

Muffler Design, Development and Validation Methods

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ABSTRACT: Internal combustion engines are typically equipped with an exhaust muffler to suppress the acoustic pulse generated by the combustion process. A high intensity pressure wave generated by combustion in the engine cylinder propagates along the exhaust pipe and radiates from the exhaust pipe termination. Exhaust mufflers are designed to reduce sound levels at certain frequencies. New regulations and standards for noise emission increasingly compel the automotive firms to make some improvements about decreasing the engine noise. On the other hand, developments on automobile technology and increasing competition between manufacturers necessitates having being reduced weight, having capability of higher sound absorption and lower back pressure mufflers. Lightness could be possible if the thickness is decreased or the volume is reduced. However, this causes high back pressure. Traditionally, muffler design has been an iterative process by trial and error. However, the theories and science that has undergone development in recent years has given a way for an engineer to cut short number of iteration. In today's competitive world market, it is important for a company to shorten product development cycle time. This paper deals with a practical approach to design, develop and validate muffler particularly reactive muffler for exhaust system, which will give advantages over the conventional method with shorten product development cycle time and validation. This paper also emphasis on how modern CAE tools could be leveraged for optimizing the overall system design balancing conflicting requirements like Noise & Back pressure.

KEYWORDS: Acoustic pulse, muffler, muffler design, transmission loss, back pressure.

I. INTRODUCTION

Since the invention of the internal combustion engine in the latter part of the nineteenth century, the noise created by it has been a constant source of trouble to the environment. Significantly, the exhaust noise in terms of pressure is about 10 times all the other noises (structural noise) combined. So the problems of reducing engine noise consist, mainly in attenuating exhaust noise.

The design of mufflers has been a topic of great interest for many years and hence a great deal of understanding has been gained. Most of the advances in the theory of acoustic filters and exhaust mufflers have come about in the last four decades. Hence good design of the muffler should give the best noise reduction and offer optimum back pressure for the engine. Moreover, for a given internal configuration mufflers have to work for a broad range of engine speed. Usually when mufflers are designed by well established numerical techniques like boundary element method or finite element method, the numerical model generation is time consuming often limiting the user to try various other possible design alternates. The process might be lengthy and laborious as it involves a more iteration with different prototypes. Mufflers have been developed over the last ninety years based on electro- acoustic analogies and experimental trial and error. Many years ago Stewart used electro – acoustic analogies in deriving the basic theory and design of acoustic filters [1]. Later Davis et al. published results of a systematic study on mufflers [2]. They used travelling wave solutions of the one-dimensional wave equation and the assumption that the acoustic pressure p and acoustic volume velocity v are continuous at changes in cross sectional area. An important step forward in the analysis of the acoustical performance of mufflers is the application of two- port network theory with use of four –pole parameters. Igarashi and

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

his colleagues calculated the transmission characteristics of mufflers using equivalent electrical circuits [3-4]. Parrot later published results for the certain basic elements such as area expansions and contractions. Sreenath and Dr. Munjal gave expression for the attenuation of mufflers using the transfer matrix approach [5]. The expression they developed was based on the velocity ratio concept. Later, Dr. Munjal modified this approach to include the convective effects due to flow [6]. Young and Crocker used the finite element method to predict four-pole parameters and then the transmission loss of complex shaped mufflers for the case of no flow [7]. Ying-change, Long-Jyi used optimized approach of maximal STL and muffler dimension under space constraints throughout the graphic analysis as well as computer aided numerical assessment [8]. Middlberg, J.M. and Barber T.J. present different configurations of simple expansion chamber mufflers, including extended inlet or outlet pipes and baffles have been modelled numerically using CFD in order to determine their acoustic response [9]. However, most of the research studies based on formulation of mathematical equation and trial and error method. The scope of our work is to establish a design methodology to make design process simpler and less time consuming by making use of acoustic theories [10, 11] and experience, in short practical approach to get better design. Also this approach will predict design quality at earlier stage of muffler design, evaluate quality of design, set targets for proto design and improves the same through out the product design steps and reduce cost of proto development.

II. DESIGN REQUIREMENTS

The properly designed muffler for any particular application should satisfy the often conflicting demands of at least five criteria simultaneously

- The acoustic criterion, which specifies the minimum noise reduction, required from the muffler as a function of frequency. The operating conditions must be known because large steady- flow velocities or large alternating velocities (high sound pressure levels) may alter its acoustic performance.
- The aerodynamic criterion which specifies the maximum acceptable average pressure drop through the muffler at a given temperature and mass flow.
- The geometrical criterion, which specifies the maximum allowable volume and restrictions on shape.
- The mechanical criterion, which may specify materials from which it is durable and requires little maintains.
- The economical criterion is vital in the marketplace. [3, 8]

III. DESIGN METHODOLOGY

The Muffler Design methodology for a given engine involves 7 steps. Following are the broad steps followed to arrive at a good design of muffler making use of practical experimental data figure 1.

Step 1: Objectives framing and benchmarking

- To obtain system inputs (usually engine)
- Compare with benchmarked system (power)
- Frame design objectives

Step 2: Calculation of targeted data (frequencies)

- Cylinder Firing Rate (CFR) based on basic engine data
- Engine firing rate calculation based on CFR

Step 3: Muffler volume calculation

- Geometrical details of engine
- Muffler volume calculation

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

Step 4: Internal configuration and concept design

- Refer benchmarking data and target frequencies
- Internal configuration calculation based on step 2
- Concept design finalization

Step 5: Virtual simulation

- Refer concept design
- Virtual acoustics for transmission loss measurement
- Comparison of results and chose best three

Step 6: Prototype manufacturing

- Refer design from step 5
- Consider all basic requirements for manufacturing
- Manufacturing of prototype without tooling cost

Step 7: Experimental testing and design finalization for prototype

- Calculation of Transmission loss
- Comparison of results
- Finalization of best concept

3.1 Benchmarking

Benchmarking is a core component of continuous improvement programs; it is a key component of quality assurance and process improvement. The role of benchmarking in process improvement is similar to that of the Six Sigma process improvement methodology. The Six Sigma methodology comprises five integrated steps: define, measure, analyse, improve, and control (DMAIC). These steps are also central to the benchmarking process forms the basis for the continuous improvement cycle. Measuring, comparing to competition, and identifying opportunities for improvements are the essence of benchmarking.

The same will be applicable to silencer, to set a target in terms of transmission loss of same engine power models of competitor benchmarking vehicles. Based on the provided engine input data and benchmark study target for back pressure and noise are range decided.

3.2 Calculation of targeted data (frequencies)

After benchmarking exercise, one needs to calculate the target frequencies to give more concentration of higher transmission loss. For calculating the target frequencies engine max power rpm is required. The exhaust noise spectrum will always contain strong tones associated with the rate of cylinder firings. In 4-cycle engines each cylinder fires once every other revolution of the drive shaft. Cylinders fire once every rev in 2-cycle engines. The lowest tone is always the CFR, which is the firing rate for any one cylinder. The engine firing rate is generally the strongest tone in the exhaust spectrum.

Engines exhaust tones:

- Cylinder Firing Rate (CFR)
 $CFR = RPM/60$ for 2-stroke engines (1)
 $CFR = RPM/120$ for 4-stroke engines (2)
- Engine Firing Rate (EFR)
 $EFR = N (CFR)$ where, N = No. of cylinders (3)

Harmonics of CFR and EFR

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

3.3 Muffler volume calculation

Based on the experience and theory of acoustics for muffler design for various Engines, the following equation works well.

Volume of the muffler (V_m):

$$V_m = V_f * \left[\frac{\pi}{4} (d^2 * 1) \right] * \frac{N}{2} \quad (4)$$

3.4 Internal configuration and concept design

Based on the benchmarking transmission loss and the target frequencies, designer draws few concepts of internal configuration that meets the packaging dimension within the volume mentioned above. Each concept and internal configuration is then formulated to the best possible configuration so as to achieve best acoustic performance and least back pressure.

Perforations: The most critical component regarding back pressure of any commercial muffler is cross flow perforated tube in which the diameter of the perforated tube hole and porosity of the perforations are most critical.

The perforated pipes are complex acoustic impedance and are evaluated using simple empirical relations. Perforated pipe forms an important acoustic element of muffler, which is tuned in line with the problematic frequencies identified in step 2.

The diameter of the hole to be drilled / punched on the pipe is calculated by a thumb rule as given below:

$$d_1 = \frac{1.29}{\sqrt{N}} \quad (5)$$

Porosity, σ is given by

$$\sigma = \frac{Area}{C^2} = \frac{\frac{\pi}{4} * d_1^2}{C^2} \quad (6)$$

A smaller hole diameter but higher porosity creates high peak power with acceptable back pressure value. While a larger hole diameters with higher porosity reduce performance due to insufficient back pressure. Sound energy gets dissipated considerably while moving through the perforations and it adds to the total attenuation in perforated tubes. The designer needs to keep in mind lesser the porosity more is the restriction and hence more will be the back pressure.

Open area ratio: The open area ratio A_{op} is given by,

$$A_{op} = \text{Area of perforation} / \text{Area of the plain sheet} \quad (7)$$

Lesser the A_{op} better the transmission loss and better the acoustic performance. At this stage, the diameter of the hole to be drilled, pitch, number of holes per row, number of rows for each pattern of holes is frozen and hence, the distance at which perforation starts and at which the perforation ends also gets frozen. Thus, the design of the perforated tube for individual hole patterns is finalized. Based on this best concepts are designed and carry forward for virtual simulations.

3.5 Virtual Simulation

Based on above mentioned approach, different concepts will be arrived with optimum combinations of different elements inside volume of the silencer. Finalized concepts will be verified virtually by calculating transmission loss and back pressure. Transmission loss (TL) in duct acoustics, together with insertion loss (IL), describes the acoustic performances of a muffler. It is frequently used in the industry areas such as muffler manufacturers and NVH

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

department of automobile manufacturers. Generally the higher transmission loss of a system it has, the better it will perform in terms of noise cancellation.

Transmission loss (TL) is defined as the difference between the power incident on a muffler and that transmitted downstream into an anechoic termination. Transmission loss is independent of the source and requires an anechoic termination at the downstream end.

3.5.1 Transmission Loss Analysis

Prediction of transmission loss virtually is an important analysis required for the development of muffler at an initial design stage. There are different software packages available in market for predicting the transmission loss. We have used virtual lab for Transmission loss measurements. It is also to be noted the limitations of the CAE tools, as the co-relation at higher frequencies is difficult since the plane wave theory holds good only up to 3000 Hz beyond which the wave is no more 2 dimensional but 3 dimensional for which the computations are far complex to match the practical results. Hence need of research to blend both strengths of CAE & Practical resulting in a Practical approach/methodology. After completion of simulation the best three concepts will (with less back pressure and higher transmission loss) be taken forward for the prototype manufacturing to check for the transmission loss and back pressure physically.

3.6 Prototype Manufacturing

All the above stages combined with the packaging of the engine evolve the design of the prototype muffler and those; can be taken up for manufacturing.

Following are some of the important manufacturing considerations summarized based on experience:

- There should not be any leakage of gas from one chamber to another.
- Full welding is better than stitch welding.
- Acoustic performance of extruded tubes with perforations is better than the tubes that are made out of perforated and welded sheets.
- CEW or ERW tubes are the common materials used.
- Either of Crimping or full welding of jacket can be used.
- Either of flanged or flared tubes can be used as end connections of the muffler. However, with leakage point of view, flanged connections are better. But at the same

However, with leakage point of view, flanged connections are better. But at the same time, this adds to the weight and cost of the exhaust system. Bearing all above in mind, a physical prototype is made in such a way that there will not be any tooling investment for the prototype.

3.7 Experimental Testing and Design Finalization for Prototype

The experimental determination of backpressure on engine and transmission loss on two source method for different concepts of verified. The prototypes of all concepts that are made at the above step are tested for the transmission loss to verify the target value. The TL is the difference in sound power level between the incident wave entering and transmitted wave exciting muffler when the muffler termination is anechoic, TL is a property of the muffler only. In this work an attempt has been made to experimentally measure transmission loss by actually using the experimental set-up. Two source techniques gives good results for the measurement of transmission loss at the different sound frequencies. Also absence of anechoic termination, the decomposition method is found to ineffective. Therefore we will be using two source methods in calculating transmission loss.

TL values obtained from these simulations are compared with experiments. At the same time if performance of muffler is found to be satisfactory as per engine noise requirement, then the above captured data becomes the input for further back pressure reduction. The iteration is continued usually 2 to 3 times to achieve an optimum balance between noise requirement and target of least back pressure and best fuel efficiency.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

IV. CASE STUDY

As per design methodology we benchmarked same kind of engine models to set the target of transmission loss of muffler.

Engine Data: Bore (D) = 69.6 mm
Stroke (L) = 82 mm
No. Cylinders (n) = 4
Engine power (P) = 88.7 bhp at 4000 rpm

Muffler Volume Calculations:

Swept volume per cylinder = $0.25 (\pi * D^2 * l) = 0.25 (3.14 * 69.6^2 * 82)$
(Vs) = 0.3119 L

Total swept volume in liters = $4 * 0.3119 = 1.247$ L

Volume to be considered for calculation = $0.5 * V_s * n = 0.6239$ L

Silencer volume: Volume of silencer must be at least 12 to 25 times the volume Considered [6]. Volume can be adjusted depending on the space constraint.

Factor considered is = 22

Silencer volume = factor * consider volume = 13.73 L

Diameter of Muffler Calculations:

$$V_m = 0.25 * \pi * d^2 * l$$

$$0.01373 = 0.25 * \pi * d^2 * 0.5$$

$$d = 0.187 \text{ m} = 187 \text{ mm}$$

Diameter of Pipe Calculations:

As per the standards of the supercritical grade of mufflers, the diameter of the body should be about three times than the exhaust pipe diameter.

$$d = 3 * e_{\text{exhaust}}$$

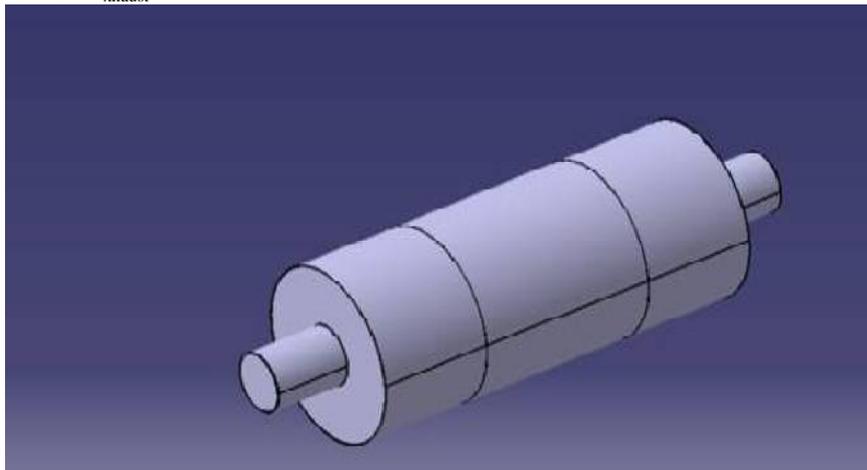


Figure 1: Numerical model of the designed muffler

The above figure is the numerical model of designed muffler, designed in catia v5 and then imported to ansys fluent for cfd analysis of fluid through the muffler. In this session we designed two models and then carried cfd analysis for the two models the below figures represent the models.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

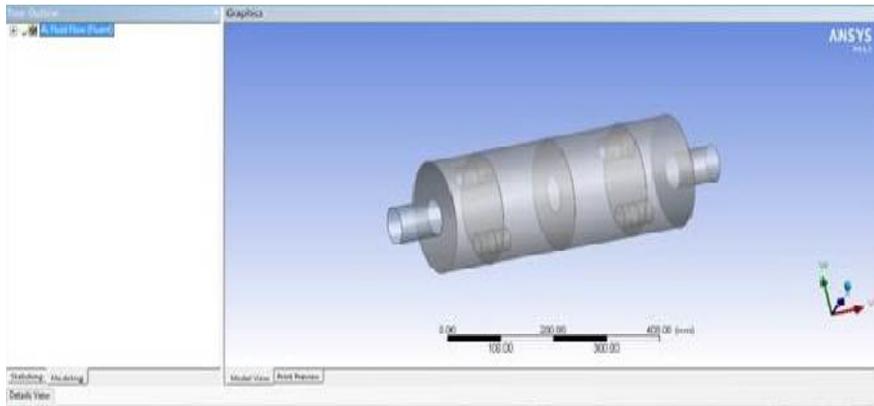


Figure 2: Internal view of designed muffler (model 1)

The above figure is internal view of the first model of the designed muffler having four expansion chambers . The first chamber ends with the two openings at the near walls with short circular tubes. The second chamber ends with the central opening and third chamber ends with two openings as like first chamber.

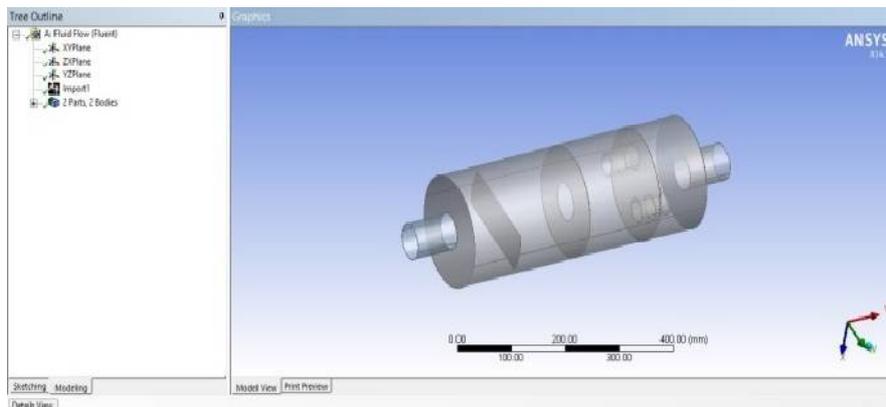


Figure 3: Internal view of designed muffler (model 2)

The above figure is internal view of the first model of the designed muffler having four expansion chambers . the first chamber ends with the two openings with special baffle arrangement. The second chamber ends with the central opening and third chamber ends with two openings at the near walls with short circular tubes.

Table 1: Material properties

Property	Units	Value
Density	Kg / m ³	7900
Specific heat	J / kg x K	500
Thermal conductivity	W/ mxK	16.2
Conductivity type	Isotropic	isotropic

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

Table 1 represents the material properties of the muffler walls which is durable and withstand the exhaust gas temperatures in the operation. From the operation of muffler the transmission loss does not depend on the muffler material so that material should satisfy only mechanical criteria.

Table 2: Dimensional data

Entity	Dimensions
Shell length	500
Shell Dia	187
Inlet pipe Dia	62.33
Inlet pipe length	85
Outlet pipe Dia	62.33
Outlet pipe length	85

Table 2 represent the dimensional data of two muffler designs calculate from the engine data and requirements of muffler like space constraint and muffler volume. Should be remind while design the acoustic properties of muffler depend on length of muffler.

V. RESULTS AND DISCUSSION

After the construction of exhaust muffler model it is analysed in ANSYS 14.5. The analysis is carried out by considering two types of inlet boundary conditions.

1. Velocity inlet
2. Pressure inlet

Table 3: Initial values for velocity inlet as the inlet boundary condition

Area(m ²)	0.003051
Temperature(K)	470
Viscosity(kg/ms)	2.7e-05
Enthalpy(j/kg)	749575.3
Density(kg/m ³)	0.696
Length(mm)	500
Velocity(m/s)	80
Pressure (bar)	3.2
Ratio of specific heats	1.4

The mean flow performance of the muffler considered in the flow analysis has been assessed. The results of the simulated muffler models obtained with the use of CFD modelling are very encouraging.

5.1 Velocity Inlet Boundary Condition for Model 1:

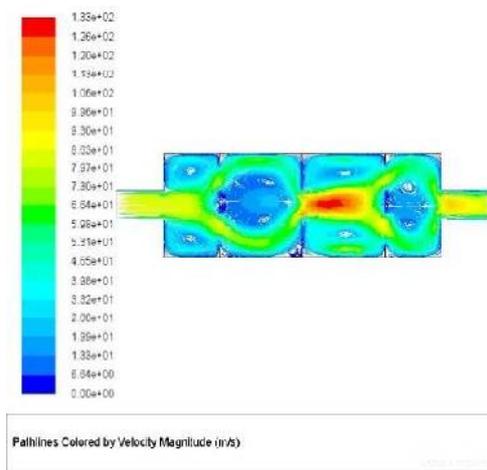


Figure 4: Velocity field cut plot

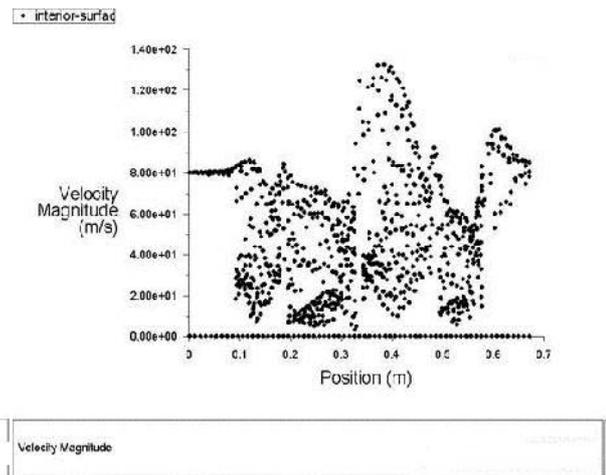


Figure 5: XY plot (velocity magnitude vs. position)

From Figure 4 it has been observed that for a velocity inlet boundary condition in model 1, the exhaust from the engine enters the muffler at a velocity of 80 m/s and increases to a magnitude of about 133 m/s in the expansion chamber once it passes through the opening.

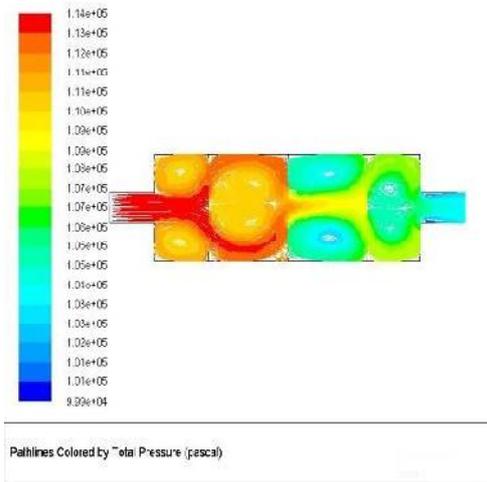


Figure 6: Pressure field cut plot

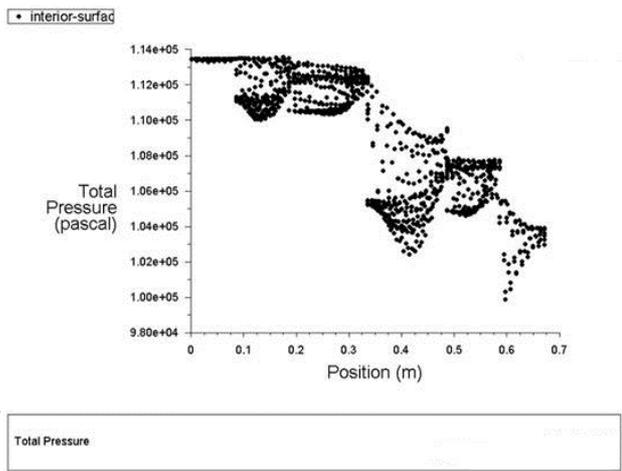


Figure 7: XY plot (total pressure vs. position)

This is observed as a result of decrease in the flow area the pressure increases and subsequently the velocity increases. After this the gases get spread in the chamber and they enter the next chamber from the other two splits. Then on hitting the baffle they swirl and come out of the exhaust at an increased speed of about 106 m/s.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

5.2 Pressure Inlet Boundary Condition for Model 1:

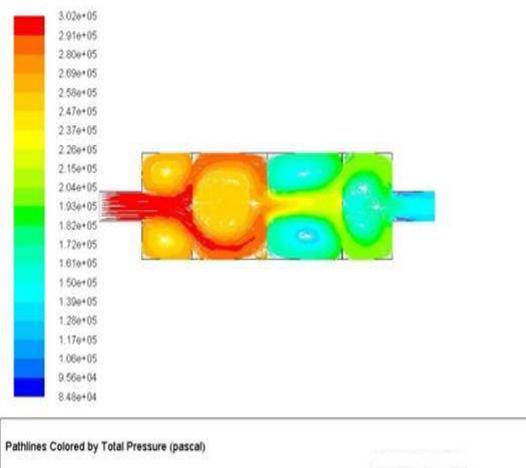


Figure 8: Pressure field cut plot

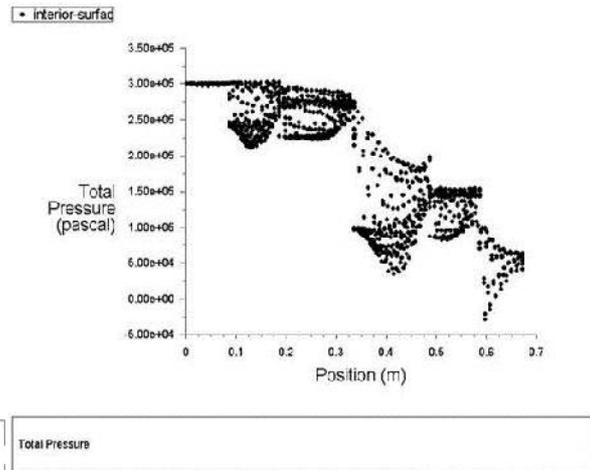


Figure 9: XY plot (Total pressure vs. position)

Also from figure 9 we observe that for a pressure inlet boundary condition in model 1, the gases enter the muffler at a pressure of 3.02 bar and hit the baffle. On making the impact the exhaust recedes a bit and swirls are observed in that particular entrance chamber. The swirls have the pressure magnitude of about 2.47 bar. The gases enter the second chamber from the baffle openings at top and bottom and hence it has been observed that the pressure intensity is more near the walls in this chamber and has a magnitude of about 2.8 bar. This is because of the fact that due to the presence of the slits or openings near the top and the bottom of the baffle, the pressurized gas or exhaust is directed towards the walls of the muffler and on hitting them the exhaust comes away from the wall

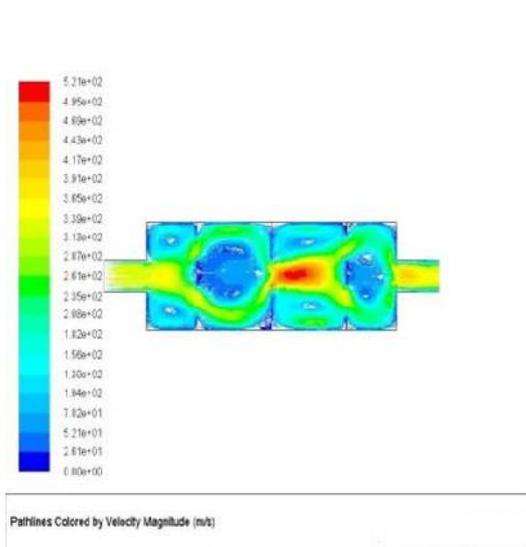


Figure 10: Velocity field cut plot

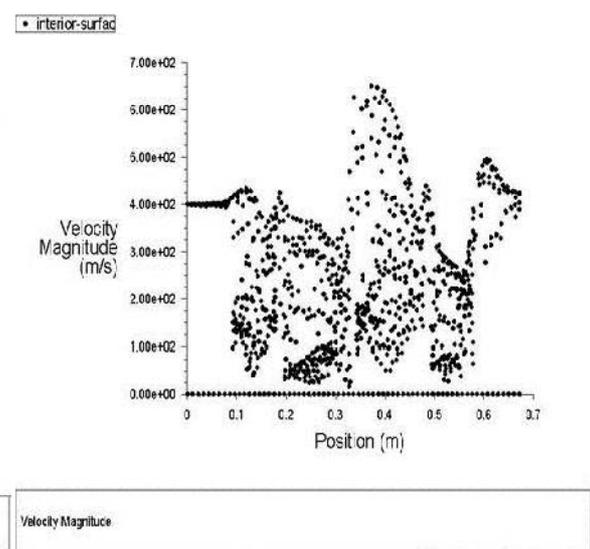


Figure 11: XY plot (Velocity magnitude vs. position)

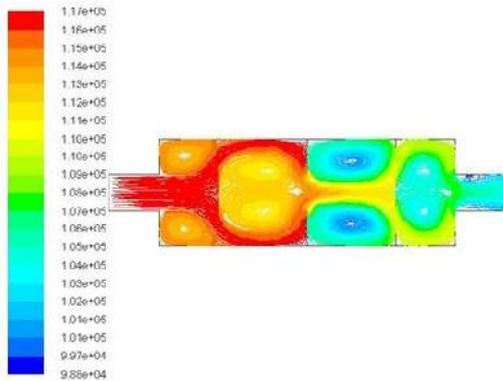
International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

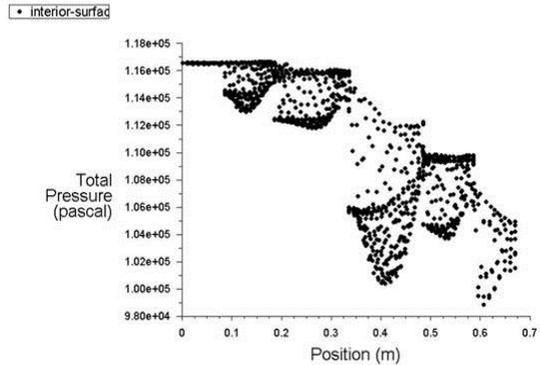
From the second chamber the exhaust enters the third chamber through an opening at the centre of a baffle and hence it is observed that there is a 2.26 bar pressure region almost throughout the centre of the chamber and the rest has a swirl region with a pressure of 1.5 bar. This is due to the reason that as the area decreases the pressure increases and the velocity decreases. So as the pressure is more hence there is a constant or maintained region in the chamber. Again the gases hit the baffle and enter the next chamber with a similar effect as it was observed in the second chamber but with a reduced pressure intensity of about 1.93 bar near the walls. Finally the exhaust gases come out of the outlet pipe at a pressure of 1.48 bars

5.3 Velocity Inlet Boundary Condition for Model 2:



Pathlines Colored by Total Pressure (pascal)

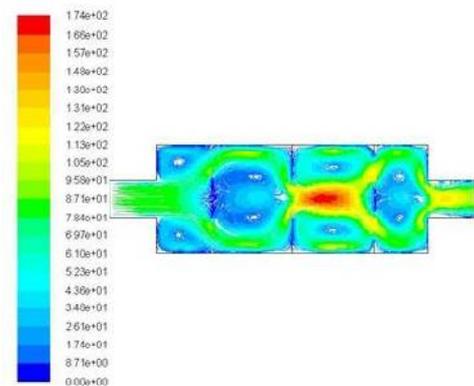
Figure 12: Pressure field cut plot



Total Pressure

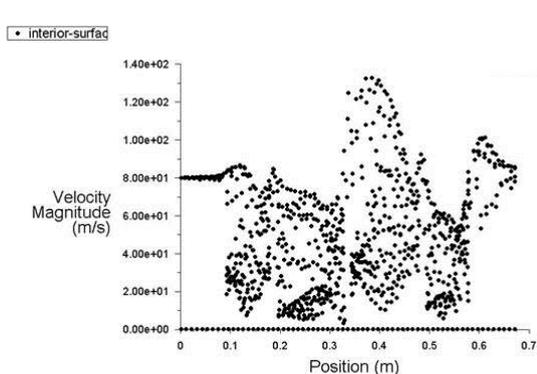
Figure 13: XY plot (Total pressure vs. position)

The velocity magnitude over this region is nearly 310 m/s according to figure 14. Due to the reduction in the flow area the velocity of the gases increases and reaches a maximum intensity of 620 m/s. The flow pattern remains similar to that of model 1 further and the gases come out of the outlet pipe at a speed of 372 m/s and a pressure of 1.32 bar.



Pathlines Colored by Velocity Magnitude (m/s)

Figure 14: Velocity field cut plot



Velocity Magnitude

Figure 15: XY plot (Velocity magnitude vs. position)

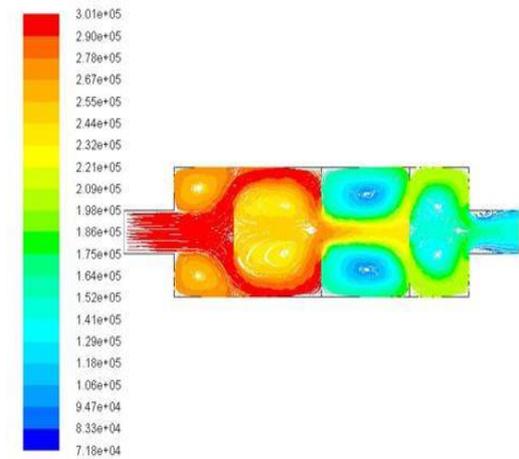
International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

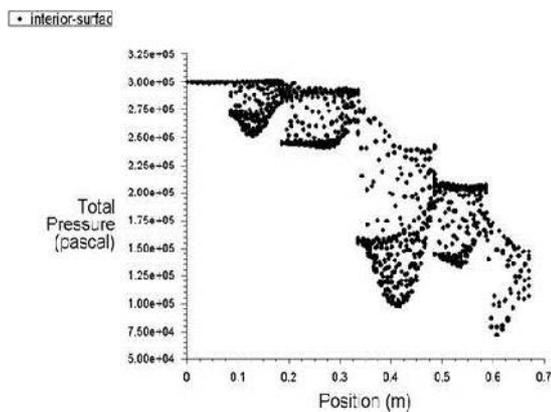
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5.4 Pressure Inlet Boundary Condition for Model 2



Pathlines Colored by Total Pressure (pascal)

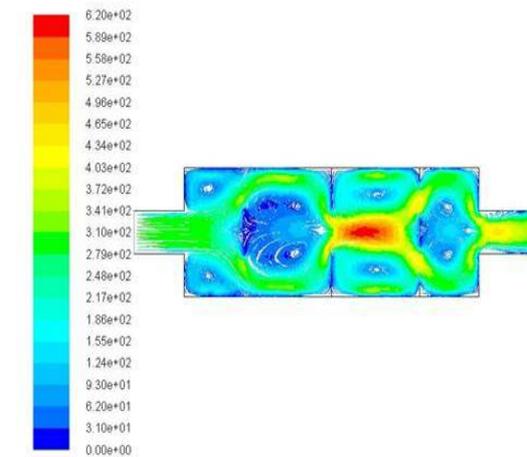
Figure 16: Pressure field cut plot



Total Pressure

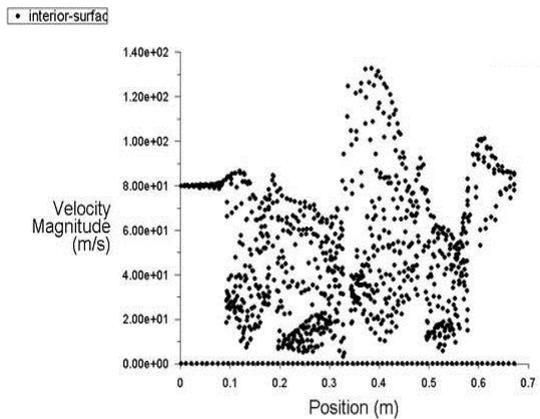
Figure 17: XY plot (Total pressure vs. position)

In model 2 as evident from figure 16 it has been observed that the gases enter the inlet pipe at a pressure of 3.01 bar and on hitting the baffle there occurs a swirl region in the first chamber which has a pressure intensity of 2.9 bar. The flow of the gases was more or less similar as that in model 1 but due to the changed arrangement of the baffles more intensity of pressure was observed near the walls.



Pathlines Colored by Velocity Magnitude (m/s)

Figure 18: Velocity field cut plot



Velocity Magnitude

Figure 19: XY plot (Velocity magnitude vs. position)

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 5, Issue 5, May 2016

Even though the first model displayed a similar behaviour, there was a more significant drop in the pressure of the exhaust gases in the second case than the first. The drop in the pressure of the exhaust gases in the first model was about 51% whereas the drop in the second model was nearly 57 %.

VI. CONCLUSIONS

1. A brief background on evaluation of muffler concept design for the prototype and validation with new approach
2. A methodology has been developed for optimum design stages and less cost for muffler design by balancing various parameters
3. A practical tool to estimate the quality of muffler design, which used for concept selection and filter out the best concept proposal at initial phase of design
4. A practical approach for muffler design to optimization of product development time & cost by balancing conflicting requirements like Noise & Back pressure
5. Design methodology emphasis on modern CAE tools for optimization of overall system design to choose the best concept

This paper emphasizes the importance of the design methodology a practical approach from the concept design to proto manufacturing and validation of exhaust muffler. This design methodology will help designers in understanding the importance of each step of designing in detail from concept level to validation level. This approach serves the purpose of reducing the number of iterations, product development time and cost with better design. Although the practical approach has become an important tool in making muffler design more of art than a science, the need for design verification will always be necessary at end of each step.

6.1 Case Study Conclusions:

In this work two different models of a muffler have been designed for the engine output of an LCV diesel engine and the flow has been simulated using ANSYS. The flow characteristics obtained through the simulation were promising.

On comparing the results and performances of the two models, we observe that though both the models have same similar design parameters, the second model was more effective in reducing the exhaust pressure than the first one because of its internal baffle arrangement.

1. Maximum velocity in model 1 for velocity inlet boundary condition is 133 m/s
2. Maximum velocity in model 2 for velocity inlet boundary condition is 176 m/s
3. Exhaust pressure reduction in model 1 is 53.82
4. Exhaust pressure reduction in model 2 is 57.14

The reduction in pressure of exhaust in model 1 is 53.82 % whereas the reduction in exhaust pressure in model 2 is 57.14 %. Hence we conclude that model 2 is more efficient in reducing the exhaust pressure when compared to model 1.

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