Effect of Length on Sound Reduction of the Expansion Type Muffler by Experiment and Flow Simulation

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Abstract—Investigate the acoustic performance of a range of different muffler as typically used in automobile represent by different technique. Muffler performance with particular engine specification was conducted using the FEM method for typical muffler elements which was also correlated to experimental results. Different parameter selected for this study with significantly affected. A number of different muffler systems are tested and their acoustic performance predicted by sound measurement techniques. A number of parameters are investigated including diameter of main pipe, length of main pipe, length of tail pipe, flow, engine load.

Key words: Muffler, Noise, IC Engine, Solid work, Measurements, Design

I. INTRODUCTION

For an automotive design engineer, exhaust noise from the internal combustion engines is one of the primary concerns. It is here that mufflers find their application as a well-designed muffler not only reduces the noise from exhaust systems but also contributes towards building a reputation about the vehicle. The performance of an exhaust system is assessed by a number of factors; the two most important being the backpressure and the attenuation of the system. High back pressure in an exhaust system affects the performance of the engine, decreasing power and increasing fuel consumption, and hence emissions; Exhaust noise is a large contributor to traffic noise, a significant source of noise pollution. An exhaust system must therefore achieve somewhat conflicting goals of low backpressure and high attenuation, while also taking into account cost, manufacturing, materials, weight, and space issues. Good design of the muffler should give the best noise reduction and offer optimum back pressure for the engine.

Day by day, noise pollution increase in our environment due to increase of number of vehicle. Numbers of cylinders per engine are also increases which are given growth in noise. Most of the advances in theory of acoustic equipment and exhaust mufflers have been developed in last decades. For the same power rating, diesel engines are noisier than gasoline engines, since the combustion characteristics of diesel engines produce more harmonics than slower combustion of gasoline.

Exposure to noise causes detrimental effects on neuroendocrine, cardiovascular, respiratory and digestive systems. Chronic exposure to noise causes fatigue and interferes with concentration, thus reducing work efficiency.[1]

A. Noise Standards in India

Noise limit for diesel generator sets (up to 1000 KVA) manufactured on or after the 1stJanuary, 2005.

The maximum permissible sound pressure level for new diesel generator (DG) sets with rated capacity up to 1000 KVA, manufactured on or after the 1st January, 2005 shall be 75 dB(A) at one meter from the enclosure surface.

The Acceptable indoor noise levels for various types of buildings as per the Indian Standard IS: 4954-1968 [2] and Indian ambient noise standards as specified by the Central Pollution Control Board, India, and are detailed in Table 1.[1]

<table>
<thead>
<tr>
<th>Area Code</th>
<th>Category of Area</th>
<th>Limit in dB(A) , Leq</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Day time</td>
</tr>
<tr>
<td>A</td>
<td>Industrial Area</td>
<td>75</td>
</tr>
<tr>
<td>B</td>
<td>Commercial area</td>
<td>60</td>
</tr>
<tr>
<td>C</td>
<td>Residential area</td>
<td>55</td>
</tr>
<tr>
<td>D</td>
<td>Silence Zone</td>
<td>50</td>
</tr>
</tbody>
</table>

Table 1: Indian Ambient Noise Standards [1]

Note:
1) Day time is reckoned in between 6 a.m. and 9 p.m.
2) Night time is reckoned in between 9 p.m. and 6 a.m.
3) Silence zone is referred to as areas up to 100 meters around such premises as hospitals. The Silence zones are to be declared by the Competent Authority. Use of vehicular horns, loudspeakers and the bursting of crackers shall be banned in these zones.
4) Mixed categories of areas should be declared as one of the four above mentioned categories by the Competent Authority and the corresponding standards shall apply.

B. Types of Muffler

1) Reactive Mufflers

The performance of a reactive muffler is determined by its internal geometry. This will define the attenuation characteristic of the muffler that can range from sharply tuned narrow band attenuation to broadband attenuation across wide frequency bands. The one or more chambers, resonators, or finite sections of pipe that are collectively make up a reactive muffler provide an impedance mismatch as shown in figure 1. This impedance mismatch results in a reflection of part of the incident acoustic energy back towards the source of the sound or back and forth between chambers, where it is eventually dissipated. The cut-off frequency of a muffler relates to the point where the presence of only one dimensional plane waves can no longer be assumed. At frequencies above the cut-off frequency, higher order waves will propagate through the muffler and dramatically decrease its performance. For this reason reactive mufflers are most effective at attenuating low frequency noise.
2) Expansion Chamber Mufflers

The simplest reactive muffler is the expansion chamber muffler, which as the name suggests is simply a section of increased area. If the wavelength ($\lambda$) of the sound of interest corresponds to the length of the chamber, and at half order multiples of this (e.g. $\lambda/2$, $\lambda$, $3\lambda/2$…), the expansion chamber is a perfect impedance match to the pipe and no sound is attenuated. For frequencies other than this, the discontinuity reflects a portion of the acoustic energy back towards its source resulting in destructive interference. Examples of expansion chamber mufflers are shown in Figure 2 along with their attenuation characteristics predicted using equation. Experimentally determined attenuation is shown at point values.

3) Extended Tube Resonators

Extended tube resonators are characterized by the protuberance of inlet or outlet pipes into an expansion chamber. At certain frequencies all the incoming acoustic energy is used to resonate the closed end cavity and almost none is transmitted downstream. Four examples of extended tube resonators are shown in Figure 3 below.

4) Helmholtz Resonators

Helmholtz resonators, also known as side branch or volume resonator, differ from expansion chamber or extended tube resonators in that there is no gas flow through the chamber. A Helmholtz resonator consists of a small opening or neck connected to a larger chamber. The basic principle is that a small mass of gas oscillates in the neck of the resonator causing compression and expansion of the volume of gas inside the chamber. At the resonant frequency of the chamber, the impedance reduces to zero preventing any transmission of noise. Figure 4 below shows the components of a simple Helmholtz resonator.

5) Absorptive Mufflers

An absorptive muffler is one whose acoustical performance is determined mainly by the presence of sound absorbing materials within the muffler. As the sound waves pass through the spaces between the tightly packed fibers of the absorptive material, the resulting viscous and inertia losses dissipate sound energy as small amounts of heat. Absorptive mufflers usually have relatively broadband noise attenuation characteristics and perform most effectively at frequencies over 1000 Hz. The most common form of an absorptive muffler is an expansion chamber packed with absorption material. Two examples of this are shown in Figure 5 below. The absorption material is placed behind a perforated pipe to prevent it from blocking the gas flow or being blown out and a barrier layer such as stainless steel wool may also be used to prevent deterioration of the packing material.
II. LITERATURE REVIEW

Limited resources in the past caused designers to make many assumptions when linking muffler insertion loss to how it will actually perform in operation. With the advancement of numerical methods, such as FEM and CFD, other key contributors to the overall muffler performance are starting to be investigated more rigorously.

The TATA INDICA TURBOMAX TDI BSIV four-cylinder diesel engine car was considered for test purposes. In that study Muffler dimensions are measured through the Benchmarking, to create CAD models. The CAD models are created in CATIA V5 R19, later these CAD models of muffler are exported to HYPER MESH for pre-processing work. Free Free analysis is carried out on this muffler by FEA Method using NASTRAN Software. The stress and stiffness of the model is studied from the results obtained from analysis to verify the success of the design. The existing mufflers have the Frequency of 281 Hz. The new muffler was found to be superior to the existing one in terms of both acoustic performance and engine performance. With the new muffler, thickness of baffles modified 2 mm into 3 mm the maximum Frequency obtained was 381 Hz.[9]

The transient acoustic characteristics of its exhaust muffler were predicted using one dimensional computational fluid dynamics. To validate the results of the simulation, the transient acoustic characteristics of the exhaust muffler were measured in an anechoic chamber according to the Japanese Standard (JIS D 1616). It was found that the results of simulation are in good agreement with experimental results at the 2nd order of the engine rotational frequency. At the high order of engine speed, differences between the computational and experimental results exist in the high revolution range (from 5000 to 6000 rpm at the 4th order, and from 4200 to 6000 rpm at the 6th order). According to these results, the differences were caused by the flow noise which was not considered in the simulation. Based on the theory of one dimensional CFD model, a simplified model which can provide an acceptable accuracy and save more than 90% of execution time compared with the standard model was proposed for the optimization design to meet the demand of time to market.[10]

Davies attributed this to a feedback interaction between the vortex generation and the flow. Figure 6 below shows flow noise generated at the entrance of an expansion chamber and its transmission to the tailpipe. [11]

III. SOUND MEASUREMENT

This section describes measurement parameters and measurement techniques that are in current use for evaluating the acoustic performance of exhaust systems and components. The first part of this section will describe the parameters used and how they are measured. The second part will cover methods set out in SAE and ISO international standards for the measurement of exhaust noise.

A. Measurement Parameters for Exhaust Noise

1) Testing Conditions

For the measurement of the acoustic performance of exhaust systems and components it is common to separate the exhaust system or specific exhaust components from the source of excitation. This is usually achieved by using an acoustically treated barrier or a wall. Figure 7 below shows the layout of one such test facility. Transmission loss (TL) is the difference in sound power between waves entering the muffler and transmitted past the muffler, assuming an anechoic termination.
2) Transmission Loss

It is therefore a property of the muffler itself and is independent of upstream and downstream conditions. Transmission loss is defined by the equation below:

\[ TL = 10 \log_{10} \frac{W_i}{W_tr} \text{ (dB)} \] ................. 3.1

Where: \( W_i \) = incident sound power
\( W_tr \) = transmitted sound power

There are a number of methods that can be used to measure the incident and transmitted sound power in order to calculate transmission loss. [13]

3) Insertion Loss

Insertion loss is the difference between sound pressure levels measured before and after a muffler has been inserted between the source and the measurement point. There are a number of different definitions for insertion loss measurements. [3]

4) Attenuation

Attenuation is the decrease in sound power between two points in an acoustic system. Attenuation is an especially useful quantity for describing wave propagation in lined ducts where acoustic material is continuously distributed along the direction that noise is travelling. Attenuation can be measured for mufflers by determining the decrease in sound pressure level per unit length of the duct measured inside the muffler away from the ends, and multiplying this by the total length of the muffler.

B. Standardized Measurements

1) Requirements of the Measurement Standards

A sound level meter for the fast exponential time averaging characteristic should be used. The meter should be calibrated before and after the measurements are taken and any deviation noted. Accuracies of all recording equipment must be within specified limits. Measurements should be repeated until they fall within 2 dB of each other.

The sound level should be measured over the entire test time. For the standard test at constant engine speed, the sound reading is measured at ¾ of the engine speed where the vehicle produces maximum power as stated by the manufacturer.

The ambient sound level at the test site must be at least 10 dB lower than the sound level produced by the vehicle during the test. It is recommended that the background noise level is 15 dB lower than that produced by the vehicle during the test.

2) The Orientation of the Microphone

The orientation of the microphone to the end of the muffler should be at a distance of 0.5m and at an angle of 45° measured from the uppermost point of the exhaust outlet and at a height in line with the highest point of the outlet itself as shown in figure 8.

Fig. 8: Microphone position

C. Sound Elements

Sound quality is an important aspect in exhaust design as the sound from an exhaust system helps to give a car its character. For example, a deep rumbling exhaust gives the impression of a powerful car whereas a quiet exhaust may give a car a feeling of refinement and quality. Sound quality can be separated into two categories: disturbing sound (e.g. boom, hollow sound) and the sound quality character (e.g. sporty, refined, four, six or eight cylinder).

IV. EXPERIMENTAL METHOD

A. Manufacturing Of Different Size Muffler

Before Experiment work, Authors has planned of different dimension design with considering length and diameter of main chamber and extrusion length of inlet and outlet. Fabricated techniques of different design mufflers conducted in Workshop.

- Purchase Raw material
- 4 Mild steel plate (8 * 4 Foot) 24 gauge thickness
- 19 foot long mild steel pipe ( Internal Diameter 1.25 inch External Diameter 1.75 inch)
- Shearing of plate in required dimension has done in tin smithy shop of workshop by using scissors.
- Rolling and fixing of roll sheet also done in tin smithy department as shown in below Figure 9.

For temporary fixing of roll sheet, roll ended welded by spot welding. Cutting of pipe with required dimension of 1 foot for inlet and outlet pipe has done in plumbing shop. Threading of pipe for fixing in exhaust system is done by lathe machine in machine shop. Different element of muffler welded permanently to make whole muffler by gas welding. Protraction length with flange welded permanently to make whole muffler by gas welding. Flange welded with main to complete whole muffler in one unit.
**B. Engine Specification**

The engine used for experiment work is the cooper diesel four stoke engine.

Different Muffler attached with engine which specification given in below table.

<table>
<thead>
<tr>
<th>Engine Model Type</th>
<th>Diesel Engine cooper CVR-5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinder</td>
<td>1</td>
</tr>
<tr>
<td>BHp</td>
<td>5</td>
</tr>
<tr>
<td>Rpm</td>
<td>1800</td>
</tr>
<tr>
<td>Tappet Clearance</td>
<td>Exhaust 0.30 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16:1</td>
</tr>
<tr>
<td>Bore</td>
<td>0.08 m</td>
</tr>
<tr>
<td>Stroke</td>
<td>0.11 m</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>0.23 m</td>
</tr>
<tr>
<td>Cooling</td>
<td>Water-cooled</td>
</tr>
<tr>
<td>Position</td>
<td>Vertical</td>
</tr>
</tbody>
</table>

**Table 2: Engine Specification**

**C. Bruel and Kjaer type 2235 precision sound level meter description**

The sound from the exhaust will measure using a Bruel and Kjaer precision Sound Level Meter Type 2235 as shown in figure 12. The meter is positioned at the same level that of flow of exhaust gas so that the noise level can be recorded effectively.

The type 2235 is the ideal choice for general purpose sound level measurement. Its precision (IEC651 type 2) and versatility enable accurate and reliable results to be obtained in most measurement situations whether the noise is continuous, rapidly repetitive or impulsive. Addition of filter set allows accurate in situ frequency analysis in octave or third octave bands.

Microphone Type 4176, which is the standard microphone fitted to the 2235 sound level meter, is design to have a linear free-field response for 0° sound incidence. Calibration performed by using Sound level calibrator type 4230 as shown in figure.

As per ANSI Standard, Select “Random” frequency weighting and under ideal free field condition optimum response is obtained by orienting the sound level meter at 45 degree and half meter away from measurement point.

**D. Change Expansion Tube Length**

Length of main tube has changed to 440 mm, 500 mm, 560 mm respectively represent as a Design-1, Design-2, and Design-3 as shown in below table 3.
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### Table 3: Change expansion tube length drawing with specification

<table>
<thead>
<tr>
<th>Design</th>
<th>Length (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-2</td>
<td>500</td>
</tr>
<tr>
<td>-3</td>
<td>560</td>
</tr>
</tbody>
</table>

![Diagram of muffler designs](image)

**Table 4: reading for different design with load varied**

<table>
<thead>
<tr>
<th>load (pound)</th>
<th>Design-1(dB)</th>
<th>Design-2(dB)</th>
<th>Design-3(dB)</th>
<th>Without muffler(dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>76.4</td>
<td>74.2</td>
<td>72.7</td>
<td>90.4</td>
</tr>
<tr>
<td>3</td>
<td>81.6</td>
<td>77.5</td>
<td>75.4</td>
<td>97.3</td>
</tr>
<tr>
<td>6</td>
<td>86.3</td>
<td>85</td>
<td>81.5</td>
<td>102.1</td>
</tr>
<tr>
<td>9</td>
<td>90.5</td>
<td>88.2</td>
<td>85.2</td>
<td>105.7</td>
</tr>
<tr>
<td>12</td>
<td>94.3</td>
<td>91.6</td>
<td>88.3</td>
<td>109.4</td>
</tr>
<tr>
<td>15</td>
<td>96.2</td>
<td>94.2</td>
<td>91.8</td>
<td>112.7</td>
</tr>
</tbody>
</table>

![Diagram of muffler attachment](image)

**Fig. 13: main tube length change muffler**

Manufacturing of mufflers with change in main tube length are done in workshop as shown in figure 13. Measuring insertion loss as per standard is possible by bruel and kjaer precision sound level meter as discussed in sound measurement section. Result obtain by finalized reading is given in result section.

Attached each muffler with engine setup one by one for measuring performance of length change muffler is complete as shown in figure 14.

![Diagram of muffler attachment](image)

**Fig. 14: Length change muffler attachment with exhaust system**

**Consideration for drawing**

- All dimensions are in millimeter.
- Thickness of main chamber which made from mild steel is 0.635.
- Dimensions in drawing for inlet and outlet pipe indicated outer dimension.
- Inside dimensions of inlet and outlet pipe are 29 mm.

After experiment work done as followed by test procedure, reading for main tube length change has gotten by researcher which is illustrated in following table 4. Reading has occupied with the load change by brake drum arrangement for 0 pound, 3 pound, 6 pound, 9 pound, 12 pound and 15 pound. Reading has also occupied for without muffler to calculate insertion loss as discussed before.

V. FLOW SIMULATION INTRODUCTION

The first step, as with any FEM simulation, is to start with the design of the model. In this work, the researcher learned the solid modeling package SOLID WORK 2009 which was used to create the muffler models. Accordingly, the muffler can either be modeled using sheets or the skins need to be extracted from the solid geometry. Each component should be created using a separate body in the part to allow for easy distinction between the components when generating the mesh.

After the model is created it needs to be output to a readable format for the mesh generating software. Output in the form of maximum velocity is obtain during flow occur to pass flow from different design.

A. **SolidWorks FloXpress used as tool [15]**

Solid Works Flo Xpress is a fluid dynamics add-in application included with Solid Works, that calculates how fluid flows through part or assembly models. Based on the calculated velocity field, you can find problem areas in your design and improve them before you manufacture any parts. SolidWorksFloXpress is a first pass qualitative flow analysis tool which gives insight into water or air flow inside your SolidWorks model.

Analysis reports can be generated with SolidWorksFloXpress. The application can create a report for Microsoft Word™ that includes the following information:

- Information about the project file
- The value of the smallest flow passage (if specified manually)
- All the information about inlet and outlet boundary conditions
- The value of the maximum velocity
- The snapped images (if any)
B. Different step for flow simulation in Solid Works Flow Xpress

In this section, researcher has considered flow in a section of an automobile exhaust pipe as planned different design, whose exhaust flow is resisted by geometry of muffler. When designing an automobile muffler, the engineer faces a compromise between minimizing the muffler's resistance to the exhaust flow while maximizing the muffler's internal surface area and duration that the exhaust gases are in contact with that surface area. Therefore, a more uniform distribution of the exhaust mass flow rate over the muffler's cross sections favors its serviceability.

Here, as a Flow Simulation analysis carried out to consider the influence of the muffler section on the exhaust mass flow rate distribution over the muffler cross sections. Observe the latter through the behaviour of the exhaust gas flow trajectories distributed uniformly over the model's inlet and passing through the muffler. Additionally, by colouring the flow trajectories by the flow velocity the exhaust gas residence time in the muffler can be estimated, which is also important from the muffler effectiveness viewpoint.

Effect of different parameter on flow velocity can be obtained by resulting in simulation Solid Work 2009 software.

Before flow simulation, it should be noted that the inlet and outlet groups should be created from the acoustic envelope as they will be used to place boundary conditions.

Mass Flow calculation for inlet condition by considering inlet pipe dimension:

\[ d = \text{inlet pipe diameter} = 31 \text{ mm} = 0.0031 \text{ m} \]

\[ \text{Inlet pipe area } A_1 = \frac{\pi}{4} \times d^2 \]

\[ = \frac{\pi}{4} \times 0.0312 \]

\[ = 0.0007547 \text{ m}^2 \]

Velocity of air \( V = 348 \text{ m/s} \)

Mass flow rate for inlet \( Q = A_1 \times V = 0.2627 \text{ m}^3/\text{s} \)

Inlet pipe temperature used for analysis is 313 K.

Outlet condition

Environment Pressure: 101325 Pa

Temperature: 293.2 K

C. Modeling

Different planed muffler is design modeled with the help of extrude the sketch and extrude cut.

Running flowxpress

Run Tool > flow xpress

Check Geometry for view fluid volume

Select fluid as air

Assign flow condition at model’s inlet

Select flow rate

Select inside face of inlet

Put value \( Q = 0.2627 \text{ m}^3/\text{s} \) & temperature 313 K

Assign flow condition at model’s outlet

Pressure by default option selected

Select inside face of outlet side of muffler

\( P = 101325 \text{ pa} \)

Solve with automatic meshing

D. Flow Velocity Effect on Sound [16]

The flow through muffler also affects the attenuation and longevity of the packing.

The flow velocity in any application should never exceed Mach 0.3 (0.3 x speed of sound) because the gas or air will begin to compress changing its properties.

The velocity along the lengths produces self-noise that can limit the attenuation of a silencer thus the velocity should not exceed Mach 0.1 as a general rule.

The flow affects the sound energy interaction with the muffler by either making the acoustic interaction either shorter or longer based on the velocity and direction of noise relative to the flow. Downstream and upstream flow cases are also affected on sound. Flow is either positive or negative and is important in analyzing performance.

As the velocity increases the flow noise increases and there are no simple procedures to calculate this affect as the geometry of the muffler highly variable.

E. Flow Trajectory Analysis

After simulation flow, trajectory obtains which is altered for different design as shown by below figure.

For Design-1 flow trajectory for inlet and outlet form shown respectively in figure 15 and figure 16.

For Design-2 flow trajectory for inlet and outlet form shown respectively in figure 17 and figure 18.
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\[ F = \frac{N}{n} \times \frac{\pi}{S} \]  

Where \( F \) = frequency (Hz)  
\( N \) = engine speed (rpm)  
\( n \) = number of cylinders  
\( S = 2 \) (4-stroke), 1 (2-stroke)

Author has worked on Cooper Diesel engine which specification already mention in experimental chapter. Engine Speed \( N = 1800 \) rpm No. of cylinder \( n = 1 \)  
\( S = 2 \) for four stroke engine All data placed in general equation of frequency  
\[ F = \frac{1800}{60} \times \frac{\pi}{2} \]  
\( F = 15 \) Hz

B. Wavelength Calculation

Knowing the speed and frequency of a sound, we can calculate the wavelength – that is distance from one wave top or pressure peak to the next.

Wavelength,  
\[ \lambda = \frac{\text{Speed of Sound (C)}}{\text{Angular Frequency (\( \omega \))}} \]

From this equation we can work out the wavelength at different angular frequencies.

Angular frequency \( \omega = 2\pi F = 2 \pi \times 15 = 94.25 \) hz  
For acoustic and sound measurement purposes, speed is expressed as 347 m/s at room temperature.  
\[ \lambda = \frac{348}{94.25} = 3.69 \text{ m} \]

C. Expansion Ratio Calculated

Calculated Expansion ratio as followed  
\[ M = \frac{\text{Cross section area of main chamber}}{\text{Cross section area of inlet to main chamber}} \]

For Design 1  
Main chamber pipe area \( A1 = \pi/4 \times D^2 \) (D = diameter of main chamber = 16 cm) = \( \pi/4 \times 16^2 = 201.06 \text{ cm}^2 \)  
Inlet pipe area \( a2 = \pi/4 \times d^2 \) (d = diameter of inlet pipe) = \( \pi/4 \times 3.1^2 = 7.55 \text{ cm}^2 \)  
Expansion ratio \( M = A1 / a2 = 26.63 \)

D. Kl Calculation

Length of main tube \( L = 0.44 \) m  
From equation As the \( \lambda = 3.69 \) m  
\[ kl = \frac{2\pi L}{\lambda} = 0.7492 \text{ rad} = 0.7492 \times 180/\pi \]  
\( n = 42.93 \) degree

E. Transmission Loss Calculation

For expansion type muffler, it is calculated as follow  
\[ TL = 10 \log [1 + \frac{1}{4} (M - \frac{1}{M})^2 \sin^2(\text{kl})] \] (dB)  
Where: \( M \) = expansion ratio  
\( kl = 2\pi l / \lambda \)  
\( l \) = length of expansion chamber (m)  
\( \lambda = \) wavelength of sound of interest (m)  
\[ TL = 10 \log [1 + \frac{1}{4} (26.66 - \frac{1}{26.66})^2 \sin^2 42.93] \] (dB)  
\( TL = 21.3048 \) db

F. TL Calculation for All Mufflers

Same calculation is done for other remaining mufflers design-2 to design-3.  
Here only consider pure expansion type muffler for calculation and not consider protraction type design-7, design-8, and design-9. There are no equations available to calculate TL for protraction type muffler.
All the three muffler calculation tabulated in below table.

<table>
<thead>
<tr>
<th>Sr. no.</th>
<th>Length of main chamber (l) m</th>
<th>Diameter of main chamber (D) m</th>
<th>Diameter of inlet pipe d</th>
<th>Expansion ratio (M)</th>
<th>kl = 2πl/λ deg</th>
<th>TL (db)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design-1</td>
<td>0.44</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>42.93</td>
<td>21.30</td>
</tr>
<tr>
<td>Design-2</td>
<td>0.55</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>53.66</td>
<td>10.77</td>
</tr>
<tr>
<td>Design-3</td>
<td>0.56</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>54.63</td>
<td>21.98</td>
</tr>
</tbody>
</table>

Table 5: TL calculation by analytical model for design-1 to design-3

VII. RESULTS

This chapter compares the results obtained using the experimental arrangement and methods presented in Analytical calculation section to predicted results obtained using the modelling procedures presented in introduction. Comparisons are made between different muffler systems to investigate the effect of mid-pipe main chamber length on muffler performance. Experimentally derived insertion loss using the engine as the source of excitation is compared to predicted transmission loss for a number of muffler systems. The differences between predicted and experimental results are due to manufacturing error, method change, temperature, flow and load effects.

A. Change Expansion Tube Length

In this study, the overall expansion tube length was changed to length of 44 cm, 50 cm and 56 cm which are represent in design-1, Design-2, Design-3. Here effects of main chamber length on performance of muffler sound are representing by different method which already discussed in previous section of thesis.

B. Analytic Result

<table>
<thead>
<tr>
<th>Sr. no.</th>
<th>Length of main chamber (l) m</th>
<th>Diameter of main chamber (D) m</th>
<th>Diameter of inlet pipe d</th>
<th>Expansion ratio (M)</th>
<th>kl = 2πl/λ deg</th>
<th>TL (db)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design-1</td>
<td>0.44</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>42.93</td>
<td>21.30</td>
</tr>
<tr>
<td>Design-2</td>
<td>0.55</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>53.66</td>
<td>10.77</td>
</tr>
<tr>
<td>Design-3</td>
<td>0.56</td>
<td>0.16</td>
<td>0.031</td>
<td>26.64</td>
<td>54.63</td>
<td>21.98</td>
</tr>
</tbody>
</table>

Table 6: Analytic Result

C. Experimental Method Result

Reading is taken as discussed in experimental section. Here different reading to represent in graphical form as shown below figure 21 to 23.

Fig. 21: load vs IL for design-1

Fig. 22: load vs IL for design-2

Fig. 23: load vs IL for design-3

Table 6: velocity for design 1 to design 3

VIII. CONCLUSION

Results gathered using the test apparatus as described in section have been shown to be reasonably repeatable. Measurements gathered corresponded well with those expected showing firing harmonics, increases in noise level with load, and flow noise at higher frequencies. Comparing
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insertion loss measured with nominally identical mufflers produced acceptable agreement with some deviation due to experimental uncertainty.

Increasing the mean flow velocity through muffler tends to increase TL and insertion loss as correlated simulation, experiment and analytical work.

The plots of Sound Pressure Level versus load at constant rpm show an increase in noise reduction with increases in load when comparing with the existing muffler.

TL is increased as increased length of main chamber by analytical method and by experiment method.

Maximum velocity of flow is decreased as increased length of main chamber by simulation method.

As the velocity decreased, backpressure on engine is reduced.

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