STUDY AND ANALYSIS OF INTERNAL FLOW IN AN EXHAUST SYSTEM OF

By

ARUN KUMAR (R160209009)
BHARTENDU PALNI (R160209013)
NAVPRABHAT BISHT (R160209035)
TARUN KABADWAL (R140208051)

COLLEGE OF ENGINEERING STUDIES

Under the guidance of

Dr. MUKESH SAXENA
HEAD OF MECHANICAL ENGINEERING DEPARTMENT

SUBMITTED

IN PATRIAL FULFILLMENT OF THE REQUIREMENT OF THE DEGREE OF

BACHELOR OF TECHNOLOGY

IN

AUTOMOTIVE DESIGN ENGINEERING

TO

UNIVERSITY OF PETROLEUM AND ENERGY STUDIES

DEHRADUN

APRIL, 2013

THESIS COMPLETION CERTIFICATE

This is to certify that the major project report on “**STUDY AND ANALYSIS OF INTERNAL FLOW IN AN EXHAUST SYSTEM OF A TWO WHEELER**” by “ARUN KUMAR, BHARTENDU PALNI, NAVPRABHAT BISHT, and TARUN KABADWAL” in partial completion of requirements of the award of Degree of Bachelors of Technology is an original work carried out by them under our joint supervision and guidance.

It is certified that the work has not been submitted anywhere else for the award of any other diploma or degree of this or any other University.

Internal Guide

Dr. Mukesh Saxena

(Name & Signature)

INTERNAL GUIDE

Prof. Narayan Khatri

(Name & Signature)

ACKNOWLEDGEMENTS

It is with a sense of great satisfaction and pride that we are sitting down to pen down our major project thesis report. First and foremost we sincerely salute our esteemed Institution **University of Petroleum & Energy Studies, Dehradun** for giving this golden opportunity for fulfilling our warm dreams of becoming a bachelor.

The first and foremost person we would like to express our deep sense of gratitude and profound thanks to our guide **Dr. Mukesh Saxena,** Head Of Department of Mechanical Engineering, UPES Dehradun for his valuable advice, suggestions, insurmountable guidance which played a vital role in carrying out our project work successfully. Our thanks to **Prof. Narayan Khatri** (Mechanical Department), UPES, Dehradun for his valuable guidance towards our project.

We would like to thank all faculty, staff, and students of Department of Mechanical Engineering and for rendering help during various stages of the project work.

Arun Kumar (R160209009)
Bhartendu Palni (R160209013)
Navprabhat Bisht (R160209035)
Tarun Kabadwal (R140208051)

ABSTRACT

One of the critical problems in engine exhaust system is to control the vibration levels. Thus design engineers have to perform a detail analysis of the whole system to calculate the exact design parameters that need to be modified.

The aim of this thesis is to provide a deeper understanding of the dynamics of automobile exhaust systems to form a basis for improved design and the development of a computationally inexpensive theoretical system model. Modeling, simulation and experimental investigation of a typical exhaust system are performed to gain such an understanding and to evaluate modeling ideas.

The investigations show that the exhaust system is essentially linear downstream of the flexible joint. Highly simplified finite element models of the major components within this part are suggested. These models incorporate adjustable flexibility in their connection to the exhaust pipes and a procedure is developed for automatic updating of these parameters to obtain better correlation with experimental results. The agreement between the simulation results of the updated models and the experimental results is very good, which confirms the usability of these models.

Furthermore, our project deals with the inner flow analysis of the silencer and studying the effect by changing the physical parameters in the geometry.

The analysis shows us the various pressure curves & velocity curves that guide us about the back pressure and gas flow velocity inside the silencer geometry.

An iterative method would be used in which a two-wheeler silencer would be analysed using FLUENT and the flow separation point will be observed. Based on the flow separation location, different elements will be associated and disassociated simultaneously and different possibility of delaying the separation point would be covered.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF GRAPHS

CHAPTER 1

INTRODUCTION

1.1EXHAUST SYSTEM:

Very often the engine exhaust system comes later to the mind of engine and chassis builders, but its design and construction strongly impacts significantly upon car performance. The exhaust system can be an effective tool for optimizing the performance of the engine, by the way in which its design manipulates the pressure waves which crucially assist cylinder filling and scavenging process.

There are two separate components to the exhaust event. These are

1. Removal of exhaust gasses from the cylinder, which occurs as a pulse of hot gas exiting the cylinder flowing down the header primary tube.

2. The second is the (much faster) travel of the pressure wave in the port caused by the pressure hike which occurs when the exhaust valve opens, and the various reflections of that wave. Taking proper advantage of these pressure waves (component two) we can produce dramatic improvements in clearing the cylinder (component one) and can strongly direct the inflow of fresh charge.

1.2 COMPUTATIONAL FLUID DYNAMICS

In the beginning of 1960s, the researchers came across computational methods to solve their flow problems around complex geometries. This provided them with a powerful tool of computational fluid dynamics to analyse the flow within or outside the body. Computational fluid dynamics provides a third approach in the philosophical study and development of whole discipline of fluid mechanics. Throughout most of the twentieth century the study and practise of fluid dynamics involved the use of pure theory on one hand and pure experiment on other hand. However the advent of high speed digital computer combined with the development of accurate numerical algorithms for solving physical problems on these computers has revolutionised the way we study and practise fluid dynamics.

However to keep things in perspective, computational fluid dynamics provides a third approach to physical problems and nothing more than that. We have to still rely on experiment and theory to validate our results. The future advancement of fluid dynamics will rest upon the proper balance of all three approaches, with computational fluid dynamics helping to interpret and understand the result of theory and experiment and vice versa.

- Pre-processor
- Solver
- Post-processor

The pre-processor involves the following steps

- Geometry acquisition/Geometry making- This is the first step in simulating any flow problem in CFD. The geometry on which the simulation has to be done is either to be made or acquired from other source.
- Discretization- This step involves the division of fluid domain and the geometry into small elements or volumes with the help of meshes. The mesh can be structured as well as unstructured.
- Boundary condition- This step involves the dividing of whole fluid domain into different physical phases like inlet, outlet, wall, symmetry etc.

The solver involves the process of simulating the flow around the meshed domain to acquire the results.

Post-processor is used to visualize the results.

CHAPTER 2

PROJECT OBJECTIVE

2.1 AIM AND OBJECTIVE

The main objective of our project is to study and perform the inner flow analysis using CFD software modify a current exhaust system of a motorbike (using a 4 stroke engine) into a sport tuned version by changing the exhaust system geometry (Length & Diameter) thereby reducing backpressure, and improving the gas flow velocity.

The engine is like an air pump; the more air that is allowed to flow through it, the more horsepower that you get out of it. In other words, if you have a free-flowing air intake and exhaust system in your average vehicle, you'll get more horsepower because of the efficient flow of air into and out of the engine. Fuel requires air to burn and thus to produce energy. The more air that is available for combustion will also improve efficiency otherwise known as gas mileage.

2.2 METHODOLOGY

In order to understand the nature of pressure curve of exhaust flowing through the silencer of a two wheeler various methodologies could be applied. It is the Analysis of flow through dynamometer by measuring pressure at various positions under varying RPM and load. For this project we have selected analysis of flow through software. Softwares like CFD and Solidworks Flow Simulation helps us to obtain practical results which are time efficient and gives accurate results.

In short, the whole project will be divided into sections:

- Model Full Exhaust System in Solid works.
- Perform the flow expression wizard analysis in solid works.
- Using CFD (Computational Fluid Dynamics), perform an Inner Flow Analysis of the System.
- Simulate it Using CFD (Computational Fluid Dynamics)

- Compare the Practical results with the Results obtained from CFD
- Make charts of pressure distribution Obtained from the Geometry & Original Geometry.
- Compile The Final Results Obtained.

CHAPTER 3

LITERATURE REVIEW

3.1 HISTORY

Expansion chambers are built to capture the sound waves (created in the combustion process) to suck the cylinder clean of spent gasses--and during the process, draw fresh air/gas mixture (known as 'charge') into the chamber and then fill all the charge back into the cylinder, filling it to greater pressures which is achieved by simply venting the exhaust port in the open atmosphere. This phenomenon was first discovered in year 1950s by Walter Kaaden, who was working at the East German company MZ. Kaaden understood that there was power in the sound waves coming from the exhaust system, and it has opened up a whole new field in two-stroke theory of exhaust and tuning. It was first appeared in the west on Japanese motorcycles after East German motorcycle racer Ernst Degner who defected to the west during racing for MZ in the 1961 in Swedish Grand Prix. He took the blueprints under his racing leathers and defected during the race by eliminating the track and claiming asylum.

3.2 EXHAUST EXPANSION CHAMBER

An expansion chamber is an exhaust system used on a two-stroke cycle engine to enhance its power output by improving its volumetric efficiency. It makes use of the energy left in the burnt exhaust exiting the cylinder to aid the filling of the cylinder for the next cycle. It is the two-stroke equivalent of the tuned pipes (or headers) used on four-stroke cycle engines. A tuned pipe is a part of a two-stroke engine's exhaust system.It should be distinguished from a muffler as a tuned exhaust pipe does more than muffling the sound. Its purpose is to retain the air/fuel mixture in the combustion chamber by using the sound wave produced by the combustion process itself and bouncing it back to the exhaust port at the appropriate time, thus precluding the fresh charge, which comes through the transfer port/s, to follow the exhaust gases. The main reason for it was not to create extra power, which is a secondary result, but to save fuel.

3.3 REMOVAL OF EXHAUST GASES

When the exhaust valve first opens in a 4-stroke engine, pressure inside the cylinder is well above atmospheric. In a normally-aspirated SI engine operating at high BMEP, the pressure can be up to 7 bar or more, and the pressure in the exhaust port at the valve is somewhere near 1 bar (atmospheric). As the valve opens, the pressure differential varies. Valve aperture (pressure ratio of approximately 7) starts exhaust gas flowing through the opening, and the outrush causes the pressure in the port (behind the valve) to increase rapidly.

The instantaneous velocity of the exhaust flow at any point can be determined by the pressure gradient & the cross-sectional area of that point. In header, a smaller tube diameter increases the velocity at a given RPM, which enhances the pressure wave tuning (the second component) and is beneficial with regard to inertia effects. However, if the diameter is very small, there will be flow losses and consequent pressure gradient increases which offset the tuning gains. So the selection of proper tube diameters is an important part of the design.

In the very first part of the exhaust cycle, the pressure difference across the valve is high. Thus, the instantaneous gas particle velocity through the small exhaust valve aperture is high. Sometime after mid-exhaust stroke, the majority of the exhaust gas left the cylinder. At that time, the valve aperture area is large and the cylinder pressure is approaching up to atmospheric, due to which the instantaneous particle velocity across the valve to be much lower. It is that phase of the exhaust cycle where the second component becomes important.

3.4 HARNESSING OF EXHAUST ENERGY

To help with the explanation of the second component, Fig. 3.4.1 shows traces of in-cylinder pressure in (black), port pressure at the intake valve (light blue) and port pressure at the exhaust valve (red), taken from a simulation of a high BMEP engine operating near the optimum tuning point for both intake and exhaust.

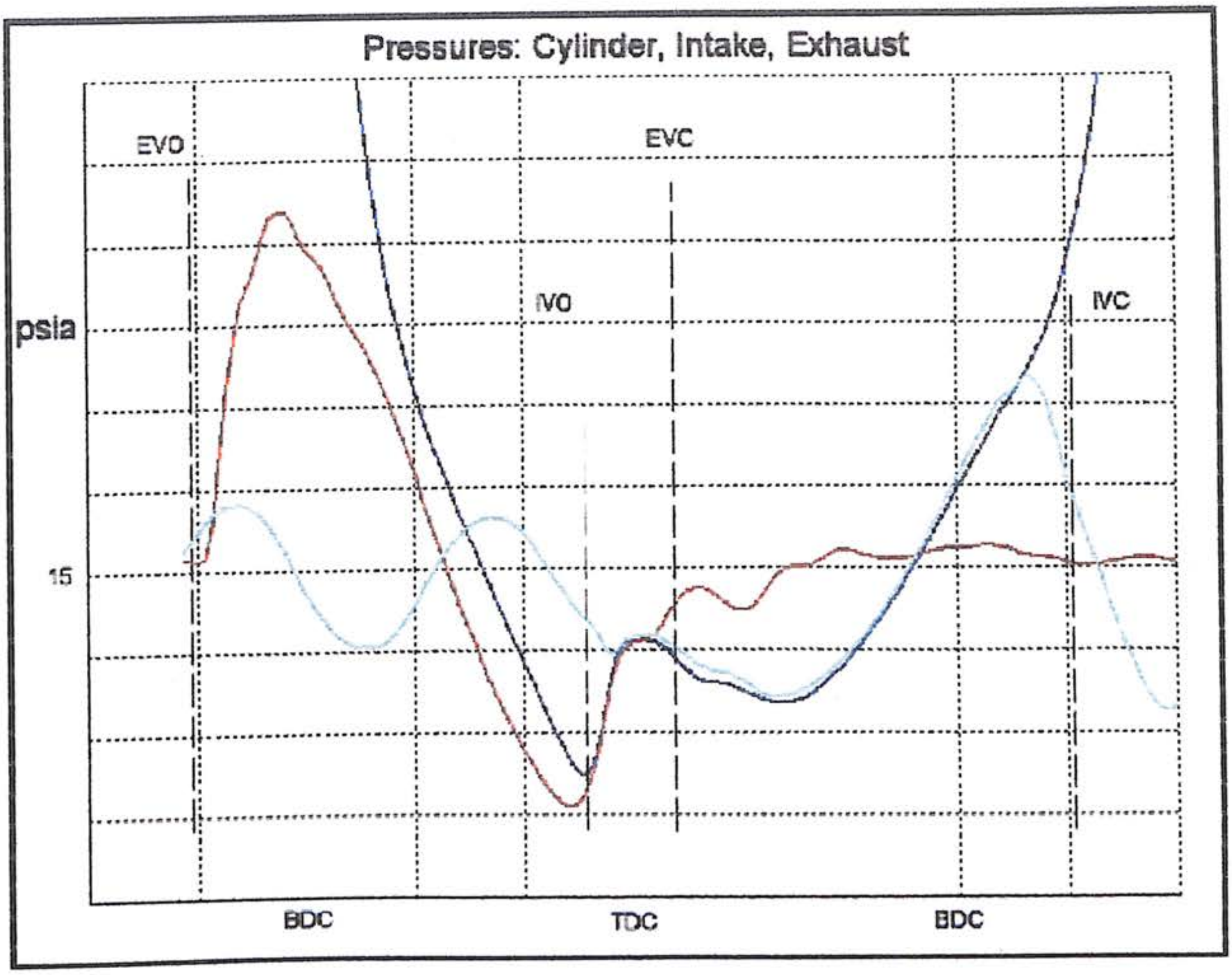


Fig.3.4.1 Intake and exhaust port Pressures with effective Tuning

The second component is the result of the pressure "spike" which occurs at EVO, shown by the peak in the red line in Fig. 5, just after EVO. That pressure spike, or pressure wave, moves down the pipe at the sum of the local sonic velocity plus the particle velocity of the gas flow. Whenever the pressure wave encounters a change in cross-sectional area of the pipe, a reflected pressure wave is generated, which travels in the opposite direction If the change in area is increasing (a step, collector, the atmosphere), the sense of the reflected pressure wave (compression or expansion) is inverted. If the change in area is decreasing (the end of another port having a closed valve, or a turbocharger nozzle, for example), the sense of the reflected wave is not inverted. The amplitude of the reflected wave is primarily determined by the proportionate change in cross-sectional area (area ratio), but the amplitude is diminished in any case. For purposes of approximation, the particle velocity can be ignored because its effect is self-canceling during the round-trip of the wave. However, highly-accurate simulations must take it into account. These waves are sometimes called finite difference waves, because of the finite difference numerical modeling techniques used to calculate their propagation characteristics.

In the case of the flow through header primary pipe, the EVO-initiated positive pressure (compression) wave is reflected back as a negative pressure (expansion) wave. If the arrival of the reflected negative pressure wave back to the exhaust valve arranged to occur during the latter part of the exhaust cycle, the resulting lower pressure in the port will enhance the easily removal of exhaust gas from the cylinder, and the pressure in the cylinder will reduce so that when the intake valve opens, the low pressure in the cylinder begins moving fresh charge into the cylinder while the piston is slowing down to a stop at TDC.

Note in Fig. 5, how the cylinder pressure (black) and exhaust port (red) pressures approaches strongly negative from approximately mid-exhaust stroke to TDC). Note also how the second-order reflected positive pressure wave of the intake tract (light blue) reaches the back of the intake valve at just before IVO, and works together with properly-timed exhaust negative pressures & begin moving fresh charge into the cylinder.

On the other hand, the negative exhaust pressure wave arrives at non-optimal time, its effect can be prejudicial to the clearing of the cylinder and in injection of fresh charge. A reflected positive wave during overlap period (from a turbocharger nozzle, for example) can push a large amount of exhaust gas back into the cylinder and also in the intake system.

Fig. 6 shows the same three pressure traces in which engine is operating well above the intake and exhaust tuning points. In addition, to reduce breathing efficiency, note the additional pumping losses occur in the higher cylinder pressure in the latter portion of the exhaust cycle, caused in part by the late arrival of the reflected negative exhaust pulse.

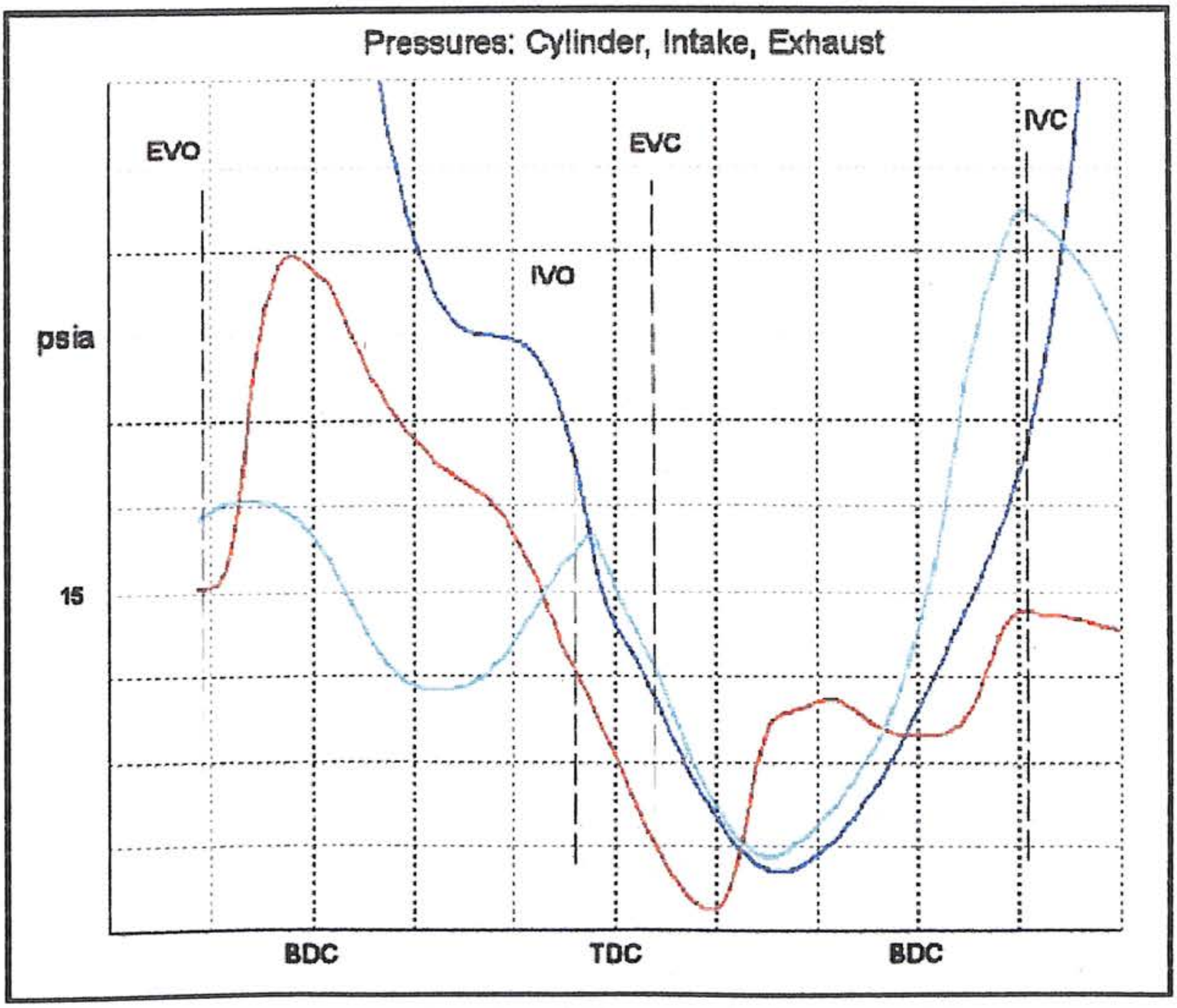


Fig.3.4.2 Intake and exhaust Port Pressures with Poor Tuning

The timing during the arrival of the negative wave at the back (port) side of the exhaust valve is determined by the engine RPM, and the speed of sound in the pipe and the distance from the valve to the relevant change in area. These three factors will cause the exhaust tuning to come in and out tune the engine operating speed range.

It is sometimes reasoned that the speed of sound is a function of pressure, density, temperature. But in really, the speed of sound in an ideal gas (which air emulates) is a function of the harshness of the gas divided by the density. When one does the arithmetic necessary to create an equation which uses the known parameters, the harshness and density terms are replaced by equivalents from the ideal gas law, producing the equation: Va (acoustic velocity m/s) = square root (S x R x T), where S is the ratio of specific heats (approximately 1.4 for air at 25°C, 1.35 for exhaust gas at

500°K), R is the gas constant (approximately 287 J/kg-°K for air, 291 for exhaust gas) and T is the absolute temperature (°Kelvin, which is °C + 273).

The pressure ratio increases across a smoothly-decreasing nozzle, the particle velocity at the smallest cross-sectional area increases by increasing pressure ratio until it reaches to the local speed of sound. Once, it reached the speed of sound no matter how much larger the pressure ratio is, the gas particle velocity remains choked. An increase in the upstream pressure will increase the mass flow rate & to the increased density upstream of the nozzle, but the particle velocity through the nozzle will remain sonic.

The air flowing in a smoothly-decreasing nozzle, the pressure ratio which is just cause's sonic flow (the 'critical pressure ratio') is slightly less. For a non-smooth and irregular nozzles (an exhaust valve, for instance) the critical pressure ratio should be higher, but the effect is the same. That means, for the same period of time after EVO, the gas particle flow velocity across the exhaust valve is at the local speed of sound.

3.5 GOALS OF A PERFORMANCE EXHAUST SYSTEM

The goals of a performance exhaust system is to:

a) To efficiently remove as much of the combusted inert exhaust gases out of the cylinder.

If it is still left in the cylinder, it takes up space in the cylinder and prevents fresh air and fuel from coming into the combustion chamber to make power.

b) To keep the velocity or speed of the exhaust gas leaving very high.

When high exhaust gas speeds are reached, a wave is created from an exhaust pulse leaving the cylinder head. Following behind this wake is a low pressure wave that acts like a vacuum. This vacuum sucks in more fresh air and fuel at cam overlap, when the intake valve is just starting to open and the exhaust valve is almost about to close. Since both the intake & exhaust valves are partially open at this time of cam overlap, header is actually "connected" to the intake manifold & intake port for a brief period. The exiting exhaust gas helps pull in the next fresh intake air & fuel. This is called scavenging. And scavenging is what helps draw in more oxygen and fuel for combustion. More fresh air and fuel coming in, with less inert burnt exhaust gases occupying combustion chamber volume, makes more power.

3.6 PARTS OF EXHAUST SYSTEM

The Exhaust Expansion Chamber consists of :

- Header Pipe
- Megaphone
- Diffuser or Divergent cone
- Expansion chamber
- Baffle cone or Convergent cone
- Tail pipe or Stinger

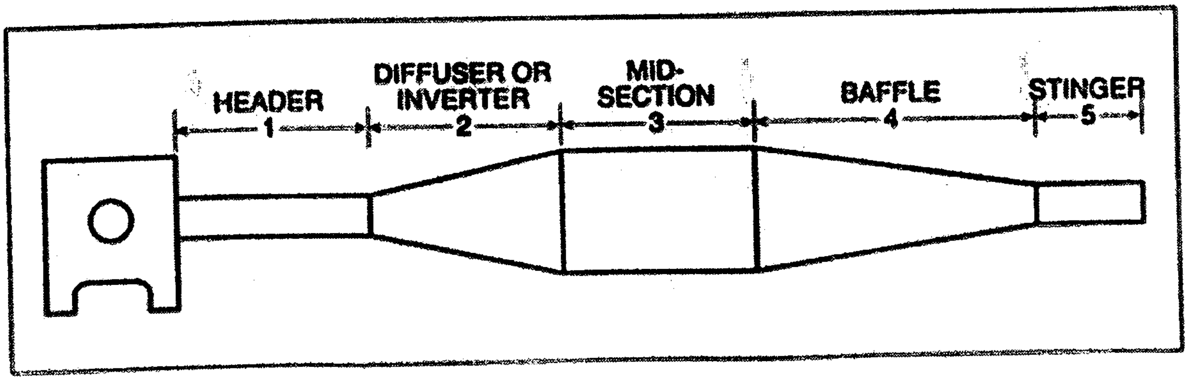


Fig.3.6.1 major components of a silencer

3.7 WORKING OF AN EXHAUST SYSTEM

When we step by step increases the diameter of the tube, a gradual but more useful negative wave is generated which helps in scavenge and pull spent gasses out of, the cylinder. Adding Divergent Tubes, which is used to be called as "Megaphones," help in making more power.

Fig.3.7.1 Divergent cone

By putting a divergent cone on the end of a straight pipe will lengthens the returning wave, extending the power band and creating a vestigial expansion chamber. To sum up when the negative wave arrives the exhaust port at the exact time, it will pull some of the exhaust gases out the cylinder and help the engine to scavenge the spent exhaust gas. By placing a divergent cone at the end of the straight (parallel) "head" pipe widens the returning wave. The returning negative wave is not as strong, but it is more retentive, so it is more likely to open the exhaust port and is capable to pull out the exhaust gases. As with plain, straight pipes, the total length of the pipe with a divergent cone welded on determines the timing of the return pulses and therefore the efficient engine speed. The divergent cone's vital dimensions are where it starts (the distance from. The exhaust port to the start of the divergent cone is called the "head" pipe), while the length of the megaphone and the rate at which it departs from the straight pipe determine the intensity and length of the returning wave--A short pipe which diverges at a acute angle from the head pipe gives a stronger and more straight-pipe-like pulse. Conversely, a long gradual divergent cone creates a smaller pulse for longer duration. In addition to the negative wave is also strong enough to help in pull of fresh mixture up through the transfer ports.

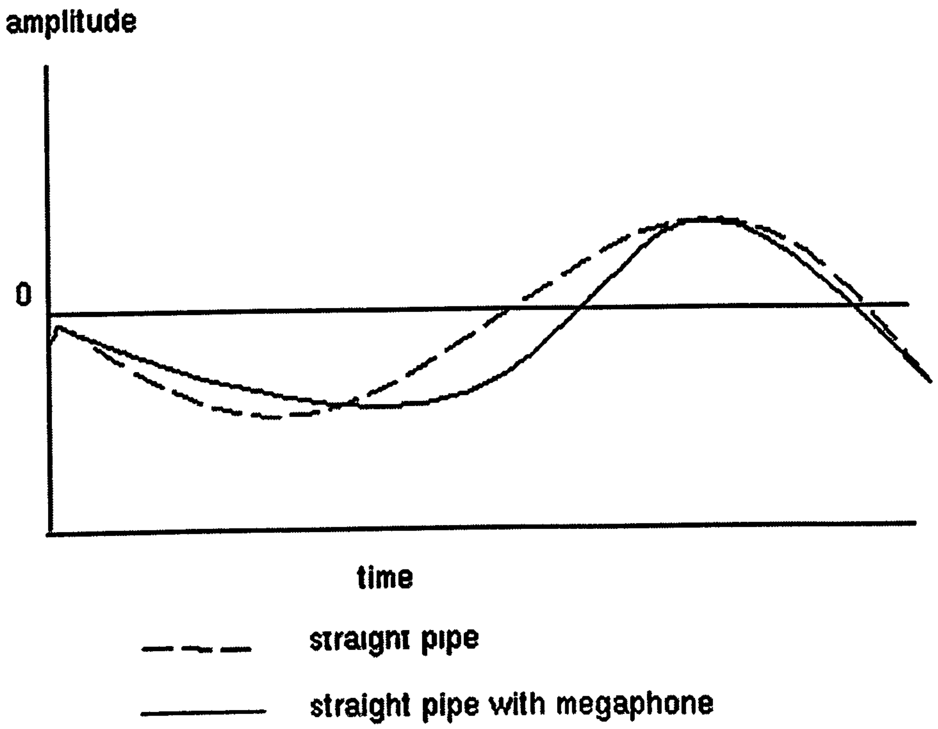


Fig3.7.2. Wave reflected from end of pipe

And while adding a divergent cone to the head pipe developed great tuning advantages, it had its limitations too: The wider negative wave from a megaphone can still arrive too early and pull fresh mixture out of the cylinder. That's the problem that Walter Kaaden was facing in factory MZs. Then he realized that putting another cone just reversed to be convergent, on the end of the first divergent pipe may be reflect positive waves back up the pipe. These positive waves would follow the negative waves back to the exhaust port, and if decently timed it would stuff the fresh mixture that was pulled into the pipe back into the exhaust port right as the piston closed the port.

In addition to head pipe length, divergent and convergent cone lengths, an expansion chamber has 3 more crucial dimensions. The length of the straight Belly between the divergent and the convergent cones, the length of the tailpiece 'stinger' and muffler, and the diameter of the belly section. The stinger acts as a pressure bleed, which allows the pressure to escape from the pipe. Back pressure in the pipe, caused by a smaller-diameter or longer stinger section, this helps the wave action of the pipe, and increases the engine's performance. This happens since the greater pressure creates a denser, uniform medium for the waves to act on--waves travel better through dense, uniform mediums.

Mr.Kaaden immediately realized a large power gain, and the expansion chamber was born.

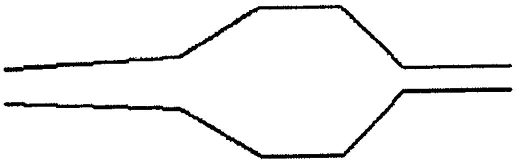


Fig 3.7.3 The middle parallel section 'belly'.

The length of the belly section determines the relative timing between the negative and positive waves. The timing of the waves is determined by the length of the straight pipe. If the belly section is too short, positive waves have a shorter distance to travel, and return to the exhaust port sooner.

The principle of two-stroke engine exhaust system tuning was experimented in four-stroke engine by an American race tuner A. Graham Bell. He discovered the black art of pulse tuning by efficiently scavenging the cylinder of exhaust gas by gas momentum or inertia tuning. The principle of inertia tuning is that exhaust gases have weight so once it gets rolling it will continue to flow even after the exhaust valve has closed. This creates a partial vacuum with a resultant suction action which can be used to scavenge the cylinder. As engine speed increase, the time available for effective cylinder exhausting will decrease, hence the need to use this suction to empty the cylinder of exhaust gas more quickly. The exhaust gas is expelled from the cylinder at a velocity of between 200 and 300 ft/s, but pulses or pressure waves are moving through that gas at around 1500 to 1700 ft/s.

As the initial charge of burnt gas bursts from the cylinder into the exhaust system, it creates a wave of positive pressure which travels at the speed of sound through the gas to the end of the pipe. As it surges into the atmosphere, the positive wave dissipates and produces a negative pressure wave (suction wave), which returns along the exhaust in to the cylinder. It arrives with a certain amount of evacuation power because its pressure is much lower than the cylinder pressure. The art of exhaust tuning is to determine the length and size of the exhaust pipe for this suction wave to arrive back at the cylinder during the valve overlap period.

CHAPTER 4

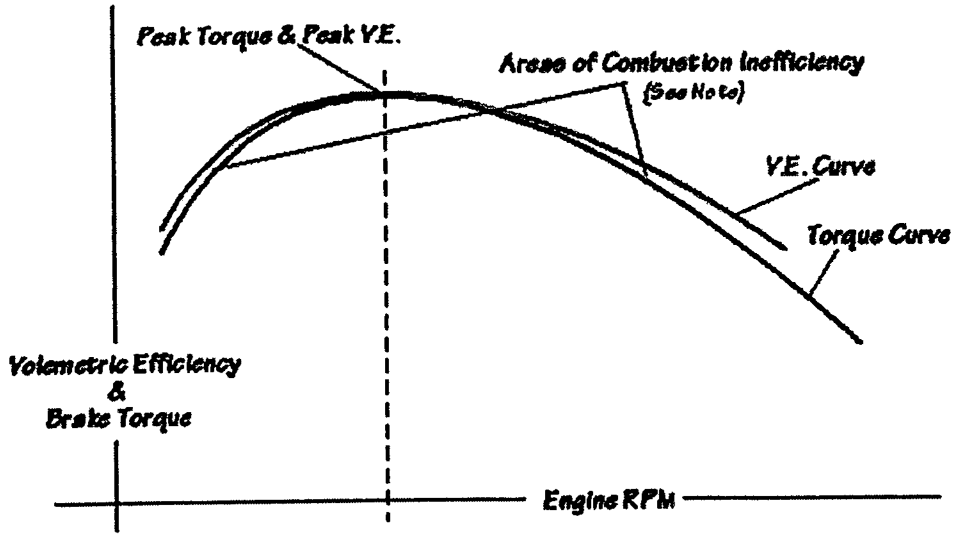
DESIGN PARAMETER

4.1 BACK PRESSURE

Back pressure refers to the resistance to a moving fluid by obstructions or tight bends in the confinement vessel along which it is moving, such as piping or air vents, against its direction of flow. The amount of back pressure produced by the exhaust system is essential but too much back pressure will be having a negative effect on the engine's top-end performance and it will restrict the flow rate of the exhaust gasses at high RPM. The result would be the engine not be able to expel the spent exhaust gasses fast enough to prevent spent exhaust gasses from fouling the fresh air/fuel mixture that is drawn into the engine on the next intake stroke. Finally , it will result in reduced engine power. If the exhaust pipe is too large, the flow velocity of the exhaust gasses gets reduced. The flow velocity of the exhaust gasses help the scavenging of the spent exhaust gasses as well as the amount of air/fuel mixture that can be drawn into the combustion chamber in the next intake stroke. This is because the flow velocity of the exhaust develops a low pressure immediately behind it that sucks more gasses out of the combustion chamber. The size of the exhaust header primary pipes is also important as it affects both back pressure and flow velocity; while the length of the primary pipes affect the power band of your engine. The size and length of the primary pipes of the exhaust manifold, as well as exhaust header design depends on the engine's displacement and maximum usable RPM & the power band required from the engine.

4.2 GAS FLOW VELOCITY

Exhaust gas flow velocity determines where peak torque occurs along the rpm range. As a general rule, when exhaust flow velocity reaches the mean value of **240-260 ft/sec.**, peak torque is achieved. Peak torque also marks when the engine has achieved its highest volumetric efficiency or maximal cylinder filling ability.



NOTE: Areas of "combustion inefficiency" show where the fact that air is present in the cylinders does not always translate into torque. Up to peak torque rpm, inefficient charge mixing decreases combustion efficiency. Above peak torque rpm, there is either insufficient time to combust the air/fuel charge or charge quality is reduced by air/fuel mixture separation or dilution with combustion byproducts, typically exhaust gas.

Fig 4.2.1 Variation of peak torque with engine RPM

4.3 LENGTH AND DIAMETER

Bigger diameter shifts peak torque to a higher rpm compared to a smaller diameter. Bigger the diameter, more the cross-sectional area. Exhaust flow must overcome this extra tube cross-sectional area and therefore the flow travels slower. It takes the engine speed to climb to a higher rpm before the speed of 240 ft/sec and therefore, peak torque is reached. So increasing diameter shifts when 240 ft/sec and peak torque is achieved to a higher or later rpm, because it takes longer for the air flow speed to reach 240 ft/sec. In addition, a bigger diameter will increase the actual peak torque number that is not only does the diameter change the location, it also increases torque. Diameter can be varied, as well, along the length of the header tube. This is called "stepping" the header. A "stepped" header will have along its length the diameters gradually increasing as it moves towards the muffler end and away from the engine. Stepping a header will prevent exhaust flow from travelling backwards to the engine (called reversion). Stepped headers therefore have anti-reversion characteristics, as well as achieving a broader power band. The pipe diameter can be used to change the peak torque rpm a reduction in diameter of 0.125 inches will drop the peak torque rpm by

500-600 rpm in engines over 2 litres and by 650-800 rpm in smaller engines. Increasing the pipe diameter by 0.125 rpm has approximately the opposite effect.

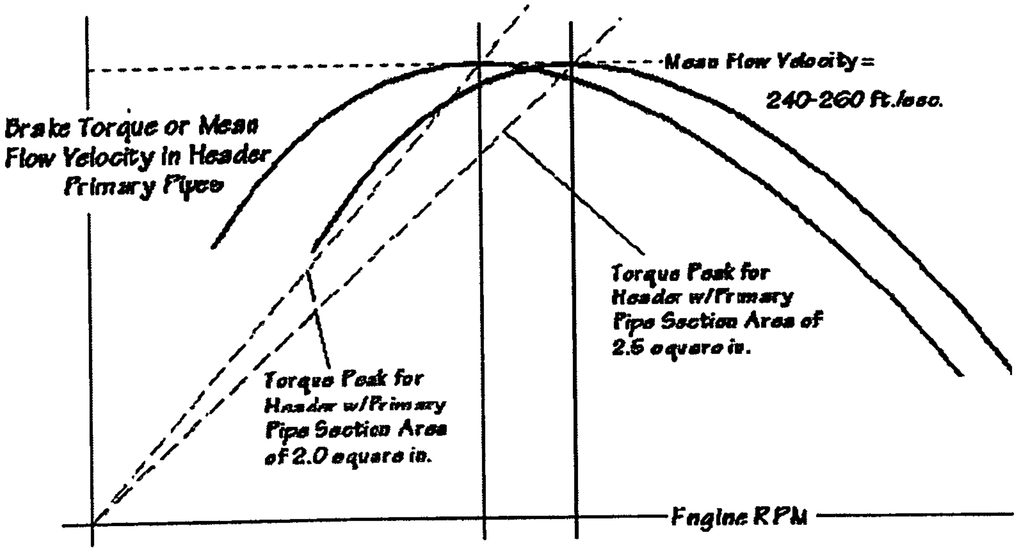


Fig 4.3.1 variation of break torque with engine RPM

Longer tubes will speed up air flow velocity. The flow velocity of 240 ft/sec and peak torque will occur at an earlier rpm compared to a shorter tube. Changing the length of the header primary tubes does not increase the value of peak torque like diameter does. Instead length changes the behaviour of the torque around peak torque along the rpm band. Therefore longer tubes shift the peak torque to a lower rpm whereas shorter tubes will shift it to a higher rpm.

CHAPTER 5

SOLIDWORKS MODELLING AND ANALYSIS

5.1 PART MODELLING

Solid Works Corporation founded in December 1993. Jon Hirsch tick had his headquarters in Massachusetts, US. He hired a team of engineers to start a company which can develop 3D CAD software that were easy-to-use, affordable and available on the Windows desktop, released its first product, Solid Works 95 in 1995. Solid Works currently markets several versions of the Solid Works CAD software in addition to e-Drawings, collaboration tools and Draft sight.

We made a model of silencer in this software to be used in Gambit for meshing & fluent for analysis at a further stage.

The very first model was made using the dimension directly took from an original stock silencer that we bought & had it cut to measure the exact dimensions.

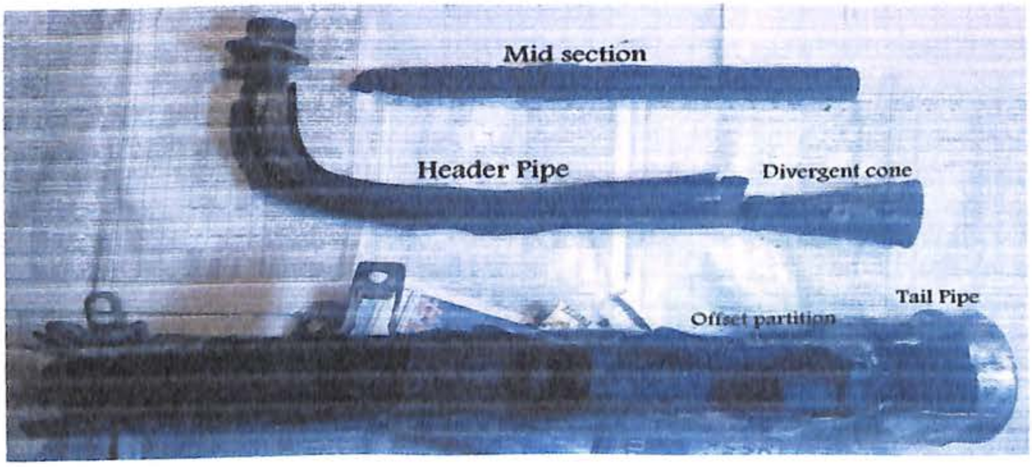


Fig.5.1.1 Original Stock Silencer Cut-Section

The geometry made in solidworks that was the replica of the original stock silencer is shown below:

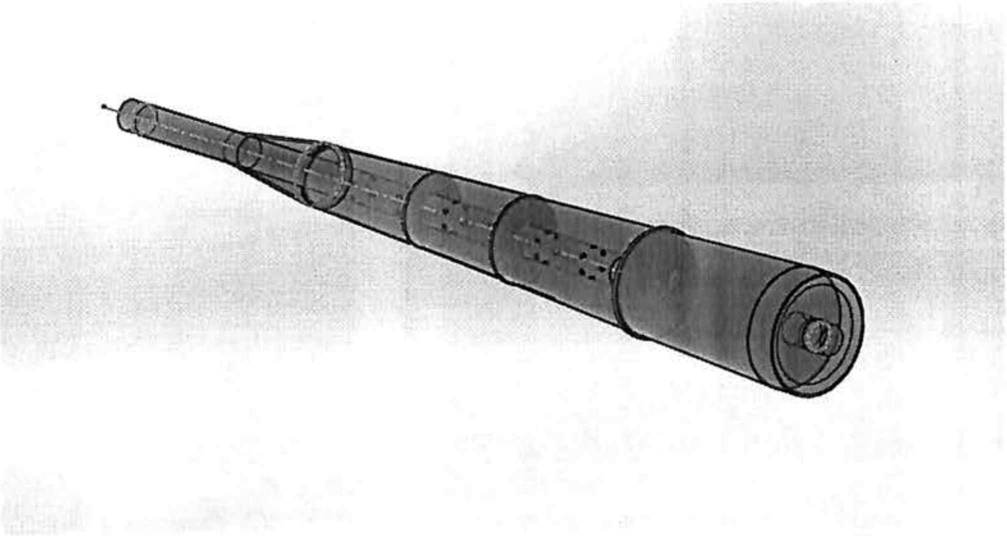


Fig.5.1.2 Solidworks Model of Silencer

Tools Used for development of the geometry:

1. Sketcher
2. Extrude commands
3. Loft commands
4. Fillets
5. Linear & Circular patterns

The geometry was made as an assembly in solid works consisting two major SLD parts.

Separate parts were made in accordance with the dimensions of the original silencer.

5.2 PART ASSEMBLY

Parts that were assembled can be seen:

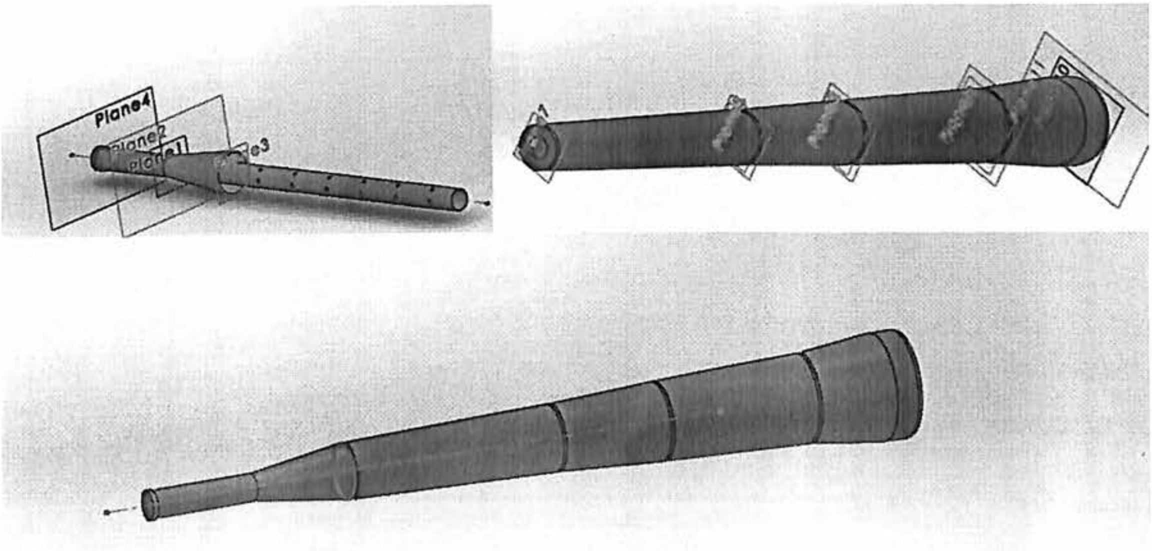


Fig. 5.2.1 assembly of silencer parts

Final assembly can be seen as:

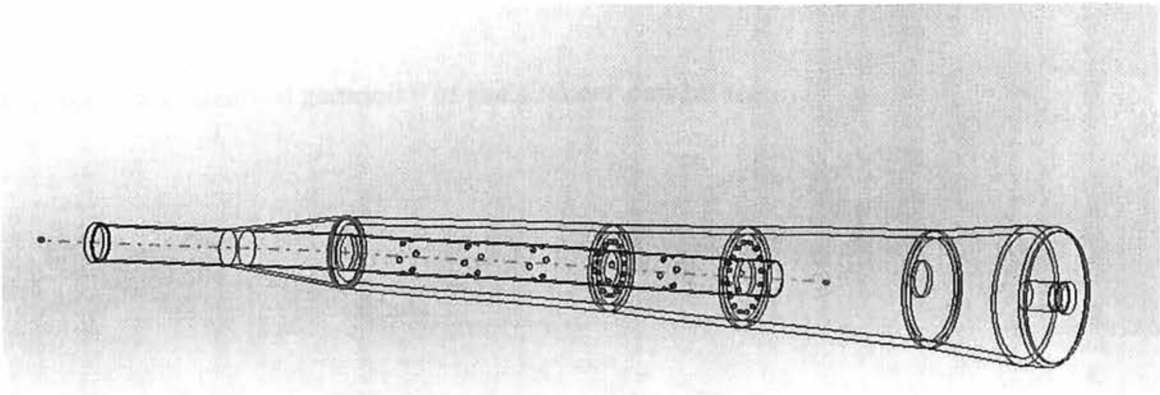


Fig. 5.2.2 final transparent view of geometory

Drawing sheet:

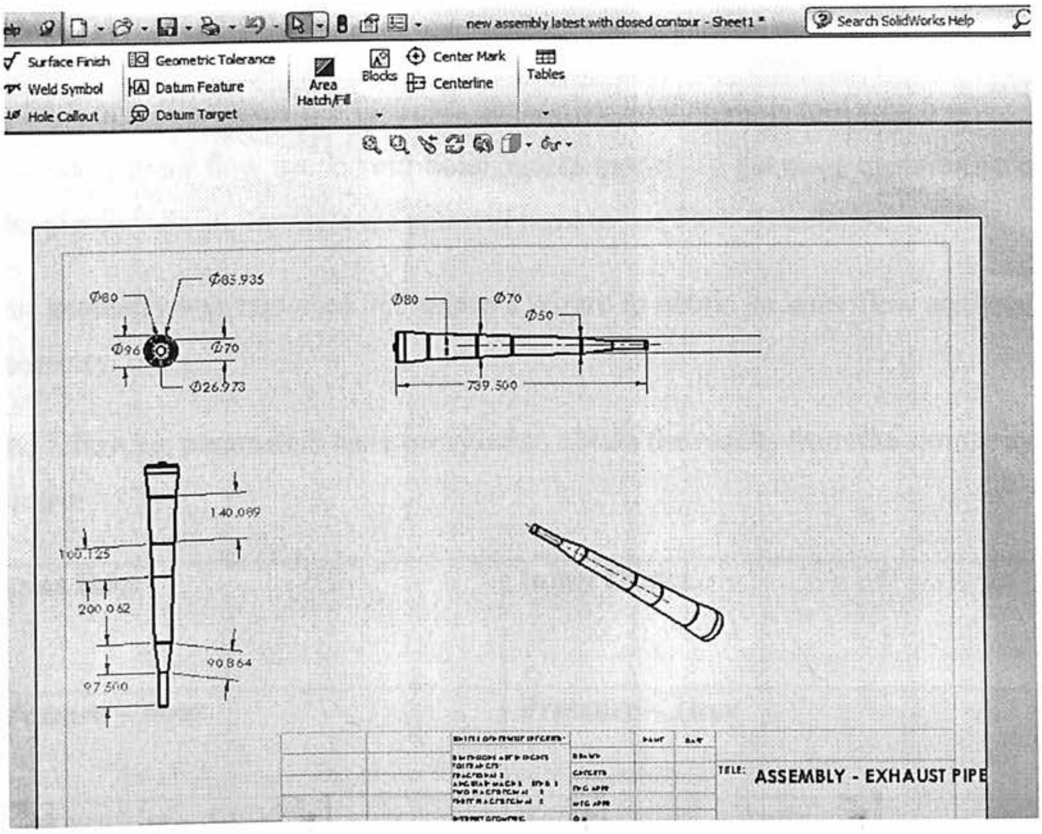


Fig.5.2.3 Drawing sheet of geometry

Cut section & drafted geometry of the silencer can be seen:

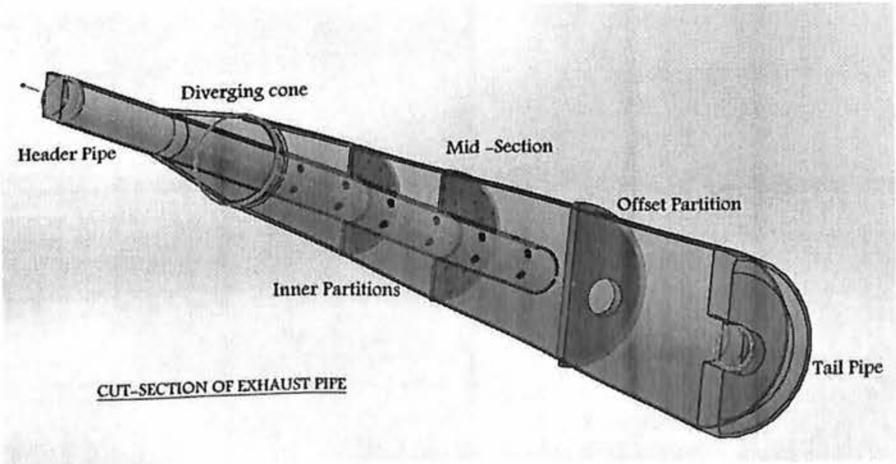


Fig.5.2.4 Cut Section View of Solidworks Model

5.3 FLOW ANALYSIS:

Solid Works FloXpress is a first pass qualitative flow analysis tool which gives insight into water or air flow inside your Solid Works model. To get more quantitative results like pressure drop, flow rate etc you will have to use Flow Simulation

Our geometry was imported in Floxpress wizard to obtain an inner flow analysis of the geometry.

The following parameters were provided to obtain the results from the simulation wizard:

Input Inlet	**Input Outlet**
Pressure – 6bar	**Pressure – 1bar**
Temperature – 540K	**Temperature – Ambient**

Analysis Results can be seen:

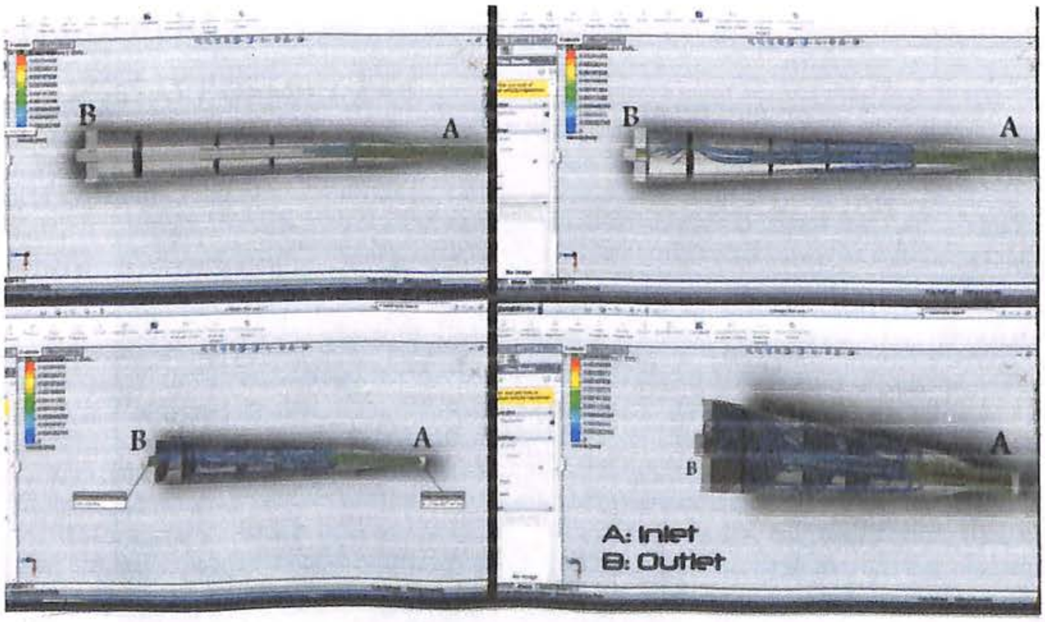


Fig.5.3.1 Flow Simulation Wizard Analysis

CHAPTER 6

GAMBIT AND FLUENT ANALYSIS

GAMBIT:

Gambit is used to create and mesh the geometry. In software the geometry is imported as STEP file. In this software we perform the following task:

- Import the STEP file.
- Specify the way in which the geometry will be colored.
- Volume meshing of exhaust pipe is done.
- Examine the mesh, and set up the initial boundary condition.
- Prepare the mesh to be read into FLUENT 6.3.26.

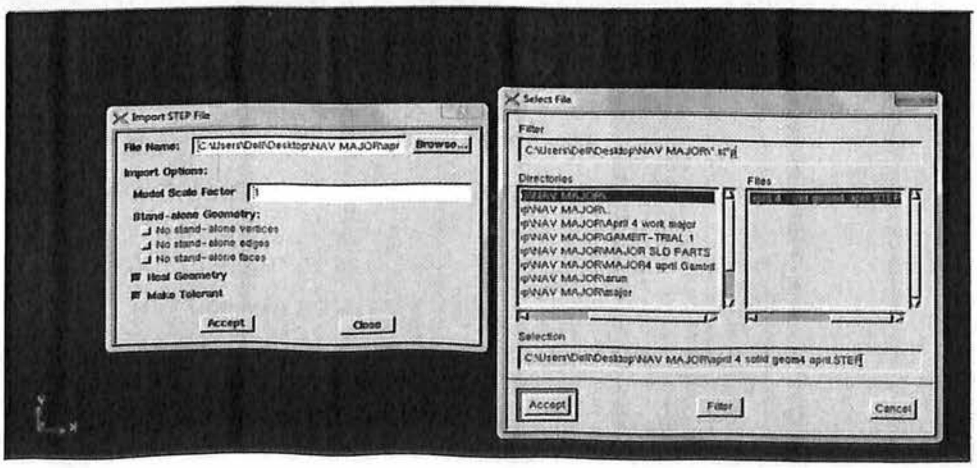
A hex/wedge mesh is created around the volume of the exhaust pipe for the analysis. After examining the raw imported geometry to identify its problems, such as unconnected edges, geometry is cleaned up using the clean up tool available in GAMBIT.

For generating mesh following steps are followed.

6.1 GEOMETORY IMPORT

PROCEDURE

- ➢ START GAMBIT 2.4.6
- ➢ Import the STEP file



File Name: C:\Users\Dell\Desktop\NAV MAJOR\apr

Model Scale Factor 1

Stand-alone Geometry:

☐ No stand-alone vertices
☐ No stand-alone edges
☐ No stand-alone faces
☑ Heal Geometry
☑ Make Tolerant

Filter: C:\Users\Dell\Desktop\NAV MAJOR*.st*

Directories:

p\NAV MAJOR\.
p\NAV MAJOR\..
p\NAV MAJOR\April 4 work major
p\NAV MAJOR\GAMBIT-TRIAL 1
p\NAV MAJOR\MAJOR SLD PARTS
p\NAV MAJOR\MAJOR4 april Gambit
p\NAV MAJOR\arun
p\NAV MAJOR\major

Selection: C:\Users\Dell\Desktop\NAV MAJOR\april 4 solid geom4 april.STEP

Fig:6.1.1 Importing step file of geometry into gambit

- Imported STEP geometry as shown below

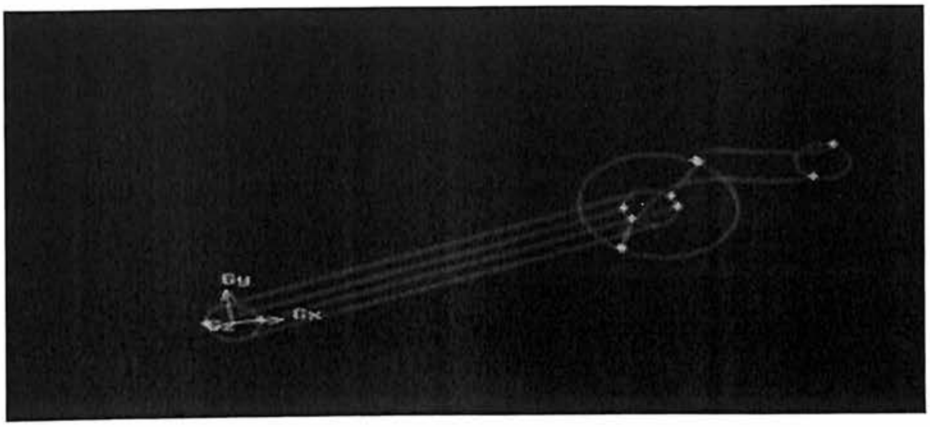


Fig: 6.1.2 imported view of geometry

6.2 MESH GENERATION

- Volume meshing

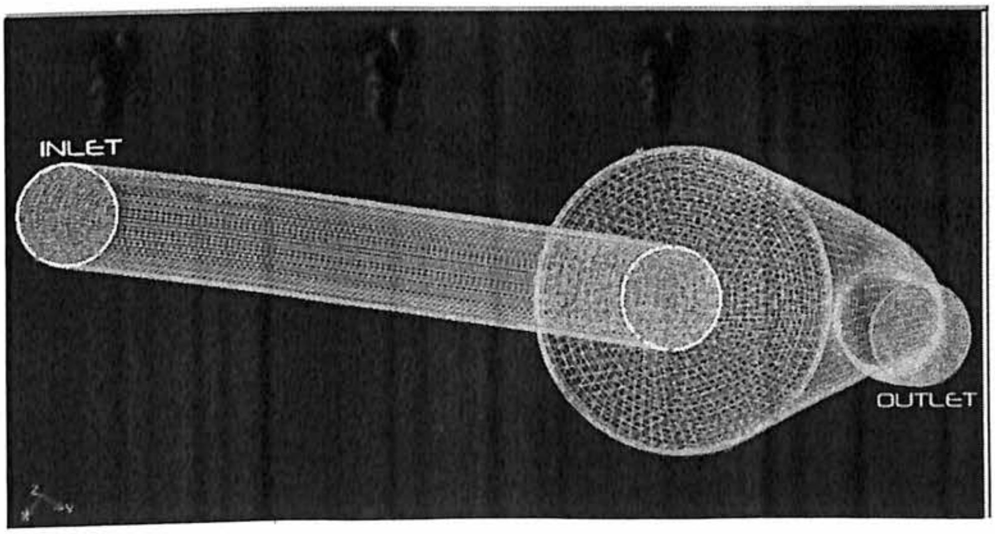


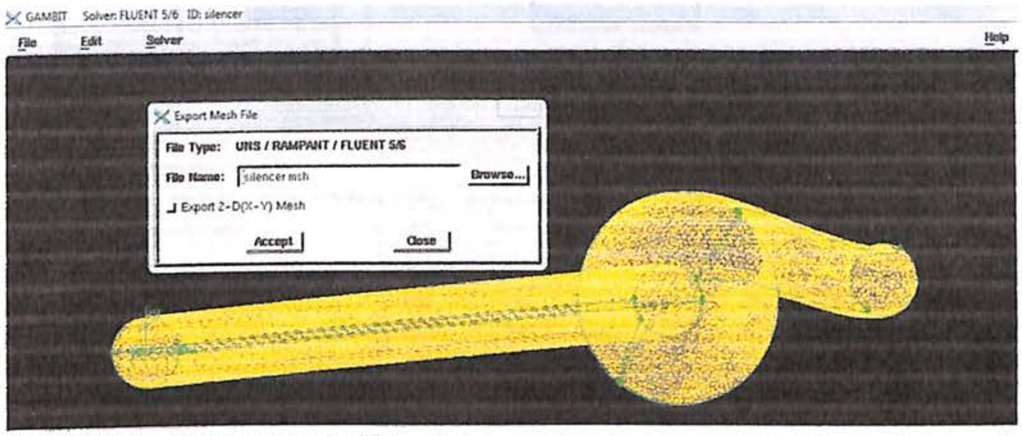
Fig:6.2.3 volume mesh

- Examine and initialize the boundary conditions

:

Examine mesh: It is important that quality of the resulting mesh is checked, because skewness can greatly affect the accuracy and robustness of the CFD solution. GAMBIT provides several quality measures with which you can assess the quality of your meshing. In case of skewness measures such as Equiangle Skew and Equisize Skew, smaller values are more desirable.

- Export the mesh file



Solver: FLUENT 5/6 ID: silencer

File Type: UNS / RAMPANT / FLUENT 5/6

File Name: silencer.msh

☐ Export 2-D(X-Y) Mesh

Fig:6.2.2 exporting mesh file

6.3 FLUENT ANALYSIS

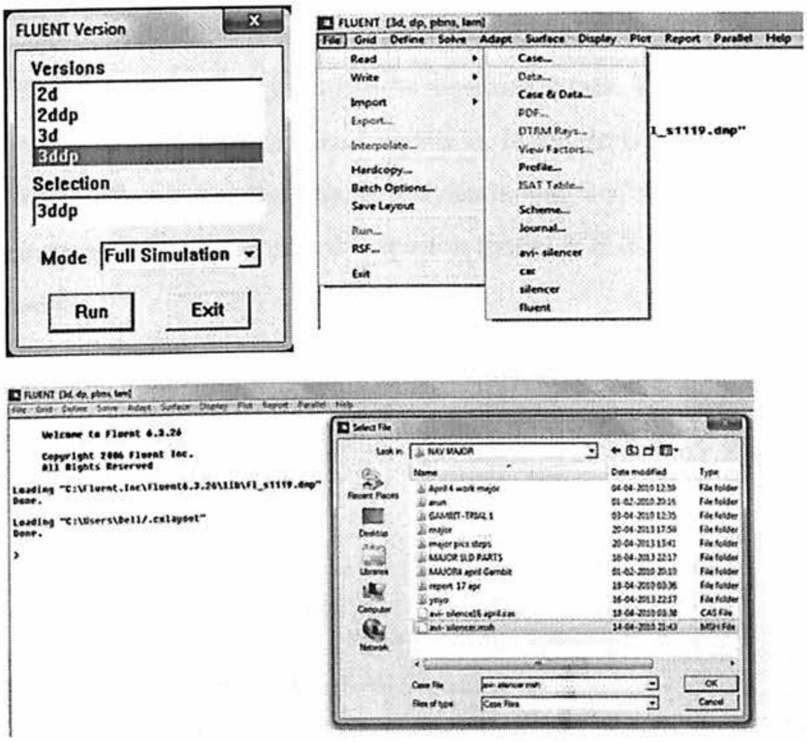
Fluent is a commercial CFD software developed in 'C' language. It follows a control volume approach. Fluent is the best CFD tool for the analysis of flow and heat transfer problems especially in process industries.

For the analysis following procedure is followed

- Read the imported grid file
- Use units to define the geometry and fluid properties
- Set material properties and Boundary condition for a k-€ problem.
- Initiate the calculation with residual plotting
- Calculate a solution using segregated solver
- Examine the pressure curves and flow field using graphics
- Enable the first order discretization scheme for improved prediction of temperature
- Find the results and plotted graph

- START 3ddp Version of FLUENT

Read the mesh file



Selection: 3ddp

Mode: Full Simulation

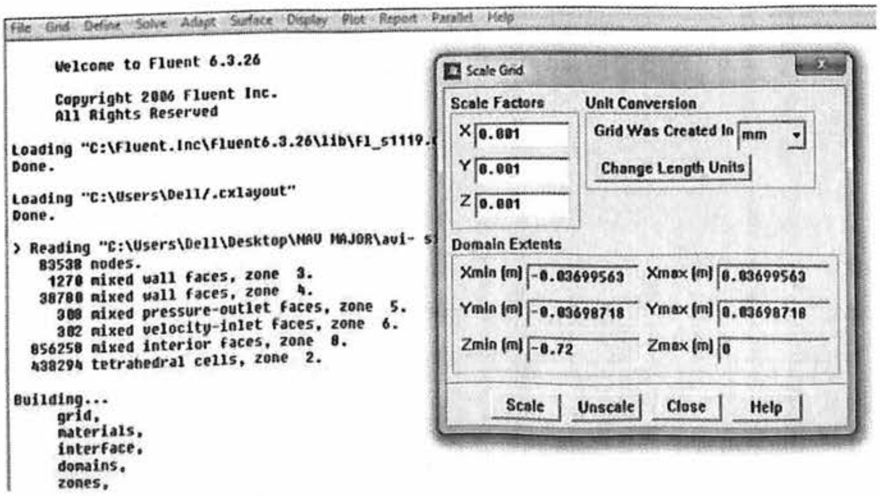
```
Welcome to Fluent 6.3.26

Copyright 2006 Fluent Inc.
All Rights Reserved

Loading "C:\Fluent.Inc\Fluent6.3.26\lib\Fl_s1119.dmp"
Done.

Loading "C:\Users\Dell/.cxlayout"
Done.

>
```

Case File: avi-silencer.msh

Files of type: Case Files

Fig: 6.3.1 importing of mesh file in fluent

- Scale the mesh file and check the grid



```
Welcome to Fluent 6.3.26

Copyright 2006 Fluent Inc.
All Rights Reserved

Loading "C:\Fluent.Inc\Fluent6.3.26\lib\Fl_s1119.
Done.

Loading "C:\Users\Dell/.cxlayout"
Done.

> Reading "C:\Users\Dell\Desktop\NAV MAJOR\avi- s
   83538 nodes.
    1270 mixed wall faces, zone  3.
   38700 mixed wall faces, zone  4.
     308 mixed pressure-outlet faces, zone  5.
     302 mixed velocity-inlet faces, zone  6.
  856250 mixed interior faces, zone  8.
  438294 tetrahedral cells, zone  2.

Building...
     grid,
     materials,
     interface,
     domains,
     zones,
```

X 0.001

Y 0.001

Z 0.001

Grid Was Created In mm

Xmin (m) -0.03699563 Xmax (m) 0.03699563

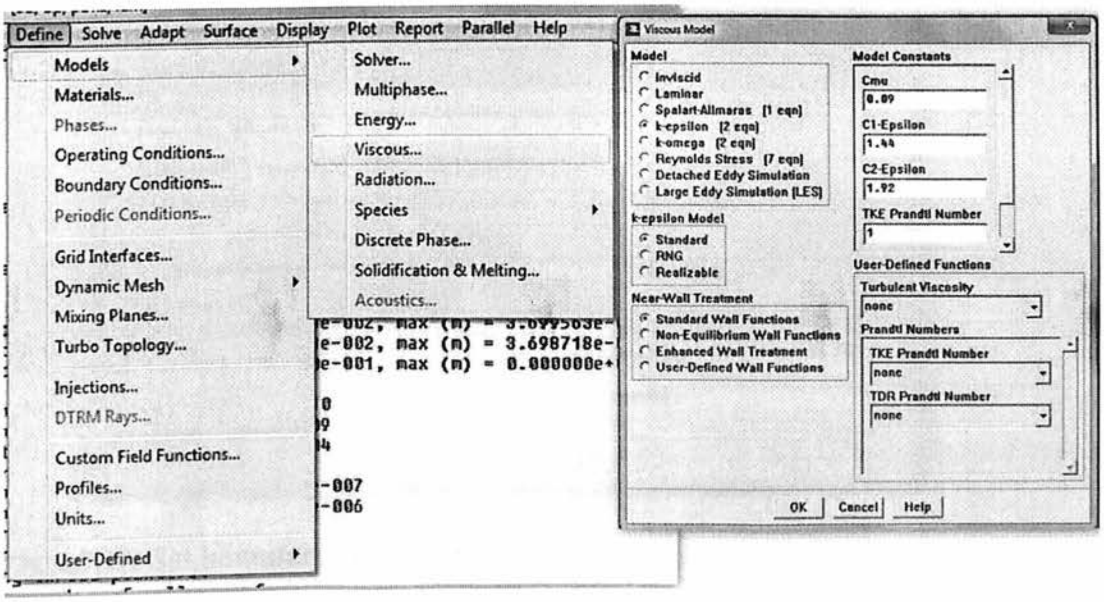
Ymin (m) -0.03698718 Ymax (m) 0.03698718

Zmin (m) -0.72 Zmax (m) 0

Fig 6.3.2 check the grid

Models and Materials

For this turbulent exhaust pipe model K-epsilon model equation is selected. The standard k-epsilon model is a semi-empirical model based on model transport equations for the turbulence kinetic energy (k) and its dissipation rate. The model transport equation for k is derived from the exact equation. In the derivation of the k-epsilon model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard k-epsilon model is therefore valid for fully turbulent flows.



```
e-002, max (m) = 3.699563e-
e-002, max (m) = 3.698718e-
e-001, max (m) = 0.000000e+
0
9
4

-007
-006
```

Viscous Model

Model
- ○ Inviscid
- ○ Laminar
- ○ Spalart-Allmaras [1 eqn]
- ● k-epsilon [2 eqn]
- ○ k-omega [2 eqn]
- ○ Reynolds Stress [7 eqn]
- ○ Detached Eddy Simulation
- ○ Large Eddy Simulation [LES]

k-epsilon Model
- ● Standard
- ○ RNG
- ○ Realizable

Near-Wall Treatment
- ● Standard Wall Functions
- ○ Non-Equilibrium Wall Functions
- ○ Enhanced Wall Treatment
- ○ User-Defined Wall Functions

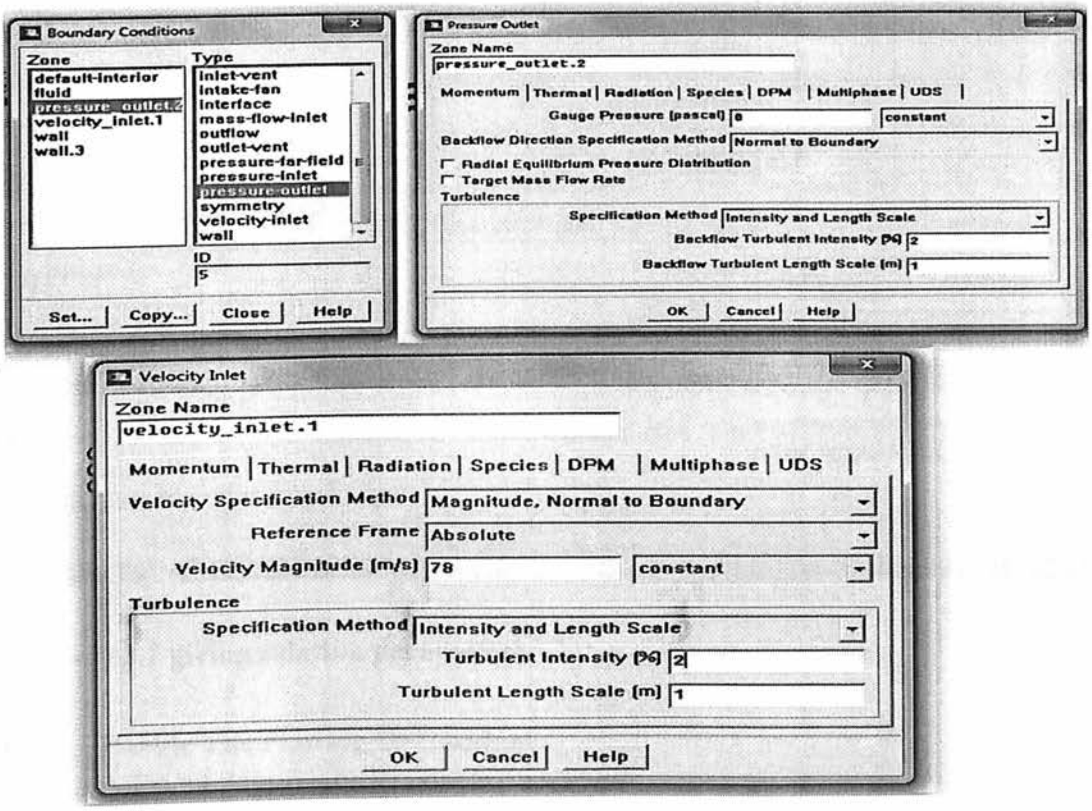
Model Constants
- Cmu: 0.09
- C1-Epsilon: 1.44
- C2-Epsilon: 1.92
- TKE Prandtl Number: 1

User-Defined Functions
- Turbulent Viscosity: none
- Prandtl Numbers
  - TKE Prandtl Number: none
  - TDR Prandtl Number: none

Fig:6.3.3 material selection

6.3.1Define Boundary Conditions

Boundary conditions are defined for inlet and outlet of the exhaust pipe.
Inlet: Velocity -78 m/s.
Outlet: Turbulent intensity 2%
Wall thickness : 0.003125m



Boundary Conditions

Zone: default-interior, fluid, pressure_outlet.2, velocity_inlet.1, wall, wall.3

Type: inlet-vent, intake-fan, interface, mass-flow-inlet, outflow, outlet-vent, pressure-far-field, pressure-inlet, pressure-outlet, symmetry, velocity-inlet, wall

ID: 5

Pressure Outlet

Zone Name: pressure_outlet.2

Gauge Pressure (pascal): 0 constant

Backflow Direction Specification Method: Normal to Boundary

☐ Radial Equilibrium Pressure Distribution

☐ Target Mass Flow Rate

Turbulence

Specification Method: Intensity and Length Scale

Backflow Turbulent Intensity (%): 2

Backflow Turbulent Length Scale (m): 1

Velocity Inlet

Zone Name: velocity_inlet.1

Velocity Specification Method: Magnitude, Normal to Boundary

Reference Frame: Absolute

Velocity Magnitude (m/s): 78 constant

Turbulence

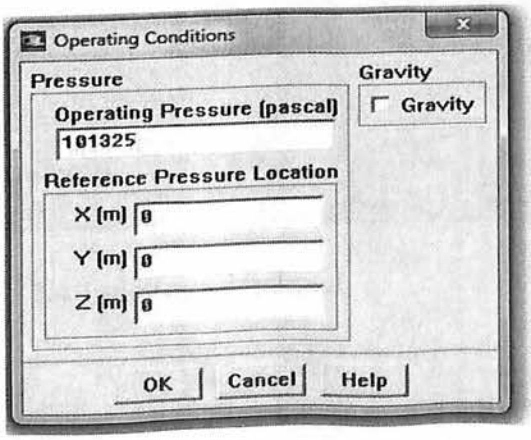
Specification Method: Intensity and Length Scale

Turbulent Intensity (%): 2

Turbulent Length Scale (m): 1

Fig:6.3.1.1 Set boundary condition

6.3.2 Define Operating Condition



Pressure

Operating Pressure (pascal): 101325

Reference Pressure Location

X (m): 0

Y (m): 0

Z (m): 0

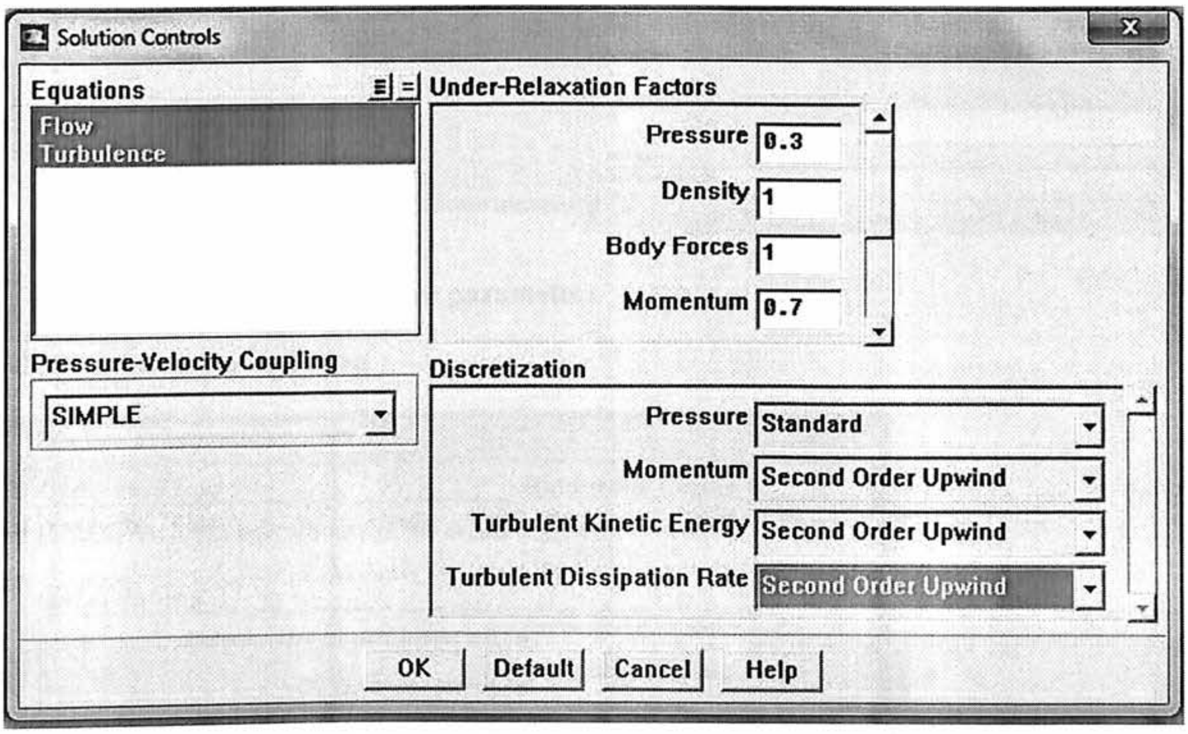
Gravity

☐ Gravity

Fig: 6.3.2.1 giving operating conditions

6.3.3Define Solution Parameter

Solve → Control → Solution



Equations: Flow, Turbulence

Under-Relaxation Factors

Pressure 0.3

Density 1

Body Forces 1

Momentum 0.7

Pressure-Velocity Coupling: SIMPLE

Discretization

Pressure: Standard

Momentum: Second Order Upwind

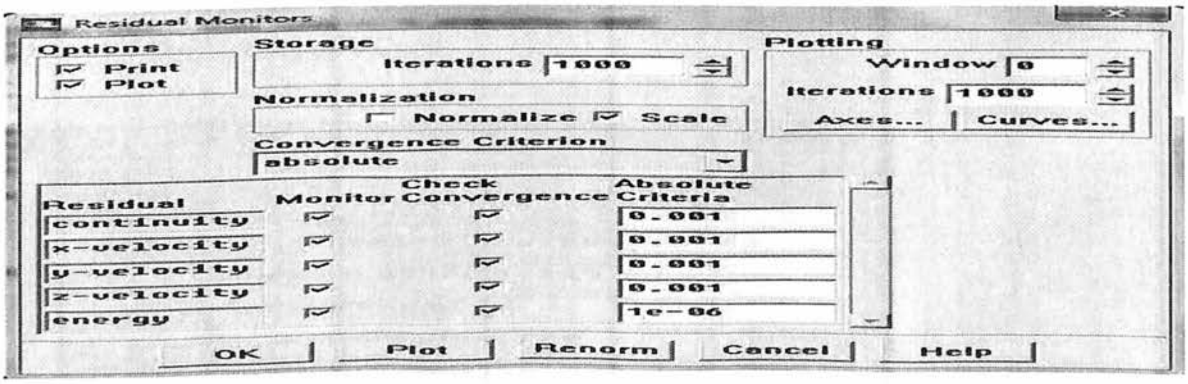
Turbulent Kinetic Energy: Second Order Upwind

Turbulent Dissipation Rate: Second Order Upwind

Fig: 6.3.3.1 giving solution parameters

6.3.4 Enable The Plotting Of Residual

Solve → Monitor → Residuals



Options: ☑ Print ☑ Plot

Storage: Iterations 1000

Normalization: ☐ Normalize ☑ Scale

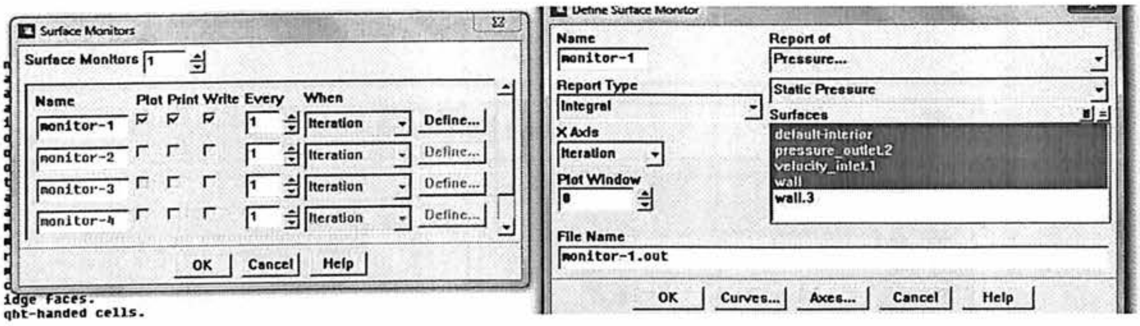
Convergence Criterion: absolute

Plotting: Window 0, Iterations 1000

Residual	Monitor	Check Convergence	Absolute Criteria
continuity	☑	☑	0.001
x-velocity	☑	☑	0.001
y-velocity	☑	☑	0.001
z-velocity	☑	☑	0.001
energy	☑	☑	1e-06

Fig:6.3.4.1 ploting of residual

6.3.5 Enable the Plotting of Other Surface Parameter (Pressure)



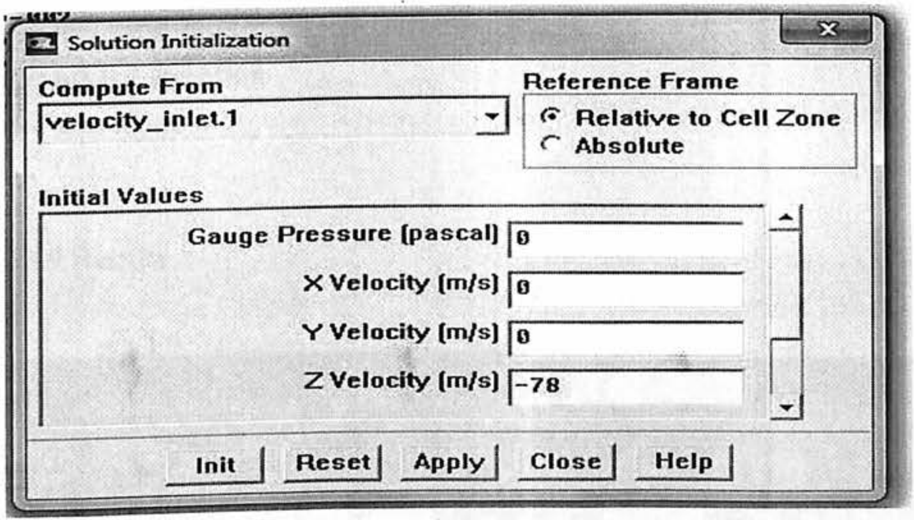
Surface Monitors: 1

Name	Plot	Print	Write	Every	When	
monitor-1	☑	☑	☑	1	Iteration	Define...
monitor-2	☐	☐	☐	1	Iteration	Define...
monitor-3	☐	☐	☐	1	Iteration	Define...
monitor-4	☐	☐	☐	1	Iteration	Define...

Name: monitor-1
Report Type: Integral
X Axis: Iteration
Plot Window: 0
Report of: Pressure...
Static Pressure
Surfaces: default-interior, pressure_outlet.2, velocity_inlet.1, wall, wall.3
File Name: monitor-1.out

Fig:6.3.5.1 plotting of surface parameters

6.3.6 Initialize The Solution

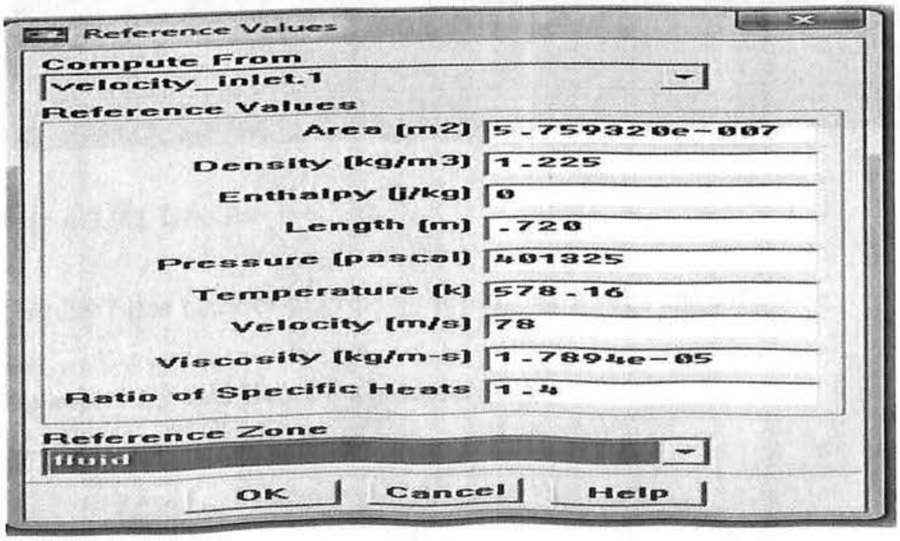


Compute From: velocity_inlet.1
Reference Frame: ● Relative to Cell Zone ○ Absolute

Initial Values
Gauge Pressure (pascal) 0
X Velocity (m/s) 0
Y Velocity (m/s) 0
Z Velocity (m/s) -78

Fig: 6.3.6.1 initialization of solution

6.3.7 Set The Reference Values



Compute From: velocity_inlet.1

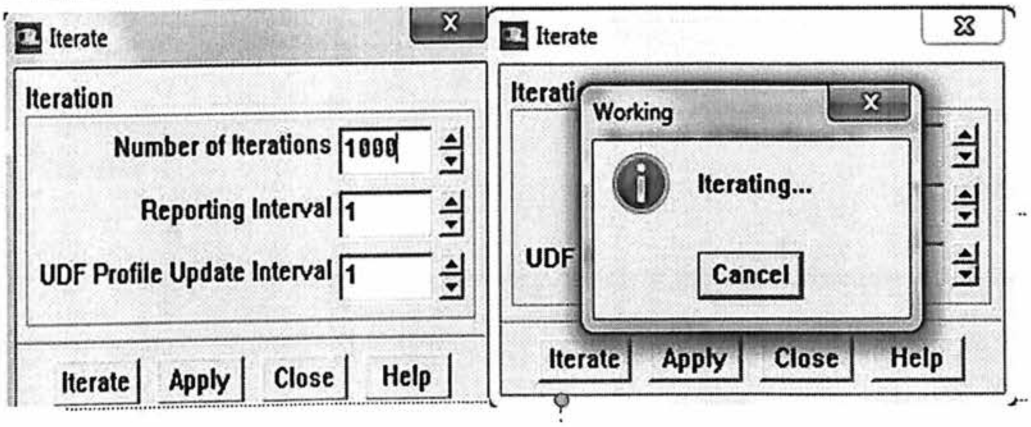
Reference Values
Area (m2) 5.759320e-007
Density (kg/m3) 1.225
Enthalpy (j/kg) 0
Length (m) .720
Pressure (pascal) 401325
Temperature (k) 578.16
Velocity (m/s) 78
Viscosity (kg/m-s) 1.7894e-05
Ratio of Specific Heats 1.4

Reference Zone: fluid

Fig:6.3.7.1 setting of reference value

6.3.8 Start the calculation by iterating to 1000 iteration.

Solve → Iterate



Iterate

Iteration

Number of Iterations 1000

Reporting Interval 1

UDF Profile Update Interval 1

Working

Iterating...

Fig:6.3.8.1 Iteration

6.3.9 Results

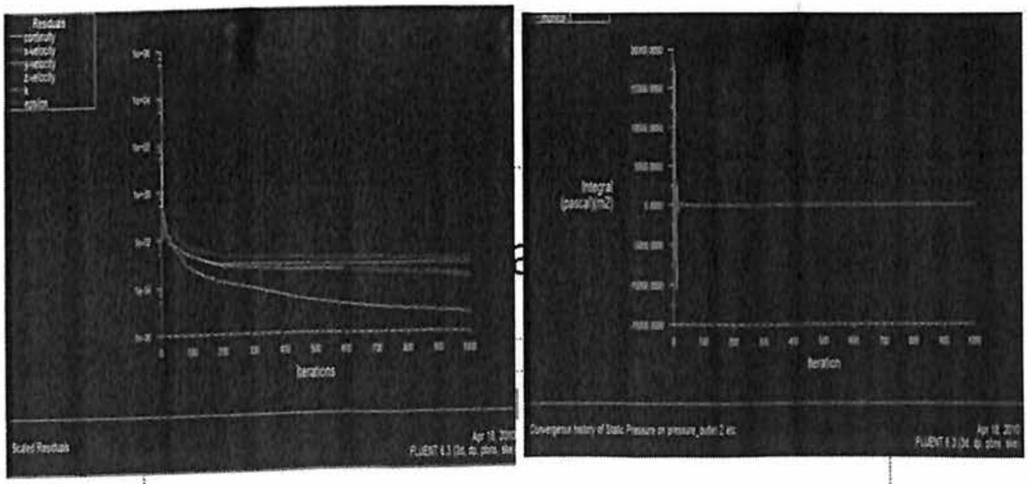


Fig: 6.3.9.1 Results

Display filled contour of Pressure for further results and inference.

Save the Case and Data files (silencer.cas)

CHAPTER 7

RESULTS AND DISCUSSION

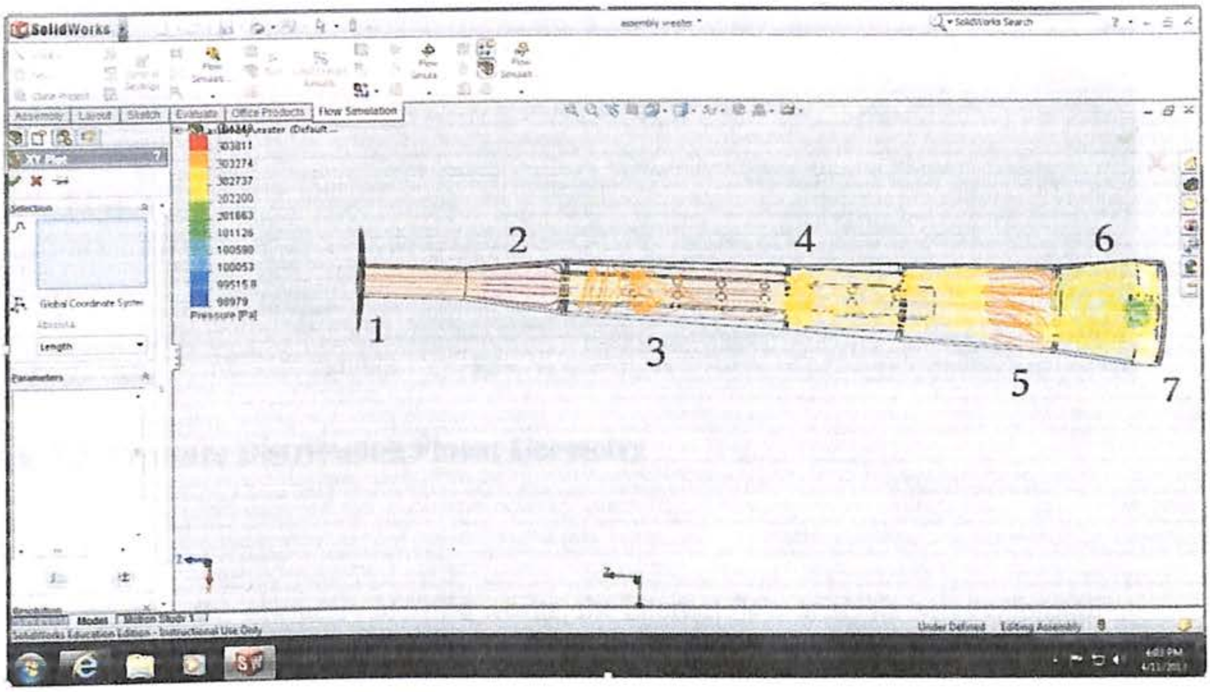


Fig.7.1 Pressure Distribution Flow-simulation

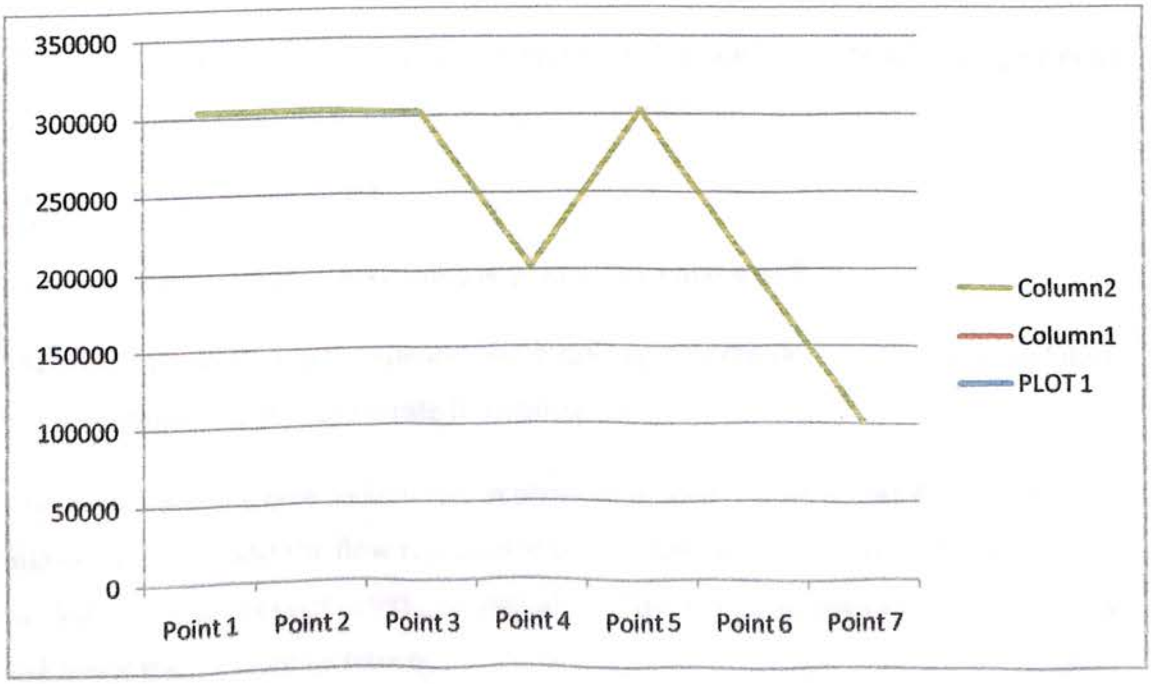


Fig.7.3 Graph Depicting Pressure Curves at Various Points on Silencer

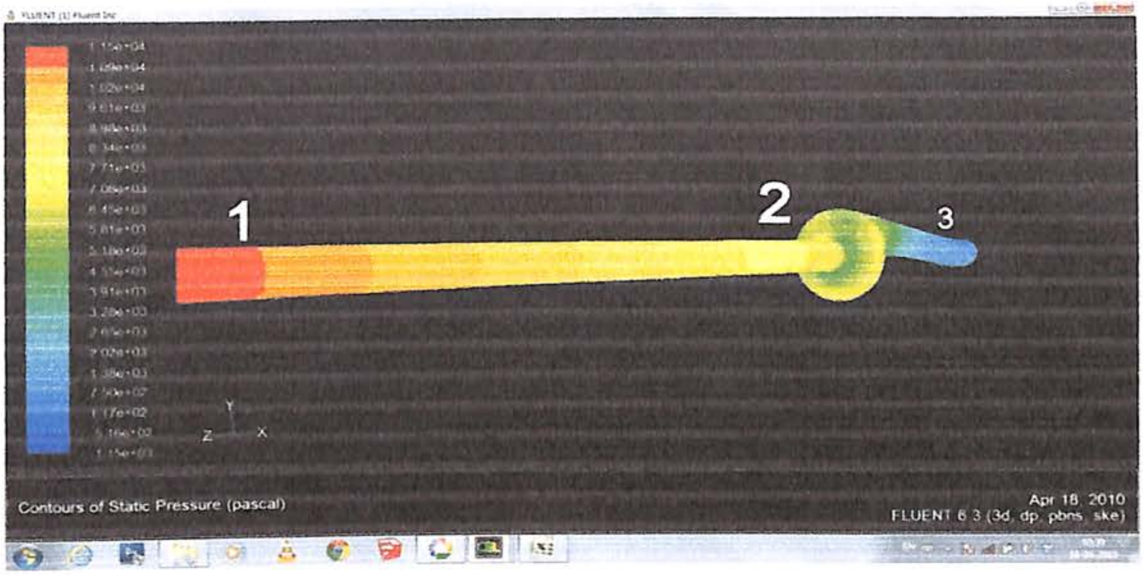


Fig.7.2 Pressure Distribution Fluent Geometry

After creating the geometry in Solid works, the internal flow analysis was performed using the Solid works as well as in Fluent after being meshed in Gambit.

The two above figures depict the Flow-analysis in Solid works and Fluent where in the Geometry different areas are numbered due to sudden change in pressure in those points.

These sudden rise or falls in Pressure at various points are highlighted here in the graph plotted between Pressure & Overall length of the silencer.

As can be seen by the Graph:

The change in pressure is noticeable at points 2 to 3 and 4 to 5.

The diverging section in the silencer geometry expands the flow immediately and thus pressure drops at a noticeable rate from point 2 to 3.

The second phase where sudden rise in pressure is seen is from points 4 to 5, where the mid-section ends and the flow reaches the offset partition. This increment is due to the sudden increment in cross-sectional area of the flow which decreases the flow velocity and hence the pressure increment.

Thus these two points majorly contribute to the back pressure and the modification in gas flow velocity.

Physical changes in geometry can alter these parameters but for sure back-pressure cannot be reduced to zero as if this happened then gas flow velocity will reduce hence silencer not able to spill out all the exhaust gases out which will alter engine life, its performance & the silencer working period as well.

CHAPTER 8

FUTURE PLANS

In future, research work can be carried out in the analysis of the effect of valve overlap duration, valve lift and advancing and retarding of the cam profile on the engine performance. Below a brief description of the future potential areas of research is given to draw the attention for researchers.

Effects of valve overlap:

Increased overlap: High torque at higher engine speeds due to pressure waves in the exhaust manifold aiding the intake of fresh charge at higher rpm.

Reduced overlap: High torque at lower engine speeds due to pressure waves in the exhaust manifold aiding the intake of fresh charge at lower rpm.

ADVANCING	RETARDING
Begins intake event sooner	Delays intake valve closing event
Open intake valve sooner	Keeps intake valve open later
Builds more low-end torque	Builds more high-RPM power
Decreases piston-intake valve clearance	Increases piston-intake valve clearance
Increase piston-exhaust valve clearance	Decrease piston-exhaust valve clearance

After studying the total analysis in two different softwares, i.e Solid works & CFD , the future scope of work still remains of modifying the Internal geometry physically using the results of these analysis.

The change in physical geometry can later be tested in the Engine Testing laboratory by using a 4-stroke engine setup and clubbing our silencer with the engine being tested.

Later the software and Physical parameters that were modified can be compared using graphs plotted by software and by the data obtained from the tests performed physically.

REFRENCES:

1. A. Graham Bell, Four-stroke performance tuning
2. http://www.motorcycle.com
3. http://www.gibsonexhaust.com/exhaustsystems/
4. http://www.supertrapp.com/
5. http://www.magnaflow.com/wideopen/wideopen.asp
6. http://www.torquecars.com/tuning/sports-exhausts.php
7. http://auto.howstuffworks.com/question636.htm
8. http://www.m4exhaust.com/
9. http://www.impulsengine.com/how/what_is_nps_about.shtml
10. http://www.wikipedia.com
11. http://auto.howstuffworks.com/
12. Philip H Smith, John C Morrison, Scientific Design of Exhaust and Intake Systems (Engineering and Performance)
13. The BG tuning manual