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Design and Tuning of Intake Manifold by Inertial and Acoustic Resonance Effects

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Abstract: ***The aim of this project is to create an intake assembly to be fitted in the FSAE race car. To comply with the rules of FSAE, each racecar must have a restriction device that has a maximum diameter of 20 mm installed between the throttle body and intake manifold be it a single or multi-cylinder engine. Naturally aspirated internal combustion (IC) engines with a fixed intake system are generally tuned to produce an induction boost at a single-engine speed by taking advantage from the induction pressure waves only for a limited speed range. This paper investigates the individual and combined effects of inertial and acoustic resonance effects on the intake manifold performance parameters of an IC engine at different engine speeds. From the data gathered through the numerous simulations in ANSYS and RICARDO WAVE, the calculations and design was done. This design made was based on our vehicle chassis, which shouldn't exceed the rules defined.***

Keywords: Wave tuning, induction pressure, valve timing, volumetric efficiency, 1D simulation, CFD.

I. INTRODUCTION

In a naturally aspirated engine, the engine creates a low pressure during the intake stroke, which sucks the air from the atmosphere into the combustion chamber and sending the burnt gases out. This process is complex and depends on many design parameters such as runner length, port volume, length, diameter, taper, valve size, number of valves, valve timing, and lift [3, 4]. The pull will be more when rpm increases which will lower the pressure created inside the cylinder. We can improve the gas exchange process by tuning of intake port lengths which are significant in providing an extra boost to the induction pressure. This pressure boost would increase the volumetric efficiency and torque of the engine [5].

As the engine speed will continue throughout its operation with respect to the throttle angle, desired torque, and vehicle velocity, a set of parameters will not be enough to optimize the performance of an internal combustion engine at all operating engine speeds.

To resolve this,

A. Design an infinitely variable transmission which operates at a single speed by varying the torque and speed as per the operating conditions to get the peak performance. However, due to low efficiency of transmission, and inability to transfer high torque at low engine speeds, its application has been limited to smaller automobiles. This approach also has issues with weight, reliability, and packaging [2].

B. Varying a set of parameters governing the engine's performance with respect to its operating speed. This would allow the engine to produce optimal performance throughout its entire operating speed range. This is more feasible and is being extensively studied in the automotive and motorsports industries and this research work would be contributing to the same.

Wave tuning of the intake manifold in an engine can add to engine performances without depending only on superchargers and turbochargers. The nature of the intake system's design will have a greater influence on the control of the fluid flow into the cylinder that is vital to achieve a great engine performance. The idea of wave tuning an engine is to increase the induction boost by taking advantage of the high-pressure waves in the intake runners and scavenging efficiently by the low-pressure wave in the exhaust headers.

A study on the effects of tuning for induction of charge into the engine versus effects of tuning for scavenging the exhaust gas out of the engine says that intake tuning can boost the performance of an IC engine by over 12% as compared to exhaust tuning which could increase the performance by 5% and that tuning intake manifold and exhaust manifold are mutually independent and the effect of tuning both ends would result in an improvement which is the sum of the contribution of them both [4]. As mentioned before, this research focuses on intake tuning, which contributes to the most improvement in volumetric efficiency and power and torque output.

We used a single-cylinder; water-cooled 373 cc engine having an output of 43 brake horsepower at 9000 RPM and a torque of 36 N-m at 7000 RPM. The engine comes with a six-speed manual gearbox and weighs 35kg. The flow of air takes place from the restrictor, plenum, and runner (intake manifold) to the combustion chamber through the throttle body.

The main difficulty with the use of a single-cylinder engine is that the measured performance does not match that with the multi-cylinder. The reasons for this may be as following,

1) Friction levels per cylinder are more in the single-cylinder engine due to the extra mechanical systems (balance shafts) which are required to balance the engine.
2) Absence of exhaust tuning found with the multi-cylinder where cylinders empty into a common exhaust pipe.
3) Cyclic nature of the intake flow is unsteady which is the subject of this paper. The term ‘wave action’ is commonly used to describe such flow and it has two effects: ‘inertial ram effect’ and a resonant acoustic behaviour.

The calculations using resonant wave action theory based only on inertia effects, which did not match experimental results very well [8]. Hence both inertial and acoustic resonance effects are considered and are found to be important, the relative importance of each varying with engine speed and intake pipe length [9]

II. METHODOLOGY

The rarefaction and compression of waves produced in the intake system are important for the overall engine’s performance. Usually, length is more evidently involved in wave tuning, as different lengths allow different timings for the waves to arrive at the start of the valve overlapping period. The diameter may not be involved with the timing of the pressure waves, but it is involved in the intensity. Increasing the pipe diameter will decrease pressure and velocity of the gas. This also reduces the amount of friction in the pipes making gases move easier. [10].

A. Objectives

1) To increase the pressure recovery of air in air restrictor.
2) To achieve enhanced engine performance by increasing the airflow and High air velocity for a given flow rate
3) To improve the throttle response of the engine.
4) Low resistance to airflow
5) Excellent fuel and air distribution throughout
6) Minimize the mass of the system which would help in reduction of mass of the vehicle.

We used a 1-D engine simulation software that is accurate to test the intake and exhaust systems of an engine. Due to lower cost, simplicity, and adjusting countless engine properties, RICARDO WAVE (ISO approved) has been frequently used and mentioned in recent studies [11].

Ricardo uses the SI Wiebe combustion model and the Woschni heat transfer model. They are given as follows:

Wiebe Function, $F = 1 - \exp\left[-a\left(\frac{\emptyset-\emptyset_0}{\Delta\emptyset}\right)^{m+1}\right]$

Where, F = mass fraction burn;	$\emptyset$= crank angle;
$\emptyset_0$ = angle of the start of the heat addition;	$\Delta\emptyset$ = combustion duration;
a = efficiency parameter;	m = shape factor.

Woschni Function, $H = 0.0128 d^{-0.2} P^{0.8} T^{-0.55} u^{0.8} K$

Where, H = heat transfer coefficient;	u = average gas velocity;
P = cylinder pressure;	T = cylinder temperature;
d = cylinder bore;	K = multiplier

B. Pressure wave

The pressure waves mentioned above have speed, tone and amplitude and they are amplified. These waves travel at the speed of sound. In hot intake air (25 to 30^{0}C) it will travel at about 381 to 396.2 m/s. The speed of these pressure waves is not a function of the engine’s operating speed and thus, wave tuning only works in a specified rpm range. [6] The runners serve the purpose of tuning the engine according to the RPM range. The runners can be tuned for Low-End Torque and High-End Horsepower. The length of the runners determines its best RPM range. The shorter runner is used for High-end Horsepower as the suction is lesser and the stroke can be completed faster, which gives more power at the Higher end. Whereas the larger runner is used for lower-end torque, as the length is more the air has to travel more than required. This depicts that air is in the form of waves that travel from the plenum to the engine as per the stroke. As the air is being sucked during the suction stroke the waves try to rush into the combustion chamber and during the closing of the valves these waves are reflected which causes them to return to plenum. As the waves reach the end of the runner they meet with a new boundary which reflects them to the valves as shown in Fig 1. However, as valves are closed, they interfere among themselves. Creating a higher pressure wave behind the valves.

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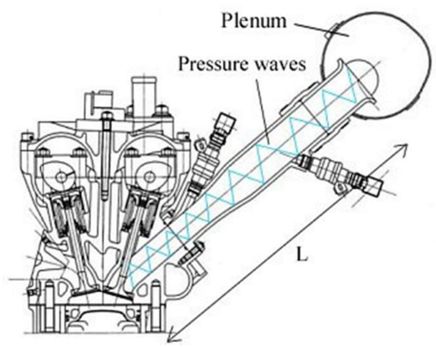


Fig. 1 Depiction of pressure wave produced

This creation of altering compression and rarefaction, an acoustic wave is formed. This serves as natural turbocharger which can be used only at a specific RPM range. This acoustic behaviour is always present irrespective of which RPM range the engine is running on.

C. Chrysler's ram air theory

The main reason of this theory is focused on amplified pressure waves and creating induction boost.

When an engine is tuned, there is an increase in the pressure differential between the amplified pressure behind the intake valve and the negative pressure created inside the cylinder due to the engine's suction. When these high-pressure waves are released into the low-pressure combustion chamber, the mass flow rate of air into the cylinder is boosted thereby increasing the filling capacity of the engine and thus, the volumetric efficiency. This process is called as intake air ramming [7]. Peak torque occurs at peak volumetric efficiency. Also, peak torque rpm corresponds to the rpm where there is a sudden increase in engine power [1]. So, the occurrence of peak torque can be controlled by varying the runner length.

D. Helmholtz resonance theory

This theory provides an equation to tune the intake lengths and justifying the need to reduce intake runner lengths by considering multiple reflections of the pressure waves.

When the intake valve is closed, port and runner system resonates like a closed-ended organ pipe, with a resonant frequency (Hz) of:

$$Fp = \frac{C}{4 \times Le} = \frac{C}{4 \times (L + 0.5D)}$$

where, C = Speed of sound at intake temperature

Le = effective length of runner = (length measured (L) + 0.5 × average runner diameter (D))

The reflection at the open end of the intake runner is by half the intake pipe's diameter. Hence, the effective runner length is more than the actual measured intake runner length by half times the average runner diameter. In the case, when the intake valve is open, tuning-peak occurs when Helmholtz resonance of cylinder and runner as a combined system is double that of piston frequency.

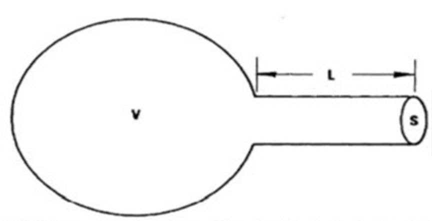
The plenum and runner system act as a single Helmholtz resonator as shown in Fig. 2, with a resonant frequency (Hz) of:

$$F_h = \frac{c}{2\pi}\left(\frac{A}{L_e \times V}\right)^{0.5} = \frac{c}{2\pi}\left(\frac{A}{V(L + 0.5D)}\right)^{\frac{1}{2}}$$

Where, A = Inlet pipe cross-sectional area

L_e = Effective length of runner = (length measured (L) + 0.5 × average runner diameter (D))

V = Effective cylinder volume



Helmholtz resonator. V = volume of the cavity; L = length of neck, and S = cross-sectional area of neck.

Fig. 1 Helmholtz resonator

The effective cam duration (ECD) will be about 20 percent of cam duration. This gives us as follows [4]:

$$L = \left(\frac{(720 - ECD)C}{N \times 2 \times R}\right) - 0.5D$$

Where, C = Speed of sound at inlet temperature
N = Engine speed
D = Average diameter of runner
R = Reflection value
ECD = Effective cam duration

E. Engelmann's formula

Using the Helmholtz theory and considering the engine cylinder dynamics, Engelmann was first who proposed the following equation which is useful over a wider range of engine speeds.

$$N = \left(\frac{162}{K}\right) * C * \sqrt{\frac{A}{LV}} * \sqrt{\frac{R-1}{R+1}}$$

Where, N = Engine Speed (rpm)
C = Speed of sound at intake temperature, ft/s
L = Measured length of intake runner,
R = Compression ratio
K = 2.0 to 2.5 for most conventional engines
V = Effective cylinder volume, in3
A = Inlet pipe cross-sectional area

III.DESIGN

The whole design formation for the intake was done in CAD part in SOLIDWORKS and is imported in ANSYS workbench. After importing in fluent the geometry edit is done in design modeller which gives an advantage of mesh continuation between inner and outer domains as shown in fig 3. Structural meshing is done in tetrahedron elements having nodes and elements as 90991 and 344376 respectively which is later converted to polyhedral mesh.

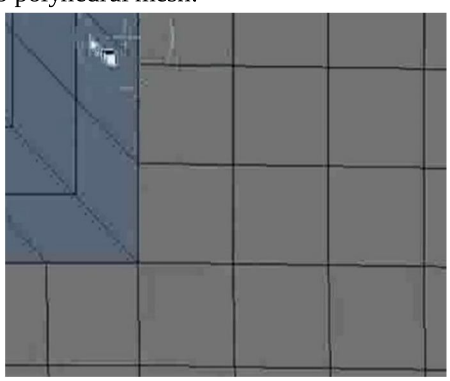


Fig. 3 Continuation of mesh at the boundary of outer and inner domain

The meshed geometry is further imported in fluent to perform CFD analysis. The reason for choosing fluent over CFX is:

1) Fluent uses a cell approach whereas CFX uses a vertex approach. Fluent is capable of handling polyhedral mesh while CFX only handles traditional tetra and hexa mesh.
2) Fluent can post-process on its own whereas CFX needs a post-processor.
3) Mesh Adaption is weaker in CFX compared to Fluent.
4) Use of GPU for Simulation acceleration can be done in Fluent.
5) Fluent is preferred for supersonic and hypersonic flows (high Mach number flows). [13]

The viscous model is K-epsilon (k-ε) turbulence as it is the most common model used in Computational Fluid Dynamics (CFD) to simulate mean flow characteristics for turbulent flow. It is a two-equation model where,

a) The first determines the energy in the turbulence and is called turbulent kinetic energy (k).
b) The second is the turbulent dissipation (ε) that is the rate of dissipation of the turbulent kinetic energy.

The boundary condition of inlet and outlet is given. Then the variables of the solution method and controls were entered into this setup and were undergone for more than 120 iterations after which the results were plotted in CFD post.

A. Restrictor

The restriction rule limits the maximum power of engine by reducing mass flow rate flowing to engine. Thus a restrictor should be designed to allow maximum possible airflow and maintain minimum pressure difference across the restrictor.

For engine to not suffocate at high speeds, either mass flow rate or the velocity of air should increase. Due to restriction of area very high velocity of air should fill the combustion chamber. For which we have to reduce pressure drop and provide charge to engine with minimum pull.

Broadly two obstruction meters used, orifice and venturi meter

The pressure loss and coefficient of discharge for Venturi tube (0.85) compared to that of orifice plate (0.65) is much better. Therefore, considering the efficiency venturi meter is reliable. The diameter of the throttle body is 48mm. As the throttle body fits in the inlet face of the Venturi the diameter of the throttle body would serve as the inlet and outlet diameter of the Venturi tube.

Mass flow rate can be calculated by using chocked flow equation,

$$\dot{m} = \frac{AP}{\sqrt{T}}\left[\sqrt{\frac{K}{G}}M\left(1+\frac{K-1}{2}M^2\right)^{-\frac{K+1}{2(K-1)}}\right]$$

Where,

A=Area=0.001256 m^2 (20 mm restriction)

G=Gas constant=0.286 KJ/Kg-K

K=Specific heat ratio=1.4

P=Total pressure=101325 pa

T=Total temperature =300 K

M= Mach number=1(choked flow)

We get mass flow rate ($\dot{m}$) =0.0703 kg/s

B. Plenum

The aim for designing the Plenum is to,

1) Minimize the pressure loss from inlet of the throttle body to the start of the Plenum and recover the pressure loss.
2) There must not be any sudden increase in the cross-sectional area in the flow to avoid the formation of contours which will retard the flow
3) The best way to choose the runner joining on plenum is by iterative 1D and CFD simulations.

The volume of plenum is an important for the performance of the vehicle. The less time required for air to be sucked in to the combustion chamber the more the throttle response on the other side, if the size of the plenum is large, more time is taken in getting fresh air thereby reducing the throttle response. As is also the case that less the volume more work required for grabbing in fresh air and filling the plenum. Alternatively, bigger the size more air being stored in the plenum at once so less work required for sucking in fresh air. Therefore, we have designed the plenum volume at approximately 5 times to the cylinder displacement volume which was altered to 4.7 times in CFD analysis.

C. Runner

The optimum design would be to tune the alternating acoustic wave at the peak torque RPM range which gives ram effect.

Power - 43BHP at 9000rpm	Torque - 35N-m at 7000rpm
Bore - 89mm	Stroke - 60mm
Valve per cylinder - 4	Intake valve opens - 2˚ before top dead center (TDC)
Intake valve closes - 44˚ after bottom dead center (BDC)	Effective Cam Duration - 226˚

Peak Torque of the engine at 7000 rpm and Max Power of the Engine at 9000 rpm but engine remains at 8000 rpm in the dynamic condition.

1) The intake valve is open 226 degrees out of 720 degrees, so time intake valve closed = 720-226 = 494 degree = 1.3722 rev.
2) Speed of sound in air is 343 m /sec at 30^0 C.
3) The engine speed in rps (revolution per second) as SI units are considered= 133.3333 rev/sec.
4) Time taken for one revolution = 1/133.333 sec = 0.007500 sec.
5) The time taken for valve to close and open again = 1.3722*0.007500 = 0.01029 sec.
6) The distance covered by wave which is at the speed of sound during this time= 343*0.01029 = 3.529 m

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7) As the pressure wave has to travel back and forth, the length for the intake runner to use the ramming effect at 8000 rpm is half of the length calculated = (1.76 m).
8) For the runner length to be adjusted to the space available. we, shorten the runner length to exactly one sixth of the calculated length. This will give us a runner length of 0.293333 m = 293.34 mm.

Which means the pressure wave will travel up and down the pipe six times before the intake valve opens again. But it still arrives at the same time to the valve.

The length of the runner covered in engine is 98 mm and the length of throttle body is 87 mm, so for the design of intake system the lengths of runner require is 293.34-185 = 108.34 mm.

According to induction wave tuning theory, intake system was tuned at 5000 RPM, resulting in total runner length of 302.332mm. Runner Diameter is selected as 48 mm as that of throttle body as both are coupled with air tight seal.

The point to note in calculating the runner length is not to change the diameter of runner as this changes the organ pipe length and results in failure of acoustical resonance effect.

IV. RESULTS AND ANALYSIS

1D simulation was done considering restrictor as duct with left diameter of 20 mm (as per FSAE rule) and right diameter of 48 mm as mentioned above. Plenum was imported as a component which was made into many ducts where we made sure that the relative volume error in each of the ducts divided was less than 6%.

Here runner was treated as a default duct where the properties were given same as of the material used to manufacture the intake manifold. This saved a lot of time making the variable (length of runner) easy to access, change and iterate many times. The exhaust header and muffler are also imported so that we can simulate even effectively for the scavenging.

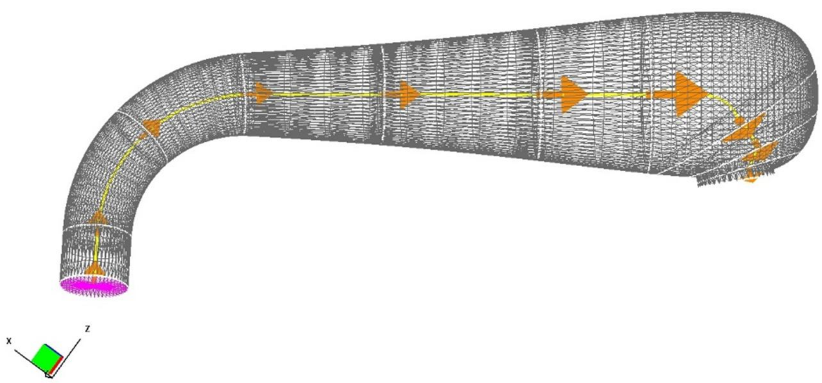


Fig. 4 Wavemesher model of plenum

The brake power obtained was 26kw and brake torque of 30.215 Nm with the tuned runner and exhaust assembly as shown in the network. The total volumetric efficiency achieved was about 87.6%.

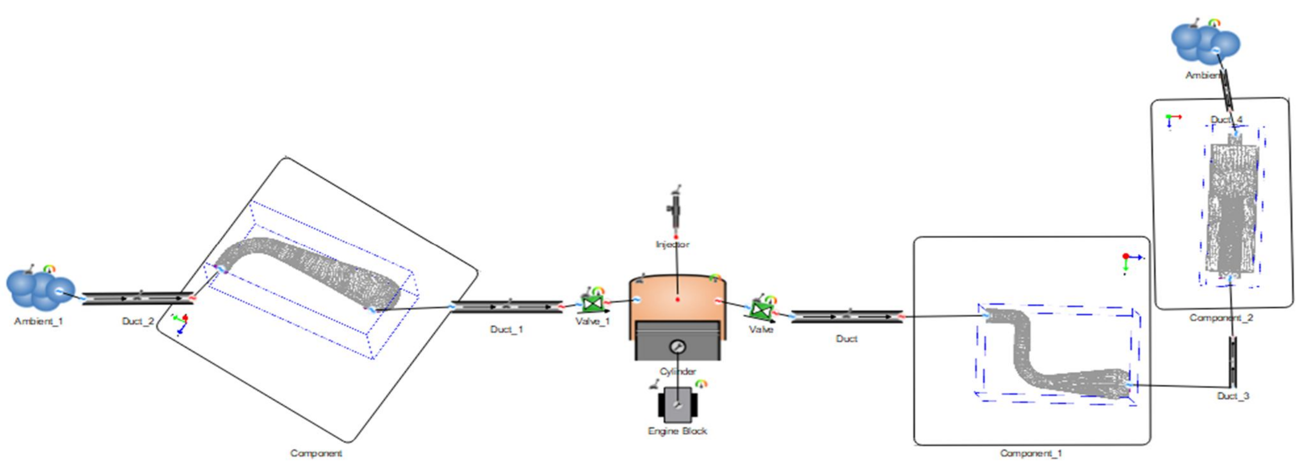


Fig. 5 Wave network

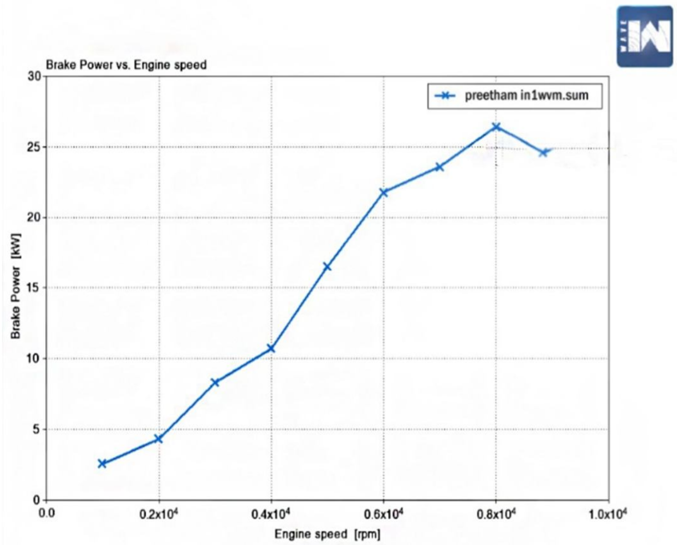


Fig. 6 Output graph of BP vs rpm

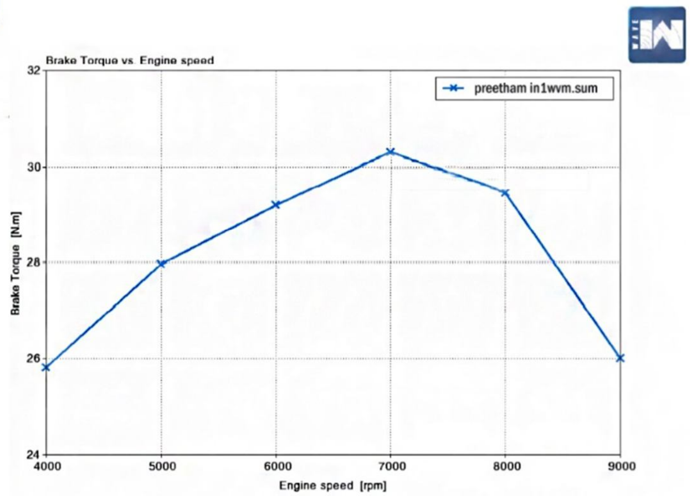


Fig. 7 Output graph of BT vs rpm

The pressure lost by restrictor was recovered in the plenum and the total loss was very minimal with an increase in velocity of 21m/s where inlet was subjected at 13.3339m/s.

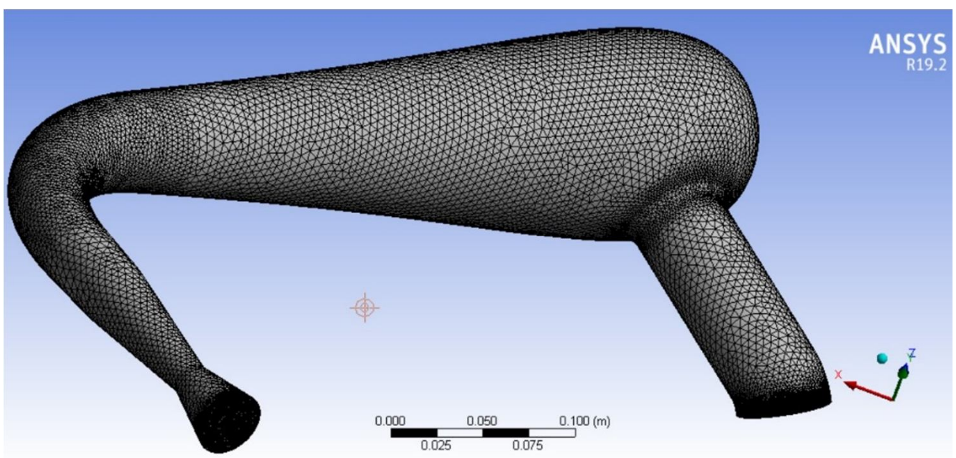
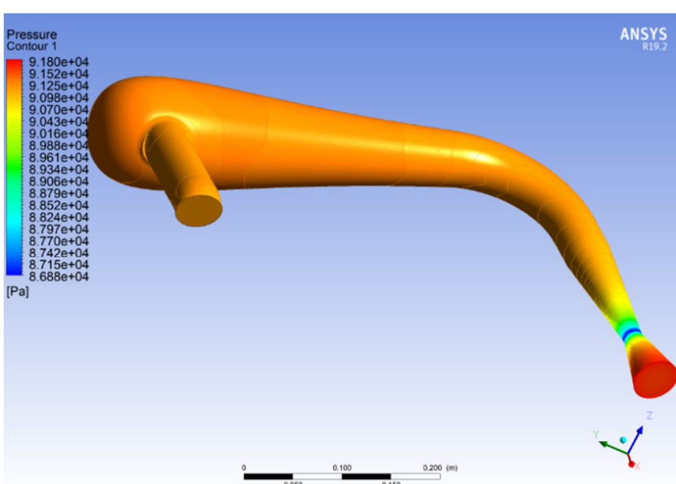
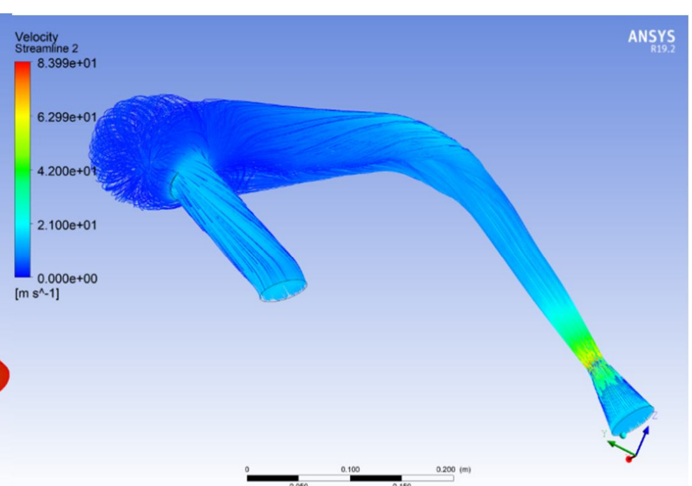
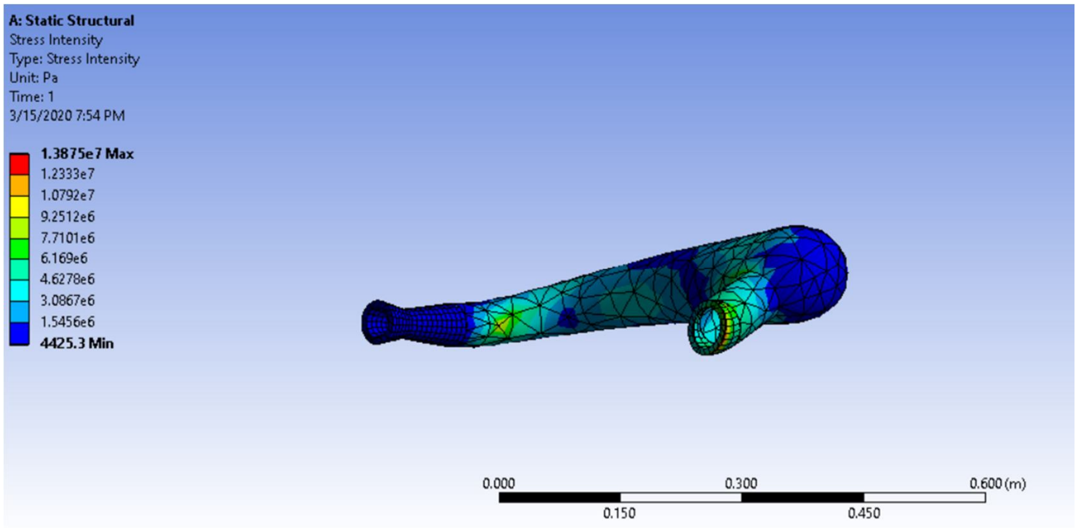


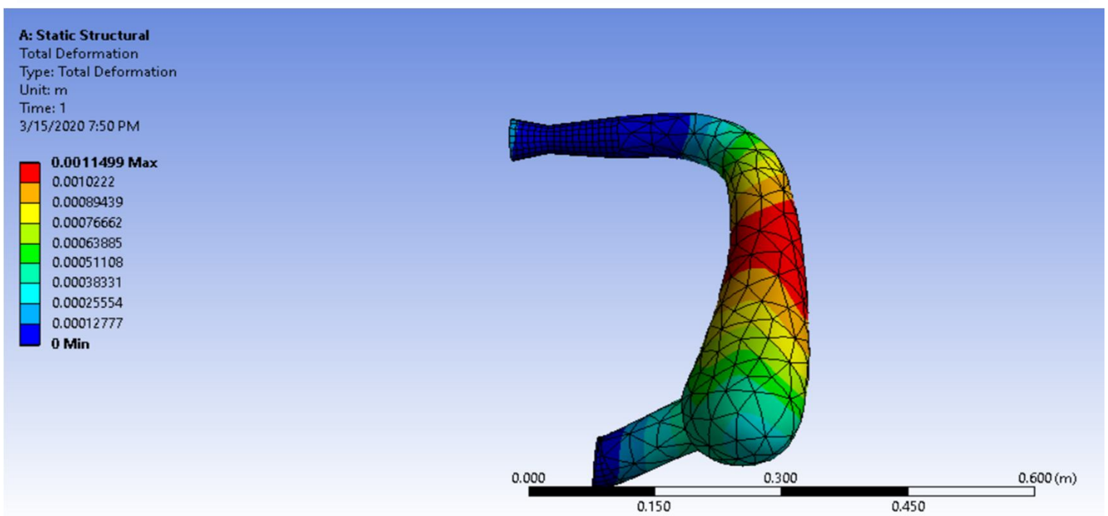
Fig. 8 Meshed model of our intake





Structural analysis was done to determine the thickness of the material to be used where the input of 6 kpa of backpressure and 45km/h of inlet air speed(for analysis) and found that 5 mm and above thickness of ABS would survive with acceptable stress intensity and deformations.





V. CONCLUSIONS

The effects of intake manifold tuning by varying intake runner lengths and the valve opening timings were successfully studied, simulated and observed. Tuning can considerably assist to increase the power produced by an internal combustion engine by increasing the airflow to the combustion chamber. But at the same time can hurt the engine's performance, if not done properly.

From the result of the study, the restrictor converging boundary must have smooth edge corner to make sure the airflow maintains with their flow path. The pressure recovery of restrictor must be maximum as much as possible to minimize the pressure drop. The airflow in the intake manifold must be distributed evenly to improve the volumetric efficiency which in turn improves the engine performance.

The material for manufacturing used was ABS(P400) which has good strength and was tested for our design by structural analysis before manufacturing. Annealing was done which is slight heating of the product and slow cooling which relives stress that is generated while printing and this is done by breaking of the semi crystalline structure within the plastic [12]. Three layers of carbon fibre fabric were applied to the manifold using a high temperature resin (PTM&W PT2520) by vacuum bagging process. This resin was chosen as it is able to survive the high temperatures (~120 F operating) that are safe from the operation of the engine.

For future validations, calibrated sensors can be used to determine velocity of throttle inlet and manifold pressure which improve the accuracy of the simulations and design with the obtained experimental data.

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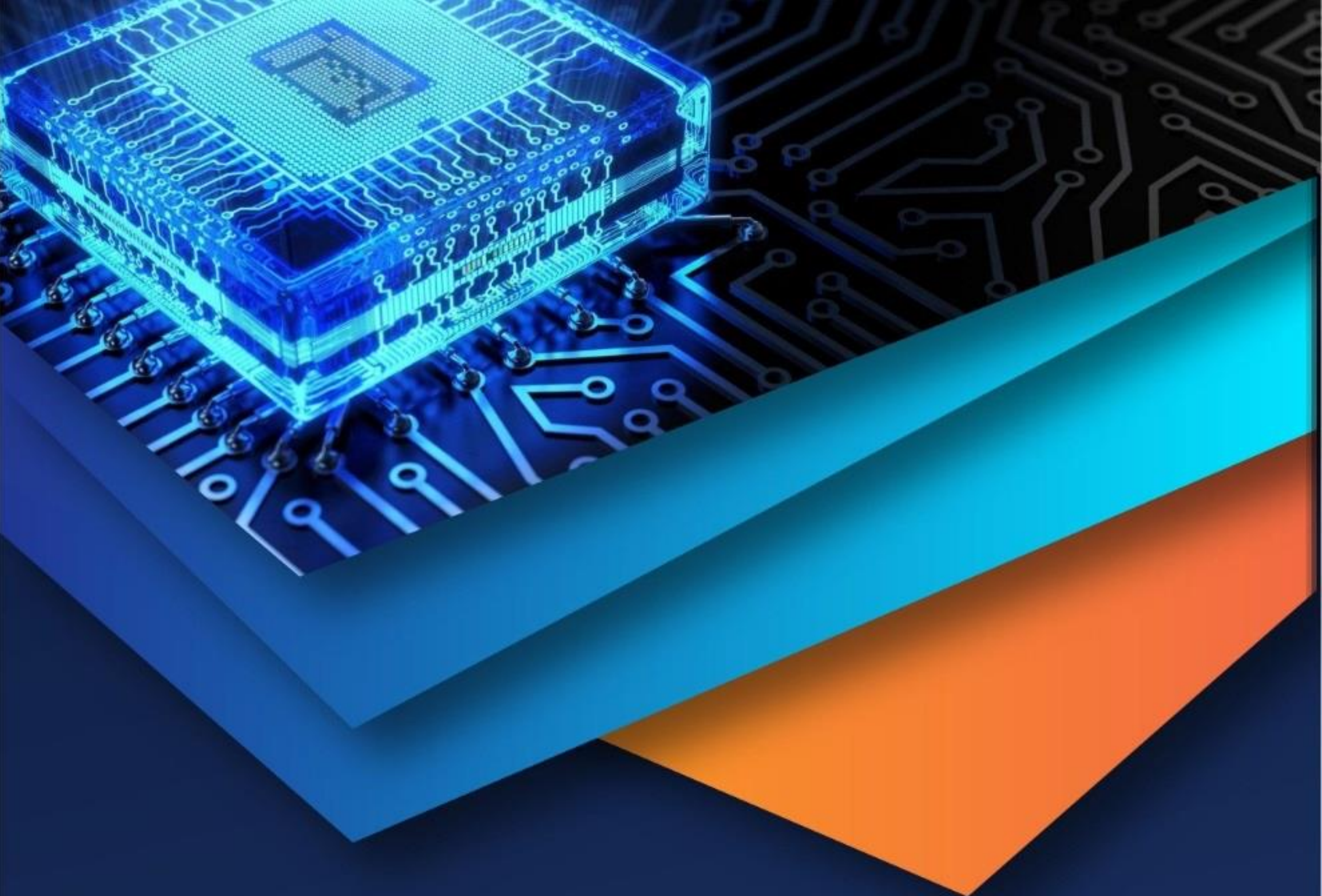

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