DESIGN AND IMPLEMENTATION OF

A TURBOCHARGER SYSTEM

A thesis

Presented to

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING

Department of Mechanical Engineering

NUST

ISLAMABAD, PAKISTAN

In Partial Fulfillment

of the Requirements for the Degree

Mechanical Engineering

By

Kaleem Ullah Khan

June 2015

Keywords: Intake, Exhaust, Transient , Turbocharger

EXAMINATION COMMITTEE

We hereby recommend that the dissertation prepared under our supervision by: Kaleem Ullah Khan 33-B, titled: "Design and Implementation of a Turbocharger System" be accepted in partial fulfillment of the requirements for the award of BE Mechanical degree with (___ grade).

Committee Chair: Dr. Riaz Ahmad Mufti	________________ Dated:
Committee Member: Dr. Nabeel Anwar	________________ Dated:
Committee Member: Dr. Aamir Mubashar	________________ Dated:
Committee Member: Dr. Hussain Imran	________________ Dated:
Committee Member: Dr. Umar Ansari	________________ Dated:
Committee Member: AP Muhammad Usman Bhutta	________________ Dated:
Supervisor: {Supervisor Name}	________________ Dated:

______________________ ____________

(Head of Department) (Date)

COUNTERSIGNED

Dated: ________________ __________________________

(Dean / Principal)

ABSTRACT

The purpose of this thesis is to design and tune the intake and exhaust manifolds over a targeted RPM range for peak power. These manifolds improve the volumetric efficiency of the engine and reduce the back pressure. Different ideas were applied during the designing process, the 3D CAD models of which were made on Solidworks. To test the designs, Solidworks flow simulation was used, which showed a drop in pressure in the headers. To conduct a more accurate result, transient flow analysis had to be done. Flow equations through an orifice (valve openings) were modelled for the pressure drop. This pressure drop was computed from the time the valve opened till it closed and fed into Solidworks flow simulation for the transient flow analysis. For testing the tuned manifolds, an engine was modelled in Ricardo wave and the designs were made in Ricardo wave3dbuild. The whole system was set up from intake to the exhaust and an analysis was run to check whether the tuned parameter actually gave the maximum torque over the desired RPM.

PREFACE

This is an independent thesis by Kaleem Ullah Khan. An equation for the change in pressure in an engine cylinder was modelled. This was further used in designing the air flow system of an automobile.

ACKNOWLEDGMENTS

During the course of this work, constant guidance from my professor was provided, he was always willing to help and clarify all of my queries.

This thesis would not have been possible with the continuous support of Dr. Samiur Rahman SHAH. He helped me through every problem I faced, although he would never tell me a direct answer. I was always pushed to the limit to figure out the solution myself.

I would like to keep working with him through the summers too as he is an inspiration for me, the knowledge he holds in his field is commendable and I would like to exploit my professor for it.

ORIGINALITY REPORT

This report has been typed by Kaleem Ullah Khan, all the content (including tables, figures, graphs) have been generated myself and nothing has been taken from the World Wide Web.

Signature

COPYRIGHT

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ABBREVIATIONS

NUST National University of Science and Technology

SAE Society of Automotive Engineering

FSAE Formula Society of Automotive Engineering

CFD Computational Fluid Dynamics

NOMENCLATURE

P0 Pressure at exhaust opening

T0 Temperature at exhaust opening

CHAPTER 1

INTRODUCTION

The performance of an automobile depends on various systems and parts, the most important and the engine, air flow systems and the aerodynamics of it. A race car is designed in a different manner than a salon. This project aims on improving the air flow system and achieving the torque peak over a desired RPM range. The main parts of an air flow system are the intake manifold, the exhaust manifold and the turbocharger.

Intake System

An intake manifolds primary function is to draw air from outside (environment) and evenly distribute the combustion mixture (or just air in a direct injection engine) to each intake port in the cylinder head(s). Even distribution is important to optimize the efficiency and performance of the engine. It may also serve as a mount for the carburetor, throttle body, fuel injectors and other components of the engine. The volumetric efficiency of the engine is dependent on the design of the manifold, with a good design and harnessing the pressure wave created in the runners of the manifold, volumetric efficiency of more than 100% can be achieved.

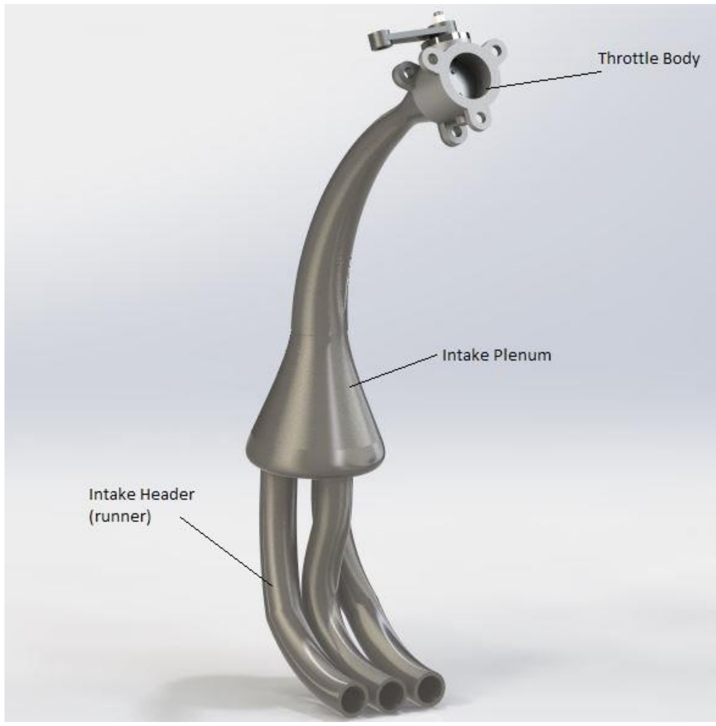

Figure 1. Intake Manifold

Exhaust System

A manifold that receives exhaust gases from the cylinders and conducts them to the exhaust pipe and into the environment through a muffler. The exhaust also helps in improving the volumetric efficiency by extracting the exhaust gasses efficiently with no trace of combustion gasses left behind in the engine for fresh mixture during the intake stroke. There are various factors to improve the exhaust system, the reduction of backpressure for free flow, the scavenging phenomena and a step in the exhaust header for reversion.

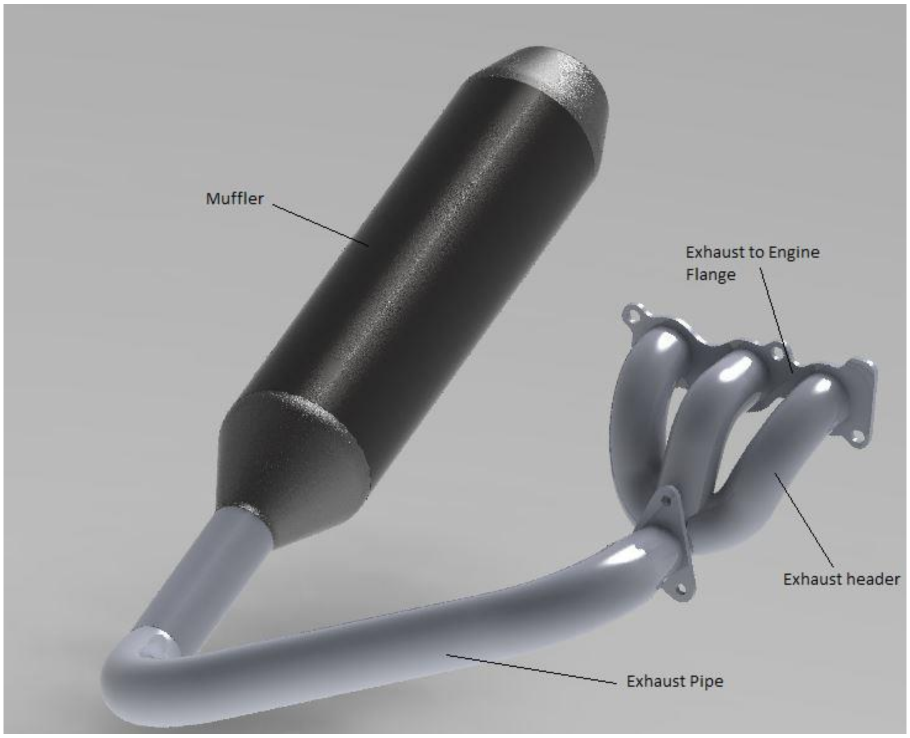

Figure 2. Exhaust system: exhaust manifold, exhaust pipe and muffler

Turbocharger

There are two types of forced induction systems, the supercharger and the turbocharger. For the Formula-Student race car we are using the turbocharger system. They increases the engine's efficiency and power by forcing extra air into the combustion chamber. This improvement over a naturally aspirated engine's output results because the turbine can force more air, and proportionately more fuel, into the combustion chamber than atmospheric pressure alone.

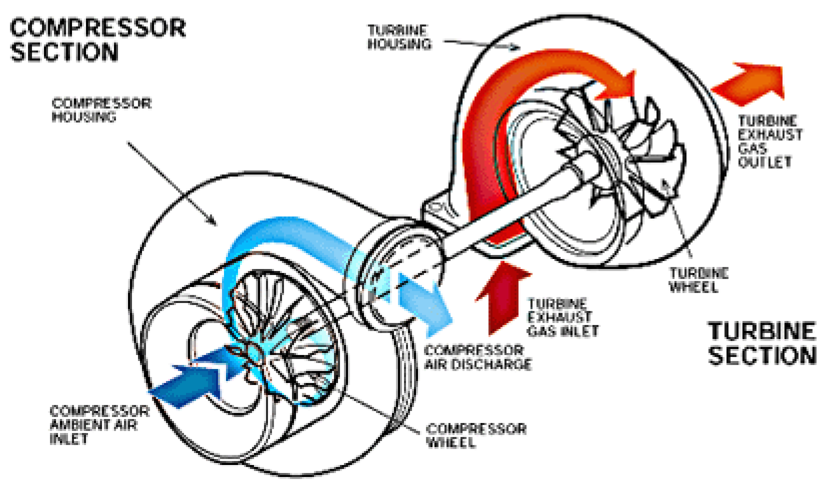

Figure 3. Turbocharger

CHAPTER 2

LITERATURE REVIEW

2.1 Engine Piston

A piston is a moving component that is confined in a cylinder and is made gas-tight by piston rings. A piston is used to transform energy imparted by a fluid entering or expanding inside a cylinder into a rectilinear motion, in an engine this rectilinear motion is transformed into rotary motion by means of a connecting rod which is connected to the crankshaft.

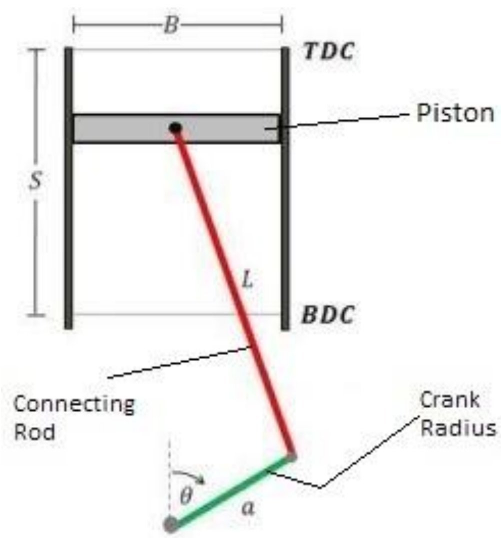

Figure 4. Piston schematic

Basic parameters for modeling an IC engine are: –

At the TDC piston height from crank origin is, $L + a$.

At the BDC piston height from crank origin is, $L - a$.

B => bore (i.e., diameter of the cylinder)

L => length of the connecting rod

S => stroke length

a => crank radius

θ => crank angle

Piston height relative to the crank origin is;

$$\Rightarrow S\theta = \mathrm{a} + \frac{a^2}{4L} - \mathrm{a}\cos(\theta) - \frac{a^2}{4L}\cos(2\theta)$$

2.1.1 Otto Cycle

An Otto cycle is an idealized thermodynamic cycle that describes the functioning of a typical spark ignition piston engine. It is the thermodynamic cycle most commonly found in automobile engines. The Otto cycle is a description of what happens to a mass of gas as it is subjected to changes of pressure, temperature, volume, addition of heat, and removal of heat. The mass of gas that is subjected to those changes is called the system. The system, in this case, is defined to be the fluid (gas) within the cylinder. By describing the changes that take place within the system, it will also describe in inverse, the system's effect on the environment. In the case of the Otto cycle, the effect will be to produce enough net-work from the system so as to propel an automobile and its occupants in the environment.

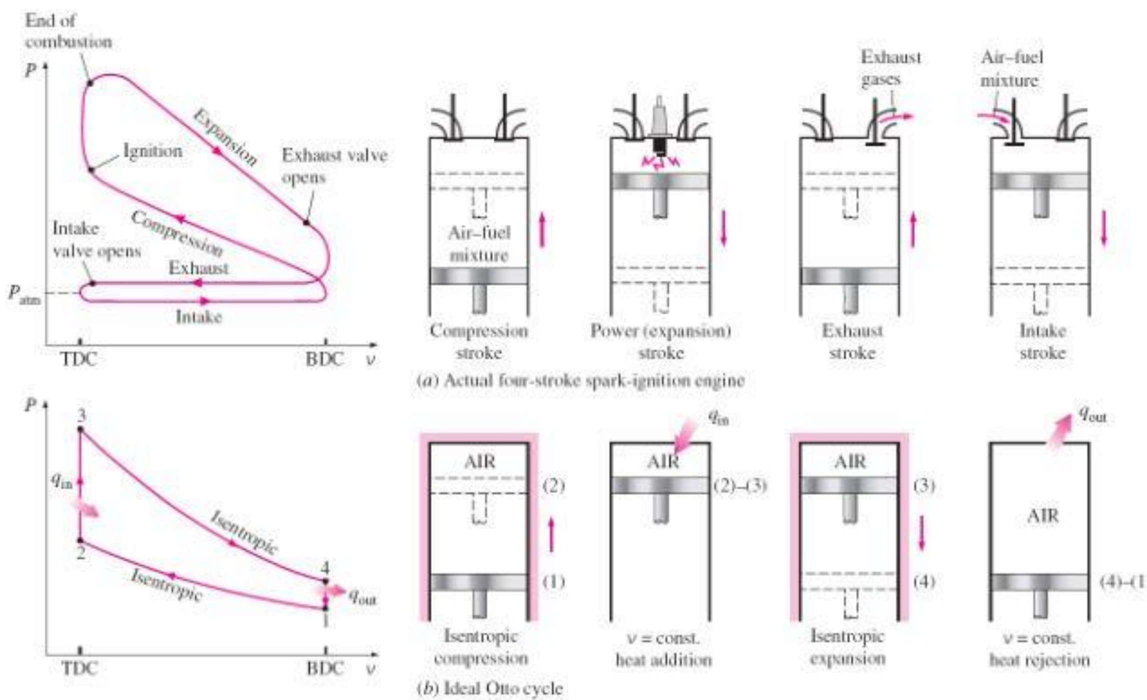

Figure 5. OTTO Ideal Cycle

2.1.2 Cylinder Pressure

Gas pressure in the cylinder of an engine varies throughout the Otto four-stroke engine cycle. Work is done on the gases by the piston during compression and the gases produce energy through the combustion process. These changes in energy combined with changes in the volume of the cylinder lead to fluctuations in gas pressure. The ability to accurately predict the pressure allows for better understanding of the processes taking place in the cylinder for designing and tuning the intake and exhaust manifolds.

The problems encountered in gathering data, the camshaft profile, spark timings and the parameters of the intake and exhaust valves were not accurately known. Some assumptions were taken due to which the modelling of pressure could not be completed; the inertia of mass was taken zero (neglected).

2.1.3 Pressure through an orifice

The pressure in the cylinder depends on whether it is an exhaust stroke or an intake stroke. The exhaust valve opening (E.V.O) occurs before the bottom dead center. At this point, the pressure inside the cylinder is much greater than the exhaust header pressure, so when the exhaust valve opens, due to the pressure gradient the mass flows out through the orifice. The piston is also moving, which assists in pushing the gasses out. Moreover the valve opening also varies with time which tends to change the mass flow rate. In brief there are many parameters that affect the pressure change, namely; the cross-section area change with the valve lift, the pressure drop in the cylinder for every time step affects the mass flow rate and the piston stroke.

The gas flow can be modelled with the following equations;

Ideal Gas equation;

$$PV = mRT$$

Isentropic gas model;

$$\frac{P_1}{P_0} = (\frac{T_1}{T_o})^{\frac{\gamma}{\gamma-1}}$$

For choked flow;

$$\dot{m} = \frac{C_d\, A_T P_o}{\sqrt{RT_o}} \gamma^{\frac{1}{2}} \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}}$$

For subsonic flow;

$$\dot{m} = \frac{C_D\, A_T P_o}{\sqrt{RT_o}} \left(\frac{P_t}{P_o}\right)^{\frac{1}{\gamma}} \left(\frac{2\gamma}{\gamma-1}\left(1 - \frac{P_t}{P_o}\right)^{\frac{\gamma-1}{\gamma}}\right)^{1/2}$$

Where;

$\dot{m}$ = real mass flow rate

C_D = discharge coefficient

A_T = cross-sectional area of throat

P_t = pressure at throat

P_o = pressure in the cylinder

T_o = temperature in the cylinder

R = characteristic gas constant of the exiting gases

γ = ratio of specific heats of the exiting gases

The valves act as flow restrictors through an orifice

2.2 Intake System

The basic layout of fuel injection manifold is determined by its application. A design focused on racing application has one throttle plate per cylinder. The advantages of this design is a better flow to each cylinder which results in more power. For street driven automobile one throttle plate attached to the intake manifold is preferred because of a crisp intake manifold vacuum generated. This vacuum greatly increases the accuracy with which low speed fuel and ignition can be calibrated.

The two application are very different from each other but the design of the manifold has a lot of things in common; an ideal shape, symmetry of design, volume of the plenum length of the intake, taper angle of the intake, injector location.

2.2.1 Ideal Shape and Symmetry

The shape of the manifold should have smooth curves so that there is no turbulent flow from the throttle to the engine port. Laminar flow assists in pushing more air into the engine. Symmetric shape is also a very important factor for even distribution to the cylinders.

2.2.2 Plenum Volume

The volume of the plenum is a function of engine displacement. Usually, 50% to 70% of the engine displacement is a good starting point. A few volumes are tested before knowing which one suits best for the maximum power. The intersection between the plenum and the runner design is vital in keeping the flow laminar. A bell mouth entry must be carefully designed to avoid flow losses.

2.2.3 Intake runner

Tuning the intake runner length for the torque peak to lie in the desired RPM range is a tricky affair. The stream of air entering the cylinder through the intake ports get blocked and squished like a spring against the intake valve when it closes. The air pushes against the intake valve and flows the other direction creating a pressure wave. The main aim is for the intake valve for the next cycle to open precisely at the time when the pressure wave arrives. This forces more air into the cylinder achieving a high volumetric efficiency. This theory is known as the ram theory. A tuned runner length is one of the most important factor in achieving more power.

How does the pressure wave bounce back at the end of the runner? Let’s take a balloon as an example. When the rubber wall of the balloon breaks, the compressed air expands. As long as the pressure of the air inside the balloon (or more accurately where the balloon was) is above atmospheric pressure, the air accelerates outward. The air rushing outward reaches maximum speed just as the pressure accelerating it outward reaches atmospheric pressure. The outward rushing air has inertia, the air moves outward reducing the pressure of the region of air which was inside the balloon. The combination of surrounding atmospheric pressure ahead and lowered pressure behind slows down the outward flowing air, brings it to a stop and accelerates it back toward the region where the balloon was. Once again inertia causes an overshoot and the pressure increases above atmospheric. The pressure oscillates above and below atmospheric. The amplitude of the pressure oscillations decreases with time.

Consider a traveling sound wave moving through a tube. The sound wave is made of regions in which air has a pressure above atmospheric pressure and regions in which the air pressure is below atmospheric pressure. A high pressure region is called a compression while a low pressure region is called an expansion. The compressed air is held in by the walls of the tube. A region of compressed air moves along the tube by compressing the air in front of itself. When the compressed air reaches the end of the tube it can expand to the side. The air in the compression accelerates outward gaining speed as long as it is above atmospheric pressure. It reaches maximum speed outward as its pressure reaches atmospheric pressure. The air continues outward carried forward by its own inertia reducing the pressure near the mouth of the tube to below atmospheric. The resulting expansion propagates down the tube. So the wave reflects off the open end, with a compression turning into an expansion.

2.2.4 Injector Location

A few basic rules apply to the location of the injector. The tip of the injector should face the center of the port, it should discharge at a point where the air velocity is the highest. For a very high air flow, a single injector cannot provide sufficient fuel, in such a case secondary injector is placed downstream to compensate for the fuel.

2.3 Exhaust System

The exhaust manifold plays a key role in the performance of a vehicle, may that be a racing or a street vehicle. Its direct responsibilities are moving the gas pulses at a steady rate without building up back pressure, scavenging the cylinder completely of the combustion gasses and in a turbocharged system it has to support the turbocharger and guide gasses into it efficiently.

There are two basic styles of exhaust manifold, the simple log style and the equal length multiple-tube, individual runner type (also called tubular/collector style).

2.3.1 Exhaust header

A well designed exhaust header helps to eliminate the manifold's back pressure. Instead of a common manifold (log style) that all of the cylinders share, each cylinder gets its own exhaust pipe (tubular style). These individual pipes join together in large vessel called the collector. By making the exhaust headers the same length, it guarantees that each cylinder's exhaust gases arrive in the collector spaced out equally so there is no back pressure generated by the cylinders sharing the collector. The header lengths can be tuned to a specific RPM using the pulse theory. This tuned length help the flow of the engine by creating a negative pressure wave in the collector to help suck the next pulse out, this phenomena is called scavenging.

2.3.2 Scavenging

In automotive usage, scavenging is the process of pushing exhausted gas-charge out of the cylinder and drawing in a fresh draught of air or fuel/air mixture for the next cycle. This process is essential in having a smooth-running internal combustion engine. If scavenging is incomplete, the following stroke will begin with a mix of exhaust fumes rather than clean air. This may be inadequate for proper combustion, leading to poor running conditions such as four-stroking. The parameters that effect the scavenging process are the length and diameter of the header pipes.

2.3.3 Back pressure

Pressure opposed to the desired flow of a fluid in a confined place such as a pipe is called back pressure. It is often caused by obstructions or tight bends in the confinement vessel along which it is moving, such as piping or air vents. Because it is really resistance, the term back pressure is misleading as the pressure remains and causes flow in the same direction, but the flow is reduced due to resistance. For example, an automotive exhaust muffler with a particularly high number of twists, bends, turns and right angles could be described as having particularly high back pressure. Back pressure caused by the exhaust system (consisting of the exhaust manifold, catalytic converter, muffler and connecting pipes) of an automotive four-stroke engine has a negative effect on engine efficiency resulting in a decrease of power output that must be compensated by increasing fuel consumption.

2.3.4 **Reversion**

Reversion is the reversal of exhaust gasses back into the combustion chamber during valve overlap, when the downward movement of the piston creates a vacuum in the cylinder. Primary pipes in the exhaust manifold should match the exhaust port diameter on the cylinder head at least; but to reduce reversion, a primary pipe that is slightly larger than the exhaust port is better, it is also called an exhaust step.

2.3.5 Catalytic converter

A vehicle emissions control device which converts the toxic pollutants in the exhaust gasses to non-toxic or less toxic gasses by catalyzing a redox reaction (either oxidation or reduction process). Internal combustion engines fueled by either petrol (gasoline) or diesel use this control device for a safe environment.

2.3.6 Muffler.

An internal combustion engine produces a lot of noise with just an exhaust manifold mounted on it, this noise is cancelled with a device known as muffler (silencer in British English, or back box in Irish English). The muffler is designed in such a way the sound waves cancel themselves out resulting in noise which to some people is like harmony.

2.3.7 General designing tips

For designing an exhaust manifold for a specific application a few points should be kept in mind. The radius of the bends that make up the exhaust primary runners should be maximized to maintain the pulse energy for the turbocharger and rapid area changes should also be avoided for the same purpose. For balanced exhaust reversion in all the cylinders, the length should be kept equal. At the collector, introduce flow from all runners at a narrow angle to minimize "turning" of the flow in the collector. For better boost response, minimize the exhaust volume between the exhaust ports and the turbine inlet. For best power, tuned primary lengths should be used.

.

2.4 Turbocharger System

The turbocharger is bolted to the exhaust manifold of the engine. The exhaust from the cylinders spin the turbine, which in turn spins the centrifugal compressor connected to the turbine through a shaft. The air is compressed above atmospheric pressure before going into the engine for combustion. The turbocharger can reach the speed of 150,000 RPM, therefore, the turbine shaft has to be supported very carefully. Most bearings explode at speeds like this, hence, fluid bearings are used for this purpose. These bearings support the shaft on a thin layer of oil that is constantly pumped around the shaft. This serves two purposes: It cools the shaft and some of the other turbocharger parts, and also assists the shaft to spin with less friction.

One of the main problems with turbochargers is that they do not provide an immediate power boost when you step on the gas. It takes a second for the turbine to get up to speed before boost is produced. This results in a feeling of lag when you step on the gas, and then the car lunges ahead when the turbo gets moving. One way to decrease turbo lag is to reduce the inertia of the rotating parts, mainly by reducing their weight. This allows the turbine and compressor to accelerate quickly, and start providing boost earlier. One sure way to reduce the inertia of the turbine and compressor is to make the turbocharger smaller. A small turbocharger will provide boost more quickly and at lower engine speeds, but may not be able to provide much boost at higher engine speeds when a really large volume of air is going into the engine. It is also in danger of spinning too fast at higher engine speeds, which might result in breaking of the blades or the shaft. Another way to lighten the weight is to use a lighter material, for e.g. ceramic. The high end turbochargers use blades made of ceramic which are lighter than the steel blades used in most turbochargers. Again, this allows the turbine to spin up to speed faster, which reduces turbo lag.

Outside Air Ambient Temp	30	°C	
Altitude above Sea Level	0	m	
Engine Displacement Volume (Vd)	610.00	cc	
Engine Compression Ratio	10.5		
Engine RPM for Maximum HP	8000	RPM	
Target HP	65	HP	
BSFC	0.55	lb/(hp.h)	Rate of fuel/power produced
Engine Air to Fuel Ratio (AFR)	12.50		Should be rich. Enter 13.0 to 13.5.
Engine Volumetric Efficiency (VE)	0.850		Typical range is 0.70 to 0.85.
Intake Manifold Temp	100	F	
Turbocharger Compressor Efficiency (CE)	1.00		Find CE on the compressor map.
Intercooler Efficiency	0.00		Set this to 0 if there is no intercooler.
Intercooler Pressure Drop in psi	0.00	psi	Set this to 0 if there is no intercooler.
Pressure Drop from ambient to Inlet	1.00	psi	restriction (caused by the air filter or restrictive ducting)
Gas Constant	636.9		
X & Y VALUES FOR CE GRAPH			***Formulas***
X = Air Flow in m^3/s	0.04836	m^3/s	*Convert lb/min to m^3/s : conv. Factor(c.f) =(0.3/39.68)(kg/s) ÷ Density of air(kg/m^3)*
X= Air Flow in lb / minute	7.45	lb/min	Target Hp(Hp) x (BSFC*0.001644/60)(lb/Hp.min) x AFR
= Turbocharger Boost Pressure Ratio PR	1.84		(Compressor discharge pressure) ÷ (compressor inlet pressure)
	OUTPUTS		
Outside Air Pressure in psia (Absolute)	14.7	psia	*(101325 x (1-0.0000225577 x Altitude)^5.25588) (Pa) x (0.000145037738) (c.f) = psi*
Air pressure at the Inlet	13.7	psia	*outside pressire(psi) - pressure drop till the inlet of the compressor(psi)*
Density of Air	1.1644	Kg/m^3	*Density (Kg/m^3)= pressure(Pa) ÷ (specific gas constant(J/(kg*degK) x ambient temperature)(K)*
Boost Pressure Ratio after Intercooler			
Turbocharger Outlet Temperature in deg F			
Intercooler Outlet Temperature in deg F			
Absolute pressure	25.18649	psia	*{(Airflow)(lb/min) x (Gas Constant) x (460 + Intake manifold temp(F))} ÷ {(V.E) x (RPM/2) x Vd(in^3)*
Turbo Boost	10.49055	psig	
Absolute pressure	1.736548	bar	
Turbo Boost	0.723298	bar	

Turbocharger calculator

2.3.2 **Wastegate**

Turbochargers have a wastegate, which allows the use of a smaller turbocharger to reduce lag while preventing it from spinning too quickly at high engine speeds. The wastegate is a valve that allows the exhaust to bypass the turbine blades. The wastegate senses the boost pressure. If the pressure gets too high, it could be an indicator that the turbine is spinning too quickly, so the wastegate bypasses some of the exhaust around the turbine blades, allowing the blades to slow down.

METHODOLOGY

3.1 Intake

3.1.1 Plenum Volume

Plenum volume is kept 50-70% of the engine displacement, but you have to test a few sizes to know which is the optimum power and volumetric efficiency.

3.1.2 Runner length

Runner lengths calculation.

The input values for desired RPM:

one cycle	720	o
Tuned for	4000	RPM
Time for 1 revolution	0.015	sec/rev
Tme for 1 cycle	0.03	sec/cycle
Duration Intake valve closed for during the cycle	0.0194583	sec

Table 0-1. Intake runner length input values

intake valve open before TDC	19.35	o
Intake stroke	180	o
Intake valve close after BDC	53.65	o
Total intake valve open duration	253	o

Table 0-2. Intake valve duration input

As the speed of sound varies with temperature, we have to calculate the speed of sound for the environment where the car is going to run.

Speed of sound		
Ambient Temperature	30	^oC
Ambient pressure	101.325	Pa
Relative humidty	50	%
Speed of sound	350.31	m/s

Table 0-3. Speed of sound for specific enviornment

Now that we have all our input values, we can calculate the runner lengths. It is fairly simple, we have to calculate the distance the pressure wave would travel up the runner and back down before it reaches the intake valve again.

Distance = Speed of sound x intake valve closed duration

= 350.31(m/s) x 0.0194583 (s)

= 8.8164m

This is the distance it travels, divide by 2 to get the distance it travels up and down the runner, 3.4082m (340.82cm).To get the distance for every consecutive wave length, divide by 2.

Intake runner lengths :			
	340.8224375	cm	1st wave
	170.4112188	cm	2nd wave
	85.20560938	cm	3rd wave
	42.60280469	cm	4th wave
	21.30140234	cm	5th wave
	10.65070117	cm	6th wave

Table 0-4. Intake Runner lengths

3.2 Exhaust

3.2.1 Header Calculation

The formula to work out the primary length (P);

$$P = \frac{850 * ED}{RPM} - 3$$

ED = 180 plus the number of degrees the exhaust valve open before BDC

Once the primary length have been calculated, calculating the internal diameter (ID) is the next step.

$$ID = \sqrt{\frac{cc}{(P+3) * 25}} * 2.1$$

Cc = cylinder volume in cubic centimeter

If you find the peak torque for specific RPM but the internal diameter is too big or small to fit the exhaust ports on the engine, you can adjust it. Generally a reduction in the primary pipes internal diameter of 0.125in will move the torque peak down by 500-600 in large engines and 650-800 in engines below 2.0L.

Exhaust valve open before BDC	19.34	º		
ED	199.34	º		
RPM exhaust tuned to	14000			
CC of engine	610			
Diameter of collector inlet	4.33071	in		
	11	cm		
Collector taper angle	8	º		

reduction multiple	13.775	
	max	min
Rpm after ID reduction	5046.25	2980

Primary length	9.102786	in	23.12	cm
ID of primary length	1.259877	in	3.2	cm
ID of secondary length	1.657014	in	4.209	cm
Collecter length	1.119523	in	2.844	cm
ID of tailpipe	4.016033	in	10.2	cm
Tailpipe length	12.10279	in	30.74	cm

Table 0-5. Exhaust Runner Calculation

3.2.2 Transient analysis

To carry out a transient analysis we need to calculate the pressure change at the exhaust valve with time step. These valves can be fed into a CFD software to analyze how the flow would change when the exhaust valve starts opening till its closing.

Following is the series of pictures on how the flow varies.

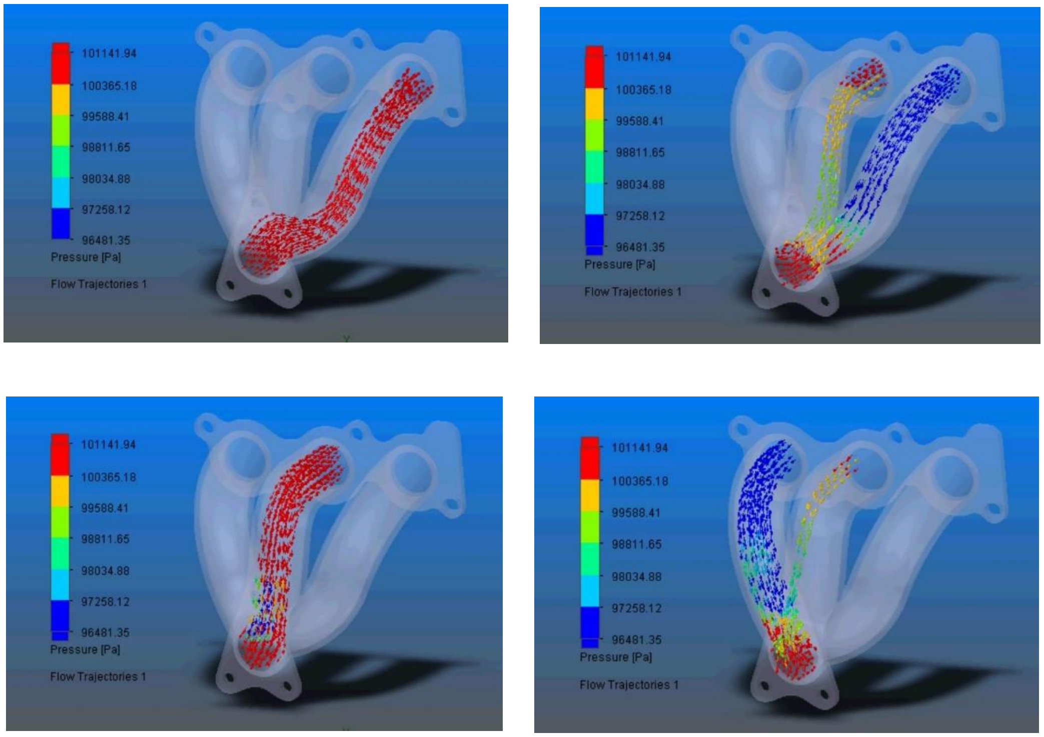

Figure 6. Air flow through the exhaust sequentially (from left to right)

3.2.3 Transient analysis calculation

volume change w.r.t stroke		
Connecting rod length (L)	120	mm
crank radius (a)	32.5	mm
cylinder bore diameter (b)	63	mm
Stroke change with crank angle (S_θ)	$a + a^2/4L - a\cos(\theta) - a^2/4L*\cos(2\theta)$	
Clearance volume	2.13285E-05	m^2
Volume of cylinder	$PI*(b/2)^2 S_\theta$	
Total Volume	0.000223949	m^2

Table 0-6. Volume change with respect to stroke

Unburned fuel coefficient (C)	0.95	
Density of air (P & T at point 3)	1.23638	kg/m3
mass of air	0.000276887	kg
mass of fuel (m_fuel)	1.83946E-05	kg
Total mass	**2.62068E-04**	**kg**

Table 0-7. Total mass Inside Cylinder

Discharge coefficient (Cd)	0.7	
Cross sectional area of throat (At)	0.000314	m^2
pressure at throat(Pt)	101325	Pa
pressure in the cylinder when exhaust valve opens (P0)	106516.9634	Pa
temperature in the cylinder (T0)	309.5873812	k
characteristic gas constant of the exiting gases (R)	286.9	J/kg.k
ratio of specific heats of the exiting gases (γ)	1.4	
constant	11.3338611	
Real mass flow rate: for choked flow	**0.053791028**	**kg/s**
for subsonic flow	**0.023878887**	**kg/s**

Table 0-8. Mass flow rate

Number of cycles for the target RPM	2500	cycles/min
2 revolutions per cycle =	720	°
Time for 1 cycle	0.024	sec
0.024 sec/cycle = 720 degrees, therefore 1.067 degree =	3.55826E-05	sec
Exhaust valve open before BDC	19.352478	°
exhaust stroke duration in degrees	180	°
exhaust valve open after TDC	53.64908	°
Duration exhaust valve open for	253.001558	°
Range	160.65 to 413.65	
Time exhaust valve open for	0.00843	sec
Cycles per sec	41.66666667	cycles/sec
Rate of fuel consumption	0.00076644	kg/s

Pressure and Temperature inside the cylinder according to ideal OTTO cycle

Inlet air pressure (P1)	1	atm
	101325	Pa
inlet air temp (T1)	27	ºC
	300.15	k
Compression Ratio (r)	10.5	
Compression Index (n)	1.32	
Density of Fuel used	737.22	kg/m3
Specific gravity at 60F of petroleum (d)	0.73722	
Heat of combustion (Q)	47,137,774	j/kg
Air/fuel Ratio (stoich)	15.05	:1
Air/fuel Ratio (actual)	12.5	:1
Specific heat formula (Cv)	881.97	207.847

Table 0-9. Start of OTTO Cycle

Process 1-2 (Compression stroke)		
Pressure (P2)	2257805.295	Pa
Temp (T2)	636.9699193	K

Table 0-10. P and T at Compression Stroke

Process 2-3 (power stroke)		
Qin	2936451.525	j/kg
$T3^2 + 8.49\ T3 - 431967.25 = 0$		
Temperature (T3)	653.0125119	k
Pressure (P3)	2314669.912	Pa
	23.14669912	bar

Table 0-11. P and T at Power Stroke

pressure in the cylinder when exhaust valve opens (P0)	106516.9634	Pa
Temp in cylinder when exhaust valve opens	309.5873812	K

Table 0-12. P and T when the exhaust Valve opens

Process 3-4 (exhaust stroke)		
Temperature (T4)	307.7095158	k
Pressure (P4)	103876.9505	Pa
	1.038769505	bar

Table 0-13. P and T at exhaust Stroke

Known Quantities;

M_i = Total mass

$P0_i$ and $T0_i$ = Pressure and Temp. when then exhaust valve opens

dM/dt = mass flowrate

Calculate:

$M_{i+1} = (M_i) * (dM/dt * timestep)$

$P0_{i+1}$ calculated using Ideal gas equation and isentropic equation

Time step (sec)	Degree	Stroke change with angle (mm)	Volume m3	mass flow rate Kg/s	Total mass (Kg)	Pressure (Pa)
3.558E-05	160.64752	63.647	0.000219732	**0**	2.62068E-04	1.065E+05
7.117E-05	161.715	63.792	0.000220185	**5.56709E-07**	0.000262068	1.054E+05
1.067E-04	162.78248	63.929	0.000220612	**1.33559E-06**	0.000262068	1.051E+05
1.423E-04	163.84996	64.058	0.000221013	**2.42078E-05**	0.000262068	1.048E+05
1.779E-04	164.91744	64.178	0.000221388	**5.02988E-05**	0.000262067	1.046E+05
2.135E-04	165.98491	64.291	0.000221738	**0.000160812**	0.000262065	1.044E+05
2.491E-04	167.05239	64.395	0.000222062	**0.000274739**	0.00026206	1.041E+05
2.847E-04	168.11987	64.490	0.000222361	**0.000447732**	0.00026205	1.039E+05
3.202E-04	169.18735	64.578	0.000222634	**0.000609216**	0.000262034	1.038E+05
3.558E-04	170.25483	64.657	0.000222881	**0.00076267**	0.000262012	1.036E+05

Table 0-14. Transient Pressure Table

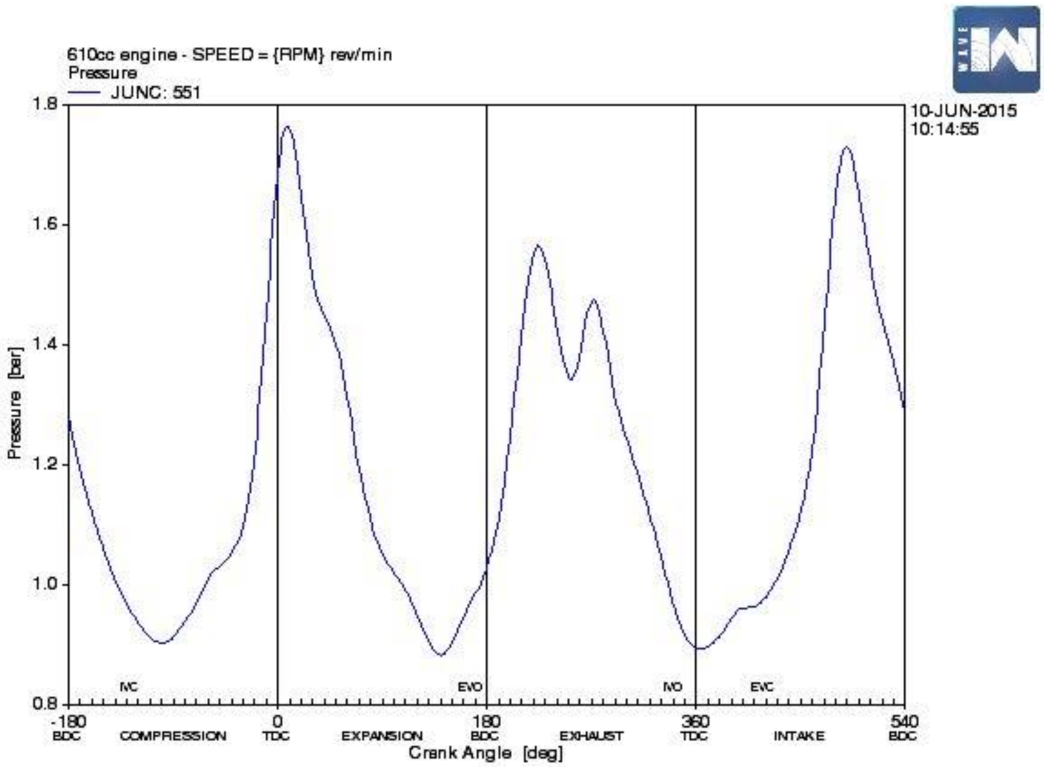

Figure 7. Pressure in exhaust valve used for transient analysis

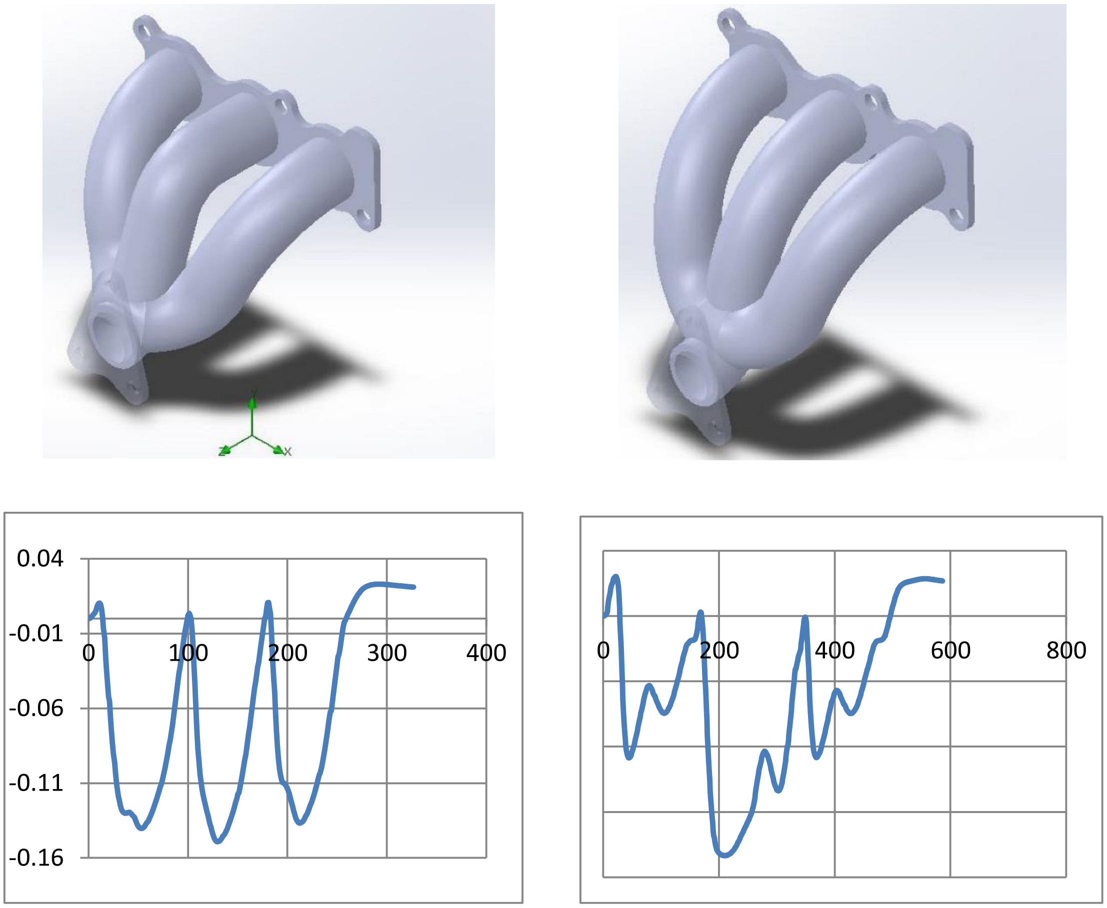

Figure 8. Equal length header vs different length

X axis – Mass flowrate m3/s

Y axis – Number of iterations

The advantage of keeping the header lengths equal reduces the back pressure (restriction in the flow). The tuned equal length exhaust manifold has equal mass flow rate through each header, which makes it easier for the gasses of the next cycle to flow out without any restriction. On the right, the un-tuned variable length exhaust manifold has a lower mass flow rate in the side pipes that the mid one, this makes it difficult for the exhaust gasses of the next cycle to flow out as the gas from the current cycle is causing a restriction for the next cycle. This back pressure reduces the volumetric efficiency and minimizes the scavenging effect in the exhaust.

CHAPTER 4

RESULTS

4.1 Ricardo wave

To test the tuned air flow system which includes the intake and exhaust manifold against an un-tuned one, a software called Ricardo Wave was used.

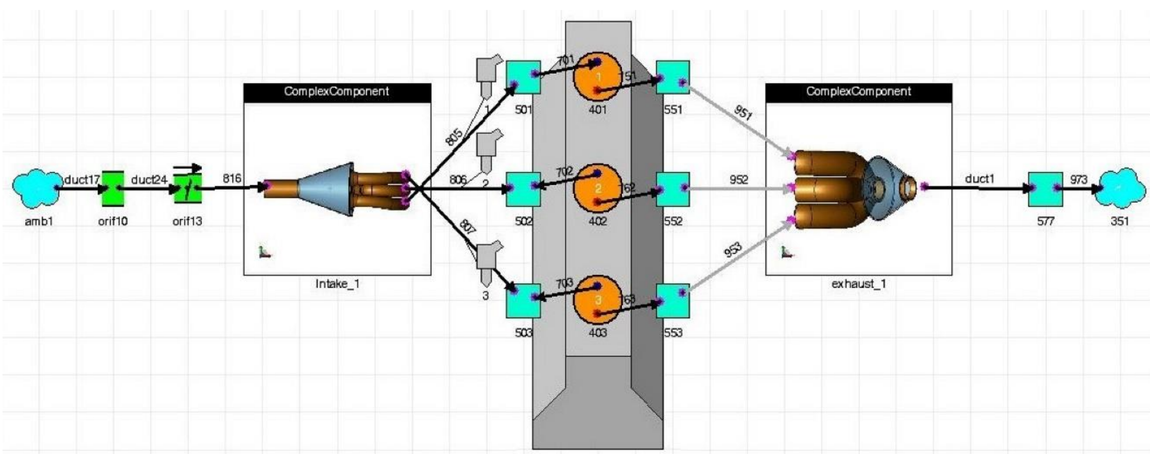

Figure 9. 610cc engine was designed using the software.

3.3.1 Results

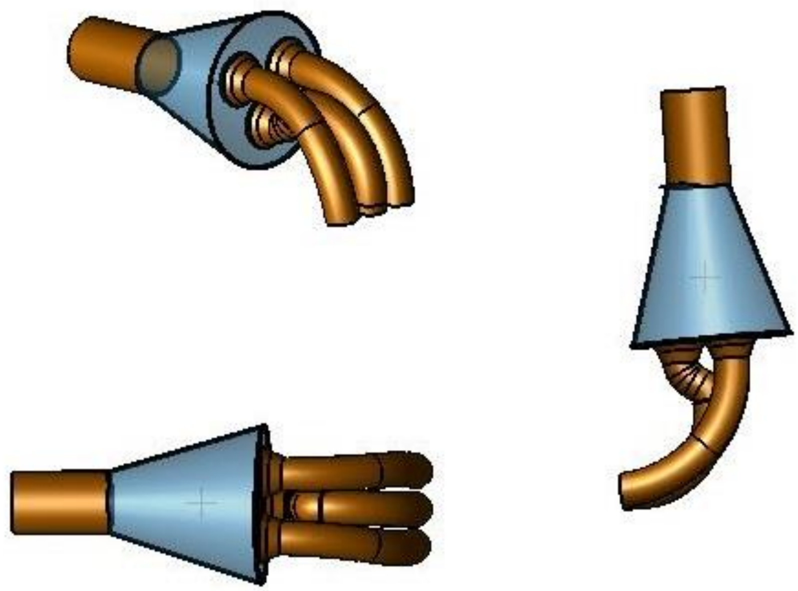

Figure 10. Intake Manifold (above), Exhaust Manifolds (below)

Two exhaust manifold were designed to compare the results of tuned (left) and un-tuned lengths (right).

Following is the comparison of the torque peaks

Figure 11. Un-tuned header length

Figure 12. Tuned header length

As you can see the tuned header lengths give us the peak torque in our desired range and also maintains a smooth curve.

On the other hand the un-tuned length has a very different trend, the torque drops at 2000 and 4000 RPM which is not suitable for our project.

Now, let's look at a comparison between the volumetric efficiencies

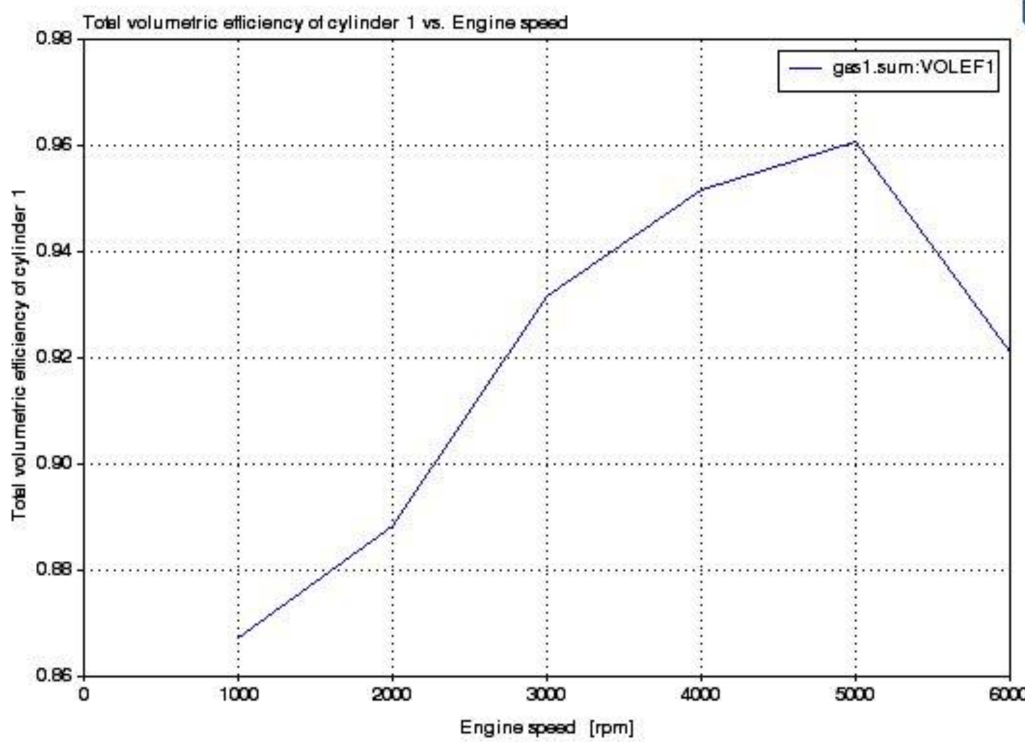

Figure 13. Volumetric Efficiency of the tuned exhaust headers

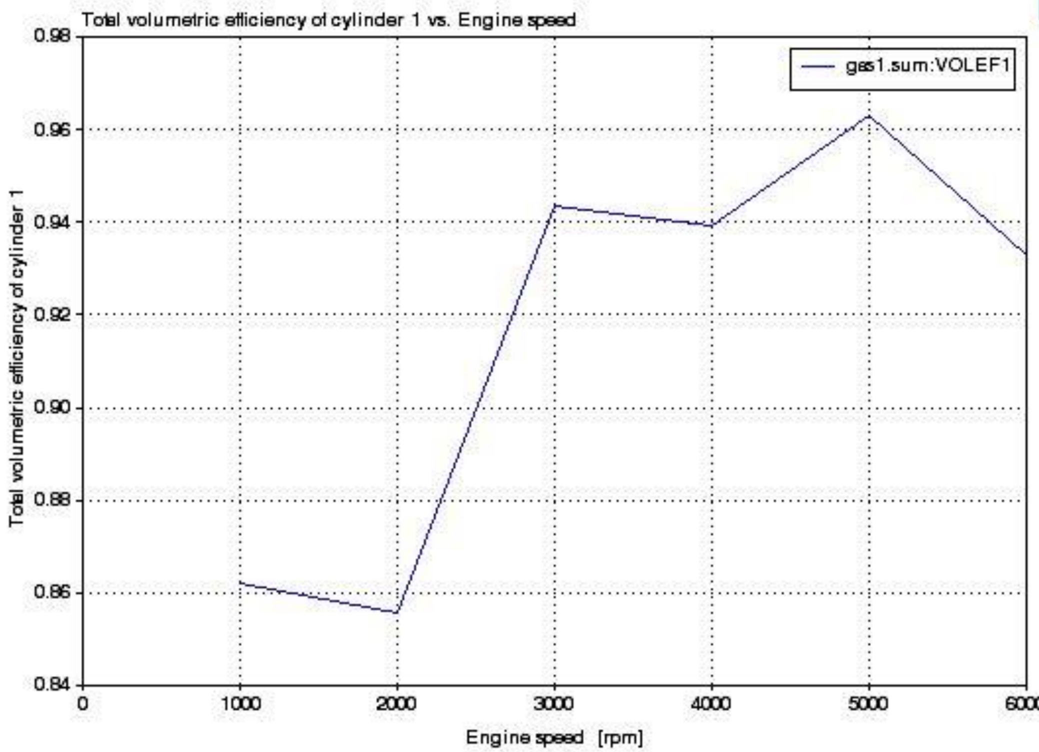

Figure 14. Volumetric Efficiency of Un-tuned exhaust header

The volumetric efficiency of the tuned header increases with the RPM and has the peak efficiency in our desired RPM band. Whereas the graph of the un-tuned header drops at 2000 and 4000 RPM which is again, not suitable for our project.

4.1.2 Conclusion

The air flow analysis of the tuned intake and exhaust lengths was a success. The tuned lengths for the intake and exhaust header gave us the desired result, within out RPM band. This will allow us to harness as much power as we can in the early RPM because the power in the late RPM is limited to us due to the 20mm restrictor before the throttle body.

4.1.3 How to Install Software

1. Extract the the .zip file in the DVD
2. Go to windows file and run “RS_2014.1_windows_installer.exe”
3. Follow the steps and install it
4. Navigate to “_SolidSQUAD_” and open “readme” follow the steps
5. Now, extract “Ricardo Suite 2014.1 x64 License Fix”
6. Open “read me” and follow the steps
7. Once installed, to open Ricardo wave, navigate to the folder where Ricardo was installed, on my laptop:

   “C:\Program Files\Ricardo\2014.1\Products\WAVE\Examples”

   Open any example, and open a new file from there.

WORKS CITED

[1] Performance Tuning in Theory & Practice By Graham Bell

[2] Maximum_boost Corky Bell

[3] Harvard Undergraduate Journal