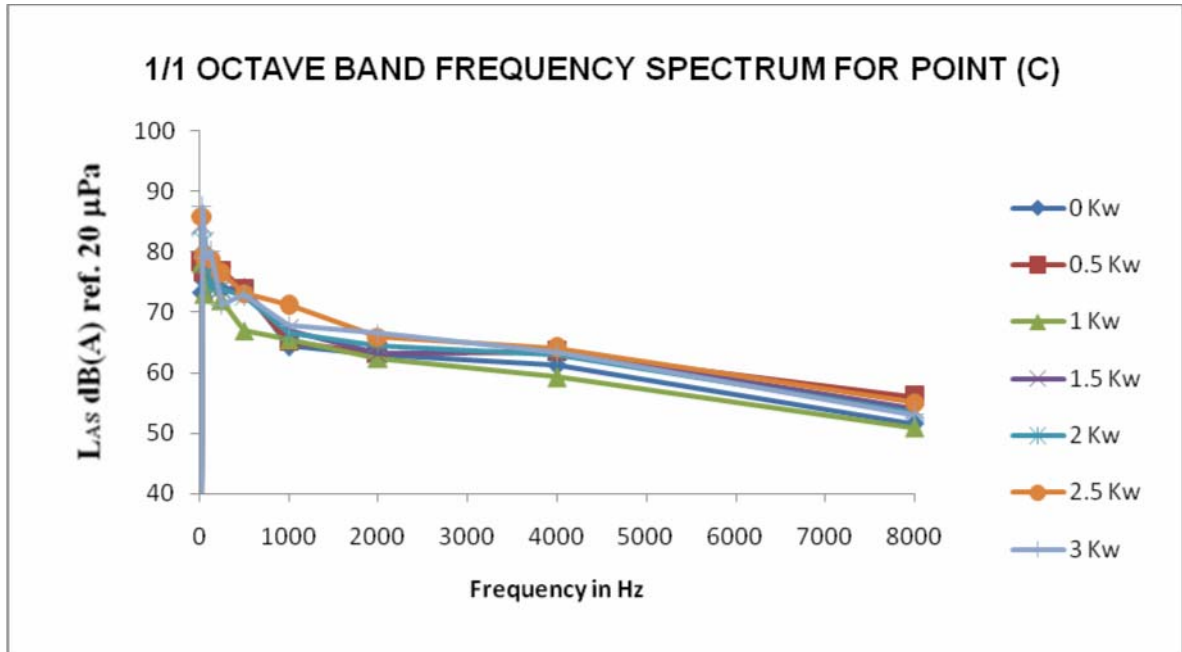


FIG 7.23 ENLARGED GRAPH

ENCLOSURE WITH SILENCER



NORMAL GRAPH

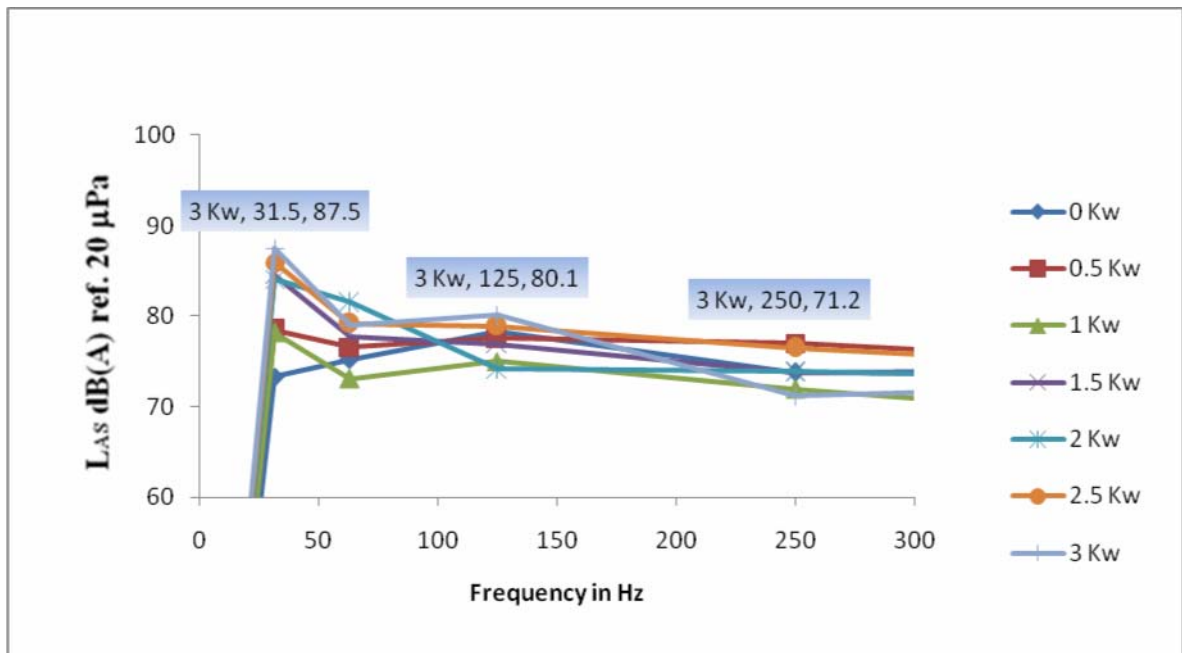
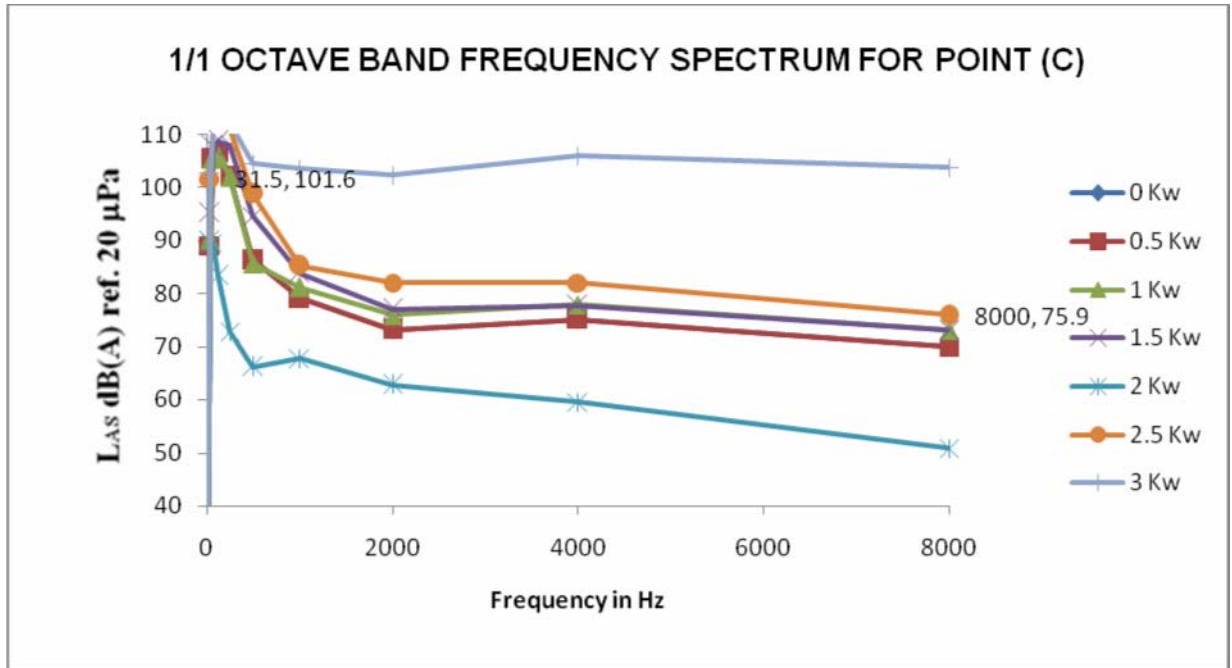


FIG 7.24 ENLARGED GRAPH

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT



NORMAL GRAPH

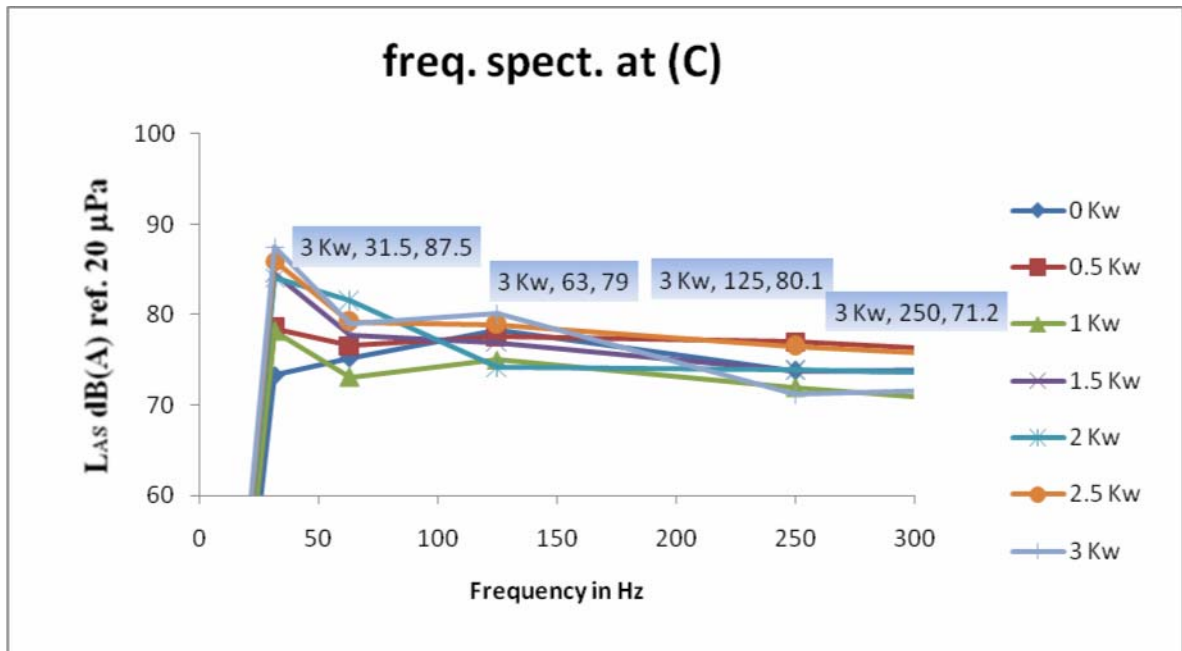


FIG 7.25 ENLARGED GRAPH

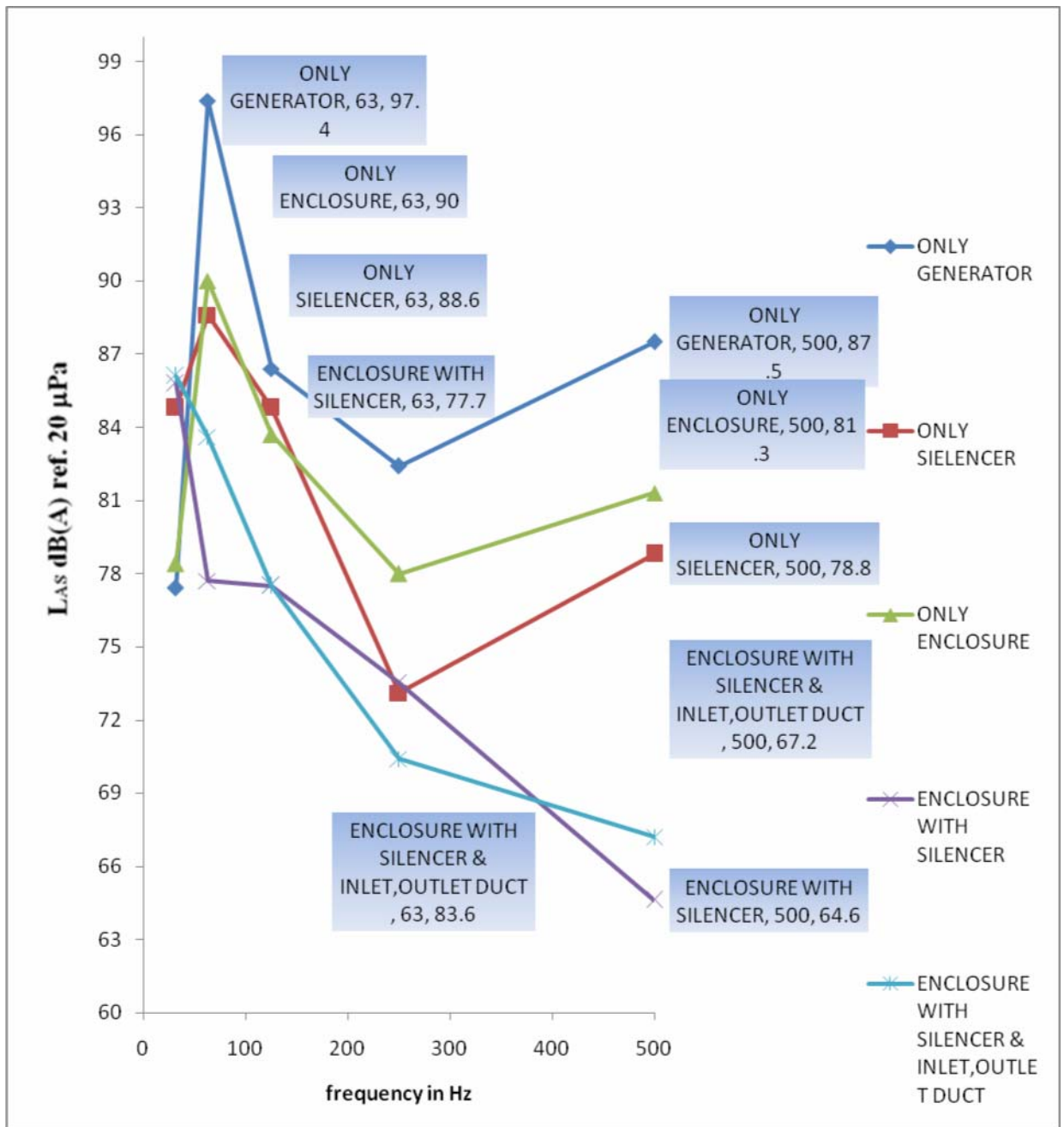


FIG 7.26 ENLARGED GRAPH

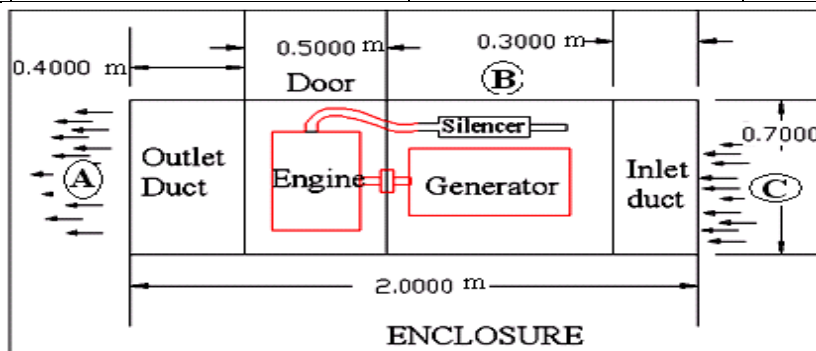
RESULTS FROM POINT (C):

S.NO.	TYPE	MAX. L_{AS} in dB(A)	PEAK FREQUENCY IN Hz	OBSERVED AT LOAD IN kW
1	ONLY GENERATOR	102.2	63	2
2	ONLY SILENCER	98.5	63	2
3	ONLY ENCLOSURE	97.3	63	1
4	ENCLOSURE WITH SILENCER	87.5	31.5	3
5	ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT	87.5	31.5	3

TABLE 7.4

1. Peak sound pressure levels are coming on lower frequencies at 63Hz, 31.5Hz.
2. Maximum sound pressure level achieving 3kW, 2kW, 1kw.
3. Spectrum gradually decreasing when moving towards higher frequencies.
4. It observed from the fig 7.26 complete enclosure with sound absorptive material can effectively reduce the peak sound pressure levels.

Linear values of Sound Pressure Level at Location C at Different Operating Conditions at Load 3 kW			
Sr. No	Operating Conditions	Sound Pressure Levels dB(A)	Effectiveness of Individuals dB(A)
1	Only Generator	94.8	
2	Only Silencer	88.4	6.4
3	Only Enclosure	90.4	4.4
4	Enclosure with Silencer	83.2	11.2
5	Enclosure with Silencer & Inlet Duct	79.1	15.7



FIG

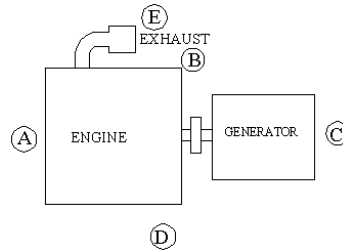
7.27LOCATIN OF A,B,C

It has been observed that

1. With the use of only silencer, Sound Pressure Level at location A reduces by 6.4 dB(A).
2. With the enclosure the Sound Pressure Level reduces by 4.4 dB(A). It has been analyzed that the use of partial barriers and poly urethane form worked effectively.
3. The Sound Pressure Level more reduces with the addition of Silencer.
4. It has been very clearly observed that Sound Pressure Level reduced by 4.5 dB(A) only with the use of inlet duct.

COMPARISON BETWEEN CONTOUR GRAPHS AT DIFFERENT HEIGHTS TO ANALYSES CHANGE IN FLOW OF NOISE AND REDUCTION IN SOUND PRESSURE LEVEL AT DIFFERENT POSITIONS WITH ONLY GENERATOR & WITH ENCLOSURE

ONLY GENERATOR SET Z-1 (30 cm)



DIRECTION OF THE GENERATOR

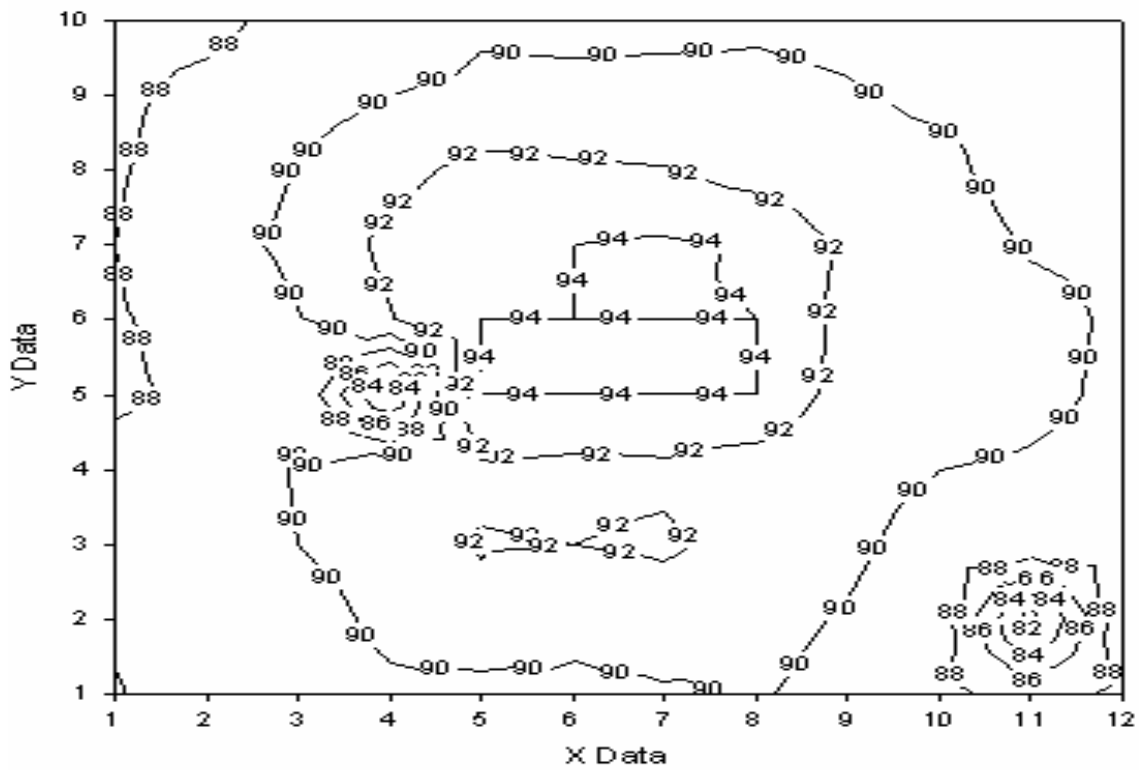


FIG 7.28

COMPLETE ENCLOSURE Z-1(30 cm)

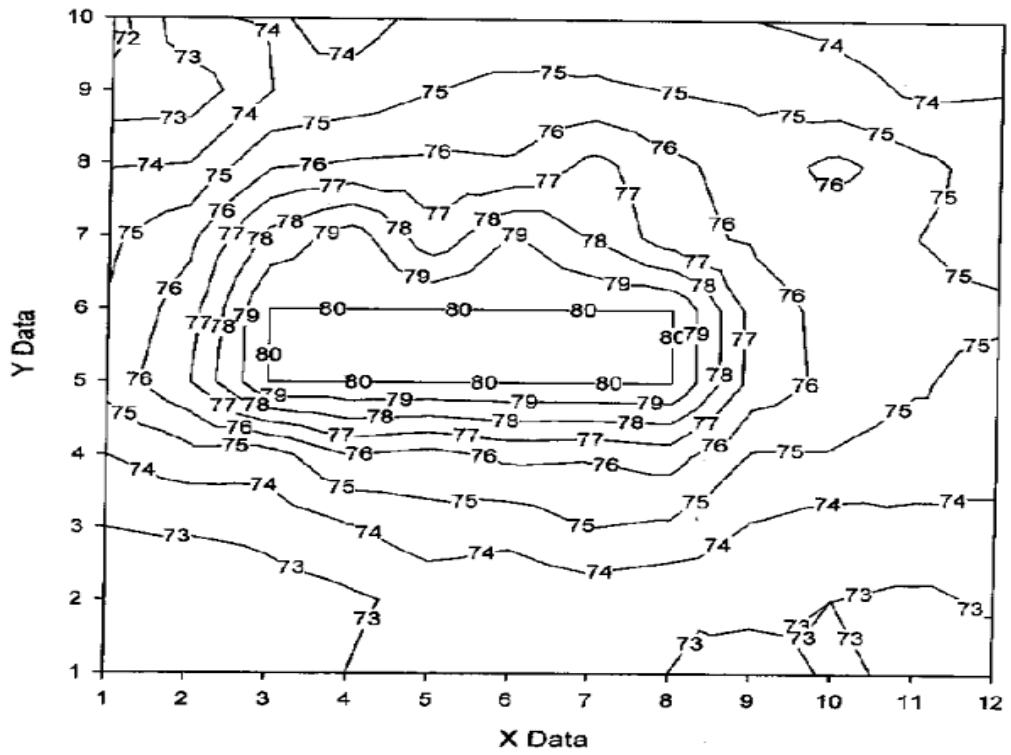
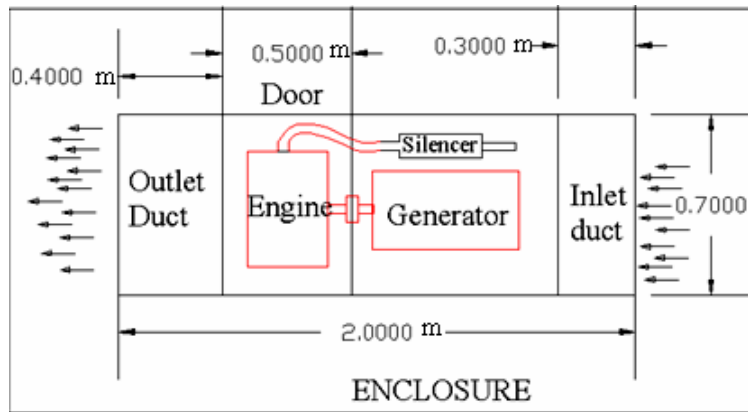
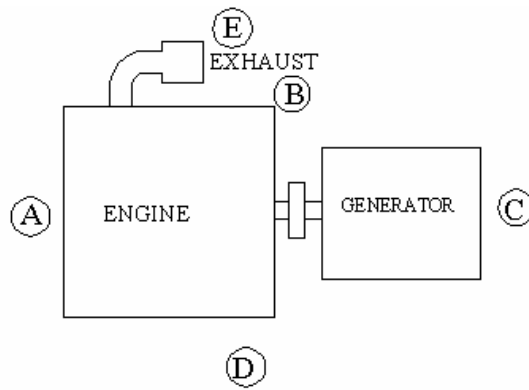


FIG 7.29

ONLY GENERATOR SET Z-2 (60 cm)



DIRECTION OF THE GENERATOR

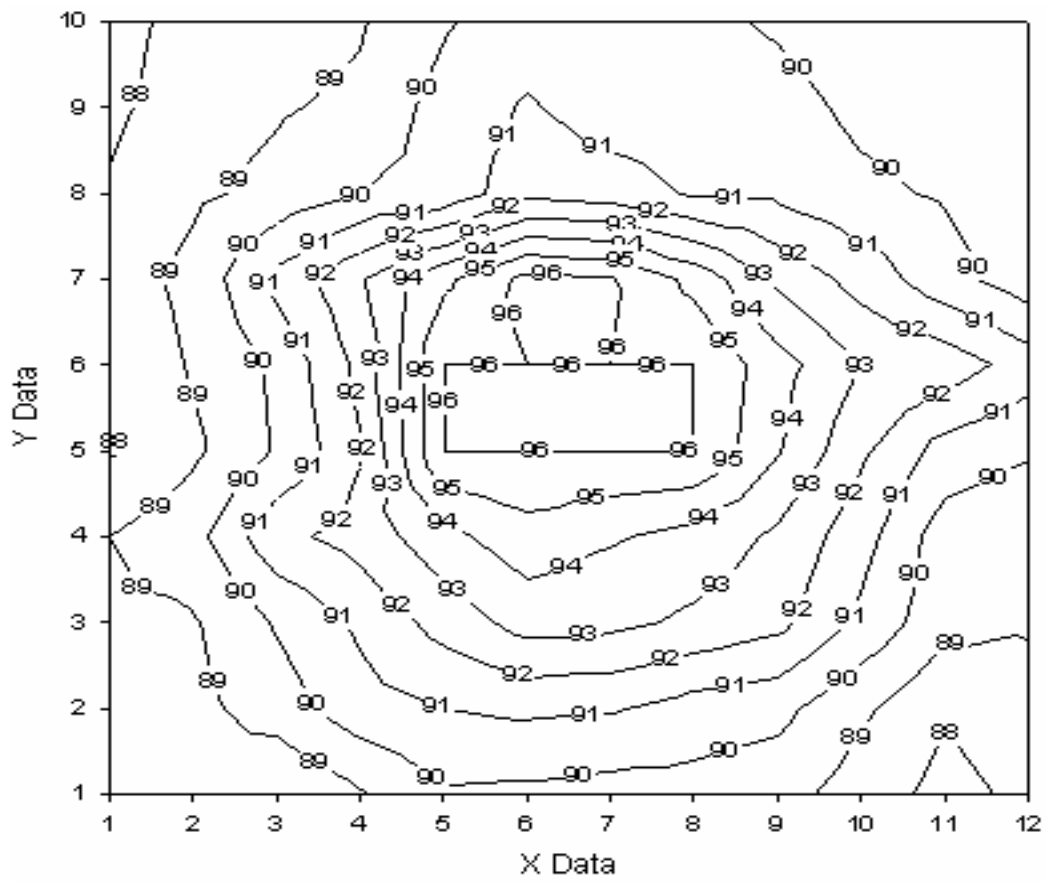
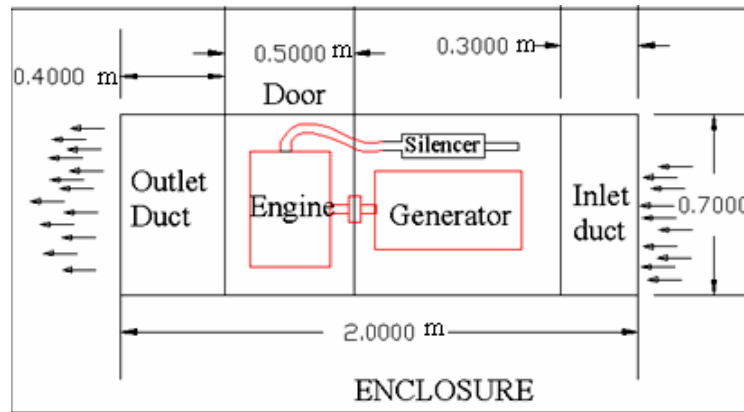


FIG .7.30 CONTOUR GRAPH OF Z-2 PLANE ONLY GENERATOR

ENCLOSURE + SILENCER + INLET & OUT LET DUCT Z-2(60 cm)



DIRECTION OF THE GENERATOR

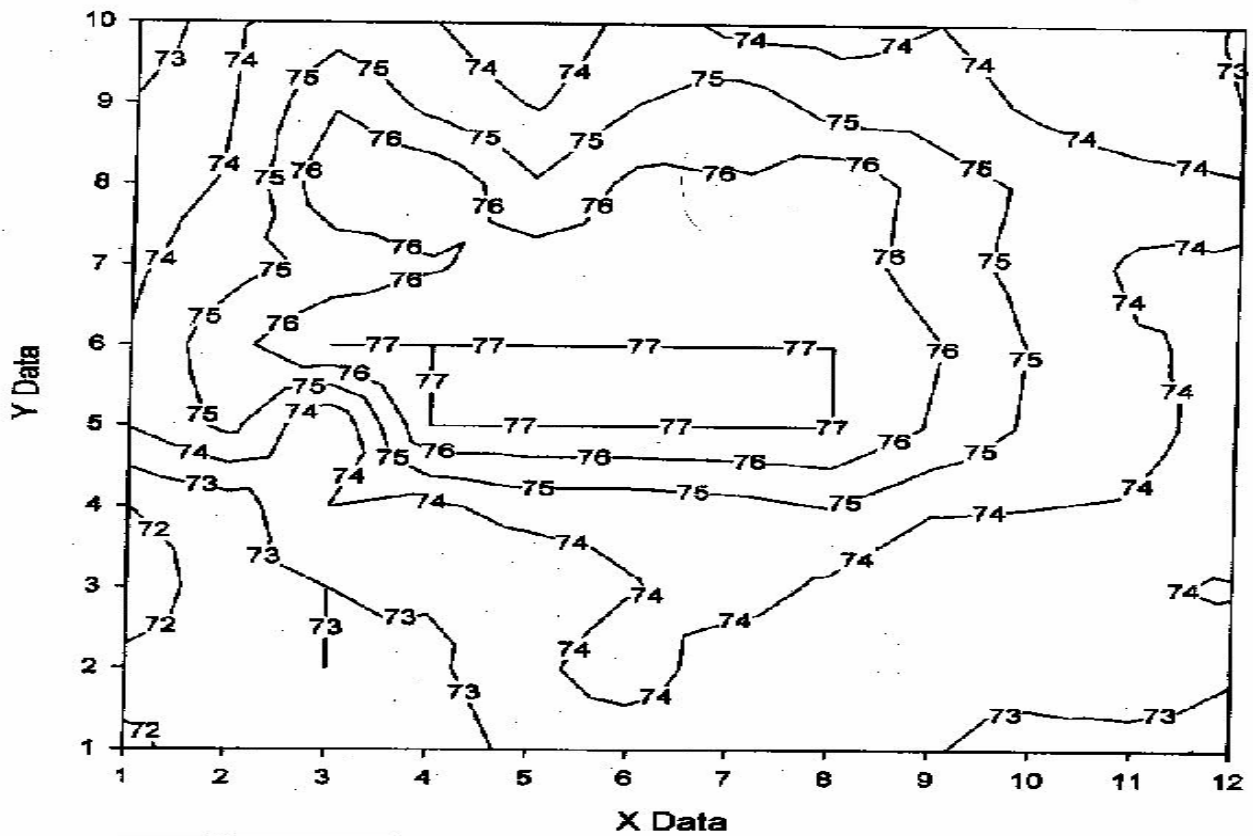
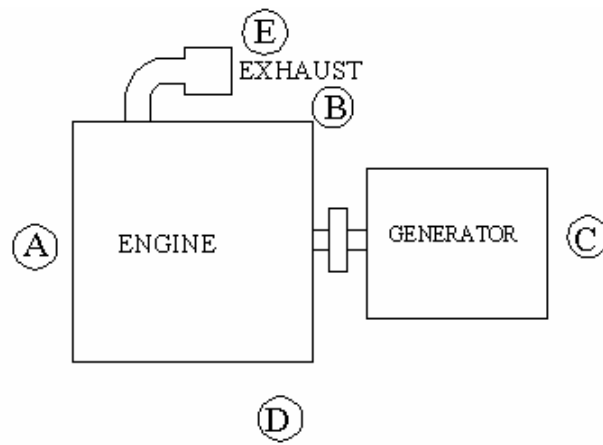


FIG .7.31 CONTOUR GRAPH OF Z-2 PLANE WITH ENCLOSURE

ONLY GENERATOR SET Z-3 (90 cm)



DIRECTION OF THE GENERATOR

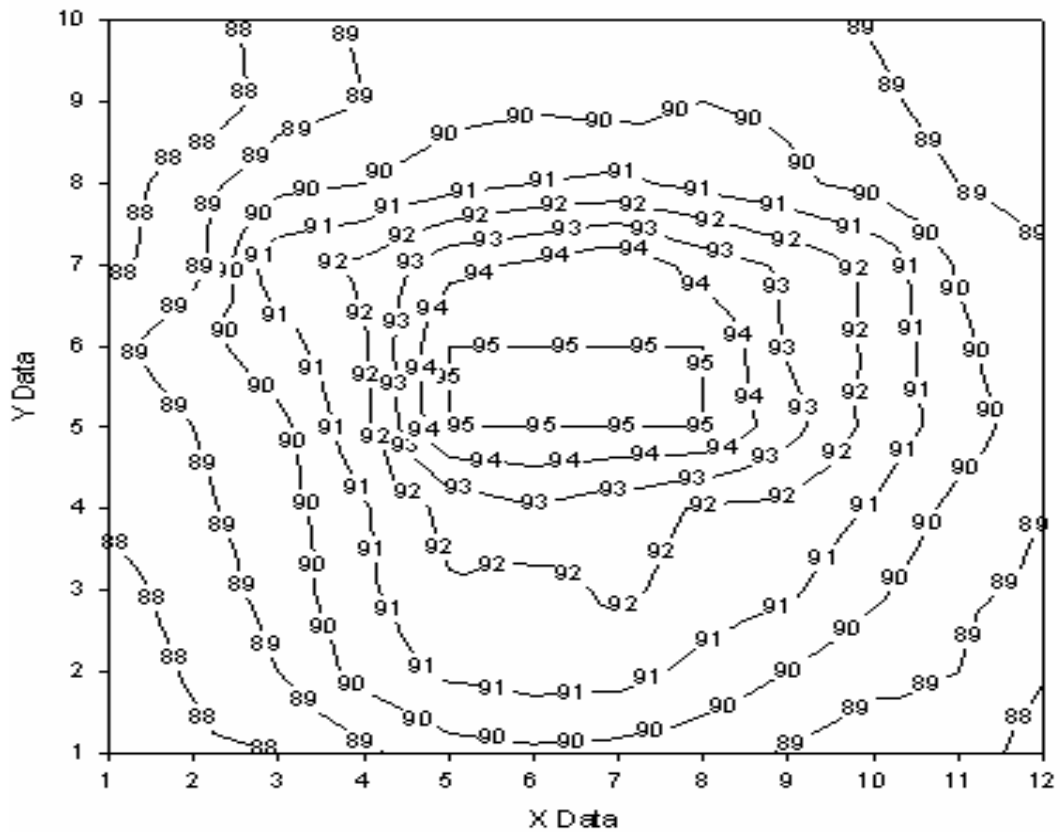
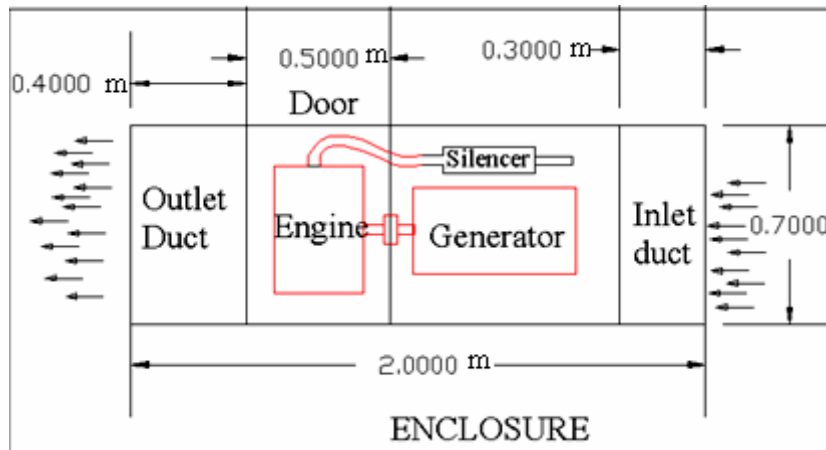


FIG .7.32 CONTOUR GRAPH OF Z-3 PLANE WITH ONLY GENERATOR

ENCLOSURE + SILENCER + INLET & OUTLET DUCT Z-3 (90 cm)



DIRECTION OF THE GENERATOR

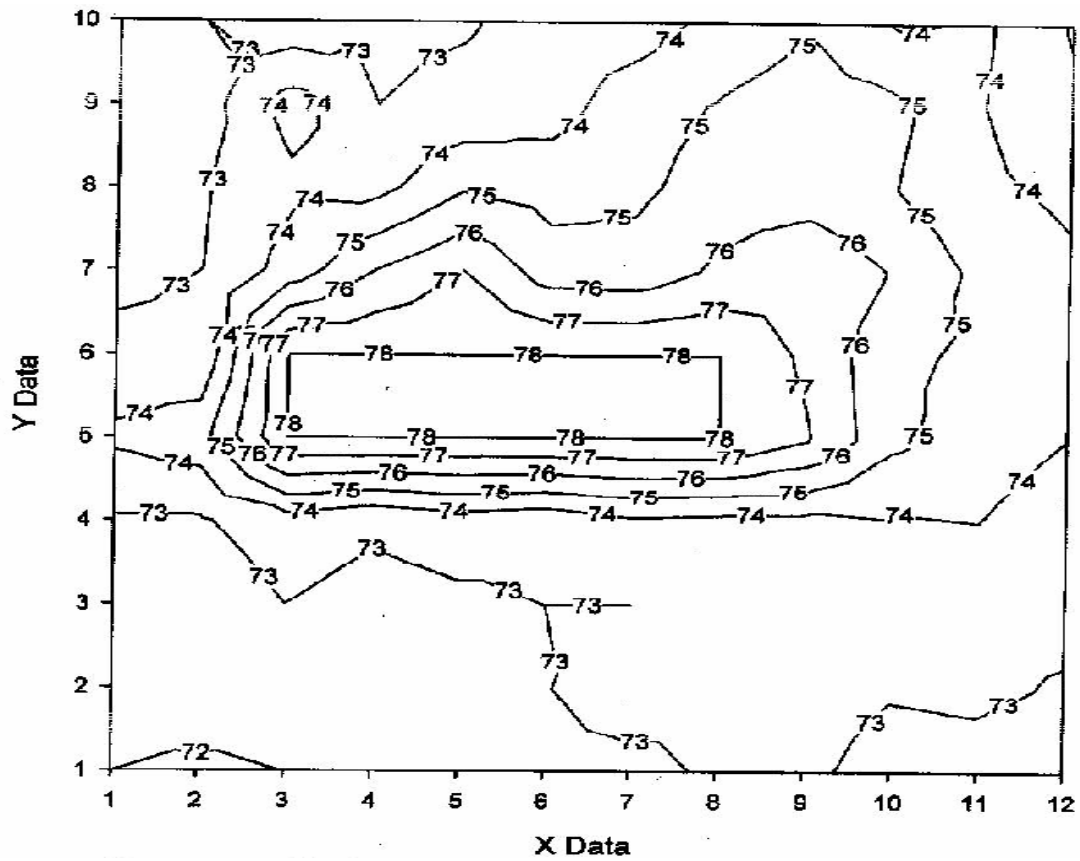
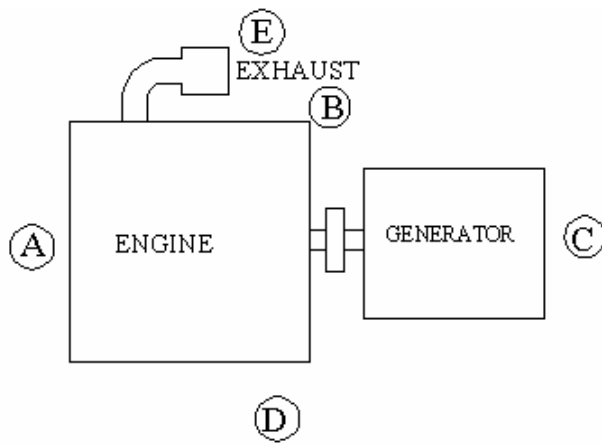


FIG .7.33 CONTOUR GRAPH OF Z-3 PLANE WITH ENCLOSURE

ONLY GENERATOR Z-4



DIRECTION OF THE GENERATOR

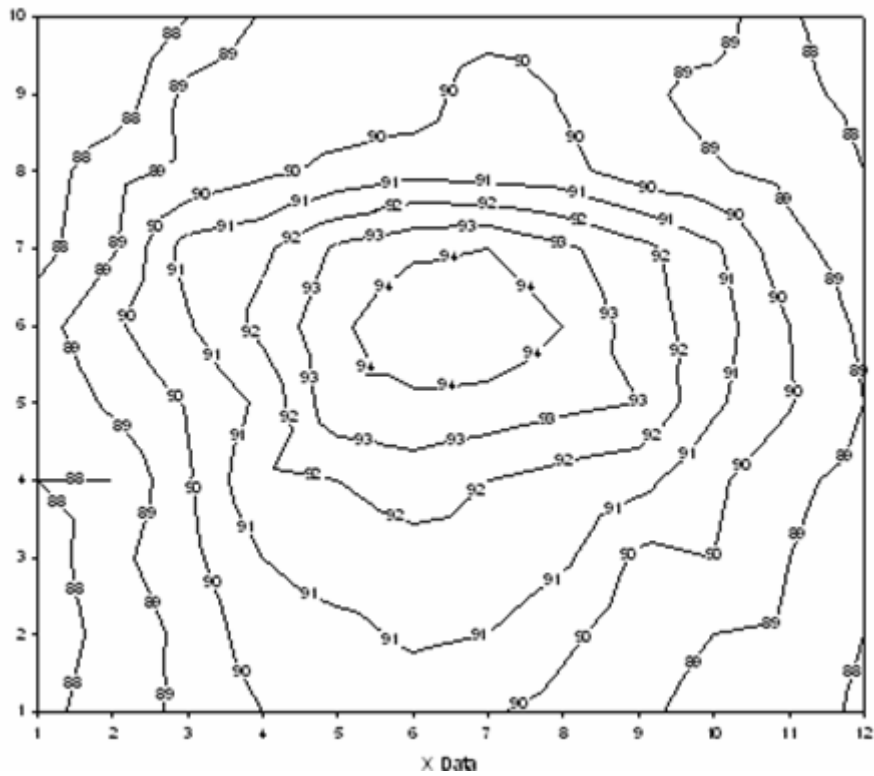


FIG .7.34 CONTOUR GRAPH OF Z-4 PLANE WITH ONLY ENCLOSURE

ENCLOSURE + SILENCER + INLET & OUTLET DUCT Z-4(120 Cm)

DIRECTION OF THE GENERATOR

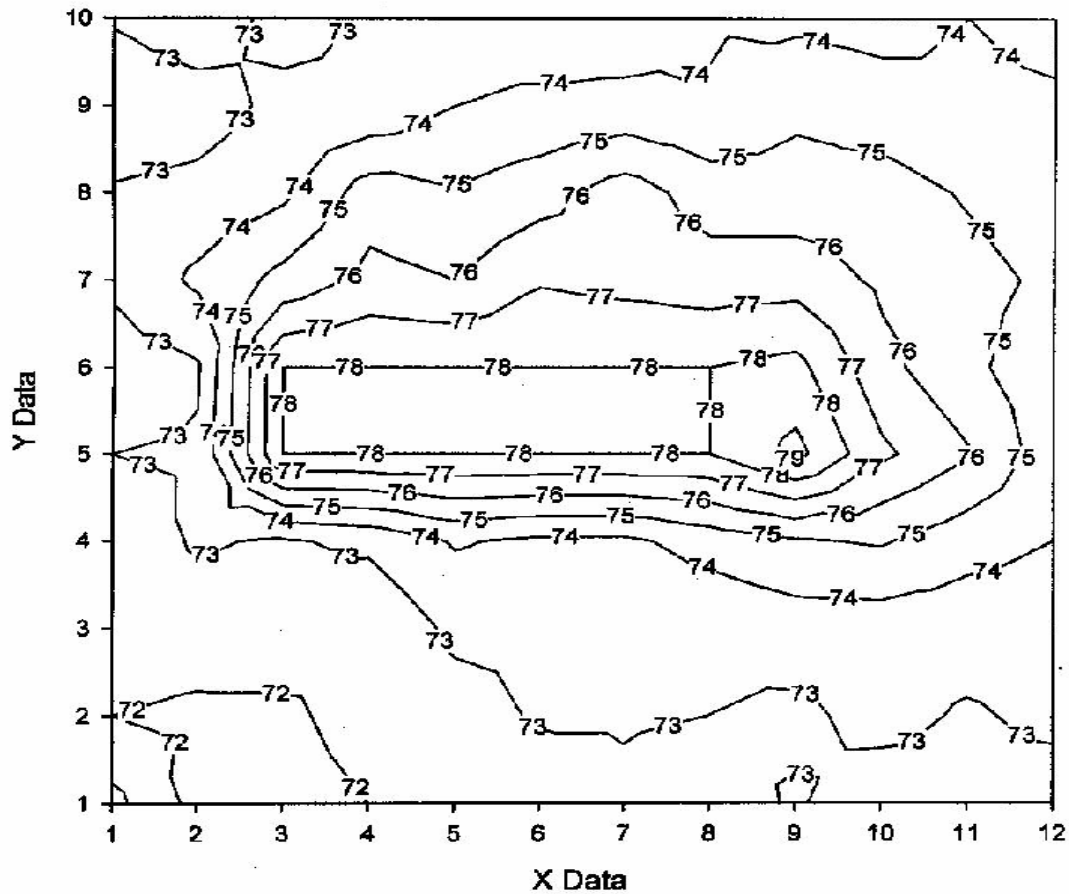
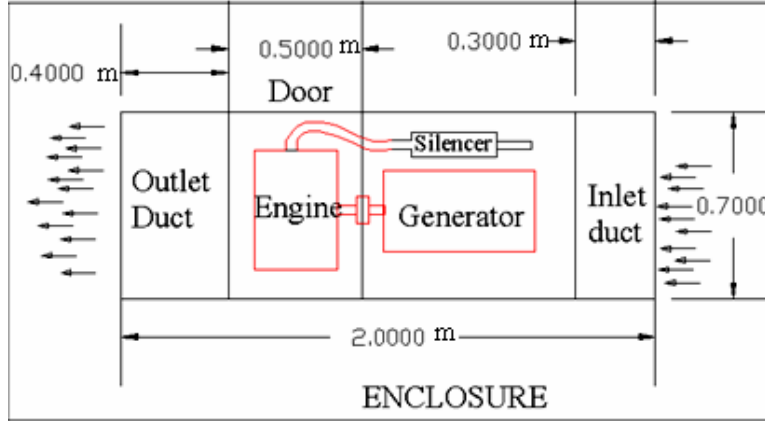


FIG .7.35 CONTOUR GRAPH OF Z-4 PLANE WITH ENCLOSURE

ONLY GENERATOR SET Z-5 (150 cm)

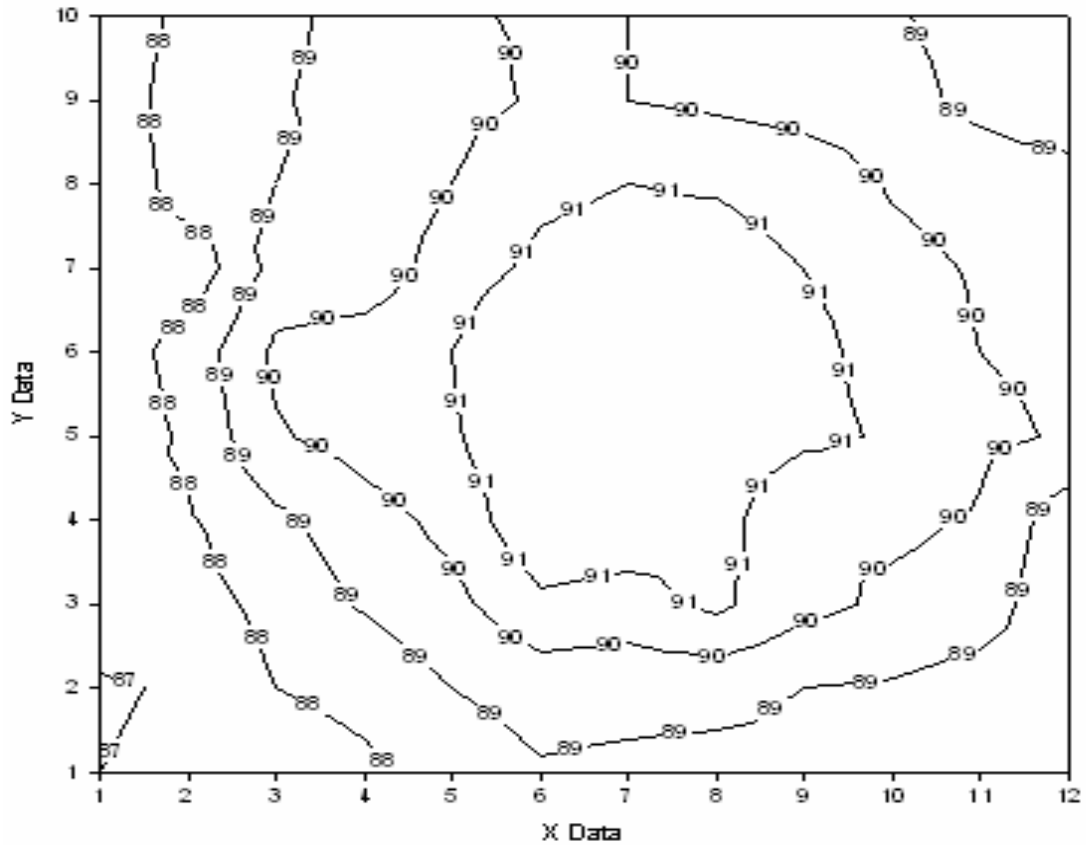
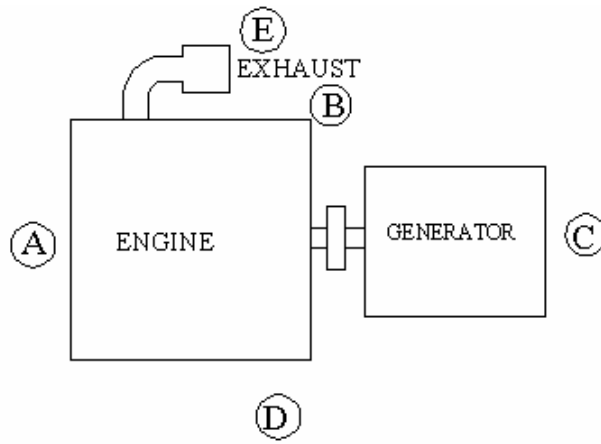
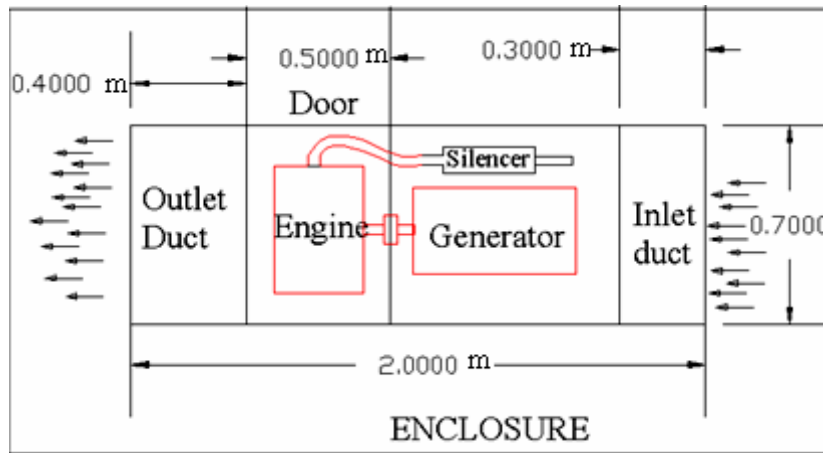


FIG .7.36 CONTOUR GRAPH OF Z-5 PLANE WITH ONLY GENERATOR

ENCLOSURE + SILENCER + INLET & OUT LET DUCT Z-5(150 cm)



DIRECTION OF THE GENERATOR

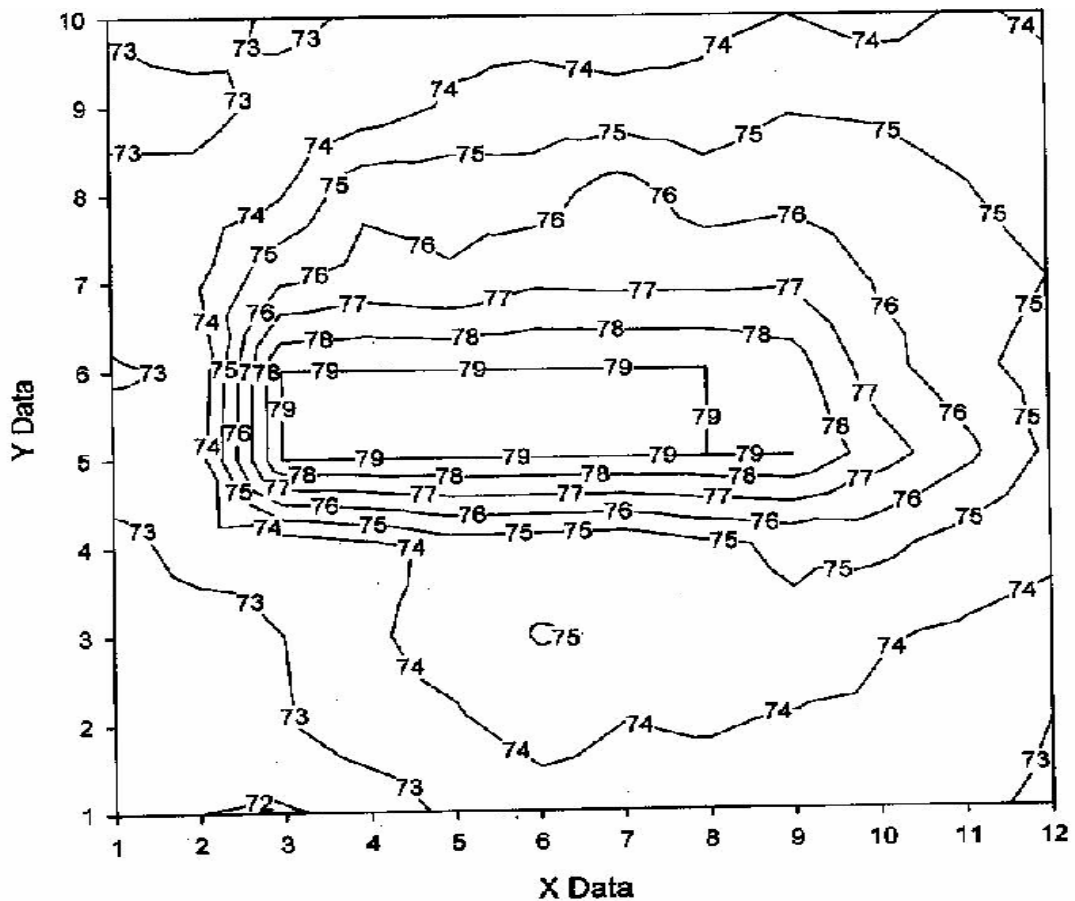


FIG .7.37 CONTOUR GRAPH OF Z-5 PLANE WITH ENCLOSURE

ONLY GENERATOR SET Z-6 (180 cm)

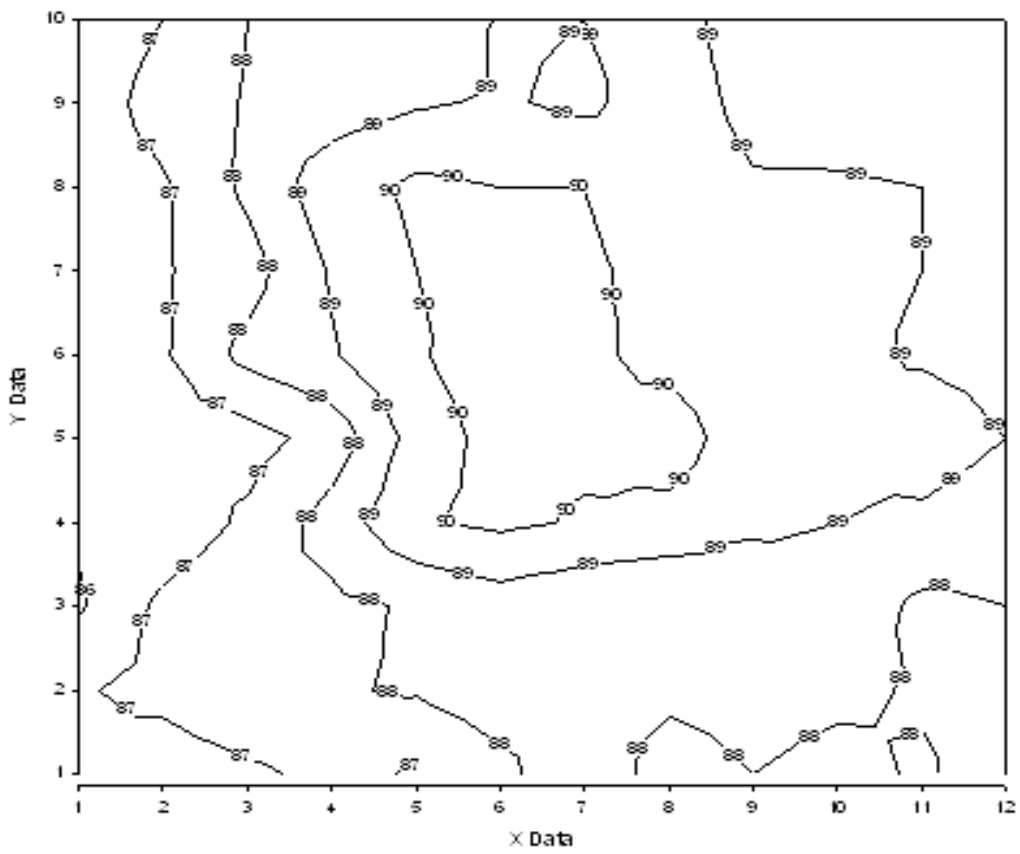
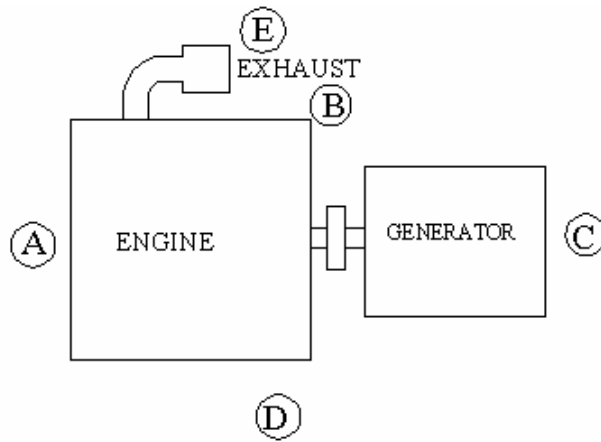


FIG .7.38 CONTOUR GRAPH OF Z-6 PLANE WITHOUT ENCLOSURE

ENCLOSURE + SILENCER + INLET & OUTLET DUCT Z-6 (180 cm)

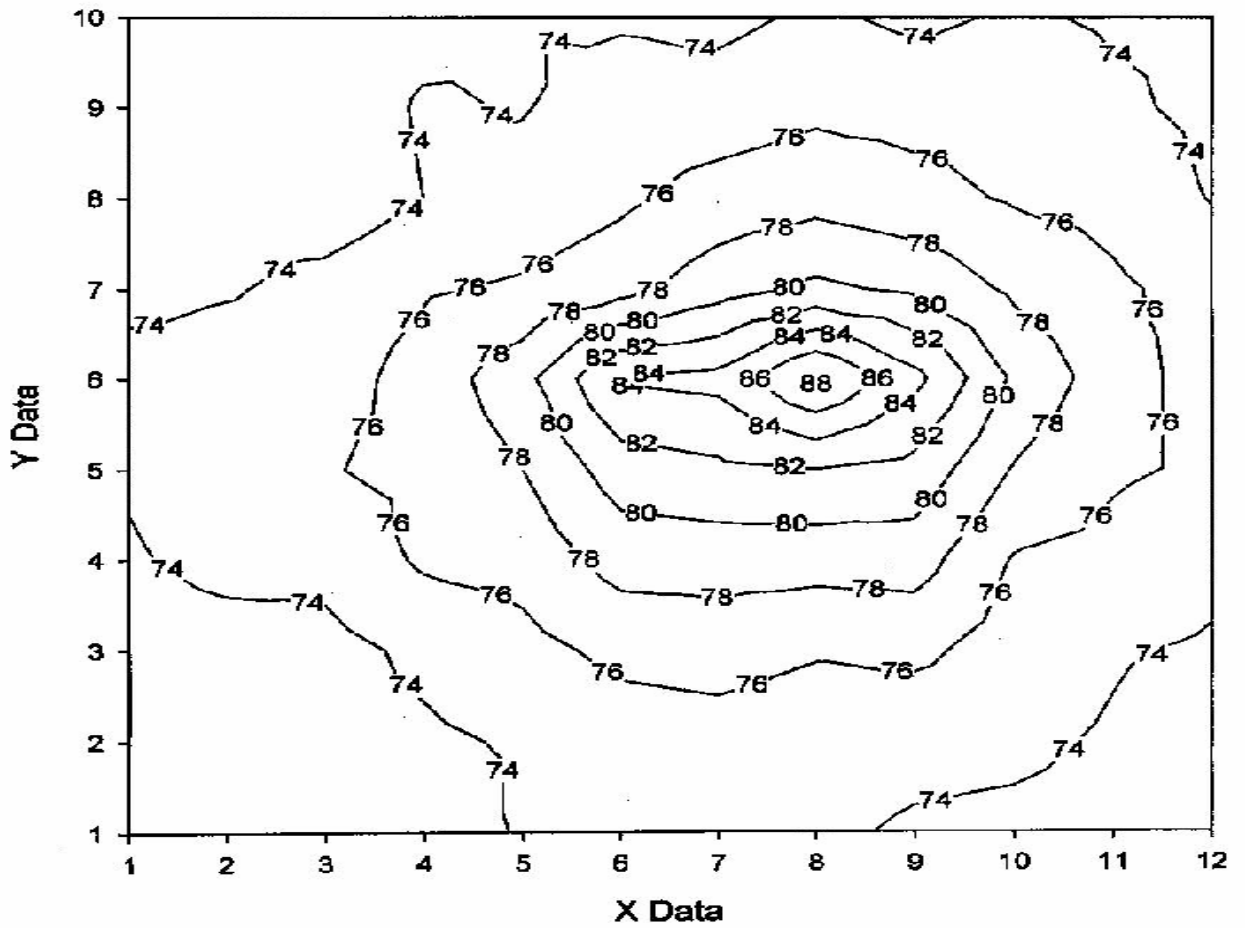
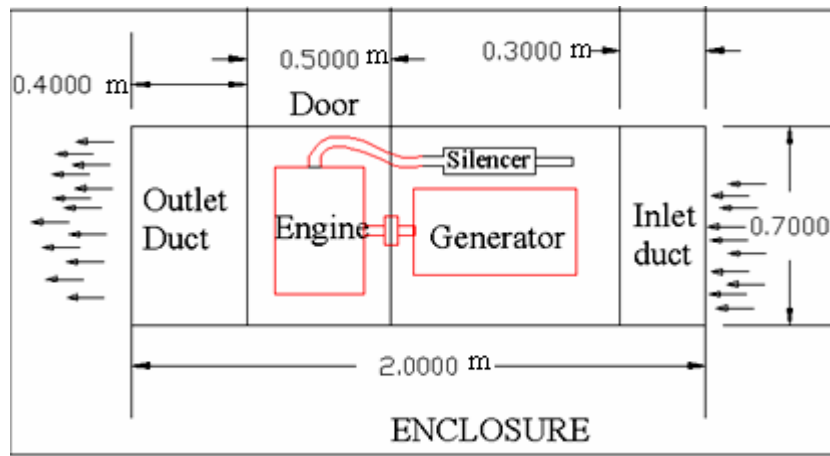


FIG .7.39 CONTOUR GRAPH OF Z-6 PLANE WITH ENCLOSURE

DISCUSSION:

For the improvement of the functional design of enclosure, there is need to find out those areas where still noise level is high than expectation. Few changes are carried out in the position's of engine parts .

For example, position of the exhaust now changed from one position to another, for that additional pipe is fitted inside the enclosure .to check the effect of all these changes on there is need to analysis the contour diagrams. Procedure of creating contour diagrams explained in chapter -6. As per that procedure contour diagrams are created. Results from them explained bellow.

1. Z-1 PLANE (30 Cm):

In this plane as shown in fig7.28 maximum sound pressure level is around the exhaust outlet.

When enclosure placed as shown in fig 7.29 sound pressure level reduces. But still maximum sound pressure level is around the door.

2. Z-2 PLANE (30 Cm):

In this plane maximum sound pressure level changes from exhaust of the generator as shown in fig 7.30. To the outlet of the hot air ventilation illustrated in fig 7.31.

3. Z-3 PLANE (90 Cm):

In this plane (fig 7.32, 7.33) new area of maximum sound pressure level is around the position of new silencer.

4. Z-4, Z-5, PLANES:

In these two planes maximum area is shifted to the air inlet opening of the enclosure.

5. Z-6 plane(Cm):

Contour diagram of this plane oriented towards exhaust pipe coming out from the enclosure. It gives assign that there is leakage of sound from packing of exhaust pipe or there is still need to improve the functional design of the enclosure

CHAPTER – 8

CONCLUSION AND SCOPE FOR FUTURE

8.1 CONCLUSION: following points are concluded from this thesis work

1. Acoustic enclosure fabricated with consideration as functional design can reduce Acoustic power of a 5kW generator up to 16 dB (A).
2. Effective silencer individually can reduce acoustic power level more than 10dB (A).
3. For noise reduction, Partial barriers are most economical and effective.
4. Peak values of sound pressure level are on lower frequencies
5. Complete enclosure can reduce the peak sound pressure levels at lower frequencies.
6. Impact of noise is more in which area can be identifying by use of contour flow diagrams.
7. Contour diagrams can help to improve the design.

8.2 SCOPE FOR FUTURE WORK

Lot of future scope is there in this experiment. as discussed above government standards can be achieve with this . But there is need to make it more economical. Following work can be further carried out in future.

1. Engine of the generator set is major source of noise as compare to the generator .So fabricate or modify acoustic enclosure only for engine and complete experiment as discussed in this thesis will be carried out. Collect the data & compare. If there is very small change in results. Then can achieve an economical solution also. Because its manufacturing cost will be the half of this enclosure (here considering to both engine and generator).
2. From the results it is clear that maximum sound pressure level is on 31.5, 63,126 Hz frequencies, so there is need to find out the particular parts of the engine which are producing these frequencies.
3. From contour graph Z-6(180 Cm.)It clears that still there is more noise at the position from where exhaust pipe coming out from the enclosure. As same few other area are making where sound level is more. So there is need to find out the alternate for that area.
4. Only functional design fabricated. Proper design of barriers and silencer need to be under taken.

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APPENDIXES

APPENDIX-A

ONLY GENERATOR

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	0				500				1000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	88.5	88.5	88.8	88.6	90.8	91.2	91	91	92.3	91.7	92.1	92
2	91.8	91.6	91.4	91.6	91.8	91.9	91.7	91.8	92.5	92.4	92.3	92.4
3	89.4	89.8	89.6	89.6	90.5	90.3	90.4	90.4	91.8	91.3	91.7	91.6
4	90.5	90.1	90.3	90.3	90.1	89.9	90.2	90.1	90.9	90.7	91	90.9
5	89.3	89.6	89.6	89.5	89.5	89.2	89.5	89.4	89.8	89.8	89.2	89.6
6	89.6	89.7	89.2	89.5	90.3	90.1	90.8	90.4	90.3	90.1	90.2	90.2
7	89.5	89.4	89.3	89.4	89.8	89.7	89.6	89.7	90.5	90.2	90.5	90.4
8	90.8	90.2	90.2	90.4	91.4	91.5	91.6	91.5	91.4	91.7	91.7	91.6
9	92.2	91.9	92.6	92.4	93.5	93.9	94.2	93.9	93.4	94.1	94.4	94
10	91.6	91.3	91.3	91.4	91.6	91.4	91.5	91.5	92.4	92.4	92.4	92.4
11	91.7	92.5	92	92.1	92.6	92.3	92.6	92.5	93.3	93.5	93.7	93.5
12	90.5	90.5	90.8	90.6	90.4	90.8	91.5	90.9	91.7	91.9	92.2	91.9
13	91.4	91.7	91.4	91.5	91.7	91.9	92.3	92	92.4	92.3	92.2	92.3
14	89.3	89.5	89.4	89.4	90.3	89.9	89.5	89.9	90.1	90	90.1	90.1
15	90.8	90.7	90.6	90.7	91.4	91.1	91.1	91.3	91.9	91.4	91.5	91.6
16	90.1	90	90.2	90.1	90.7	90.6	90.2	90.5	90.2	90.5	90.8	90.5
17	92.3	92.6	92.3	92.4	91.6	92.4	92.6	92.2	93.2	93.2	93.2	93.2
Ls	90.6				91.1				91.7			
Lw	101.9				102.4				103			

Lp	73.9	74.1	75
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	85.2	85.4	86.3
-----------	-------------	-------------	-------------

$$L_W = L_P + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_P + 10\log_{10}(S/S_0) \text{ dB(A)}$$

10^{-12} W) A-1

$$L_W = L_P + 11.3 \text{ dB(A) (ref.}$$

ONLY GENERATOR

Observations for sound pressure level in dB(A) ref. 20 µPa

LOAD (W)	1500				2000				2500				
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.
1		92.1	92.9	92.5	92.5	93.4	93.2	93.3	93.3	94.1	93.9	94	94
2		93.5	93.5	93.8	93.6	94.5	94.7	94.9	94.7	95.4	95.3	95.2	95.3
3		92.1	91.3	92.6	91.9	92.4	92.5	92.9	92.6	93.8	93.7	93.9	93.8
4		91.5	91.9	91.1	91.5	92.5	91.6	92.1	92	92.9	92.3	92.9	92.7
5		89.8	89.9	89.4	89.7	90.4	90.4	90.7	90.5	91.2	91.3	91.1	91.2
6		91.1	91	91.1	91.1	91.6	91.9	92.3	91.9	91.3	91.4	91.8	91.5
7		91.4	91.7	91.1	91.4	90.8	91.3	90.8	90.9	91.4	91.3	91.5	91.4
8		92.3	92.5	92.4	92.4	93.3	92.7	93.1	93	93.3	93.2	93.1	93.2
9		94	94.3	94.5	94.3	95.1	95.4	94.8	95.1	95.9	95.6	95.9	95.8
10		93.3	93.3	93.9	93.5	93.7	93.9	94.3	94	94.4	95.5	95.1	95
11		94.7	94.9	94.8	94.8	95.2	95	94.9	95	95.8	95.7	95.9	95.8
12		92.3	92.9	92.6	92.6	92.9	92.7	92.8	92.8	94.1	94	94.1	94.1
13		93	93.3	92.6	92.9	92.8	92.8	93.3	92.9	93.9	93.9	93.6	93.8
14		91.6	91.6	91.9	91.7	91.3	91.1	91.2	91.2	91.1	91	91.1	91.1
15		91.5	91.8	91.5	91.6	92.5	92.2	92.5	92.4	92.4	92.6	92.5	92.5
16		91.2	91.2	91.2	91.2	91.9	91.6	92.3	91.9	92.1	92.3	92.2	92.2
17		93.7	93.4	93.4	93.5	93.6	94.3	93.8	93.9	94.3	94.4	94.5	94.4

Lp	92.4	92.9	93.4
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Observations for Acoustic Power in (dB(A) ref. 10⁻¹² W)

Lw	103.7	104.2	104.7
-----------	--------------	--------------	--------------

$$L_W = L_P + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_P + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_W = L_P + 11.3 \text{ dB(A) (ref.}$$

ONLY GENERATOR

Observations for sound pressure level in dB(A) ref. 20 µPa

LOAD (W)	3000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	94.6	94.8	94.8	94.7
2	96.1	96.1	96	96.1
3	94.3	94.6	94.9	94.6
4	93.1	92.8	93.1	93
5	91.6	91.7	91.8	91.7
6	92.1	92.1	92	92.1
7	92.3	92.2	92.1	92.2
8	94.2	94.1	94.3	94.2
9	96.5	97.4	97	96.9
10	96.3	95.8	96.1	96.1
11	97.8	97.7	97.6	97.7
12	94.6	94.2	94.7	94.5
13	93.6	93.6	94.6	93.9
14	91.8	91.7	91.9	91.8
15	92.8	93.4	92.6	92.9
16	92.3	92.4	92.2	92.3
17	95.1	95.4	95.7	95.4

Observations for Acoustic Power in (dB(A) ref. 10⁻¹² W)

Lp	94.2
Lw	105.5

$$L_W = L_P + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_P + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_W = L_P + 11.3 \text{ dB(A) (ref.}$$

A-1

ONLY SILENCER

Observations for sound pressure level in dB(A) ref. 20 µPa

LOAD (W)	0				500				1000			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	81.9	81.7	81.8	81.8	82.3	81.8	81.9	82	82.6	82.5	82.7	82.6
2	83.8	83.6	83.4	83.6	83.3	83.1	82.9	83.1	83.7	83.9	83.2	83.6
3	83.44	83.5	83.3	83.4	83.5	83.4	82.6	83.1	83.5	83.3	83.8	83.5
4	83.1	83.1	82.8	83	83.2	83.4	83.3	83.3	83.2	83.3	83.4	83.3
5	82.4	82.6	82.5	82.5	82.5	82.5	82.2	82.4	83.2	83.2	83.2	83.2
6	83.9	83.3	83.9	83.7	82.8	82.5	83.6	83	84.1	84	84.2	84.1
7	82.2	82.2	82.2	82.2	83.5	83.6	83.4	83.5	83.1	82.4	83.5	83
8	83.1	83.2	83.3	83.2	82.8	83.3	82.9	83	82.9	82.9	83.2	83
9	85.9	85.4	85.8	85.7	86.1	86.4	85.7	86.1	85.5	84.9	84.6	85
10	82.9	82.7	83.3	82.9	82.5	82.4	82.3	82.4	82.7	82.7	82.9	82.8
11	83.7	83.8	84.2	83.9	84.9	84.8	84.7	84.8	85.5	84.8	84.7	85
12	84.7	84.6	84.4	84.5	84.2	84	84.1	84.1	84.5	84.7	84.6	84.6
13	83.7	83.9	83.8	83.8	84.8	84.4	84.9	84.7	85.6	85.1	85.2	85.3
14	3.6	83.8	83.7	83.7	82.9	82.7	82.9	82.8	83.1	83.6	82.3	83
15	8	83.6	84.4	84	83.5	83.5	83.5	83.5	84.9	84.5	84.7	84.7
16	2.5	82.1	82.5	82.3	82.9	82.3	82.6	82.6	83.1	83	83.2	83.1
17	8.1	83.4	83.7	83.4	83.8	83.4	83.9	83.7	83.4	83.8	84.5	83.9

Ls	83.4	83.5	83.7
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Observations for Acoustic Power in (dB(A) ref. 10⁻¹² W)

Lw	94.7	94.8	95
----	-------------	-------------	-----------

$$L_W = L_P + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W})$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

A-2

ONLY SILENCER

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	1500				2000				2500			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	83.4	83.6	83.1	83.4	83.4	83.8	83.5	83.9	84.2	84.3	84.4	84.3
2	84.2	84.3	84.1	84.2	85.1	85.5	84.6	85.1	85.6	85.3	85.6	85.5
3	83.6	83.9	84.3	83.9	84.3	84.4	84.2	84.3	84.8	84.7	84.9	84.8
4	84.1	84.2	84.3	84.2	84.9	84.7	84.8	84.8	85.2	85.3	85.4	85.3
5	83.8	83.8	83.8	83.8	84.4	84.2	84.3	84.3	84.6	84.4	84.8	84.6
6	84.5	84.6	84.4	84.5	85.4	84.5	84.8	84.9	85.4	85.6	85.5	85.5
7	83.5	83.8	83.5	83.6	84.2	84.2	84.5	84.3	84.4	84.9	84.8	84.7
8	83.2	83.6	83.4	83.4	84.5	83.5	84	84	84.2	84.5	84.8	84.5
9	85.3	85.4	85.2	85.3	85.8	85.4	85.6	85.6	85.5	85.7	85.6	85.6
10	83.3	83.4	83.8	83.5	83.9	83.6	84.5	84	84.5	84.6	84.4	84.5
11	85.4	85.9	85.8	85.7	85.9	85.7	86.2	85.9	86.2	86.6	86.4	86.4
12	85.9	84.4	84.6	85	85.5	85.4	85.3	85.4	85.6	85.7	85.8	85.7
13	85.2	85.8	85.2	85.4	85.9	86.3	85.8	86	86.4	86.7	86.3	86.4
14	83.4	83.4	83.7	83.5	83.7	83.4	83.7	83.6	84.2	83.4	84.3	83.9
15	85.3	85.1	84.9	85.1	85.2	85.3	85.4	85.3	85.6	85.7	85.5	85.6
16	83.4	82.6	83.3	83.1	85.5	85.4	85.3	85.4	85.5	85.4	85.9	85.6
17	84.2	84.3	84.1	84.2	84.5	84.6	84.4	84.5	84.8	84.8	84.5	84.8

Lp	84.2	84.8	85.2
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	95.5	96.1	96.5
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W})$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

A-2

ONLY SILENCER

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	3000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	85.2	84.8	85.1	85
2	86.2	86.4	86	86.2
3	85.4	85.2	85.6	85.4
4	85.6	85.4	85.8	85.6
5	85.3	86.5	86.2	86
6	85.3	85.4	85.2	85.3
7	85.5	85.2	85.5	85.4
8	85.4	85.3	85.8	85.5
9	86.2	86.2	86.2	86.2
10	85.4	85.2	85.3	85.3
11	87.2	86.8	86.8	86.9
12	86.1	86.2	86.3	86.2
13	86.8	86.8	86.5	86.7
14	84.9	84.4	84.5	84.6
15	85.6	85.7	86.4	85.9
16	86.1	86.6	86.5	86.4
17	85.9	85.8	85.4	85.7

Lp	85.8
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	97.1
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

a = Length = 1.65 m

L_w = Acoustic Power

b = Width = 1.35 m

L_p = Average L_{AS}

c = Height = 1.90 m

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-2

ONLY ENCLOSURE

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	0				500				1000			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	87.4	87.1	86.5	87	87.6	87.7	87.8	87.7	88.4	88	87.6	88
2	87.8	87.7	87.9	87.8	86.3	85.9	85.7	86	87.3	87.2	86.8	87
3	85.5	85.3	85.4	85.4	85.9	85.8	85.7	85.8	86.5	86.4	86.3	86.4
4	87.6	88.2	88	87.9	89.7	89.8	89.9	89.8	90.4	90.3	90.5	90.4
5	85.8	85.6	85.7	85.7	85.7	85.6	85.5	85.6	86.3	86.2	86.4	86.3
6	86.6	86.8	86.7	86.7	86.8	86.8	87.4	87	87.8	87.7	87.6	87.7
7	86.4	86.3	86.2	86.3	86.8	86.9	86.7	86.8	88.8	88.9	88.7	88.8
8	88.8	88.7	89.3	89.1	89.5	90.2	90.2	89.9	90.7	90.6	90.5	90.6
9	99.5	99.7	99.9	99.7	100.8	101.1	101.3	101.1	102.6	102.5	102.7	102.6
10	84.5	84.6	84.7	84.6	86.5	86.4	86.3	86.4	85.8	86.2	86.3	86.1
11	85.1	84.8	85.2	85	85.9	85.8	85.7	85.8	86.5	86.7	86.6	86.6
12	84.5	84.4	84.3	84.4	84.8	84.7	84.6	84.7	84.8	84.7	84.9	84.8
13	85.3	85.5	85.4	85.4	86.8	86.7	86.9	86.8	86.6	85.5	85.9	86.1
14	85.3	85.2	85.1	85.2	85.6	85.5	85.4	85.5	86.1	86.2	86.3	86.2
15	85.3	85.4	85.2	85.3	86.6	86.5	86.7	86.6	87.2	86.7	86.9	86.9
16	85.5	86.4	86.2	86	86.3	86.1	86.2	86.2	87.6	87.7	87.8	87.7
17	86.4	86.3	86.2	86.3	86.9	86.8	86.7	86.8	87.2	87.4	86.8	87.1

Ls	87	87.6	88.2
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	98.3	98.9	99.5
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-3

ONLY ENCLOSURE

Observations for sound pressure level in dB(A) ref. 20 μPa

LOA D (W)	1500				2000				2500			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	88.9	88.7	89.4	89	89.6	89.8	89.7	89.7	92.1	91.7	92.2	92
2	87.8	87.9	87.7	87.8	88.7	88.9	88.7	88.9	90.2	90.4	90.3	90.3
3	86.7	86.8	86.6	86.7	87.5	87.4	87.3	87.4	81.1	88.3	88.2	88.2
4	90.4	90.5	90.6	90.5	91.6	91.7	91.8	91.7	92.6	92.5	92.7	92.6
5	86.7	86.6	86.5	86.6	87.7	87.6	87.8	87.7	88.4	88.5	88.3	88.4
6	88.1	88.2	88.3	88.2	88.3	88.5	88.4	88.4	90.3	89.5	90.4	90
7	88.7	88.6	88.8	88.7	89.4	89.3	89.2	89.3	90.8	90.7	90.6	90.7
8	90.5	91.2	90.8	90.9	81.6	81.8	81.7	81.7	92.4	92.5	92.3	92.4
9	103.1	102.8	103.3	103.1	103.3	103.5	103.5	103.4	104.4	103.6	104	104
10	87.4	86.9	86.9	87.1	87.8	87.6	87.7	87.7	88.1	87.8	88.2	88
11	87.3	87.2	87.4	87.3	88.4	87.5	87.8	87.9	88.7	88.6	88.8	88.7
12	85.3	85.2	85.1	85.2	86.3	86.4	86.2	86.3	87.2	87.4	87.3	87.3
13	87.9	88.2	87.9	87	87.5	87.3	87.4	87.4	87.7	87.6	87.8	87.7
14	86.4	86.5	86.3	86.4	87.8	87.8	88.1	87.9	87.7	88.5	87.7	88.1
15	87.8	87.6	87.7	87.7	88.7	88.6	88.8	88.7	89.2	89.4	89.3	89.3
16	87.8	87.7	87.9	87.8	88.6	88.9	88.6	88.7	89.5	89.4	89.6	89.5
17	87.6	87.4	87.5	87.5	87.4	88.4	88.6	88.1	89.3	86.6	89.4	89.1

Lp	88.7	88.9	90.4
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Observations for Acoustic Power in (dB(A) ref. 10⁻¹² W)

Lw	100	100.2	101.7
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-3

ONLY ENCLOSURE

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	3000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	92.3	92.4	92.2	92.3
2	91.6	91.4	91.5	91.5
3	88.7	88.6	88.5	88.6
4	92.7	92.8	92.9	92.8
5	88.8	88.7	88.9	88.8
6	89.8	90.4	90.1	90.1
7	90.5	91.5	90.8	90.9
8	92.4	92.3	92.5	92.4
9	104.1	104.2	104.3	104.2
10	88.3	88.1	88.1	88.2
11	88.7	89.4	88.9	89
12	87.7	88.4	88.1	88.1
13	88	88.2	87.8	88
14	88.2	88.3	88.1	88.2
15	89.3	89.5	89.4	89.4
16	89.6	89.7	89.8	89.7
17	89.4	89.5	89.3	89.4

Lp	90.7
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	102
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-3

ENCLOSURE WITH SILENCER

Observations for sound pressure level in dB(A) ref. 20 μ P

LOAD (W)	0				500				1000			
	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	72.9	72.4	72.8	72.7	73.9	73.9	74.2	74	74.2	74.5	74.5	74.4
2	72.8	73.4	72.8	73	74	74.1	74.5	74.2	74.4	74.3	74.2	74.3
3	72.2	72.9	72.4	72.5	72.5	72.6	72.9	72.8	73.2	73.3	73.1	73.2
4	78.5	78.1	78.3	78.3	79.1	79.9	79.5	79.5	79	79	79	79.8
5	71.9	71.9	72.5	72.1	73.5	72.6	72.9	73	73.1	73.4	73.7	73.4
6	73.9	73.5	73.7	73.7	73.7	73.4	73.7	73.6	73.6	74.2	74.4	74.1
7	73.6	73.7	73.5	73.6	73.7	74.5	73.9	74.1	74.8	74.2	74.5	74.5
8	78.3	78.9	78.9	78.7	80.4	79.5	80	79.9	80.6	81.5	80.9	81
9	78.9	78.2	78.9	79	79.1	79.3	79.2	79.2	80.2	80	80.4	80.2
10	72.7	72.9	72.5	72.7	72.6	73.5	72.9	73	74.8	74.1	74.6	74.5
11	72.3	72.2	72.7	72.4	73.5	72.8	73.1	73.1	73.5	73.1	73.6	73.3
12	71.8	72.6	71.9	72.1	72.2	71.9	71.5	71.9	72.4	72.2	72.3	72.3
13	80.8	80.9	81.3	81	81.5	81.2	81.8	81.5	81.9	81.5	81.4	81.6
14	73.4	73.2	73.3	73.3	73.8	74.2	74	74	74.2	74.1	74.9	74.4
15	73.6	73.2	73.9	73.5	73.4	73.8	73.9	73.7	83.9	74.3	73.8	74
16	72.8	73.3	73.1	73.1	73.7	73.2	73.6	73.5	74.2	74.7	74.3	74.4
17	76.9	76.5	77.4	76.9	77.1	77.6	77.5	77.4	78.1	78.3	78.2	78.2

Lp	74.6	75.2	75.7
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	85.9	86.5	87
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W})$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

A-4

ENCLOSURE WITH SILENCER

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	1500				2000				2500			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	74.5	75.6	74.7	75.1	75.9	75.8	75.7	75.9	77.2	77.3	77.7	77.4
2	75.6	75.2	75.7	75.5	76.3	76.2	76.4	76.3	77	77.2	76.8	77.1
3	74.2	74.6	74.2	74.3	74.4	74.6	74.2	74.4	75.8	75.7	75.9	75.8
4	80.2	80.3	80.1	80.2	80.6	80.9	80.6	80.7	80.2	81.1	81.2	80.9
5	74.4	74.5	74.9	74.6	74.8	74.7	74.6	74.7	75.4	75.7	75.4	75.5
6	74	74	74.9	74.9	75.2	75.5	75.2	75.3	76.5	76.6	76.4	76.5
7	75.1	75.7	75.4	75.4	75.7	75.8	75.9	75.8	76.2	77.2	77	76.9
8	80.7	81.4	80.9	81	81.5	81.4	81.3	81.4	82.8	82.7	82.6	82.7
9	80.9	80.2	80.4	80.5	81	81.2	81.1	81.1	83.2	82.8	83.9	83
10	74.9	75.7	74.7	75.1	75.2	75.4	75	75.2	76.6	76.9	76.6	76.7
11	75.8	73.3	73.7	73.6	73	74.1	74.3	73.9	75.8	75.5	75.5	75.6
12	72.1	72.5	73.4	72.9	73.8	73.7	73.9	73.8	74.5	74.8	75.3	74.9
13	81.9	81.7	81.5	81.7	81.1	81.6	81.5	81.4	82.9	82.8	82.7	82.8
14	74	74	74	74.8	74.1	74.8	75.2	74.9	76.8	76.7	76.6	76.7
15	74.8	74.4	74.9	74.7	75.1	75.2	75.3	75.2	75.5	76.3	76.1	75.9
16	74.8	74.8	75.2	74.9	76	76.2	75.8	76	77.4	77.5	77.3	77.4
17	78.2	78.7	78.3	78.4	78.9	78.7	78.5	78.7	80.9	81.2	81	81

Lp	76.3	76.8	78.1
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	87.6	88.1	89.4
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W})$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

A-4

ENCLOSURE WITH SILENCER

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	3000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	78.9	78.6	78.9	78.8
2	77.4	77.2	77.3	77.3
3	77.2	77.3	76.8	77
4	80.5	81.1	81.2	80.9
5	76.7	76.8	76.9	76.8
6	77.2	77.5	76.9	77.2
7	78.8	78.9	78.7	78.8
8	84.3	83.8	84.4	84
9	83.9	83.4	83.5	83.6
10	77.8	77.3	77.7	77.6
11	75.8	76.2	76	76.1
12	76.3	75.9	75.8	76.1
13	83.4	83.2	83.5	83.4
14	77.6	77.9	77.8	77.7
15	76.5	77.1	76.7	76.9
16	78.4	77.8	78	78.1
17	83.2	83.6	83.4	83.4

Lp	79.1
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	90.4
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

A-4

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	0				500				1000			
	Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3
1	73.4	73.5	73.6	73.5	72.5	73.1	72.8	73	73.8	74.4	74.4	74.2
2	73.1	73.3	73.2	73.2	73.3	73	73.8	73.3	74.5	74.8	74.5	74.6
3	72.5	73	73.2	72.9	72.8	73.2	73	73	74.6	74.5	74.7	73.6
4	73.9	74.3	74.1	74.1	74.5	74.4	74.6	74.6	75.3	75.3	75.6	75.4
5	72.8	73.1	72.4	73	72.8	73	73.3	73.1	73.9	74.3	74.3	74.2
6	73.8	73.3	73.6	73.6	73.6	74.2	74.2	74	74.6	74.7	74.8	74.7
7	73.1	73.5	73.3	73.4	73.5	73.5	73.8	73.6	74.8	75.4	75.4	75.2
8	74.5	74.8	74.3	74.8	75.7	75.9	75.5	75.7	76.8	77.4	77.4	77.2
9	81.9	82.4	82.8	82.6	82.8	82.5	82.2	82.5	82.8	82.8	83.3	82.9
10	73	72.4	72.4	72.7	72.6	72.9	72.9	72.8	74	74.3	74.3	74.2
11	72.6	73	73.2	72.9	72.5	72.9	72.7	72.7	73.1	73.3	73.2	73.2
12	72.3	72.3	72.6	72.6	71.3	71.4	70.8	71.2	71.8	72.2	72	72
13	73.2	73.7	73.6	73.5	73.4	73.1	73.4	73.3	73.8	74.3	73.8	73.9
14	73.5	73.3	73.6	73.5	74.5	74.4	74.6	74.5	74.9	74.6	74.6	74.7
15	73.2	73.1	72.8	73	73.5	73.6	73.4	73.5	75.3	74.7	74.8	74.9
16	73.4	73.4	73.4	73.4	74.3	73.7	73.8	73.9	74.7	74.9	74.8	74.8
17	73.6	73.3	73.8	73.7	73.6	74.3	74.1	74	75.1	75	74.9	75

Lp	73.9	74.1	75
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	85.2	85.4	86.3
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.}$$

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	1500				2000				2500			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	74.8	75.4	75.4	75.2	75.3	75.3	75.6	75.4	76.8	77	76.8	76.9
2	101.4	101.7	101.5	75.9	75.9	76.4	76.3	76.2	75.7	75.5	75.4	75.5
3	73.8	74.3	73.8	73.9	73.8	74.4	74.4	74.2	75.2	75.4	75.4	75.3
4	75.4	75.5	75.6	75.5	75.7	75.9	75.5	75.7	76.7	76.7	76.5	76.6
5	74.2	74.4	74.3	74.3	74.6	74.7	74.8	74.7	75.6	75.9	75.9	75.8
6	75	75.2	74.8	75	102	101.8	102	75.7	76.5	76.8	76.5	76.7
7	75.5	75.8	75.5	75.6	76.3	75.9	76.1	76.1	76.8	76.9	77.6	77.1
8	71.5	71.5	71.8	71.6	78.4	77.8	78.5	78.2	79.9	79.6	80	79.6
9	102.6	102.7	102.9	82	81.8	82.5	81.7	82	83.2	83.6	83.5	83.4
10	74.6	74.7	74.8	74.7	74.9	74.9	74.5	74.8	76.9	76.8	77.4	77
11	101.4	101.3	101.3	74.3	73.6	74.2	74.2	74	75	75.2	75.1	75.1
12	72.8	73	73.3	73.1	73.6	73	73	73.2	74.4	73.8	73.9	74
13	73.8	74	74.3	74.1	74.3	73.9	73.8	74	75.3	75.7	75.6	75.5
14	75.1	75	74.9	75	75.1	75.5	75.6	75.4	76.4	75.9	76	76.2
15	74.9	75.3	75.1	75.1	75.5	74.9	74.8	75.1	76	75.7	76	75.9
16	75.3	75.4	75.2	75.3	76	76.1	76	76	77.3	76.9	76.8	77.1
17	76.1	75.9	76.6	76.2	76.6	76.8	76.4	76.6	77.5	77.3	77.5	77.5

Lp	75.2	75.7	76.8
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Observations for Acoustic Power in (dB(A) ref. 10^{-12} W)

Lw	86.5	87	88.1
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$$L_W = L_P + 10 \log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_W = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_P = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-5

ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT

Observations for sound pressure level in dB(A) ref. 20 μ Pa

LOAD (W)	3000			
Grid Pts.	Obs. 1	Obs. 2	Obs. 3	Avg.
1	77.8	78	77.8	77.9
2	78.4	77.8	78.5	78.2
3	79.9	79.6	80	75.6
4	77.5	77.5	77.8	77.6
5	75.5	75.8	75.5	76.6
6	77.6	77	77	77.2
7	79.6	78.2	78.2	79
8	82.4	82.5	82.5	82.5
9	84.1	84.5	84.6	84.4
10	78.3	78.7	78.6	78.5
11	75.8	76	76	75.9
12	75.9	75.9	76	75.9
13	75.5	75.8	75.5	75.6
14	73.8	74	74.3	77.1
15	77.4	77	77	77.2
16	78.5	78.4	78.2	78.3
17	79.9	80.3	80	80.1

Lp	78.1
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Observations for Acoustic Power in (dB(A) ref. 10⁻¹² W)

Lw	89.4
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$$L_w = L_p + 10\log_{10}(S/S_0)$$

$$a = \text{Length} = 1.65 \text{ m}$$

$$L_w = \text{Acoustic Power}$$

$$b = \text{Width} = 1.35 \text{ m}$$

$$L_p = \text{Average } L_{AS}$$

$$c = \text{Height} = 1.90 \text{ m}$$

$$S = \text{Surface area} = ab + 2(bc + ca) = 13.63 \text{ m}^2$$

$$S_0 = \text{Reference Area} = 1\text{m}^2$$

$$\text{Acoustic Power} = L_p + 10\log_{10}(S/S_0) \text{ dB(A)}$$

$$10^{-12} \text{ W)}$$

$$L_w = L_p + 11.3 \text{ dB(A) (ref.)}$$

A-5

APPENDIX-B

ONLY GENERATOR							
1/1 octave band Frequency Spectrum for point (A)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	79.4	80	81.4	81.6	81.4	83	83.2
63	90.6	90.1	91.8	91.4	91.8	93	93.3
125	86.1	86.1	87.6	87.1	87.6	89.3	89.3
250	78.4	79	80.6	81.3	80.6	82.3	82.2
500	84.4	82.5	84.6	83.9	84.6	86.2	86.6
1000	83.2	81.2	81	82.6	81	83	84
2000	84.3	83.4	82.9	84.4	82.9	82.8	83.2
4000	84.5	83.2	84	85	84	84	85.4
8000	77.2	78.5	76.9	80.3	76.9	79	77.4

B-1.1

ONLY GENERATOR							
1/1 octave band Frequency Spectrum for point (B)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	80	81.6	83.4	85.2	89.8	92.2	93.2
63	89.3	91.4	93	94.9	97.2	99.2	102.2
125	85	87.1	88	88.4	89.8	92.8	92.5
250	79.9	81.3	78.9	79.2	80.1	83	84.2
500	82.1	83.9	85.2	88.2	90.8	91.5	91.2
1000	82.7	82.6	83.8	83.9	85.4	86	88.2

2000	85	84.4	83.3	89.6	88.8	89.4	91.3
4000	83.5	85	87.3	86.5	85.4	87.3	89.2
8000	79.6	80.3	79.5	86.3	84.1	86.1	84.2

B-1.2

ONLY GENERATOR							
1/1 octave band Frequency Spectrum for point (C)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	73.1	73.5	73.5	77.1	73.1	75.2	78
63	92.2	92.8	92.8	94.6	92.2	93.3	95.4
125	81.4	80.4	80.4	85.1	81.4	82.3	84.3
250	76.8	81.6	81.6	79.7	76.8	77.5	80.2
500	82.9	83.8	83.8	86.8	82.9	84.3	84.2
1000	82.5	82.4	82.4	87.8	82.5	83.8	82.5
2000	79.2	79.5	79.5	81.7	79.2	80.2	80.9
4000	77.5	77.2	77.2	81.3	77.5	79.1	79.5
8000	70.5	70.9	70.9	72	70.5	72.2	75.2

B-1.3

ONLY GENERATOR							
1/1 octave band Frequency Spectrum for point (D)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	79.4	80.7	82.7	84.3	75.7	76.5	77.4
63	90.6	92.2	94.2	95.8	95.2	97.4	97.4
125	86.1	87.9	89.6	89.4	84.1	85.3	86.4
250	78.4	79.6	80.7	78.8	80.4	81.4	82.4
500	84.4	83.2	85.1	87	85.2	86.9	87.5
1000	83.2	81.9	81.1	81.5	83.1	83.2	84.3
2000	84.3	82.3	83.3	86.5	81.7	82.8	83.2

4000	84.5	85.4	85	82.5	80.8	81.9	83.2
8000	77.2	77.5	79.6	79.4	71.7	72.3	73.2

B-1.4

ONLY GENERATOR							
1/1 octave band Frequency Spectrum for point (E)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	80	78.4	81.5	95.6	97.4	99.4	99.2
63	97.3	98.6	100.1	100.8	104.3	107.3	108.2
125	81.7	83.5	95.7	93.3	95.8	98.2	99.3
250	91	76.8	76.3	77.6	95.9	97.9	98.1
500	82.1	95.4	95.3	96.5	96.2	98.3	100.8
1000	83.6	95.5	95.9	83.1	93.8	93.8	94.5
2000	91	80.2	95.1	95.6	95.7	96.5	95.7
4000	79.3	81.9	91.2	95.1	96.1	98.1	99.1
8000	72.2	75.1	75.9	77.3	83.8	91.3	83.7

B-1.5

ONLY SILENCER							
1/1 octave band Frequency Spectrum for point (A)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	74.7	77.9	81.2	82	86.1	81.9	85.4
63	78.1	81.2	85.2	84	82.9	84.4	82.9
125	82.7	80.7	82.2	80.3	84.1	81.1	79.4
250	74.9	69.8	71.9	74.1	76.6	74.5	70.4
500	75.7	77.2	76.5	75.8	76.2	75.7	75.9
1000	80.6	82.8	82.5	80.7	80.1	82	80.5
2000	74	77.2	77	75.1	75.6	75.4	77.8
4000	73.8	75.8	73.7	73	73.9	72.7	72.9
8000	66	73.9	66.9	66.2	67.3	64.3	65

B-2.1

ONLY SILENCER							
1/1 octave band Frequency Spectrum for point (B)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	78.4	81.5	83.8	86.5	86.1	88.5	88.7
63	81.3	84.6	88.3	88.2	88.4	87.2	87.8
125	83.9	84.7	83.5	83.5	83.6	84.8	85.3
250	75.7	71.5	73	78.7	74.8	74.8	75.2
500	76.4	77.2	76.6	75.8	77.5	73.5	77.6
1000	80.2	80.1	81.4	76.8	81.8	84.2	83.2
2000	76.4	78.6	78.6	84.5	79.2	79.2	81.4
4000	76	76.1	76.5	79.2	78.5	77.2	76.9
8000	69	70.6	70	72.2	71.2	69.5	69.4

B-2.2

ONLY SILENCER							
1/1 octave band Frequency Spectrum for point (C)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	77.6	82.8	82.8	85.7	86.5	88.6	90.5
63	84.8	90.1	90.6	92	93	93.9	94.3
125	81.9	81.5	84.6	84.7	85.3	84.4	85.5
250	68.8	66.5	71.8	68.7	70.8	70	71.8
500	74.9	73.9	76.7	75.3	74.1	74.3	73.6
1000	78	76.9	79.3	79.4	79.3	79.4	78.8
2000	74.4	76.2	75.7	75.7	75.8	75.7	76.5
4000	73.2	71.4	73	72.8	74.1	73.4	73.1
8000	64.3	63.4	64.9	63.6	63.9	63.2	64.5

B-2.3

ONLY SILENCER							
1/1 octave band Frequency Spectrum for point (D)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	76	76.4	80.9	82.6	89.3	85.2	84.8
63	81.2	84	88.1	89	84.7	89.3	88.6
125	79.8	80.9	80.5	83	74	84.6	84.8
250	73	73.2	73.3	74.2	77.5	73.5	73.1
500	78	76.6	78.2	80.2	81.8	79.6	78.8
1000	79.9	79	79	82.5	79.3	81	81.8
2000	77.8	77.4	78.2	78.5	77.2	77.7	78.5
4000	78	76.2	75.2	74.5	75.2	76.7	74
8000	72.6	69.4	70.8	69.4	67.6	67.5	65.3

B-2.4

ONLY SILENCER

1/1 octave band Frequency Spectrum for point (E)

Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	85.4	88.7	93.5	90.2	92.4	85.4	93.2
63	82.5	85.6	83.4	99.2	81.6	81	102.3
125	78.1	78.1	92.5	96	99.5	80.4	104
250	80.4	81.6	81.3	88	87.8	72.1	99.5
500	76.9	75.7	76.8	74.3	101.4	76.9	87.8
1000	72.1	76.3	78.1	76.9	103.1	81.6	83.1
2000	74.3	75.7	76.3	79.5	77.5	75.7	77.5
4000	72.1	85.6	72.8	82.3	83.4	73.9	71.6
8000	76.9	66.7	66.6	64.5	70.3	64.3	69.9

B-2.5

ONLY ENCLOSURE							
1/1 octave band Frequency Spectrum for point (A)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	74.6	75.9	90.1	96.6	76.7	93.8	74.9
63	81.7	83.4	95.1	97.3	87.5	97.7	79.7
125	80.5	81.9	91.6	94.2	80	93.5	80.4
250	81.7	81.5	97.1	91.4	80.9	94.1	80.3
500	81.3	83.5	99.4	96.1	86.1	99.2	82.8
1000	72.4	74.9	93.1	86.8	78.6	88.2	77.1
2000	69.1	70.8	88.8	89.1	81.4	94.3	74.9
4000	66.6	69	81.9	82.6	73	80.3	68.4
8000	56.1	60.2	76.5	79.5	66	80.2	58.5

B-3.1

ONLY ENCLOSURE							
1/1 octave band Frequency Spectrum for point (B)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	76	75.9	78.9	86.6	84.9	80.1	80.3
63	79.5	83.4	87.1	86.5	87.1	86.5	87.9
125	81.5	81.9	83.6	81.3	78.7	77.2	81.3
250	85.1	81.5	81.2	78.9	81.1	80.4	82.1
500	83.3	83.5	81.5	81.5	82.3	80.1	82.2
1000	73.7	74.9	75	75.2	76.7	74.9	74.4
2000	68.1	70.8	73.4	76.7	75.3	74.8	72.9
4000	65.4	69	70.6	71.2	70.3	69	68.6
8000	53	60.2	61.5	61.6	59.4	58.7	58.9

B-3.2

ONLY ENCLOSURE							
1/1 octave band Frequency Spectrum for point (C)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	76.6	77.1	77.4	83.4	84	80.7	77.4
63	84.4	85.2	84	85.3	85.7	86.5	83.3
125	79.5	79.4	81.7	83.3	86.6	84.6	84.1
250	81.9	82.1	84.6	83.1	80.9	82.3	80.7
500	79.4	80.4	83.1	82.9	85.3	80.7	82.5
1000	74.1	75.4	75.5	78.2	78.3	76.8	75.3
2000	71.1	72.2	73.6	79.2	77.7	75.9	75.7
4000	71.1	70.9	67.7	71.7	71.2	70.4	68.6
8000	60.2	60	55.4	62.6	57	59.6	57.1

B-3.3

ONLY ENCLOSURE							
1/1 octave band Frequency Spectrum for point (D)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	68.8	70.1	76.2	83.7	82.8	82.3	78.4
63	82.1	84.6	88.5	91.3	89.7	91.3	90
125	78.5	78.6	83.2	81.1	78.1	82.2	83.7
250	82.8	79	81.1	82.4	82.4	82	78
500	84.2	80.5	80.7	83.4	83.5	81.6	81.3
1000	73.9	75.9	76	77.4	78.2	77.2	74.5
2000	67.1	68.7	76.3	81.9	79.4	78.6	76.4
4000	64.4	66.3	71.2	75	73.2	73	72.7
8000	50.9	53.7	59.5	64.9	64.2	62	60.8

B-3.4

ONLY ENCLOSURE

1/1 octave band Frequency Spectrum for point (E)

Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	83.8	90.3	98.5	78.2	73.7	72.2	70.5
63	89.9	95.4	99	88.5	88.2	89.5	87.9
125	89.1	88.6	96	82.8	83.4	80.2	76.8
250	95	85.6	92.2	82.2	81.5	79.7	82.4
500	97	77.1	97.5	81.5	82.9	82.1	82.4
1000	90.4	96	91.1	77.1	76	76.1	73.7
2000	84.2	87.9	96.2	78.4	77.3	69.7	73.7
4000	79.6	82.4	86.2	72.6	71.1	59.9	67.9
8000	71.3	79.7	86.5	64.8	62.2	60.6	57.5

B-3.5

ENCLOSURE WITH SILENCER							
1/1 octave band Frequency Spectrum for point (A)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	73.3	78.5	78.2	84.6	84.1	85.9	87.5
63	75.2	76.6	73.1	77.8	81.5	79.2	79
125	78.3	77.6	75	76.9	74.2	78.9	80.1
250	73.8	77	72	73.8	73.9	76.5	71.2
500	74.2	73.7	67	73.5	72.7	73.2	72.9
1000	64.4	65.2	65.5	67.1	66.6	71.2	67.8
2000	62.9	63.3	62.4	63.2	64.5	65.9	66.6
4000	61.3	63.5	59.3	63.6	63	64.1	63.4
8000	51.6	56.1	51	54	53.5	55.1	53.1

B-4.1

ENCLOSURE WITH SILENCER							
1/1 octave band Frequency Spectrum for point (B)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	76.2	80.6	79.4	86.6	88.1	88.3	88.1
63	75.6	79.1	73.6	79.1	82.6	81	78.9
125	74.6	73.2	75.7	76.9	76.1	76.6	74.2
250	73.7	74	73	74.7	75.7	77	72.5
500	70.3	70.9	67.3	68.9	70.4	70	69.8
1000	65.5	64.9	65.5	64.1	65.5	66.2	66.6
2000	61.9	63.2	61.9	63.5	63	64	64.5
4000	58.9	62.9	60.4	61.1	60.5	61.9	62.5
8000	47.6	50.7	51.8	49.4	49.5	49.9	50.9

B-4.2

ENCLOSURE WITH SILENCER							
1/1 octave band Frequency Spectrum for point (C)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	79	82.4	86.7	89.9	90.7	89.4	91.2
63	78.1	81	83.4	87.8	89.1	87	87.8
125	76.4	75.8	75.8	77.8	79	76.7	83.4
250	74	71.5	73.5	75.8	75.4	76	75.9
500	72.9	73.6	73.2	72.2	71.2	71.1	72.5
1000	65.6	66.9	68	68.9	69	67.8	71.1
2000	66.8	65.4	66.9	67	69	65.1	69.9
4000	69.3	66.8	67.6	68.3	68.9	63.9	70.1
8000	58.1	54.9	57.6	58.2	60.3	53.2	58.7

B-4.3

ENCLOSURE WITH SILENCER							
1/1 octave band Frequency Spectrum for point (D)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	69.2	73.3	74.9	78.1	80	82.1	85.8
63	77.9	78.6	82.2	82.7	86.4	84.3	77.7
125	72.8	73.3	74.7	75.4	77.5	80.8	77.5
250	73.3	73.5	73.1	69.1	72.1	71.6	73.5
500	65.9	64.9	65.2	63.9	64.4	65.4	64.6
1000	61.8	63.2	63.9	64	64	65.8	64.5
2000	58.4	59	59.8	60.4	61.1	61.9	63.6
4000	57	57.2	57.4	58.1	58.7	60	61.3
8000	45.1	45.1	46.2	47.1	47.9	49	50.4

B-4.4

ENCLOSURE WITH SILENCER

1/1 octave band Frequency Spectrum for point (E)

Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	87.5	78.5	89.6	94.7	95.6	95.8	99.2
63	98	76.6	98.2	100.8	103.9	104.7	106.7
125	95.3	77.6	94	95.9	99.9	101.9	106.3
250	86.7	77	85.9	87.3	93.2	96.6	104.5
500	74.4	73.7	75.8	76.1	78.1	80.5	92
1000	79	65.2	79.2	80	81.7	81.8	83.4
2000	71.9	63.3	73.2	74.7	73.6	74.9	79.2
4000	67.5	63.5	67.8	71.4	77.9	80.8	83
8000	66.1	56.1	68.3	70.7	71.7	75.3	78.3

B-4.5

ENCLOSURE WITH SILENCER & INLET , OUTLET DUCT							
1/1 octave band Frequency Spectrum for point (A)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	67.7	72	80.2	81.7	83.2	83.7	86.3
63	77.9	77.9	79.9	80.5	84.9	79.4	78
125	76.7	74.7	74.7	76.3	74.6	77.2	78.8
250	73.3	66.6	69.8	68.6	69.3	70.8	70
500	65.9	65.8	70.4	64.3	63.5	65.4	67.9
1000	60	61.2	65.5	62.5	64.4	63.3	63.9
2000	56.8	58.5	60.6	59.2	61.5	61	62.2
4000	51.8	54.9	57.9	55.5	58.8	58.2	59.6
8000	40.3	41.8	47.4	43.4	47.8	47.2	48.3

B-5.1

ENCLOSURE WITH SILENCER & INLET , OUTLET DUCT							
1/1 octave band Frequency Spectrum for point (B)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	71.7	77.9	81.2	84.3	99.5	88.9	89
63	77	77	80.5	82.2	110.1	83.1	79.4
125	74.7	73.8	75.7	77.6	110.8	79.3	78.9
250	68.5	70.5	71.4	70.3	111.8	74.4	73.8
500	63.8	71.8	69	68.5	101.1	68.8	69.4
1000	64.5	65.5	67	66.9	86.3	66.3	66.8
2000	59.8	63	62.5	62.5	82.2	63.6	64.4
4000	58.8	59.3	59.1	60.2	80.4	61.5	62.8
8000	47	51.9	49	50.1	75	50	51.4

B-5.2

ENCLOSURE WITH SILENCER & INLET , OUTLET DUCT							
1/1 octave band Frequency Spectrum for point (c)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	74.5	78.7	81.9	87.8	78.2	92.7	92
63	83.3	84.8	85.2	88	78.6	91.9	90.1
125	79.5	79.1	78.7	84	76.8	87.3	86.3
250	73.5	75.5	75.5	73.6	68	76.4	75.9
500	64.4	66.7	67.5	66.8	71.9	67.1	67.6
1000	62.8	63.2	65.1	66	66.1	67.1	69.7
2000	58.3	58.9	61.1	61.1	61.2	63.8	65.3
4000	55.7	55.7	57.2	58.3	61.4	60.7	62.4
8000	46	45.1	47.3	48.8	47.5	51.7	53.3

B- 5.3

ENCLOSURE WITH SILENCER & INLET , OUTLET DUCT							
1/1 octave band Frequency Spectrum for point (D)							
Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	69.4	73.4	77.2	80.5	87.5	84.5	86.1
63	82.2	81.1	83.6	84.3	82.3	85.9	83.6
125	75.7	77.3	73.8	76.5	78.9	79.7	77.5
250	68.8	69.7	70.3	72	72	69	70.4
500	61.8	64.9	64.2	66.4	70.5	67.1	67.2
1000	63.4	62.7	62.7	66.6	66.2	64.2	65.6
2000	58	59	60.4	59.1	63.6	63.2	63
4000	55.4	56.4	57.2	57.2	61.2	60.2	60.6
8000	43	44.2	46	46.1	52.2	49.4	50

B-5.4

ENCLOSURE WITH SILENCER & INLET , OUTLET DUCT

1/1 octave band Frequency Spectrum for point (E)

Load (W)	0	500	1000	1500	2000	2500	3000
Frequency(Hz)	0	0	0	0	0	0	0
31.5	89	89	90.1	95.2	89.8	101.6	102.6
63	105.3	105.3	105.3	107.7	89.5	110.6	110.8
125	106.6	106.6	105.9	108.7	83.3	111.1	111.6
250	102	102	102	107.8	72.8	111	112.3
500	86.3	86.3	85.6	94.6	66.2	98.8	104.5
1000	79.1	79.1	81.1	83.8	67.7	85.2	103.5
2000	73.2	73.2	75.9	77.1	62.8	82	102.3
4000	75.1	75.1	77.9	77.8	59.5	82	105.9
8000	69.9	69.9	73.1	73	50.8	75.9	103.8

B-5.5

APPENDIX-C

CAPARISON BETWEEN CONTOUR GRAPHS OF ONLY GENERATOR &
GENERATOR WITH ENCLOSURE WITH SILENCER & INLET, OUTLET DUCT
AT DIFFERENT HEIGHTS TO ANALYSIS THE DIFFERENCE IN SOUND
PRESSURE LEVEL AT DIFFERENT POSITION

ONLY GENERATOR SET Z-1 (30 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	87.9	88.2	88.5	88.7	87.6	87.8	88	87.8	87.6	87.1
2	88.8	88.9	89.1	89.3	88.5	88.8	89.3	89	88.6	87.4
3	88.6	88.8	90	90.1	89.8	89.9	90.4	90.1	89.2	88.7
4	89.5	90.7	91.2	91.3	81.6	91.9	92.5	91.4	90	89
5	89.3	91.6	92.1	91.7	94	94	93.5	92.6	90.3	89.8
6	89.6	90.5	92	91.5	94	94	94	92.3	90.3	89.7
7	89.8	91	92.3	91.6	94	94	94.3	92.1	90.5	89.6
8	90.1	90.4	91.3	90.9	94	94	93.8	91.2	90.7	89.6
9	89.6	89.9	90.3	90.5	91.1	91.4	91.6	90.7	90.2	89.4
10	88.7	88.9	89.4	90	91.2	91.3	91.4	90.6	89.6	88.7
11	86.9	81.2	89.5	89.7	90.6	90.8	89.8	88.9	89.2	88.1
12	88.5	89	89.1	89.3	89.5	89.6	89.7	88.6	88.6	88.2

C-1.1

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-1(30 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	72.1	73	73	74	75.4	75.1	74.8	73.9	72.3	71.6
2	72.6	72	73.1	74.6	76.9	76.5	75.7	74	72.4	73.9
3	72.7	72.5	73.3	74.5	80	80	78.3	75.8	74	74.1
4	73	72.8	73.9	76	80	80	79.5	76.1	74.4	73.6
5	73.6	73.3	74.6	75.6	80	80	77.3	76.2	75	74.4
6	73.8	73.3	74.3	76.2	80	80	79.1	76.1	75.4	74
7	73.5	73.4	75	76.1	80	80	77.9	77.3	75.2	74.4
8	73	73.1	74.8	76.4	80	80	76.4	76.2	75	74
9	72.5	73.3	73.9	74.9	76.6	76.6	75.8	75.5	74.8	74
10	73.1	73	73.5	74.9	75.7	75.6	75.1	76.3	74.3	73.8
11	72.9	72.8	73.7	74.5	75.1	75.2	75	75.3	73.9	73.6
12	72.6	73.1	73.3	74.9	74.6	75.2	74.6	74.5	74	73.7

C-1.2

ONLY GENERATOR SET Z-2 (60 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	88.5	88.2	88.8	89	87.9	88.2	88.5	88.1	87.8	87.6
2	88.6	88.7	88.9	89.6	88.8	89.1	89.3	88.8	88.5	88.4
3	88	89.5	90.1	91.7	90.1	90.2	91.3	89.3	88.9	88.6
4	88.9	90.6	91.2	92.4	91.9	92.3	92.9	90	89.2	88.9
5	89.9	91	92.4	93.7	96	96	94.8	90.3	90.4	89.9
6	89.8	91.2	93.4	94.6	96	96	96.3	91.7	91.1	90.5
7	89.6	91.1	93.3	94.1	96	96	96.1	91.5	90.5	90.3
8	89.6	90.6	92.8	93.7	96	96	94.7	90.9	90.4	90.2
9	89.4	90.3	92.3	92.8	94.1	94.4	92.8	90.8	90.3	89.9
10	88.5	89.2	90.8	91.4	92.2	93	91.6	90.2	89.8	89.5
11	87.7	88.1	89.3	89.5	90.6	92.6	90.4	89.9	89.6	89
12	88.2	88.6	89.1	89.3	90.1	91.5	89.4	89.3	89.3	89

C-1.3

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-2 (60 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	71.8	72.4	71	72	74.1	74.1	73.8	73.6	73.1	72.1
2	72.4	72.4	72.8	72.5	75.3	75.7	74.6	74.1	74	73.9
3	72.8	73	73	74	77	77	75.3	76.9	75.9	74.5
4	72.4	72.8	73.1	73.8	77	77	75.9	76.8	74.6	74
5	73.3	73.6	73.1	74.4	77	77	76.5	75.1	73.9	73.1
6	73	74.8	73.9	74.4	77	77	76.3	76.3	75	74.5
7	73.3	73.3	74.5	74.6	77	77	76.3	76.1	75.6	73.7
8	73.2	73.3	73.8	75	77	77	76.5	76.8	74.6	73.6
9	73.1	73.3	73	74.1	75.9	76.1	75.5	75.5	74.5	74
10	72.6	73.4	73.3	74	74.8	74.9	74.6	74.8	73.8	73.4
11	72.5	73.8	73.9	73.9	74.3	74.1	73.8	74.5	73.2	73.5
12	72.6	73.1	74.1	73.4	73.6	73.8	73.9	74.2	73	72.9

C-1.4

ONLY GENERATOR SET Z-3 (90 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	87.5	87.5	87.6	88.2	88.3	88.8	87.7	87.5	87.2	87.1
2	87.8	88.1	88.4	88.8	89	89.6	88.7	88.6	87.1	87.2
3	88	89	89.4	89.5	89.7	90.6	91.7	89.7	88.5	88.7
4	88.8	90.4	90.8	90.9	91.7	91.9	92.2	90	89	89.1
5	89.6	91.2	91.9	92.3	95	95	93.7	90.6	89.6	89.3
6	89.8	91.5	91.6	92.9	95	95	94.2	91	89.8	89.5
7	89.7	91.4	92.2	92.4	95	95	94.8	91.2	89.6	89.6
8	89.4	90.7	91.6	91.9	95	95	93.6	90.7	90	89.7
9	88.9	90	91.2	91.7	93.4	92.9	92.8	90.2	89.8	89.5
10	88.4	89.3	90.2	90.9	91.7	91.8	91.8	89.7	89.2	88.9
11	88.6	89	89.2	89.7	90.5	90.2	89.9	89	88.6	88.4
12	87.5	88.1	88.8	89	89.4	89.3	89.2	88.5	88.4	88.2

C-1.5

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-3(90 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	72	72.3	72.7	72.9	74.2	73.2	72.8	72.7	73	72.5
2	71.8	72.8	72.2	72.9	74.6	73.2	73	72.9	72.6	73
3	72	72.3	73	73.6	78	78	74.4	73.8	74.4	72.3
4	72.4	72.5	72.6	73.2	78	78	76	73.7	73	72.8
5	72.7	72.8	72.8	73.5	78	78	77	74.9	73.3	72.9
6	72.6	72.9	73	73.3	78	78	75.5	74.6	73.6	73.5
7	72.5	73.9	73	73.8	78	78	75.4	74.8	74.6	73.5
8	73.2	73.2	73.4	73.7	78	78	76.2	75.5	75.1	74.4
9	73.3	73.6	73.2	73.7	77.1	76.8	76.9	75.5	75.5	74.9
10	72.5	73.1	73.5	73.9	75.2	75.2	75.9	75	75.2	73.9
11	72.6	73.2	73.3	74	74.5	74.8	74.7	74.2	74	74.1
12	72.3	72.9	73.3	73.4	74	74.1	74.2	73.8	73.2	72.9

C-1.6

ONLY GENERATOR SET Z-4 (120 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	87.6	87.5	87.4	88	88.2	88.6	87.6	87.4	87	87.1
2	88.6	88.3	88.7	88	89.2	89.8	88.9	88.7	87.2	87.2
3	89.2	89.3	89.7	89.9	90.1	90.9	91.4	89.2	89.3	88
4	90	90.6	91	91.9	91.2	92.3	91.8	89.8	89.5	89.1
5	90	90.7	91.5	92	93.8	93.8	93.2	90.2	89.5	89.1
6	90.3	91.2	91.6	92.5	93.8	94.9	93.8	90.7	89.3	89.2
7	90.1	91	91.8	92	93.7	94.8	94	90.5	90.8	89.3
8	89.7	90.2	91.2	91.6	93.3	94	93.2	90.3	89.9	89.5
9	89.2	89.5	89.8	91.3	93	92.5	92.3	89.5	89.2	89.2
10	88.6	89	90	90.2	91.2	91.5	91.2	89.1	88.7	89.5
11	88.8	88.9	89	89.2	90.1	90	89.3	88.7	88.2	88.1
12	87.7	88	88.5	88.7	89	88.8	88.5	88	87.8	87.5

C-1.7

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-4(120 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	73.3	72	72.6	72.4	73	72.5	73.2	73.1	72.3	73.1
2	71.7	71.8	72.5	73.1	73.1	72.9	74.2	73.3	72.5	73.7
3	71.6	71.8	72.6	72.9	78	78	75.3	73.8	73.3	72.6
4	72	72.4	72.2	73.2	78	78	76.3	75.5	73.2	73.1
5	72.7	72.6	73.2	74.1	78	78	76	75.1	74	73.6
6	72.4	73.1	73.1	73.8	78	78	76.9	75.6	74.2	73.4
7	72.8	73.1	73	73.8	78	78	76.7	76.5	74.3	73.4
8	72.5	73	73.2	74.4	78	78	76.5	75.5	74.1	73.9
9	73.1	72.8	73.5	74.9	79.3	78.3	76.6	75.4	74.8	73.8
10	72.5	73.3	73.5	75.1	77.2	76.3	75.8	75.4	74.5	73.6
11	72.7	72.9	73.4	74.4	76.1	75.2	75.3	74.9	74.1	74
12	72.6	73.2	73.4	74	74.4	74.4	74.8	74.4	74.1	73.8

C-1.8

ONLY GENERATOR SET Z-5 (150 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	87	86.9	87.5	87.6	87.2	87.4	87.2	87	87.1	87
2	87.2	87.1	87.6	87.9	88.2	88.4	87.3	88.6	88.7	88.4
3	87.4	88	88.3	88.8	89.9	90.2	89.4	89	88.9	88.6
4	87.8	88.3	89.1	89.6	90.4	90.6	89.3	89.1	89.4	89.6
5	88.4	89	89.7	90.3	90.9	91	90.6	90	89.7	89.9
6	88.9	89.4	90.8	91.9	91.8	92	91.2	90.8	90.1	90.1
7	88.8	89.3	90.6	91.6	91.7	91.9	91.4	91	90	90
8	88.7	89.3	91.2	91.4	91.5	91.8	91.5	90.9	89.8	89.7
9	88.7	89	90.3	90.1	91.2	91.4	91	90.6	89.6	89.4
10	88.6	88.9	89.8	90.2	90.9	90.5	90.3	89.9	89.4	89.1
11	88.4	88.5	89.6	89.9	90.2	90	89.9	89.6	88.7	88.6
12	88.3	88.1	88.1	88.4	89.9	89.8	89.7	89.5	88.1	88.1

C-1.9

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-5(150 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	72.3	72.1	72.6	72.7	73.5	72.9	73.4	73.3	72.7	73.1
2	72	72.1	72.6	73.3	73.4	73.1	73.8	73.4	72.6	73.6
3	71.9	72.9	73	73.1	79	79	75.9	73.9	73.3	72.8
4	72.2	73.8	73.9	73.6	79	79	76.4	75.8	73.4	73.1
5	73.4	73.9	74.3	74.4	79	79	76.1	75.7	74.1	73.7
6	73.7	74.3	75.1	74.3	79	79	76.8	75.5	74.4	73.6
7	73.7	74	74.9	74.2	79	79	76.7	76.5	74.3	73.4
8	73.6	74.1	74.6	74.8	79	79	76.7	75.5	74.3	73.7
9	73.4	73.9	74.8	75.2	79.2	78.5	76.8	75.7	74.9	74
10	73.2	73.8	74.1	75.3	77.5	76.4	75.9	75.6	74.8	73.6
11	73.1	73.3	73.9	74.6	76.3	75.3	75.6	75.1	74.4	74.1
12	72.9	73	73.6	74.3	74.8	74.6	75	74.4	74.2	73.9

C-1.10

ONLY GENERATOR SET Z-6 (180 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	86.7	86.9	85.9	86.1	86.5	86.6	86.5	86.6	86.6	86.5
2	86.4	87.3	87.2	86.3	86.4	86.9	86.9	86.9	87.3	87
3	86.8	87.9	87.9	87.2	86.6	88.3	87.6	88.2	88.1	88
4	87.3	87.9	87.8	88.4	87.4	88.9	89.1	89.6	88.5	88.3
5	86.9	88.1	88.1	89.9	89.4	89.9	90	90.2	88.9	88
6	87.7	88.6	88.5	90.2	90.4	90.5	90.2	90	89.1	89.1
7	88.9	88.5	88.2	89.9	90.2	90.2	90.1	90	88.8	89
8	87.4	88.3	88	89.7	90.5	89.7	89.8	89.7	89.5	89.4
9	88	88.4	88.2	89.2	89.4	89.2	89.2	89.1	88.7	88.5
10	87.7	88.2	88.7	89	89.1	89.3	89.1	89.1	88.6	88.4
11	88.1	87.9	87.8	88.8	89.5	88.9	89	89	88.5	88.6
12	87.6	87.8	88	88.4	89	88.7	88.6	88.7	88.9	88.4

C-1.11

GENERATOR WITH ENCLOSURE, SILENCER & INLET, OUTLET DUCT Z-6(180 Cm)										
Points	1	2	3	4	5	6	7	8	9	10
1	72.3	72	72.6	73.7	74.3	75.2	73.1	72.6	72.9	72
2	72.4	72.5	72.8	74.8	75.6	75.3	73.8	73.1	73.4	72.4
3	72.9	73.3	73.4	74.6	75.9	75.1	74.7	72.7	72.9	72.8
4	72.8	73.7	74.4	76.3	76.4	76.8	75.8	74	74.2	73.4
5	74.2	74.2	75.4	76.7	78	79.3	76.3	74.6	73.9	73.9
6	74.3	75	76.5	78.8	81	84.2	77.3	75.6	74.8	73.8
7	74.4	75.2	76.8	78.9	81.6	84.6	79.1	76.8	74.9	73.5
8	74.3	74.8	76.2	78.8	82	88.4	80.4	77.3	75.6	74.3
9	73.8	74.5	76.5	78.9	81.4	84.6	79.6	76.6	75.4	73.6
10	73.6	74.4	75.1	75.9	77.9	79.7	77.6	75.8	75.2	74.2
11	73.4	73.8	74.2	75.4	76.3	76.9	76.3	75.4	74.3	73.8
12	73.3	72.9	73.8	74.5	75.7	75.1	75.3	73.9	73.6	73.7

C-1.12