ACOUSTIC DESIGN OPTIMIZATION OF CLOSE-FITTING ENCLOSURE USING GENETIC ALGORITHM TOOL FOR DIESEL POWER GENERATOR SOUNDPROOFING APPLICATION

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Dissertation submitted in partial fulfillment of the requirements for the degree

Master of Science

Msc in Industrial Automation.

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Sri Lanka

May 2015

DECLARATION OF THE CANDIDATE AND SUPERVISOR

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ABSTRACT

Reciprocating engine power generator sets produce uncomfortably loud noise when it is in operation. Installation of Power Generator (PG) shall comply with local noise regulations. Further price, space and weight ensure a commercially appealing product in PG market. However in Sri Lanka most of the locally fabricated soundproof PGs has failed to meet SPL regulations even though it is in reasonable low price compared with imported soundproof PG. Open discussion about this problem with the local PG suppliers revealed that, absence of simple and fast acoustic enclosure design procedure customized with the SPL spectrum of open PG and the customer requirement of space, weight and cost has created this problem.

Close-fitting enclosure fabricated with sheet metal enclosure face insulated with sound absorption materials is method of Passive Noise Controlling (PNC) used in PG soundproofing. SPL model of soundproof PG was developed considering Insertion Loss (IL) of the enclosure and SPL spectrum of the open PG. The model was constrained for customer required SPL, cost and weigh. Effective deign variables of the model were identified and developed an optimization code for selecting optimum minimum values for the identified variables using Genetic Algorithm (GA) optimization tool in MATLAB. Optimization were converted to user friendly deign application through a Graphical User Interface (GUI).

Validity of the developed design methodology was done by comparing the model predicted data with manufacturer given data for selected set of "Cummins" power generators. After that design variables were predicted for open type standby power 22kVA "Cummins" PG with 75% load at 3m distance and the acoustic enclosure for the model was fabricated accordingly. SPL measurement of fabricated enclosure realized the developed methodology is substantially accurate and result can be used for the preliminary design of the enclosure. Accuracy of deign can be developed further by considering the effect of noise leak through opening and the effect of sound attenuator.

DEDICATION

To my family who proved lovely relationships beyond the logic

ACKNOWLEDGEMENT

I would like to gratefully acknowledge research supervisor Dr. D.P. Chandima, Senior Lecturer, Department of Electrical Engineering, University of Moratuwa for his dedication, inspiration and great effort throughout the research. Also I would like to thank senior lecturers of Department of Electrical Engineering, University of Moratuwa for their valuable feedbacks and advices during progress reviews that directed my research towards a realistic end.

I gratefully thank Eng. P.K. Chadrasekars, Chef Engineer, Power Plant section, Ceylon Electricity Board for providing entire resource for the research.

I am thankful Power Generator engineering team of the Trade Promoters (Pvt) Ltd, Hayleys Industrial Solutions (Pvt) Ltd and Metropolitan Engineering (Pvt) Ltd. for their immense support on problem oriented survey and providing manufacturer data of commercially available Power Generators.

Last but not least, I am grateful to Department of Electrical Engineering, University of Moratuwa for providing me the opportunity to read MSc in Industrial Automation successfully.

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LIST OF ABBREVIATIONS

Abbreviation	Description
SIL	Sound Insertion Loss
dB	Decibel
OSHA	Occupational Safety and Health Administration
dB(A)	A weighted Decibel
ANC	Active Noise Control
PNC	Passive Noise Control
TL	Transmission Loss
IL	Insertion Loss
GA	Genetic Algorithm
GUI	Graphical User Interface

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CHAPTER 01
INTRODUCTION

Sound is what the human ear hears and noise is the unwanted sound. It is produced by vibrating objects and reached the listener's ear as pressure waves in the air or other media.

Sound Pressure Level (SPL) at the point of hearing is measured in Pascal (Pa). Expression of SPL in that form is inconvenience since ratio of softest sound the ear can detect to loudest sound ear can experience without damage is approximately 1:106. Therefore in acoustic SPL measurements are reported in terms of decibel (dB) scale. Decibel scale is base 10 logarithmic scale define in terms of reference pressure of $20\mu Pa$. The whole range of human hearing can be described by number range from 0 dB threshold of normal hearing to 140 dB threshold of pain. In 1972, Occupational Safety and Health Administration (OSHA) has adopted frequency weighted decibel scale as the officially regulated Sound level descriptor. With this regulation SPL is measured with "A weighting filter". This is an approximation used to correct the SPL, to reflect what the human year perceives more accurately. Therefore all the noise regulations and sound level of machine are described in "A weighted" decibel scale dB (A).

Federal safety regulation of OSHA and municipal noise ordnance are two main regulations affected for the noise level exposed by individual or public. In Sri Lanka, maximum permitted overall noise level at the property line ranges from 45dB (A) to 75dB (A) [1] depending on the location and zoning. OSHA regulation is only applicable for the workers who would exposed to machine noise above 80dB (A) for any appreciable period of time.

Noise can be controlled by two methods. Active Noise Controlling (ANC) is a method of reducing or cancelling unwanted sound by generating a sound pressure wave which is specifically designed to cancel the unwanted sound wave component. ANC method works best for the standing wave with low frequencies ($< 500\ Hz$) [2].

As an example noise reduction in industrial exhaust stacks can be done by using ANC. However ANC is an expensive controlling method which requires state of the art electronic hardware and precision computer software.

Widely used cost effective method of noise controlling is Passive Noise Controlling (PNC). Introduction of an obstacle to the sound propagation path in between the source and receiver is the method controlling which is used in PNC. The obstacle is a stiff and dense barrier. Sometimes the sound pressure wave incidental surface of the barrier is insulated with sound absorption material. Sound pressure wave reflection property of weighted barrier materials and sound wave absorption property of sound absorption material control sound radiation through the barrier. PNC is effective for broad range of frequencies, though they are often inefficient at low frequency [3].

PNC can be done as a barriers or an enclosures depending on the size of noise source and receiver. Barrier is effective when noise source or the receiver/s is too large and difficult to cover by an enclosure. As an example high way traffic noise is controlled by locating noise barriers. When the noise source or the receiver is small enough to cover by all sides, barrier effectively become an enclosure.

Acoustic Performance of any type of noise controller shall be given by a noise terminology. Two noise terminologies are used to discuss the acoustic performance in PNC. Transmission Loss (TL) is the base 10 logarithmic ratio of the amount of sound Pressure wave transmitted through a surface to the pressure wave incident on the surface. Acoustic performance of noise barriers can be measured in TL. This is done by measuring the SPL immediately before and after the barrier assuming sound wave is not diffracted around the barrier [4], because the barrier is considered infinite. However measurement of SPL near the panel barrier is impractical since the panel surface behaves as a non-rigid wall which is subjected to vibration and near field effect. Therefore it is difficult to measure the incident and transmitted pressure wave without using sensitive measuring equipment. Therefore acoustic performance are measured in Insertion Loss (IL) in most of engineering perspectives.

IL is define as the difference between the SPL before and after the installation of a barrier or an enclosure, measured at a location which lies outside the barrier or the enclosure after its installation.

Enclosures are commonly used for soundproofing of machineries. With a careful design and fabrication of an enclosure, noise can be attenuated up to 50 dB(A) [5]. There are different types of enclosure deigns which are fabricated based on the operation and maintenance requirements of the machine to be enclosed. Sealed enclosure versus partially open enclosure accommodate the air ventilation requirement of the machine during the operation. Distance between the noise source and the enclosure panel categorizes the enclosures as far fitting enclosure and close fitting enclosure. In acoustic theories, if source to panel distance is less than 1m or noise source occupy more than 1/3 [6] of the enclosure volume is defined as a close fitting enclosure. Further the enclosure can be constructed as separate free standing type or machine mounted type. Acoustic behavior with in the enclosure is different with different enclosure contractions.

To date, researchers [7, 8] have developed lot of mathematical models for enclosures using the Newton's mass low and the stiffness effect of the panel. But the acoustics of close fitting enclosure is totally different than that model since the noise source and the enclosure panel is too close and air gap is small. The close fitting enclosure panel are considered to be in near field developing an acoustic coupling between the source and panel [9]. This acoustic coupling makes the enclosure panel vibration. Panel vibration against the sound pressure wave create resonance.

In 1991, Oldham [9, 10] has developed a mathematical model to predict the IL of close fitting enclosure. His work is heavily concerned with vibration of the enclosure panel caused by the acoustic pressure from the noise source. This model influenced not only by the material properties and thickness of the panel but also by the distance the panel is from the noise source. The theory developed by Oldham has being using for different researches [6] and had proved that this theory closely follow the actual behavior of close fitting enclosure.

Mass low based TL model and oldham's theory based IL model behavior of an enclosure is shown in Figure 1.1. According to the figure both TL and IL show increasing trend with respect to the frequency. But IL model represent nulls at certain frequencies ($f_0$and f_{sw})which are created by the resonances.

Resonance at $f_0$is structural resonance in enclosure panel which occurred at low frequencies. This happens where the enclosure wall mass is opposed by the wall and air gap stiffness. This resonance frequency can be increased by increasing panel stiffness and the amount of IL reduction at this frequency can be control by good mechanical damping of enclosure panels

Resonance at f_{sw} is standing wave resonance in the air gap between the machine and the enclosure. This occurs at high frequency level about 1000Hz. IL reduction due to this resonances can suppressed by the placement of sound absorbing material at wave incidental surface.

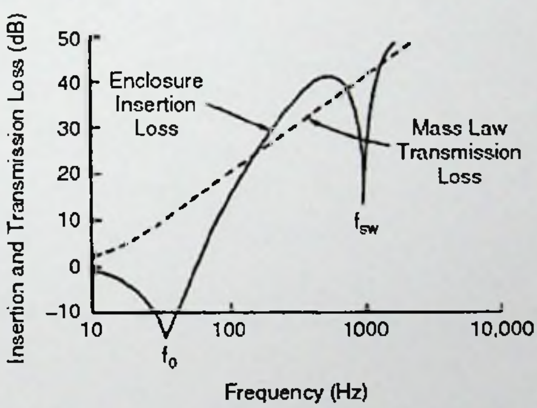

Figure 1.1: TL and IL model behavior of a close fitting enclosure (Source: Hand book of noise and vibration control, Malcolm J Crocker)

Reciprocating engine powered generator set produce uncomfortable noise when it is in operation. Noise level of unenclosed Power Generator (PG) vary from 90 dB(A) to 110 dB (A) at 1m point of hearing depending on model of PG. Table A.1 in Appendix A is shown the SPL spectra of selected "Cummins" unenclosed PGs.

Figure 1.2 shows the SPL spectra over octave band frequency plot of selected PG models.

SPL regulated by local ordinance belongs to the rage bounded by the dotted lines in the figure 1.2.The lower limit is 45dB (A) and upper limit is 75 dB (A). SPL of all the PG models not within the above mentioned range through the frequency spectrum beyond the 63Hz.

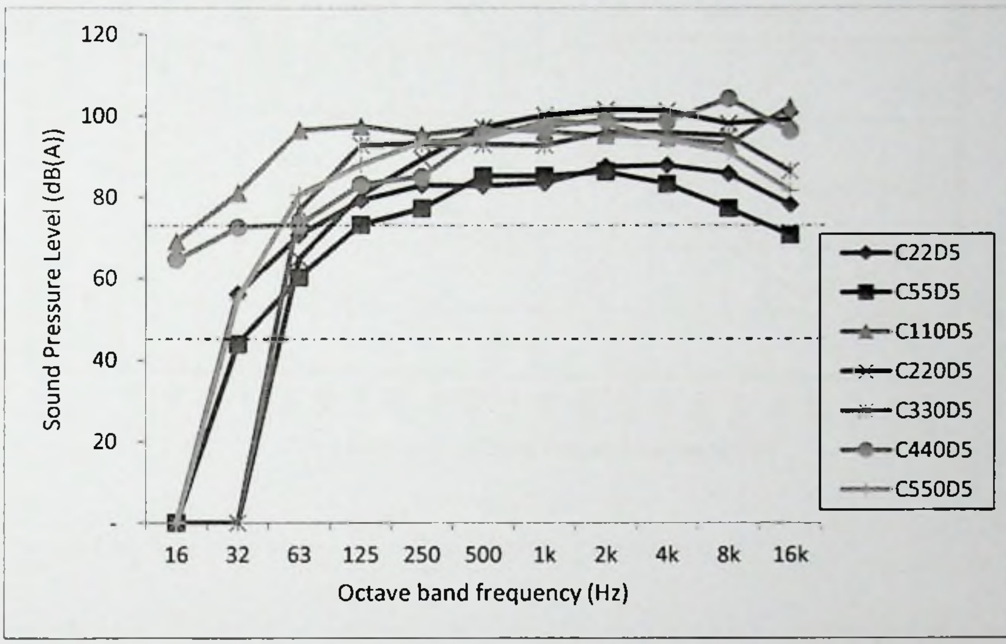

Figure 1.2: SPL spectrum of "Cummins" PGs @ 1m distance 110% load (Source: Power Suit Library, www.cumminspower.com)

Diesel engine noise can be said to arise from three sources. These are engine air flow interaction, combustion noise and mechanical noise [11]. Figure 1.3 represents the SPL spectra of various sources of diesel PGs. According to the figure noise from air flow interaction (exhaust and intake and the cooling fan ventilation) is predominant at low frequency range which requires proper controlling using the passive as well as active noise controlling. However the active noise controlling for the diesel power generator soundproofing has not adopted yet since there are some limitations with the technology such as high cost and SPL spectra is not represent a standing wave behavior. Normally passive noise controlling is economical choice for the diesel power PG soundproofing since it is best at frequencies above about 200Hz [11].

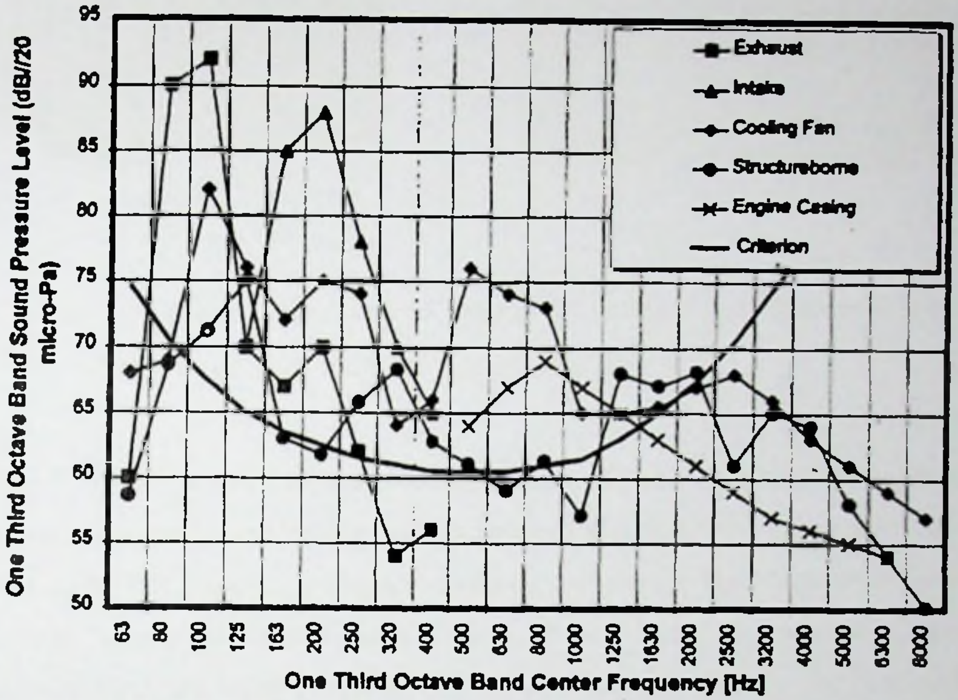

Figure 1.3: Source of noises from 30kW Diesel PG (Source: [11])

This air borne noise due to air interaction is controlled by three methods as follows.

a. Introduction of passive silencer for the exhaust air.
b. Covering of the entire machine with canopy
c. Ventilation air inlet and exhausted to and from the canopy through sound attenuation system.

At designing stage, firstly the silencer shall be designed and then the canopy shall be designed as a noise barrier to overall sound radiation. Enclosed PGs are categorized in to two considering the canopy construction. They are packaged standard type and Power Box type as shown in figure 1.4. Low power rated PGs up to 550kVA are enclosed as packaged standard type and the higher power rated PGs beyond 550kVA are enclosed as Power Box.

(a)

(b)

Figure 1.4: Power Generator with soundproof enclosure; (a) Standard type (b) Power Box type (Source: www.cumminspower.com)

Standard size containers are used as the enclosure in Power Box type. PG in installed inside the container and soundproof the container to suppress the noise up to required SPL using PNC method. Free space within the enclosure is big which allow the walk around service and maintenance facility. Power Box type enclosure can be considered as far fitting enclosure and enclosure performance can be predicted by using techniques developed in architectural acoustics.

Packaged standard type soundproof PGs consist with specially designed soundproof enclosure as a compact, light and low price product. This type of enclosure is fabricated as a close fitting enclosure and typically reduce radiated noise by a minimum of 10dB (A) [12].

With the growth of standby power installation in highly polluted areas, demand for soundproof PGs is increasing. Further compact and light PG at low cost has become commercially appealing in modern market. Currently in Sri Lanka, most of PG agents have tends to import PG parts separately and assembled locally or import open PG and do the modifications as per the customer requirements locally. Under this practice most of soundproofing of PG is done locally. According their point of view locally assembled PG is cheaper than original factory assembled PG which helps to be competitive in local market. But customer satisfaction on locally soundproof PG is poor because most of the locally soundproof PGs have failed to comply with the SPL level requested by the customer at the end of the fabrication of enclosure.

Open discussion with the local agents regarding this problem revealed that absence of proper enclosure design methodology is the problem. Normally the enclosures are fabricated as duplicate of equivalent product without considering acoustics of the unenclosed sound source. But the study of sound propagation pattern of the assembled PG is important to develop the noise controller for that noise. Because SPL spectrum of any type of machine is different from machine to machine even with the same model. For example PG noise is produced by six major components. They are engine, cooling fan, alternator, induction, engine exhaust and structural vibration. The operation characteristics of each item as well as the quality of assembling of them define the noise spectrum of the PG. Therefore it is required to measure the SPL spectrum of open PG and shall do the enclosure design based on this.

With the above mentioned problem I motivated to development of an acoustic design procedure for standard type PG enclosure. I focused further to facilitate customized constrains of cost, weight and dimension of the enclosure with the design. This help to give solutions for acoustic problems with a great customer satisfaction.

Hence the research was casted to achieve the objective of acoustic design optimization of close-fitting enclosure for packaged standard type power generators with cost, space, weight and sound pressure level constrains to create customer satisfaction in locally made enclosures. The optimization of design involves shape optimization of the enclosure by selecting the best dimensions and the selection of optimized sheet metal material and sound absorption material property and thickness combination to achieve the customer defined SPL requirement at minimum cost. The Weight constrain can be imposed on the design model if it is prevailing factor by customer side.

The work done to achieve the objective is organized in next four chapters. In the 2nd chapter, development of mathematical model to calculate the SPL of PG with the soundproof close fitting enclosure will be discussed with all the theories, assumptions and equations. Hereafter effects of model variables on IL of enclosure will be discussed with IL versus octave band frequency plots. In effect analysis noise source dimension, enclosure panel material (sheet metal and sound absorption material) properties and thickness and the source to panel distance were considered as model variable. This chapter will be ended up with the discussion of noise leak correction factor and customer required constrains assignment on the original model.

Chapter 3 describes the design methodology developed to calculate optimum minimum values for the model variables under the customized constrain. This method was developed using the Genetic Algorithm (GA) optimization tool box in MATLAB. Design methodology was concluded with a Graphical User Interface (GUI) in MATLAB as software base application to make the design process simple and fast for the users.

Under Chapter 4, validation of the developed design methodology by actual implementation will be discussed. Acoustic enclosure for 22kVA "Cummins" PG at 75% load of operation was design using the developed methodology and the enclosure was fabricated accordingly. SPL measurements were taken and results were compared.

Finally in chapter 5 ultimate achievement of the research, credibility and the gaps of the developed methodology will be concluded. Recommendations for further development for the initiated methodology will be discussed at last.

CHAPTER 2

DEVELOPMENT OF SPL MODEL FOR PACKAGED STANDARD TYPE PG ENCLOSURE

Standard type Power Generator enclosure is fabricated by reverting sheet metal panel to a steel frame clamped to heavy metal base structure. Open PG is installed on the heavy metal base structure trough flexible support which absorb the structural vibration of PG mechanically. The inner surface of the sheet metal enclosure is insulated with fibrous or open cell foam type sound absorption material. There are two openings for ventilation air intake and exhaust. The combustion is exhausted through a passive silencer. This type of enclosure can be modeled as Partial, closing fitting and separately mounted acoustic enclosure.

The acoustic behaviors of those enclosures cannot be predicted by the theories use in architectural acoustics since the close fitting enclosure acoustics represent internal impedance effect and resonance effect which are not addressed in architectural acoustics. In the internal impedance effect, if the noise source has low internal impedance close fitting enclosure can "load" the source so that it produce less sound power [13]. However in most machinery including PG noise problem, the internal impedance of the noise is high enough to make this effect negligible.

Foremost effect is resonances where the structural resonance in the panels and standing wave resonance or the cavity resonance in the air gap. In structural resonance, certain structural modes of the enclosure are excited from the noise and cause the panel to vibrate. The vibration of enclosure panel effectively becomes a vibrating piston that the structure can increase or decrease the IL depending on the frequency of the excited mode and the frequency content of the noise in the enclosure. If the panel vibrate in phase with the incident pressure wave, the transmitted pressure is virtually unaffected by the enclosure. In cavity resonance, might produce a pressure that is greater than the driving pressure which results in a negative IL. These excited resonate mode will cause structure to vibrate and produce new pressure wave, adding to the pressure wave emitted from the noise.

2.1 IL Model for close fitting sealed enclosure

Figure 2.1 illustrate schematic diagram sealed closefitting enclosure with the noise source and the sound measurement point. IL model for the enclosure shown in figure 2.1 was developed using the Oldham's theory.

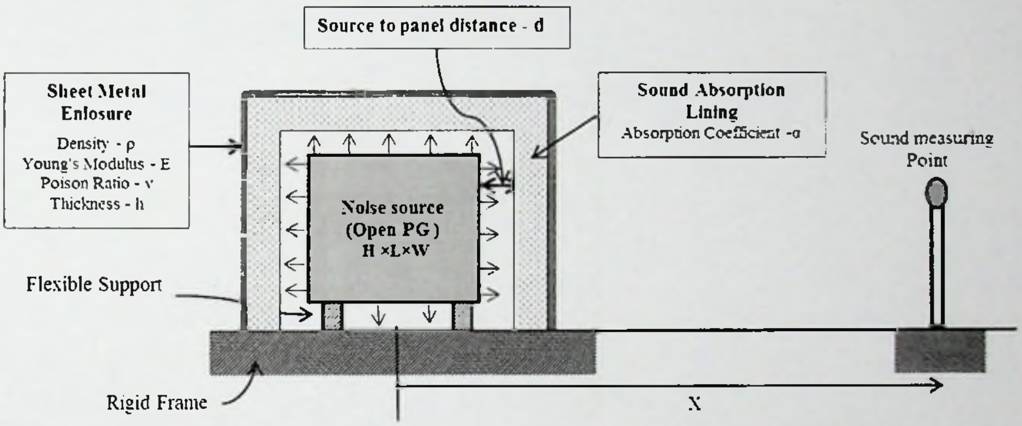

Figure 2.1: Schematic diagram of sealed close fitting enclosure with PG noise source and noise measurement point

The system was modeled with the following assumptions.

1. Sound source (open PG) is approximated as cuboid in shape which is placed on a rigid frame. Five sides of sound source are assumed to be a vibrating panel which is positioned closed to the enclosure panel. The enclosure panel is therefore exposed to uniform pressure field over a large frequency range. The simplified one dimensional model is shown in figure 2.2.
2. The open PG is mounted to rigid frame trough vibration isolators and passive silencers are passed through vibration isolated holes. Therefore and noise transmission as vibration through the Structure borne path were neglected.

3. The panels are excited by higher frequencies from the noise source and high modes of the panel are excited. The high order mode shapes will, however, essentially cancel their effects, causing them to be ineffective at radiating sound. Therefore it is assumed that the 1:1 mode of structure is the only effective radiator of sound
4. The panels are assumed as sheet metal with single side insulation of absorption material.
5. Enclosure is fabricated with n numbers of panels

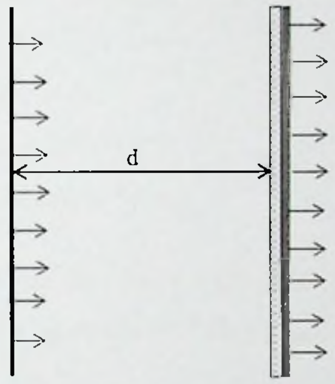

Figure 2.2: Simplified one dimensional model for closefitting enclosure panel

According to the Blank's finding [6] based on Oldham's theory. SPL modeling of the PG enclosure can be intuited with Oldham's IL model for Clamped boundary conditions. IL of panel without sound absorption material insulation ($IL_{Bare\ Panel}$) was given by equation (1)

$$IL_{Bare\ Panel} = 10\log_{10}\left[\left(\cos(kd) + \left(\frac{\pi^2}{4K\omega\rho_0 c}\right)\sin kd\right)^2\right] \quad (1)$$

$$K = \frac{1.35}{\left[3.86D\left(\frac{129.6}{a^4}+\frac{78.4}{a^2b^2}+\frac{129.6}{b^4}\right)-\omega^2\rho h\right]} \quad (2)$$

$$D = \frac{Eh^3}{12(1-v^2)} \quad (3)$$

where,

k - Acoustic wave number, $k = \dfrac{2\pi f}{c}$

d – Source to panel distance in m

f - Frequency of the sound wave in Hz

ρ_0- Density of fluid medium inside the enclosure (Air with density of406 kg/m^3)

c - Speed of sound in air (343 m/s)

D – Bulk Modulus of panel

a – Length of the panel in m

b – Width of the Panel in m

h – Thickness of the Panel in m

E – Panel's' Young modulus N/m^2

ν – Poison Ratio of the Panel Material

ρ- Density of Panel Material kg/m^3

Equation (1) was modified to the equation (4) given below to calculate IL of panel with sound absorption material insulation ($IL_{AB\ Panel}$).

$$IL_{AB\ Panel} = 10\log_{10}\left[\left(\frac{ab\alpha}{1-\alpha}\right).\left(\cos(kd) + \left(\frac{\pi^2}{4K\omega\rho_0 C}\right)\sin(kd)\right)^2\right] \qquad (4)$$

Where, α is he sound absorption coefficient of insulation material.

At the design stage all noise terminologies shall be calculated for the effective frequency range in order to identify their acoustic behavior with respect to the frequency. Because the noise wave is a blend of different frequencies and pressure wave has different strength at different frequencies. Therefore the spectrum of data helps to identify the alarming strengths to control them accordingly.

To calculate the IL at different frequencies using equation (4), α at the frequencies considered shall be known. Normally α vs. frequency data of absorption material can be find using the manufacturer data sheet. But it is difficult to collect the manufacturer data sheets from the material supplier in local market. The problem can overcome by calculating α using empirical relationship [14]. The empirical relationships have developed by considering the density and material thickness of sound insulation materials. Delaney & Bazley's relationship [15] for fibrous material and Dunn & Davern's relationship [16] for open cell foam material are the commonly used empirical equation to calculate α.

Under this research mineral wool of type rock wool and glass wool materials were considered as sound absorption material because this type of fibrous material are durable with high temperature and oily PG operation environment. Therefore Delaney & Bazley's empirical relationship was used to calculate α as given below.

$$\alpha = 1 - \left|\left(\frac{(Z-1)}{Z+1)}\right)^2\right| \tag{5}$$

$$Z = z_c \coth(R.t) \tag{6}$$

$$z_c = (1 + 0.0571C^{0.754}) + (0.087C^{0.732})i \tag{7}$$

$$R = k(0.189C^{0.595}) - k(1 + 0.978C^{0.7})i \tag{8}$$

$$C = \frac{\tau}{\rho_0 f} \tag{9}$$

where,

R - Flow resistivity of Absorption Material

t - Thickness of the Absorption material

Flow resistivity of rock wool and fiber glass wool material with known density can be calculated by using flow resistivity vs. density plots given in Figure B.1 under Appendix B

2.2 Development of SPL model of PG with acoustic enclosure.

SPL of PG with enclosure is depends on the SPL of the open PG and the IL of the enclosure. The enclosure is fabricated with n numbers of panels and equation (10) gives the SPL at the point of noise measuring after introducing an acoustic panel.

$$\mathrm{SPL}_i = SPLO_i - IL_i \tag{10}$$

where,

i - Frequency

$SPLO_i$ - SPL of the open PG at the noise measuring point

IL_i - IL of one enclosure panel.

SPL_i - SPL after introducing a panel at noise measuring point.

But in commercial term SPL is given as single figure of overall SPL. Overall SPL is calculated as logarithmic sum of SPL in all frequencies. Resolution of the frequency is not affected to the final results. In PG acoustics all the measurement are taken at octave band frequency. Hence overall SPL is calculated using equation (11).

$$SPL_{overall} = 10 * \log_{10}\left\{\sum_{i=1}^{11} 10^{SPL_i/10}\right\} \tag{11}$$

SPL at the point of noise measuring after introducing n numbers of panels (i.e. enclose) ($SPL_{Soundproof\ PG}$ was calculated by equation (12)

$$SPL_{Silenced\ Soundproof\ PG} = 10 * \log_{10}\left\{\left(\sum_{j=1}^{n} 10^{SPL_{overall\,j}/10}\right)/n\right\} \tag{12}$$

where, j is the panel number.

Numbers of panel used for the fabrication of the enclosure is depends on the ultimate dimension of the enclosure. The dimension of the enclosure depend the dimension of the open PG (H×L×W) and the source to panel distance(d).d is determined by the design and difficult to fix at the initial stage. Therefore, the equation (12) was simplified by assuming numbers of panels in on side of cuboid of enclosure as single panel. Hence there are only five panels and the equation (12) can be re write as in equation (13)

$$SPL_{Silenced} = 10 * \log_{10}\left\{\left(\sum_{z=1}^{5} 10^{SPL_{overall_z}/_{10}}\right)/5\right\} \tag{13}$$

where, z is the side of the cuboid, i.e. left, right, back, front and top

2.3 Effects of model variable on SPL model

SPL of the open sound source and IL of the enclosure defines the SPL of enclosed sound source. But the SPL spectrum of the sound source is specific to the sound source which cannot control at enclosure design stage. Therefore controlling of IL to match with SPL of the open sound source is controlling method which can be done by careful deign of the acoustic. IL of enclosure depend on following variable as per the equation (4).

a. Panel dimension(a & b).
b. Source to panel distance(d)
c. Sheet metal material property (ρ, E, υ)
d. Sound absorption material property (α)
e. Sheet metal material thickness (h)
f. Sound absorption material thickness (t)

Variation effect of those model variables on IL model was analyzed to get the idea on controlling the IL. Variation of only one variable was considered at a time and the rage and step size of variation was limited defined by considering the material type, property and dimension of locally available materials as well as the fabrication capability, suitability for machine operation environment and realistic nature of cost and weight.

Effect of each model variable were analyzed by using two plots. First one was IL vs. octave band frequency of complete audible range. Second plots was IL vs. low frequency octave band up to 500 Hz. In the first plot, standing wave resonance can be observed clearly. At variable standing wave resonance is at 1000 Hz except in source to panel variation plot. Air gap between source and panel vary with the source to panel distance variation and this result in standing wave resonance at different frequency. The structural panel resonance can be observed in low frequency plots in frequencies less than 100Hz. But it is difficult to identify this resonance clearly in most of the variable effect plot. Because the plot are located only in octave band frequencies, but the structural panel resonance can be happen in a frequency in between two octave band frequencies.

2.3.1 Effect of panel dimension on enclosure IL model

As per the equation (13) panel dimension is defined by the enclosure dimension. Enclosure dimension is function of the dimension of the sound source ($H \times L \times W$) and the source to panel distance d . In the panel size effect analysis d was fixed and the PG size was varied by selecting five different standard type PGs under the make of "Cummins". Dimensions of the selected open PGs are given in the Table A.2 in Appendix A.

Effect of open PG dimension on IL is given in figure 2.3 and 2.4. According to the figures the dimension of the sound source is not effected to the IL significantly throughout the frequency range. However according to the theory, as the panel gets larger panel stiffness decrease and this result in decreasing structural resonance frequency. According to figure 2.4 the structural resonance has happened around the frequency of 16 Hz. At that frequency, there is significant IL drop when the PG dimension is increased.

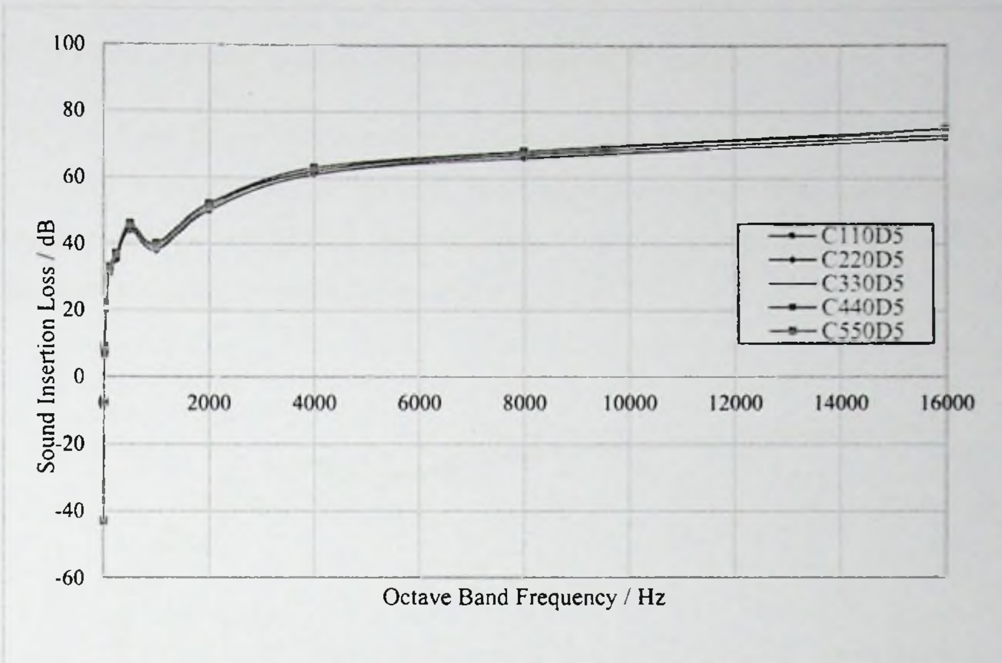

Figure 2.3: Effects on IL due to changes in PG dimension

(2mm thick Galvanized Steel sheet metal, 50mm thickness Rock Wool with the density of 50 kg/m^3 and $d = 0.5m$)

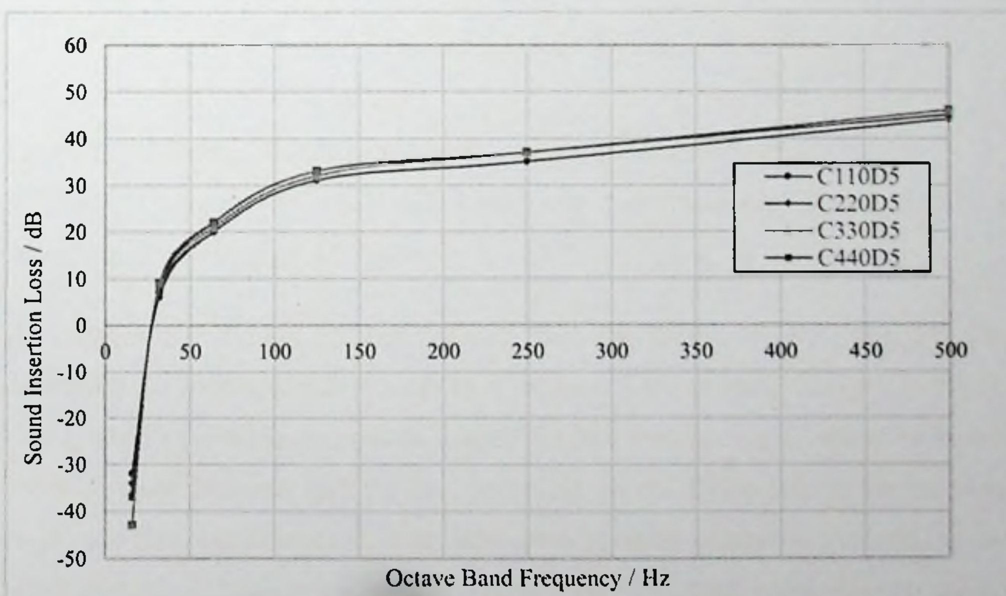

Figure 2.4: Effects on IL due to changes in PG dimension at low frequency

2.3.2 Effect of source to panel distance on enclosure IL model

Figure 2.5 and figure 2.6 illustrate the effect of source to panel distance variation on IL model. According to the figures IL is heavily influenced by this variable because this variable not only change the panel dimension as discussed under section 2.2.1 but also change the air gap between source to panel.

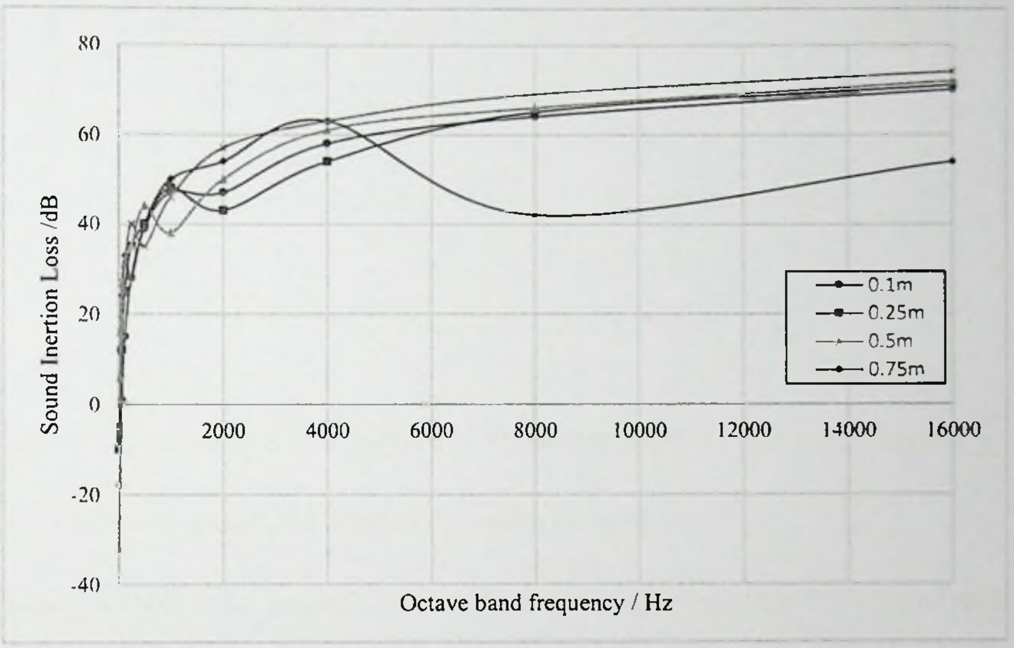

Figure 2.5: Effect of IL due to changes in source to panel distance
(C110D5 "Cummins" PG, Galvanized steel with 2mm thickness, 50mm thick Rock Wool with the density of 50 kg/m^3)

As illustrated in figure 2.5, the distance from the source to panel (i.e. air gap) determine the frequencies of the cavity resonances and resulting decrease in the IL. Cavity resonance frequency moves toward the low frequency side when the source panel distance increase and the IL. According to the figure 2.6 structural panel resonance frequency decrease when the source to panel distance increased. In this effect analysis IL represent a chaos behavior because the both resonances are highly influenced by this modal variable

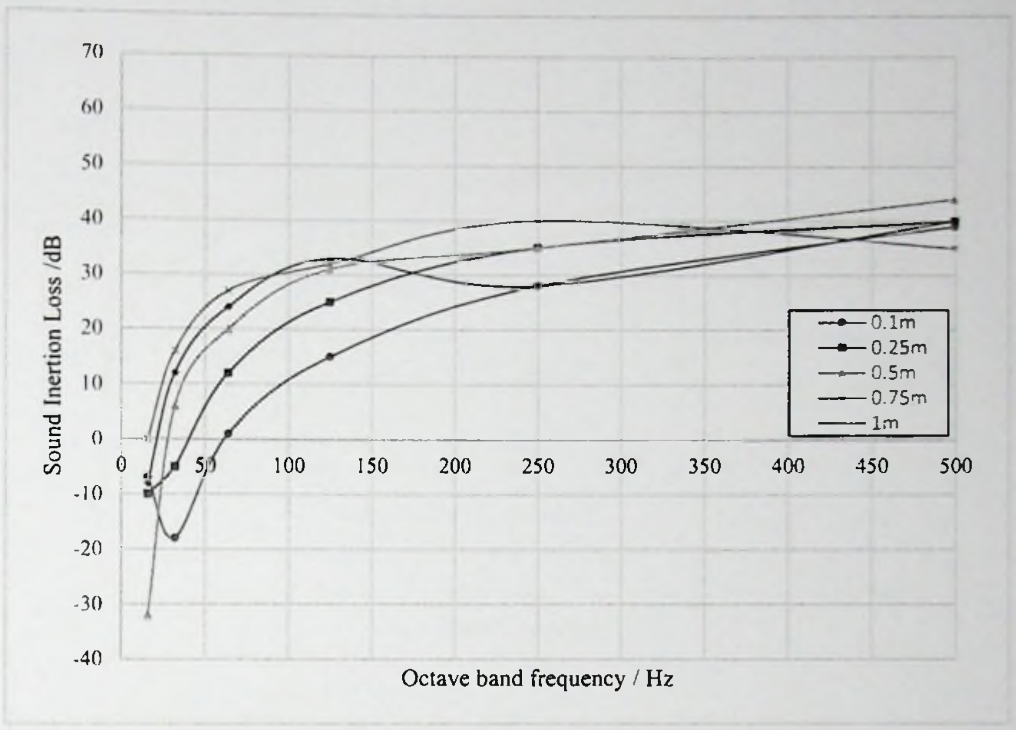

Figure 2.6: Effect of IL due to changes in source to panel distance at low frequency

2.3.3 Effect of sheet metal material Properties on IL Model

Effect of sheet metal property on IL was analyzed by considering five type of sheet metal with different density. The selected materials were Aluminium composite, Aluminium, Galvanized Steel, Copper and Led. Material properties of sheet metal are tabulated in Table-B.1 in Appendix B.

IL behaviors with different sheet metal material are illustrated in figure 2.7 and 2.8. According to the figures IL increase with increasing the panel density over all octave band frequencies except at the 16Hz frequency. At that frequency structural panel resonance of high density panels has occurred and IL reduction is higher. The structural panel resonance of low density panel has occurred at 32Hz. Therefore figure 2.8 means stiff and low density panels represent good IL performance at low frequencies.

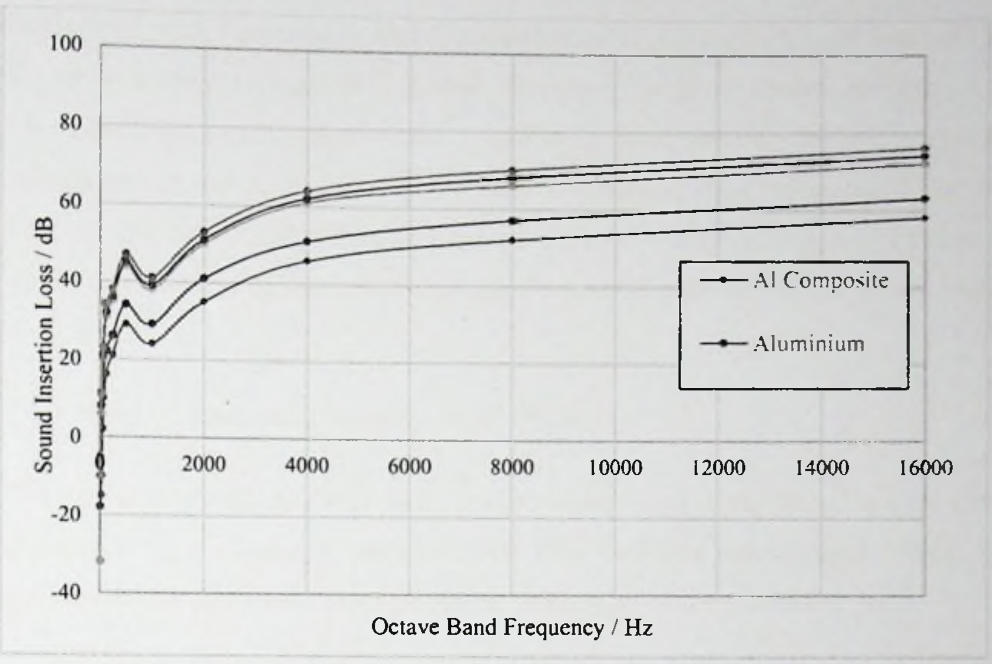

Figure 2.7: Effect of IL due to changes in sheet metal material property.
(C110D5 "Cummins" PG. Sheet metal with 2mm thickness. 50mm thick Rock Wool with the density of 50 kg/m^3 and d = 0.5m)

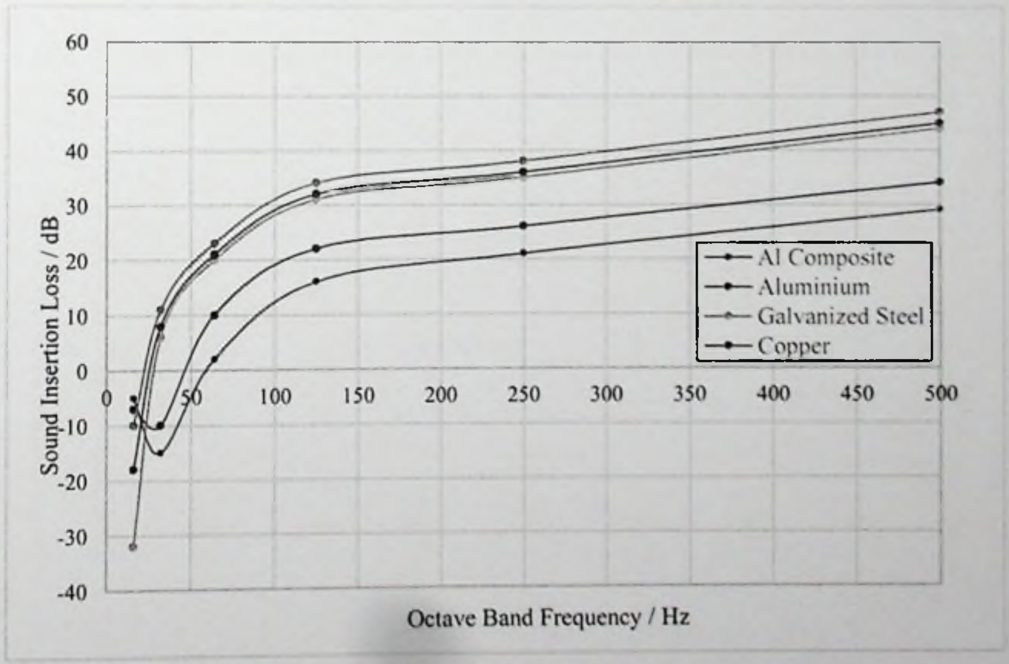

Figure 2.8: Effect of IL due to changes in sheet material properties at low frequency.

Galvanized steel, Copper and Led represent good IL behavior at high frequencies. PG noise consist of high SPL in high frequency range as shown in figure 1.2. Therefore high density sheet metal would be a good solution for PG enclosure fabrication. However IL increment among Galvanized steel, copper and led are insignificant further the copper and led are expensive compared to the Galvanized steel. Galvanized steel was selected as the sheet metal material for the PG enclosure fabrication.

2.3.4 Effect of sheet metal thickness on IL Model

Effect of sheet metal thickness on IL model was analyzed using the galvanized sheet metal with 1mm, 2mm and 3mm thickness. The thickness were limited to 3mm by considering the fabrication feasibility, cost and weight of the material.

Figure 2.9 and figure 2.10 shows the IL behavior with respect to sheet metal thickness. By increasing the panel thickness the amount of IL is increased as shown in figures. This is due to the fact that the IL is controlled by the effective stiffness or Bulk modulus (D) of the panel. The effective stiffness is a function of the Young's Modulus, Poison's Ratio and material thickness as described by the equation (3). Panel thickness is proportionate to Bulk Modulus and this result in increasing IL.

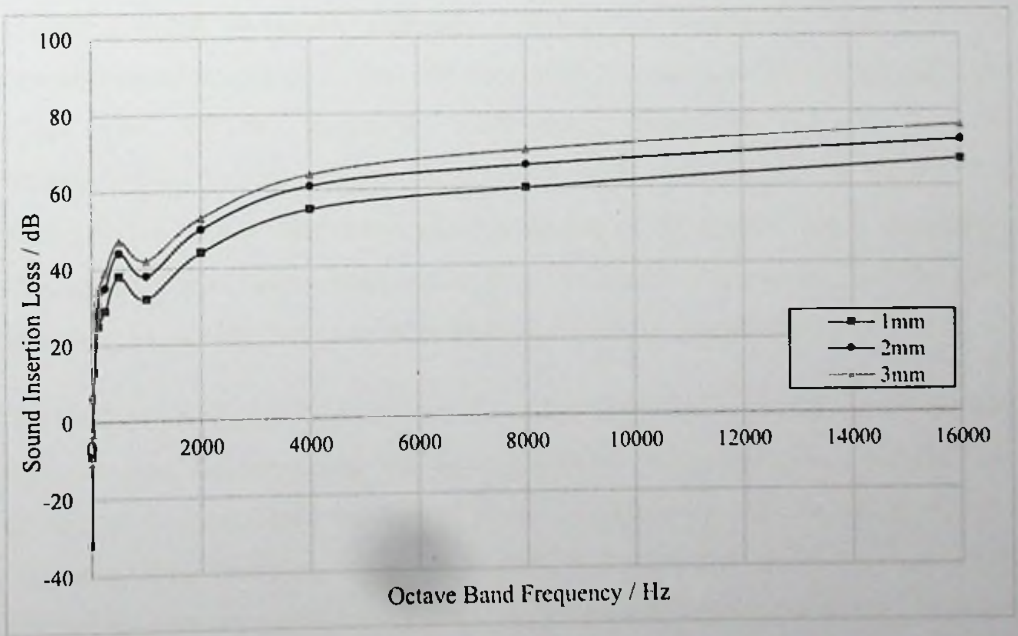

Figure 2.9: Effect of IL due to changes in sheet metal thickness
(C110D5 "Cummins" PG, Galvanized steel, 50mm thickness Rock Wool with the density of 50 kg/m^3 and $d = 0.5m$)

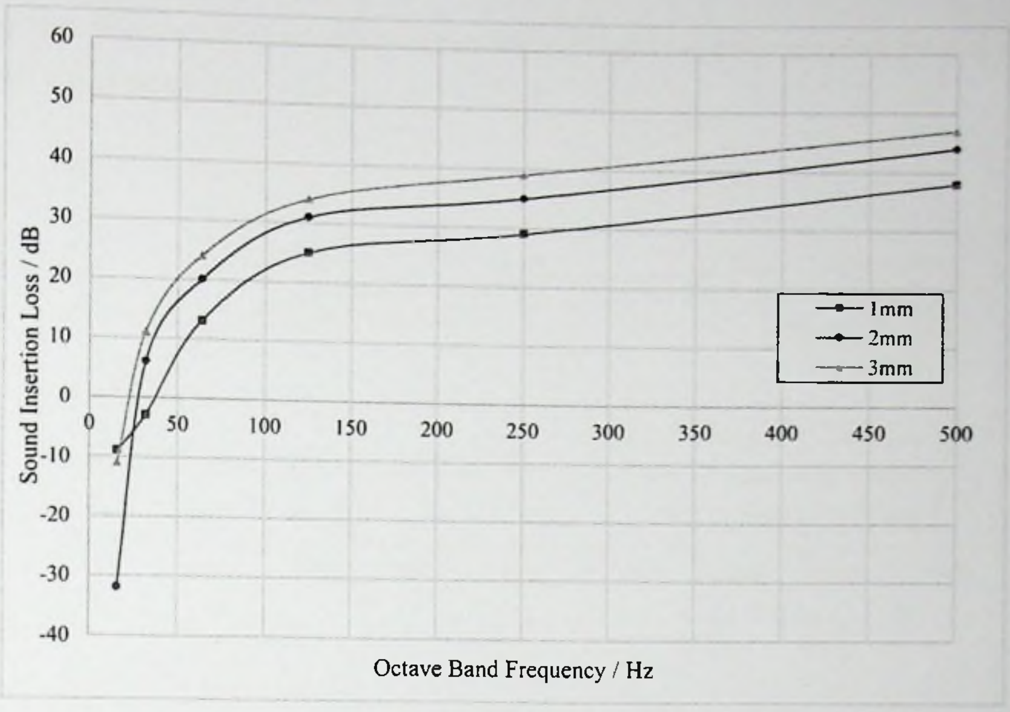

Figure 2.10: Effect of IL due to changes in sheet material thickness at low frequency.

2.3.5 Effect of sound absorption material type on IL model

There are different type of fibrous and open cell foam materials are available in market a sound absorption materials. However the high temperature and oily operation environment of PG limit the choices of that material. Rock wool and Fiber Glass wool are the commonly used sound absorption materials for PG soundproofing. In this analysis Rock wool with densities of 50 kg/m^3 and 80 kg/m^3 and Fiber Glass wool with the density of 50 kg/m^3 were considered. Properties of sound absorption materials are tabulated in Table B.1 and the flow resistivity vs. density plot is given in figure B.1 under Appendix B

Figure 2.11 and figure 2.12 illustrate the effect on IL due to changes in absorption material type. By increasing the flow resistivity of the absorption material the amount of IL is increased throughout broadband frequency. This is due to that the increasing flow resistivity result in friction to transmitted sound pressure wave through the barrier causing energy conversion from pressure to heat energy. Furthest to that face absorption material suppress the standing wave resonances occurs in the air gap of enclosure.

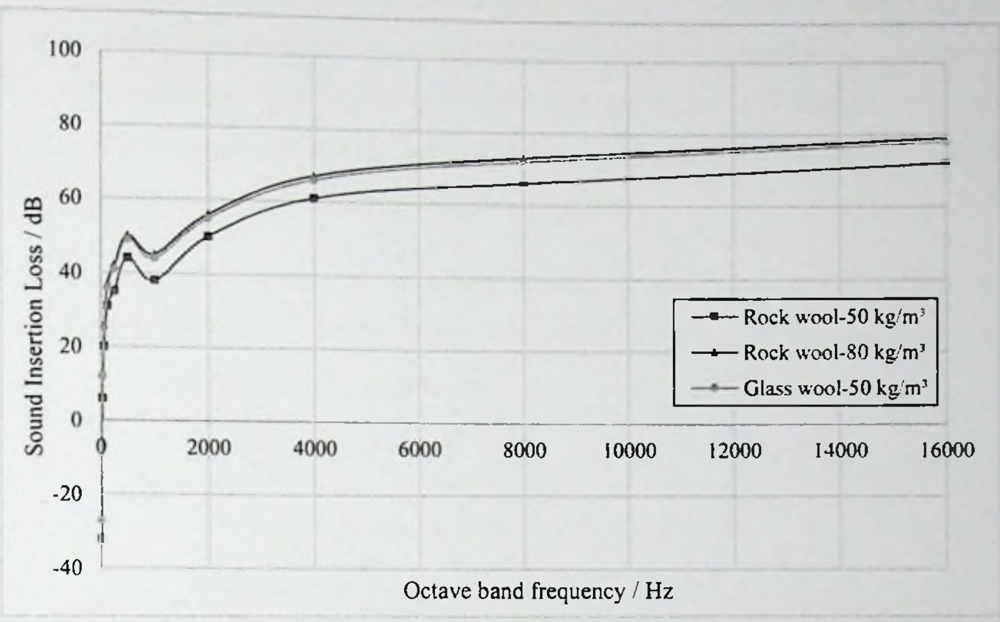

Figure 2.11: Effect of IL due to changes in sound absorption material type (C110D5 "Cummins" PG, Galvanized steel with 2mm thickness. 50mm thickness sound absorption material, $d = 0.5m$)

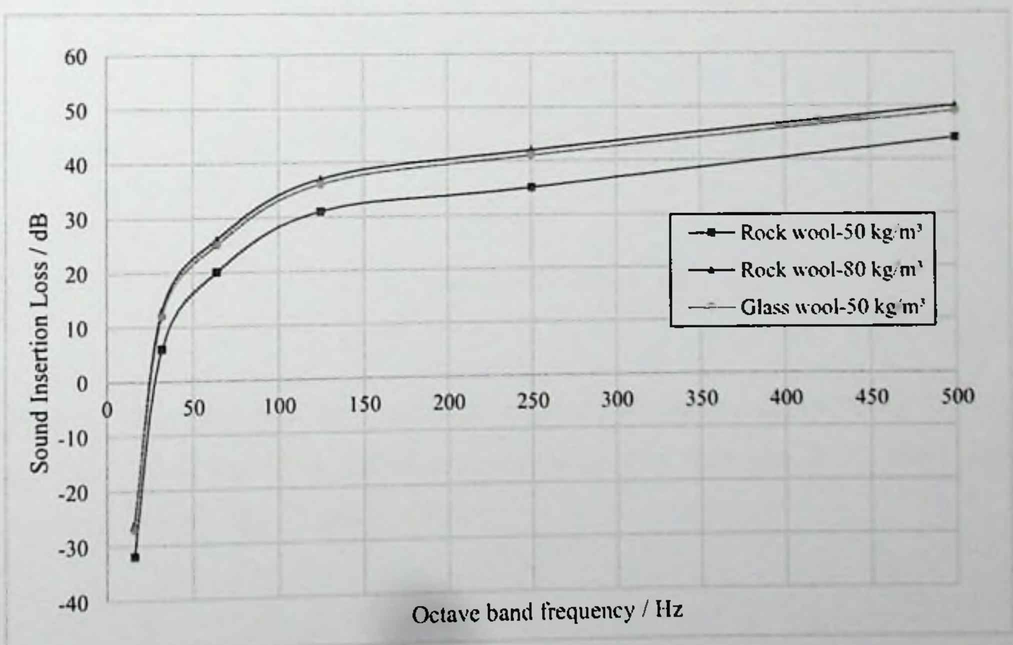

Figure 2.12: Effect of IL due to changes in sound absorption material type at low frequency.

2.3.6 Effect of absorption material thickness on IL model

Sound absorption fiber material are available in 1", 2" and 3" thickness in the market. Those three thickness were selected to observe the effect of sound absorption material thickness on IL of enclosure. Figure 2.13 and figure 2.14 show the effect on IL due to changes in absorption material thickness. As per the plots the IL increase with increment of sound absorption material thickness.

Effect of sound absorption material type and sound absorption material thickness revealed that the Rock wool with 80 $kg/m^3$and Fiber Glass wool with the density of 50 kg/m^3 represent similar IL behavior.

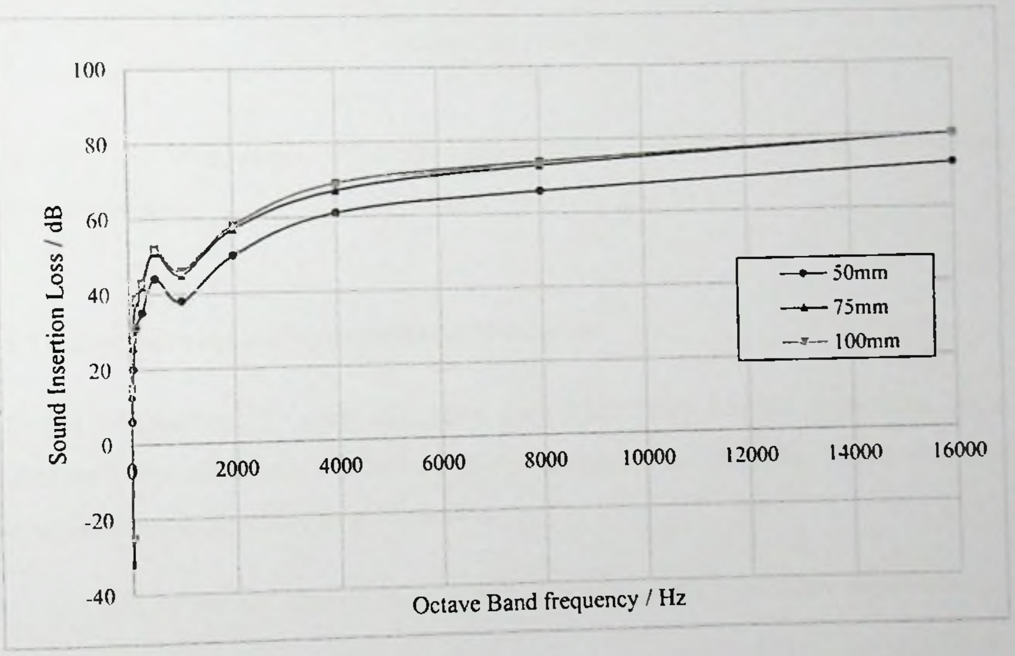

Figure 2.13: Effect of IL due to changes in sound absorption material thickness (C110D5 "Cummins" PG, Galvanized steel with 2mm thickness. Rock Wool with the density of 50 kg/m^3 and $d = 0.5m$)

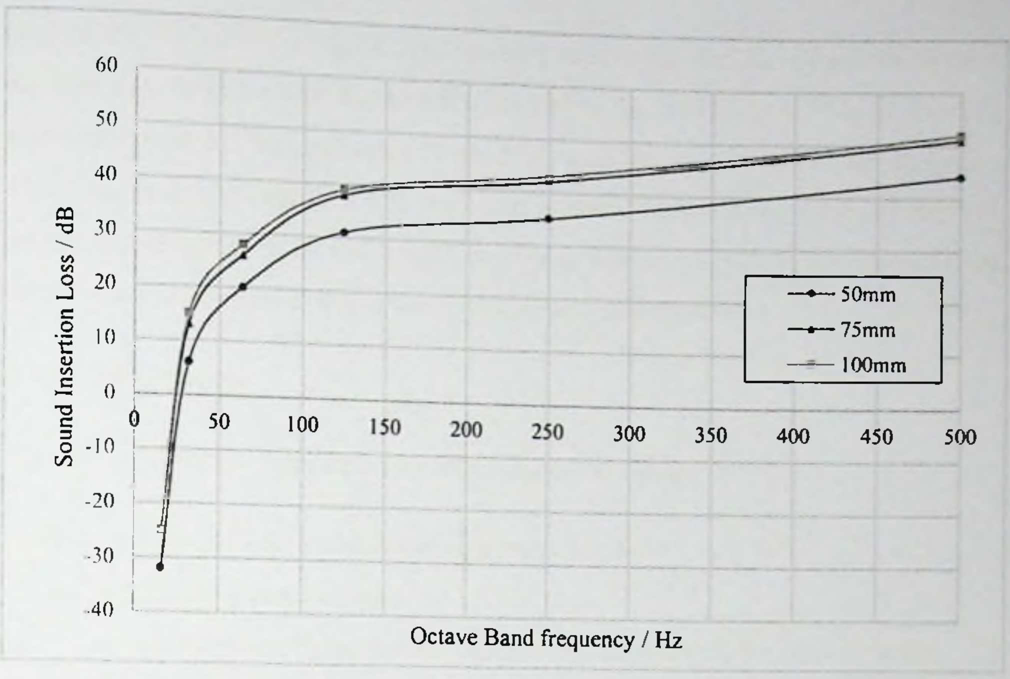

Figure 2.14: Effect of IL due to changes in sound absorption material thickness at low frequency

2.4 Customer required constrains on SPL model

Sound attenuation of open PG have embedded with several constrains as a commercially appealing product at higher customer satisfaction. There are four constrains were identifies as given bellow.

a. SPL of the soundproof PG which is complying with local noise regulation
b. Size
c. Cost
d. Weight

Foremost objective of the design is achieving the customer required SPL which is modeled in equation (3). Therefore SPL of enclosed PG can be addressed in the SPL model as the objective function.

Size of the enclosed PG can be controlled by the design variable of source to panel distance (d). Source panel distance has limited to the maximum of 1m by the SPL model because all the equations were developed for the close fitting enclosure. Therefore it is obvious that SPL and size contain has been imposed on the SPL model automatically. Cost and weight constrains was developed as functions of sheet metal material and sound absorption material type and thickness.

2.4.1 Cost constrain

PG enclosure cost is contributed by following components.

a. Sheet metal and sound absorption material cost.
b. Metal frame material cost.
c. Fabrication cost including the material and sheet metal fabrication work.
d. Labour cost.
e. Overhead cost.

Cost estimations for different sizes of enclosures considering the above components revealed sheet metal and absorption material cost is approximately a 40% of the total cost of enclosure. With this approximation enclosure cost equation was developed as follows.

$$\text{Enclosure Cost} = \frac{S_E(P_S + P_{AB})}{0.4} \tag{14}$$

where,
S_E- Surface are of enclosure m^2
P_S- Unit surface price of sheet metal LKR/m^2
P_{AB}- Unit surface price of sound absorption material LKR/m^2

Under the scope of this research. Sheet metal was selected as the Galvanized steel with the thickness of 1mm, 2mm and 3mm. Absorption material were selected as Rock wool with densities of 50 kg/m^3 and 80 $kg/m^3$and Fiber Glass wool with the density of 50 kg/m^3. Thicknesses of the sound absorption materials were 50mm, 75mm and 100mm.

Equation was developed to calculate P_s and P_{AB} based on the market prices of the selected material. Equations are illustrated in figure 2.15 and figure 2.16.

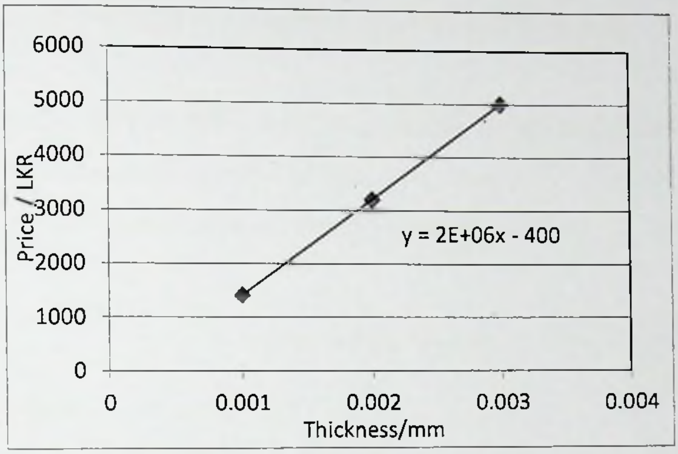

Figure 2.15 Price equation of Galvanized sheet metal

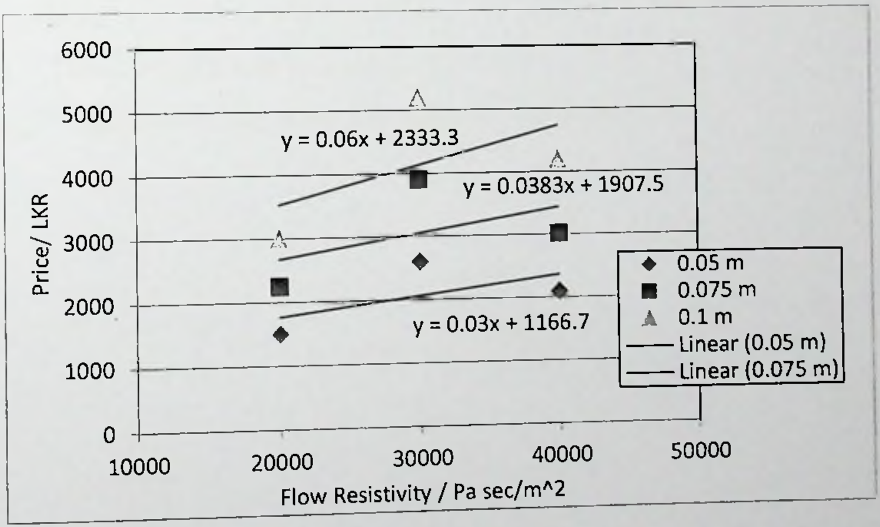

Figure 2.16: Price equation of the absorption material with the 0.05m, 0.075m and 0.1m thickness.

2.4.2. Weight constrain

Weight of a PG enclosure associated with following components

a. Weight of the Sheet metal and sound absorption material.
b. Weight of the metal Frame material.
c. Weight of the fabrication material like fastens.

Estimation on the weight of enclosure revealed that the weight of panel material is approximately a 60% of the total weight of enclosure. With this approximation enclosure cost equation was developed as follows.

$$\text{Enclosure Weigth} = \frac{S_E h \rho_s + S_E t \rho_{ab}}{0.6} \quad (15)$$

where;

S_E- Surface are of enclosure m^2

h - Thickness of sheet metal in m

ρ_s- Density of sheet metal in kg/m^3

t - Thickness of sound absorption metal in m

ρ_{ab}- Density of sound absorption material in in kg/m^3

CHAPTER 3

DESIGN METHODOLOGY

Objective functions and the constrain functions in designing an acoustic close fitting enclosure were developed under the chapter 2. Development of an optimization method to calculate the optimal minimum value for the design variable in order to achieve the required SPL at constrained cost and weight. This calculation can be done by an optimization method.

There are several optimizations methods range from liner programming to more advance mathematical calculus. Selection of suitable optimization by considering the numbers of design variables, complexity of the mathematical model and constrain model is important. In this case determination of optimum result using a manual liner programming theories is complex as well as time consuming due to following reasons.

a. Acoustic behavior of objective function is chaos with different deign variable values
b. There are six design variables and two constrain functions
c. Single step follows for a particular calculation have to be repeat for eleven different octave band frequencies

In order to efficiently do similar optimization many researches have used computer based advanced and fast optimization method like Simulated Annealing method [17], Artificial Immune method [18] etc. In this research Genetic Algorithm (GA) optimization was selected for the optimized numerical assessment.

3.1. Genetic Algorithm (GA) tool for optimization

GA is a computational procedure that mimics the natural process of evolution. This is good as algorithm which consider a larger, potentially huge, search space and navigating them looking for an optimal combination of things and solutions. A more striking difference between the GA and typical optimization method is that, GA uses a population of points at one time in contrast to single point approach by typical optimization method. All individuals in a population evolve simultaneously without central coordination [19]. The major steps of the GA optimization are;

1. Create the initial population / Genotype (i.e. create initial set of solutions to the selected problem)
2. Phenotypic decoding and objective function calculation
3. Ranking and selection
4. Apply genetic operators like crossover, mutation to create offspring
5. Evaluate the objective function
6. Repeat the above procedure until stopping criteria

3.2 Development of GA optimization code using MATLAB

GA tool is the built in Graphical user interface (GUI) to run GA in Matlab. This tool enable us to perform GA optimization without using commend line. Figure 3.1 illustrate the GA optimization GUI used for SPL minimization of enclosed PG.

Fitness function and nonlinear contain function of the SPL minimization problem is given in Appendix C. Galvanized steel was fixed as the sheet metal in the optimization. related upper and lower bounders of design variables were,

a. Sheet metal thickness, h/m : [0.001, 0.003]
b. So Source to panel thickness in length side, d_1/m : [0.1, 1]
c. Source to panel thickness in width side, d_2/m : [0.1, 1]
d. Source to panel thickness in height side, d_3/m : [0.1, 1]
e. Sound absorption material thickness t/m : [0.05, 0.1]
f. Flow resistivity of sound absorption material γ / $kPa.s/m^2$: [20000,400000]

GA optimization can be customized by changing the stopping criteria. Population size, Selection, reproduction, mutation, crossover and other parameter in option menu according to the problem to be solved. There are some critical parameters in GA tool which determines the best optimization results. Those parameters are called run time parameter.

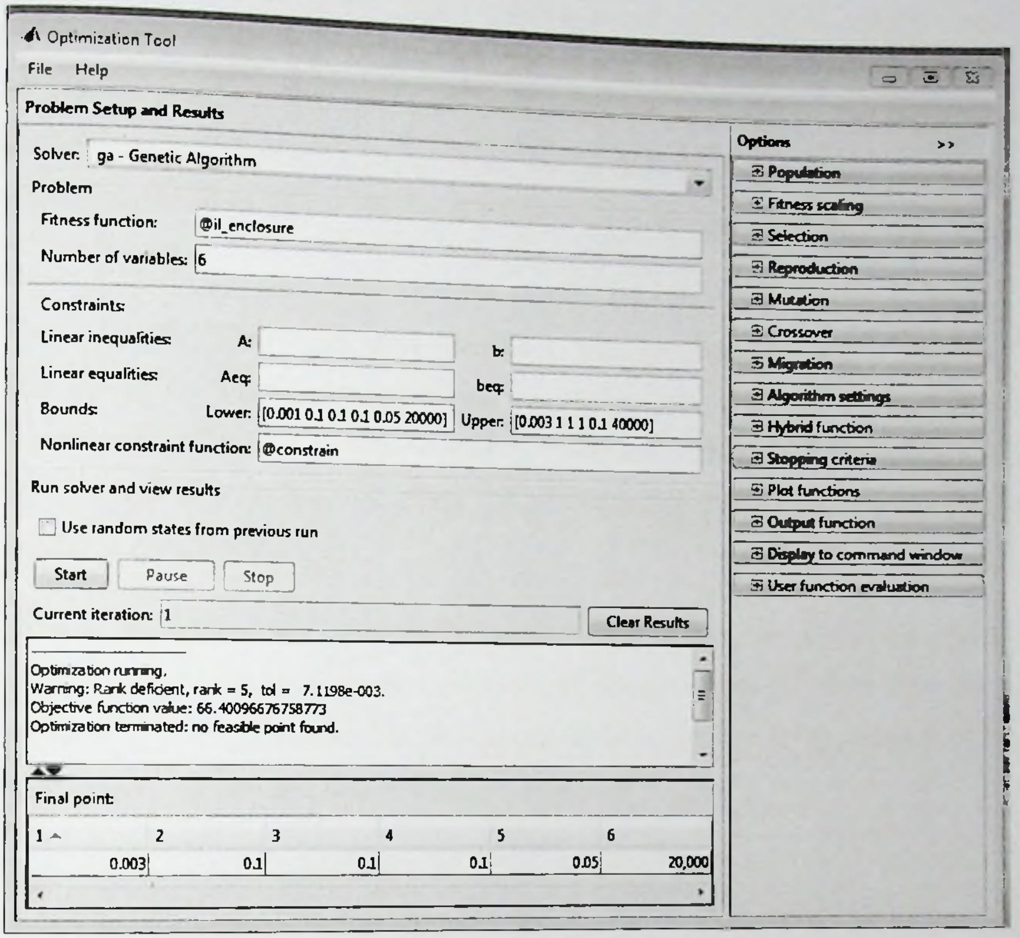

Figure 3.1. GUI interface of GA Toolbox

Selection of values for run time parameter is critical step in the application of GA. Population Size, Mutation rate and the crossover rate are the most important parameter and numerous investigations have discussed parameter set selection, both theoretically and empirically. Schaffer in 1989 [19] has suggested set of best run time parameter and there are some researches have been done based on that parameter [20]. According to their findings following rage were selected

a. Population Size : [6, 20]
a. Mutation Rate : [0.005 0.1]
b. Crossover Rate : [0.65 0.95]

Selection of best set of run time parameter among the above rage was investigated by doing iterative analysis for the for the optimum minimum SPL of "Cummins" PGs Table D.1 in Appendix D illustrate the results for different GA parameter combination. According to the data Population rate, Crossover rate and mutation rate combination of [20, 0.65. 0.05] gives the lowest SPL for all PG model analyzed. Therefore this selection was taken as GA parameter for the optimization.

Cost and weight constrains were imposed to the optimization which control the direction of the optimization search and stop the iteration process of optimization when the results reached to specific constrain irrespective of others. This implies that the values of contains assigned on the objective function shall be realistic and shall be matched each other.

As a set objective of development of simple and fast acoustic design procedure, GUI was developed as user application to obtain the optimized design values. The GUI was developed using Matlab and the relevant m files are given in Appendix E (CD rom). Figure 3.2 illustrate the relevant GUI interface.

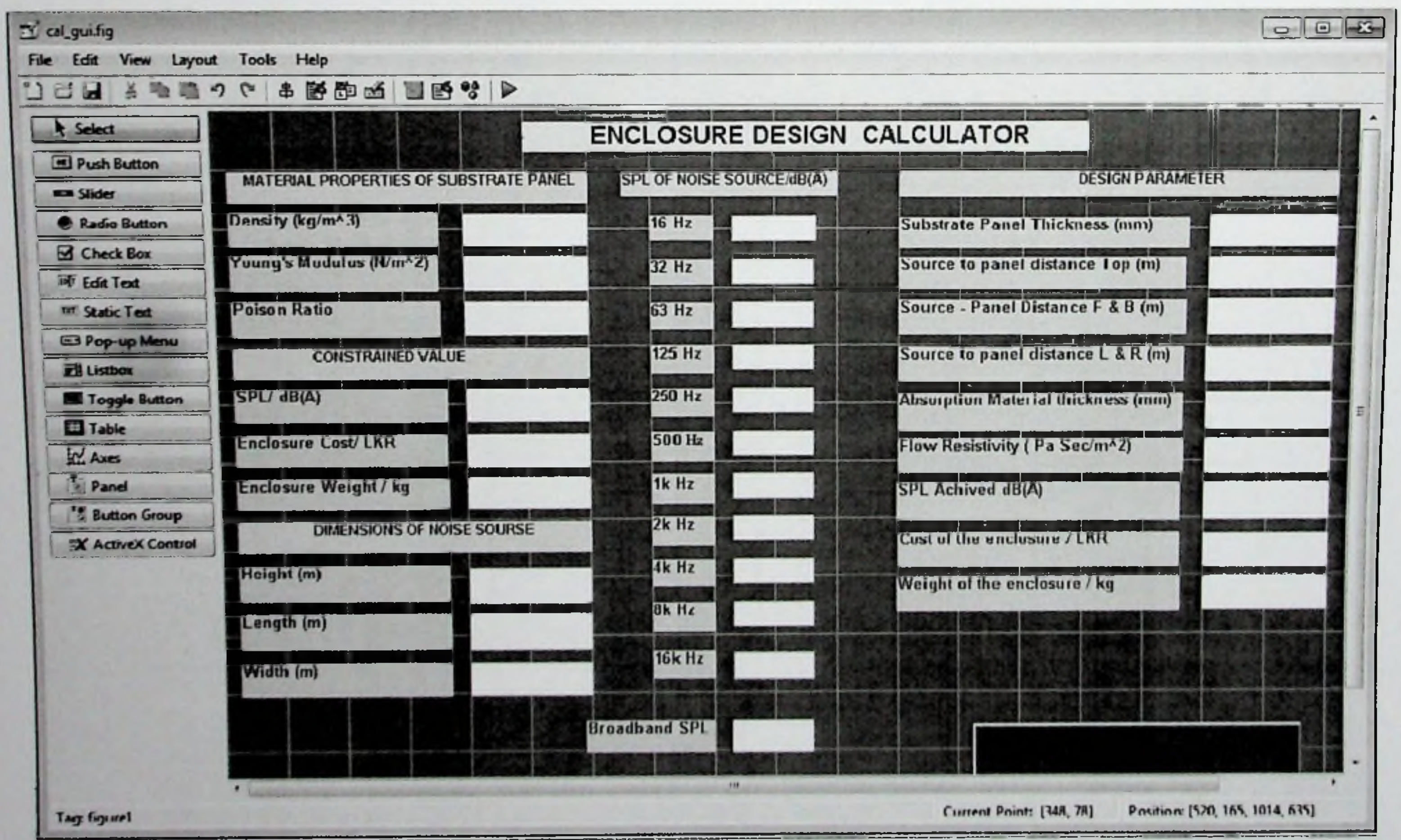

Figure 3.2: Optimization GUI developed using MATLAB

3.3 Design optimization steps of acoustic close fitting enclosure using developed method

Close-fitting enclosure deign steps using the developed optimization method is described by the flow chart given under the figure 3.3.

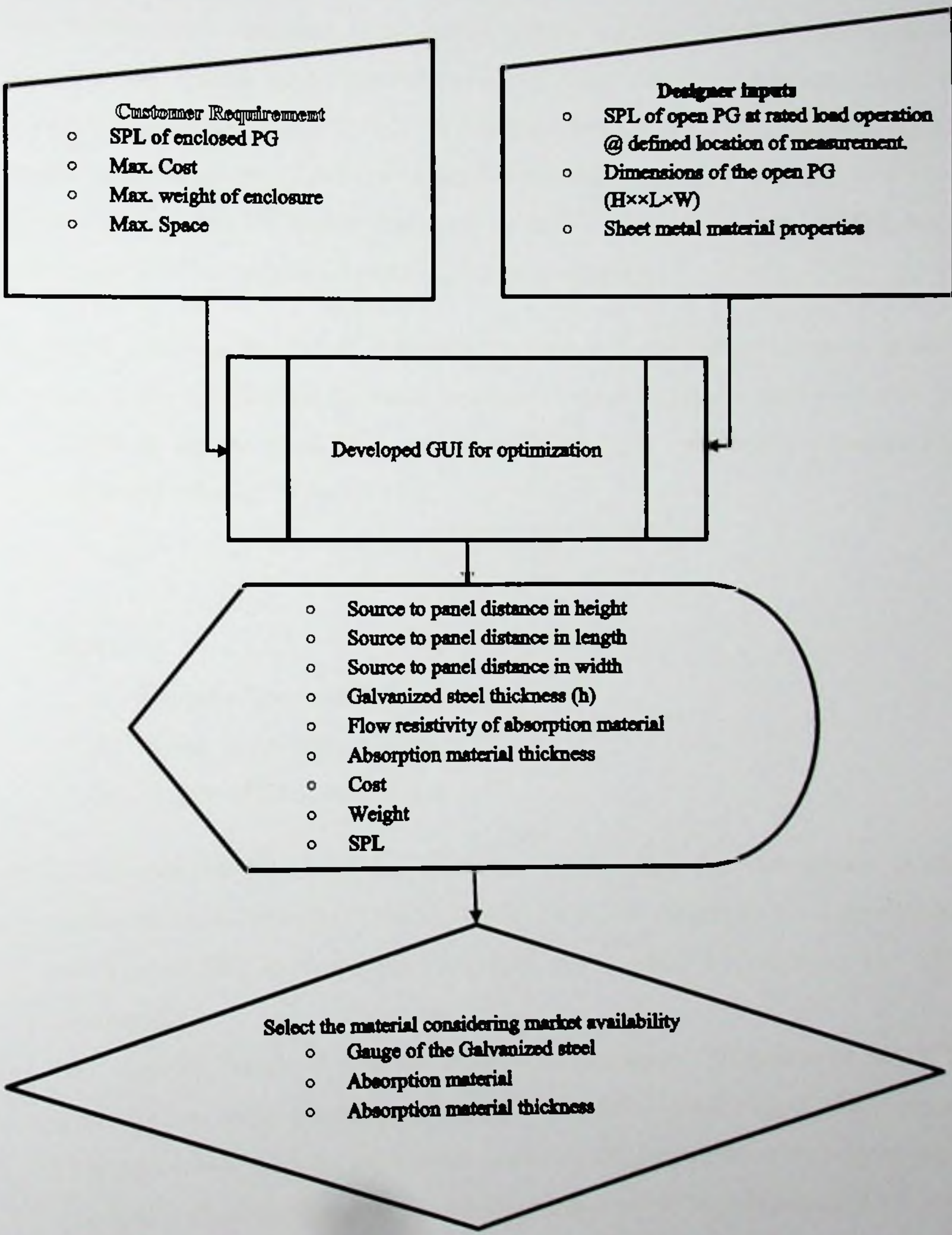

Figure 3.3: Flow chart of enclosure methodology

There are several steps to be followed when taking noise measurements of open PG. Foremost the measurement shall be taken with the silencer of the PG. Because the silencer separate the internal combustion exhaust noise from the other noises which do not considered in enclosure deign.

The International Standard Organization (ISO) outlines detailed procedure to calculate the overall sound pressure level of noise source which are placed in anechoic, reverberant and free field. In this test condition the sound source is placed in free field and the ISO 3744 can be used as the SPL measuring standard. ISO 3744 gives measurements for engineering grade accuracy. Under these standard following conditions shall be maintained when taking measurements.

a. Point of measuring shall be determined according to the size of the noise source. The point of measurement shall be located equal or greater than the twice of minimum dimension d_0 and not less than to 1m. Minimum dimension is calculated using the equation (16).

$$d_0 = \sqrt{\left(\frac{L}{2}\right)^2 + \left(\frac{W}{2}\right)^2 + (H)^2} \tag{16}$$

where,

L – Length of the open PG in m

W- Width of the open PG in m

H – Height of the open PG in m

b. Noise shall be measured at 8 locations by locating the microphone at the points defined under ISO3744 (Refer figure F.1 in Appendix F). Logarithmic average of SPL measurements at eight points shall be taken as the SPL measured.

c. SPL shall be measured with the facility of frequency analysis to obtain SPL over octave band frequency. MATLAB based virtual sound level meter (VSLM v0.41) [20] was used to measure the SPL spectrum over octave band frequency. Calibrated digital sound level meter with "A weighted" filter was used to measure the overall SPL and for calibration purpose of VSLM v0.41. GUI of VSLM v0.41 is given as figure F.2 in Appendix F.

CHAPTER 4

RESULTS AND DISCUSSION

Validity of the developed design methodology was done under two steps and those validation procedures will be discussed under the following two sections.

4.1 Comparison of model predicted data with manufacturer given data for commercially available PGs.

Five numbers of commercially available PG under the make of "Cummins" were selected for the comparison of the commercially available product data with the predicated data by the developed design methodology. All of the investigations and testing done in this research were based on " Cummins" PGs since there are only the PG manufacturer who is maintaining e-format technical library with SPL data over the octave band frequency. C110D5, C220D5, C330D5, C440D5 and C550D5 are the "Cummins" PGs models selected for this discussion. Manufacturer given data on SPL spectrum of the open PGs, the dimensions of the open and enclosed PGs, and the overall SPL of the enclosed PGs are given under the table A-1, table.A-2 and table A-3 respectively.

The data were predicted using the developed model for the above mentioned five numbers of PGs under three conditions as follows.

a. Minimum overall SPL which can be achieved using the selected design limits within the model.
b. Constrains of the source to panel distance which is similar to the enclosure size of the commercially available PGs.
c. Constrain of overall SPL of the soundproof medal which is similar to the overall SPL of commercially available soundproof PGs.

Predicted design values and the selected design values based on the dimensions and the properties of the material available in the market are tabulated in Appendix G. Cost, weight and the overall SPL of the commercially available PGs and Predicted modals were tabulated in table 4.1, table 4.2, table 4-3 and table 4-4 respectively.

Table 4.1: Overall SPL, cost & weight of commercially available PG

PG Model	Commercially available PG enclosure		
	Overall SPL/ dB(A)	Enclosure Cost/LKR	Enclosure weight / kg
C110D5	71	423,000.00	669
C220D5	68	762,000.00	1536
C330D5	67	780,000.00	1577
C440D5	66	1,090,000.00	1412
C550D5	66	1,250,000.00	1458

Table 4.2: Predicted overall SPL, cost & weight of minimum overall SPL modal

Genset Model	Predicted Data		
	Overall SPL/ dB(A)	Enclosure Cost/LKR	Enclosure weight / kg
C110D5	52	1,201,300.00	2481
C220D5	37	1,041,700.00	2152
C330D5	39	1,417,500.00	2928
C440D5	46	1,531,300.00	3163
C550D5	38	1,733,200.00	3580

Table 4.3: Predicted overall SPL, cost & weight of source panel distance constrained modal

Genset Model	Predicted Data		
	Overall SPL/ dB(A)	Enclosure Cost/LKR	Enclosure weight / kg
C110D5	65	423,960.00	876
C220D5	40	611,030.00	1262
C330D5	44	735,970.00	1520
C440D5	49	952,090.00	1966
C550D5	56	973,650.00	2011

Table 4.4: Predicted overall SPL, cost & weight of overall SPL constrained model

Genset Model	Predicted Data		
	Overall SPL/ dB(A)	Enclosure Cost/LKR	Enclosure weight / kg
C110D5	71	293,100.00	679
C220D5	68	450,850.00	705
C330D5	67	598,740.00	908
C440D5	66	703,800.00	1431
C550D5	66	837,740.00	1222

According to data given under the table 4.2 minimum overall SPL achieved by the model are too low. All of the models have achieved minimum overall SPL which is lower than the below limit of local noise regulation (45 dB (A)) excluding the C110D5 model. According to the table 4.2, table 4.3 and table 4.4 noise performance of C110D5 model is low. This because open PG of C110D5, emit higher SPL noises at lower frequencies below 200Hz. The enclosures are passive noise controller which is poor in reducing SPL at lower frequencies.

Compared to table 4.1 and table 4.3, commercial product and the predicted model have similar enclosure size. But the overall SPL of commercial product is higher than the predicted model. This is because the acoustic model was developed by assuming zero noise leaks through the opening. But in actual situation ventilation opening leaks considerable amount of noises which causes to retarded the acoustic performance of the sealed enclosure. Cost and the weight of the predicted model are lower than the commercial model. However the cost and weight of the commercial models reflected both the enclosure and sound attenuation muffler at exhaust and inlet air opening. Therefore the cost and weight gap between the predicted data and commercial data have to compensate accordingly.

4.2 Construction of design variable based enclosure for predicted overall SPL validation

Validity of the results given by the developed design methodology was found by doing a noise control design for an selected open type "Cummins" PG with the capacity of 22kVA.

Dimensions of the unenclosed PG was Length(L) $= 1.667$m, width $(W) = 0.930m$ and Hight $(H) = 1.13\ m$. Using the equation (16) minimum radius was calculated.

$$d_0 = \sqrt{\left(\frac{1.667}{2}\right)^2 + \left(\frac{0.930}{2}\right)^2 + 1.13^2}$$

Subsequently minimum radious was 1.47 m. Then the 3m was selected as the noise measuring surface according to the ISO 3744 standard. The noise of the open PG was measured and the overall SPL was 99 dB(A). For this investigation maximum funds availability was LKR 120,000.00. With this cost constrain of LKR 120,000.00 the developed optimization gives the design variable as [0.001, 0.5, 0.1, 0.1, 20000, 0.05] and the estimated cost of the enclosure was LKR 116,950.00 and estimated weight of the enclosure was 200kg. With this optimization results enclosure was fabricated and the noise measurements were taken. The SPL spectra of unenclosed PG, predicted and the measured SPL of enclosed PG are illustrated in figure 4.1.

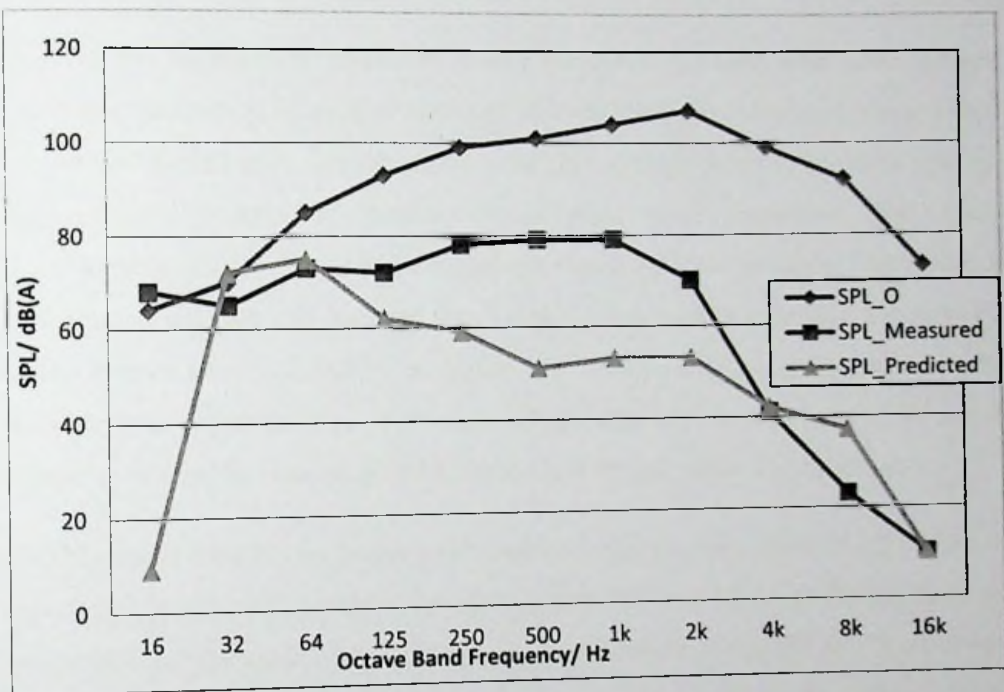

Figure 4.1: SPL spectra of unenclosed PG and predicted and measured SPL of enclosed PG

Predicted overall SPL was 77 dB (A) and the measured overall SPL was 84 dB (A) which was 7dB (A) higher than the predicted data.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

Noise generation due to engine air interaction is predominant in PG operation. This air borne noise consist of inlet and exhaust air in combustion process as well as the inlet and exhaust air in ventilation or cooling processes. The combustion air exhaust is silenced by the passive silencers and resulted the overall noise radiation to the receiver is obstructed by the enclosure.

Typical PG enclosure is fabricated with sheet metal material with inner surface insulation with sound absorption material. The best sheet metal material was selected as the Galvanized steel and the sound insulation material were selected as mineral type fibrous material of Rockwool and glass wool. Economic and space consideration usually dictate that the enclosure be as small as possible. This makes a close-fitting enclosure. It has been found that it is impossible to predict the acoustic performance of close-fitting enclosure by using technically well-developed architectural acoustics. This is because of the structural resonance due to panel vibration and cavity resonance within the narrow air gap within the enclosure.

A SPL model for a PG enclosure was developed with the help close-fitting acoustics developed by researches to date. The developed model consists of enclosure material properties and thickness and the source to panel distance. Once the SPL model was realized, the optimization program was developed using the GA optimization tool in MATLAB. Design of a PG enclosure is not minimization of SPL. It is the requirement of achieving SPL governed by the local ordinance with one for few the customized constrains. Constrains behind soundproof PG can be identified as cost, space and weight. With those constrains, the direction of the optimization process can be controlled.

The developed design methodology was validated by designing a close-fitting enclosure for 22kVA "Cummins" PG at 75% loading condition. In this project cost constrain was introduced since funding was limited to LKR120, 000.00. The overall SPL of open PG at 3m distance was 99 dB (A). The optimization process gives the

77dB (A) as minimum optimum SPL which can be achieved under the cost contain. With given results the enclosure was fabricated and the SPL measurement was taken. According to the results measured overall SPL was 84 dB (A) which is 7dB (A) difference form predicted value.

This difference is produced by the opening of the enclosure. The percentage of noise leaks depends on the size and the orientation of the openings. The size of the opening depends on volume of ventilation air required and the orientation depends on location of installation. So a noise leak compensation factor shall be imposed for the model to reduce the result gap between the actual and predicted SPL. Further the noise leaks trough the structure vibration was approximated as zero. But the mechanical vibration damping of selected PG was in good condition and portions of structural vibration loaded the SPL.

5.1 Future work and consideration

This research is the initiation of designing for a close-fitting PG enclosure. The validation of design methodology by actual implementation revealed the research gap. This gap can be reduced by treating to the causes which makes that gape. Foremost investigation shall be focused on the noise leaks trough the opening for air ventilation. More acoustical study should be done through a research to compensate the reduction of acoustic performance of enclosure trough the openings.

To date most of the researches have focused on PNC for the PG soundproofing. However a hybrid noise controlling method can be investigated for controlling of SPL since the PNC works best for high frequency pressure waves. The exhaust and inlet air noise of PG associated with low frequency pressure waves which are unable to mitigate by PNC successfully. As a solution, the noise in ventilation air opening paths can be controlled by hybrid noise controlling. Firstly, air ventilation path can be designed with duct which will be provided passive noise controlling. Secondly, the uncontrolled low frequency pressure waves can be controlled by introducing an Active Noise controller to the duct.

The design variables obtained by the developed methodology can be used to do the preliminary design of close-fitting enclosure. With the preliminary design the ultimate performance can be improved by introducing the sound attenuators to the inlet and exhaust openings. At last but not least I conclude that this research opened a path to local PG enclosure fabrication industry to do their design with a background of acoustic design rather than duplicating an existing enclosure.

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APPENDIX A

MANUFACTURER DATA OF COMMERCIALLY AVAILABLE DIESEL PGs

Table A.1: SPL spectra of "Cummins" unenclosed PGs

Frequency	Sound Pressure Level at 1m 110% load							Sound Pressure Level at 3m 110% load						
	C22D5	C55D5	C110D5	C220D5	C330D5	C440D5	C550D5	C22D5	C55D5	C110D5	C220D5	C330D5	C440D5	C550D5
16	-	-	69	-	-	65	-	0	0	59	0	0	55	0
32	56	44	81	-	-	73	55	47	34	71	0	0	63	45
63	71	60	96	65	77	73	81	61	51	87	55	67	64	71
125	79	73	97	81	93	83	88	70	64	88	71	83	74	78
250	83	77	95	89	93	85	94	73	68	86	80	84	75	84
500	83	85	97	97	93	96	94	73	76	87	88	83	86	85
1000	83	85	96	100	93	97	99	74	76	87	91	83	88	89
2000	88	86	95	102	96	99	98	78	77	86	92	86	89	89
4000	88	83	95	101	96	99	94	78	74	85	92	86	89	84
8000	86	77	93	98	95	104	91	76	68	83	89	85	95	82
16000	78	71	102	99	86	96	82	69	61	93	90	77	87	72
Overall	94	92	107	108	103	107	104	84	82	97	98	93	98	94

Table A.2: Dimensions of "Cummins" open and packaged standard type enclosed PG

#	PG Model	Open PG Dimension (mm)			Soundproof PG Dimensions (mm)			Source to Panel distance (mm)		
		Length	Width	Height	Length	Width	Height	Length	Width	Height
1	C110D5	2268	1094	1576	3151	1142	1714	441.5	24	138
2	C220D5	2656	1100	1658	3900	1100	2062	622	0	404
3	C330D5	3135	1100	1928	4254	1424	2215	559.5	162	287
4	C440D5	3549	1100	2115	5110	1563	2447	780.5	231.5	332
5	C550D5	3433	1500	2065	5110	1563	2447	838.5	31.5	382

Table A.3: Overall SPL of "Cummins" packaged standard type enclosed PG

#	PG Model	SPL of soundproof PG /(dB(A)	
		At 1m distance	At 3m distance
1	C8D5	69	59
2	C11D5	69	59
3	C22D5	75	65
4	C33D5	75	65
5	C55D5	77	67
6	C110D5	78	68
7	C220D5	77	67
8	C330D5	77	67
9	C440D5	76	66
10	C550D5	76	66

APPENDIX B

ENCLOSURE MATERIAL PROPERTIES

Table B.1: Material properties of sheet metal material

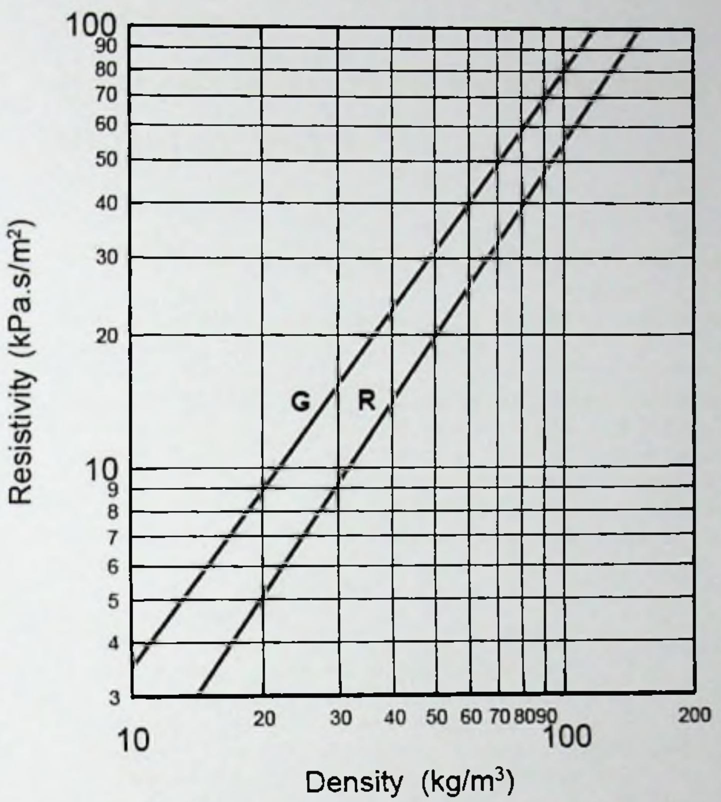
Material	Density (kg/m^3)	Young's Modulus (N/m^2)	Poisson Ratio
Aluminium Composite	1400	7×10^{10}	0.3
Aluminium	2700	7.16×10^{10}	0.34
Steel	7700	1.96×10^{11}	0.31
Copper	8900	1.30×10^{10}	0.35
Led	11000	1.58×10^{10}	0.43

Table B.2 Material properties of material use for passive noise barriers

Material	**Density kg/m^3**	**Modulus of elasticity 10^9 Pa**	**Poisson's ratio**	**Loss factor (internal) 10^{-3}**
Metal, glass etc.				
Aluminium	2700	66–72	0.33–034	~ 0.1
Copper	8900	110–120	0.35–0.36	~ 0.2
Magnesium	1750	42–45	~ 0.35	
Steel	7700–7800	190–210	0.28–0.31	~ 0.1
Glass	2300–2600	50–65	-	0.6–2.0
Plexiglas	1150	3.8		2–4
Concrete	2300	32–40	0.15–0.2	4–8
Concrete (reinforced)	2400	33–45	0.15–0.2	10–50
Concrete (lightweight aggregate)	1300	3.8	~ 0.2	10–20
Concrete (autoclaved aerated)	400–600	1.0–2.5	~ 0.2	10–20
Panel materials				
Plywood (fir, spruce)	500–600	8–10		10–30
Plywood (birch)	650–700	9–10		10–30
Fibre board (pressed, 5 – 10 mm)	700–950	2–4		10–30
Gypsum board (9 – 13 mm)	800–900	4.1	~ 0.3	10–15
Wooden chipboard	650–800	3.8	~ 0.2	10–30
Mineral wool				
Rock wool [1)]	110–135	0.00025–0.00030 [2)]		
Glass wool [1)]	~ 125	0.00011–0.00013 [2)]		
Plastic materials:				
PVC (hard)	1380–1550	2–3		20–40
Polystyrene	980–1110	1.5–3.9		
Polystyrene (expanded)	10–20	0.0003–0.003		
Polyurethane (foam)	33–72	0.007–0.019	~ 0.4	

[1)] High density types intended for vibration isolation, sandwich element etc. [2)] At 2 kPa static load

Figure B.1: Flow resistivity for mineral wool of type rock wool and glass wool



APPENDIX C

MATLAB CODES (.M FILES) DEVELOPED FOR THE DESIGN METHODOLOGY

D.1 MATLAB .m file of SPL calculation of PG with sealed enclosure

```
function SPL=il_enclosure(x)     %vector input x= [ h d1 d2 d3 t r]
% SPL model of sealed Enclosure
% Modal Variables
    x(1)=0.002;      %  Sheet metal Panel Thickness in m
    x(2)=0.5;        %  Source to Panel Distance Length side
    x(3)=0.5;        %  Source to Panel Distance Width side
    x(4)=0.5;        %  Source to Panel Distance Height side
    x(5)=0.1;        %  Sound absorption material thickness
    x(6)=40000;      %  Flow resistivity of sound absorption material
%.....................................................................

% SPL of open PG
    SPL_O=[69 81 96 97 95 97 96 95 95 93 102];
% Octave band frequency spectrum
    freq=[16 32 63 125 250 500 1000 2000 4000 8000 16000];
    omega = 2.*pi.*freq; %rad/sec
% Properties of fluid media between panel and source (air)
    rho_c=1.2;            % Density of air
    c_air=343;            % Speed of sound wave in air, m/s
    k=omega./c_air;       % Acoustic wave number (rad/m).
% Material properties of sheet metal panel (Galvanized Steel)
    rho=7700;        % Density
    E=1.96e11;       % Young's Modulus
    mu=.31;          % Poison Ratio
% Dimensions open PG
    a=2.065;          % Height of the Open Genset in m
    b=3.433;          % Length of the Open Genset in m
    c=1.500;          % Width of the Open  Genset in m
%a1,b1,c1 are hight, length and width of enclosure respectively
    a1=a+(x(4));
    b1=b+(2*(x(2)));
    c1=c+(2*x(3));
%Bulk modulus of the face material
    D1=(E*(x(1)^3))/(12*(1-mu^2));
% Absorption coefficient
    C=x(6)./(rho_c.*freq);
    k0= (2*pi.*freq)./c_air;
% Normalized Characteristic impedence
    Zc=((1+(0.0571.*C.^0.754)))-1i.*(0.087.*C.^0.732);
% Propergation Constant
    R=(k0.*(0.189.*C.^0.595))-1i.*k0.*(1+(0.978.*C.^0.7));
    Z=Zc.*coth(R.*x(5));
```

```
    A=(1-((real((Z-1)./(Z+1))).^2));        % Absorption coefficient

  % IL of top panel
    Ab1=((b1*c1).*A)/(1-A);
    kd1=k.*x(4); %rad
    K1 = 1.35 ./ (3.86 * D1 * ((129.6/b1^4) + (78.4/(b1^2 * c1^2)) +...
        (129.6/c1^4)) - (omega .^2 *rho * x(1)));
    IL_top = 10 .* log10(Ab1.*(((cos(kd1) +...
        (pi^2 ./ (4 .* K1 .* omega .* rho_c.*c_air)) .*
sin(kd1)).^2)));
  % IL of Length side
    Ab2=((b1*a1).*A)/(1-A);
    kd2=k.*x(2); %rad
    K2 = 1.35 ./ (3.86 * D1 * ((129.6/b1^4) + (78.4/(b1^2 * a1^2)) +...
        (129.6/a1^4)) - (omega .^2 *rho* x(1)));
    IL_lgt = 10 .* log10(Ab2.*(((cos(kd2) +...
        (pi^2 ./ (4 .* K2 .* omega .* rho_c.*c_air)) .*
sin(kd2)).^2)));
  % IL of Width side
    Ab3=((c1*a1).*A)/(1-A);
    kd3=k.*x(3); %rad
    K3 = 1.35 ./ (3.86 * D1 * ((129.6/c1^4) + (78.4/(c1^2 * a1^2)) +...
        (129.6/a1^4)) - (omega .^2 *rho * x(1)));
    IL_width = 10 .* log10(Ab3.*(((cos(kd3) +...
        (pi^2 ./ (4 .* K3 .* omega .* rho_c.*c_air)) .*
sin(kd3)).^2)));
  % Calculation of IL over the band width
    for this =1:length(freq)
        sum(this)=10^(IL_top(this)/10)+ 10^(IL_lgt(this)/10)+...

10^(IL_width(this)/10)+10^(IL_lgt(this)/10)+10^(IL_width(this)/10);
    end
    % Average the IL for five sides of the panel
    for this=1:length(freq)
        il_band_total(this)=10*log10(sum(this)/5);
    end
    IL= real(il_band_total);
    SPL=10*log10(10^((SPL_O(1)-IL(1))/10)+10^((SPL_O(2)-IL(2))/10)+...
        10^((SPL_O(3)-IL(3))/10)+ 10^((SPL_O(4)-IL(4))/10)+...
        10^((SPL_O(5)-IL(5))/10)+10^((SPL_O(6)-IL(6))/10)+...
        10^((SPL_O(7)-IL(7))/10)+10^((SPL_O(8)-IL(8))/10)+...
        10^((SPL_O(9)-IL(9))/10)+10^((SPL_O(10)-IL(10))/10)+...
        10^((SPL_O(11)-IL(11))/10));
  end
```

D.1 MATLAB .m file of cost and weight constrain

```
function [C,ceq]=constrain(x)

% GA optimized design variables are x(1),x(2),x(3),x(4),x(5), x(6),
%     x(1 Sheet metal Panel Thickness in m
%     x(2 Source to Panel Distance Length side
%     x(3)Source to Panel Distance Width side
%     x(4)Source to Panel Distance Height side
%     x(5)Sound absorption material thickness
%     x(6)Flow resistivity of sound absorption material
%..........Dimensions of the open PG................................
     a=2.065;         % Height of the Open Genset in m
     b=3.443;         % Length of the Open Genset in m
     c=1.500;         % Width of the Open Genset in m
%...........Dimensions of enclosed PG................................
     a1=a+(x(2));
     b1=b+(2*(x(3)));
     c1=c+(2*x(4));

%...................Cost of the enclosure...........................
      Pa=((2e6)*x(1))-400;        % Price of the Barrier Material LKR/m^2
if x(5)<=0.05
            Pb=(0.03*x(6))+1166.70;
         elseif 0.075>=x(5)>0.05
            Pb=(0.0383*x(6))+1907.50;
        else
            Pb=(0.06*x(6))+2333.30;
end
      AE = ((2*(a1*b1)+2*(a1*c1)+(b1*c1))); % Panel Area;
      Ct = (AE*(Pa+Pb))/0.4                 % cost of the Enclosure

%...................Weight of the enclosure.........................


      da=7700;
if x(6)<=20000
            db=50;
      elseif 30000>=x(6)>20000
            db=80;
         else
            db=50;
end
W = ((AE*x(1)*da)+(AE*x(5)*db))/0.6 % Weight of the Enclosure
C = [Ct-150000; W-500];
ceq=[];
```

APPENDIX D

GENETIC ALGORITHM OPTIMIZATION PARAMETER SELECTION

Table D.1: Optimal SPL results with respect to various GA parameters

#	GA Parameter			SPL for different PG					No of minimum
	ps	cr	mr	C110D5	C220D5	C330D5	C440D5	C550D5	
1	6	0.65	0.0050	41	25	29	38	32	0
2	6	0.65	0.0100	41	25	30	29	31	2
3	6	0.65	0.0500	40	23	26	35	31	3
4	6	0.65	0.1000	39	24	27	31	31	2
5	6	0.75	0.0050	43	24	30	31	33	0
6	6	0.75	0.0100	42	28	29	31	33	0
7	6	0.75	0.0500	42	23	28	30	33	1
8	6	0.75	0.1000	39	24	27	29	30	3
9	6	0.85	0.0050	41	26	27	37	35	0
10	6	0.85	0.0100	42	26	26	33	34	1
11	6	0.85	0.0500	38	22	26	31	31	3
12	6	0.85	0.1000	39	23	26	32	29	1
13	6	0.95	0.0050	45	24	28	32	33	0
14	6	0.95	0.0100	42	24	28	32	34	0
15	6	0.95	0.0500	43	25	31	35	31	0
16	6	0.95	0.1000	45	23	26	33	35	1
17	10	0.65	0.0500	38	22	27	29	30	4
18	10	0.85	0.0500	38	24	26	30	31	2
19	10	0.75	0.1000	40	22	27	29	29	2
20	20	0.65	0.0500	38	22	26	29	24	5

APPENDIX E

Matlab .m file of GA optimization design calculator developed as a GUI (CD Rom)

APPENDIX E

NOISE MEASURING STANDARDS

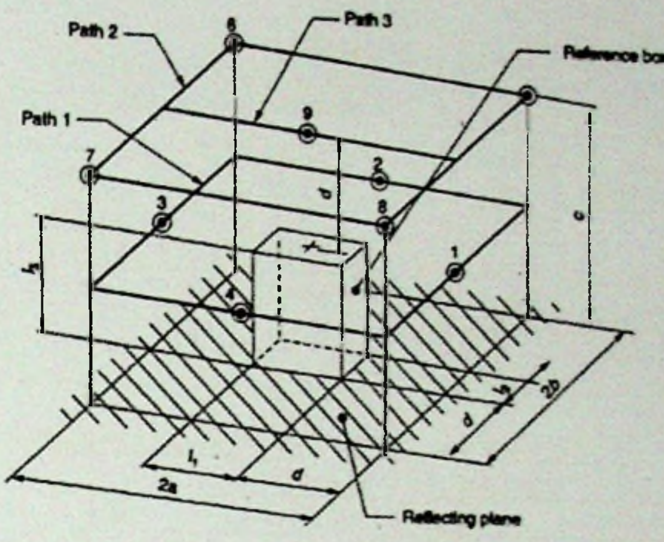


Figure F.1 Microphone positions for rectangular shape measuring shape according to ISO3744

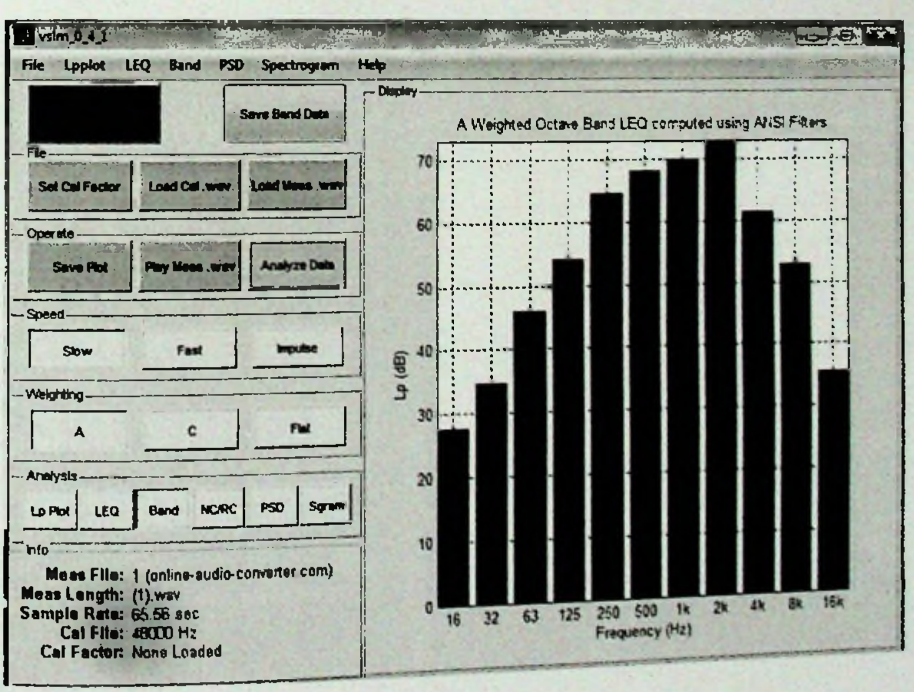


Figure F.2: GUI of virtual sound level meter VSLM 0.4

APPENDIX G

PREDICTED DESIGN VARIABLE DATA FOR COMMERCIALLY AVAILABLE PGS

Table G-1: Predicted design data for minimum SPL

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.003	0.969	0.997	1	0.1	21250
C220D5	0.003	0.919	0.958	0.554	0.071	34634
C330D5	0.003	0.906	0.962	0.922	0.1	21563
C440D5	0.003	0.908	0.928	0.925	0.08	29208
C550D5	0.003	0.956	0.912	0.925	0.097	21329

Table G-2: Selected design values for fabrication for minimum SPL

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.003132	1	1	1	0.1	20000
C220D5	0.003132	0.9	1	0.5	0.075	30000
C330D5	0.003132	0.9	1	0.9	0.1	20000
C440D5	0.003132	0.9	0.9	0.9	0.1	30000
C550D5	0.003132	1	0.9	0.9	0.1	20000

Table G-3: Predicted design data for source to panel distance constrained model

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.003	441.5	24	138	0.099	21563
C220D5e	0.003	622	0	404	0.09	24565
C330D5	0.003	559.5	162	287	0.091	24438
C440D5	0.003	780.5	231.5	332	0.094	23107
C550D5e	0.003	838.5	31.5	382	0.1	20000

Table G-4 : Selected design values data for source to panel distance constrained model

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.003132	441.5	24	138	0.1	20000
C220D5e	0.003132	622	0	404	0.1	20000
C330D5	0.003132	559.5	162	287	0.1	20000
C440D5	0.003132	780.5	231.5	332	0.1	20000
C550D5e	0.003132	838.5	31.5	382	0.1	20000

Table G-5: Predicted design data for overall SPL constrained model

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.001	1	0.982	0.748	0.093	23561
C220D5e	0.001	0.179	0.173	0.162	0.05	20000
C330D5	0.001	0.685	0.702	0.663	0.062	39575
C440D5	0.002	0.546	0.573	0.447	0.072	23104
C550D5e	0.001	0.43	0.385	0.977	0.083	36328

Table G-6: Selected design data for overall SPL constrained model

Genset Model	Results given by GA optimization					
	h/m	d1/m	d2/m	d3/m	t/m	τ (kPa.s/m^2
C110D5	0.001006	1	0.982	0.748	0.1	20000
C220D5e	0.001006	0.179	0.173	0.162	0.05	20000
C330D5	0.001006	0.685	0.702	0.663	0.075	40000
C440D5	0.001994	0.546	0.573	0.447	0.075	20000
C550D5e	0.001006	0.43	0.385	0.977	0.1	40000