THE DESIGN AND VALIDATION OF ENGINE INTAKE MANIFOLD USING PHYSICAL EXPERIMENT AND CFD

GURU DEEP SINGH, KESHAV KAUSHIK & PRADEEP KUMAR JAIN

Department of Mechanical Engineering, Delhi Technological University, Main Bawana Road, New Delhi, India,

ABSTRACT

Race-car engineers aim to design an intake manifold which can maintain both low-end and top-end power without compromising the responsiveness of the engine throughout the power band. A major obstacle in achieving this goal is the rule requirement by FSAE for the mandatory presence of air intake restrictor which limits top-end power.

In this paper, the selection criteria for design parameters such as runner length, plenum volume and intake geometry have been discussed. The effect of runner length and plenum volume on throttle response and manifold pressure has been studied through a physical exp. on a prototype variable geometry intake manifold. CFD simulations have been performed on ANSYS CFX to optimize the geometry for venturi and plenum. The geometry for which there was minimum pressure loss and maximum mass flow rate was chosen in the final design. The adopted approach was validated by conducting the same exp. on the designed intake manifold.

KEYWORDS: *Air Intake Manifold, CFD, FSAE, Engine, Converging- Diverging Nozzle & Variable Length Intake Manifold*

Received: Jun 13, 2019; **Accepted:** Jul 04, 2019; **Published:** Jul 22, 2019; **Paper Id.:** IJAEIRDDEC20191

1. INTRODUCTION

FSAE is the largest engineering design competition in the world which gives students an opportunity to design and manufacture a race pertaining to a series of rules whose purpose is both to ensure on-site event operations and promote clever problem solving. The rules dictate that engines used in Formula SAE are limited in capacity to no greater than 710cc and the entire intake flow must pass through a single circular restrictor (20mm in diameter) located between the engine and throttle. This limits the maximum power obtained from an engine because it decreases the amount of air inducted in every cycle. However, the influence of a restrictor can be reduced by a proper intake manifold design.

Expansion and compression waves are formed in IC engine systems due to unsteady nature of flow through the intake and exhaust systems. These waves are of finite amplitude in nature and are energy-charged with extremely high-pressure ratio[1,2]. Experiment conducted by Margary[3] proved that tuned intake duct behaved as quarter-wave length pipe with 9% increase in volumetric efficiency. However, these tuned intake runners achieved best performance for a limited range of engine speeds. Cauchi J, [4] investigated optimum converging and diverging angles for minimum pressure drop across the restrictor using 1-D engine simulation software and experimental techniques. Various empirical equations for determining the appropriate dimensions for runner length, runner diameter, plenum volume, etc. have been provided by David Vizard [5]. Gordon P. Blair, [6] conducted 3-D simulations to study the effect of bell mouth shape on engine mass flow rate and found that

elliptical profile resulted in minimum losses. CFD analysis of non-symmetrical intake manifold was performed by J. Ling [7] to illustrate the role of plenum in acting as a buffer volume and moderate the flow fluctuations. Adam Vaughan [8] succeeded in developing a continuously variable intake manifold with help of cable linkage mechanism and servo motor to produce an engine which was tuned over a greater range of engine speeds.

It is important to understand the limitations and assumptions associated with the equations available for intake manifold design. Expertise in fluid mechanics, combustion theory and finite element techniques is essential for using engine simulation and CFD software. These are highly capable software used by the industry, but inadequate knowledge can result in a bad intake manifold design causing loss of engine power, high fuel consumption and sluggish engine response.

Therefore, a systematic approach was adopted to design the intake manifold for KTM 500 EXC. Physical experiments were conducted to study the effect of plenum volume and runner length on peak engine power and throttle response. This data was used to validate the engine simulation model developed in Ricardo WAVE. Steady state simulation for determining the optimal venturi converging-diverging angles and transient simulations based on pressure vs. crank angle data from Ricardo WAVE were carried out to optimize intake geometry using Ansys CFX.

2. METHODS

2.1 Variable Volume and Runner Length Intake Design

2.1.1 System Description

For obtaining the response of the engine at various configurations i. e. different plenum volume and different runner length, we had two options – either constructing multiple intakes with various configurations or making a Variable Volume as well as Variable runner length. In the present study, the second option was employed. Figure 1 shows the various parts of the built intake. This design allows us to vary the volume as well as runner length with the help of two actuation cables. Based on theoretical calculations and competition rules three different settings were selected for the experiment:

Table 1: Variable Length Intake Manifold Geometric Specifications

Setting	Plenum Volume	Runner Length
Minimum	1.8 l	32cm
Intermediate	2.6 l	28 cm
Maximum	3.4 l	24 cm

Figure 1 shows the design of variable length intake manifold used for physical experiment.

Figure 1: Intake Manifold: Maximum Volume (Left) & Minimum Volume (Right)

Figure 2: Exploded View

The designed intake has eight parts namely Throttle body, Venturi, Upper Plenum Cap, Plenum Drum, Movable Ram, Internal Runner, Lower Plenum Cap and External Runner. The throttle body is attached to the venturi and venturi to upper plenum cap with the help of nut and bolt. The upper plenum cap is threaded to the plenum drum which is in turn threaded to the lower plenum cap. The external runner containing injector and a Manifold Pressure Sensor is threaded to the lower plenum cap. The internal mechanism is constituted of a movable ram whose outer diameter is just smaller than the internal diameter of the plenum drum and is welded to an internal runner whose outer diameter is just smaller than the internal diameter of the external ram, an actuation cable is connected to each of the sides of the movable ram which helps to actuate the internal ram.

2.1.2 Actuation

Two actuation cables are attached to two sides of the movable ram so that when the actuation cable of the upper plenum cap is pulled the volume of plenum decreases and the runner length increases simultaneously. The exact opposite occurs when the actuation cable of the lower plenum cap is pulled.

2.1.3 Experimental Setup

Two 1-bar DENSO pressure sensors were used to measure the pressure inside the plenum which was placed on the Upper and Lower plenum cap as shown in Figure 3 and the reading of the throttle was obtained from the onboard throttle position sensor. The average of the two pressure sensors readings was used for further evaluation. The analog signals were obtained on Arduino Uno and the voltage readings were calibrated to provide lucid data.

Figure 3: Intake Manifold with Sensors in Position

3.2 Steady State Simulations for Venturi Design

A Converging – Diverging nozzle was designed with throat diameter of 20mm. Steady state simulations were conducted on ANSYS CFX (with atmospheric pressure inlet and mass flow outlet) to determine an optimal restrictor geometry which would result in minimum pressure loss.

Figure 4: Venturi 16-6 Pressure Profile

3.3 Transient Simulations for Geometry Design

The runner length (30 cm), runner diameter (42mm) and Venturi geometry (16-6) were fixed and changes were made to the plenum geometry. The design considerations were to allow for gradual expansion from venturi to plenum for minimum losses due to sudden expansion and abrupt convergence at the runner end to cause turbulence in the runner for better atomization of the fuel.

Figure 5: RICARDO Pressure vs Crank Angle

The pressure values at intake runner outlet for a cycle at 6250 RPM were obtained from the RICARDO WAVE software. This data was distributed into 60-time steps.

Time dependence boundary conditions were used to model the 3- dimensional unsteady flow inside the manifold.

Although, the k- ω model is more accurate for internal flows involving strong curvature, the standard k- ϵ being the simplest model, was used due to its good convergence rate, low memory requirements and reasonable accuracy. It was observed that the simulation achieved mesh independence at 27,80,052 cells. The maximum mass flow rate was obtained at outlet at about 56th time step for each iteration.

3. RESULTS

3.1 Variable Length Intake Manifold

As the volume increased, the average pressure in the plenum increased and as the pressure inside the plenum is directly proportional to the volumetric efficiency thus the volumetric efficiency increased but the percentage increase was very less at low RPM.

Table 2: Mean Pressure Readings at Lower Rpm

Plenum State	RPM	Throttle Percentage	Mean Pressure (Psi)
Minimum	2220	20	10.23
Maximum	2280	21	10.34

It was observed that there was a significant increase in the average pressure and volumetric efficiency at 6250 rpm. However, the throttle percentage required to maintain 6250 rpm at Max volume setting was higher than that required in case of Min volume setting which shows that there was a decrease in throttle response as we increased plenum volume.

Table 3: Mean Pressure Readings at Higher RPM

Plenum State	Rpm	Throttle Percentage	Mean Pressure (Psi)
Minimum	6250	25	7.655
Intermediate	6250	28	9.02
Maximum	6250	30	10.17

In a Formula Student competition both throttle response as well as power at wheels are required. After discussion with the driver of the car on which the Variable Length Intake Manifold (VLIM) was used, it was noted that the throttle response was way too much to handle at minimum plenum volume and just enough at maximum plenum volume. It was also noted that at a maximum plenum volume there was wheel spin, which leads to increase in lap time of the event. Thus, there was a need to reduce wheel spin and improve throttle response, which could be achieved by reducing the plenum volume. At intermediate volume, not much wheel spin was noted, and the throttle response improved as well.

3.2 Venturi Design

It was found that the pressure loss across the 16 – 6 geometry was minimum (6942 Pa). The nozzle was machined on CNC Lathe and the material used was Aluminium 6 Series. The choked mass flow rate through a 20mm restrictor was calculated using Eq. (1) (Appendix).

Table 4: Venturi Design Parameters

Converging Angle – Diverging Angle	Input Conditions		Result
	At Inlet [Pa]	At Outlet [Kg/s]	At Outlet [Pa]
12-6	101325	0.075	93425
14-6	101325	0.075	93996
16-6	101325	0.075	94383
18-6	101325	0.075	93991

3.3 Geometry Design

Figure 6: Plenum Iteration 1 (Plenum Volume: 2.27 L)

Figure 7: Plenum Iteration 2 (Plenum Volume: 2.55l)

Figure 8: Plenum Iteration 3 (Plenum Volume: 2.63 L)

Table 5: Plenum Design Parameters

Geometry (Plenum)	Input Conditions		Result
	At Inlet [Pa]	At Outlet [Pa]	At Outlet [Kg/s]
ITERATION 1	101325	F(Pa)	0.0524
ITERATION 2	101325	F(Pa)	0.0608
ITERATION 3	101325	F(Pa)	0.0631

In Table 5, as per the transient CFD results as the geometry tended towards more streamlined shape the mass flow at the outlet of runner increased and the maximum obtained mass flow rate was for the 3rd iteration with the mass flow rate of 0.0631 Kg/s. Thus, the volumetric efficiency increased. Hence, more fuel could be provided to engine to produce more power.

4. CONCLUSIONS

The same test was performed on 3D printed intake shown in Figure. 9 to record a plenum pressure of 9.81 *psi* at 26.5 throttle percentage at 6250 *rpm* (No Load). This performed better than the Variable Length Intake Manifold at intermediate Volume.

Table 6: 3D Printed Intake Pressure Reading

Plenum State	RPM	Throttle Percentage	Mean Pressure (Psi)
3 D Printed	6250	27	9.81

Thus, following the systematic approach from physical testing data followed by simulations to improve the design resulted in a high-performance intake manifold. Additional information about the design is mentioned below:

Table 7: Component Specifications

Part	Material	Manufacturing Method	Specifications
Throttle Body	Aluminium Alloy	OEM (Yamaha R15)	Bore – 28mm
Venturi	Aluminium 6 Series	CNC Lathe	16° – 6° (Converging Angle – Diverging Angle)
Plenum	Nylon	3D Print (SLS)	Volume – 2.6 l
Runner	Aluminium 6 Series	Tube Bending	ID – 42mm

Figure 9: Intake CAD Model

Figure 10: Intake Installed on Fsaе Car

REFERENCES

1. Heisler H. *Advanced Engine Technology* 1995.
2. Plank M. *Engine Optimisation And Performance Characteristics For A Formula SAE Race Car* 2005.
3. Margary R, Nino E, Vafidis C. *The Effect Of Intake Duct Length On The In-Cylinder Air Motion In A Motored Diesel Engine*. SAE Trans 1990:322–33.
4. Cauchi J, Farrugia M, Balzan N. *Engine Simulation Of A Restricted Fsaе Engine, Focusing On Restrictor Modelling*. 2006.
5. Vizard D. *How To Build Horsepower: Volume 1: Methods For Building Horsepower In Any Engine*. SA Design Books; 1990.
6. Umesh, K., Pravin, V., & Rajagopal, K. *An Approach (Performance Score) For Experimental Analysis Of Exhaust Manifold Of Multi-Cylinder SI Engine To Determine Optimum Geometry For Recreational And Commercial Vehicles*.
7. Blair GP, Cahoon WM. *Special Investigation: Design Of An Intake Bellmouth*. Race Engine Technol 2006;17:34–41.
8. Ling J, Tun LTY. *CFD Analysis Of Non-Symmetrical Intake Manifold For Formula SAE Car*. 2006.
9. Vaughan A, Delagrammatikas GJ. *A High Performance, Continuously Variable Engine Intake Manifold*. 2011.

APPENDIX

$$M = C * A \sqrt{k * \square * P * (2/(k + 1))^{(k+1)/(k-1)}} \quad (1)$$

Where, M = mass flow rate

C = discharge coefficient

A = discharge hole cross-sectional area

k = (c_p / c_v) of gas

c_p = specific heat of the gas at constant pressure

c_v = specific heat of the gas at constant volume

P = absolute upstream pressure of gas

Inserting following values,

$C = 1$; $A = 3.14 \times 10^{-4} m^2$; $c_p = 29.19 J/mol \cdot K$; $c_v = 20.85 J/mol \cdot K$; $K = 1.4$; $\dot{\rho} = 1.2041 Kg/m^3$; $P = 101325 Pa$

This gives, $M = 0.075 kg/s$