Designing an Air Intake System of a Formula Student Race-Car

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Abstract— Extracting the most power from the engine and tuning the engine to perform in the particular RPM range is quintessential for any race car. This paper will focus on the Principles and foundation behind the working of Intake system and how to design the same for naturally aspirated single cylinder engine according to the performance requirement with the help of 1-D and 3-D simulations.

Key words: Air Intake System, Formula Student Race-Car

I. Introduction

Air intake system comprises of all the components which assist internal combustion engine to suck air from the atmosphere. Along with the primary function of allowing air for combustion, it ensures air is clean and also in some case provides casing for mixing of fuel. Air intake system comes with variety of configuration and main components of air intake system are: Runner; Plenum; Throttle body; and Fuel injector. . Engine used is ktm 390cc with actual displacement of 373.2 cc.By varying above components geometry one can finely tune the engine according to their requirement. For instance, short runner length improves high RPM range performance and vice versa.

A. Design

The most suitable configuration for the intake system was selected after following analytical steps and then it was validated through 1 D simulations in RICARDO wave tool. Analytical solutions predict best configuration for the individual component but in reality, every component are assembled together and their combined effect can only be predicted by 1-D simulation. After, narrowing down to best configuration, computation flow dynamics tool like ANSYS fluent was used to visualise the air flow and reduce the pressure losses and vortex formation.

B. Design Constrain

According to the event rules, air must pass through the 20 mm air restrictor and that air restrictor must be located after the throttle body of an intake system.

C. Nomenclature

Intake system constitutes 3 major components which are:

1) Intake Runner

A series of tubes and channels which provides air-fuel mixture to the cylinders. Acoustics or pressure wave theory was used to find the appropriate runner length.

2) Plenum

Air chamber which assist in equally distributing air to all the cylinder in case of multi cylinder engine and also damp the pulsations formed by cylinders. Helmholtz resonator theory was used to find the appropriate volume of chamber and CFD analysis was done to determine the shape.

3) Throttle Body

It is actuated by the help of accelerator pedal and provides driver to control the engine revs.

D. Design Data & Calculation

1) Restrictor Design

According to rule all air must pass through 20 mm circular restrictor.so to obviate this restrictor effect to some extent we design cd nozzle for efficient breathing of engine even at high rpm.

Providing the hole of 20 mm will induce lot of pressure loss and low volumetric efficiency. For achieving maximum volumetric efficiency, nozzle of ISO: 9300 was selected with four different variables.

a) inlet area
b) inlet radii curvature
c) diffuser angle
d) diffuser length

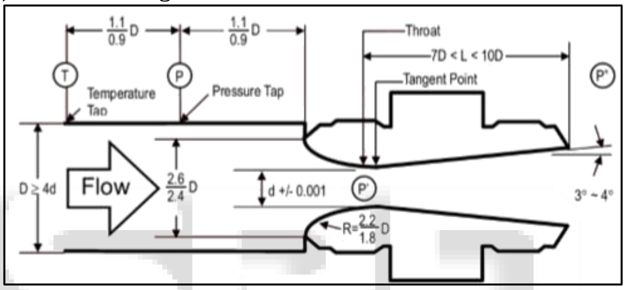

Fig. 1: ISO 9300 Nozzle

With D =20 mm

- Range of values for other parameters were determined.
- It gives throttle opening= inlet area diameter in range of 48mm to 56mm. but, this large throttle opening gives poor throttle response.
- So, throttle opening was reduced to 34mm as stock throttle is available of this size and throttle opening is fairly good.
- For determining diffuser length and diverging angle of nozzle, CFD analysis was done in ANSYS fluent.
- For 20mm throat converging diverging nozzle, maximum mass flow rate achieved during choking is 0.0703kg/s.

$$\dot{m}_{air} = \frac{A_{throat}P_{total}}{\sqrt{T_{total}}}\sqrt{\frac{\gamma}{R}}M\left(1+\frac{\gamma-1}{2}M^2\right)^{\frac{-\gamma+1}{2(\gamma-1)}}$$

- $\dot{m}_{air}$= mass flow rate of air
- A_{throat}= cross-sectional area of throat
- P_{total}= total pressure
- T_{total}= total temperature
- R= ideal gas constant
- γ= specific heat ratio
- M= mach number

For ktm 390cc, max mass flow rate required is 0.030kg/s. so, only consideration while designing nozzle was of pressure loss

Physical model chosen: K-epsilon with pressure based solver.

Inlet conditions: velocity = 10m/s

Pressure at exit: atmospheric.
Inference after analysis
Increasing diffuser length helps in pressure regain.
b.Less is diffuser angle, less is pressure loss and more diffuser angle leads to flow separation.
Final dimension of converging diverging nozzle: Diffuser length: 11cm
Inlet diameter: 34mm
Diffuser half angle: 4°

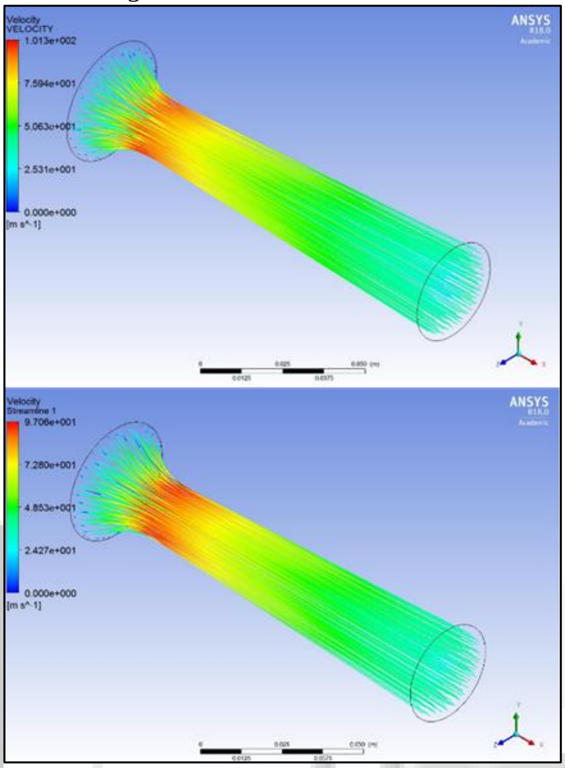

Fig. 2: Velocity Streamlines

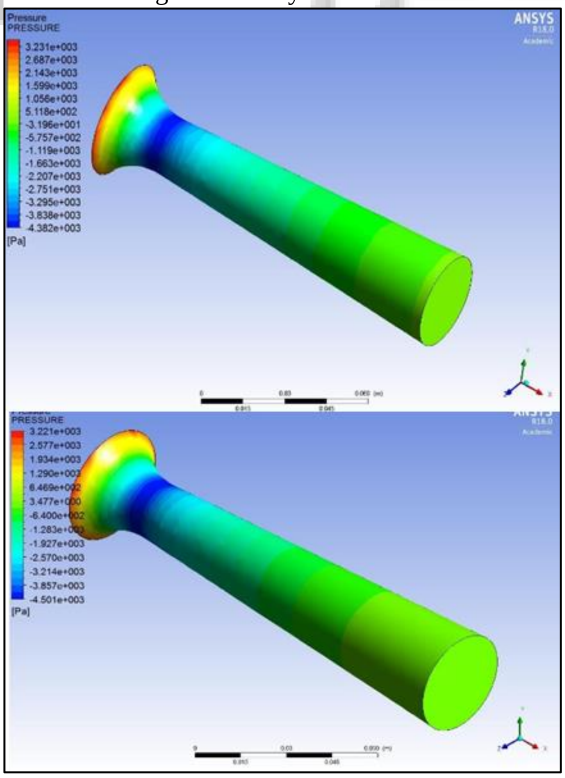

Fig. 3: Pressure Drop Analysis

II. ANALYTICAL CALCULATION FOR INTAKE RUNNER

For designing cylinder runner, deep understanding of wave theory is required.

A. *Induction Waves*

Let's first look at what happens in the manifold to better understand how to use it to our advantage. When an engine is moving there are high and low pressure waves moving in manifold caused by inertia of air and opening and closing of intake and exhaust valves. The idea of port tuning is to have high pressure wave approach intake valve before intake valve closes or just opens, forcing in little more charge.

The most commonly known cause of pressure wave is the piston as it moves down the bore. During intake stroke, piston makes negative pressure waves that travels from piston to open intake tract. Once the negative pressure wave reaches the plenum area, it is reflected as positive pressure wave. The positive pressure wave travels down back into the cylinder. If it reaches intake valve just before it closes, it will force little more air in cylinder.

The second cause is exhaust system, if exhaust system scavenges well then it will also create negative pressure wave that will travel up the intake tract and positive pressure wave will get reflected which in turns add more intake charge.

Third cause of pressure wave is when the intake valve closes, any velocity left in the intake column of air will create high pressure wave at back of valve and that pressure wave will travel up the intake tract and will get reflected as negative pressure wave, again when it reaches the intake valve, it should be closed which will reflect wave as negative pressure wave and it will again travel up the tract and reflect as positive pressure wave. This wave should be timed to push in more intake charge.

B. *Combined Effect*

In a well-tuned intake setup, there will be high pressure wave at intake valve during the time of opening and if at the same time, exhaust valve is open, that is engine is in its overlap period, there will be high pressure at intake valve and low at exhaust opening and this will help in filling charge more rapidly.

C. *Calculation for Determining Intake Runner Length*

L= (EVCD*0.25*V*2)/(rpm*RV) - (1/2*D)
Where,
L= runner length
EVCD = effective valve closing duration
RV = reflective value (4,8)
V= pressure wave speed
D= runner diameter = 48mm (initially, same as that of port)
EVCD = 720°-(Exhaust closing duration)
EVCD= 720-226+20
(20° is added to get effective valve open duration)
EVCD= 514°
This give us runner length of 400mm.
This value is approximate value and cannot be used directly given the complexity of the system.

III. PLENUM VOLUME CALCULATIONS

Plenum volume calculation is done by Helmholtz resonator theory. With known intake runner we can estimate the plenum volume and set our power band as our requirement.

Intake tuning is done for power band 5000-7000 rpm.

Helmholtz resonator is constituted by a cavity and a short duct, which connects it to the system. It behaves like a system composed by a pneumatic spring (the cavity) and a mass (the gas inside the short duct): in consequence, it has a natural frequency of pressure oscillations which can be calculated as follows:

$$f_H = \frac{c}{2\pi}\sqrt{\frac{S}{LV}}$$

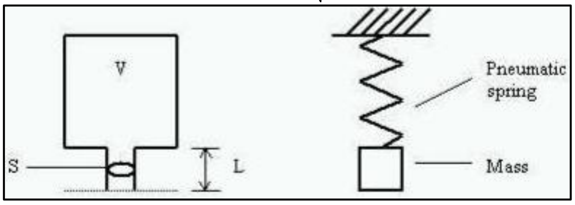

Fig. 4: Helmholtz Resonator Analogy

Where c is the speed of sound. The resonators are utilized in the field of acoustics to control the sound pressure level inside cavities (rooms, boxes, machine cavities) or ducts (conditioning, ventilation, exhaust systems), and their efficiency is largely proved and experimented.

The intake system without any resonator, composed by the cylinder (spring) and a simple intake duct (mass), can be seen like an Helmholtz resonator, whose resonance frequency is given by the equation above (the volume to consider is the mean cylinder volume, and the length should be corrected considering the little mass of gas outside the duct that moves together with the inner one). The optimum filling is obtained when the natural frequency is about double the piston frequency, and the torque curve shows only one peak.

The following figure shows an example of an intake system with an in-line resonator, for a single cylinder engine. The whole intake system behaves like double Helmholtz resonator, being constituted by two ducts and two volumes. Choosing the dimensions of the pipes, it is possible to calculate the behaviour of the filling index by varying the piston speed Np [rpm] and the ratio RV between the volume of the cylinder and the volume of the resonator. The figure below shows the lumped element equivalent system.

The presence of second spring adds second natural frequency which gives us one more frequency at which peak occurs. Natural frequency obtained is twice the piston frequency.

$$f_1^2 = \frac{1}{8\pi^2}\frac{c^2}{V}\left(\frac{1}{L_1}+\frac{R_v}{L_2}+\frac{R_v}{L_1}-\sqrt{\left(\frac{1}{L_1}+\frac{R_v}{L_2}+\frac{R_v}{L_1}\right)^2-4\frac{R_v}{L_1L_2}}\right)$$

$$f_2^2 = \frac{1}{8\pi^2}\frac{c^2}{V}\left(\frac{1}{L_1}+\frac{R_v}{L_2}+\frac{R_v}{L_1}+\sqrt{\left(\frac{1}{L_1}+\frac{R_v}{L_2}+\frac{R_v}{L_1}\right)^2-4\frac{R_v}{L_1L_2}}\right)$$

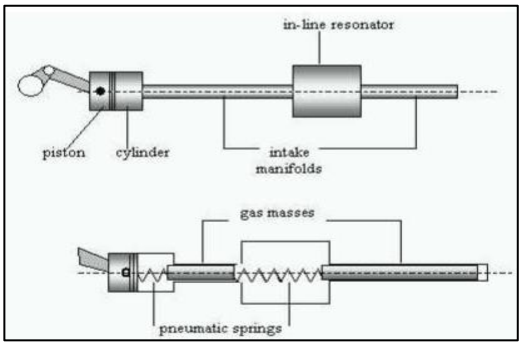

Fig. 5: Equations of double Helmholtz resonator and double engine with inline resonator representation

Where, L_1 and L_2 are duct length to area ratio of runners attached to cylinder and open to atmosphere respectively.

L_2 was assumed to be 15 cm long and with 400mm intake runner length, code for Helmholtz resonator is formed in JAVA and frequencies were calculated.

Analytical predictions:

1) Volume: 1200 cc
- Runner length: 400mm
- L_1: 184.03
- L_2: 91.46
- F_1: 181 = 2500 rpm
- F_2: 314 = 4500 rpm

2) Volume: 1600cc
- Runner length: 400mm
- L_1: 194.86
- L_2: 91.46
- F_1: 176 = 2350 rpm
- F_2: 325 = 4750 rpm

3) Volume: 2500 cc
- Runner length: 400mm
- L_1: 204
- L_2: 91.46
- F_1: 170 = 2100 rpm
- F_2: 333 = 5100 rpm

Now, with the value obtained from theoretical calculations; 1-D calculations are done in RICARDO wave.

A. 1-D Engine Simulations

RICARDO wave is used for engine simulation. 1-D simulations can predict performance when multiple systems work together.

1) Setup

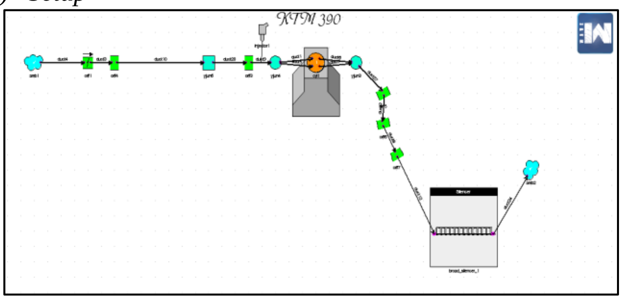

Fig. 6: Setup in RICARDO Wave

Some graphs of torque and Bhp with varying plenum volume:

Results are displayed for plenum volume of 1000cc to 2500cc.

It can be inferred that the increasing the volume of plenum helps in improving mid-range rpm performance, keeping the runner length constant which is equal to 400mm.

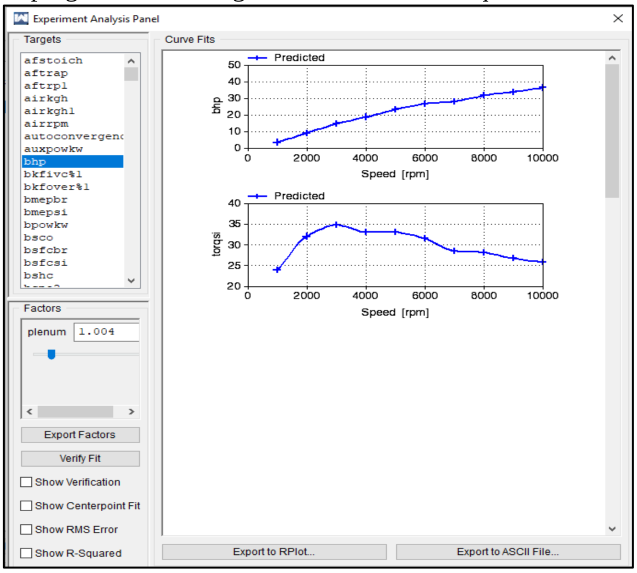

Fig. 7: Bhp and Torque curve for 1000cc plenum

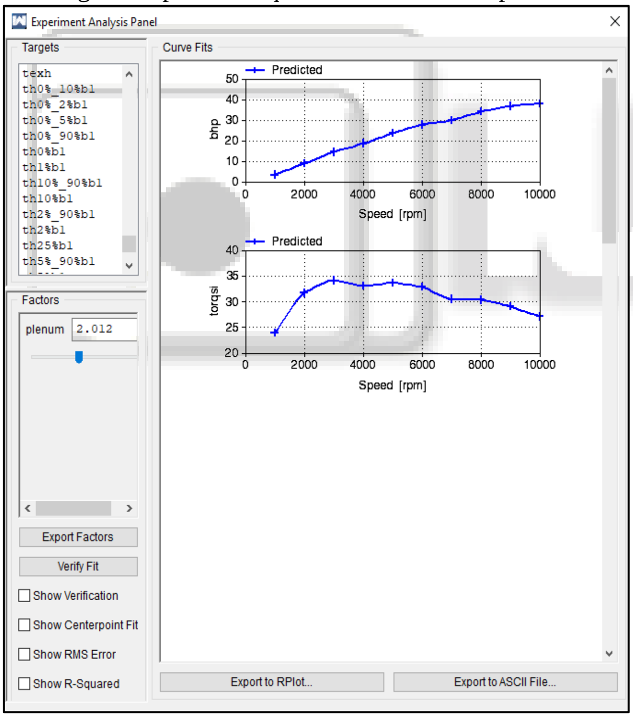

Fig. 8: Bhp and Torque curve for 1500cc plenum

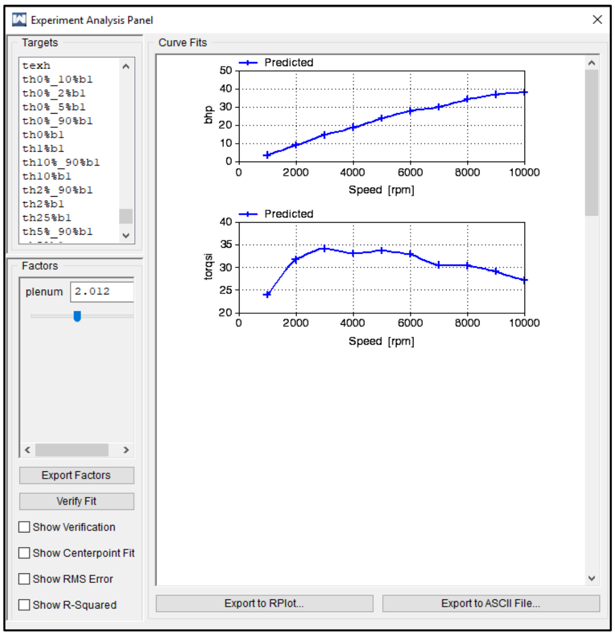

Fig. 8: Bhp and Torque curve for 1500cc

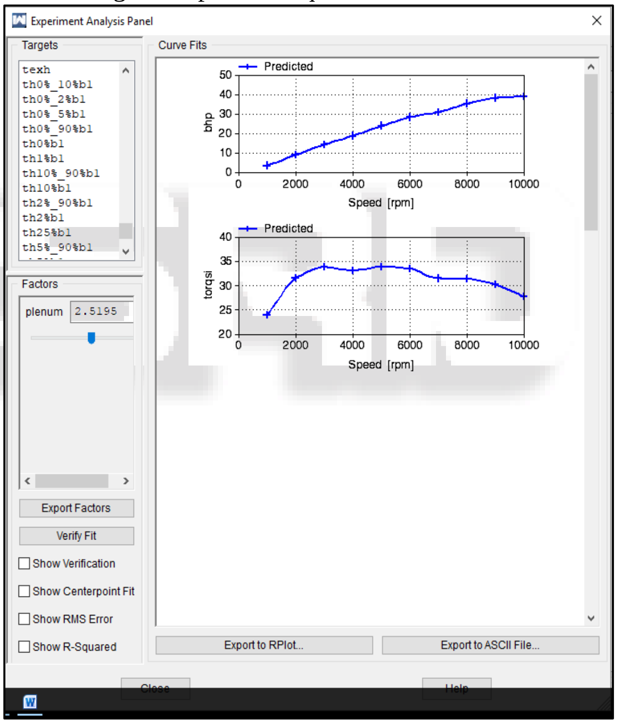

Fig. 9: Bhp and Torque curve for 2500cc plenum

Due to space constrain, 400mm runner length was not possible and optimization study was carried out with all the factors including plenum volume, runner length and diffuser length.

Final configuration:

– Runner length: 200mm
– Plenum volume: 2000cc
– Diffuser length: 150mm

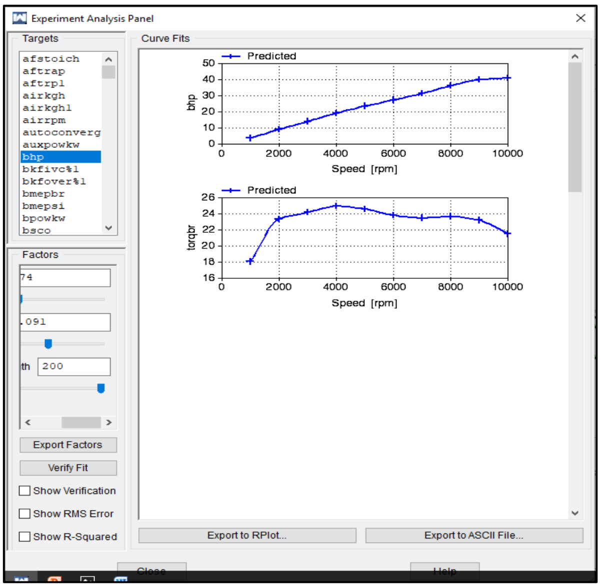

Fig. 10: Bhp and torque curve of final configuration

This shifted the rpm range a little to the left from the intended. New rpm range was 3500-6000 RPM.

Flow simulations were carried out in ANSYS fluent with different shapes of plenum. As plenum volume is large, flow circulation was seen in flow analysis.

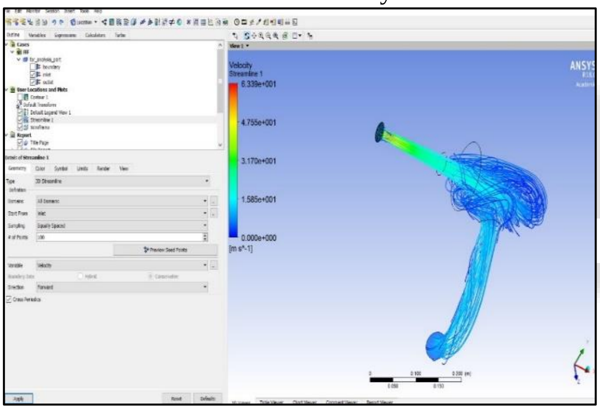
Fig. 11: Flow Simulation of Geometry

To reduce and the flow circulation, other geometry was modelled and analysis was done.

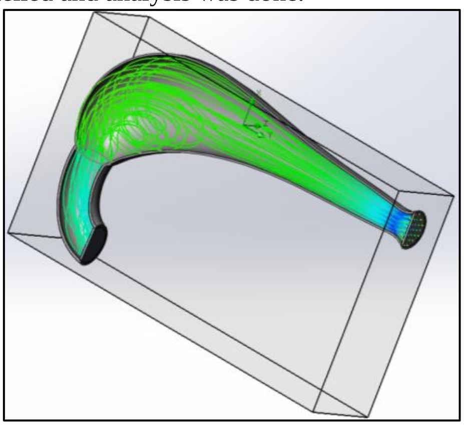
Fig.12: Final Configuration of Intake

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