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Two Wheeler Exhaust Pressure Drop Optimization

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Abstract – An exhaust system is usually a piping used to guide reaction exhaust gases away from the controlled combustion inside an engine.Pressure drop across the exhaust manifold requires to be dealt with by engine and extra power of engine will be used in forcing the exhaust gases from exhaust valve to exit of the exhaust pipes. In this paper the focus is on finding remedies to reduce the pressure drop across the system of the exhaust. A geometry of the exhaust system for 100 CC four stroke SI engine for two wheeler vehicles is created using SOLIDWORKS 2015 software and flow analysis is ran on the module using ANSYS fluent 16.2. Different changes in the geometry of the exhaust assembly are done and effect of those changes on the pressure drop across the exhaust is measured. In final iteration it is successfully identified the area of improvement to reduce pressure drop and have achieved maximum pressure drop reduction up to 70 % is achieved.

Keywords –Exhaust system, S I Engine, Pressure Drop Optimization

I. INTRODUCTION

"Exhaust systems are developed to control emissions and to attenuate noise vibration and harshness to meet the regulatory requirements. The exhaust system components are manifold, close coupled and underbody catalytic converters, flexible bellow, muffler, resonator, connecting pipes, flanges, and tailpipe."

A well-designed exhaust system collects exhaust gases from engine cylinders and discharges them as quickly and silently as possible. The primary system design considerations of an exhaust system include the following:

- Minimizing resistance to gas flow (back pressure) and keeping it within the limits specified for the particular engine model and rating to provide maximum efficiency.
- Reducing exhaust noise to meet local regulations and application requirements.
- Providing adequate clearance between exhaust system components and engine components, machine structures, engine bays, enclosures and building structures to reduce the impact of high exhaust temperatures on such items.

Fig.1. Two wheeler exhaust system

II. PROBLEM STATEMENT AND OBJECTIVE

Total pressure drop across the exhaust system is mistaken for the required phenomenon for the engine to perform by many engineers. Actual case is engine do not require to be applied with any back pressure and it is just a efforts produced by the engine while burning fuel to push exhaust gases from inlet of the exhaust to the outlet of the exhaust. In this present work it is proposed that the design is to be created for the exhaust system which has lower pressure drop across the system which will save the work by the engine and indirectly will save a lot of fuel consumption by the vehicle

III. LITERATURE REVIEW

S. M. Rabia and M. Abd-El-Halim presented research on **"**Effect of Valve Timing and Exhaust Back Pressure on the Performance of Gasoline Engine". In this research the conclusion on the effect of valve timing and exhaust back pressure on the performance of gasoline engine. The engine performance was tested with combination of Late Intake Valve Closing and reduction in the Engine Back Pressure. The effect of these parameters on fuel saving, residual gas, and volumetric efficiency was studied. The main conclusions from this study, the engine volumetric efficiency increased due to reduction of exhaust back pressure and the maximum pressure inside the cylinder at different loads shows low differences at the same load.[1]

Twinkle Panchal, Dhruv Panchal, Bharat Dogra, Krupal Shah presented research on "Effect of Exhaust Back Pressure on Exhaust Emissions by Altering Exhaust Manifold Position". In this research the main conclusions are Manifold at 30^0 and 345^0 provide best HC emission results for lower loading conditions at 3000 rpm. Manifold at 20^0 and 30^0

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provide best HC emission results for higher loading conditions at 3000 rpm. HC emission increases 2.60 ppm with increase in manifold angle by 1^0. Manifold at 20^0, 30^0 and 60^0 provide best CO emission results at 3000 rpm. CO emission increases 0.01 % with increase in different load conditions. Manifold at 20^0 and 45^0 preserve their position for providing favorable results in CO_2 emission for 3000 rpm at low load condition.[2]

Sidharam Ambadas Basargi presented a paper "Design and Development of Automobile Silencer for Effective Vibration Control" This paper postulates the first stage in the design analysis of an exhaust system. With the specified properties of the material, the exhaust system is modelled by using a conventional FEM package. This presents a computational approach for the lifetime assessment of structures. One of the main features of the work is the search for simplicity and robustness in all steps of the modelling, in order to match the proposed method with industrial constraints. The proposed method is composed of mechanical finite element computation. The results are compared with the reading taken on FFT analyser, so as to distinguish working frequency from natural frequency and avoid resonating condition. The validation of the silencer is done by physical prototype development. While the Silencer is made available in the physical form, the trials and testing would address the phase of validation. The correlation between the experimental and theoretical results will be analysed and recommendations can be made for future scope of work. [3]

Tushar J. Awari & Dr. S. Y. Gajjal presented a preview study named "Review Study of 2-Wheeler Silencer's Natural Frequency by FFT Analyser" stating ideally natural frequency of different parts should never match with engine Excitation frequencies but practically there are moments when resonance could not be avoided. Due to this they have to find not only the resonance frequency but also the vibration amplitude at the same time. To reduce these vibrations we design and modify two wheeler silencer. This can be achieved by two methods namely experimental and FEM analysis. The experimental analysis is carried out with the help of FFT analyser to evaluate the natural frequency and vibration amplitude and to distinguish it from the working frequency to avoid resonating condition. [4]

I. P. Kandylas, A. M. Stamatelos conducted a study on "Engine exhaust system design based on heat transfer computation". In this study they said that, when one faces the problem of designing an engineering approach to the design of a spark-ignition-engine vehicle exhaust system that aims at the attainment of specific exhaustemissions targets, high accuracy in heat transfer computations is imperative. Furthermore, the compromises related to converter efficiency and durability targets (catalyst aging) involve extensive heat transfer computations. Thus, an in depth study of exhaust piping heat transfer and the development of related exhaust system heat transfer codes are essential in supporting Computer Aided Engineering methodologies for exhaust after-treatment systems. They concluded that the following exhaust system design parameters may be optimized: exhaust manifold material, thickness and insulation exhaust manifold and downpipe design (geometry),position of catalytic converter in gasoline engines, position of particulate trap in Diesel engines, effect of distributed or concentrated metallic masses (e.g., flanges) on transient response, different types of pipe insulation.[5]

IV. THEORETICAL ANALYSIS

1. Mass of air that the engine breaths in + mass of fuel = mass of exhaust gases
2. Volume of air the engine takes in = Displacement of the engine x rpm/2
3. To make calculations easy, assume perfect combustion, there aren't any by products and unburned fuel etc.

Intake system needs to flow 1.5 cfm per engine horsepower and your exhaust system needs to flow 2.2 cfm per engines horse power.

- Hero Splendor

Engine – Air cooled, 4 Stroke, Single Cylinder
BHP – 4 @ 8000 rpm
Torque @ 5000 rpm
Displacement – 97 cc
Top Speed – 37 mph
Several Assumptions made to design calculations:

1. Combustion will be stoichiometric and complete
2. Compression ratio
3. The engine is throttled (no variable valve timing)
4. Normal aspirate engine (no turbocharger)
5. Volumetric efficiency (the amount of air that makes it into cylinder during induction strokes) is 1.00

Engine is of displacement 97 cc = 0.097 litre
Intake volume of air = Engine displacement x rpm/2
Intake volume of air in engine = 388 litre/min

- That will approximately be the intake volume flow for an engine with the throttle wide open.
- Neglecting the addition of the fuel mass, the mass of the exhaust gas will be the same as the intake gas.

From the ideal gas laws,

$$PV = mRT$$

It is known that the increase in volume of the exhaust gas will be proportional to the increase in absolute temperature.

Intake temperature = 26 ^{0}C (Average ambient air) = 80 ^{0}F

And exhaust gas temperature of 4 stroke petrol engine = 1400 ^{0}C (max) = 2552 ^{0}F

Hence converting temperature to absolute Rankine temperature,

Tin = 80 ^{0}F ≈ 540 ^{0}R

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Tout = 2552 ^{0}F ≈ 3012 ^{0}F

From Gas Laws,

$$P1V1 = mRT1$$
$$P2V2 = mRT2$$
$$\frac{P1V1}{T1} = mR$$
$$\frac{P2V2}{T2} = mR$$
$$V2 = V1 \times \frac{T2}{T1}$$
$$V2 = 388\ lit/\min \times \frac{3012}{540}$$
$$V2 = 2164.18\ lit/min$$

Converting it into CFM (cubic foot per minute)

1 CFM = 28.3 lit

$$V2 = \frac{2164.18}{28.3}$$
$$V2 = 76.47\ CFM$$
$$V1 = \frac{388}{28.3} = 13.71\ CFM$$

Now, contribution of combustion products, assuming stoichiometric combustion, there will be one pound of fuel burned for each 14.55 lbs of air

As in air, there is 21%, oxygen available 60 taking 21% of 14.55, it is 3.05 lbs of oxygen available to burn each pound of fuel.

A resonance chemical approximation for gasoline is octane, which has a chemical of C8H18

The molecular weight is (12 X 8 + 18 X 1) = 114

The combustion formula is,

C8H18 + 12.5 O2 = 8CO2 +9H2O

For each 114 grams of C8H18, there will be 12.5 moles of oxygen consumed, producing 8 moles of CO2 & 9 moles of H2O

For gas volume purposes, since equal volume produced by equal mole of gas.

Volume of exhaust gas replacing,

Oxygen will be equal to 17/12.5 = 1.35

The volume percentage of oxygen in air is about 21%

This volume will be removed and replaced by exhaust gas with a volume of (21 X 1.36) = 28.56%

The resulting post combustion volume is (79% + 28.56% = 107.56%) of the pre-combustion volume assuming no temperature increase.

So combining the increase in volume flow from combustion reactions and thermal expansion

Total Exhaust Volume = 76.47 CFM X 1.0756 = 82.25 CFM

Maximum allowable back pressure for gasoline engine, Gasoline (all types) is 4” of Hg (54.38” of H2O)

Now the Data available,

1. Engine exhaust flow (CFM) = 82.25 CFM
2. Engine exhaust temperature (max) = 2552 ^{0}F
3. Max back pressure (inch of H2O) = 58.38

Now calculating exhaust gas velocity,

$$V = 4005\sqrt{\frac{\Delta P}{C\left(\frac{530}{T+360}\right)}}$$

Where,

V = gas velocity,

ΔP = back pressure in inches of water

C = silencer pressure drop co-efficient

T = exhaust gas temp, ^{0}F

$$V = 4005\sqrt{\frac{58.38}{1\left(\frac{530}{2552+360}\right)}}$$
$$V = 71729.68\ Ft/min$$

Now calculating required flow area,

$$Flow\ Area\ Required\ (Ft^2) = \frac{CFM}{V}$$
$$= \frac{82.25}{71729.68}$$
$$Af = 0.001147\ Ft^2$$

From standard table,

For Af = 0.0055 Ft^2 and less.

Diameter of exhaust is 1 inch.

Taking D = 1.2 for safety.

$$\therefore D_{in} = 30.48\ mm$$

- inner diameter of exhaust pipe.

Therefore from selected inner diameter,

$$V_{actual} = \frac{CFM}{Silencer\ Flow\ Area}$$
$$= \frac{82.85}{0.0055}$$
$$V_{actual} = 14954.54\ Ft^2/sec$$
$$\Delta P = C\left(\frac{Vact}{4005}\right)^2\left(\frac{530}{T+460}\right)$$
$$= 1\left(\frac{14954.54}{4005}\right)^2\left(\frac{530}{2552+460}\right)$$
$$\Delta P = 0.17596\ inch\ of\ H2O$$
$$-\ less\ than\ design\ value\ hence\ safe$$

Exhaust Pipe Diameter

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$$D = 30.48\ mm\ or\ 1.2\ inch$$

Generally an exhaust muffler is requirement to satisfy some basic requirements such as adequate insertion loss, low back press muffler sizing which could affect the cost and durability to withstand rough use and extremely high temperature.

In order to select a suitable muffler type, some basic information is necessary regarding muffler as per the ASHRAE Technical Committee.

Muffler Grades,

1. Industrial / Commercial:

Insertion Loss (IL) = 15 to 25 dBA
Body dia / Pipe dia = 2 to 2.5
Length / Pipe = 5 to 6.5

2. Residential Grade:

IL = 20 to 30 dBA
Body dia / Pipe = 2 to 2.5
Length / Pipe = 6 to 10

3. Critical Grade:

IL = 25 to 35 dBA
Body / Pipe = 3
Length / Pipe = 8 to 10

4. Super Critical Grade:

IL = 35 to 45 dBA
Body / Pipe = 3
Length / Pipe = 10 to 16

IL = Insertion Loss i.e. the level of sound reduction after attaching the muffler.

The proposed muffler is assumed to the Super Critical Grade & the mixed type of absorptive and reactive muffler with '3' chambers and tunable resonator. The chamber is performed and wrapped with the glass wool for absorption.

Design data Required:

Max engine speed – 8000 rpm
No. of cylinder – 1
Inlet pipe diameter – 1.2 inch (30.48 mm)
Maximum temperature at inlet of muffler

Considering frequency ranges as below for calculation purpose,

f1 = 125 Hz
f2 = 253 Hz
f3 = 381 Hz
f4 = 509 Hz

Considering lower three frequencies for calculation from resonance point of view.

Chamber Design

A. Chamber length and diameter according to ASHRAE,

For critical grade,

IL = 25 to 35 dBA
Body diameter = 3 x Pipe Diameter
= 3 x 1.2
= 3.6 inch or 91.44 mm
Length of chamber = 10 x 1.2 to 16 x 1.2
= 12 inch to 19.2 inch
= 304.8 mm to 487.68 mm

B. Resonance Method:

Maximum attenuation occurs when,

$$L = \frac{n\lambda}{4}$$

Where, λ is wavelength of sound (m)
And, n = 1, 3, 5 (odd integers)
For economical considerations, we take
n = 1
Reference value of speed of sound
V_s = 330 m/s

The length is calculated for frequencies 125 Hz, 253 Hz, 381 Hz, 509 Hz, are experimental exhaust noise frequencies. So the length of chambers to satisfy 'I'

$$\lambda_1 = \frac{Vs}{f_1} = \frac{330}{125} = 2.64\frac{m}{s}$$
$$\lambda_2 = \frac{Vs}{f_2} = \frac{330}{253} = 1.304\frac{m}{s}$$
$$\lambda_3 = \frac{Vs}{f_3} = \frac{330}{381} = 0.866\frac{m}{s}$$
$$\lambda_4 = \frac{Vs}{f_4} = \frac{330}{509} = 0.65\frac{m}{s}$$

Therefore, length of chamber

$$l_1 = \frac{n\lambda_1}{4} = \frac{2.64}{4} = 0.66\ m$$
$$l_2 = \frac{n\lambda_2}{4} = \frac{1.304}{4} = 0.326\ m$$
$$l_3 = \frac{n\lambda_3}{4} = \frac{0.866}{4} = 0.2165\ m$$
$$l_4 = \frac{n\lambda_4}{4} = \frac{0.65}{4} = 0.16\ m$$

We choose the length of chamber,
1st Chamber = 0.326 m
2nd and 3rd Chamber = 0.2165 m
Therefore total length of chamber,

$$L = l_1 + 2l_3$$
$$= 0.326 + 0.2165 + 0.2165$$
$$= 0.759\ m$$

Therefore Diameter of chamber,
D_1 = 91.44 mm = 92 mm
Total length of Chamber,
L = 759 mm
Baffle Pipes Design:

Diameter of pipes inside the baffles is so that the cross section area doesn't reduce. So,

Area of inlet pipe = Total Area of baffle pipe

$$\frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times {d_1}^2$$
$$d_1 = 21.56\ mm$$

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V. FINITE ELEMENT ANALYSIS

The Finite Element method is a numerical method, which can be used for accurate solution of complex engineering problems. The method was first developed in 1956 for the analysis of aircraft structural problems. Thereafter, within a decade, the potentialities of the method for the solution of different types of applied science and engineering problems were recognized. Over the years, the finite element technique has been so well established that today it is considered to be one of the best methods for solving a wide variety of practical problems efficiently.

The solution of a general continuum problem by the finite element method always follows an orderly step-by-step process. With reference to static structural problems, the SIX-STEP PROCEDURE

- Discretization of the structure.
- Selection of the interpolation polynomials.
- Formulation of the element characteristic matrices.
- Assembly of element matrices and derivation of system of equations.
- Solution of the finite element (system) equations.
- Computation of the element resultants.

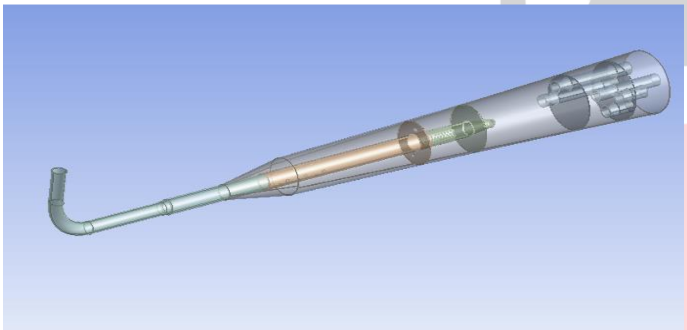

Fig.2. Two wheeler exhausts system

CAD model was created by measuring the actual model of exhaust system used for 100 cc SI engine of splendour. Model is shown in the figure above. SOLIDWORKS is used for modelling the exhaust. From design calculations **velocity of 50 m/s is deduced and applied as a boundary condition to the inlet** of the exhaust system. Temperature of the inlet gases to the exhaust system are assumed as **400 degree Celsius** which is at the higher end of the 100 cc engine exhaust for SI engines. Outlet is applied with atmospheric pressure outlet boundary condition. **All the part boundaries which are exposed to atmosphere are provided with heat convection rate of 200 W/m^2 K** according to reference table of the convection heat transfer coefficients. 1 lakh 86 thousand nodes and 9 lakh elements are used to mesh the model of exhaust

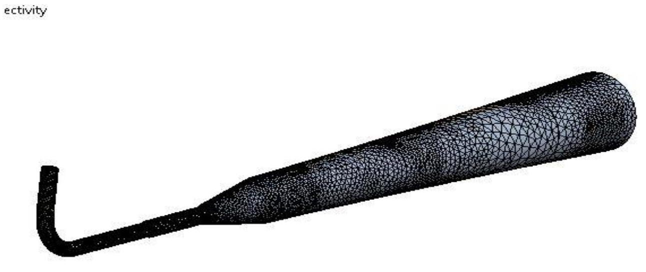

Fig.3. Meshed model for exhaust system

Basic flow analysis is to be performed on the exhaust system model. With inlet boundary condition is provided with inlet velocity and temperature, all the wall of the body exposed to the cooling air stream are provided with 200 convection heat transfer coefficient.

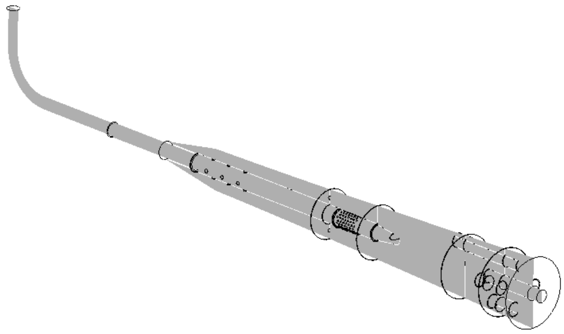

Fig.4. Sectional view of exhaust system

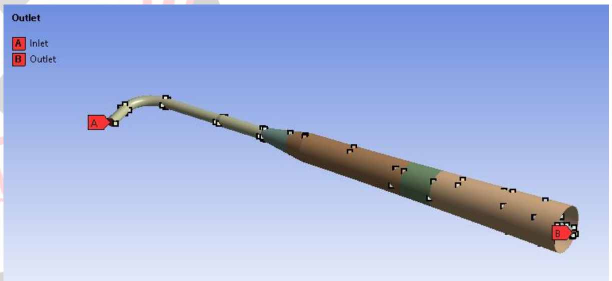

Fig.5. Boundary condition positions

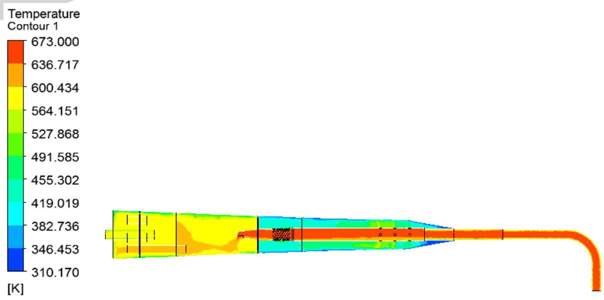

Fig.6. Temperature Plot of Exhaust system @ section

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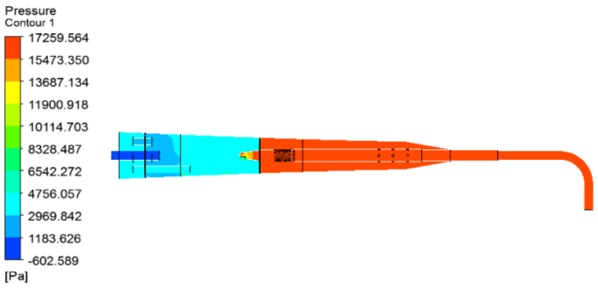

Fig.7. Pressure Plot @ section

Pressure drop across the exhaust system is that means there is almost 18.8 KPa pressure drop across the system. Mostly maximum permissible pressure drop allowed in the system

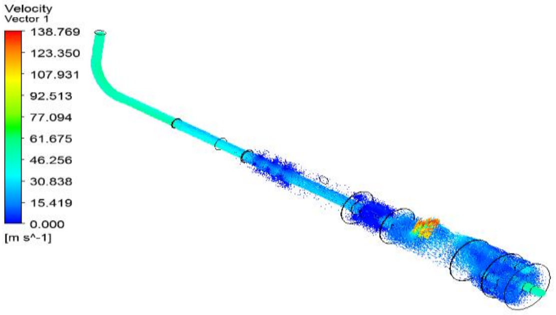

Fig.8. Velocity vector plot for Baseline flow analysis

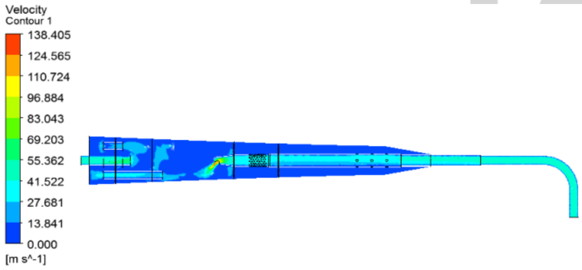

Fig.9. Velocity plot at the cross section

Above plots shows the information about the velocity, pressure and temperatures in the flow zone. Unit of the velocity used is m/s, temperature unit is Kelvin and pressure unit is Pascal.

Colour contour on the right hand side of all the plots show us the range of variable the colour represents, and colours in the plot gives us an idea about the values of the variables in the zone.

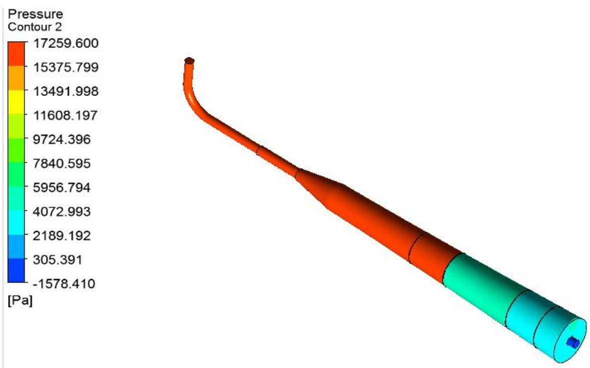

Fig.10. Pressure plot across system Baseline

Pressure drop across the system can, As there is 0 atm pressure boundary condition at outlet.

Total pressure drop = 17.3 KPa

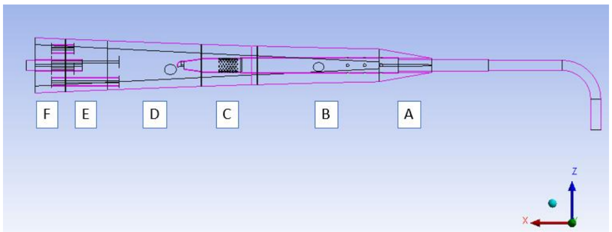

Fig.11. Wireframe diagram for iteration 1 side view

From inlet side, if the chambers are divided in 6 parts calling it A to F. Consideration for iteration 1 length of the chamber C is increased by 10 mm and analysis is done. Results are compared with the baseline.

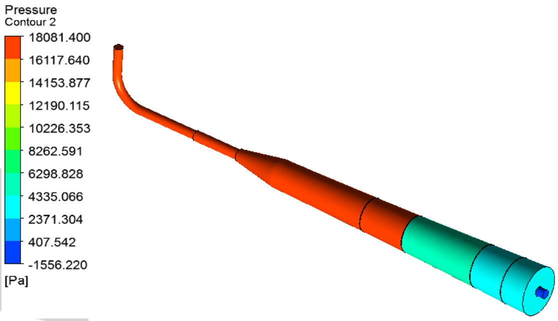

Fig.12. Pressure plot at Iteration 1

Pressure drop = 18.1 KPa

Pressure drop increased by 0.8 KPa.

So increasing the length of the air travel will increase the pressure drop as the resistance to the flow increases.

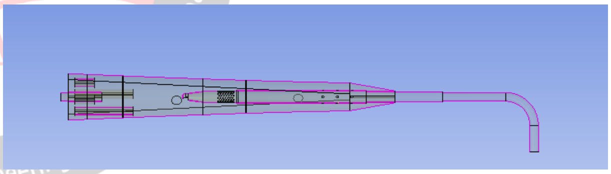

Fig.13. Wireframe diagram for iteration 2 side view

In Iteration 2 reduction of the lengths of the flow travel is tried. In iteration 2 the length of chamber F is reduced by 15 mm. And pressure plot is given below

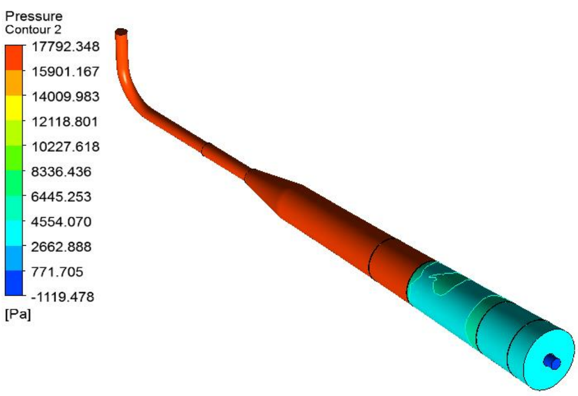

Fig.14. Pressure plot at Iteration 2

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Pressure drop = 17.792 =17.8 KPa, as there is 0 gauge pressure or atmospheric pressure applied on the outlet. 17.8 KPa. Pressure drop is increased by 0.5 KPa in this iteration when compared with baseline.

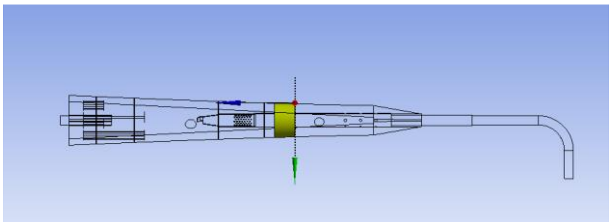

Fig.15. Iteration 3 design model

In Iteration 3 Chamber C length is reduced by 40 mm, and results for the same are observed.

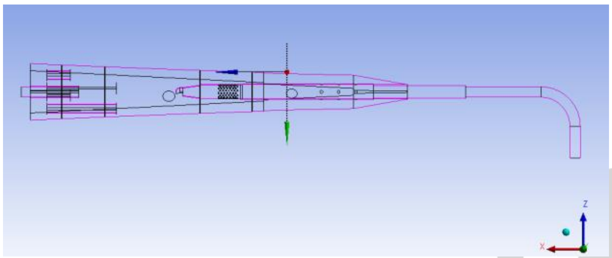

Fig.16. Wireframe diagram for iteration 3 side view

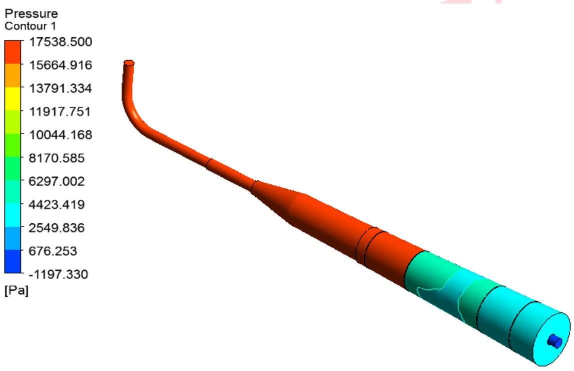

Fig.17. Pressure plot Iteration 3 model

Pressure Drop Increased by 0.2 KPa due to step in the geometry.

Step in the geometry is removed, and inlet pipe before entering the exhaust is reduced by 50 mm

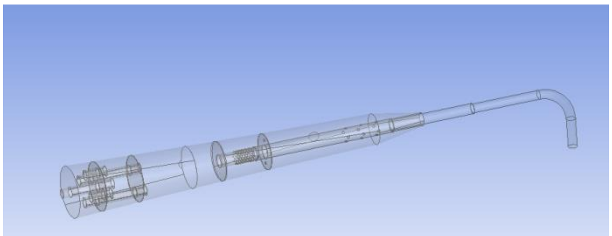

Fig.18. Iteration 4 Design model

In Iteration 4 design change pipe extruded and obstruction to flow in chamber D was removed and flow is directed directly in to the chamber D. This will reduce the resistance to the flow drastically and less effort will be required to force the gases out of the exhaust system. Fig.19 the wireframe diagram of the model.

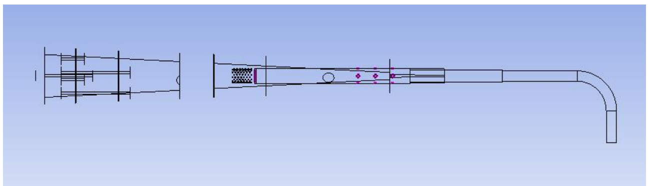

Fig.19. Iteration 4 design wireframe model

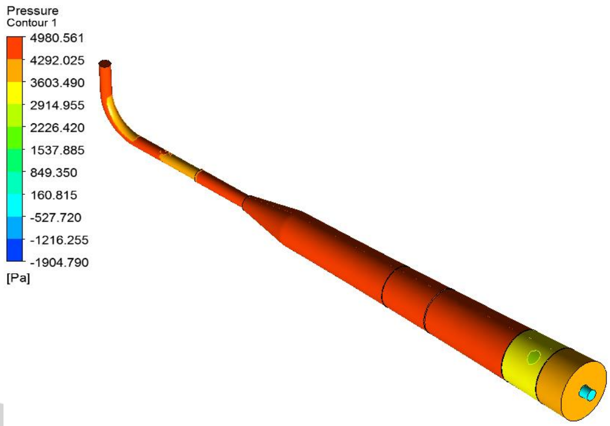

Fig.19. Iteration 4 pressure drop contour

Above figure shows pressure plot for Iteration 4 geometry. Only removing the obstruction and small geometrical changes like minimizing the length of inlet of exhaust pipe which reduce the pressure drop across the system drastically. Removing the portion in front in the chamber D results in pressure drop across the system as 4.9 KPa that is 12.4 KPa pressure drop reduction from the original system. Now according to this design, changes will be made in the fabrication model and system will be analyzed for the exhaust gas constituents as well as pressure drop in practical applications. Final iteration other result plots are shown in the images below.

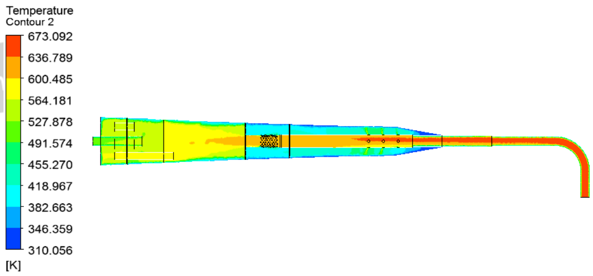

Fig.20. Temperature Plot of Iteration 4 design

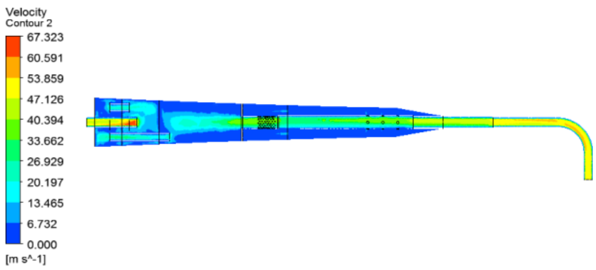

Fig.21.Iteration 4 velocity plot @ cross section

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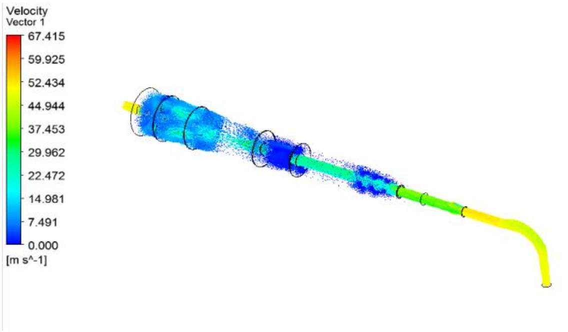

Fig.22.Velocity vector plot at Iteration 4 Design

VI. EXPERIMENTAL TESTING

Experimental testing of the exhaust system will be carried out by manufacturing the proposed design changes in the current exhaust system design for the Hero splendor. New exhaust system will be fit to two wheeler. Engine will be run for different loading conditions and pressure drop across the exhaust system will be measured. Comparison of the results from modified silencer with the results from FEA analysis. After manufacturing the modified silencer with the results of FEA checking natural frequency by using FFT analyzer.

VII. RESULTS AND DISCUSSION

Table.1. Result summary table CFD

Design	Pressure Drop (Kpa)	Pressure Drop Reduced
Baseline	17.3	
Iteration 1	18.1	-0.8
Iteration 2	17.8	-0.5
Iteration 3	17.5	-0.2
Iteration 4	5	12.3

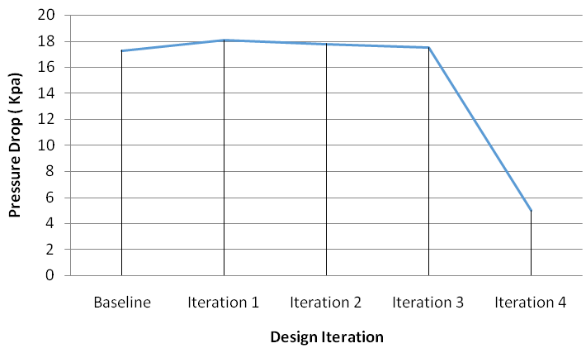

Fig.23.Graph of Pressure drop across the silencer vs Design iteration

Table 1 and graph plotted above in fig 23 shows that results for iterations which says baseline design has pressure drop across the system 17.3 Kpa . Iteration 1st- Length of chamber C increased by 10 mm gives pressure drop of 18.1 KPa . Iteration 2nd- Length of chamber C reduced by 15 mm gives pressure drop of 17.8 KPa . Iteration 3rd- Length of chamber C reduced by 40 mm pressure drop to shift to 17.5 KPa. Iteration 4th length of exhaust pipe at starting reduced by 50 mm and removing the obstruction to flow in chamber D pressure drop is changed to 5 KPa across the system.

VIII. CONCLUSION

In this present work flow analysis on the two wheeler exhaust system is performed. It is found out that approximately 70 % pressure drop can be reduced if length of exhaust pipe at starting reduced by 50 mm and restriction to the flow in D chamber is removed. It can be seen that first 3 iterations show slight increase in the pressure drop across the exhaust system which will result in to increased work for the two wheeler engine to overcome the resistance by exhaust system. In iteration 4 design pressure drop across the exhaust system drop down by 12.3 KPa and it is observed to be 5 KPa which reduces the required work by the engine to force exhaust gases out of the system drastically. This will result in increased engine efficiency and low fuel consumption for similar work production by engine.

IX. FUTURE SCOPE

The scope of study can be expanded by also considering effect on Noise, Vibration and Harshness by the changes. Pulsating pressures on the walls of the exhaust system due to flow and perform static structural analysis to find out structural fatigue caused by the flow in the joints of exhaust system. Also analyze the vibrations by provision of vibration absorber or vibration isolator. Effect of different welding processes on the silencer can be examined.

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