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Noise Reduction of an I.C. Engine by using Enclosure with Glass Wool as Noise Absorbing Medium

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ABSTRACT:- *The main noise sources in I.C. engine were considered to be mechanical in nature. For noise reduction muffler, silencer, vibration control, barriers and enclosures can be used and should be considered at the design stage. A of method of noise reduction is depends upon the application of an I. C. Engine. Enclosures are specifically designed structures that reduce the amount of transmitted sound power impinging on the space of interest. In either barriers or enclosures, passive control is achieved by the use of an acoustic absorbing material or sound reflection. An absorbing material is used in this context to describe the materials ability to attenuate or "absorb" the sounds incident pressure wave, thus reducing the amplitudes of the transmitted and reflected pressure waves. The resulting sound pressure level on the opposite side of the absorbing material will therefore be reduced. The absorbing material is something as simple as a metal or plastic panel, or as elaborate as a multilayered partition which consists of specially engineered acoustic foam or fiberglass, different thicknesses of panel layers, as well as different panel materials. In this paper we have considered I.C. engine as a prototype of mobile generator and enclosure design for that with glass wool of thickness 50 mm as absorbing material. First SPL is measured without acoustic enclosure and then with acoustic enclosure of 1mm thick M.S. material with glass wool of 50 mm thickness.*

***Keywords:-* I.C. engine, Noise reduction, Acoustic enclosure, Acoustic material, Glass wool, thickness, Octave frequency band.**

INTRODUCTION

Noise is defined as a sound, generally of a random nature, the spectrum of which does not exhibit distinct frequency component. Different methods have used to reduce noise such like acoustic enclosure. Enclosures are specifically designed structures that reduce the amount of transmitted sound power impinging on the space of interest. In either barriers or enclosures, passive control is achieved by the use of an acoustic absorbing material or sound reflection. An absorbing material is used in this context to describe the materials ability to "attenuate" or "absorb" the sounds incident pressure wave, thus reducing the amplitudes of the transmitted and reflected pressure waves. The resulting sound pressure level on the opposite side of the absorbing material will therefore be reduced. The absorbing material is something as simple as a metal or plastic panel, or as elaborate as a multilayered partition which consists of specially engineered acoustic foam or fiberglass, different thicknesses of panel layers, as well as different panel materials [2].

While designing the enclosure, two types of enclosure resonances are taken into account. The first is the mechanical resonance of the enclosure panels and the other is the acoustic resonance of the air-space between the enclosed machine and the enclosure walls [6]. The light weight, close fitting enclosure will have some attenuation of the radiated sound due to the mass of the structure. Its effectiveness will also depend on the source to panel distance, the stiffness of the enclosure material, the enclosure panel vibration and the panel thickness [10]. Such element must be considered in turn in the design of an enclosure wall, and the transmission loss of the wall determined as an overall area weighted average of all of the elements.

For this calculation the following equation is used:

$$TL = 10\log 10 \frac{\sum_{i=1}^{q} S_i 10^{-TL_i/10}}{\sum_{i=1}^{q} S_i}$$

Where,

S_i is the surface area (one side only), and

TL_i is the transmission loss of the *i*th element.

The first few panel resonance frequencies of the enclosure should not be in the frequency range in which sound attenuation is desired, acoustic resonances occur at the standing wave frequency given by (Ubhe 1996) [2].

$$F = c / 2d,$$

Where c is the speed of sound and d is the distance between source and panel.

Among various kinds of sound absorption materials, porous materials such as glass wool quilting and polyurethane foams are the most common and significant technique which are widely used for room acoustics and various electric devices [7]. For noise absorption fibrous sound-absorbing materials, 15 types of glass wool (thickness 854mm and bulk density 20102 kg/m^3) are available commercially [8]. Mostly for high temperature applications glass wool and rock wool is used as insulating materials due to their great thermal insulation properties. Glass wool can be used as noise absorbing material [9]. So we use glass wool of density 24kg/m^3 and thickness 50mm as a noise absorbing material for acoustic enclosure.

DESIGN OF EXPERIMENT

Calculation of sound pressure level is done by two methods rectangular parallelepiped and hemisphere parallelepiped. In the present work, rectangular parallelepiped method is used. In this method, the first step is to make a grid according to the dimensions of engine. Length breadth and height of engine are approximately 375 mm, 175 mm and 100 mm. The grid is made by placing an engine at centre position and with the help of wire at required positions mark the different points. There are minimum 10 Grid points formed as shown in figure 1. Initial noise levels of an I C engine were measured in order to have reference values for comparison after applying control measures. The sound pressure levels obtained at a distance of 1m from an engine [1].

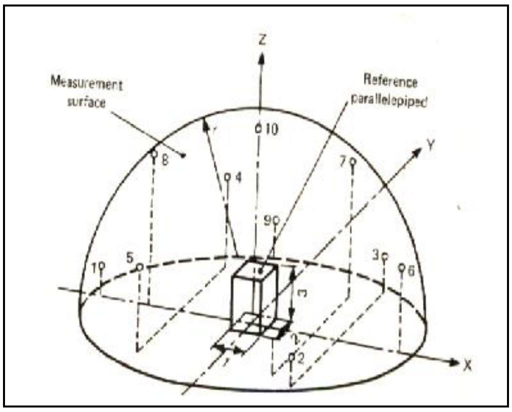

Figure 1. Layout of microphone positions

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Sound pressure level can be measured for every grid point shown in figure 2 for different throttle positions to change engine rpm in increasing manner by adjusting the throttle position at N1, N2 and N3 of engine respectively such that, speed at N1< speed at N2< speed at N3.

Figure 2. Experimental set up

EXPERIMENTATION

FFT analyser of model Bruel & Kjaer Photon+ 986A0186, Frequency limit1Hz to 20 KHz, Operating temperature range of 10^0C to 50^0C with microphone Bruel & Kjaer 4188 is used.

First SPL is measured by using FFT analyser for condition of without acoustic enclosure (EC1) at throttle position N1, N2, N3 in increasing manner. Then sane procedure is applied for the condition of acoustic enclosure with 1mm thick M.S. wall and glass wool of 50mm thickness (EC2) as shown in figure 3.

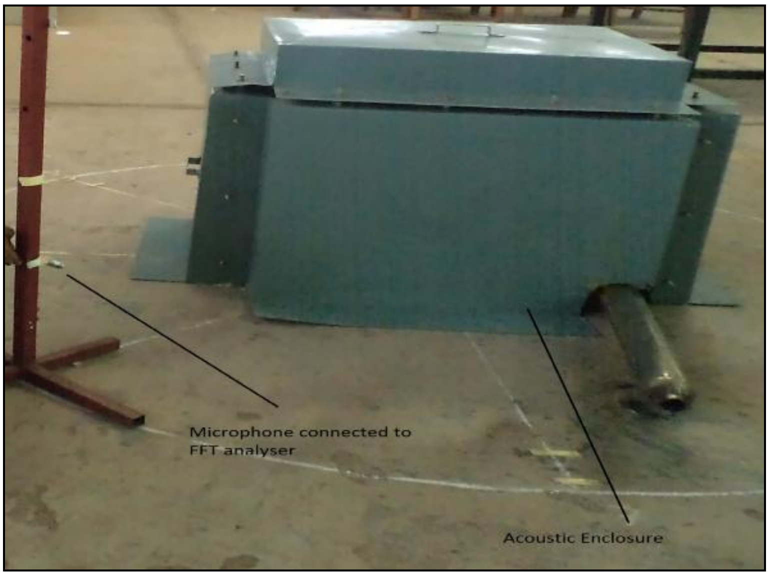

Figure 3. Measurement of SPL at different microphone positions

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EXPERIMENTAL DATA

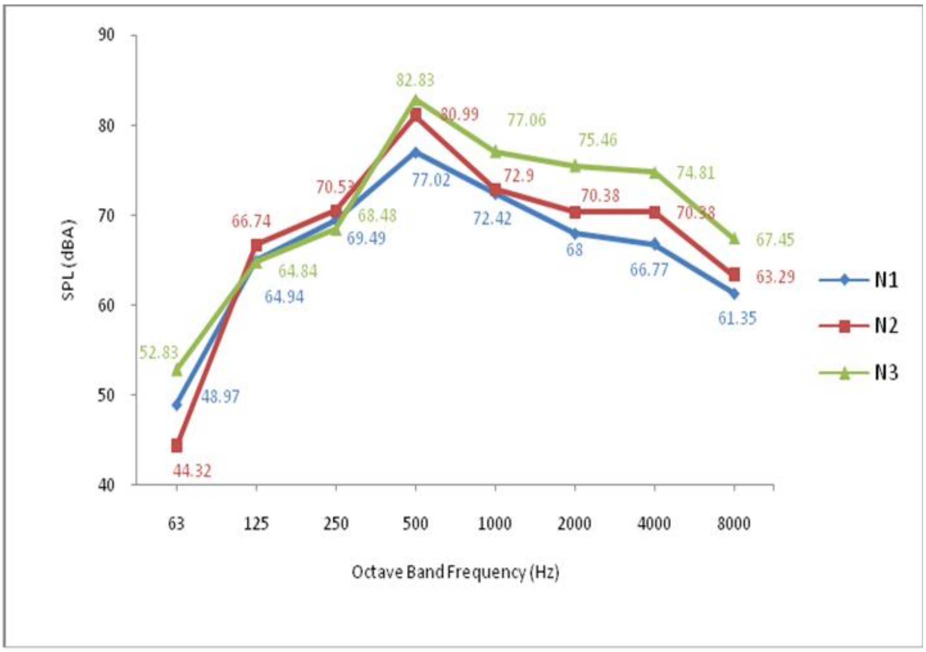

Figure 4. Avg. SPL at Octave Band Frequency for engine without an enclosure (EC-1)

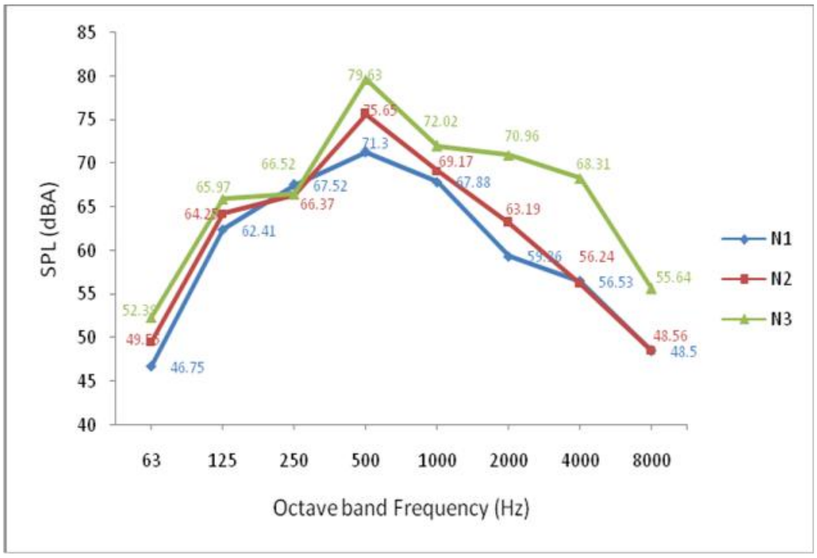

Figure 5. Avg. SPL at Octave Band Frequency for engine with an enclosure having M.S. sheet wall and glass wool of 50 mm thickness as an absorbing material (EC-2)

Figure. 4 and Figure. 5 shows the nature of variation of overall SPL at all three throttle positions with both enclosure conditions for octave frequency band. Which shows that for all frequencies of octave band maximum SPL is for Throttle Position 3 and enclosure condition 1.

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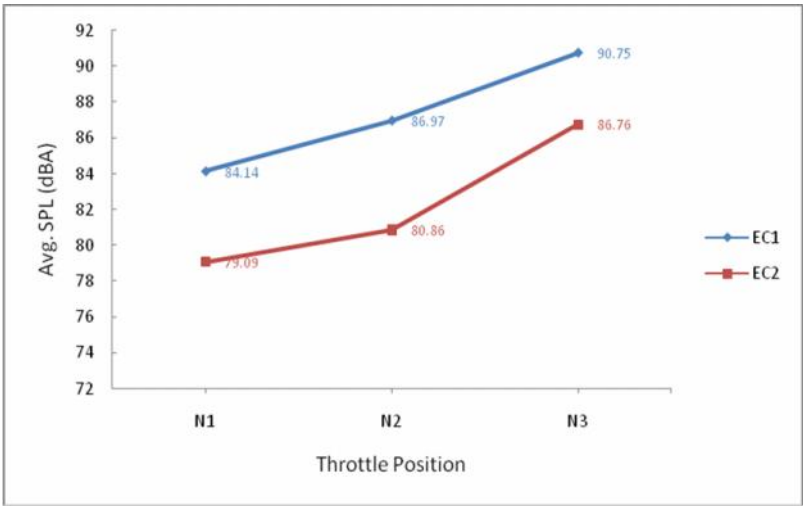

Figure 6. Avg. SPL for throttle positions N1, N2 and N3 for both conditions i.e. EC-1 and EC-2

Figure 6 shows the nature of variation of overall SPL at all three throttle positions for both enclosure conditions. Which shows that maximum SPL is for Throttle Position 3 and enclosure condition 1.

RESULTS

From this experiment following results can be calculated,

1. It is very difficult to reduce SPL for smaller frequencies of octave band as shown by graph in figure 7, on the other hand reduction in SPL for higher frequencies can be achieved up to 14.73dBA.

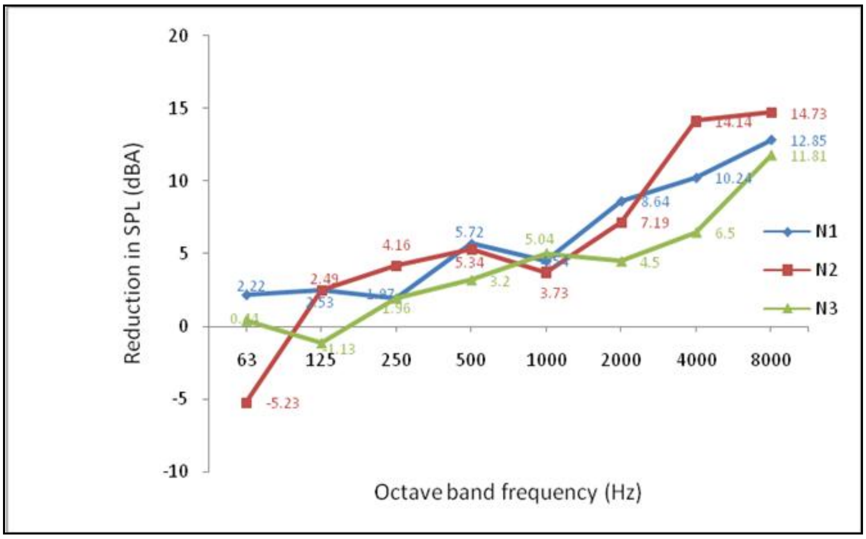

Figure 7. Reduction in an avg. SPL at Octave Band Frequency for EC-2 incomparisn with EC-1 for throttle positions N1, N2, and N3

2. The overall SPL is maximum for more throttle opening, and for enclosure condition 1 i.e. without enclosure up to 90.75dBA.

For throttle position N1 overall SPL is reduced by 5.05dBA, for N2 by 6.11dBA and for N3 3.99dBA as shown in graph of figure 8.

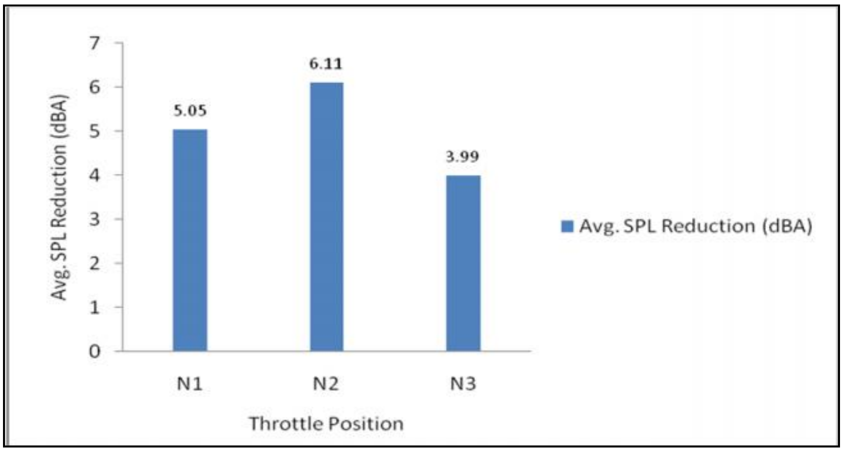

Figure 8. Reduction in an avg. SPL for an enclosure condition EC-2 compare with EC-1 at throttle positions N1, N2 and N3

CONCLUSIONS

1. As the throttle opening increases overall SPL increases irrespective of enclosure condition.
2. After application of acoustic enclosure SPL reduces for all throttle openings N1, N2 and N3.
3. Average overall SPL increases from centre band frequency 63Hz up to 500Hz and then again starts decreasing
4. Difference between average overall SPL is very less for lower frequencies and for higher frequencies this difference increases widely.
5. As overall SPL can be reduced by application of acoustic enclosure up to 6.11dBA, it can be used for less noise reduction applications such as portable generator of medium speed.

REFERENCES

[1] N. Tandon, B. C. Nakra, D. R. Ubhe and N. K. Killa, “Noise Control of Engine Driven Portable Generator Set”, Applied Acoustics, Vol. 55, No. 4, pp. 307-328, 1998.
[2] N Tandon, “Noise-reducing designs of machines and structures”, Sadhana, Vol. 25, Part 3, June 2000, pp. 331-339.
[3] Indian Institute of Technology Roorkee, “Vehicle Noise and Vibration Control”.
[4] Professor Colin H. Hansen, Dr Berenice I.F. Goelzer “Engineering Noise Control”, Department of Mechanical Engineering World Health Organization University of Adelaide South Australia 5005.
[5] C.R. Krishna, J. E. Wegrzyn, “Survey of Noise suppression systems for engine generator sets”, Brookhaven National Laboratory, Communications and Electronics Command, U.S. Army, October 1999.
[6] Joseph E. Blanks, Optimal Design of an Enclosure for a Portable Generator, Thesis submitted to the faculty of the Virginia Polytechnic Institute and State University in partial fulfillment of the requirements for the degree of Masters of Science in Mechanical Engineering February 7, 1997, Blacksburg, Virginia.
[7] Toshihiko Komatsuzaki and Yoshio Iwata, “Modeling of Sound Absorption by Porous Materials using Cellular Automata”, Graduate School of Natural Science and Technology, Kanazawa University, Kakuma-machi, Kanazawa, 920-1192, Japan.
[8] Takeshi Komatsu, “Improvement of the Delany-Bazley and Miki models for fibrous sound-absorbing materials”, 2008 The Acoustical Society of Japan, Industrial Research Institute of Shizuoka Prefecture, 2078 Makigaya, Aoi-ku, Shizuoka, 4211298 Japan.
[9] R.M. Prascevic, A.M. Milosevic, S.D. Cvetkovic, “Determination of absorption characteristic of materials on basis of sound intensity measurement”, JOURNAL DE PHYSIQUE IV, Colloque C5, suppl6ment au Journal de Physique 111, Volume 4, (1994), pp. 159-162.
[10] Professor Colin H. Hansen, Dr Berenice I.F. Goelzer, Department of Mechanical Engineering and World Health Organization, University of Adelaide, South Australia.