An Experimental Investigation on Noise Reduction by Using Modified Helmholtz Resonator

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Abstract

Noise is a frequently encountered problem of all the automotive engines. When subjected to external excitation, a Helmholtz resonator can show a strongly resonant response at a well-defined frequency. This resonant characteristic can be used to effectively attenuate the noise, by attaching the resonator to the exhaust. It is particularly useful for the reduction of low frequency and narrow band noise. In this investigation, modified Helmholtz resonators are designed and fabricated by noise frequency spectrum analysis. Modifications of the Helmholtz resonators are carried out by inserting the resonator neck inside the cavity. Noise level by modified Helmholtz resonator at different loading condition of engine was measured. Effect of parallel and series arrangement of resonators on noise reduction was found out. In addition, exhaust back pressure due to the designed resonators was measured to find out its effectiveness. Finally, effect on noise reduction by modified and non-modified Helmholtz resonator was investigated. The results shall enable engineers to know the effect of modified Helmholtz resonator on noise reduction.

CHAPTER 1
INTRODUCTION

1.1 Background and Present State of the Problem

Noise is defined as unwanted sound. It is a frequently encountered problem in modern society. Presence of noise deteriorates working environment and human comfort. Therefore, since past, efforts were taken to reduce equipment noise by many ways and means. Sound is a pressure wave formed from pulses of alternating high and low pressure air. In an automotive engine pressure waves are generated when the exhaust valves repeatedly open and let high pressure gas into the exhaust system. These pressure pulses are the sound we hear. It is to be mentioned that all noise emitted by an automobile does not come from the exhaust system. Other contributors to vehicle noise emission include intake noise, mechanical noise and vibration induced noise from the engine body and transmission.

In general, sound waves propagating along a pipe can be attenuated using either absorptive or a reactive silencer. Absorptive silencer uses a straight through perforated tube wrapping with sound absorbing material to take energy out of the acoustic motion in the wave, as it propagates through the silencer. Reactive silencers, which are commonly used in automotive applications, reflect the sound waves back towards the source and prevent sound from being transmitted along the pipe. The reactive type silencer is usually good for noise of low frequency noises and absorptive type silencers are good for high frequency noises. Reactive silencer design is mainly based either on the principle of a Helmholtz resonator or an expansion chamber. There are different types of reactive silencer. These are Helmholtz resonator, side branch resonator, expansion chamber silencer, tube in tube resonator, bent resonator etc. These silencers are used for different desired application. Knowledge of Helmholtz resonance and its theoretical description have been used over the years to design and analyze various systems. An advantage of the Helmholtz resonator is that it has the characteristic of strong sound attenuation, even though its geometry is relatively simple. When it is appropriately tuned, it can substantially reduce noise over the low frequency domain. Many researchers and engineers have been interested in and employed the Helmholtz resonator for a variety of applications.

In this paper an experimental investigation is carried out on noise reduction by designing and fabricating a modified Helmholtz resonator through noise spectrum analysis. As the Helmholtz resonator is particularly useful for low frequency and narrow band noise reduction, many resonators are combined together to reduce low frequency noise. Effect on noise reduction in series and parallel of modified resonators, varying the number of resonators and at different loading conditions are also tested. Finally, Effect on noise reduction by modified vis-a-vis non-modified Helmholtz resonator was experimented by spectrum analysis also analyzed.

1.2 Internal Combustion Engine Noise Signal Components

An internal combustion engine noise signal component is composed of many components from different sources. These sources include combustion, mechanical, and the combination of both. The combustion noise is produced by a rapid rate of pressure rise, which besides being a source of engine structural vibrations also excites resonance in the gas inside a combustion chamber cavity.

A high intensity pressure wave generated by combustion in the engine cylinder propagates along the exhaust pipe and radiates from the exhaust pipe termination. In a normal condition, the combustion noise is mostly in a frequency range above 100 Hz as the combustion energy below this range is mostly transformed into useful work by pushing pistons forward [1]. However, degradation in the combustion quality may produce some low frequency content in the combustion noise. Measurements of the exhaust pipe pressure pulse show that the majority of the pulse energy lies in the frequency range of 50 to 600 Hz [2]. However, few significant noises also observed up to 1500 Hz. Therefore, exhaust silencers are designed to attenuate a broad band range of low and medium frequencies.

An important feature of IC engines is that they have both reciprocating and rotating parts. Different type of parts will produce different signal components. Rotating parts, such as the flywheel and front pulley, can excite harmonic components to the noise decided by the engine speed, these harmonic components mainly distribute in the low-frequency range. An increase in the amplitude of the harmonic components indicates condition variations of these rotating parts. Contributions of different rotating parts to the noise can be identified with reference to their speeds. Injectors and valves are reciprocating moving parts. They produce impacts to the engine structure and hence contribute transient components to the noise. In a normal condition, there are two major impacts generated in the operation of an injector. The first impact is the needle of the injector striking the backstop, and the second impact occurs when the needle sits back in its seat. In an injector, the needle is held onto its seat by a high rate spring. This spring also serves to control the injection pressure and regulate the injection time. A decrease in the stiffness of the spring will bring forward the injection time. As a consequence, the combustion quality will be degraded. Low spring stiffness may also fail to push the needle onto its seat against the combustion pressure. In some other abnormal conditions, the needle may remain open because of the deterioration of the guide or the seat. In such cases, there will be no impact at all. Degradation in fuel pumps, such as low pressure and piston leakage, will also result in a change in the injection time and pressure. In summary, time and amplitude of some transient components are indications of the condition of an injection system. An engine has many inlet valves and exhaust valves. A valve is opened by a camshaft and pushed back to its seat by a valve spring. Any problems with valve seats, tappets, and mechanisms can cause a change to the transient vibrations produced during opening and closing, and thus the corresponding transient components of the noise signal. These valves open and close at different times, and so the contribution of different valves to the noise can be identified from the times of events. Fluid-induced noise, such as exhaust and inlet noise, is also an important part of the noise. Along with the sudden release of gas into the exhaust system or the rush of a sharp pulse of fresh air into the cylinder, oscillation of the air volume in the cylinder and the exhaust system is excited and hence noise is produced. When inlet and exhaust valves close, noises will also be generated for a change in the fluid field. The fluid-induced noise contributes transient components to the whole noise. Some early research shows that fluid-induced noise usually has high frequencies. With modern fluid passage designs, the level of fluid-induced noise is normally very low. Damage or problems with the exhaust and inlet system will increase the magnitude of the fluid-induced noise.

1.3 Helmholtz Resonance

Helmholtz resonance is widely known as the phenomenon of air resonance in the cavity or chamber that contains a gas. The name comes from a device created in the 1850s by Hermann Ludwig Ferdinand von Helmholtz (31 August 1821 – 8 September 1894). In a Helmholtz resonator design, a cavity is attached to the exhaust pipe. At a specific frequency the cavity will resonate and the waves in the exhaust pipe are reflected back towards the source. However there are also pass band frequencies where the resonator has no effect and so resonator silencer design is targeted to specific frequencies where the majority of the attenuation is required. In some designs, the silencer has several resonators of different sizes to target a range of frequencies.

A Helmholtz resonator produces sound frequencies by a method analogous to the oscillation of a mass-spring oscillator[3]. For producing sound, the resonator has a neck. A cavity is connected to neck containing a large volume of air. The air in its neck behaves as a discrete mass, while the air in the cavity has the role of a spring. When air is forced into the cavity, the pressure in the cavity is increased and air will be pushed out and thereby more air will be expelled out of the resonator than necessary. Hence, the pressure inside the resonator will now be less than the pressure outside and in order to compensate the pressure differences, more air will be sucked into the cavity. This process repeats until the system finally reaches equilibrium. Basically, the movement of air into and out of the resonator is identical to the movement of a spring along the vertical axis. Therefore, during oscillation, the gas within the volume of the resonator is alternately compressed and expanded at very low magnitudes. The inertia of the air in the neck of the resonator plays an important role.

Helmholtz resonators are typically used to attenuate sound pressure when the system is originally at resonance and reduces potentially loud and obnoxious engine noise. Therefore, dimensions are calculated so that the waves reflected by the resonator help cancel out certain frequencies of sound in the exhaust.

1.4 Objectives

The aim of this thesis is to experimentally investigate the reduction of noise of modified Helmholtz resonator by spectrum analysis. The objectives of the experimental investigations are as follows:

(1) To design and fabricate modified Helmholtz resonators by engine noise frequency spectrum analysis.

(2) To measure noise level by modified Helmholtz resonator at different loading condition of engine.

(3) To find out the effect of combination (parallel and series) and number of modified Helmholtz resonators on noise reduction.

(4) To measure the back pressure due to the designed resonators.

(5) To analyze effect on noise reduction by modified and non-modified Helmholtz resonator using noise level meter and spectrum analysis.

(6) To analyze effect of modified Helmholtz resonator and absorptive silencer on noise reduction.

The results shall enable us to know the effect of modified Helmholtz resonator on noise reduction.

1.5 Scope of the Study

This section contains the brief description of the different themes which has been presented in the various chapters. Internal combustion engine noise characteristics and ways and means to attenuate noise and thereby problem was stated in **chapter 1.** The importance of the investigation on noise reduction by modified Helmholtz resonator through spectrum analysis and the aim of the thesis have also been included in this chapter.

In **chapter 2** the brief survey of the various related literatures has been provided. Usually, the research works which are directly related to the present study has been included in this chapter. Some works which are in line with the present study have also been included.

The theoretical outline of Helmholtz resonator and transmission loss is explained in **Chapter 3.** The description of the experimental set-up and the measuring equipment has been given in **chapter 4** in a nutshell.

The most important part of the thesis is the results and discussion, which have been provided in **chapter 5.** The effect of modified Helmholtz resonator is explained this chapter.

Finally, in **chapter 6** the conclusions and the recommendations for future researchers have been given.

1.6 Limitations and Assumptions

(1) Environmental noise hampered during taking engine exhaust noise reading in the laboratory.

(2) Noise from engine other than exhaust also hampers the exact exhaust noise.

1.7 Outline of Methodology

Noise was generated from an engine through exhaust manifold and noise level was measured at its exit. Spectrum of unsilenced engine noise was also analyzed by Audacity software to find out frequencies corresponding to maximum noise level. Thereby target frequencies were identified. Helmholtz resonators were designed for target frequencies. Helmholtz resonators were modified by inserting the neck of the resonator inside the resonators cavity. Thus modified Helmholtz resonators are fabricated and fitted with the exhaust manifold. Reduction of noise was investigated through spectrum analysis by the designed Resonators at different loading condition and by varying number of those. Finally, effect on noise reduction by modified and non-modified Helmholtz resonator and effect of modified Helmholtz resonator and absorptive silencer was analyzed using noise level meter and spectrum analysis.

CHAPTER 2
REVIEW OF THE LITERATURE

2.1 Previous Studies

A brief description of some papers related to the present state of the problem is mentioned here. Seo et. al [4] studied silencer design by using array resonators for low-frequency band noise reduction. They experimentally studied the serial and parallel arrangement of Helmholtz resonators and combination of those. They proposed a new design method that optimizes the arrangement of resonators for transmission loss that has a broadband characteristic in low frequency.

Little et al. investigated a fluidic Helmholtz resonator for use as an adaptive engine mount. Dimensions of the cross-sectional area of the neck of the Helmholtz resonator were modified to attenuate the noise. A creative electro-rheological fluid component was developed to continuously alter the cross sectional area of the neck; this approach is different from the configuration of a widely utilized valve, which has a discontinuous characteristic. However, the algorithm for the controlling parameter that relates the opening of the surface area was not explained.

Lamancousa designed a changeable cavity of the Helmholtz resonator to substitute for expansion chamber silencers in automobiles. Two types of modifiable arrangements of the cavity volume were considered. In the first device, the volume was continuously changed by increasing or decreasing the length of the cavity through a moveable piston inside the cavity. In the second type of device, separate volumes were employed; that is, the volume of the chamber was divided into several sub-volumes, which could be closed off. Using the approaches described in the foregoing, it was possible to attain either a continuous or discrete variation of volume of the resonator according to the revolution signal of the engine. In these devices, it was determined that insertion losses of more than 30 dB were achieved by manipulating the continuously changeable volume of the resonator. It should be noted that Krause et al. also experimentally investigated the effect of alterations of the volume and the neck of a Helmholtz resonator on attenuation of the source of noise in automotive tailpipes.

Matsuhisa et al. investigated the consequences of the variable volume of a resonator by using a removable piston inside the chamber. The resonator was attached to a duct in the manner of a side branch and the adjustment of the volume of the resonator was guided by comparison with the phase of the sound pressure in the duct, relative to that in the resonant cavity. The chamber of the resonator was controlled to maintain a constant phase difference of ninety degrees. Using this procedure, anti-resonance of the duct-resonator system was accomplished. In this experiment, three sensors were utilized to measure and compare the sound pressure in different positions. One microphone was used to measure the excitation frequency, one was employed to measure the pressure in the duct, and the remaining one measured the pressure in the cavity. It was found in this investigation that the use of an adjustable resonator produces reductions in sound pressure up to 30 dB for a speaker driven system and 20 dB for a fan-driven system.

McDonald et al performed tonal noise control by using a variable Helmholtz resonator, similar to that used in the experiments of Matsuhisa et al. The phase difference between the pressure and the resonator cavity in the duct system was employed to guide adjustment of the volume of the cavity and the length of resonator neck, in order to achieve sound reduction.

Selamet et. al.[5] showed that the individual dimensions of a Helmholtz resonator can play a great role in determination of the resonant frequency and the transmission loss characteristics. An increase of the ratio of the length scale of the volume to the diameter decreases the predominant resonant frequency. This phenomenon is similar to the result of using an effective length, which includes a correction length. Experiments show agreement with the analytical expression and numerical simulation.

DeBedout [6] investigated an adaptive Helmholtz resonator, which optimized its performance according to changes in environmental conditions and excitation frequency. It was found that reduction in sound pressure up to 30 dB could be attained through a combination of a variable resonator and an appropriately controlled algorithm. For the case of this adaptive-passive noise control device, the control algorithm is simple and the efficiency of the process is optimized. Furthermore, with the tunable Helmholtz resonator, it is possible to achieve optimal reductions of sound in response to changing environmental conditions and excitation frequency.

Tang [7] investigated the effects of the taper and length of the resonator neck on the characteristics of a Helmholtz resonator. It was investigated that an increase of the tapered length leads to improvement of sound reduction and an increase of the cavity volume results in increased capacity for sound absorption of the Helmholtz resonator. These experiments showed that sound attenuation via the Helmholtz resonator of more than 50% could be achieved by changing the length of the tapered neck, compared to the untapered neck. The increase of the resonant frequency is proportional to the tapered length and is decreased by expanding the cavity volume at a fixed slope of the tapered section. In addition, this investigation showed that the resonant frequency is proportional to the slope of the tapered section at constant volume of the Helmholtz resonance chamber.

Han [8] investigated sound reduction via selected Helmholtz resonator. The resonant frequencies of the Helmholtz resonator were evaluated by experiments and an analytical method, while changing the geometrical dimensions of the Helmholtz resonator, including the neck cross-sectional area, the length of the neck, and the magnitude of the volume.

Hannink [9] applied tube resonator for the reduction of sound radiation and sound transmission. He investigated the applicability of this method to develop and validate efficient models for the prediction of sound radiation by and sound transmission through panels with tube resonators.

Prydz et. al [10] studied the acoustic characteristics of panel using multiple array resonators to obtain a high absorption coefficient at low frequency. Koai et.al [11] studied the muffling effect of Helmholtz resonators installed in different environment.

Rahman et. al [12] designed and constructed a silencer for engine exhaust noise. They studied muffling effect of conventional silencer with tube resonator. With regard to characteristics of the silencer using resonators, Anderson [13] studied the effect of flow when a single side branch Helmholtz resonator is attached to a circular duct.

Koopman and Neise (1980, 1982) studied the use of adjustable resonators to dampen the tone produced by blade passage of centrifugal fans. The volume of the Helmholtz resonator was changed by use of a moveable Teflon piston. Their experimental results showed that the amplitude of the tone of the blade passage frequency could be decreased up to 29 dB without generating a negative side effect on the fan frequency. However, no definitive methods were suggested for achieving the optimal condition of sound reduction by variation of the cavity depth.

2.2 Scope of Work

Several studies on the applications of resonator are in progress in various fields. The previous studies of the Helmholtz resonator have provided fundamental knowledge for the present experiments. Taking into account these investigations, sound reduction via modified Helmholtz resonator was pursued. Therefore, endeavor was taken to investigate noise reduction by fabricating modified Helmholtz resonator. Experiment was carried out by frequency spectrum analysis. Noise level was measured at different loading conditions. Thereby, noise reduction was investigated at different arrangement and number of resonators. Effect of modified Helmholtz resonator and absorptive silencer on noise reduction was also analyzed. The outcome of the present work will contribute to control of exhaust noise of engine effectively.

CHAPTER 3
THEORETICAL OUTLINE

3.1 Helmholtz Resonator

A Helmholtz resonator consists of a closed volume connected to the noise source through a tube that is properly sized to tune the resonant frequency of the combination to a specified value. The operation of the resonator is based on reflection of waves at the source, that is, when passing through the exhaust manifold a small portion of energy propagates through the resonator, and most of the energy is reflected back to the source. Although large amounts of attenuation are theoretically obtainable at a given frequency, limited only by damping effects, the use of resonators has been somewhat restricted in practical silencer configurations. Intake and exhaust noise typically are quite broadband and do not lend themselves well to narrow band treatment. In addition, due to the high sound levels and flow present in internal combustion engine exhaust systems, nonlinear effects are predominate at resonant frequencies, and the attenuation predicted by linear acoustical calculations is often not obtained. Geometry of a typical Helmholtz resonator is shown in fig.3.1.

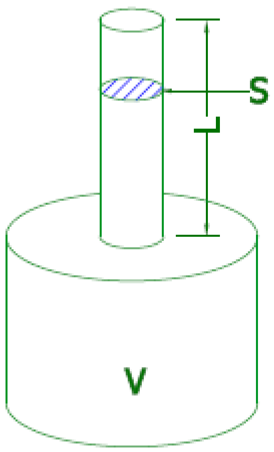

Fig 3.1: A typical Helmholtz resonator. (L=Neck length, S=Neck cross-section area, V= Volume of resonating chamber)

3.2 Resonant Frequency of Helmholtz Resonator

Consider a basic force balance F=ma, where m is mass and a is acceleration, $a = \frac{d^2x}{dt^2}$. Therefore, $\frac{d^2x}{dt^2} = \frac{F}{m}$ (1)

For adiabatic system with air as an ideal gas, the thermodynamic process equation for the resonator is

$pV^{\Upsilon} = Constant$

Consider

$\frac{p}{p_0} = \Upsilon \frac{\Delta V}{\mathrm{V}}$ (2)

Where p = Pressure at the neck entrance, p_0 is atmospheric pressure

The change in the cavity volume is

$\Delta V = -Sx$ (3)

S is neck cross sectional area, x is displacement. Putting the value of equation (3) in to (2)

$P = \Upsilon \frac{Sxp_0}{\mathrm{V}}$ (4)

Where $x(t) = e^{i\omega t}$ and ω is angular frequency, $\omega = 2\pi f$

Force F can also be expressed in terms of pressure

$F = PS = \Upsilon \frac{Sxp_0}{\mathrm{V}} S = \Upsilon \frac{S^2xp_0}{\mathrm{V}}$ (5)

$m = \rho v = \rho SL$, (6)

Where L is length of neck, m=acoustic mass of the resonator

Equation (1) becomes, by substitution of equations (4), (5) and (6)

$\frac{d^2x}{dt^2} = (i\omega)^2 \, e^{i\omega t} = \frac{\Upsilon \frac{S^2xp_0}{\mathrm{V}}}{\rho SL} = \frac{\Upsilon Sxp_0}{\rho VL} e^{i\omega t}$ (7)

$-\omega^2 = \frac{\Upsilon Sp_0}{\rho VL}$ (8)

$2\pi f = \sqrt{\frac{\Upsilon Sp_0}{\rho VL}}$

$$f = \frac{1}{2\pi}\sqrt{\frac{YSp_0}{\rho VL}} \qquad (9)$$

The speed of sound is $c = \sqrt{Y\frac{p_0}{\rho}}$

Then, $f = \frac{1}{2\pi}\sqrt{\frac{YSp_0}{\rho VL}} = \frac{1}{2\pi}\sqrt{Y\frac{p_0}{\rho}}\sqrt{\frac{S}{VL}} = \frac{c}{2\pi}.\sqrt{\frac{S}{VL}}$ (10)

Therefore, the resonant frequency is defined as

$$f = \frac{c}{2\pi}\sqrt{\frac{S}{VL}}$$

3.3 Transmission Loss of a Helmholtz Resonator

Transmission Loss (TL) is the accumulated decrease in acoustic intensity as an acoustic pressure wave propagates outwards from a source. As the acoustic wave propagates outwards from the source the intensity of the signal is reduced with increasing range due to spreading and attenuation or absorption. Silencer using a Helmholtz resonator reduces noise by an impedance mismatch [14]. Acoustic impedance indicates how much sound pressure is generated by the vibration of molecules of a particular acoustic medium at a given frequency. Acoustic impedance Z (or sound impedance) is frequency (f) dependent. Mathematically, it is the sound pressure p divided by the particle velocity v and the surface area S, through which an acoustic wave of frequency propagates. Impedance mismatch causes reflection of the incident acoustic energy and attenuation in the resonator‟s neck. When a resonator is attached to a duct by a side branch, as depicted in Fig 3.2, the basic assumption is that plane waves propagate in a duct and the reflected waves from downstream of a duct do not exist in the absence of mean flow. Considering effects of grazing flow, if the mean flow‟s velocity is less than *M*=0.1 (*M*: Mach number), its effect is not serious [15]. The sound pressure (P) and the volume velocity (U) can be expressed as follows:

$$P_1 = \left(Ae^{-jkx} + Be^{jkx}\right),\ P_2 = Ce^{-jkx} \qquad (11)$$

$$U_1 = \frac{1}{Z}\left(Ae^{-jkx} - Be^{jkx}\right),\ U_2 = \frac{1}{Z}\left(Ce^{-jkx}\right) \qquad (12)$$

Where A, B, and C are the magnitude of the incident wave, reflected wave, and transmitted wave, respectively, and $Z = \frac{\rho c}{S}$ is the acoustic impedance of the duct(S=surface area). Here $k = \frac{2\pi f}{c}$ is the wave number, ρ is the density of air, and c is the sound speed.

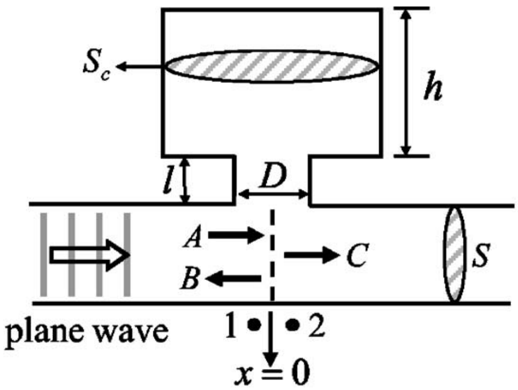

Fig 3.2: Transmission loss in a Helmholtz resonator

The transfer matrix between point 1 and point 2 can be obtained as follows by using the continuity of the sound pressure and the volume velocity:

$$\begin{pmatrix} P_1 \\ U_1 \end{pmatrix} = \begin{bmatrix} 1 & 0 \\ \frac{1}{-jZ_c\,Cot\,kh + Z_h} & 1 \end{bmatrix} \begin{pmatrix} P_2 \\ U_2 \end{pmatrix}$$

$$= \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{pmatrix} P_2 \\ U_2 \end{pmatrix} \qquad (13)$$

Where,

Acoustic impedance of a resonator‟s cavity, $Z_c = \frac{\rho c}{S_c}$.

The impedance of resonator Z_r can be expressed as

$$Z_r = -jZ_c \cot kh + Z_h \qquad (14)$$

$$Z_h = \frac{\rho c}{S_h}[0.0072 + jk(l + 0.75)] \qquad (15)$$

Where, Z_h is the hole impedance of a resonator, as suggested by Sullivan [16]. Sullivan obtained the hole impedance of perforated elements in a concentric tube resonator by measurement. Here, the resistance of Z_h was modified in consideration of the experimental results of this study, and S_h is the cross-sectional area of the hole.

Transmission loss (TL) can be represented as follows by using the transfer matrix:

$$TL = 20 \ \log_{10} \left| \frac{A}{C} \right|$$

$$= 20 \log_{10} \left| \frac{T_{11} + \frac{T_{12}}{Z} + T_{21}.Z + T_{22}}{2} \right| \qquad (16)$$

Where A is acoustic pressure of the incident wave and C is the acoustic pressure of the transmitted wave. Using Equation (13) and (16), the TL of a branch resonator in the duct can be obtained as:

$$TL = 20 \ \log_{10} \left| \frac{2 + Z\left(\frac{1}{-jZ_c Cot \ kh + Z_h}\right)}{2} \right|$$

3.4 Sound Signal

In analog electronics, a time varying signal can be represented by voltage or current waveform, which represents some quantity changing with time. Sound is a mechanical wave and a microphone can be used to convert it to a time varying analog signal. The three primary characteristics of a time varying analog signal include level, shape, and frequency. Microphone or sound sensor measures the sound signal in time domain (level vs. time). Fourier transformation can be used to obtain the frequency domain representation from the time domain (power/amplitude vs. frequency).

3.4.1 Decibel Representation

The decibel (dB) is a logarithmic unit used to express the ratio between two values of a physical quantity, often power or intensity. When referring to measurements of power or intensity, a ratio can be expressed in decibels by evaluating ten times the base-10 logarithm of the ratio of the measured quantity to the reference level [17]. Thus, the ratio of a power value P_1 to another power value P_r is represented by dB, which is calculated using the formula:

$$dB = 10\ log_{10}\frac{p}{p_r}$$

The base-10 logarithm of the ratio of the two power levels is the number of bels. The number of decibels is ten times the number of bels. P and P_r must measure the same type of quantity, and have the same units before calculating the ratio. If $P = P_r$ in the above equation, then dB = 0. If P is greater than P_r then dB is positive; if P is less than P_r then dB is negative [18].

When referring to measurements of field amplitude, it is usual to consider the ratio of the squares of A (measured amplitude) and A_r (reference amplitude). This is because in most applications power is proportional to the square of amplitude and it is desirable for the two decibel formulations to give the same result in such typical cases [18]. Thus, the following definition is used:

$$dB = 10\ log_{10} \frac{A^{2}}{{A_r}^{2}}$$

$$dB = 20\ log_{10} \frac{A}{A_r}$$

Similarly, in electrical circuits, dissipated power is typically proportional to the square of voltage or current when the impedance is held constant. For voltage, this leads to the equation:

$$dB = 20\ log_{10} \frac{V}{V_r}$$

Where, V is the voltage being measured, V_r is a specified reference voltage. A similar formula holds for current [19].

The decibel unit can also be combined with a suffix to create an absolute unit of electric power. For example, it can be combined with "m" for "milliwatt" to produce the "dBm". 0 dBm is the level corresponding to 1 milliwatt [18].

In professional audio specifications, a popular unit is the dBu. The suffix u stands for unloaded, and was probably chosen to be similar to lowercase v, as dBv was the older name for the same unit. The dBu is a root mean square (RMS) measurement of voltage that uses as its reference approximately 0.775 V_{RMS}. The reference value is the voltage level which delivers 1 mW of power in a 600 ohm resistor, which used to be the standard reference impedance [20].

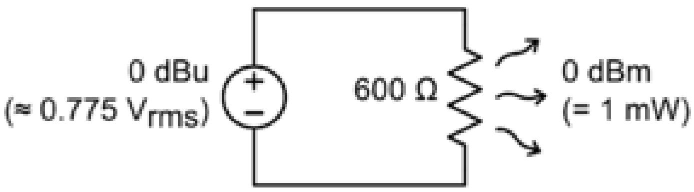

Fig 3.3: A schematic showing the relationship between dBu and dBm

The most common usage of "decibels" in reference to sound level is dB SPL, sound pressure level referenced to the nominal threshold of human hearing. The reference sound pressure is 20 μPa (rms) or 0.00002 N/m^2, which is usually considered the threshold of human hearing (roughly the sound of a mosquito flying 3 m away)[17]. This corresponds to 0 dB. Threshold of pain is 200 N/m^2 and corresponds to 130 dB. The noise meter measures "Sound Pressure" and they take the threshold of human hearing as the reference. Sound pressure is a field quantity; therefore the field version of the unit definition is used.

$$dB = 20\ log_{10} \frac{p}{p_r}$$

Where p_r is equal to the standard reference sound pressure level of 20 micro pascals in air or 1 micropascal in water.

3.4.2 Digital Sound Level

dBFS is a dB reference level equal to “Full Scale” or “Full Code.” It is used in specifying A/D and D/A audio data converters, but also increasingly used to refer to signal levels in the digital domain since they are almost always referenced to the full code value. A full Code signal is the maximum theoretical output of a given digital device, which refers to the maximum voltage level possible before “digital clipping,” or digital overload of the data converter. The actual Full Scale voltage is fixed by the internal data converter design, and varies from model to model.

Digital devices have a very finite and exact amount of dynamic range depending upon how many bits are used in recording (8-bit, 16-bit, 24-bit, etc). As dBFS is referenced to Full Scale / Full Code value, 0 dBFS is when all of the one‟s and zero‟s of the digital signal become one‟s for a given sample. All the other levels will result in a negative dB measurement, as those values are smaller than the reference value (Full Code).

Assuming 16 bit audio, 0 dBFS indicates the digital number with all digits ="1", the highest possible sample = $(1111\ 111111111111)_2 = (65535)_{10}$

The lowest possible sample is 0000 00000000 0001, which equals to 20 log(1/65535) = -96.33 dBFS. So, for 16-bit systems, all levels will be between 0 dBFS (the highest level) to -96.33 dBFS (the lowest level). For 20-bit digital audio, all measurements will be between 0 dBFS to -120.41 dBFS. For 24 bit digital audio, the measurements will be between 0 dBFS to -144.5 dBFS.

For an n-bit system, the lowest measurement will always equal to

$$= 20 \log \left(\frac{1}{2^n}\right)$$
$$= -20n \log(2)$$
$$= -6.02n$$

3.4.3 Relation between Analog dBm / dBu and Digital dBFS

On an analog meter, 0 dB is the optimal recording or output level of a device. If the voltage is much higher, the signal may distort. If the voltage is much lower, the signal may be lost in the noise inherent in the device. On a digital meter, 0 dBFS refers to the highest audio level allowed before clipping. In digital metering a level of 0 dBFS is ostensibly equal to 0 dBm in analog measurement, but in practice they are not equal due to discrepancies added at several point due to converter design decisions, quantization levels, resolution, and etc issues. There are several common digital levels used to correspond to 0 dB on an analog meter.

3.4.4 Spectrum Analysis of Digital Audio

Analog spectrum analyzers can take an analog signal and apply fourier transformation on it. Alternatively, digital spectrum analyser uses computer software and it can use Fast Fourier Transform algorithm to compute the discrete Fourier transform (DFT) and its inverse. Software always works on digital data, and so the dB levels it computes for different frequency levels are dBFS.

3.5 Modified Helmholtz Resonator

Helmholtz resonator was modified by inserting the neck inside the cavity (Fig.: B1). Such modification was carried out by making thread in cavity and end of the neck. Impact of modified Helmholtz resonator was measured and attenuation characteristic was compared with actual Helmholtz resonator. Modified Helmholtz resonator is shown in fig. 3.3.

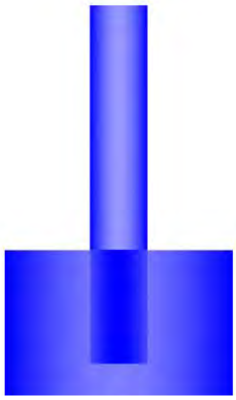

Fig 3.4: Modified Helmholtz resonator

3.6 Functional Requirement

The primary requirement of the designed resonator is desired insertion loss characteristic. Numerous secondary requirements such as exhaust back pressure, size, weight, durability, cost etc are also considered. Backpressure represents the extra static pressure exerted by the resonator on the engine through the restriction in flow of exhaust gases. Backpressure should be kept to a minimum (below 40 mbar) to avoid power losses of engine. Volume, weight and cost were balanced to obtain the desired performance. Styling is strongly related to the ease of manufacture and to performance particularly for designs require unusual shapes or inadequate volumes.

Reasonable amount of durability was attained through proper material selection, although at an increase in cost. Typically, silencer placement and piping arrangements are restricted because of operational requirements for the application being considered. Carbon steel was used to fabricate the resonator.

From the initial volume specification the inlet and outlet locations for the silencer were determined. Resonators were designed and fabricated on the basis of the attenuation required as well as the frequencies of maximum noise levels. Following the analytical evaluation, refinements were made. A final test was performed using an engine its effectiveness under actual operating conditions was determined.

CHAPTER 4
DESIGN OF MODIFIED HELMHOLTZ RESONATORS

4.1 Experimental Procedure

Noise was generated from an engine through exhaust and noise level was measured at its exit by noise level meter. An isolating duct was fitted after exhaust manifold to get the exact exhaust noise. Noise spectrum of unsilenced engine was analyzed by Audacity software to find out frequencies corresponding to maximum noise level. Thereby target frequencies were identified. Helmholtz resonators were designed for target frequencies. On fabrication of Helmholtz resonator again noise level was measured. Noise attenuation was measured by noise level meter. Noise spectrum was analyzed to find out effectiveness of the designed resonators. Attenuation of noise for target frequencies was measured through spectrum analysis. Helmholtz resonators were modified by inserting the neck of the resonator inside the resonators cavity. Thus modified Helmholtz resonators are fabricated and fitted with the exhaust manifold. Reduction of noise was investigated through spectrum analysis by the designed Resonators at different loading condition and by varying number of those. Different arrangements of modified Helmholtz resonators were also tested. Exhaust back pressure was measured for modified and non modified Helmholtz resonator. Finally, effect on noise reduction by modified and non-modified Helmholtz resonator and effect of modified Helmholtz resonator and absorptive silencer was analyzed using noise level meter and spectrum analysis.

4.2 Engine for Experiment

For the experiment a diesel engine of Heat Engine Laboratory was used. Calculations were done on the basis of data collected from the engine. Engine specifications are appended below:

Engine brand: Changtuo
Engine model: S1100A2 N

Engine maximum power: 16 hp

Engine rated power: 8 hp

Engine maximum rpm: 2200 rpm

Engine rated rpm (economical): 1515 rpm

Engine rated power (economical): 50% load

Fuel used: Diesel

Number of cylinder: 1

Dynamometer: Hydraulic type

Exhaust manifold inner diameter: 4 cm

4.3 Experimental Setup

Fig. 4.1 shows the experimental setup for finding out noise level and spectrum analysis of the noise generated through engine exhaust using modified Helmholtz resonator. Experimental setup is shown in Fig. 4.1. In Fig B2 of Appendix shows the experimental setup in the laboratory.

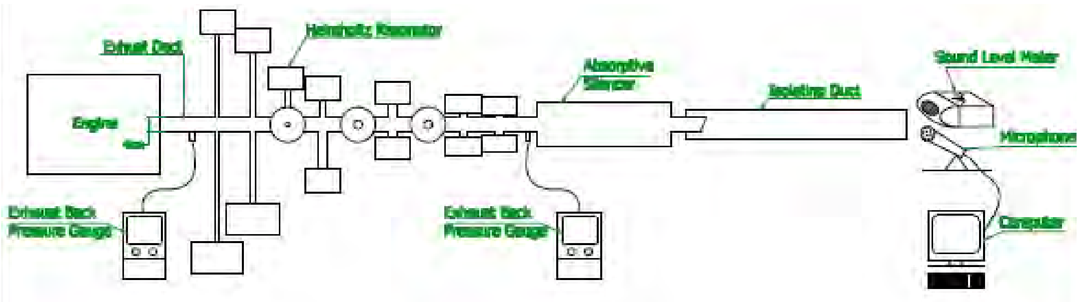

Fig 4.1: Experimental Setup for investigation of noise reduction by Modified Helmholtz resonator

4.4 Calibration of Measuring Equipment

In this experiment microphone is used for the input of spectrum analysis. Before starting the experiment calibration of microphone was carried out. Three microphones are used in this regard. Following features of each microphone is compared with another one and thereby tested.

a. Amplitude linearity

b. Time response

c. Distortion

d. Phase match

4.5 Engine Noise Level Measurement (without Silencer)

Noise generated from a model engine was measured. The sound level meter (Brand: YEW, Type: 3604) was positioned at a distance of one meter away from the exhaust manifold. The meter was positioned at the same level that of flow of exhaust gas so that the noise level can be recorded effectively. It was found that due to other frictional and vibrating parts of engine, pure engine exhaust noise cannot be obtained. Therefore, cumulative noise nearby engine found higher and it was measured 105 db. In this experiment, only exhaust noise is considered to attenuate. Therefore, a duct was fabricated and fitted over the exhaust manifold to isolate exhaust noise from other noises. Maximum noise level without silencer and without isolating duct was found 105 db and with an isolating duct it was found 102 db at 2200 rpm. All the measured noise level without isolating duct, without silencer and with existing silencer is shown in table 1 of Appendix A. All the measured noise level with isolating duct without silencer and with existing silencer is shown in table 2 of Appendix A.

4.6 Spectrum Analysis of Engine Noise for Selection of Target Frequencies

In practice the sound spectrum of an engine exhaust is continually changing, as it is dependent on the engine speed that is continually varying when the engine is being driven. It is impossible to design a resonator that achieves complete attenuation. Noise spectrum variation with varying speed and load makes resonator design quite difficult. In this experiment noise level was measured and noise spectrum was analyzed for three different speeds i.e. for 1435 rpm, 1515 rpm and 2200 rpm at no load, 50% load and 100% load conditions. From the spectrum analysis target frequencies was selected considering above mentioned engine speed and in particular 1515 rpm at 50% loading condition. The band of frequency that contributes to the maximum noise level was found between 50 to1500 Hz. Sixteen dominating frequencies within the range was identified by spectrum analysis. These are 63 Hz, 75 Hz, 125 Hz, 139 Hz, 188 Hz, 214 Hz, 265 Hz, 340 Hz, 401 Hz, 457 Hz, 529 Hz, 595 Hz, 725 Hz, 940 Hz, 1280 Hz, 1419 Hz. Sixteen resonators were designed to attenuate dominating frequencies. Spectrum analysis of noise generated during engine running at 1515 rpm and at 50% load shows that frequencies where the noise level are at peak selected for resonator design.

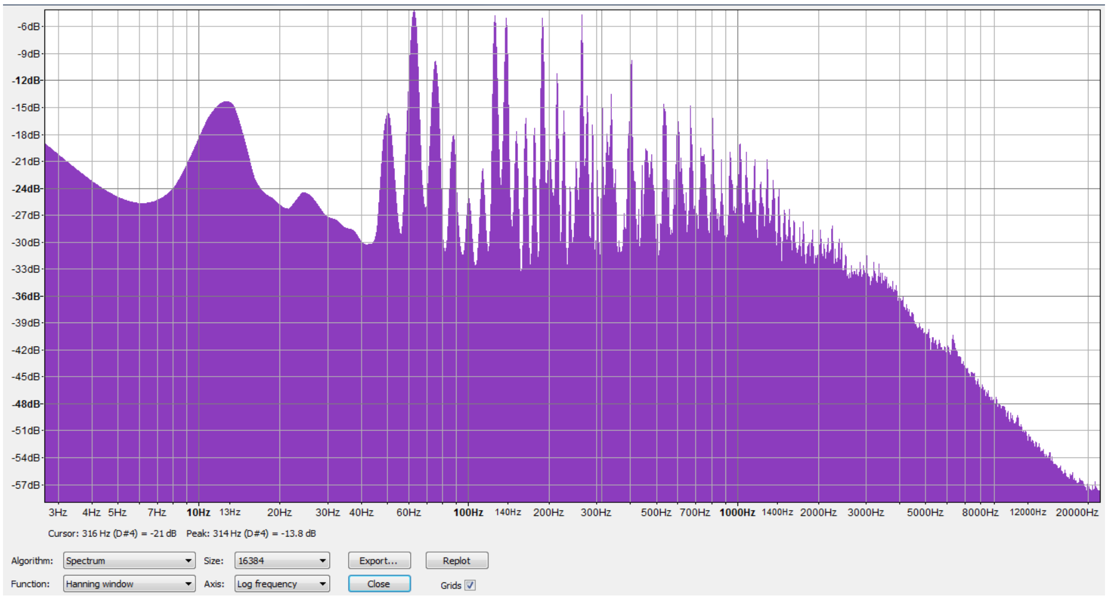

Fig 4.2: Noise spectrum of engine without silencer running at 1515 rpm with 50% load

4.7 Design of the Resonating Chambers

According to the target frequencies obtained from spectrum analysis, resonators are designed for those resonance frequencies using following equation:

$$f = \frac{c}{2\pi}\sqrt{\frac{S}{L \times V}}$$

Where,

V = volume of resonator

S = Cross sectional area of resonator neck

L= Length of resonator neck

c = Sound velocity (increases 0.6 m/sec at 1^0 temperature increase)

Exhaust temperature after manifold=160^0 C

$V = \pi r^2$h, where r is the resonator cavity radius and h is the height of the cavity

$S = \frac{\pi d^2}{4}$, where d is the resonator neck diameter

The silencer using Helmholtz resonators has many design parameters i.e. volume of cavity V, neck length of resonator *L*, neck hole diameter of resonator *d*. If all design parameters are considered, the silencer model becomes very complex. Therefore, it is necessary to minimize the design parameters. Here *d, r, s* can be fixed by geometric shape. Therefore, above equation may be expressed for finding out neck length:

$$L = \frac{c^2 d^2}{4\pi^2 \ f^2 \, r^2 h}$$

For 125 Hz target frequency, resonator geometry is shown as follows:

f = 125 Hz

d= 0.0127 m

r= 0.05 m

h= 0.066 m

c = 428 m/sec

L = ?

Putting values in the equation we get,

$$L = \frac{428^2 0.0127^2}{4\pi^2 \; 125^2 \, 0.05^2 0.066}$$

L= 0.29 m

Similarly, neck length was calculated for other 15 Helmholtz resonators. Detail design parameters are shown in Appendix A, table 3. Designed resonators are shown in fig. 4.3 and 4.4 and fabricated accordingly.

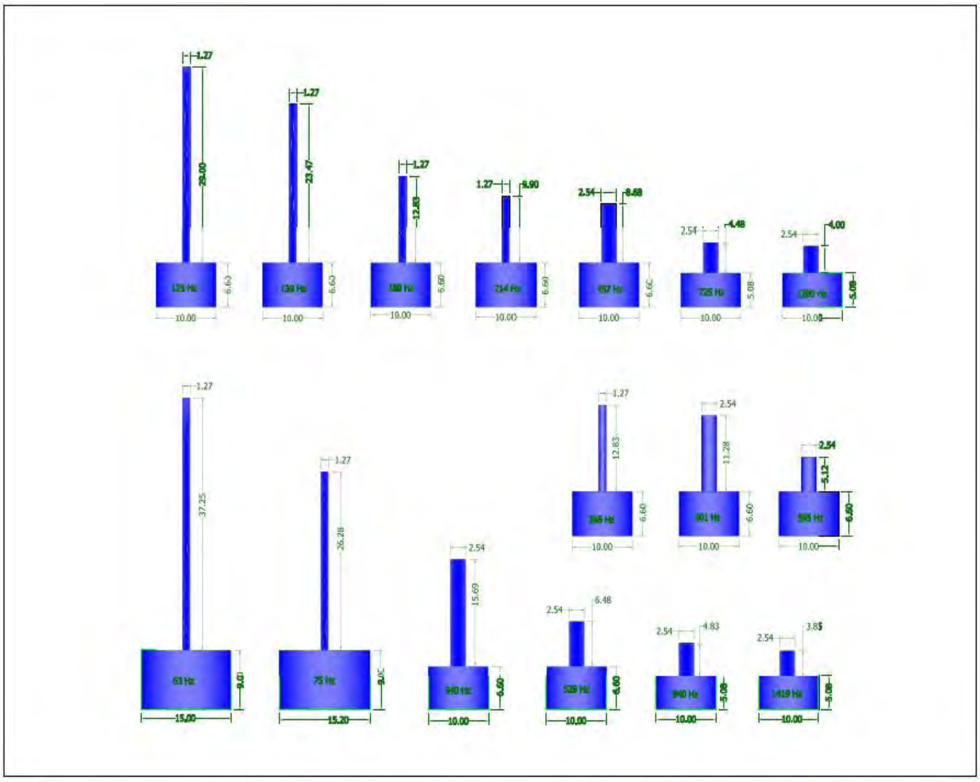

Fig 4.3: Designed 16 x Helmholtz resonators against target frequencies

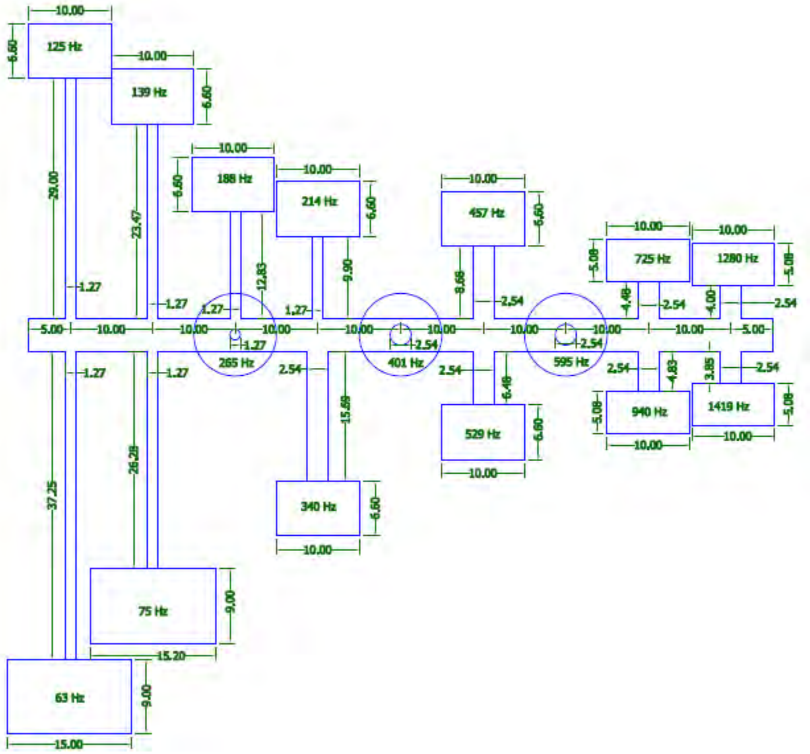

Fig 4.4: Arrangement of designed non-modified Helmholtz resonator

4.8 Distances between Resonators

Seo et. al [14] determined the optimal distance between resonators hole to obtain a high TL in the objective frequency band by equivalent impedance analysis. The distance between resonators hole can be determined as:

$$L_i = \frac{\frac{\lambda_i}{4} + \frac{\lambda_{i+1}}{4}}{2} \quad and \;\; \lambda = \frac{c}{f}$$

Here, λ is in m, L is in m, c is in m/s and f is in Hz

$$= \frac{c\left(\frac{1}{f_i} + \frac{1}{f_{i+1}}\right)}{8}$$

$$= \frac{c\left(\frac{1}{f_1}+\frac{1}{f_2}\right)}{8}$$

$$= \frac{428\left(\frac{1}{63}+\frac{1}{75}\right)}{8}$$

$$= 1.56\ m$$

When the distances between resonators are λ/4, the TL has a higher value in the objective frequency band. But the silencer will have a long length because of the wavelength relatively long. Seo et. al [14] shows that considering the compactness of silencer hole to hole distance may be kept 100 mm and it doesn‟t have serious impact in attenuation of noise. Therefore, considering the compactness distances between resonators were kept 100 mm. Fig. 4.5 and 4.6 show the distance between resonators.

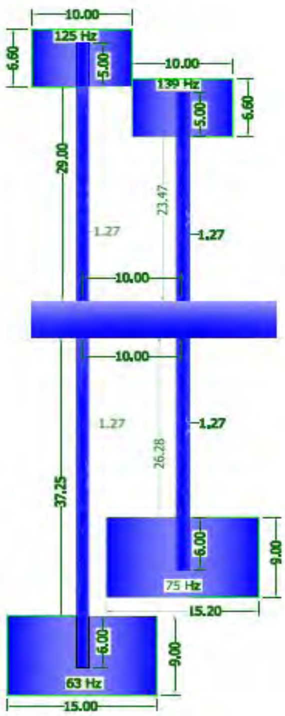

Fig 4.5: Distance between Helmholtz resonators

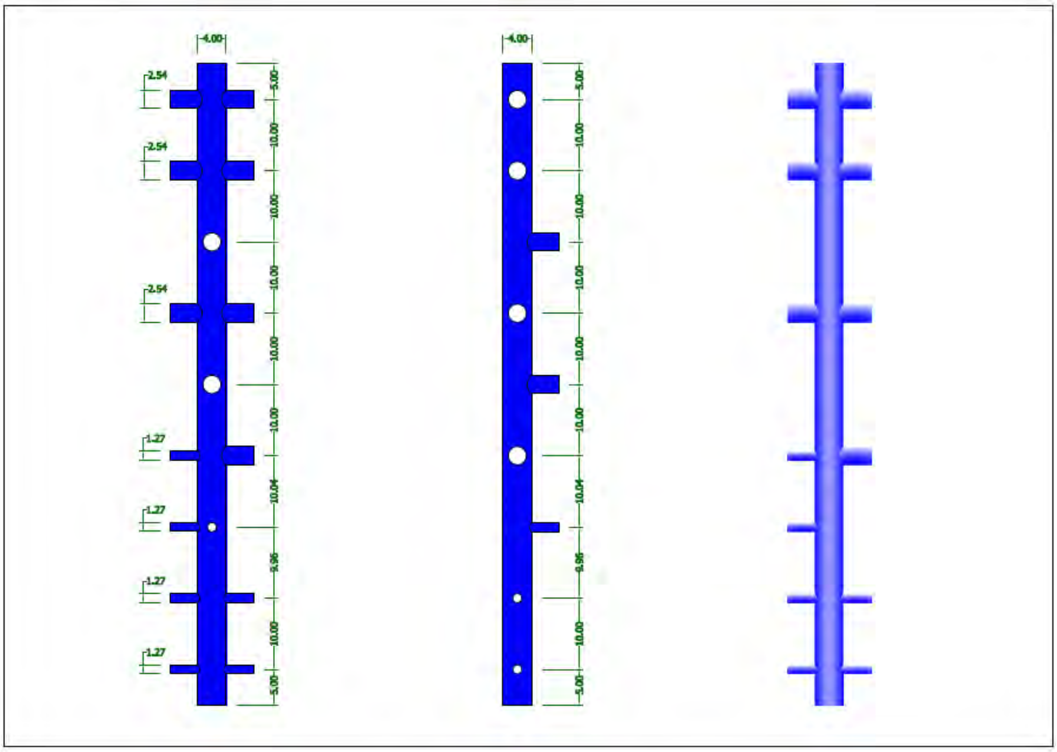

Fig 4.6: Manifold for attachment of Helmholtz resonators

4.9 Modification of Helmholtz Resonator

Helmholtz resonator was modified by inserting the neck inside the cavity. Such modification was carried out by making thread in cavity end of the neck. Impact of modified Helmholtz resonator was measured and attenuation characteristic was compared with actual Helmholtz resonator. Modified Helmholtz resonator is shown below fig. 4.7 and 4.8.

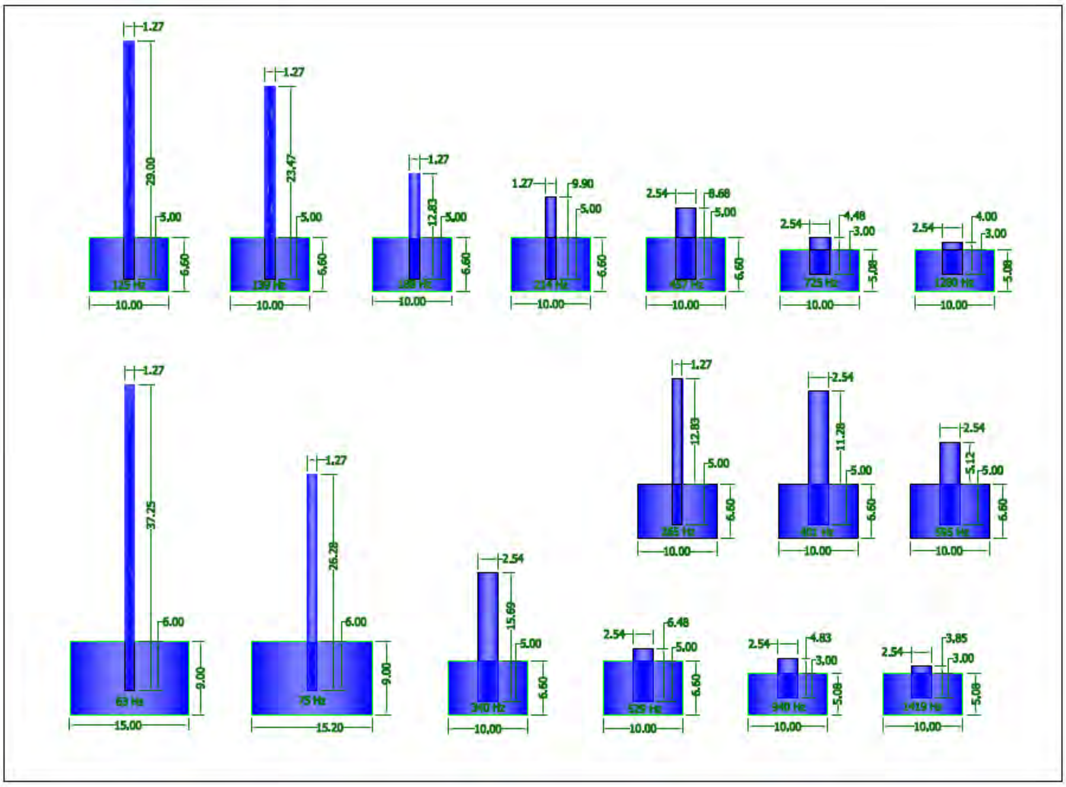

Fig 4.7: Diagram of Modified Helmholtz resonator separately

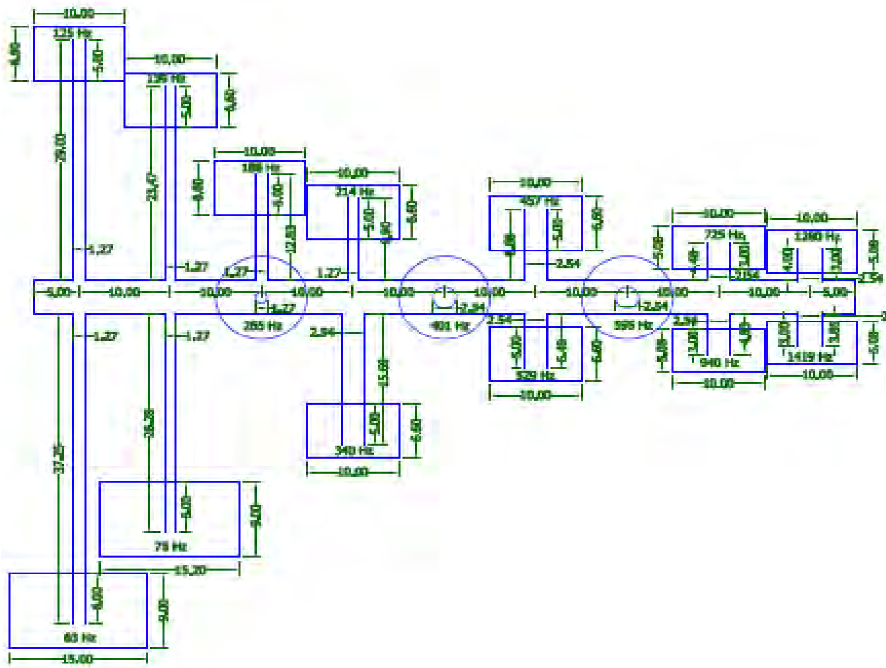

Fig 4.8: Arrangement of Modified Helmholtz resonator attached with exhaust pipe

4.10 Fitting of an Isolating Duct and Calibration of Designed Helmholtz Resonator

Initially, Helmholtz resonator was designed based on the noise nearby the engine. By spectrum analysis, it was found that the noise generated is discrete in nature. Different moving parts of the different system and echo produced from those created such discrete noise. Therefore, accurate data could not be measured and it was very difficult to get exact frequencies responsible for exhaust noise. In this regard, data measured was not accurate and initial designed Helmholtz resonator was also not accurate. Thereafter, it was required to isolate the exhaust noise from other engine noise to deal with only exhaust noise. To overcome such problem an isolating duct was fabricated and fitted over the exhaust manifold to isolate exhaust noise from other noises. All the readings were taken at the exit of the isolating duct. Based on the isolated noise the Helmholtz resonator was designed finally and thereby calibrated also.

4.11 Tailpipe Length Determination

The section of pipe downstream of the final resonator will have acoustic resonances that can amplify engine noises if they match. Resonances can be avoided by keeping the length of the tailpipe less than ½ wavelengths at the tone frequency. Even better size of the tailpipe can be made exactly ¼ wavelengths i.e. $\lambda/4$ to cancel the tone. Wave length calculation against target frequencies is shown in table 3 of Appendix A. From calculation minimum wavelength was found 0.3 m and maximum 6.79 m. Therefore, tail pipe length to be in between 0.075 m to 1.70 m. Therefore, considering compactness and ease of fabrication tail pipe length was kept 0.10 m.

4.12 Absorptive Silencer Design

The absorptive silencer is the classic dissipative design, deriving its noise control properties from basic fact that noise energy is effectively absorbed by various types of fibrous packing materials. As the sound waves pass through the spaces between the tightly packed, small diameter fibers often absorptive material, the resulting viscous friction dissipates the sound energy as small amounts of heat.

Absorptive silencers are highly effective on high frequency noise (over 1000Hz). At frequencies below this range attenuation performance progressively diminishes with common absorptive materials unless special design considerations are implemented. Since noise is absorbed by the acoustic packing media, absorptive silencers generally employ straight through or annular internal designs, which impose very little restrictions on air flow. Typically, the greater the ratio of packing surface area to flow area, the greater is attenuation capability of the silencer. Many different packing materials can be used in absorptive silencers and are chosen for use based on varying absorptive performance, price, temperature and corrosion characteristics.

The effect of the thickness of absorptive material and spacing play an important role in sound attenuation. The attenuation increases sharply at high frequencies as the spacing is narrowed. Better performance at lower frequency is obtained as the thickness of the absorbing material is increased.

In order to attenuate high frequency noise, a metal tube surrounded by acoustical-quality mineral wool inside the silencer outer containment shell has been used here. The sides of the tube are perforated that permit sound waves impinge on the absorbing materials. According to „ASHRAE Technical committee 2.6‟ range of chamber length:

$$\frac{Length(inch)}{Pipe\ diameter(inch)} = \frac{10}{16}$$

$$\frac{Length}{1.5} = \frac{10}{16}$$

$$Length = 15\ to\ 24\ inch = 38\ cm\ to\ 61\ cm$$

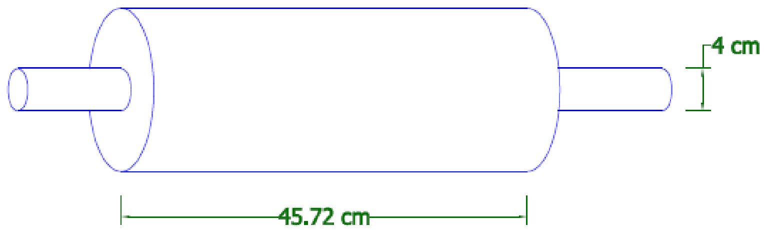

Fig 4.9: Diagram of absorptive type silencer

Considering the compactness and ease of fabrication the length was kept 45.72 cm.

CHAPTER 5
RESULT AND DISCUSSION

5.1 Measurement of Noise with Non-Modified Helmholtz Resonator

As per the experimental set up readings were taken at the exit of the duct. Initially, noise levels were taken using non-modified Helmholtz resonator. Thereafter, spectrum analysis was carried out for different arrangement at different rpm and loading condition.

Maximum noise level at 1515 rpm and 50% load without silencer found 90 dB. Thereafter, total 16 resonators were fitted with the exhaust manifold as shown in the fig. 5.1

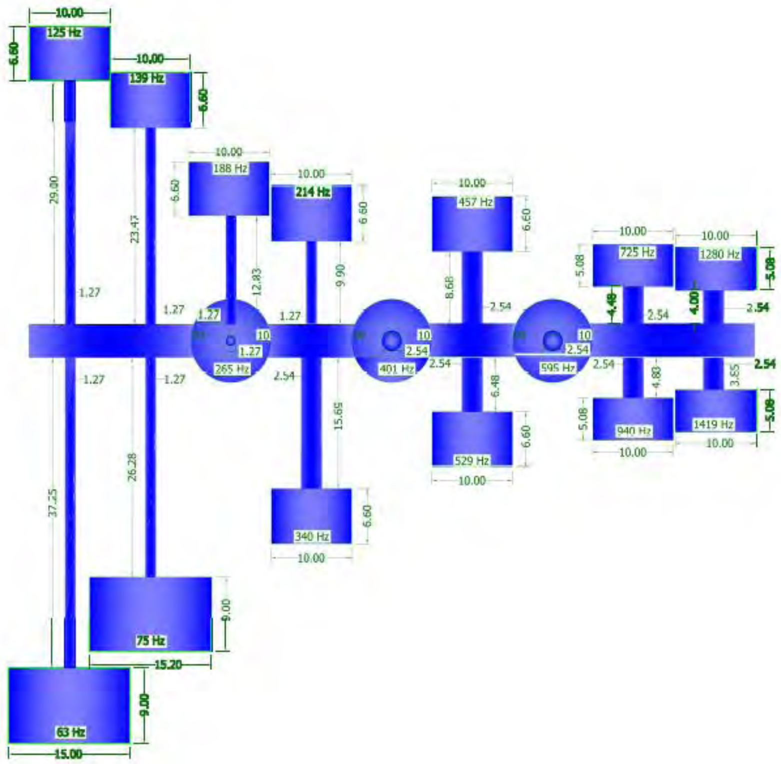

Fig. 5.1: Non-modified Helmholtz resonator arrangement in the exhaust pipe

After attaching 16 in number non-modified Helmholtz resonators, noise level at 1515 rpm with 50% load found 72 db and thereby insertion loss was found 18 db. Detail measured noise level using 16 in number non-modified Helmholtz resonators are shown in table 4 of Appendix A. Spectrum analysis shows that by attenuating noise level of 16 in number target frequencies by 16 in number Helmholtz resonators overall noise level was reduced significantly (fig. 5.2). Noise spectrum with 16 in number Helmholtz resonators and noise spectrum graph of without silencer and with Helmholtz resonator during running engine at 1515 rpm with 50% load is shown in fig. 5.3.

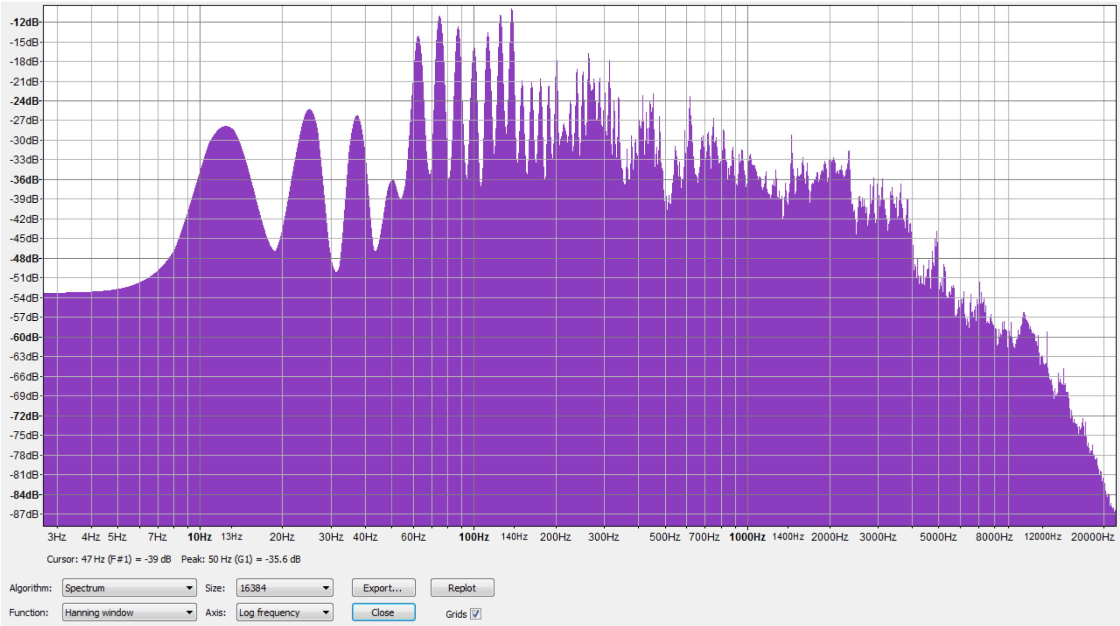

Fig 5.2: Noise spectrum of engine during running at 1515 rpm with 50% load attaching 16 in number non-modified Helmholtz resonators.

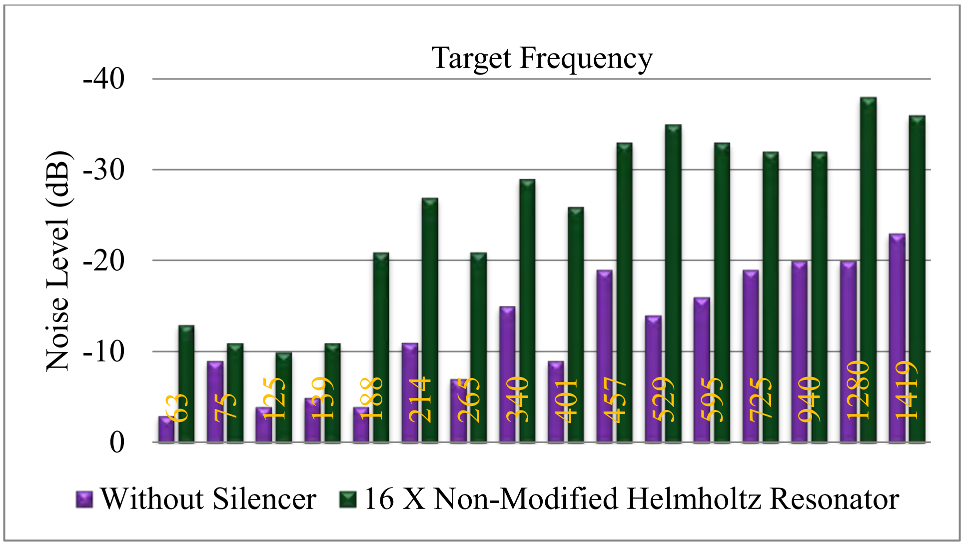

Fig 5.3: Comparative Noise graph of target frequencies during engine running at 1515 rpm without silencer and with 16 in number non-modified Helmholtz resonator

5.1.1 Measurement of Noise in Series and Parallel Arrangement of Non-Modified Helmholtz Resonator

Maximum noise level without silencer found 90 dB at 1515 rpm at 50% load. Now 6 in number non-modified Helmholtz resonators (63 Hz, 75 Hz, 340 Hz, 529 Hz, 940 Hz, 1419 Hz) were fitted with the exhaust manifold in series and parallel as shown in the fig. 5.4 and 5.5. After attaching same 6 in numbers non-modified Helmholtz resonators in series and in parallel maximum noise level was found in both case 78 db and thereby insertion loss was found 12 db at 1515 rpm with 50% loads. Detail noise level measured with 6 in numbers non-modified Helmholtz resonator in parallel and series arrangement are shown in table 5 and 6 of Appendix A. The result shows that the arrangement order of non modified Helmholtz resonator is not significant in noise reduction. Spectrum analysis with series arrangement during running engine at 1515 rpm and 50% load is shown in fig. 5.6. Spectrum analysis of target frequencies is shown in fig. 5.7.

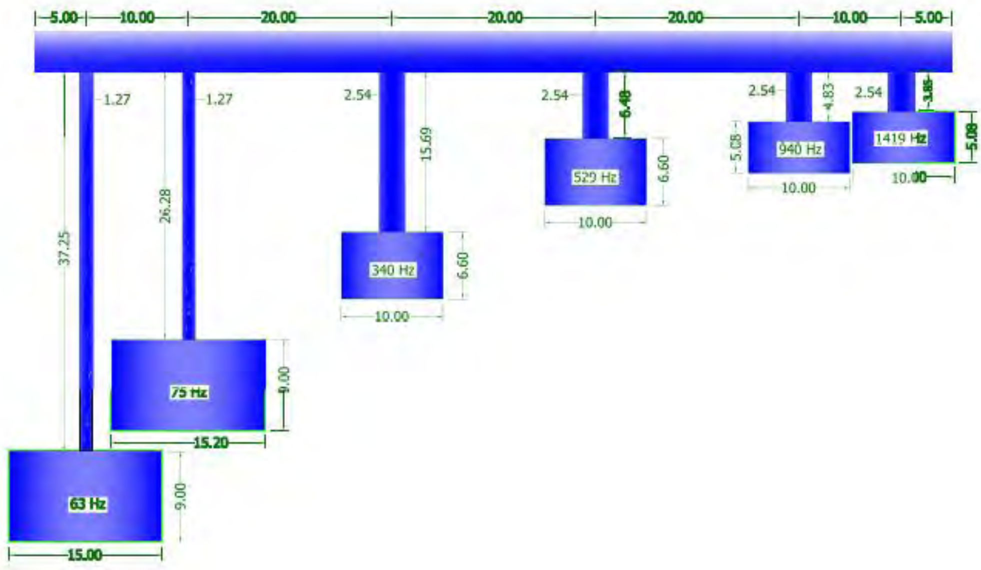

Fig 5.4: Series arrangement of 6 x non modified Helmholtz resonator

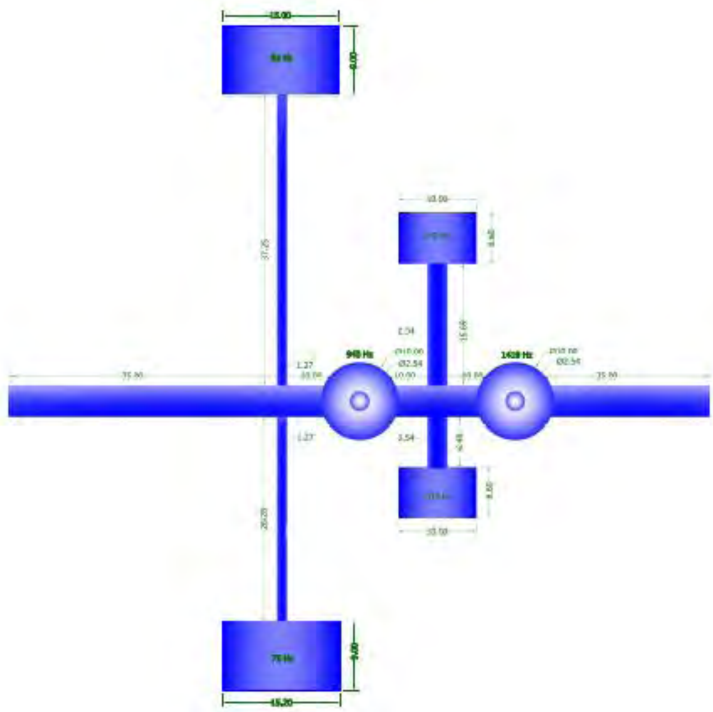

Fig 5.5: Parallel arrangement of 6 x Non modified Helmholtz resonator

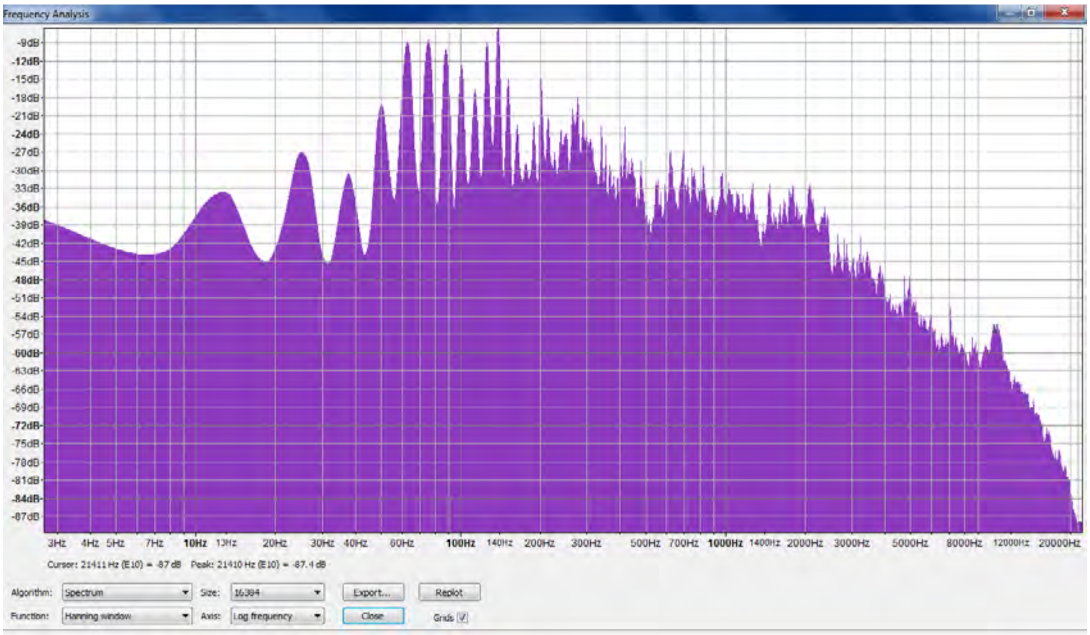

Fig 5.6: Noise spectrum of engine during running at 1515 rpm with attaching 6 non-modified Helmholtz resonators in series.

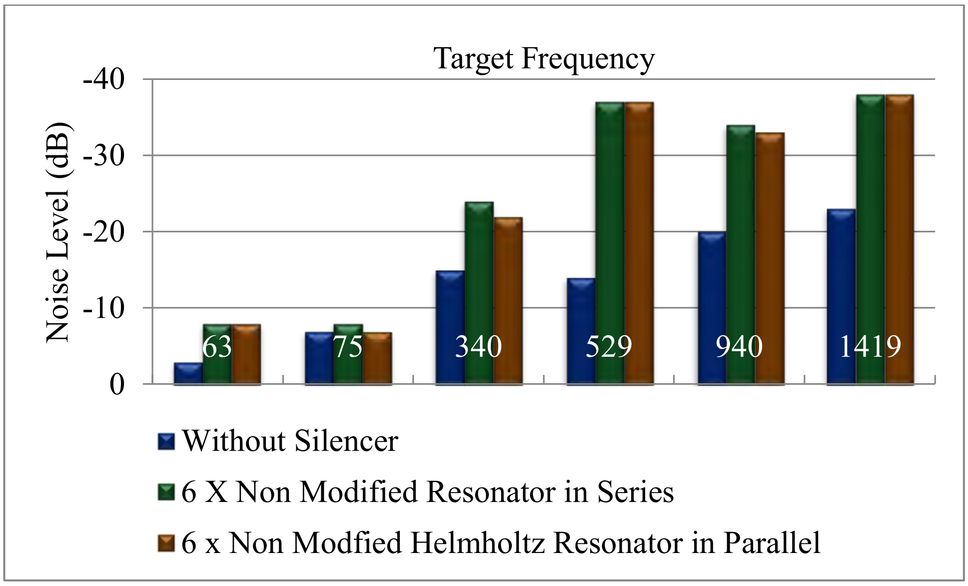

Fig 5.7: Spectrum analysis of target frequencies without silencer, with 6 x non modified Helmholtz resonator

5.2 Measurement of Noise with Modified Helmholtz Resonator

With the same experimental set up now 16 x Helmholtz resonators are modified by inserting the neck gradually inside the cavity as shown in fig. 5.8. All necks connected with the cavity were made threaded so that all the necks can be gradually inserted. Effects on noise for such modification were measured. Total 16 modified Helmholtz resonators were fitted with the exhaust manifold as shown in the fig. 5.9.

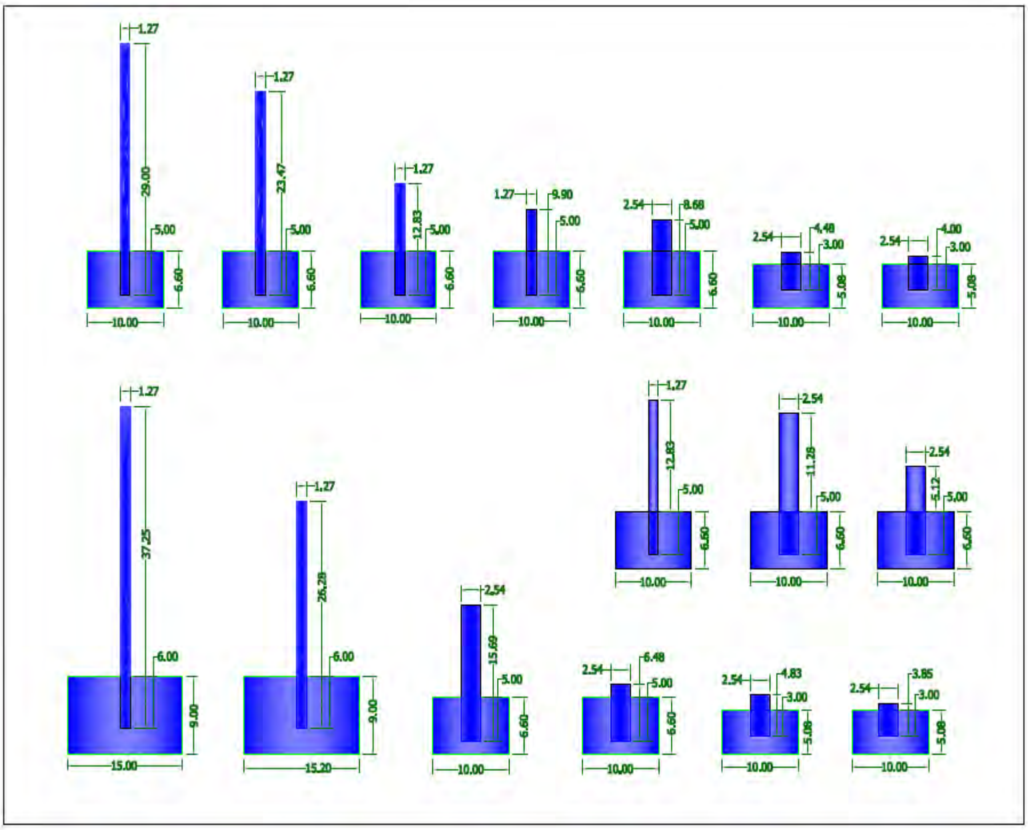

Fig 5.8: Modified Helmholtz resonator

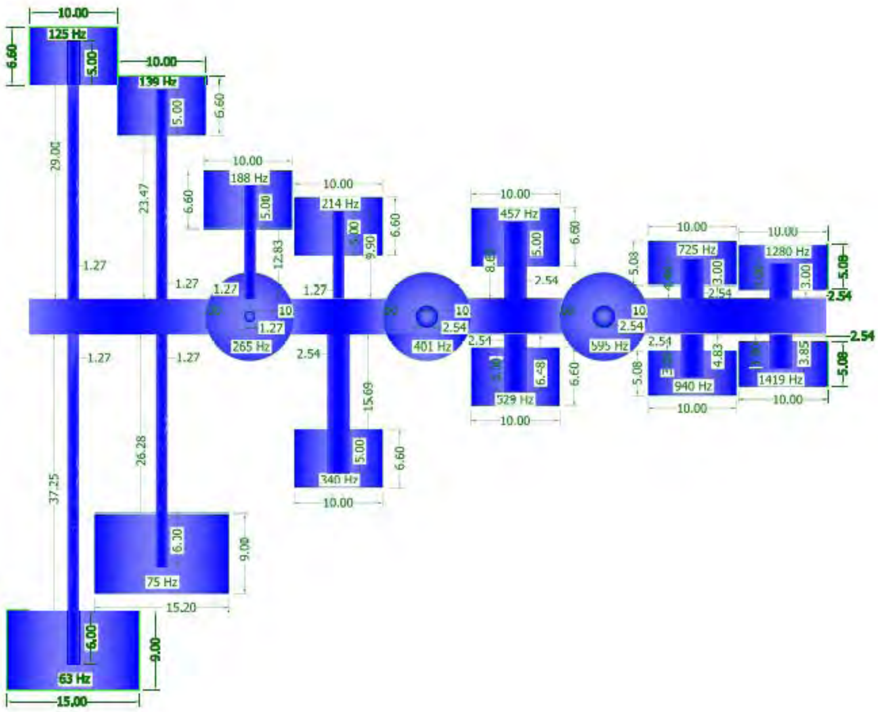

Fig 5.9: Modified Helmholtz resonator arrangement

After attaching modified Helmholtz resonators, noise level was found 73 dB at 1515 rpm with 50% load whereas noise level without silencer was 90 dB. Thereby insertion loss was found 17 db. Detail noise level measured attaching modified Helmholtz resonators are listed in table 7 of Appendix A. Spectrum analysis shows that by attenuating noise level of 16 target frequencies by 16 modified Helmholtz resonators overall noise level could be reduced but performance was better with non-modified Helmholtz resonator. Detail spectrum analysis for 16 modified Helmholtz resonators in fig. 5.10 shows that with non-modified Helmholtz resonator maximum target frequencies noise level was better compared with modified one.

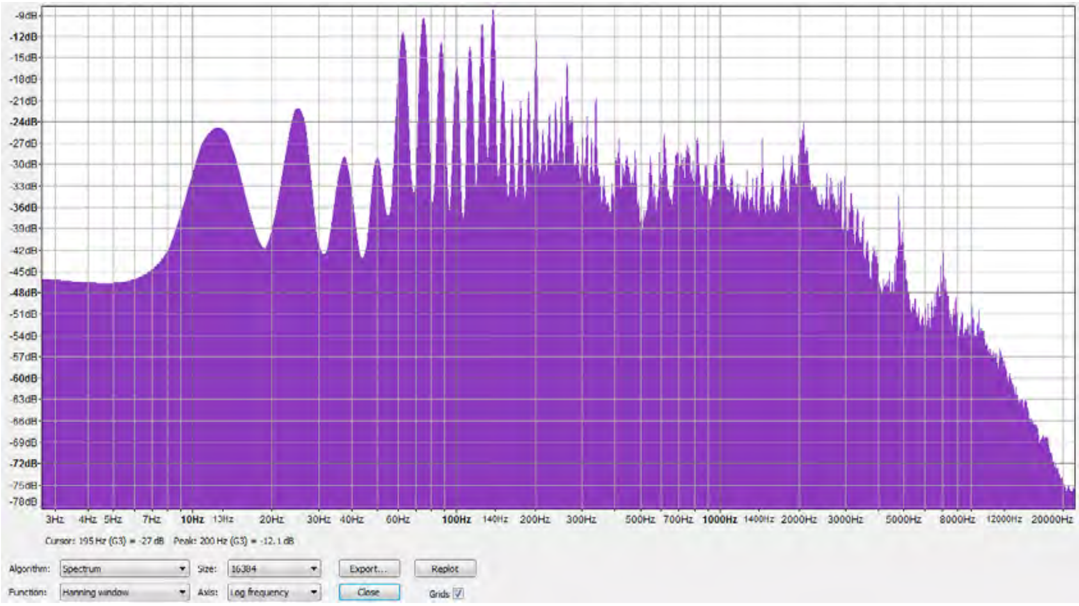

Fig 5.10: Spectrum analysis during running engine at 1515 rpm and 50% load by attaching 16x modified Helmholtz resonators.

5.2.1 Measurement of Noise with Parallel and Series Arrangement of Modified Helmholtz Resonator

Total 6 modified Helmholtz resonators (63 Hz, 75 Hz, 340 Hz, 529 Hz, 940 Hz, 1419 Hz) were fitted in parallel and in series as shown in fig. 5.11.

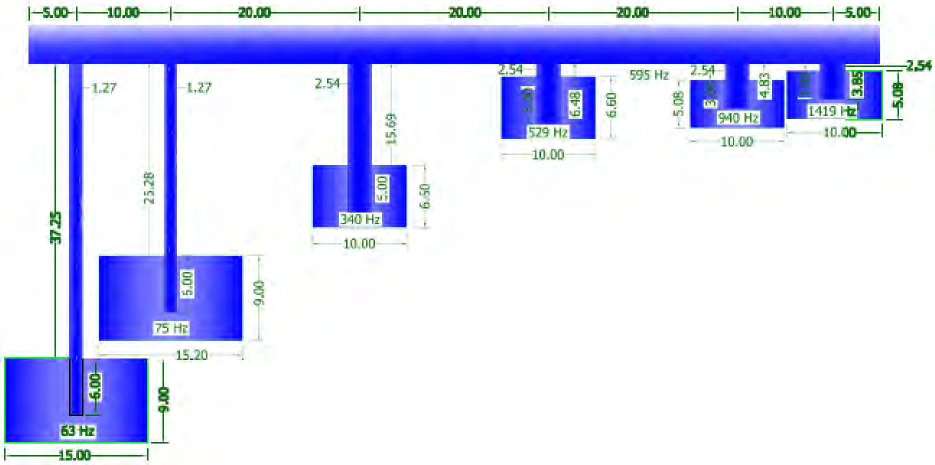

Fig 5.11: Series arrangement of 6 x modified Helmholtz resonator

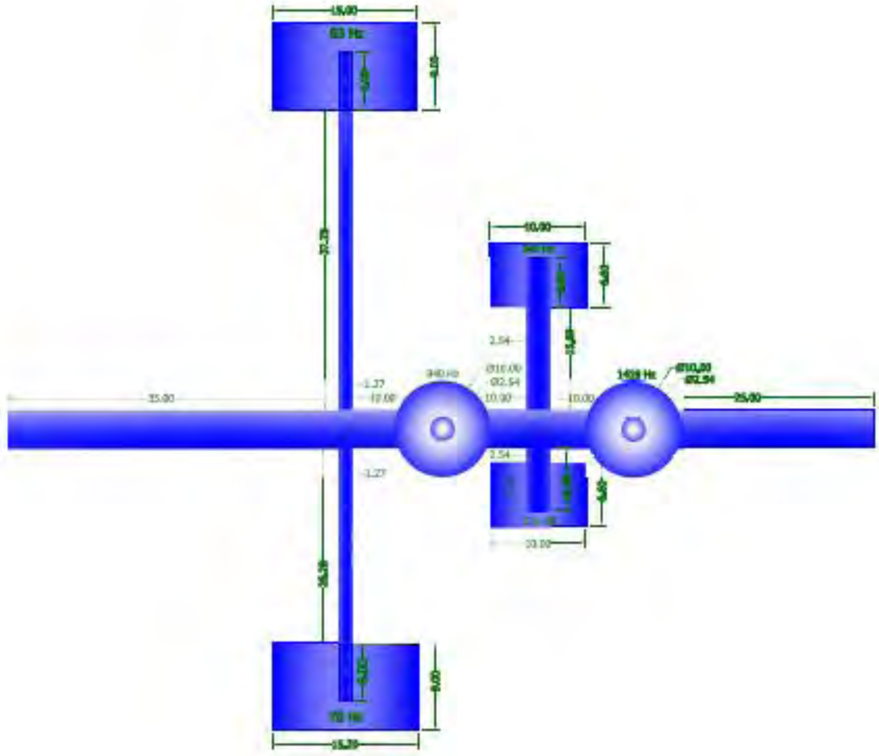

Fig 5.12: Parallel arrangement of 6 x modified Helmholtz resonator

Noise level was found 79 db at 1515 rpm with 50% load whereas noise level without silencer was 90 db. Thereby insertion loss was found 11 db. Detail noise level measured with modified resonator in series and parallel was shown in table 8 and 9 of Appendix A. The result shows that the arrangement order is not significant in noise reduction. Frequency spectrum analysis with series arrangement of 6 in number modified resonator during running engine at 1515 rpm and 50% load is shown in fig. 5.13. Comparative curve of series and parallel arrangement of modified Helmholtz resonator is shown in fig. 5.14. Comparative frequency spectrum of 6 in number non-modified and 6 in number modified Helmholtz resonator in series is shown in fig. 5.15.

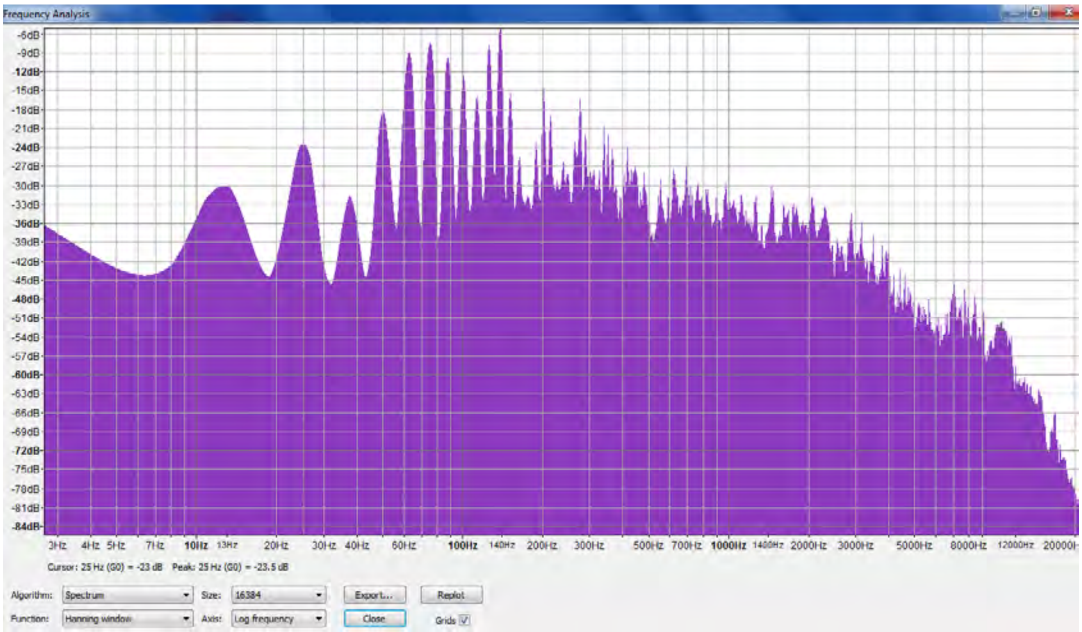

Fig 5.13: Spectrum analysis of series arrangement with 6xModified Helmholtz Resonator

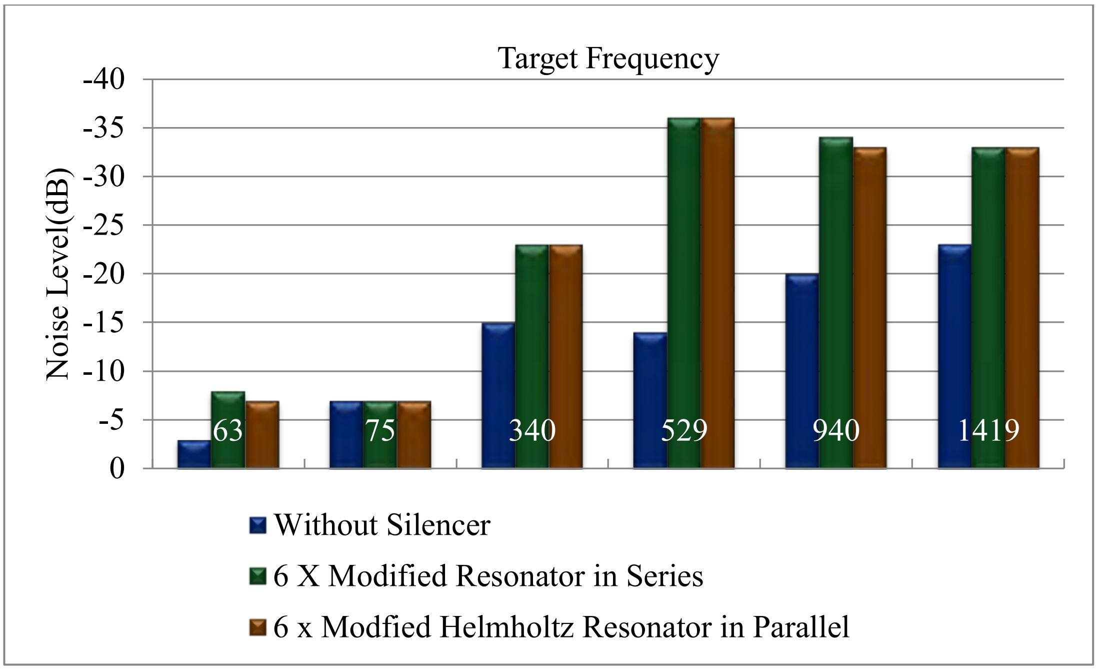

Fig 5.14: Comparative Frequency spectrum graph of series and parallel arrangement

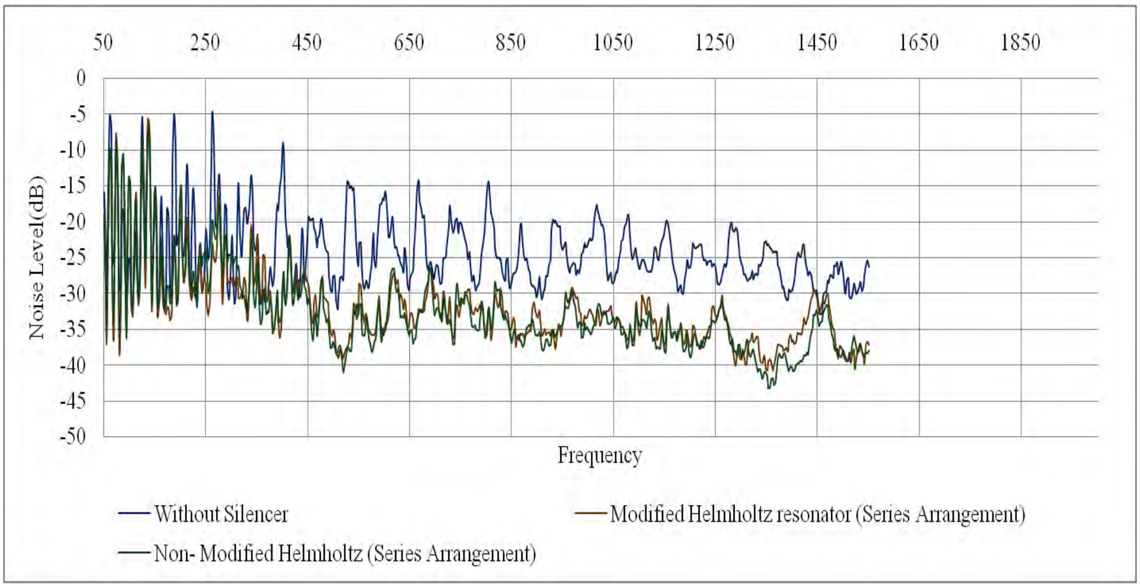

Fig 5.15: Comparative frequency spectrum analysis of 6 in number modified and non-modified Helmholtz resonator in series arrangement

5.3 Measurement of Noise by Varying Number of Modified Helmholtz Resonator

Noise levels were measured by varying the number of Helmholtz resonator. Initially, by removing 3 in nos Helmholtz resonator noise level found 74 db at 1515 rpm and with 50% load (fig. 5.16). Thereafter, 6 x resonators were removed and noise level found 76 db at same rpm and load. Finally, total 10 in nos Helmholtz resonators were removed and noise level was found 79 db at same rpm and load (fig. 5.17). Therefore, it is observed that with the increase of resonator, noise level decreases. Detail noise level at different resonator number is shown in table 10 and 11 of Appendix A.

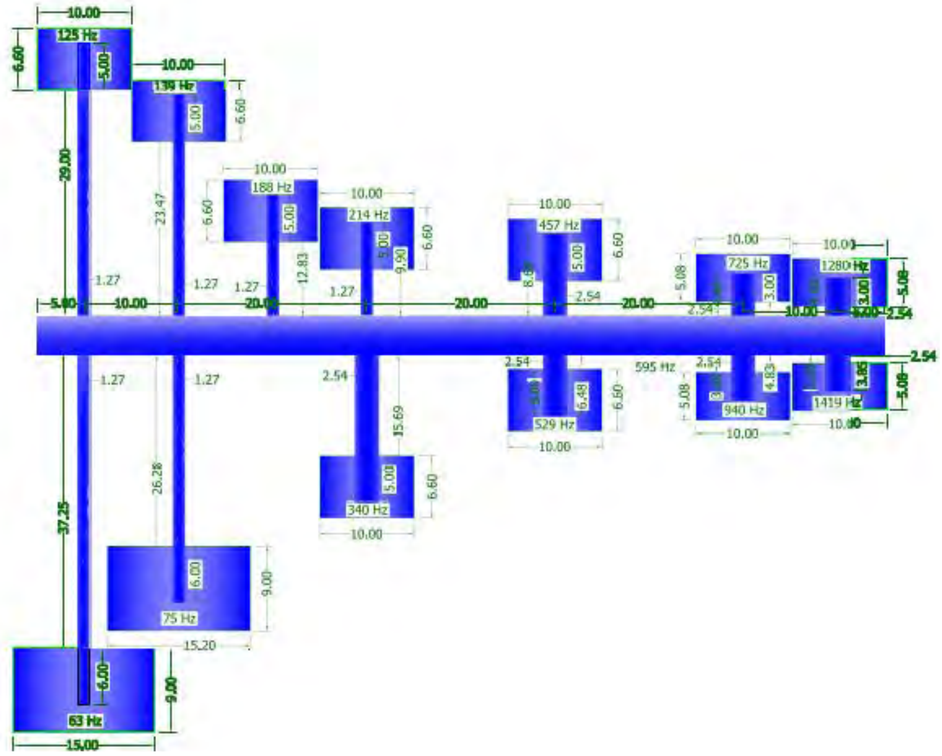

Fig 5.16: Arrangement of 13 x modified Helmholtz resonator

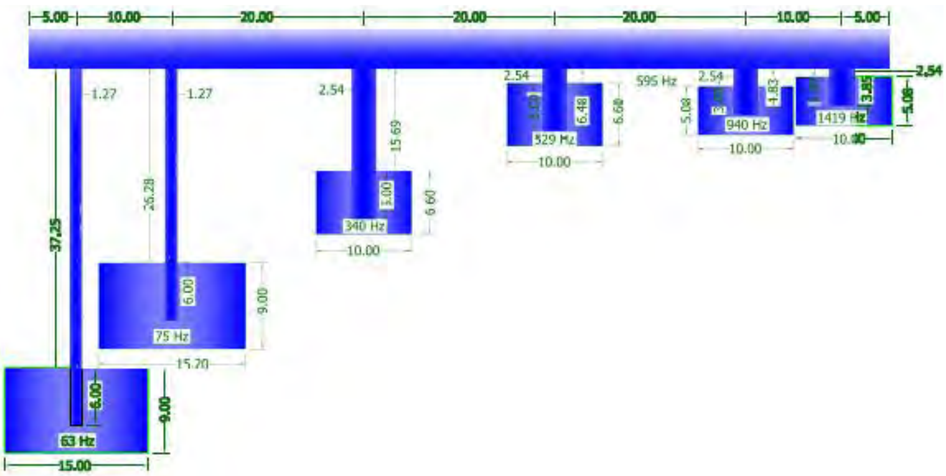

Fig 5.17: Arrangement of 6 x modified Helmholtz resonator

5.4 Investigation on Noise Reduction: Non-modified Helmholtz Resonator vis-à-vis Modified Helmholtz Resonator

It has been found through spectrum analysis and noise level meter that noise attenuation is slightly less in modified Helmholtz resonator compared to non modified Helmholtz resonator. Comparative spectrum analysis of noise level using non-modified Helmholtz resonator and modified Helmholtz resonator is shown in fig. 5.18 and fig. 5.19. In fig. 5.18 it is shown that unsilenced engine noise curve level is highest and thereafter modified Helmholtz resonator curve and finally lowest curve by using non-modified Helmholtz resonator.

The reason for slightly less attenuation with modified Helmholtz resonator is perceived that due to the insertion of neck inside the cavity, cavity volume has been reduced. Therefore, when air is forced into the cavity, the pressure in the cavity is increased and air pushed out in advance. As a result, resonance frequency of the Helmholtz resonator changes from the target frequency. Another reason may be perceived that while air gets inside the cavity through inserted neck, it is impeded by the cavity end. Therefore, resonant frequency is slightly deviated due to modification. However, the attenuation variation is considered negligible because of modification. Therefore, while fabricating Helmholtz resonator, there will not be any significant problem if there is little outcropping or a flange in the neck.

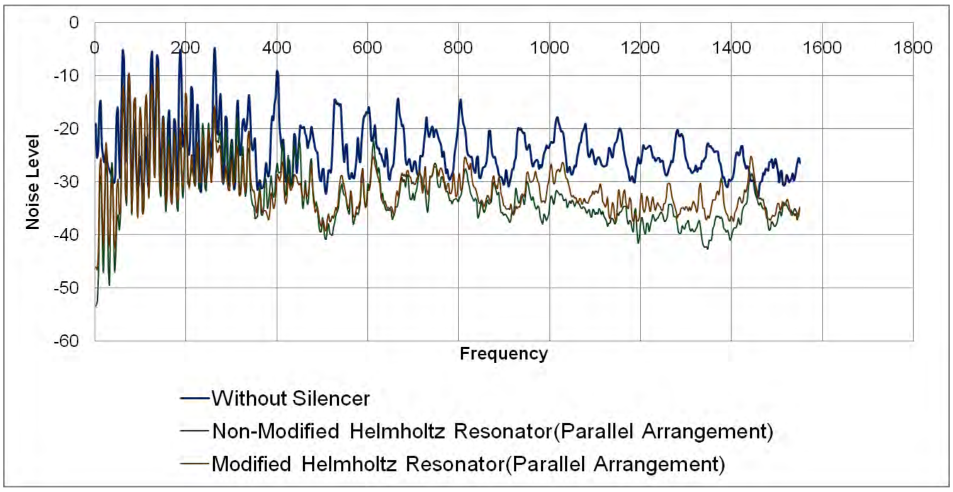

Fig 5.18: Comparative spectrum analysis of noise level using non-modified Helmholtz resonator and modified Helmholtz resonator

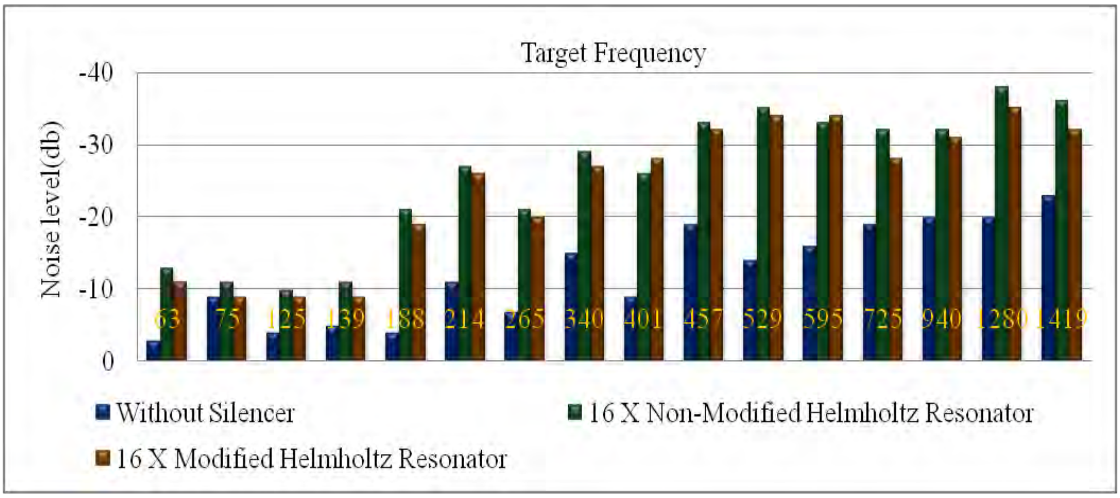

Fig 5.19: Comparative spectrum analysis of target frequencies using non-modified Helmholtz resonator and modified Helmholtz resonator

5.5 Effect on Noise Reduction: Helmholtz Resonator vis-à-vis Absorptive Silencer

Noise spectrum analysis shows that when non-modified and modified Helmholtz resonator is attached, low and medium frequency noise (up to 1800 Hz) significantly reduced (fig. 5.20 and 5.21). Attaching absorptive silencer, it is found that noise frequency above 700 Hz absorptive type silencer attenuation performance is better (fig. 5.20 and 5.22).

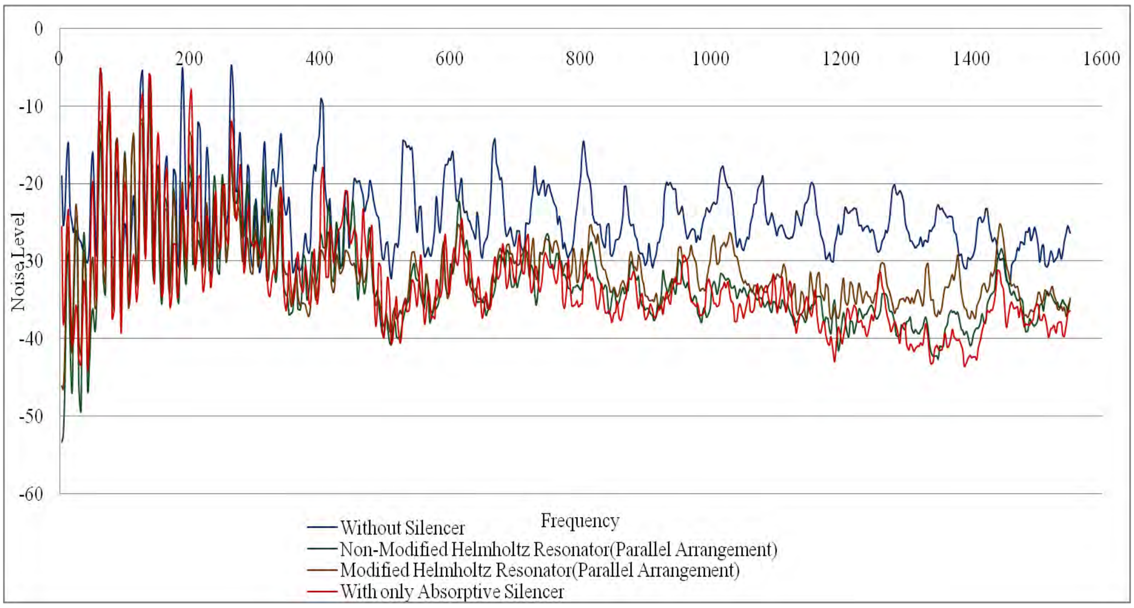

Fig 5.20: Comparative noise spectrum (up to 1500 Hz) of engine without silencer, with Helmholtz resonator and with Absorptive

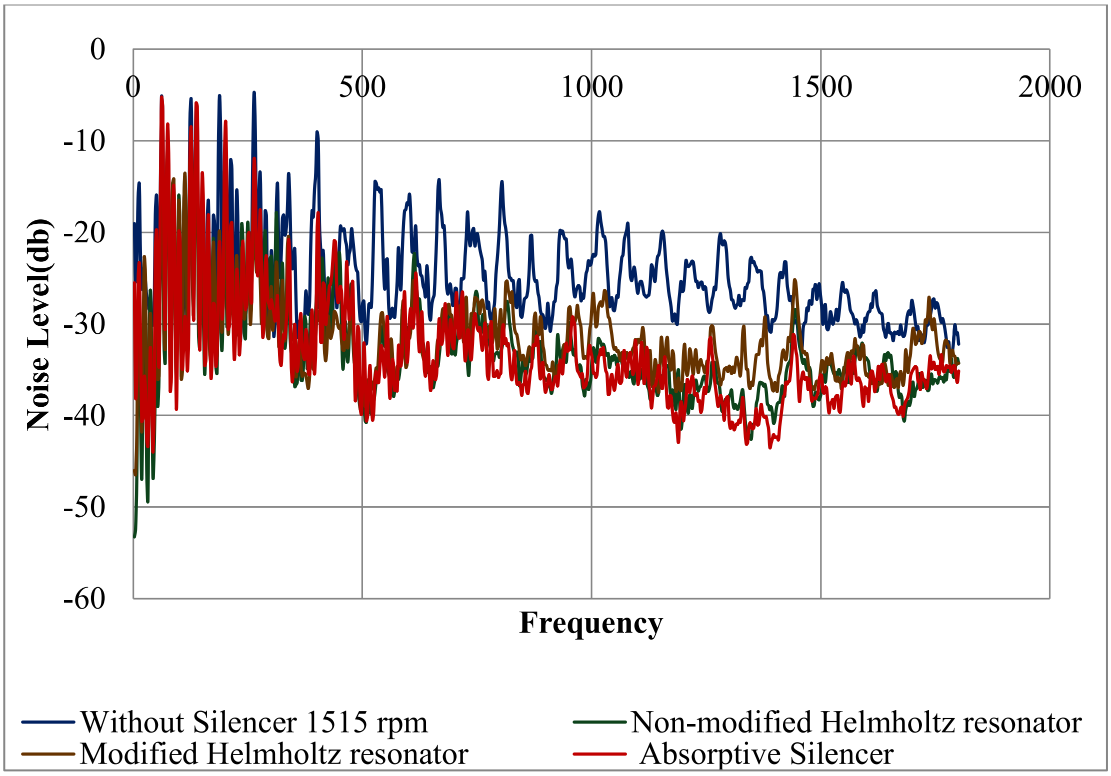

Fig 5.21: Comparative noise spectrum of engine (up to 1800 Hz) without silencer, with Helmholtz resonator and with Absorptive silencer

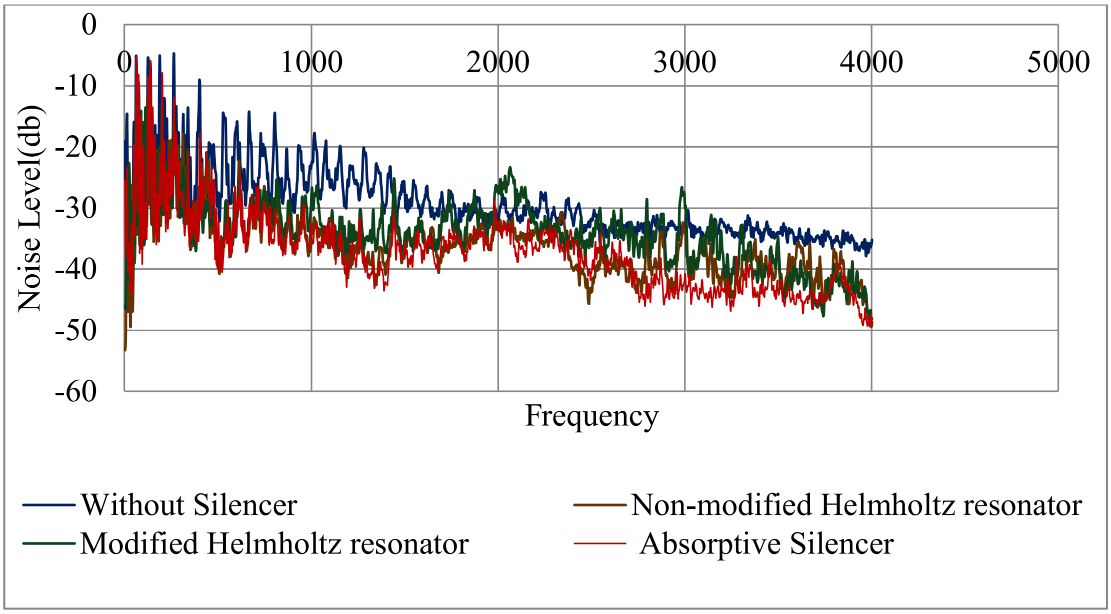

Fig 5.22: Comparative noise spectrum of engine (up to 4000 Hz) without silencer, with Helmholtz resonator and with Absorptive silencer

5.6 Comparison of Noise Level with Modified Helmholtz Resonator at Different Engine Speeds

Experiment was carried out with Helmholtz resonator at 1515 rpm with 2200 rpm. And the graph shows that attenuation performance at 1515 rpm is better than 2200 rpm. As the resonator is designed in particular for 1515 rpm, performance found better.

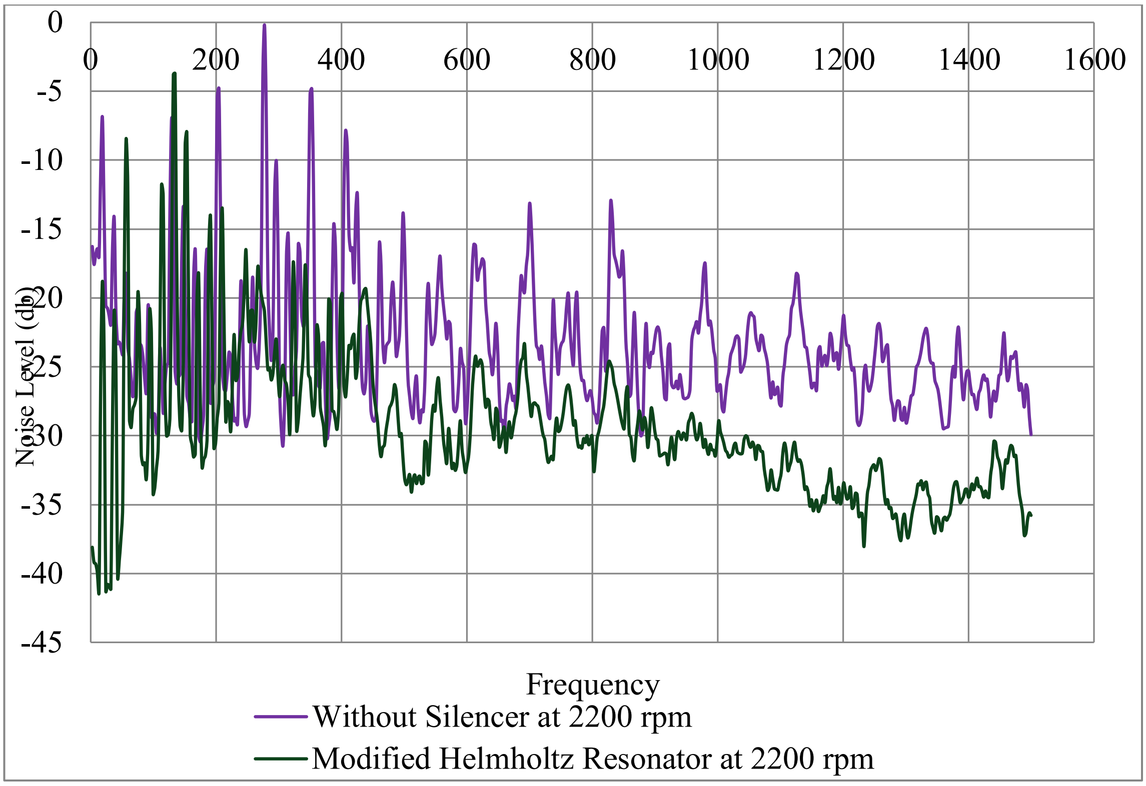

Fig 5.23: Frequency spectrum analysis of engine noise running at 2200 rpm

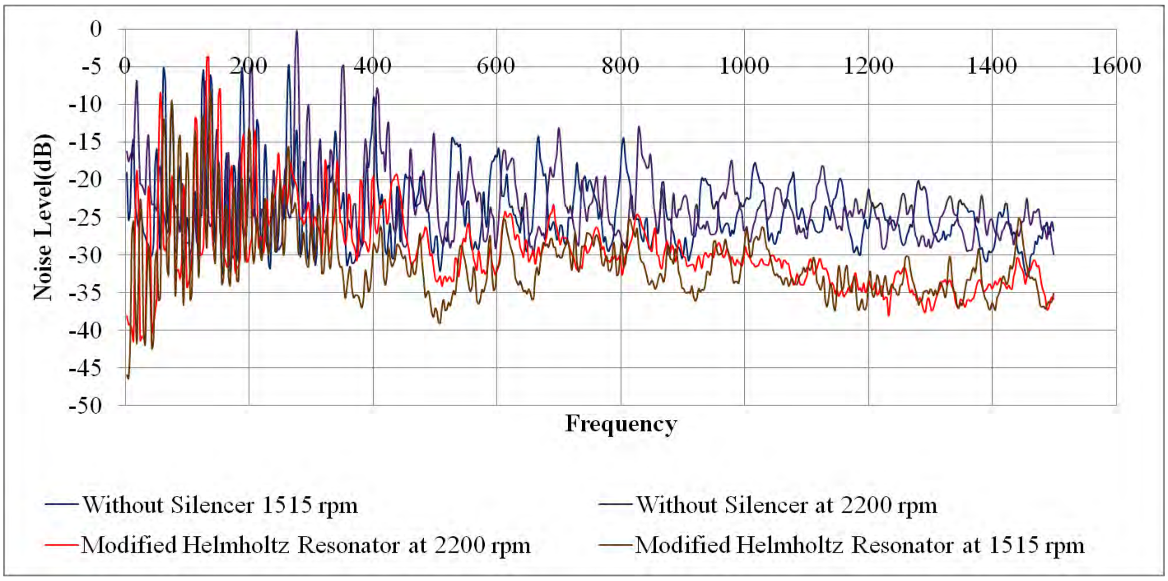

Fig 5.24: Comparative Frequency spectrum analysis of engine noise running at 2200 and 1515 rpm

5.7 Measurement of exhaust back pressure

Exhaust back pressure is very important in designing resonators as high exhaust back pressure reduces engine output. Therefore, exhaust back pressure was measured by a digital differential pressure gauge for the designed non-modified and modified Helmholtz resonator (Fig. B3, Appendix B). It was found maximum 14 mbar for non modified Helmholtz resonator and 17 mbar for modified Helmholtz resonator at full load and at maximum rpm. However, while attaching isolating duct over exhaust manifold the exhaust back pressure increases 3.5 mbar more. It has been observed that at lower rpm and at lower load exhaust back pressure is not significant. But at higher rpm and load exhaust back pressure is high. Beside Exhaust back pressure is higher in modified Helmholtz resonator. As the neck is inserted inside the cavity exhaust flow is impeded in the cavity end. Therefore, exhaust flow is not smooth with the modified Helmholtz resonator and thereby exhaust back pressure is higher. However, the designed Helmholtz resonators exhaust back pressure is within the limit (40 mbar). Exhaust back pressure in different loading condition and rpm is given in the table 12 of Appendix A.

CHAPTER 6
CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The aim of this thesis is to experimentally investigate the reduction of noise of modified Helmholtz resonator by spectrum analysis. Therefore, Helmholtz resonators were designed and modified those by inserting neck inside the cavity. Noise level was measured using modified and non modified Helmholtz resonator for parallel and series arrangement running engine at different rpm. Frequency Spectrum Analysis was done by Audacity software to find out the real time performance of modified and non-modified Helmholtz resonator and different arrangement of those. Following conclusions are drawn investigating the noise reduction by modified Helmholtz resonator:

1. 16 frequencies that contribute to the peak noise level were selected by frequency spectrum analysis of noise produced from running designated engine. The band of frequency that contributes to the maximum noise level was found between 50 to 1500 Hz.

2. Engine running at 1515 rpm without silencer maximum noise level found 90 dB and 102 dB at maximum 2200 rpm.

3. 16 Helmholtz resonators were designed, fabricated and arranged in a single manifold to test with the designated engine.

4. After attaching 16 non-modified Helmholtz resonator noise level was found 72dB engine running at 1515 rpm with 50% load. Therefore, insertion loss found 18 dB.

5. After attaching 16 modified Helmholtz resonator noise level was found 73 dB at 1515 rpm with 50% load. Therefore, insertion loss found 17dB.

6. Noise reduction is slightly less in modified Helmholtz resonator compared to non-modified Helmholtz resonator.

7. The change in volume of the Helmholtz resonator cavity influences its resonance characteristics. The reason for slightly less noise reduction with modified Helmholtz resonator is perceived that due to insertion of neck inside the cavity, cavity volume has been reduced. As a result, resonance frequency of the Helmholtz resonator changed from the target frequency. Another reason may be perceived that while air gets inside the cavity through inserted neck, it is impeded by the cavity end. The attenuation variation due to modification may be considered negligible. Therefore, it may be concluded that while fabricating Helmholtz resonator, there will not be any significant problem if there is little outcropping or a flange in the neck.

8. Using Helmholtz resonator, low and medium frequency noise (up to 1500 Hz) significantly reduced. However noise frequency above 700 Hz absorptive type silencer attenuation performance is better.

9. Parallel and series arrangement of Helmholtz resonator do not have any significant effect on noise reduction. However, considering compact shape and size parallel arrangement is advantageous.

10. Helmholtz resonators are used to reduce noise in the narrow frequency band. Combining many resonators in series and parallel broadband characteristics can be obtained.

11. Measuring exhaust back pressure it was found maximum 14 mbar for non-modified Helmholtz resonator and 17 mbar for modified Helmholtz

resonator at full load and at maximum rpm. It has also been observed that at lower rpm and at lower load exhaust back pressure is not significant. But at higher rpm and load exhaust back pressure is high. Exhaust back pressure is also higher in modified Helmholtz resonator. As the neck is inserted inside the cavity exhaust flow is impeded in the cavity end. Therefore, exhaust flow is not smooth with the modified Helmholtz resonator and thereby exhaust back pressure is higher. However, the designed Helmholtz resonators exhaust back pressure (17 mbar) is within the maximum allowable limit of 40 mbar.

6.2 Recommendations

For further study in relation to the present work the following recommendations are provided below:

1. The study has been done using engine noise. As engine noise fluctuates rapidly due to other factors, it is very difficult to get constant same noise. Therefore performance of modified Helmholtz resonator may be evaluated using separate sound source in an encapsulated duct.

2. Using a straight through perforated tube wrapping with absorbing material as a neck of the Helmholtz resonator. Such modified Helmholtz resonator noise reduction performance may be analyzed.

3. In this experiment neck is inserted inside the cavity and noise attenuation level was measured in this regard. Furthermore, inserted neck can be perforated and thereby attenuation for such modification may be investigated.

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Appendix A

Table 1: Engine noise without and with existing reactive silencer without isolating duct

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without silencer(db)	Noise Level with existing silencer(db)	Insertion Loss(db)
Without Isolating Duct	0% Load	1435 1515 2200	90 92 95	85 87 90	5 5 5
	50% Load	1435 1515 2200	95 96 102	86 88 93	9 8 9
	100% Load	1435 1515 2200	98 102 105	88 89 94	10 13 11

Table 2: Engine noise without and with existing reactive silencer with isolating duct

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without silencer(db)	Noise Level with existing silencer(db)	Insertion Loss(db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	82 82 85	4 5 5
	50% Load	1435 1515 2200	89 90 97	83 83 88	6 7 9
	100% Load	1435 1515 2200	93 95 102	84 86 91	9 9 11

Table 3: Helmholtz resonator's Dimensions

Vel of Sound	Resonance Frequency	Neck Diameter	Cavity radius	Height of Cavity	Length of Neck	Wave length
c (m/s)	f (Hz)	d (m)	r (m)	h (m)	L (m)	λ (m)
428	63	0.0127	0.075	0.09	0.37247	6.79
	75	0.0127	0.075	0.09	0.26281	5.71
	125	0.0127	0.05	0.066	0.29029	3.42
	139	0.0127	0.05	0.066	0.23476	3.08
	188	0.0127	0.05	0.066	0.12833	2.28
	214	0.0127	0.05	0.066	0.09904	2.00
	265	0.0127	0.05	0.066	0.06459	1.62
	340	0.0254	0.05	0.066	0.15695	1.26
	401	0.0254	0.05	0.066	0.11283	1.07
	457	0.0254	0.05	0.066	0.08687	0.94
	529	0.0254	0.05	0.066	0.06483	0.81
	595	0.0254	0.05	0.066	0.05125	0.72
	725	0.0254	0.05	0.0508	0.04485	0.59
	940	0.0254	0.05	0.0508	0.02668	0.46
	1280	0.0254	0.05	0.0508	0.01439	0.33
	1419	0.0254	0.05	0.0508	0.01171	0.30

Table 4: Engine noise with 16 x Non-Modified Helmholtz resonator

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	72 72 77	14 15 13
	50% Load	1435 1515 2200	89 90 97	72 72 80	17 18 17
	100% Load	1435 1515 2200	93 95 102	74 75 81	19 20 21

Table 5: Engine noise with 6 x Non-modified Helmholtz resonator in parallel

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	76 77 82	10 10 8
	50% Load	1435 1515 2200	89 90 97	77 78 83	12 12 14
	100% Load	1435 1515 2200	93 95 102	80 81 86	13 14 16

Table 6: Engine noise with 6 X Non-modified Helmholtz resonator in series

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	76 77 82	10 10 8
	50% Load	1435 1515 2200	89 90 97	77 78 83	12 12 14
	100% Load	1435 1515 2200	93 95 102	80 81 86	13 14 16

Table 7: Engine noise with 16 x Modified Helmholtz resonators

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	72 73 78	14 14 12
	50% Load	1435 1515 2200	89 90 97	73 73 79	16 17 18
	100% Load	1435 1515 2200	93 95 102	75 76 82	18 19 20

Table 8: Engine noise with 6 X Modified Helmholtz resonator in parallel

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	77 78 83	09 09 7
	50% Load	1435 1515 2200	89 90 97	78 79 84	11 11 13
	100% Load	1435 1515 2200	93 95 102	81 82 87	12 13 15

Table 9: Engine noise with 6 in numbers Modified Helmholtz resonator in Series

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435 1515 2200	86 87 90	77 78 83	09 09 7
	50% Load	1435 1515 2200	89 90 97	78 79 84	11 11 13
	100% Load	1435 1515 2200	93 95 102	81 82 87	12 13 15

Table 10: Engine noise with 13 X Modified Helmholtz resonator

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435	86	73	13
		1515	87	74	13
		2200	90	79	11
	50% Load	1435	89	74	15
		1515	90	74	15
		2200	97	80	17
	100% Load	1435	93	77	16
		1515	95	78	17
		2200	102	83	17

Table 11: Engine noise with 10 X Modified Helmholtz resonator

Set Up condition	Engine Load %	Speed (rpm)	Noise Level without Silencer(db)	Noise Level with Helmholtz resonators(db)	Insertion Loss (db)
With isolating Exhaust Duct	0% Load	1435	86	74	12
		1515	87	75	12
		2200	90	80	10
	50% Load	1435	89	75	14
		1515	90	76	14
		2200	97	81	16
	100% Load	1435	93	78	15
		1515	95	79	16
		2200	102	84	16

Table 12: Exhaust Back Pressure of Designed Helmholtz resonator

Engine Load (kg)	Speed (rpm)	Exhaust back pressure with non-modified Helmholtz resonator(mbar)	Exhaust back pressure with modified Helmholtz resonator(mbar)
0 (0% Load)	1500	3.0	3.5
	1600	3.2	3.7
	1700	3.4	3.8
	1800	3.5	3.9
	1900	3.7	4.2
	2000	3.7	4.5
	2100	3.8	4.8
	2200	4.0	5.0
3 kg (50% Load)	1500	4.5	4.5
	1600	5.0	6.0
	1700	5.4	6.5
	1800	6.0	7.1
	1900	6.5	7.5
	2000	7.0	8.0
	2100	7.5	8.6
	2200	8.0	9.0
5 kg (100% Load)	1500	8.0	9.0
	1600	10	12
	1700	10.5	13
	1800	11	14
	1900	12	15
	2000	12.5	15.5
	2100	13	16
	2200	14	17

Appendix B

Fig B-1 Modified Helmholtz resonator fabrication in the workshop

Fig B-2: Experimental set up in Heat Engine Laboratory, BUET

Fig B-3: Measurement of exhaust back pressure by Differential pressure gauge

Fig B-4: Sound Level Meter

An Experimental Investigation on Noise Reduction by Using Modified Helmholtz Resonator

by

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Department of Mechanical Engineering

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May, 2014

CERTIFICATE OF THESIS APPROVAL

The thesis titled "An Experimental Investigation on Noise Reduction by Using Modified Helmholtz Resonator", Submitted by Md. Shahidullah Al Faruq, Roll no: 1009102019, Session: October-2009, has been accepted as satisfactory in partial fulfillment of the requirement for the degree of Master of Science in Mechanical Engineering on 11th May, 2014.

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CANDIDATE’S DECLARATION

It is hereby declared that this thesis or any part of it has not been submitted elsewhere for the award of any degree or diploma.

Md. Shahidullah Al Faruq

Contents

List of Tables

List of Figures

List of Abbreviations of Technical Symbols and Terms

Unit		Symbol
Noise, decibel	=	dB
Force, Newton	=	F
Acoustic mass of the resonator, kg	=	m
Acceleration, m/sec^2	=	a
Atmospheric pressure, bar	=	P_0
Angular frequency, rad/sec	=	ω
Length of neck, m	=	L
Frequency, Hz	=	f
Sound pressure, kg/cm^2	=	P
Density of air, kg/m^3	=	ρ
Volume of resonator cavity,m^3	=	V
Speed of sound, m/sec	=	c
Transmission Loss, dB	=	TL
Acoustic impedance , N s/m^3	=	Z
Velocity, m/sec	=	v
Mach number	=	M
Volume velocity, m/sec	=	*U*

Wave number	=	k
Height of the cavity, m	=	h
Cross sectional area of the cavity,m^2	=	S_c
Neck length, m	=	l
Acoustic impedance of a resonator's cavity, N s/m^3	=	Z_c
Impedance of resonator, N s/m^3	=	Z_r
Hole impedance of a resonator, N s/m^3	=	Z_h
Cross sectional area of resonator neck, m^2	=	S
Resonator neck diameter, m	=	d
Resonator cavity radius, m	=	r
Sound Wave length, m	=	λ
Displacement, m	=	x
Temperature, T	=	^0C